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AEROSPACE FLUID COMPONENT DESIGNERS' HANDBOOK

VOLUME I

Revision D

TECHNICAL DOCUMENTARY REPORT NO. RPL-TDR-64-25

FEBRUARY 1970

AIR FORCE ROCKET PROPULSION LABORATORY
RESEARCH AND TECHNOLOGY DIVISION
AIR FORCE SYSTEMS COMMAND
Edwards, California

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TRW
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VOLUME I
Revision D

TRW SYSTEMS GROUP
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Editors

[redacted]

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AEROSPACE FLUID COMPONENT
DESIGNERS' HANDBOOK

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Qualified requesters may obtain copies of this report from the
Defense Documentation Center (DDC), Cameron Station
Alexandria, Virginia

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FOREWORD

This handbook is the result of a program which was originated to document state of the art information on aerospace fluid component technology. The program consists of compiling, organizing, and editing basic information concerning the design and selection of aerospace fluid components. The program was initiated in June 1962, and has been conducted under contract by TRW Systems Group (formerly TRW Space Technology Laboratories), Science and Technology Division, for the Air Force Rocket Propulsion Laboratory, Air Force Systems Command. Mr. Jack G. Hartley succeeded Captain John L. Feldman who succeeded Messrs. Roy A. Silver and James R. Lawrence as the Air Force Program Manager.

Mr. Terry M. Weathers succeeded Mr. Glen W. Howell as the Project Manager directing the program at TRW Systems Group, Although Mr. Howell retained overall responsibility for the handbook program as Manager of the Fluid and Electrical Systems Department. Subsequent editions of the handbook will be published and updated in a new format as part of the Air Force Design Handbook (DH) series. The handbook, to be identified as AFSC DH 3-6, Fluid Components, will be published and distributed by the Design Handbooks Branch (ASNPS-40), Aeronautical Systems Division, Wright-Patterson AFB, Ohio 45433.

A first edition of this new handbook will be available about mid-1970 (see subsection 1.7).

SPECIAL NOTICE

Subsequent Editions of This Handbook will Appear in a New Format as Explained in the Foreword Above.

ISSUED: FEBRUARY 1970
SUPERSEDES: NOVEMBER 1968 — AD-865276

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This handbook represents a team effort that involved many individuals at TRW Systems Group, as well as members of other aerospace industries. Members of the TRW Systems Group Technical Staff who were contributing authors are listed alphabetically as follows:

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Grateful acknowledgement is also made to the following individuals who made major contributions to the handbook program:

- A. L. Huntington, S. Inase, R. M. Williams, and M. A. Pupkin, Copy Preparation
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Acknowledgement is made to the many members of aerospace industries whose cooperation in submitting information and whose constructive suggestions concerning the initial edition of the handbook have contributed to the success of the program. In addition, special acknowledgement is made of the many individuals and organizations who reviewed material and offered recommendations prior to publication. The number of these reviewers has now grown so large as to make the publication of their names here impracticable.

The Editors wish to express sincere gratitude to all of the above, plus the many others whose valued assistance contributed to the publication and updating of this handbook.

G. W. Howell
T. M. Weathers
Editors

ABSTRACT

The Aerospace Fluid Component Designers' Handbook is a compilation of basic information on the design, analysis, selection, and specification of valves and associated fluid components used in aerospace fluid systems. The handbook is intended to be used as a basic reference for engineers and other technical personnel who are involved in any phase of aerospace fluid component technology. It contains sections dealing with heat transfer, fluid mechanics, fluid systems, fluid components, modules, analysis, computers, specifications, contamination and cleaning, reliability, materials, environments, stress analysis, component testing, and fluidics. The handbook has been revised periodically for the purposes of maintaining the currency of the data and adding applicable new material. The handbook content will subsequently be published as part of the AFSC Design Handbook Series, AFSC DH 3-6, Fluid Components.

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1.1 SCOPE

The Aerospace Fluid Component Designers' Handbook has been prepared to provide fluid component and fluid system engineers with a single source of basic information on the design, analysis, specification, and selection of components for aerospace fluid systems. The information presented in the handbook has been collected, collated, and correlated from a wide variety of sources. In addition to the books, technical papers, articles in technical journals, and manufacturer's literature listed in the bibliography, much material has been gathered from TRW Systems' fluid component experience in ballistic missile and spacecraft programs.

The need to collect available knowledge on fluid component technology, and to disseminate this knowledge in an appropriate way to facilitate its use, was recognized by the Air Force through experience gained on ballistic missile programs. In response to this need, the Air Force Rocket Propulsion Laboratory sponsored a program to search, collect, assemble, and disseminate fluid component information in the form of a handbook designed to (1) facilitate its use by the engineer, (2) provide for ease of updating, and (3) adapt to future expansion through a continuing program.

As a specific part of the continuing effort performed by TRW Systems Group under Air Force sponsorship, the following sections, which were not included in the first edition of the handbook, were subsequently included in the October 1965 edition, Revision A:

Sub-Section 5.12	Connectors
Sub-Section 6.2	Valving Units
Sub-Section 6.3	Static Seals
Sub-Section 6.4	Dynamic Seals
Section 9.0	Specifications
Section 12.0	Materials

Revision A provided for dividing the handbook into two volumes.

Revision B of the handbook was published in March 1967 and comprised an updating of previously published material.

Revision C was published in November 1968 and includes major additions in the form of:

Section 14.0	Stress Analysis
Section 15.0	Component Testing

Revision D, published in February 1970, includes the addition of Section 16.0, Fluidics, and major revision of Sub-Sections 6.6, Bellows, and 6.7, Diaphragms.

In the interests of keeping the size of the handbook within reasonable bounds, every attempt has been made to compile that information which has the broadest application and the greatest interest to a wide variety of potential users. For the reader desiring more detailed information, selected references are cited in the text indicating sources of additional or more detailed information on a particular subject.

Readers interested in details concerning the actual researching, writing, editing, and publication of this handbook are referred to "Preparation of the Aerospace Fluid Component Designers' Handbook," Technical Report AFRPL-TR-65-238, December 1965, TRW Systems Report No. 8676-6041-RU000, and "Updating of the Aerospace Fluid Component Designers' Handbook," Technical Report TR67-126, May 1967, TRW Systems Report No. 4712.5-67-141.

1.2 HANDBOOK CONTENT

The primary emphasis of the handbook is on aerospace valves; however, a number of components associated with valves have also been included. These components include fluid connectors and couplings, filters, flow meters, and pressure, temperature, and level sensors. Presentation of basic data and information on aerospace fluid components would be incomplete without an appropriate discussion of fluid mechanics, heat transfer, dynamic analysis, contamination, materials, reliability, and environments. Content and organization is based on the philosophy that valve design and development may be appropriately considered as the design, selection, and integration of basic modules or building blocks. Therefore, Section 6.0, Modules, reflects more of a design orientation than those sections dealing with components. The technical content of the handbook is divided into the following sixteen sections:

Section 2.0, Heat Transfer, presents basic heat transfer theory, working relationships, and data useful to the fluid component designer.

Section 3.0, Fluid Mechanics, covers basic fluid flow theory and data on pressure drop and flow rate determinations for pneumatic and hydraulic components.

Section 4.0, Fluid Systems, discusses the operation of a variety of typical aerospace fluid systems in terms of the function of components making up the fluid system.

Section 5.0, Fluid Components, describes design and selection factors, design types, applications, and limitations of fluid components. This section also includes a glossary of fluid component terminology.

Section 6.0, Modules, presents design and selection information on basic fluid component elements such as seals, valving units, springs, bearings, bellows, diaphragms, and actuators common to many types of fluid components.

Section 7.0, Dynamic Analysis, covers dynamic analysis techniques including servo theory, vibration and shock analysis, and component dynamic performance analysis.

Section 8.0, Computers, describes the use of analog and digital computers as tools useful to the fluid component designer.

Section 9.0, Specifications, discusses factors that are important in the preparation of adequate component specifications.

Section 10.0, Contamination and Cleaning, discusses techniques for controlling contamination, and presents design considerations for minimizing contamination susceptibility of fluid components.

Section 11.0, Reliability, briefly summarizes reliability definition and mathematics, and discusses design considerations for maximizing fluid component reliability.

Section 12.0, Materials, includes material property data important to the fluid component designer, with major emphasis on the selection of materials compatible with fluids handled in aerospace applications.

Section 13.0, Environments, describes environmental extremes to which aerospace fluid components must be designed and discusses design techniques for making components compatible with imposed environmental conditions.

Section 14.0, Stress Analysis, is a compilation of basic structural design criteria, stress and deflection equations, and design data from a wide variety of sources selected to assist the fluid component designer in those aspects of stress analysis particularly related to aerospace fluid components.

Section 15.0, Component Testing, contains definitions, suggestions, and recommendations on both the general types of tests to which components are subjected and the unique aspects of those tests performed only on certain types of components.

Section 16.0, Fluidics, has been prepared in response to an exceptionally high level of interest reflected in the questionnaires returned by users of the handbook. This section summarizes the technology of fluidics and introduces the basic language of this new technology, to assist the fluid component designer in evaluating the applicability of fluidics to various aerospace systems.

1.3 HANDBOOK FORMAT

The general format was designed to facilitate use and to accommodate periodic revisions as the handbook was updated and expanded. The use of a ring binding and looseleaf pages makes the handbook useful as a repository for pertinent information collected by the individual user. In subdividing the major topics, every attempt has been made to make the breakdown logical, thus facilitating lookup and use. Appropriate tab dividers are provided for quick location of sections, and a detailed table of contents is presented at the beginning of each section. Each section is subsequently divided into numbered sub-sections, subtopics, and detailed topics, each having an appropriate descriptive title. The commonly accepted method of paragraph numbering has been adopted throughout, consisting of a decimal system outlined as follows:

3.0	SECTION
3.1	SUB-SECTION
3.1.1	SUB-TOPIC
3.1.1.1	DETAILED TOPIC

1.3.1 Figures, Tables, and Equations

Figures, tables, and equations are numbered according to the lowest numbered subdivision in which they appear, such as Figure 3.1.1.1b, Table 3.1.1.1b, etc. The letter (a, b, c ...) following the numeric designation is used when more than one figure, table, or equation appears within any numbered subdivision. Occasionally a number is followed by -1, -2, -3, etc., where figures, tables, or equations have been added to the original text.

1.3.2 Page Format

In the lower outside corner of each page a number precedes a dash, indicating the sub-section or sub topic appearing on that page. These numbers serve to locate easily sub-sections and sub-topics, since the handbook indexing system is based on paragraph numbers. The dash numbers have been provided for aiding in collating the handbook, and have no lookup value. Dash numbers followed by a, b, c, etc., are used where necessary when new pages are added to the original text. As an additional lookup feature, each page is identified in the upper righthand corner by the major topic or topics presented on that page. Each page also has the date of issue and/or revision to indicate the currency of data appearing therein.

1.4 REFERENCES AND BIBLIOGRAPHY

All references mentioned in the text are listed in the bibliography. Each reference is identified by a number which was assigned to it as it was located during the handbook literature search. Missing numbers are accounted for by the fact that not all references located during the search were actually used in the preparation of the handbook. References are cited by reference number in the text and in lists at the end of each section or sub-section. Reference numbers are actually two numbers, joined by a hyphen, as in 19-24. The number preceding the hyphen (19) identifies the source of the reference, the same number being used for all references from one given source. The number following the hyphen (24) identifies the specific reference. Manufacturers' literature (catalogs, brochures, drawings, etc.) is designated by the letter "V" preceding the hyphen, in which case the number following the hyphen identifies the manufacturer source rather than citing a specific catalog, brochure, etc. An alphabetical list of numerical reference sources may be found immediately following the bibliography, followed in turn by a Manufacturers' Source List. A complete listing of all documents and sources cataloged in the handbook program through the preparation of Revision A (October 1965) may be found in "A Bibliography of Aerospace Valve and Fluid Component Technology," Technical Report AFRPL-TR-65-239, December 1965, TRW Systems Report No. 9736.5-66-225, and Addendum A, AFRPL-TR-67-157, May 1967, TRW Systems Report No. 07891-6001-R000.

1.5 SYSTEM OF UNITS AND SYMBOLS

Two systems of units have been used in the handbook in accordance with commonly accepted usage. The unit force-mass system is used in the sections dealing with fluid mechanics, while the English gravitational system is used in those sections dealing primarily with solid mechanics. The ideal system of units where changes in gravitational field must be considered, such as space systems, is an absolute system having the quantity of matter (mass) as a basic dimension. Unfortunately, an absolute system of units, such as the cgs system, is not in common usage in American engineering practice.

The unit force-mass system used in the fluid Mechanics Section has gained general acceptance in the fields of fluid mechanics and thermodynamics. This system is a compromise between the English gravitational system, which has fundamental dimensions of force (F), length (L), and time (t), and the absolute system, which has fundamental dimensions of mass (M), length (L), and time (t). The unit force-mass system which has the pound-force (lb_f) as the force unit also has a basic dimension of mass, the pound-mass (lb_m). In contrast, mass in the English gravitational system is a derived quantity having units of lb_f—sec^2/ft, called the *slug*. Although the unit force-mass system has the advantage of having a fundamental dimension for mass, it has the disadvantage of having four dimensions—force (F), mass (M), length (L), and time (t) instead of three as in the absolute system (mass (M), length

(L), and time *(t)*) and in the gravitational system (force *(F)*, length *(L)*, and time *(t)*). Another disadvantage of the unit force-mass system is the requirement for a proportionality constant relating force, mass, and acceleration, thus complicating many equations by the addition of the proportionality constant $1/g_c$. These disadvantages, however, are outweighed by the distinct advantage of dealing with physical quantities that are independent of gravitational considerations (density lb_m/ft^3, specific heat Btu/lb_m).

Since the English gravitational system is commonly accepted in the fields of aerodynamics and solid mechanics, this system has been used in the handbook for the sections dealing primarily with solid mechanics, i.e., Section 6.0, Modules, Section 7.0, Dynamic Analysis, Section 13.0, Environments, and Section 14.0, Stress Analysis. As stated above, this system has a significant advantage over the unit force-mass system in that equations are less complicated by eliminating the conversion constant $1/g_c$. Symbols used are either defined where they are used, or are defined on foldout symbol lists such as those appearing at the end of Sections 2.0 and 3.0. For clarity, in Section 14.0, Stress Analysis, the dimension of force *(F)* is shown in units of lb_f, borrowing from the unit force-mass system.

The addition of Section 16.0, Fluidics, presented a unique problem concerning units. The basic standard documents for fluidics, such as MIL-STD-1306 and SAE ARP 993A, present both International System (SI) and "Standard (English)" units, the latter referring to the English gravitational system. In order to make the Fluidics Section as compatible as possible with the Fluid Mechanics Section, the unit force-mass system has been substituted for the English gravitational system in Section 16.0, Fluidics.

1.6 HANDBOOK REVISIONS

Because the Aerospace Fluid Component Designers' Handbook has been part of a continuing program to compile and disseminate basic information on aerospace fluid component technology, revised sections and/or new material have been issued to handbook recipients as necessary, giving appropriate directions regarding the insertion of new and discarding of old material. Revised pages indicate the new issue date and the issue date of superseded material. A complete revision record giving information on pages added and/or deleted is included as Appendix B.

Readers are encouraged to submit any constructive comments, recommended changes, and additional data for inclusion in the handbook to the Design Handbooks Branch Aeronautical Systems Division (ASNPS-40), Wright-Patterson AFB, Ohio, 45433.

Individuals and organizations presently on the controlled distribution list for the Aerospace Fluid Component Designers' Handbook will be placed on the distribution list for AFSC DH 3-6 automatically. Controlled distribution handbooks are identified by a number between 1 and 2000 on the upper right hand corner of the title page. Handbooks numbered 5000 and above, and those received through DDC are *not* on the controlled distribution list and should request DH 3-6 as outlined below.

1.7 FUTURE HANDBOOK AVAILABILITY

The Aerospace Fluid Component Designers' Handbook has been very well received throughout the industry. Accordingly, it has been decided by the Air Force Systems Command (AFSC) Headquarters to incorporate the handbook into the new AFSC Design Handbook series in AFSC DH 3-6, Fluid Components, part of series 3-0, Space and Missile Systems (brown volumes). The overall AFSC Design Handbook Program is described in detail in DH 1-1, General Index and Reference.

The Design Handbooks are provided without charge to qualified agencies supporting USAF technical interests and objectives. However, they are subject to special export controls, and each transmittal to foreign governments or foreign nationals may be made only with prior approval of the Design Handbooks Branch (ASNPS-40), Aeronautical Systems Division, Wright-Patterson AFB, Ohio 45433. U.S. Government organizations must justify requests based on their need for specific handbooks in carrying out their assigned missions and functions. Non-Government organizations must satisfy at least one of the following requirements and submit the appropriate justification in writing to ASD (ASNPS-40):

a) The specific handbook is cited in an active Air Force contract or current Invitation for Bid. In such cases, identify the handbook(s) and state the contract or IFB number, title of the work, and responsible Government office.

b) The specific handbook is needed in private or education technical efforts, the results of which will directly benefit the Air Force. In such cases, identify the handbook(s) and describe the requesting organization, work, and expected benefits to the Air Force.

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ILLUSTRATIONS

TABLES

2.1 INTRODUCTION

The purpose of this section is to provide the designer of fluid components with a ready reference to the basic concepts of heat transfer, along with some practical applications of these concepts in the design of fluid components. Symbols, dimensions, and units used are on a foldout sheet at the end of the section.

2.2 METHODS OF HEAT TRANSFER

The three essentially different methods of heat transfer are termed conduction, convection, and radiation. *Conduction* may be described as the transfer of heat from one point of a body to any adjacent point, or from one body to another in contact with it without displacement of the matter within the body. Heat is transferred by *convection* when bodily movement of the heated substance takes place. Heat transfer by convection applies only to mobile substances such as liquids, vapors, and gases. When a fluid becomes heated at the bottom of a container it moves to the top because its density is less than the colder fluid above it. This process is called *natural convection.* If the fluid is transported by means other than density variations — such as by circulating air with a fan or by flowing liquid through a pipe by means of a pressure difference — the process is referred to as *forced convection. Radiation* is the transfer of heat from a high temperature source to a body of lower temperature some finite distance away by the passage of radiant energy. The radiant energy is then converted back to internal energy when it is absorbed by the receiving body. A common example of heat transfer by radiation is the heating of objects exposed to the sun. Radiation, unlike conduction and convection, does not require a transmitting substance and, therefore, can take place through a vacuum.

Heat transmission is seldom confined to just one method at a time. For example, conduction problems often involve radiation and convection, while convection problems often involve radiation and conduction. Radiation, in other than a vacuum environment, involves conduction and convection. Sometimes it is possible to calculate separately the heat quantities transmitted by each process, but more often only total heat transfer can be determined.

2.2.1. Conduction

In the transmission of heat by conduction, the quantity of heat flowing per unit time is directly proportional to (1) the conductivity of the material involved, (2) the cross-sectional area of the conductor perpendicular to the direction of heat flow, and (3) the temperature gradient across the conductor being considered. Under steady-state conditions, the amount of heat flowing into a body or through an area is exactly equal to the amount of heat flowing out, so that both the temperature and heat quantity during any time interval at each point remain constant. Conversely, if either the temperature or heat quantity at any given point varies with the time, heat flow is transient or unsteady.

2.2.1.1 STEADY-STATE CONDUCTION. In 1822, Gautheir-Villars of Paris published J. B. Fourier's "*theorie analytique de la chaleur*" in which the following basic equation for steady-state heat flow by conduction was presented

$$q = -kA \frac{dT}{dL} \qquad \text{(Eq 2.2.1.1a)}$$

where

q = rate of heat transfer, Btu/hr
k = thermal conductivity, Btu/(hr) (ft²) (°F/ft)
A = area of surface perpendicular to heat flow, ft²
T = temperature, °F
L = material thickness in direction of heat flow, ft

The minus sign in Fourier's equation is introduced so that the heat flow, q, may be expressed as a positive number since the temperature change, dT, decreases or is negative while passing through a positive distance, dL. The expression $\frac{dT}{dL}$, the rate of change of temperature with thickness, is called the temperature gradient, G. For practical purposes under steady-state conditions $G = \frac{T_2 - T_1}{\Delta L}$, where $T_2 - T_1$ is the temperature difference across the material thickness, ΔL. By substitution, Equation (2.2.1.1a) can be written

$$q = -kAG \qquad \text{(Eq 2.2.1.1b)}$$

2.2.1.2 CONDUCTIVITY OF SOLIDS. The value of thermal conductivity varies from one solid to another. It also varies for any given solid, depending upon the temperature at which it is measured and the exact composition and purity of the sample. For example, minute amounts of arsenic in copper will reduce its conductivity to approximately one-third that for pure copper. Figure 2.2.1.2a shows how k varies with temperature for several common metals and alloys, and Figure 2.2.1.2b shows how k varies with temperature for several non-metals. Tables of thermal conductivities are given in Section 12.0, "Materials." It should be noted that the values of k given in handbook tables are usually accurate for homogeneous solids only. They are not accurate for non-homogeneous solids such as porous materials which have a variable moisture content and density.

2.2.1.3 CONDUCTIVITY OF GASES AND LIQUIDS. The superior heat transfer characteristic of liquids over gases is illustrated by the larger values of conductivities of liquids. The values of k for most non-metallic pure liquids lie between 0.05 and 0.40 Btu/(hr) (ft²) (°F/ft), while for most gases at standard conditions k ranges between 0.005 and 0.015. By theoretical analysis, J. C. Maxwell devised the following equation for the conductivity of a gas expressed as a function of its specific heat and viscosity (Reference 130-1) :

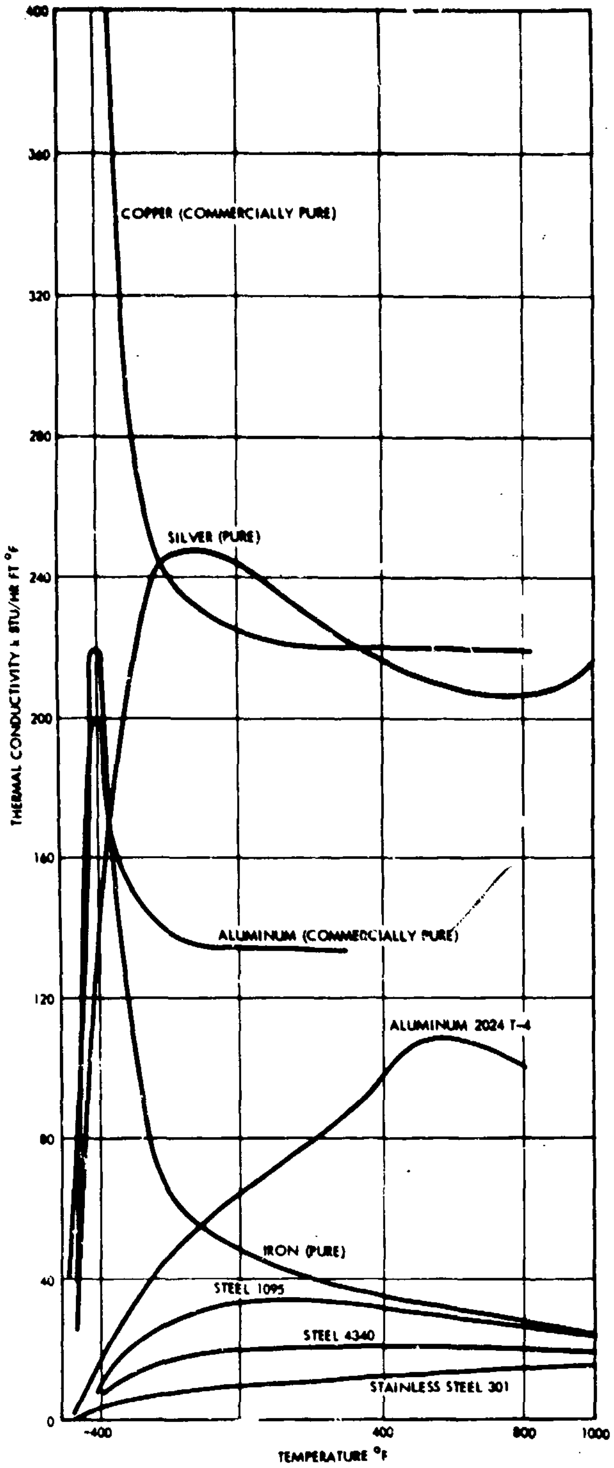

Figure 2.2.1.2a. Variations in Thermal Conductivity with Temperature for Several Common Metals and Alloys

(From References 23-38, 82-5, and 82-8.)

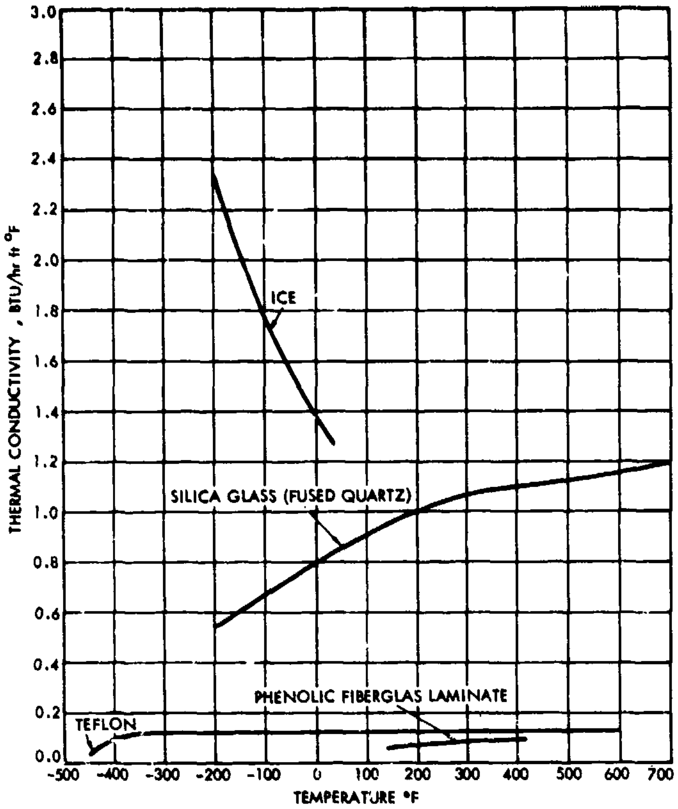

Figure 2.2.1.2b. Variations in Thermal Conductivity with Temperature for Several Common Nonmetals

(Adapted from "Aerospace Applied Thermodynamics Manual," SAE, Inc., Committee A-9, pp. D-24 and D-29)

$$k = c_v \mu_m a \qquad \textbf{(Eq 2.2.1.3)}$$

where k = thermal conductivity, Btu/(hr) (ft²) (°F/ft)

c_v = specific heat of gas at constant volume, Btu/lb_m °F

μ_m = absolute viscosity, lb_m/ft-hr

a = a constant having theoretical values as follows:

2.45 for monatomic gases (e.g. He, A)
1.90 for diatomic gases (e.g. H_2, N_2, O_2, CO)
1.70 for triatomic gases (e.g. CO_2, NO_2, H_2O, O_3)
1.30 for gases with more complex molecules (e.g. CH_4, C_2H_6, C_2H_2)

The above equation is based on the kinetic theory of gases and checks well with experimental results. Accurately determining the conductivities of gases is complicated by convection currents in the test samples, but over the years quite accurate values have been determined.

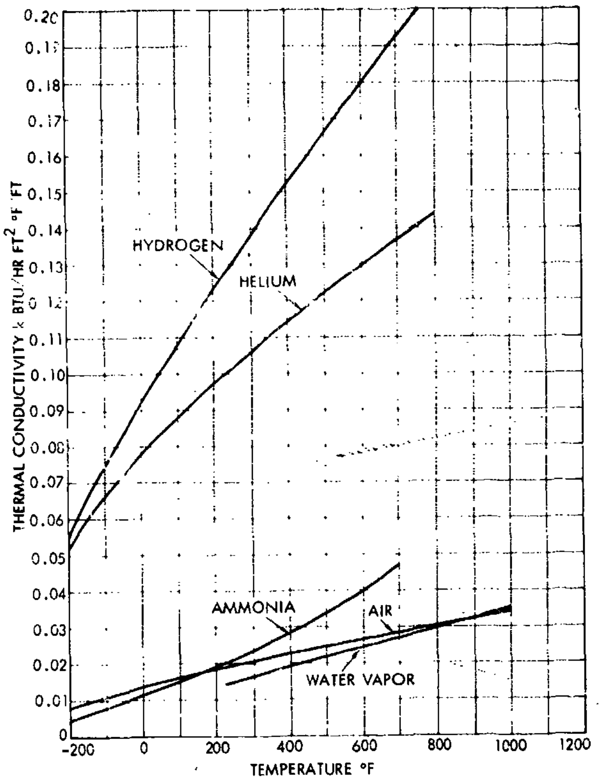

Figure 2.2.1.3a. Variations in Thermal Conductivity with Temperature for Several Common Gases and Water Vapor

(Adapted from "Aerospace Applied Thermodynamics Manual," SAE, Inc., Committee A-9, pp. B-3, 4, 8, 9, 12)

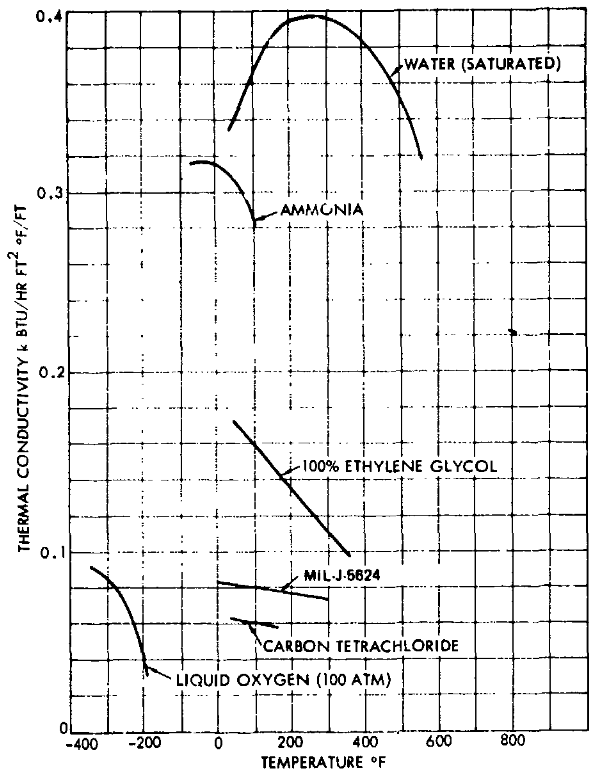

Figure 2.2.1.3b. Variations in Thermal Conductivity with Temperature for Several Common Liquids

(Adapted from "Aerospace Applied Thermodynamics Manual," SAE, Inc., Committee A-9, pp. C-1, [illegible], 21, 30, 36)

While the thermal conductivity of a gas will increase with temperature, it is virtually independent of pressure, provided the pressure is below the critical pressure of the gas and above the point where the mean free path of the gas molecule is large compared to the width of gas space measured along the heat transfer path. Heavier gases usually have lower k values than lighter gases.

In most cases, the thermal conductivity of liquids decreases with increase in temperature. For water, which is the best conductor among the non-metallic liquids, k increases to a temperature of approximately 250°F, then decreases with further increases in temperature. The relationships between k and temperature for a number of gases and liquids are illustrated in Figures 2.2.1.3a and 2.2.1.3b, respectively.

2.2.1.4 TRANSIENT CONDUCTION. Heat transfer by transient conduction is determined from the following basic consideration:

$$Q_{in} - Q_{out} = Q_{stored} \qquad \textbf{(Eq 2.2.1.4a)}$$

where Q_{in} = total heat input, Btu

Q_{out} = total heat output, Btu

Q_{stored} = heat absorbed by the conducting material, Btu

In differential form:

$$Q_{in} = -kA \frac{\partial T}{\partial L} dt$$

$$Q_{out} = Q_{in} + \frac{\partial}{\partial L} Q_{in}\, dL$$

$$= -kA \frac{\partial T}{\partial L} dt + \frac{\partial}{\partial L}\left(-kA \frac{\partial T}{\partial L} dt\right) dL$$

$$Q_{stored} = c_p \rho A\, dL \frac{\partial T}{\partial t} dt$$

and Equation (2.2.1.4a) becomes

$$-kA \frac{\partial T}{\partial L} dt - \left[-kA \frac{\partial T}{\partial L} dt + \frac{\partial}{\partial L}\left(-kA \frac{\partial T}{\partial L} dt\right) dL\right]$$

$$= c_p \rho A\, dL \frac{\partial T}{\partial t} dt \qquad \textbf{(Eq 2.2.1.4b)}$$

For one-dimensional transient conduction through a flat slab, Equation (2.2.1.4b) reduces to

$$\frac{\partial T}{\partial t} = \frac{k}{\rho c_p}\left(\frac{\partial^2 T}{\partial L^2}\right) = \alpha \left(\frac{\partial^2 T}{\partial L^2}\right) \qquad \textbf{(Eq 2.2.1.4c)}$$

where L = distance along heat path

$\frac{\partial T}{\partial t}$ = change in temperature with time at a given location along the heat path

$\frac{\partial^2 T}{\partial L^2}$ = rate of change in temperature gradient with position along the heat path determined at any given location

$\alpha = \frac{k}{\rho c_p}$ = thermal diffusivity

Several techniques have been developed for numerical and graphical solutions to this second order differential equation. The Dusinberre method is one technique wherein a continuous process is replaced by a stepwise one.

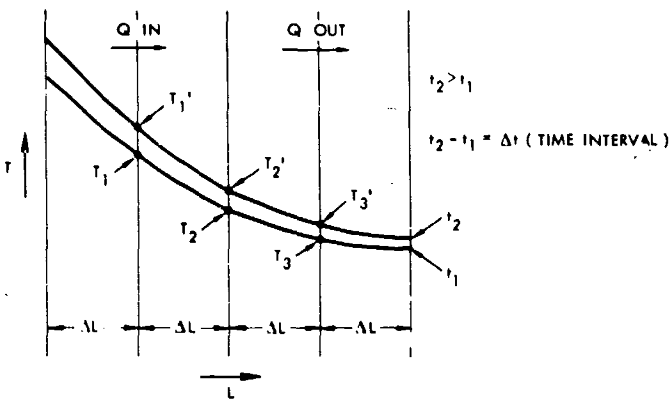

Figure 2.2.1.4. Incremental Sectioning of a Flat Plate for Determining One Dimensional Transient Heat Transfer Using the Dusinberre Numerical Method

Referring to Figure 2.2.1.4

$\frac{\partial T}{\partial t}$ is replaced by $\frac{T_2' - T_2}{\Delta t}$

$\frac{\partial^2 T}{\partial L^2}$ is replaced by $\dfrac{\frac{T_2 - T_1}{\Delta L} - \frac{T_3 - T_2}{\Delta L}}{\Delta L}$

This technique is the basis for programming digital computer solutions for transient conduction problems. The Dusinberre method and several other numerical methods are described in detail in References 130-1 and 2-31.

For one-dimensional transient conduction through the wall of a right circular cylinder, Equation (2.2.1.4b) becomes

(Eq 2.2.1.4d)

$$\left(\frac{\partial T}{\partial t}\right)_R = \alpha\left[\left(\frac{\partial^2 T}{\partial R^2}\right) + \frac{1}{R}\left(\frac{\partial T}{\partial R}\right)\right]$$

where R = radius of cylinder

$\frac{\partial T}{\partial t}$ = change in temperature with time at any given radial location

$\frac{\partial^2 T}{\partial R^2}$ = ratio of change in temperature gradient with radial position along the heat path determined for any given radial position

$\frac{\partial T}{\partial R}$ = radial temperature gradient

α = thermal diffusivity, $\frac{k}{\rho c_p}$

Where R is large in relation to wall thickness the final term can be neglected, reducing the equation to the flat plate configuration. The final term accounts for the fact that the cross-sectional heat transfer area increases with increasing R.

Transient conduction plays an important role in many missile and space applications involving high temperatures for limited time periods. Under such conditions many materials which are completely unsatisfactory for high temperature service under steady-state conditions are found to be perfectly satisfactory for short time applications. Silicone rubber, for example, although limited under steady-state temperature conditions to approximately 400°F, has been used satisfactorily in applications where the material is exposed to direct flame at temperatures of over 5000°F for time durations limited to a few minutes. Materials particularly well suited to short time applications at high temperatures are those having low thermal conductivities, k, and high volumetric specific heats (product of density and specific heat at constant pressure ρc_p) as indicated by Equation (2.2.1.4c).

2.2.2 Convection and Conduction

In almost all engineering problems involving convection, conduction also plays a significant role in the heat transfer process. Therefore, convective heat transfer is treated in the following paragraphs in terms of combined convection and conduction.

2.2.2.1 FLOWING FLUIDS. A common heat transfer problem involving both convection and conduction is the transfer of heat from a flowing fluid (liquid or gas) to a flat or cylindrical wall, through the wall, and then from the wall to the atmosphere. The solution of such a problem involves the use of a combined conduction, convection, and radiation coefficient, h, for the fluid and for the air. The expression for the heat transfer rate between a fluid and an adjacent surface takes the form

$$q = h\,A\,(T_B - T_W) \qquad \textbf{(Eq 2.2.2.1a)}$$

where q = heat transfer rate, Btu/hr

h = total surface or film coefficient of heat transfer, Btu/(hr) (ft²) (°F)

A = surface area, ft^2

T_B = bulk fluid temperature (average temperature of flowing fluid), °F

T_W = temperature of surface or wall temperature, °F

The value of h takes into account the effects of ambient conduction, radiation, and convection. It varies widely for different fluids depending upon such parameters as viscosity, density, thermal conductivity, velocity of fluid stream, specific heat, and the characteristic dimensions of the apparatus involved (tube diameter and surface roughness) and can be expressed as follows:

$$h = h_c + h_r \qquad \text{(Eq 2.2.2.1b)}$$

where h = total film coefficient

h_c = film coefficient for combined conduction and convection

h_r = film coefficient for radiation

For many practical fluid flow problems where temperature differences do not exceed several hundred degrees Fahrenheit, the influence of h_r is small and can be neglected without significant error.

The film coefficient of convection and conduction alone for fluids undergoing turbulent flow in horizontal tubes may be written

$$h_c = 0.023\,\frac{k}{D}\left(\frac{DV\rho}{\mu_m}\right)^{0.8}\left(\frac{c_p\mu_m}{k}\right)^{0.4} \qquad \text{(Eq 2.2.2.1c)}$$

where h_c = film coefficient of combined conduction and convection, Btu/(hr) (ft^2) (°F)

k = conductivity of fluid, Btu/(hr) (ft^2) (°F/ft)

D = diameter of fluid stream, ft

$\left(\frac{DV\rho}{\mu_m}\right)$ = Reynolds number

$\left(\frac{c_p\mu_m}{k}\right)$ = Prandtl number

The Nusselt number $\left(\frac{h_cD}{k}\right)$ can be obtained by rearranging Equation (2.2.2.1c). Equation (2.2.2.1c) applies where the Reynolds number is in the range of 10,000 to 120,000, the Prandtl number is between 0.7 and 120, the length of the tube is at least 60 diameters, and the temperature drop across the film is not large. The physical properties of the fluid noted in Equation (2.2.2.1c) should be determined at the bulk temperature of the fluid stream. A tabulation of film coefficient relationships for laminar and turbulent flow through and around various shapes is given in Reference 23-38 (Part C of Section 1).

The expression for the heat transfer rate from a fluid

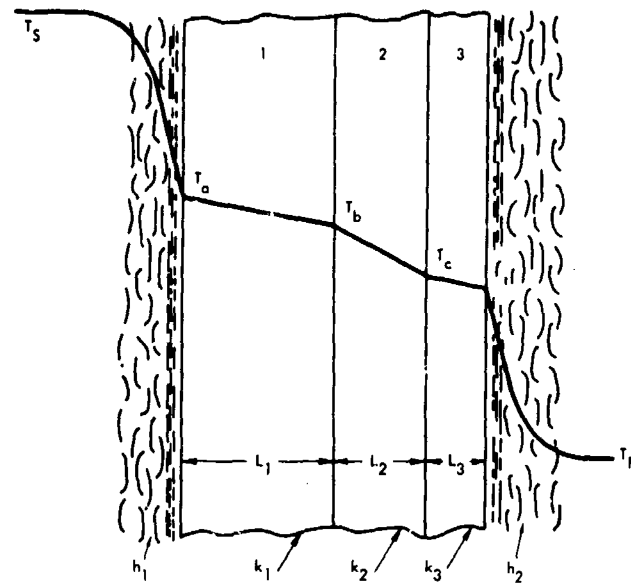

Figure 2.2.2.1. Temperature Gradients Across a Multilayered Plane

(Reprinted with permission from "Applied Thermodynamics," V. M. Faires, p. 425, Copyright 1947, Macmillan Co., New York)

through any composite wall (Figure 2.2.2.1) to the surrounding medium can be written

$$q = \frac{A\,(T_S - T_F)}{\frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \cdots \frac{L_n}{k_n} + \frac{1}{h_2}} \qquad \text{(Eq 2.2.2.1d)}$$

Equation (2.2.2.1d) in turn may be written

$$q = U\,A\,(T_S - T_F) \qquad \text{(Eq 2.2.2.1e)}$$

where U, the transmittance, is equal to the reciprocal of the sum of individual thermal resistivity terms $\frac{1}{h_1}, \frac{L_1}{k_1}$ ... etc. Typical values of U are given in Table 2.2.2.1. The

Table 2.2.2.1. Value Range for the Transmittance, U, of Several Common Hydraulic Components
(Reference 19-182)

RANGE OF U Btu/(hr) (ft^2) (°F)	COMPONENT DESCRIPTION
2 - 5	Steel tank surrounded by inhibited air circulation
5 - 10	Steel tank in normal indoor surroundings
10 - 13	Steel tank surrounded by good natural convection (guided air current)
25 - 60	Oil-to-air heat exchanger
80 - 100	Oil-to-water heat exchanger

term ($T_S - T_F$) is the total differential temperature across the area under consideration. Equation (2.2.2.1d) is useful in helping the engineer keep track of all thermal resistance terms. By increasing the value of one or more terms, heat will flow at a slower rate which would be the desired effect when designing insulation for hot gas or cryogenic lines. Conversely, by reducing the number of resistances or their size, the rate of transfer will increase. If one term is considerably larger than all others, little benefit is gained by reducing any but the largest. Detailed analysis of conduction-convection heat transfer problems involves full knowledge of fluid characteristics and requires the calculation of the surface film coefficient. This subject is treated in more detail in References 130-1, 132-1, and 134-1. The use of a digital computer to solve Equation (2.2.2.1d), where variations of k with temperature are considered, is presented in Detailed Topic 8.3.3.2.

2.2.2.2 **BOILING LIQUIDS.** Heat transfer to boiling liquids occurs in several kinds of heat exchange apparatus such as steam boilers, evaporators, refrigerating plants, and regeneratively cooled liquid rocket motors. In spite of the wide use of equipment using boiling liquids, information concerning heat transfer to boiling liquids is still far from complete. When a liquid is heated without boiling, the heat transfer can be predicted by the laws of convection and film conductance as indicated in Detailed Topic 2.2.2.1. When boiling occurs, vapor bubbles of various sizes and shapes are formed, then liberated at varying rates at the heating surface in contact with the liquid. This creates turbulence of uncertain magnitude, and makes mathematical prediction of heat transfer difficult.

Boiling takes place when heat is added to a fluid which has been heated to its saturation temperature. The rate at which heat is transferred to the fluid, which determines the rate of boiling, is a function of the temperature of the heated surface (wall) and the saturation temperature of the fluid. Figure 2.2.2.2 shows the change in heat transfer rate to a pool of water (pool boiling) as a function of the difference between the temperature of a heated wire and the water saturation temperature, ΔT. Boiling at moderate values of ΔT is termed "nucleate boiling" because it takes place through the action of tiny cells of air or other gases, or particles that adhere to the heating surfaces and serve as nuclei for the formation of vapor bubbles. These nuclei may be attached to the heating surface when it is first wetted or they may precipitate out of the liquid after contact.

As the heating surface temperature rises and ΔT increases, liquid boiling takes place more and more rapidly, with the density of bubbles and speed of formation increasing until a temperature is reached where the bubbles become dense enough to form a solid film of vapor between the heating surface and surrounding liquid. The vapor film thus formed acts as an insulator, and the temperature of the heated surface can be increased considerably without an increase in heat transfer. In fact, a lowering of heat transfer occurs until the radiation mechanism comes into full play as a result of the increased temperature. If heat is continually added, the surface temperature can increase to a point where burn-through or melting is possible. At a ΔT, just before film boiling begins, a point of maximum nucleate boiling heat transfer is reached which is known as the upper limit of nucleate boiling, or q ultimate (Point C on Figure 2.2.2.2).

In most fluid systems where boiling is encountered, the fluid is in a flowing state rather than in a pool. Increasing fluid velocity causes a significant increase in rate of heat transfer because of the action of the flowing fluid in carrying away vapor bubbles as they are formed. As indicated, boiling heat transfer cannot be accurately predicted mathematically; however, empirical data has been determined for most common fluids. Further discussion of nucleate, film, and pool boiling may be found in References 130-1, 134-1, 2-24, and 325-1.

2.2.2.3 **COOLING FINS.** Another common conduction-convection problem encountered by fluid component designers is the calculation of the heat transfer characteristics of cooling fins. The solution of such problems depends on the configuration of the fins. Two recommended references on the design of fins are Reference 23-38 (Part C of Section 1) and Reference 325-1.

2.2.3 Radiation

Heat transfer by radiation is the process of transferring heat energy between two bodies by electromagnetic radiation similar to light. Like light, the exact nature by which radiant heat energy is transmitted is explained by two

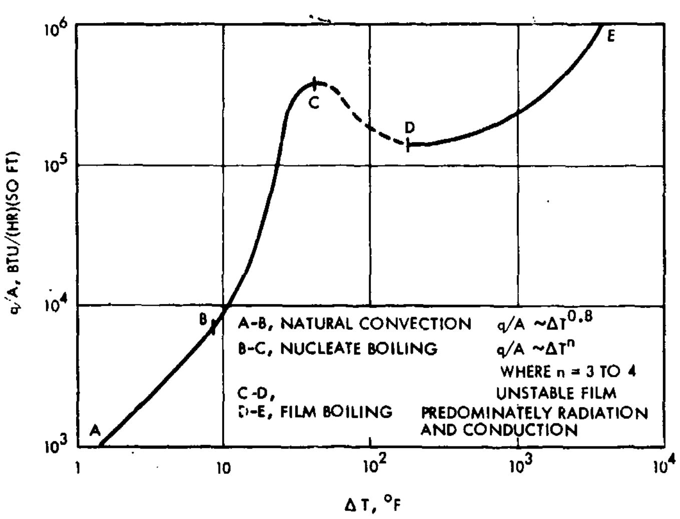

Figure 2.2.2.2. Typical Boiling Data
(A wire heated electrically in a pool of water at atmospheric pressure)
(From "Journal of the Society of Mechanical Engineers," (Japan) S. Nukiyama, vol 37, p. 367, 1934)

different theories — wave propagation and quantum propagation.

In 1792 the "theory of exchanges" was proposed by Prevost. This theory stated that a continuous exchange of energy exists between bodies as a result of the bilateral process of radiation and absorption. In other words, if two bodies at different temperatures are completely isolated in a vacuum from all other bodies but not from each other, there will be a radiant energy interchange between the two bodies. The warmer body receives less energy from the colder body than it radiates to it and, consequently, its temperature will decrease. The colder body temperature increases until a state of equilibrium is reached, at which time each body is radiating and absorbing the same amount of energy. In line with this theory, which agrees well with laboratory experiments, all bodies continue to radiate thermally to other bodies at all temperatures above absolute zero.

2.2.3.1 ABSORPTIVITY, TRANSMISSIVITY, AND REFLECTIVITY. Thermal radiation striking a body may be absorbed by it, transmitted through it, or reflected from it. The degree to which each of these phenomena occurs is dependent on several factors, including the substance of which the body is composed, its surface finish and color, and the wave length of the radiant energy. For example, most solids (with the exception of transparent solids such as glass) absorb and reflect practically all thermal radiation and transmit very little. Liquids and gases, on the other hand, have a higher degree of transmission and absorb and reflect less than solids. The fraction of total energy received and absorbed by a body is called the *absorptivity* of the body. The fraction of total energy transmitted through it is termed the *transmissivity*, and the fraction reflected by it is called the *reflectivity*.

2.2.3.2 BLACK BODY RADIATION. In nature no materials or substances exist which absorb all radiation received and reflect none. Certain substances such as lamp black and platinum black absorb all but very small portions of the incident radiation; these substances are termed "black" from a thermal radiation standpoint. An idealized "black" substance or "black body" is one which would absorb all radiant energy falling upon it and reflect or transmit none. In addition, the idealized black body would emit or radiate the maximum amount of energy for any surface at any given temperature. A black body is also termed, therefore, a perfect radiator. Although a true black body does not exist in nature, one can be closely approximated for experimental purposes by a hollow enclosure blackened on the inside and penetrated by a small hole through which thermal radiation can pass. Radiation entering the enclosure will undergo repeated reflection inside and very little will escape. One such device commonly used for laboratory experimentation is a closed hollow cylinder 8 inches long, 2 inches in diameter, and with a one-half inch diameter hole in one end. When such an enclosure is heated, the radiation emitted from the hole is termed *black body radiation*.

2.2.3.3 STEFAN-BOLTZMANN EQUATION. The manner in which the temperature of a black body affects the amount of radiation emanating from it was proposed from empirical data by the Austrian physicist Josef Stefan (1835-1893), and was later theoretically calculated by his contemporary, Ludwig Boltzmann (1849-1906).

The relationship developed by Stefan-Boltzmann states that the amount of radiation from a black body is proportional to the fourth power of the absolute temperature.

Thus

$$E_B = \sigma T^4_{abs} \qquad \textbf{(Eq 2.2.3.3a)}$$

where E_B = black body radiant energy density, Btu/hr ft^2

σ = Stefan-Boltzmann constant of proportionality, 0.1713×10^{-8} Btu/(hr) (ft^2) ($^\circ$R)^4

T_{abs} = absolute temperature, $^\circ$R

This relationship shows that the amount of radiant energy emitted from a body is constant if its temperature remains constant. The rate of radiant energy emission from a body is usually stated in Btu/hr and the radiant energy density is stated in Btu/hr ft^2.

In contrast to conduction and convection, which vary directly with temperature difference, radiation varies with the fourth power of the absolute temperature. For example, if the absolute temperature doubles, the radiant energy will increase sixteen-fold.

The Stefan-Boltzmann Equation (2.2.3.3a) expresses the radiant energy emitted by a black body. Since such a body exists in theory only, it is necessary to introduce a proportionality constant which will make the equation usable for real materials and various surface finishes. In 1859 Kirchhoff derived a law which states that at a given temperature the total radiant energy from any body is equal to its absorptivity, multiplied by the total emissive power of a perfect black body at that temperature. This is known as Kirchhoff's Law, and can be expressed mathematically as

$$E = a\,E_B \qquad \textbf{(Eq 2.2.3.3b)}$$

where E = radiant energy of the body at a given temperature

a = absorptivity of the body for the same temperature

E_B = radiant energy of a black body at the same temperature

The ratio of the total radiant energy of any body to the total radiant energy of a black body at the same temperature is called the emissivity, ϵ. The emissivity of a surface is numerically equal to its absorptivity when the radiating source is a black body at the same temperature as the surface.

$$\frac{E}{E_B} = a = \epsilon \quad \text{(Eq 2.2.3.3c)}$$

or

$$E = \epsilon E_B \quad \text{(Eq 2.2.3.3d)}$$

When Equation (2.2.3.3b) is combined with Equation (2.2.3.3a) it yields the expression

$$E = \epsilon \sigma T^4_{abs} \quad \text{(Eq 2.2.3.3e)}$$

This equation allows the calculation of the radiant energy of any body or surface providing the temperature, T, and emissivity are known. Emissivity varies with surface temperature and the condition of the surface. Table 2.2.3.3a lists the emissivities for a wide variety of materials and surface finishes. These values are normal emissivities for radiation in a direction normal to the emitting surface. Examination of the data in Table 2.2.3.3a reveals the following general characteristics:

a) Highly polished metals have low emissivities

b) The emissivity of most metals and their oxides increases with increase in temperature

Table 2.2.3.3a. Emissivity of Metals and Their Oxides

(Reprinted with permission from "Mechanical Engineers Handbook," L. S. Marks, Copyright McGraw-Hill Book Company, Inc., 1952, p. 4-110)

SURFACE	TEMPERATURE, °F*	EMISSIVITY*
Aluminum		
Highly polished	440-1070	0.039—0.057
Polished	73	0.040
Rough plate	78	0.055—0.07
Oxidized at 1110F	390-1110	0.11—0.19
Oxide	530-1520	0.63—0.26
Alloy 75ST	75	0.10
75ST, repeated heating	450-900	0.22—0.16
Brass		
Highly polished	497-710	0.03—0.04
Rolled plate, natural	72	0.06
Rolled, coarse-emeried	72	0.20
Oxidized at 1110F	390-1110	0.61—0.59
Chromium	100-1000	0.08—0.26
Copper		
Electrolytic, polished	176	0.02
Comm'l plate, polished	66	0.030
Heated at 1110F	390-1110	0.57—0.57
Thick oxide coating	77	0.78
Cuprous oxide	1470-2010	0.66—0.54
Molten copper	1970-2330	0.16—0.13
Dow metal, cleaned, heated	450-750	0.24—0.20
Gold, highly polished	440-1160	0.02—0.40
Iron and steel		
Pure Fe, polished	350-1800	0.05—0.37
Wrought iron, polished	100-480	0.28
Smooth sheet iron	1650-1900	0.55—0.60
Rusted plate	67	0.69
Smooth oxidized iron	260-980	0.78—0.82
Strongly oxidized	100-480	0.95
Molten iron and steel	2730-3220	0.40—0.45
Lead		
99.96%, unoxidized	260-440	0.06—0.08
Gray oxidized	75	0.28
Oxidized at 390F	390	0.63
Mercury, pure clean	32-212	0.09—0.12
Molybdenum filament	1340-4700	0.10—0.29
Monel metal K5700		
Washed abrasive soap	75	0.17
Repeated heating	450-1610	0.46—0.65
Nickel and alloys		
Electrolytic, polished	74	0.05
Electroplated, not polished	68	0.11
Wire	368-1844	0.10—0.19
Plate, oxid. at 1110F	390-1110	0.37—0.48
Nickel oxide	1200-2290	0.59—0.86
Copper-nickel, polished	212	0.06
Nickel-silver, polished	212	0.14
Nickelin, gray oxide	70	0.26
Nichrome wire, bright	120-1830	0.65—0.79
Nichrome wire, oxid.	120-930	0.95—0.98
ACI-HW (60Ni, 12Cr) firm black ox. coat	520-1045	0.89—0.82
Platinum, polished plate	440-2960	0.05—0.17
Silver, pure polished	440-1160	0.02—0.03
Stainless steels		
Type 316, cleaned	75	0.28
316, repeated heating	450-1600	0.57—0.6
304, 42 hr at 980F	420-980	0.62—0.73
310, furnace service	420-980	0.90—0.97
Allegheny #4, polished	212	0.13
Tantalum filament	2420-5430	0.194—0.33
Thorium oxide	530-1520	0.58—0.21
Tin, bright	76	0.04—0.06
Tungsten, aged filament	80-6000	0.03—0.35
Zinc, 99.1%, comm'l, polished	440-620	0.05
Galv., iron, bright	82	0.23
Galv., gray oxid.	75	0.28

*When two temperatures and two emissivities are given they correspond, first to first and second to second, and linear interpolation is permissible.

c) The emissivity of any metal varies widely with the condition of its surface.

The absorptivity of a surface depends on the surface condition as well as the temperature of the emitter. When a surface is exposed to solar radiation, its absorptivity can vary appreciably from its emissivity due to the extreme temperature difference between the temperature of the sun (approximately 10,000°F black body temperature) and the radiated surface. Values of solar absorptivity and ratios of a/ϵ for several materials are shown in Table 2.2.3.3b. A common radiation problem involves the determination of net total heat transferred between surfaces separated by a non-absorbing medium. For most practical applications, air and other common gases at atmospheric pressure can be considered non-absorbing media. For such cases the following expression may be used:

$$q = \sigma F_e F_A A \left(T^4_{abs_1} - T^4_{abs_2}\right) \quad \text{(Eq 2.2.3.3f)}$$

where

q = net radiant heat transfer rate, Btu/hr

σ = Stefan-Boltzmann constant = 0.1713×10^{-8} Btu/(hr) (ft^2) (°R)^4

F_e = function of the emissivities of the two surfaces

F_A = function of the configuration of the two surfaces

A = area factor depending on configuration, ft^2

T_{abs_1} and T_{abs_2} = respective absolute temperatures of surfaces 1 and 2, °R

Values of F_e, F_A, and area A for various common configurations are available in standard heat transfer references (Reference 130-1, pages 62 and 64).

2.2.3.4 RADIATION FROM FLAMES AND GASES. Radiation from flames and gases is an important consideration in component design for ballistic missiles and space boosters. At high altitude flight, rocket engine exhaust gases can expand such that the flow reverses, causing flaming gases to envelop the engine system. Propellant feed system components are thereby directly exposed to high temperature radiant heat transfer. A technique used to prevent base recirculation (the reverse exhaust flow phenomenon) is to locate a scoop at the upper end of the engine compartment so that air flowing along the missile skin is ducted into the compartment. The scoop, however, does not eliminate heat transfer problems, since the boundary layer gas is aerodynamically heated by skin friction to temperatures as high as 2000°F. A technique used in missiles for protecting heat-sensitive components from thermal damage is the wrapping of exposed surfaces with reflective aluminum tape.

For detailed information on the theory of flame and gas radiation see References 130-1, 132-1, 134-1, and 325-1.

2.3 HEAT TRANSFER CONSIDERATIONS IN FLUID COMPONENT DESIGN

The following paragraphs describe some practical considerations for promoting or reducing heat transfer in fluid component design.

2.3.1 Design Techniques to Promote Heat Transfer

2.3.1.1 INCREASE CONDUCTION. Two techniques for increasing conduction are the use of materials having good thermal conductivity and the design of joints in the conduction path which have good physical contact.

Use Good Conductors. One of the simplest ways of increasing heat transfer involves the selection of materials

Table 2.2.3.3b. Solar Absorptivities and a/ϵ Ratios

(Reference 473-1)

MATERIALS	SOLAR ABSORPTIVITY, a	RATIO a/ϵ	EMISSIVITY, ϵ	TEMPERATURE, °F
2024 Aluminum As received	0.242 — 0.49	2.69 — 4.45	0.09 — 0.11	0 — 300
2024 Aluminum Polished	0.302 — 0.25	3.36 — 2.50	0.09 — 0.10	0 — 300
301 Stainless Steel As received	0.44	2.09	0.21	0
301 Stainless Steel Polished	0.38	2.375	0.16	0
Zinc, Polished	0.55	25.0	0.022	30
Silver	0.07	3.5	0.02	80
Epoxy Paint	0.25	0.28	0.90	30

which have good thermal conductivity. It is often necessary that fluid components be designed to transfer heat from the fluid media to the external environment (dissipation of heat in hydraulic systems), or to transfer heat from one fluid to another (heat exchangers). Rapid conduction of heat can be an important consideration in fluid components which contain elastomeric seals or other materials subject to damage by excessive heat. Metals which possess relatively good thermal conductivity are copper, aluminum, brass, gold, silver, molybdenum, and beryllium. As a general rule, a material which is a good electrical conductor is also a good thermal conductor. The following general statements can be made concerning the use of materials where it is desired to improve heat transfer by conduction (Reference 130-1):

a) The conductivities of all homogeneous, solid materials are relatively high compared to the conductivity of porous, cellular, fibrous, or laminated materials.

b) The conductivity of a material increases with its density.

c) The conductivity of solid materials increases with increasing temperature.

Where weight is an important consideration, the use of alkali metal boiling and condensation as a technique for increasing high rate heat transfer is feasible. Two of the most common alkali metals used for this purpose are potassium and sodium. Several research studies (References 2-24 and 2-25) have recently been conducted to determine the best methods for handling and using these metals.

The light weight and low melting point of potassium and sodium combined with relatively good thermal conductivity are the desirable properties for heat transfer applications. One well known use of sodium for heat transfer purposes is in the sodium-filled aircraft engine valve poppet. This poppet, unlike conventional auto and aircraft engine poppets, has a cavity in the head and stem which is filled with sodium. The sodium serves as a high-rate heat conductor, transfering heat more effectively from the head of the poppet into the stem where it can be dissipated into the stem guide and cylinder head. This construction results in prolonged poppet life, minimizing wear and burning by reducing the operating temperature of the thin sections.

Use Good Physical Contact. A second important way to improve heat transfer by conduction is to assure that good physical contact is maintained between the elements in the conduction path. Some considerations in designing fluid components where good conduction is important are:

a) Maintain good metal-to-metal contact at joints, avoiding non-metallic and insulating materials in the conduction path wherever possible.

b) Specify in detail good surface finish and flatness of mating surfaces. The exclusion of as much air as possible between mating parts will assure less joint resistance to heat transfer.

c) Make the surface areas of joints as large as possible consistent with weight limitations and good design. Generally speaking, if the surface area of the joint between two mating parts is larger than the cross-sectional area of the parts adjacent to the joint, impairment of heat transfer across the joint will be kept to a minimum.

d) Clamping or mating forces between two adjacent parts should be a maximum value consistent with the design. Heavy clamping forces will often overcome surface irregularities which might otherwise result in a barrier to good heat conduction.

e) When it is desirable to conduct heat from reciprocating and rotating elements in fluid components, heat transfer can be improved by the use of plain or journal bearings rather than ball and roller bearings, since the area of contact will be greater. In certain cases the use of dry lubricated bearings should be considered, because the conductivity of many oils and organic lubricants is relatively low.

2.3.1.2 INCREASE CONVECTION. Heat transfer by convection is probably the most important single method for the removal of heat as far as fluid component design for aircraft or ground equipment is concerned. Several ways for improving the convection process are given in the following paragraphs.

Use Flow Turbulators. Flow turbulators (fins or baffles) may be placed within a fluid component in the path of fluid flow to cause turbulence or agitation of the fluid as it passes through the component, thereby increasing heat transfer. In addition to increasing the heat transfer coefficient through increased Reynolds number (turbulence), turbulators also increase conductive heat transfer by increasing the surface area over a given length of flow path. There is a practical limit to the addition of turbulators, dictated by the allowable pressure drop of the line or component and by heat transfer limitations of the outside of the system (Reference 59-9, page 94).

Promote Natural Convection. In aircraft and ground system hot gas and cryogenic components, natural convection can be utilized for promoting heat transfer. Since a gravitational field is required to establish natural convection currents, natural convection does not exist under conditions of zero gravity, thus eliminating this mode of heat transfer for spacecraft systems (References 2-21 and 2-23). Where natural convection is possible, the following practices should be adhered to:

a) Provide as much free space around hot components as possible.

b) Use cooling fins freely whenever possible, making certain that surfaces are smooth enough to avoid the accumulation of grease and dirt which insulate and, therefore, reduce heat transfer.

c) Arrange hot components in series to form a horizontal grouping of minimum height, thus reducing natural convection of heat from one component to the other.

d) Stagger components or parts when used in vertical arrangement, to prevent heating of adjacent parts by natural convection.

e) Isolate temperature-sensitive components, or place them below sources of heat.

f) Use louvers, vents, and ducting whenever possible to promote natural drafts.

An example of a fluid component which has been designed for cooling by natural convection, as well as conduction and radiation, is illustrated in Figure 2.3.1.2. In this component, heat is transferred from the hot valve body outward to the cooling fins by conduction. The air between the fin surfaces, heated by conduction and radiation, passes outward and upward across the valve by natural convection.

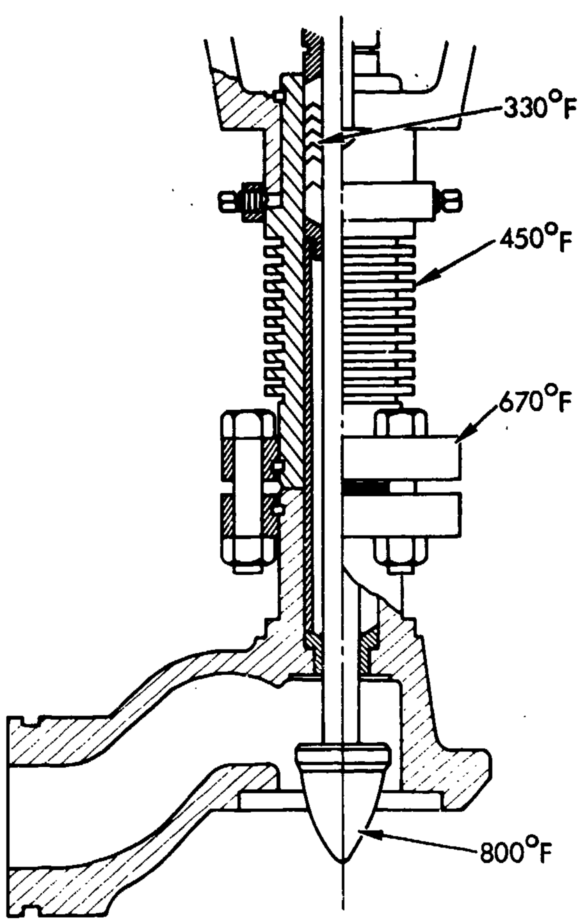

Figure 2.3.1.2 Hot Gas Valve

(Courtesy of Annin Company, Montebello, California.)

As the heated air rises, cool air moves in to take its place and the process continues. This valve design can also be used for handling cryogenic fluids when it is desirable to protect the valve stem packing and actuator from the detrimental effects of cryogenic temperatures. In this case the fins promote heat transfer to, rather than away from the components.

Use Forced Convection. Forced convection of a cooling fluid over the external surfaces of a component provides a practical means of cooling for aircraft and ground systems. By using pumps and blowers, the designer is afforded considerable latitude in the control of heat transfer by merely sizing the forcing unit to obtain, within limits, the flow rate required to maintain satisfactory operating temperatures.

Some factors which should be kept in mind when designing equipment to be cooled by forced convection are:

a) Air cooling is preferred to liquid cooling, since it usually results in a simpler, less expensive design.

b) Where large quantities of air are to be forced over hot surfaces to effect cooling, a manifold or duct system with dust filters should be provided. The size of the intake filter should be selected to provide reasonable service life with a low pressure drop. Exhaust filters should always be two or three times larger than intake filters to minimize pressure drop. Unfiltered leaks or openings should be eliminated from the flow system whenever possible.

c) Fans, pumps, and blowers should be designed to insure a velocity sufficient to maintain turbulent flow over the heat dissipating surfaces.

d) Cooling is increased by providing fins over which the cooling fluid is directed. Fins should be integral with the part to be cooled or in perfect metal-to-metal contact with it.

e) Safety switches should be included to shut down the system, if convection power fails.

f) The fluid intake port of the convection system should be isolated from the exhaust port to eliminate heat transfer between the influent and effluent streams.

g) In air-cooling systems, ducting should be designed so as to utilize the effects of induced natural drafts.

2.3.1.3 INCREASE RADIATION. A third method of improving heat transfer in fluid components, especially those components in a space environment, is to improve the radiating ability. Equation (2.2.3.3f) indicates that heat transfer by radiation can be increased by raising the emissivity of the component in question and by increasing component surface area. The removal of external radiation barriers is also important in promoting radiation heat transfer.

Raising Emissivity. Two simple methods of raising the emissivity of a component are:

a) Blacken or darken radiating surfaces.

b) Increase roughness of radiating surfaces.

Techniques often used to darken surfaces include painting, anodizing, and parkerizing. Surfaces can be roughened by grinding, brushing, shot peening, sand blasting, coating with metallic oxides, etc., without materially affecting the strength or function of the component. Neglecting to finish machine certain surfaces which will operate hot reduces the cost of manufacturing the parts and eliminates the necessity for surface roughening operations on finished parts. Using dark paints or organic coatings on components which operate at high temperatures to increase their emissivity may be impractical because of the temperature limitations of such coatings. For such items metallic oxide coatings serve best. More detailed information can be found in Reference 2-22.

Remove Radiation Barriers. Another method of increasing radiation from a component is to remove radiation barriers which may be present, assuring that the surrounding atmosphere or environment which is the sink for radiation reception has as high an absorptivity coefficient as possible. Reflective surfaces such as white or polished metal should not be placed in close promimity to hot components.

2.3.2 Design Techniques to Reduce Heat Transfer

2.3.2.1 REDUCE CONDUCTION. Often in the design of fluid components such as cryogenic valves and regulators, the designer is faced with the problems of (1) reducing the transfer of heat from the surroundings to the component in order to prevent excessive loss of the cryogenic fluid by evaporation; and (2) reducing heat transfer between temperature-sensitive parts in cryogenic fluids. Reduction of heat transfer is also important in systems designed to handle hot gases or liquids where it is important to keep the fluid from losing heat. A common technique for thermally isolating temperature-sensitive parts such as actuators, from the valve body containing the cold or hot fluid is to place a thermal barrier, such as a low conductivity non-metal (for example, Teflon), between the valve body and the actuator.

Metals which have relatively low conductivity and are often used in cryogenic components are nickel alloys and stainless steels. Other less frequently used metals with low conductivities are antimony, bismuth, mercury, lead, indium, platium, and tantalum. However, conductivity alone is often not enough to cut heat losses. Other techniques which must be used are discussed in the following paragraphs.

Conduction Path Geometry. When the designer is not free to choose materials with low conductivities, it is necessary to employ other techniques to reduce heat transfer, such as breaking the heat conduction path completely. This may be accomplished by separating the hot and cold parts by a dead air space, vacuum, or other insulators. A design which employs this technique is illustrated in Figure 2.3.2.1a.

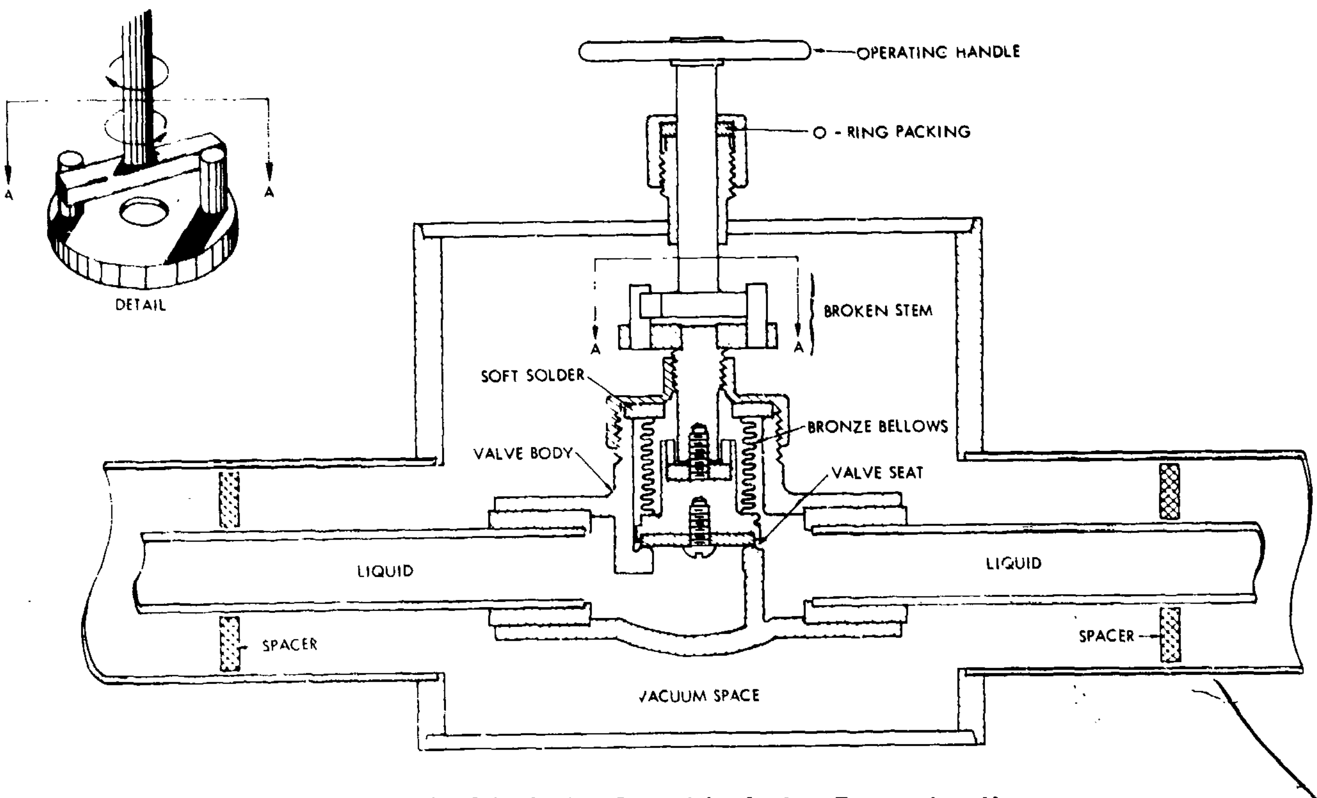

Figure 2.3.2.1a Broken Stem Valve for Low Temperature Use
(Courtesy of NBS Cryogenic Engineering Laboratory, Boulder, Colorado.)

In this device, called a broken stem valve, the valve actuator may be isolated from the valve body by a slight back turn of the stem. The entire valve is then enclosed in an evacuated chamber which further reduces heat loss.

Another technique for reducing heat transfer by separation of critical parts is the use of extended stems and bonnets as shown in Figures 2.3.1.2 and 2.3.2.1b. Figure 2.3.1.2 shows the effect of a long conduction path between the 800°F poppet and the temperature-sensitive packing. Further reduction in heat transfer to the packing is accomplished by using a low conductivity stainless steel bonnet and stem and providing integral fins to direct heat away from the packing to the surrounding air. Radiation transfer to the air is increased by coating the fins black. The hollow portion of the bonnet surrounding the stem will become filled with condensation of steam (or other condensable vapor from the transported fluid) to further reduce heat transfer to the packings.

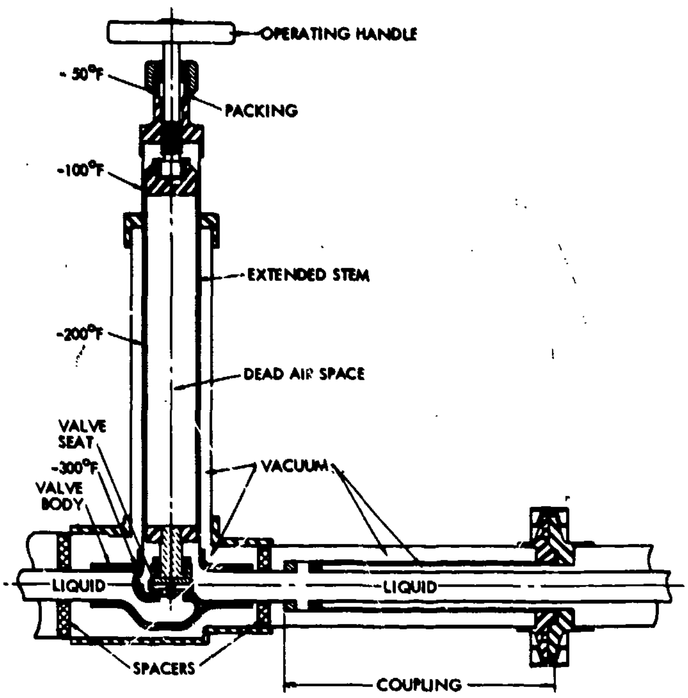

Figure 2.3.2.1b. Extended Stem Valve for Low Temperature Use
(Courtesy of NBS Cryogenic Engineering Laboratory, Boulder, Colorado.)

In Figure 2.3.2.1b the cryogenic valve stem is a torque tube design. The small cross-sectional area of the stem partially isolates the low temperature-sensitive valve packing from the cryogenic fluid temperature. The stem is surrounded by a vacuum to isolate the packing from the cryogenic fluid and to reduce heat loss from the surrounding air to the fluid.

Insulation for Reducing Heat Transfer. Insulation is used in high temperature systems to prevent heat loss to the environment from the hot fluid. In cryogenic systems it is used to restrict heat transfer from the atmosphere to the fluid, and to protect sensitive parts from the low temperature. Cryogenic components require protection by thermal insulation to a degree unapproached in many other fields. The choice of insulation for any particular component must be a compromise between such variables as thermal conductivity, reliability, cost, replaceability, weight, compatibility, and size.

Foamed insulation materials are relatively inexpensive and are satisfactory for many cryogenic applications. Foamed insulation has a cellular structure formed by the addition or the evolution of gas during manufacture. When the cells of such a material are sealed from one another, the material is an effective insulation for extreme temperature application. Foams of this type have been made from polystyrene, polyurethane, rubber, silica, and glass. Since gas can permeate such insulation only by diffusion through the cell walls, the material behaves as though it were completely impermeable when exposed to a foreign gas for limited time periods. Also, if the temperature of the cold side of the foam is so low that the gas in the nearby cells is condensed and has negligible vapor pressure, the thermal conductivity is reduced by eliminating gaseous conduction in these regions.

It is possible to have foamed insulation bonded to the surfaces of the insulation cavity. This type of insulation, usually termed "*foamed in place,*" is quite adaptable to valves and other components having irregular surfaces. Further information on foamed insulation is given in References 20-5 and 213-1, pages 163-165.

Another type of cryogenic insulation which has merit is composed of low density materials such as powders and fibers, with gas at atmosphere pressure in the interstitial spaces. These materials have been used successfully to insulate components at liquid nitrogen temperatures. In this type of insulation the volume of gas in the void may be 10 to 100 times the volume of the solid material used (Reference 29-5). Common filler materials used for insulation are fiberglass, asbestos, perlite (a volcanic glass), expanded SiO_2, calcium silicate, and diatomaceous earth. In addition to being low thermal conductivity materials, powders and fibers provide effective barriers to heat transfer by reducing the contact area between individual components of the insulation and by providing multiple radiation barriers. For further information on powder and fiber insulations see References 213-1, and 326-1.

A vacuum is a widely used and very effective insulation for cryogenic systems. This insulation is formed by constructing a gas-tight closure around the component, then evacuating it by means of a vacuum pump, and positively sealing it. Figure 2.3.2.1c shows the change in conductivity between two separated conductors as a function of the gas pressure between the surfaces. From this figure it is seen that an effective vacuum insulation is achieved between 10^{-2} and

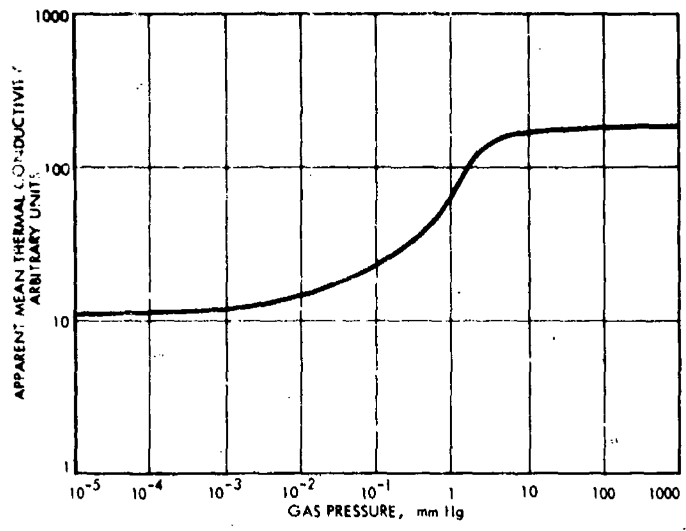

Figure 2.3.2.1c. Change in Apparent Thermal Conductivity with Gas Pressure

10^{-4} mm Hg. The knee in the curve occurs when the gas pressure is reduced to a point where the mean-free path of the gas molecules is large compared to the space between conductors. To aid in maintaining a vacuum over long time periods, a small quantity of gas absorbent such as activated charcoal can be introduced into the vacuum space. A disadvantage of vacuum insulation is the fact that a gas leak can occur without any warning and without making itself readily evident. Consequently, when effective insulation is imperative to the operation of the component, it is best to provide some sort of vacuum indicator which can easily be monitored to ascertain the effectiveness of the insulation. In a cryogenic system, loss of insulation will result in the formation of frost on the exterior of components. By eliminating conduction and convection, vacuum insulation essentially reduces heat transfer to radiation only.

An improvement over powder and vacuum insulations is a combination of both, where a cavity containing powdered insulation material is evacuated to eliminate gas conduction. The powder, as previously stated, provides a radiation barrier (References 326-1 and 213-1).

The best insulation systems available, called super insulations, consist of multi-layer radiation barriers in a vacuum. The multi-layer material is composed of thin layers of insulating material such as fiberglas alternating with thin layers of a radiation barrier such as aluminum foil. Where radiation shields are in series, as is the case with powdered and multi-layer reflective insulations, the heat transfer rate by radiation (from Equation (2.2.3.3f) becomes

(Eq 2.3.2.1)

$$q = \sigma \frac{F_e F_A A}{n + 1} \left(T^4_{abs_1} - T^4_{abs_2} \right)$$

where n = the number of radiation shields

Although it is the most effective insulation material available, multi-layer insulation is also the most expensive, costing 10 to 20 times as much as powdered insulation. Comparative conductivity values for various insulation materials having a cold side temperature of −300°F and a warm side temperature of 70°F are shown in Table 2.3.2.1.

2.3.2.2 REDUCE CONVECTION. One method of reducing heat transfer by convection is through the use of insulation, as discussed under Detailed Topic 2.3.2.1. In the case of convection, the insulator serves primarily to isolate the component from the convecting medium by a physical barrier.

A dead air space formed by a double wall provides an effective insulation for reducing convection. If the air trapped within the walls is maintained at nearly constant temperature, convection currents are negligible. Heat transfer through such a wall is by conduction through the supports, by radiation and through the trapped air. To make dead air space insulation effective, thickness should be kept very small compared to the length in any direction.

If for economic reasons or from a weight standpoint double walls cannot be provided to form dead air spaces, the next best type of protection against heat transfer by convection is the use of baffles or barriers to prevent the formation of convection currents. A familiar example of this technique is the use of thermostatically controlled movable shutters in front of cooling system radiators, which can be closed to prevent the passage of air through the radiator during warm up.

2.3.2.3 REDUCE RADIATION. Equation (2.2.3.3f), an expression for heat transfer by radiation, indicates that reduction of the emissivity or the temperature of a radiating body will reduce the heat transfer rate. Reduction of radiation requires that the opposite of the technique recommended in Detailed Topic 2.3.1.3 be used for increasing the radiating ability of a surface. Because it is often impossible or impractical to alter operating temperatures of a component, it is necessary to concentrate on reducing the emissivity (or absorptivity) of components to lower heat transfer rates. This can be most easily accomplished by altering surface color and finish. Components which are, and should remain, colder than their surroundings may be finished in light colors or enclosed in containers with silvered surfaces. Polishing of dark or rough surfaces will increase reflectivity and thereby reduce absorbed heat.

Another simple technique involves the use of radiation barriers, a common example of which is a sun shade to protect against direct solar radiation. In this connection insulation material used to reduce heat transfer by convection and conduction will also serve as a radiation barrier and should be selected with this purpose in mind.

2.3.2.4 USE OF ABLATIVE MATERIAL TO REDUCE HEAT TRANSFER. Although not widely used at this time for fluid components, ablative materials can be useful for

Table 2.3.2.1. Apparent Mean Thermal Conductivity, k, for Various Insulations (Cold Temperature = −300°F, Warm Temperature = 70°F, Pressure <10^{-5} mm Hg unless otherwise noted)

(References 213-1 and 28-22)

INSULATIONS	REMARKS	DENSITY (lb_m/ft^3)	THERMAL CONDUCTIVITY* Btu/(hr) (ft^2) (°F/ft)
Foams			
Polystrene		2.5	185×10^{-4}
Polyurethane	at 1 atm	5-9	191×10^{-4}
	at 10^{-3} mm Hg		69.5×10^{-4}
Rubber	at 1 atm	5	208×10^{-4}
Evacuated Powders			
Silica aerogel		5	12.2×10^{-4}
Perlite	$\geq$80 mesh	8.7	5.8×10^{-4}
Perlite	30 to 80 mesh	8.4	7.2×10^{-4}
Perlite	$\leq$30 mesh	66	5.8×10^{-4}
Diatomaceous earth		20	6.4×10^{-4}
Lampblack		12	7.3×10^{-4}
Phenolic spheres		12	7.5×10^{-4}
Talc		75	9.3×10^{-4}
Fused alumina		125	10.4×10^{-4}
Laminar alumina		4.5	13.3×10^{-4}
Multiple Layer			
Glass fiber paper — aluminum foil		4-12	0.23×10^{-4} — 0.75×10^{-4}
Glass fiber mat — aluminum foil	26 shields	7	0.41×10^{-4}
Nylon net, aluminum foil	24 shields	5.6	1.3×10^{-4}
Aluminized mylar	24 layers	2.8	0.49×10^{-4}
	47 layers	5.6	1.04×10^{-4}
Linde CS-5		11	2.3×10^{-4}
Linde SI-12		2.5	1.2×10^{-4}
Linde SI-44		4.7	0.23×10^{-4}

*These values are generally accurate to ±5%.

Table 2.3.2.4. Relative Thermal Durability of Materials

(Adapted from "International Science and Technology," I. Gruntfest and L. Shenker, no. 19, July 1963, Copyright 1963 by the Conover-Mast Publications, Inc.)

MATERIAL	RELATIVE WEIGHT LOSS
Graphite	1.0
Phenolic and Nylon fibers	1.48
Silicon carbide	2.1 — 7.8
Phenolic and silica fibers	2.7
Phenolic and glass fibers	2.7*
Silica	2.9
Alumina	8.5 — 17.0
Aluminum silicate	10.0
Zirconia	16.0

*Depends on purity of glass. Impurities reduce thermal resistance.

reducing heat transfer in high temperature flow systems. An ablative material may be defined as a substance which absorbs heat energy by mass removal through erosion, vaporization, sublimation, and thermal decomposition. Typical among such materials are silica, reinforced phenolic plastics, phenolic nylon, graphite cloth phenolics, and Teflon. Originally such materials were developed for heat protection of re-entry vehicles, but they are now used in many high temperature applications such as lining rocket engine thrust chambers and nozzles, solid propellant cases, and gas generators. These materials are relatively light in weight, simple to fabricate, low in cost, and capable of simple quality control. Because ablation is a material removal process, this technique of heat transfer control is limited to applications where operational times are short, such as in space boosters and re-entry vehicles (Reference 332-1). Table 2.3.2.4 compares the thermal durability of three phenolic reinforced ablative materials with several other high temperature materials using the weight loss of graphite as the basis of comparison.

REFERENCES

2-21	29-5	134-1
2-22	59-9	213-1
2-23	64-4	324-1
2-24	82-3	325-1
2-25	82-5	326-1
2-31	82-8	332-1
19-182	130-1	411-1
20-5	131-1	463-1
23-38	132-1	473-1
28-22	133-1	V-8

SYMBOLS, UNITS, AND DIMENSIONS

SYMBOL	QUANTITY	UNIT	DIMENSION
a	Absorptivity	dimensionless	
A	Cross-sectional area	ft^2	L^2
c_p	Specific heat at constant pressure	Btu/[illegible] °F	FL/MT
c_v	Specific heat at constant volume	Btu/lb_m °F	FL/MT
D	Diameter	ft, in	L
E	Radiant energy	Btu/hr ft^2	FL/tL^2
E_b	Black body radiant energy	Btu/hr ft^2	FL/tL^2
F_A	Configuration factor	dimensionless	
F_e	Emissivity factor	dimensionless	
G	Temperature gradient	°F/ft	T/L
h	Surface or film coefficient of heat transfer	Btu/(hr)(ft^2)(°F)	F/tLT
k	Thermal conductivity	Btu/(hr)(ft^2)(°F/ft)	F/tT
q	Quantity of heat transferred	Btu/hr	FL/t
R	Radius	ft	L
t	Time	sec	t
T	Temperature	°F	T
T_{abs}	Temperature, absolute	°R	T
U	Transmittance of overall coefficient of heat transfer	Btu/(hr)(ft^2)(°F)	F/tLT
V	Velocity	ft/sec	L/t
μ_m	Absolute viscosity	lb_m/ft-hr	M/Lt
α	Thermal diffusivity	ft^2/hr	L^2/t
ϵ	Emissivity of a radiating surface	dimensionless	
ρ	Density	lb_m/ft^3	M/L^3
σ	Stefan-Boltzmann constant of proportionality = 0.1713×10^{-8}	Btu/(hr)(ft^2)(°R)^4	F/LtT^4

SUBSCRIPTS

1	Upstream (with respect to heat flow)
2	Downstream (with respect to heat flow)
c	Conduction and convection
r	Radiation
w	Wall
B	Bulk fluid
[illegible]	Initial
F	Final

ISSUED: MAY 1964

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ILLUSTRATIONS

ILLUSTRATIONS (Continued)

TABLES

3.1 INTRODUCTION

The Fluid Mechanics Section presents a review of the fundamentals of fluid mechanics plus a compilation of data and formulae which are used in the design, selection, and evaluation of fluid components. Since the topic of fluid mechanics and its close association with thermodynamics is a very extensive and far-reaching subject, not all areas of interest can be covered here. It is the intent of this section to present fundamental theory and equations and to refer the reader to selected references which treat the subject in greater detail. A more important part of this section is the compilation of data and valve coefficients which seldom find their way into specialized texts. For the most part, liquids are considered incompressible fluids, while gases are considered ideal or perfect fluids. Two-phase flow is limited to gas-liquid mixtures.

3.2 NOTATIONS AND DEFINITIONS

3.2.1 Units, Dimensions, and Symbols

The system of units used throughout this section is the unit force-mass system, which provides a compromise between the absolute and gravitational systems. A comparative list of the fundamental units and a few derived quantities for the three systems are shown in Table 3.2.1. A complete list of derived quantities used in Section 3.0 with their symbols, units, and dimensions is presented on a foldout sheet at the end of the section. Conversion tables for the primary quantities are given in Appendix A.

The unit force-mass system has five basic dimensions: length (L), mass (M), force (F), time (t), and temperature (T). The basic units assigned to the dimensions are as follows:

3.2.1.1 LENGTH (L). For most engineering purposes, the basic unit of length in the English system is the foot, in the metric system, the meter.

3.2.1.2 MASS (M). In the unit force-mass system, the unit of mass is the pound-mass (lb_m), in the metric system, the kilogram.

3.2.1.3 FORCE (F). The unit force in the unit force-mass system is defined as the gravitational force on a unit mass at a selected standard location. The selected standard location is taken on the earth's surface where the acceleration of gravity = 32.1740 ft/sec². The value is often rounded off to 32.2 ft/sec². If the mass is assumed constant, Newton's Second Law of Motion is expressed

$$F = \frac{1}{g_c} ma \qquad \textbf{(Eq 3.2.1.3)}$$

where F = force, lb_f
m = mass, lb_m
a = acceleration, ft/sec²
g_c = constant of proportionality
$= 32.2 \frac{lb_m}{lb_f}$ ft/sec²

At the standard location ($a = g$), the weight (gravitational force) of a body using the unit force-mass system will have the same numerical value as its mass. Thus, $g/g_c = 1\ lb_f/lb_m$.

In the English system of units, force is expressed in pounds and is designated in pound-force units, lb_f. In the metric

Table 3.2.1 Systems of Units

UNIT FORCE-MASS SYSTEM UNITS			ABSOLUTE UNITS			GRAVITATIONAL UNITS		
Quantity	English (ft-lb_m-lb_f-sec)	Metric (m-kg_f-kg-sec)	Quantity	English (ft-lb_m-sec)	Metric (cm-gm-sec)	Quantity	English (ft-lb_f-sec)	Metric (cm-gm-sec)
Fundamental			Fundamental			Fundamental		
Length	ft	m	Length	ft	cm	Length	ft	cm
Mass	lb_m	kg	Mass	lb_m	gm			
Force	lb_f	N(Newton)				Force	lb_f	gm
Time	sec	sec	Time	sec	sec	Time	sec	sec
Temperature	°F, °R	°C, °K	Temperature	°F, °R	°C, °K	Temperature	°F, °R	°C, °K
Derived			Derived			Derived		
Area	ft²	m²	Area	ft²	cm²	Area	ft²	cm²
Volume	ft³	m³	Volume	ft³	cm³	Volume	ft³	cm³
Speed	ft/sec	m/sec	Speed	ft/sec	cm/sec	Speed	ft/sec	cm/sec
Acceleration	ft/sec²	m/sec²	Acceleration	ft/sec²	cm/sec²	Acceleration	ft/sec²	cm/sec²
			Force	poundal	dyne			
						Mass	slug	no name
Energy	ft-lb_f	m-kg_f	Energy	ft-poundal	erg	Energy	ft-lb_f	cm-gm
Pressure	lb_f/ft²	N/m²	Pressure	poundal/ft²	dyne/cm²	Pressure	lb_f/ft²	gm/cm²
Power	ft-lb_f/sec	m-kg_f/sec	Power	ft/poundal/sec	erg/sec	Power	ft-lb_f/sec	cm/gm/sec

system, the basic unit of force is the kilogram.

3.2.1.4 **TIME** (t). The basic unit of time in both the English and metric systems is the second.

3.2.1.5 **TEMPERATURE** (T). Temperature may be classed as either relative or absolute. *Relative* temperatures are subdivisions based on the interval between the freezing and boiling points of water. The two scales used are Centigrade and Fahrenheit.

a) The *Centigrade* scale divides the temperature interval between the freezing and boiling points of water into 100 parts. On this scale, the ice point is 0°C and steam point is 100°C.

b) The *Fahrenheit* scale divides the temperature interval between the freezing and boiling points of water into 180 equal parts. The ice point and steam point are 32°F and 212°F, respectively.

The Centigrade and Fahrenheit scales are related by the equation

$$°F = \frac{9}{5}°C + 32 \qquad \text{(Eq 3.2.1.5)}$$

Absolute temperatures are temperatures referred to a zero degree datum (reference) plane. The two scales used are Kelvin and Rankine.

a) On the *Kelvin* scale the lowest temperature attainable is 0°K, which is defined as 273.16° below the freezing point of water. Thus, 0°K = −273.16°C.

b) On the *Rankine* scale the lowest temperature attainable is 0°R, which is defined as 491.69° below the freezing point of water (32°F); or 459.69 below 0°F. Thus, 0°R = −459.69°F.

In the English system, degrees Fahrenheit and Rankine are used extensively; in the metric system, degrees Centigrade and Kelvin are more commonly found.

3.2.2 Definitions and Simplifying Assumptions

3.2.2.1 **FLUID DEFINITIONS.** The terms fluid, gas, liquid, and vapor are defined as follows:

a) A *fluid* is a substance which will deform continuously while being subjected to shear stress. The term fluid includes both gases and liquids, but the distinction between a gas and a liquid is not always easy to make, especially near the so-called critical point.

b) A *gas* may be readily compressed. With no boundary restraints, it expands somewhat indefinitely as a result of molecular energy. When bounded, it tends to completely fill its container.

c) A *liquid* is a considerably less compressible fluid than a gas. Its constituent molecules are closely held together by inter-molecular attraction. Even at zero gravity, liquids usually have a definite surface from which molecules escape and re-enter.

d) A *vapor* is a gas which may begin to liquify (or solidify) as a result of a small pressure increase or temperature decrease.

It is common to find both liquids and gases flowing simultaneously as, for example, during cavitation or the transport of a cryogenic material and its vapor. This is called *two-phase flow*. Two-phase flow may also apply to the transport of solids suspended in a liquid matrix. For example, the conveyance of minute, suspended, metallic particles in thixotropic propellants is typical of slurry flow. In this section, two-phase flow is limited to gas-liquid mixtures.

3.2.2.2 **SIMPLIFYING ASSUMPTIONS FOR FLUIDS.** For the purpose of simplifying analyses, simple properties or characteristics are sometimes ascribed to fluids. For example, when friction-dependent properties (e.g., viscosity) are neglected, the fluid may be considered as *ideal*. When such properties cannot be neglected, an assumption is made that the ratio of shear stress to rate of angular displacement is a constant. This fluid is considered *Newtonian*. Similarly, when density changes can be neglected, the fluid may be regarded as incompressible. Density changes for liquids are usually small, hence liquids are often considered *incompressible*. For gases, density changes are significant and normally cannot be neglected. Gases are normally considered *compressible* fluids.

3.2.2.3 **PROCESS DEFINITIONS.** The term process and types of processes are defined as follows:

A *process* is an event in which energy is transformed or redistributed within a system. Processes may be classified as flow or non-flow, reversible or irreversible, steady or non-steady.

a) A *non-flow process* is one in which the fluid does not flow in or out of a container but may be acted upon while enclosed in the container. An example of a non-flow process is the compression and expansion of a gas in a piston-cylinder combination.

b) A *steady-flow process* is one in which the fluid moves continuously through a region without conditions (velocity, pressure, density, etc.) changing with time. Flows through pipe lines, valves, and nozzles are closely approximated by a steady-flow process.

c) A *non-steady flow process* is one in which system properties and conditions vary with time. A typical example is the discharge of a gas (blowdown) from a constant volume tank.

d) A *reversible process* is one which, after completion, may be returned to its initial state by reverse order of the process. All other systems associated with it may be returned to their initial state, and all energy transformed may be returned to its original form, position, and amount.

e) An *irreversible process* does not satisfy the above criteria. Some factors influencing irreversibility are fluid friction, turbulence, inelastic deformation, heat transferred through a finite temperature difference, mixing of two different substances, unrestrained expansion, and combustion.

3.2.2.4 SIMPLIFYING ASSUMPTIONS FOR PROCESSES. To simplify analyses of processes, simple conditions may be assumed to occur during the process. For example, if no heat is added or extracted, the process can be considered *adiabatic*. If, in addition to being adiabatic the process is reversible (entropy change is zero), the process is *isentropic*. A process completed with a transfer of heat is called a *diabatic* process.

During flow processes, additional restrictions may be ascribed to the fluid. Two of the most common are *uniformity* and *unidirectionality*. At a given time, the velocity of each particle may be different. By uniformity, each particle is restricted to the same velocity (magnitude and direction) at a given instant. By unidirectionality, each particle has only one direction of motion.

3.3 FLUID PROPERTIES

3.3.1 Density, Specific Weight, and Specific Volume

Density, ρ, is defined as the mass per unit volume and is written dimensionally as M/L^3. In the English system of units, density can be expressed in either slugs/ft³ or lb_m/ft^3. In the metric system, density is usually expressed as grams per cubic centimeter, gm/cm^3. The units of density throughout this section are lb_m/ft^3.

The relative effect of changes in pressure and/or temperature on the density of liquids is minor as compared with gases. In most engineering problems, liquids are considered to be incompressible fluids while gases are said to be compressible. However, if pressure and temperature variations are very small, gases can be considered as incompressible fluids. Figure 3.3.1 illustrates the general influence of temperature and pressure on the density of a liquid and gas. Densities of commonly used fluids are presented in Section 12.0, "Materials."

Specific weight, w, represents the weight (force exerted by gravity) per unit volume of fluid, and has dimensions of F/L^3. The units of specific weight in the English system are lb_f/ft^3. In the unit force-mass system of units, the relationship between ρ and w is expressed

$$\rho = \frac{g_c}{g} w \qquad \textbf{(Eq 3.3.1a)}$$

where g = the acceleration of gravity, ft/sec²

$g_c = 32.2 \frac{lb_m}{lb_f}$ ft/sec²

In the use of relations involving specific weight, it is important to recognize that values of specific weight change

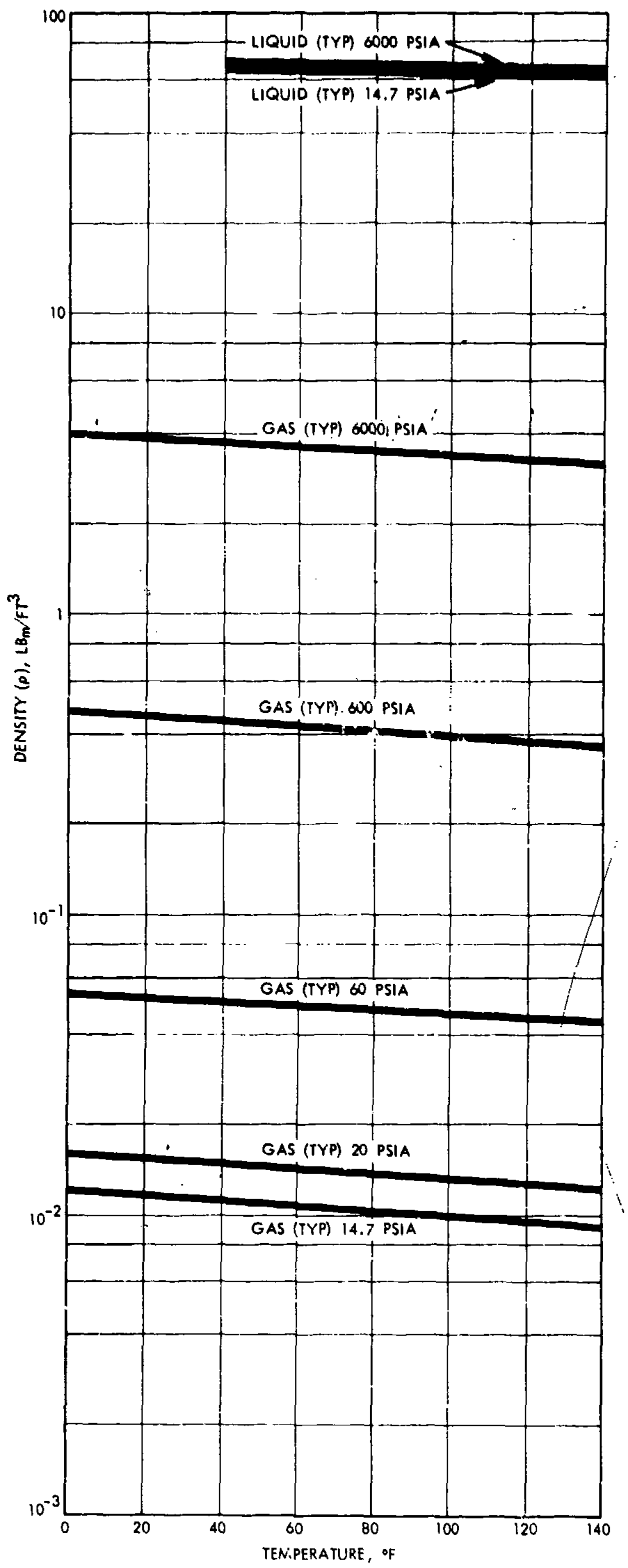

Figure 3.3.1. Effect of Pressure and Temperature on the Density of a Liquid and Gas

with variations in acceleration of gravity. Density, ρ, on the other hand, is independent of the gravitational system. The specific weight is numerically equal to the density in the unit force-mass system only if the acceleration of gravity is 32.2 ft/sec^2.

Although for most engineering calculations g can be considered to be a constant (32.1740 ft/sec^2 — international standard value at sea level and 45° latitude), the value of g is a function of distance from the center of the earth and latitude. The effect of changes in elevation on g is indicated by Table 3.3.1a.

To correct g for altitudes above-sea level, subtract 0.003 ft/sec^2 for each 1000 feet. This can be expressed

$$g = 32.1740 - 0.000003\,h \quad \textbf{(Eq 3.3.1b)}$$

where h = feet above sea level.

Changes in g with latitude are indicated by Table 3.3.1b, and can be approximated by the following relationship:

$$g = 32.08\,(1 + 0.0052 \sin^2 \phi) \quad \textbf{(Eq 3.3.1c)}$$

where ϕ = geographical latitude.

Values of g for bodies within the solar system are given in Table 3.3.1c.

Specific volume, v, is the volume occupied by a unit mass of fluid and is the reciprocal of the density, namely

$$v = \frac{1}{\rho} \quad \textbf{(Eq 3.3.1d)}$$

In the English system, units are commonly cubic feet per pound-mass (ft^3/lb$_m$). Sometimes specific volume is defined as the reciprocal of specific weight, $\frac{1}{w}$. In the unit force-mass system specific volume, based on density, is numerically equal to the specific volume based on specific weight, if the acceleration of gravity is 32.2 ft/sec^2.

3.3.2 Specific Gravity

Specific gravity, S, is a dimensionless ratio of the density or specific weight of a fluid to that of a reference fluid.

If the specific gravity of a liquid is based on water as the reference fluid, then

$$S = \frac{\rho_{liquid}}{\rho_{H_2O}} = \frac{W_{liquid}}{W_{H_2O}} \quad \textbf{(Eq 3.3.2a)}$$

Although the specific gravity of a liquid can be determined with the measured liquid at any temperature and pressure based on water at any other temperature and pressure, engineers normally standardize with both fluids at 60°F and 14.7 psia. This can be specified as $S_{60/60°F}$ where the upper number refers to the temperature of the fluid and the lower number refers to the temperature of the water. Some references, such as the *Handbook of Chemistry and Physics*, give specific gravity under conditions such as $\frac{20}{4}$ °C, $\frac{15}{4}$ °C, etc. The density of water is a maximum at 4°C.

Table 3.3.1a. Variations in "g" With Elevation

(From "Handbook of Chemistry and Physics," 38th Ed., Copyright 1956 by Chemical Rubber Publishing Co., Cleveland)

LOCATION	ELEVATION (ft)	LATITUDE	"g" (ft/sec²)
Cambridge, Mass.	46	42° 23′	32.1652
Worcester, Mass.	558	42° 16′	32.1628
Denver, Colorado	5380	39° 40′	32.1393

Table 3.3.1b Variations in "g" With Latitude

(From "Handbook of Chemistry and Physics," 38th Ed., Copyright 1956 by Chemical Rubber Publishing Co., Cleveland)

LOCATION	LATITUDE	ELEVATION (ft)	"g" (ft/sec²)
Canal Zone	9° 00′	20	32.0944
Jamaica	17° 58′	7	32.1059
Bermuda	32° 21′	7	32.1548
Cambridge	42° 23′	46	32.1652
Standard Station	45°	0	32.1740
Greenland	70° 21′		32.2353

Table 3.3.1c Gravitational Constants for the Solar System

SOLAR BODIES	"g" (ft/sec²)
Sun	897.07
Moon	5.190
Mercury	10.449
Venus	28.297
Earth	32.172
Mars	12.95
Jupiter	85.27
Saturn	37.62
Uranus	33.85
Neptune	47.61
Pluto	—

Since the density of a liquid varies with pressure and temperature, the specific gravity varies in direct proportion. However, the density of water used as the standard of comparison remains a constant.

The specific gravity of liquids is measured by a hydrometer. Special hydrometer scales are used in certain trades and industries. The most common of these are the API and Baumé scales. Degrees Baumé (units of the Baumé scale) are related to specific gravity by the following equations:

For liquids lighter than water

$$S_{60/60°F} = \frac{140}{130 + °Be'} \quad \text{(Eq 3.3.2b)}$$

For liquids heavier than water

$$S_{60/60°F} = \frac{145}{145 - °Be'} \quad \text{(Eq 3.3.2c)}$$

The American Petroleum Institute has adopted the following relation between specific gravity and degrees API:

$$S_{60/60°F} = \frac{141.5}{131.5 + °API} \quad \text{(Eq 3.3.2d)}$$

The specific gravity of a gas is a ratio of its density to that of either hydrogen or air at specified conditions of temperature and pressure. Although there are no recognized agreements as to the conditions or types of gas for a standard, air at 14.7 psia and 60°F has been used extensively.

3.3.3 Specific Heat

Specific heat is the amount of heat which must be applied to a substance of unit mass to increase its temperature one degree. Specific heats have dimensions of *FL/MT*, and in the English system have units of ft-lb$_f$/(lb$_m$)°R. In the metric system, specific heats are given as gm-cal/gm°C. The two types most often referred to are the specific heats at: (a) constant volume, c_v, and (b) constant pressure, c_p. The specific heat c_v is the change in internal energy of a fluid per change in temperature in a constant volume process. The specific heat c_p is the change in enthalpy per change in temperature in a constant pressure process.

For gases, values of c_v are lower than for c_p, since in a constant volume process no work is done and the heat added goes only to increase the temperature. In a constant pressure process part of the heat added produces work and, consequently, more heat must be added to produce a temperature rise of one degree. Both c_v and c_p increase as the temperature increases, but c_v increases at a faster rate than c_p (Figure 3.3.3). When heat is added to a liquid (or solid) during a constant pressure process, there is little change in volume and practically no amount of work is done. The process is almost equivalent to a constant volume process. Thus, specific heats c_p and c_v are almost identical, and usually no distinction is made.

The ratio of specific heats, γ, is a familiar parameter in most gas dynamic equations, and is defined as

$$\gamma = \frac{c_p}{c_v} \quad \text{(Eq 3.3.3)}$$

The ratio γ is dimensionless and decreases as the temperature increases. For perfect gases, γ is assumed constant and independent of temperatures variations.

Values of functions of γ which frequently appear in gas flow process equations are tabulated in Table 3.3.3.

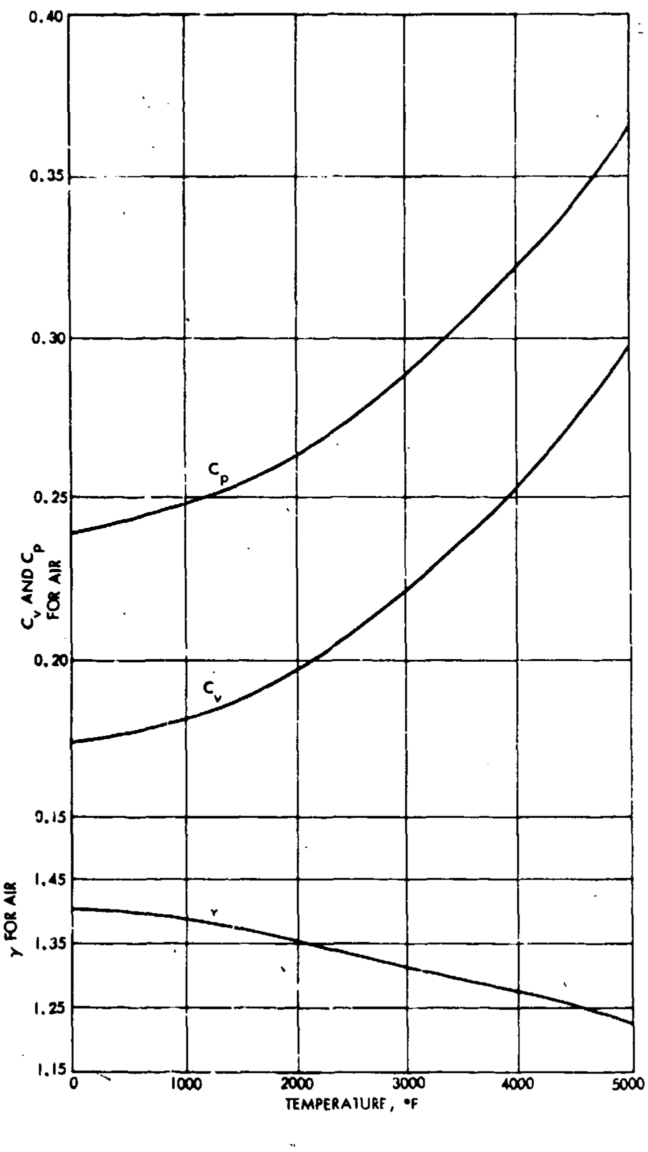

Figure 3.3.3. Effect of Temperature on c_p, c_v, γ for Air

Table 3.3.3. Values of Functions of the Specific Heat Ratio γ

γ	$\sqrt{\frac{2g\gamma}{\gamma-1}}$	$\frac{\gamma-1}{\gamma}$	$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$	$\gamma\sqrt{\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$
1.10	26.61	0.0909	0.5847	0.6590
1.15	22.21	0.1304	0.5744	0.6848
1.20	19.65	0.1667	0.5645	0.7104
1.21	19.26	0.1736	0.5626	0.7155
1.22	18.89	0.1803	0.5607	0.7205
1.23	18.55	0.1870	0.5588	0.7257
1.24	18.23	0.1936	0.5569	0.7307
1.25	17.94	0.2000	0.5549	0.7356
1.26	17.66	0.2064	0.5532	0.7408
1.27	17.40	0.2126	0.5513	0.7457
1.28	17.15	0.2188	0.5494	0.7508
1.29	16.92	0.2248	0.5475	0.7558
1.30	16.70	0.2308	0.5457	0.7608
1.33	16.10	0.2481	0.5405	0.7757
1.36	15.59	0.2647	0.5352	0.7906
1.40	15.01	0.2857	0.5283	0.8102
1.50	13.89	0.3333	0.5120	0.8586
1.60	13.10	0.3750	0.4968	0.9062
1.66	12.72	0.3980	0.4881	0.9340

3.3.4 Viscosity

Absolute viscosity, μ, is a proportionality factor relating the shear stress in a fluid, τ, to its angular rate of deformation, dV_x/dy, as

$$\mu = \frac{\tau}{\frac{dV_x}{dy}} \qquad \textbf{(Eq 3.3.4a)}$$

If the relationship between shear stress and deformation rate is linear, the fluid is said to be *Newtonian* in character (straight line in Figure 3.3.4a). Water, fuels, and most mineral oils are considered Newtonian. If the viscosity varies with deformation rate, the fluid is designated *non-Newtonian*. Non-Newtonian fluids include the broad class of plastics (Bingham body) such as chewing gum; pseudoplastics such as catsup; dilatants such as quicksand; thixotropics such as silica gel; and rheopectics such as gypsum in water. If viscosity is zero, the fluid is defined as being ideal or perfect.

Viscosity contributes to characterizing fluid flow as laminar or turbulent and creates a boundary layer and velocity profile.

Absolute viscosity can be expressed either in terms of force or mass. The symbol μ_m is used to denote absolute viscosity in mass units, while μ_f is used to denote absolute viscosity in force units. In some references μ_f is called the dynamic viscosity. Absolute viscosity has the following dimensions:

$$\mu_m = M/Lt$$

$$\mu_f = Ft/L^2$$

In the English system the units are

$$\mu_m = \text{lb}_m/\text{ft sec}$$

$$\mu_f = \text{lb}_f\ \text{sec}/\text{ft}^2$$

Absolute viscosity expressed as $\text{lb}_f\ \text{sec/in}^2$ is called a *Reyn*. In the metric system, absolute viscosity has units of poise or centipoise. A poise is 100 centipoise and has units of grams per centimeter-second (gm/cm sec). An expression relating μ_m and μ_f is stated

$$\frac{\mu_m}{\mu_f} = g_c \qquad \textbf{(Eq 3.3.4b)}$$

where $g_c = 32.2 \frac{\text{lb}_m}{\text{lb}_f}\ \text{ft/sec}^2$

Kinematic viscosity, ν, is the ratio of the absolute viscosity to the density,

$$\nu = \frac{\mu_m}{\rho} \qquad \textbf{(Eq 3.3.4c)}$$

Note the effect of the density by comparing Figures 3.3.4b and 3.3.4c.

Kinematic viscosity has the dimensions of L^2/t. In the English system, the units are commonly ft^2/sec; in the metric system, the units are expressed in stokes or centistokes. A

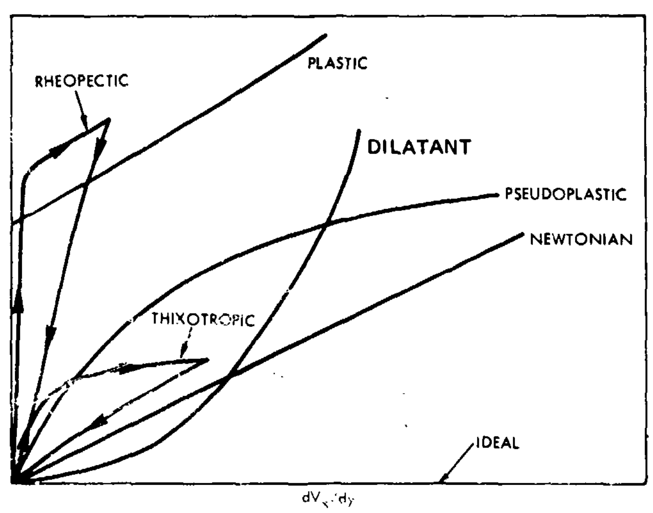

Figure 3.3.4a. Types of Fluids

stoke is equivalent to 100 centistokes, and has units of a square centimeter per second (cm^2/sec).

Kinematic viscosity measurements are more common than those of absolute viscosity, since higher accuracies can be attained. Kinematic viscosity is determined by noting the time required for a given volume of fluid to pass under its own head through a small diameter tube, and is reported in number of seconds. A Saybolt viscosimeter is the most commonly used, of which there are two types, the Universal for low viscosity liquids, and the Furol for highly viscous liquids. Other types of viscosimeters include the Redwood and the Engler. In the use of the Saybolt viscosimeter no standard temperature is specified, but those commonly employed for the Universal are 70, 100, 130, 210°F. For the Furol, 77, 100, 122, 210°F are used. Saybolt Universal is used for flow times between 32 seconds and 1000 seconds. The minimum time for the Furol is 25 seconds (Reference 141-1). Units are termed Saybolt-seconds and can be converted to stokes by the formula

(Eq 3.3.4d)

$$\nu \text{ (stokes)} = 0.00220\, t - \frac{1.80}{t}$$

where t = time, sec.

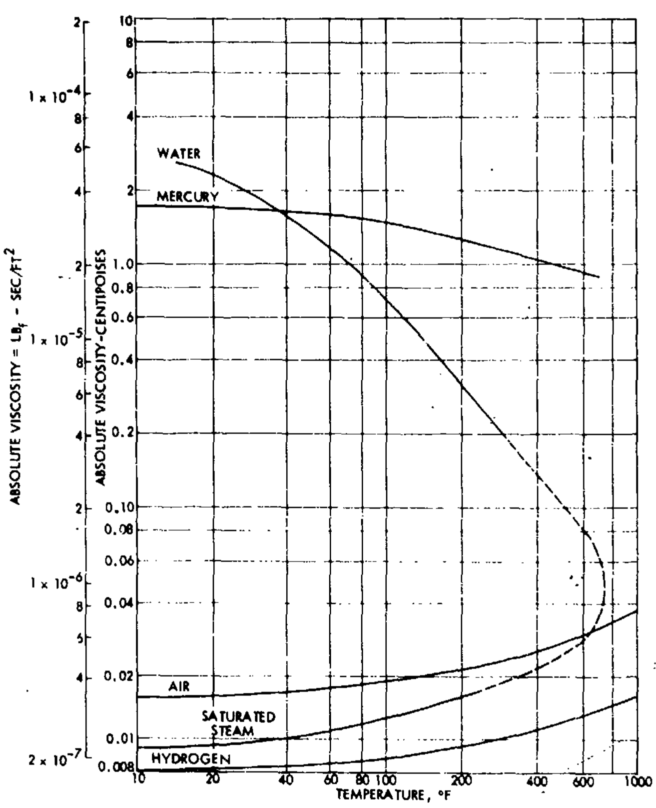

Figure 3.3.4b. Effect of Temperature on Absolute Viscosity for Several Fluids

The viscosity of a gas is relatively constant at pressures ranging from atmospheric to 1000 psia, but the viscosity of a liquid increases rapidly with pressure. Formulae relating vicosity and pressure can be found in Reference 45-1, page 21.

The viscosity of gases usually increases with temperature, and the viscosity of most liquids decreases with an increase in temperature (see Figures 3.3.4b and 3.3.4c). The difference may be explained by considering the fluid's molecular behavior. Resistance to shear is dependent upon the cohesion of the molecules and the transfer of momentum from one fluid layer to another. Cohesion predominates in liquids and since it decreases with temperature, viscosity does likewise. While cohesion is of minor importance in gases, molecular activity increases with temperature, thereby causing an increase in momentum

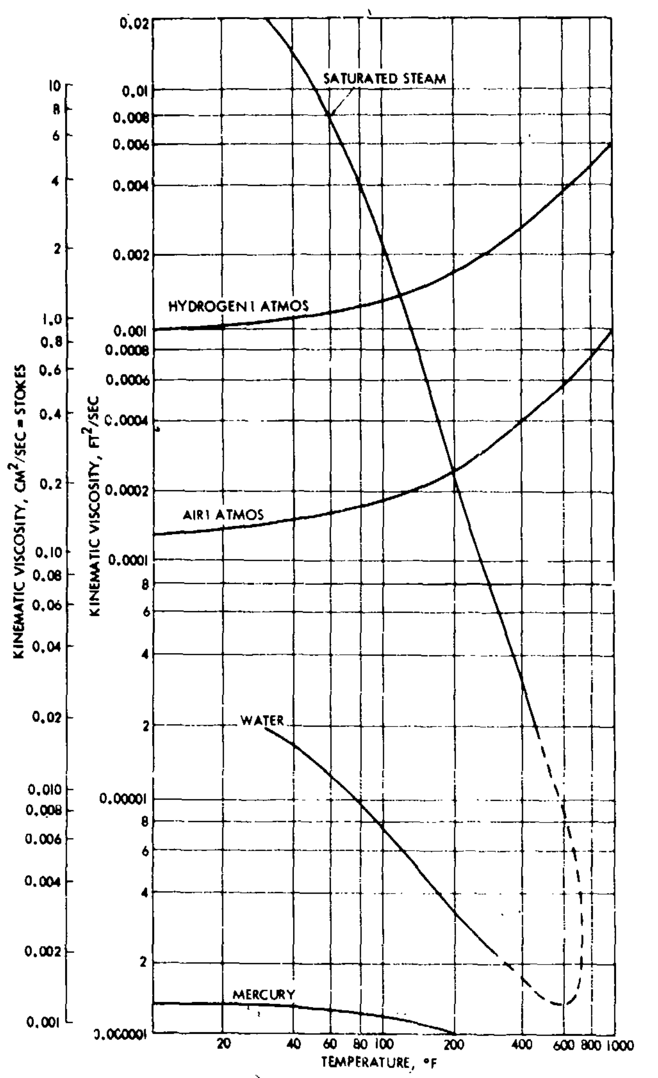

Figure 3.3.4c. Effect of Temperature on Kinematic Viscosity for Several Fluids

transfer and a corresponding increase in viscosity (References 45-1, 137-1). The change in viscosity of gases due to temperature variation may be computed by formulae presented in Reference 45-1, pages 22 and 23.

The viscosity of several fluids can be found in Section 12.0, "Materials."

3.3.5 Cohesion and Adhesion and Their Effects

Cohesion and adhesion result from molecular attraction. Cohesion is the attraction of like molecules and enables a liquid to resist tensile stress. *Surface tension* is a result of cohesion of molecules at the surface of a liquid. Adhesion is the affinity of molecules to adhere to a body composed of a different molecular structure, enabling a liquid to wet a surface. The predominance of either cohesion or adhesion has a decisive effect on the behavior of a liquid. If cohesion predominates, a liquid will possess a high surface tension. It will not wet the surface but will shrink away, leaving the liquid depressed at the surface of contact with a container. The capillary effect will be to depress the liquid below its true height. If adhesion predominates, the liquid will possess a low surface tension and will wet a container surface, rising at the point of contact above the average height of its surface. Surface tension decreases as temperature increases, but the effect on capillarity is relatively small. Capillary rise or depression is shown in Figure 3.3.5. The physical theory of capillary flow phenomena is presented in References 163-2 and 164-1, and general discussions are presented in References 138-1 and 141-1. The units of surface tension (σ) are lb_f/ft (FL^{-1}).

3.3.6 Heats of Fusion and Vaporization

The *heat of fusion* is defined as the heat necessary to change at a constant temperature one unit mass of liquid into its solid state. To return the solid to its liquid state, the heat extracted must be resupplied. In the English system the units of heats of fusion are Btu/lb_m. However, in most references heats of fusion are given in metric units of gram-calories per gram (gm-cal/gm).

The heats of fusion of most organic liquids range between 20 and 60 gm-cal/gm. Water has a value of 79.7 gm-cal/gm, while mercury has a value of 2.66 gm-cal/gm.

The *heat of vaporization* is the heat required to convert one unit mass of liquid (or solid, if sublimation takes place) to the gaseous state. The units of the heat of vaporization in the English system are Btu/lb_m; in the metric system, gm-cal/gm. Heats of vaporization for so-called permanent gases range from 5.97 gm-cal/gm for helium to 106.7 gm-cal/gm for hydrogen. For water, the value is 539.4 gm-cal/gm.

3.3.7 Bulk Modulus and Compressibility

Bulk modulus, β, is a measure of fluid compressibility, influencing the stiffness of a fluid system. To achieve rapid response of control systems, liquids with high bulk moduli (low compressibilities) are used. The bulk modulus has the dimensions of F/L^2, and in the English system units commonly assigned are lb_f/in^2. The value of the bulk modulus is the reciprocal of compressibility, k,

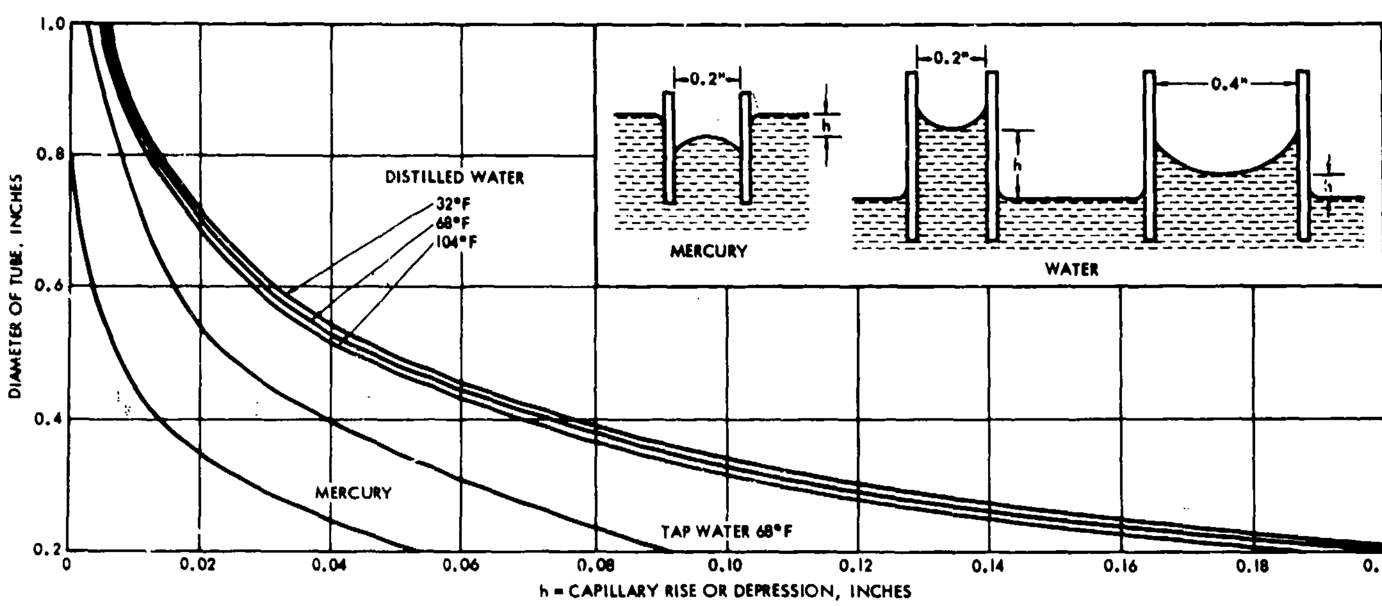

Figure 3.3.5. Capillary Rise in Circular Tubes

(Reprinted with permission from "Fluid Mechanics," Daugherty and Ingersoll, McGraw-Hill Book Company, Inc., 1954, p. 12)

$$\beta = \frac{1}{k} \qquad \textbf{(Eq 3.3.7a)}$$

where β = bulk modulus, lb_f/in^2
k = compressibility, in^2/lb_f

Two equations commonly used for defining bulk modulus are

$$\beta = -V\left(\frac{\partial p}{\partial V}\right) \qquad \textbf{(Eq 3.3.7b)}$$

$$\overline{\beta} = V_i\left(\frac{p - p_i}{V_i - V}\right) \qquad \textbf{(Eq 3.3.7c)}$$

where β = bulk modulus, lb_f/in^2
$\overline{\beta}$ = "mean" bulk modulus, lb_f/in^2
p = pressure, lb_f/in^2
V = volume, in^3
p_i = reference pressure (usually atmospheric), lb_f/in^2
V_i = volume at p_i, in^3

Bulk modulus can be determined under either adiabatic or isothermal conditions. If pressure changes take place slowly, allowing sufficient time for heat transfer between the fluid and its surroundings so that temperature is maintained constant, the bulk modulus is *isothermal.* If pressure changes occur so rapidly that no heat is transferred, the bulk modulus is *adiabatic.* During the operation of aircraft and missile control systems, adibatic conditions are closely approached.

Subscripts s and T are used to indicate adibatic and isothermal bulk moduli, respectively. β_s and β_T are related by the expression

$$\frac{\beta_s}{\beta_T} = \gamma \qquad \textbf{(Eq 3.3.7d)}$$

Values of γ for most liquids range between 1.1 and 1.3.

For gases, by differentiating the perfect gas equation $\left(\frac{p}{\rho} = RT\right)$ and combining with the relationship for compressibility $\left(k = \frac{1}{V}\frac{\partial V}{\partial p}\right)$, the isothermal compressibility factor is

$$k_T = \frac{1}{p} \qquad \textbf{(Eq 3.3.7e)}$$

F[illegible] an isentropic compression

$$k_s = \frac{1}{\gamma p} \qquad \textbf{(Eq 3.3.7f)}$$

Two methods commonly used for measuring bulk modulus or compressibility are the pressure-volume-temperature method and the ultrasonic method. The former is used under isothermal conditions to determine β_T or $\overline{\beta}_T$ and the latter is used to determine β_s.

Sonic velocity and β_s are related by the equation

$$\beta_s = \frac{1}{144\, g_c}\,\rho\, V_s^2 \qquad \textbf{(Eq 3.3.7g)}$$

where ρ = density, lb_m/ft^3
V_s = sonic velocity, ft/sec

At a constant temperature bulk modulus increases with pressure, and at constant pressure the bulk modulus generally decreases with increasing temperature. Water is an exception, reaching a maximum at 120°F and decreasing in both directions as temperature changes from that point, as shown in Figure 3.3.7.

3.3.8 Perfect Gases

3.3.8.1 PERFECT GAS LAW. For a gas to be perfect it must be thermally and calorically perfect. A *calorically* perfect gas specifies that the specific heat at constant volume, c_v, and the specific heat at constant pressure, c_p, are constants. A *thermally* perfect gas follows the equation of state, called the perfect gas law

$$Pv = RT \qquad \textbf{(Eq 3.3.8.1a)}$$

where P = pressure, lb_f/ft^2
v = specific volume, ft^3/lb_m
T = absolute temperature, °R
R = constant for any gas, $ft\text{-}lb_f/lb_m$ °R

The gas constant

$$R = \frac{R_o}{M.W.} \qquad \textbf{(Eq 3.3.8.1b)}$$

where R_o = universal gas constant
= 1544 $ft\text{-}lb_f$/lb-mole °R
M.W. = molecular weight of the gas

The gas constant R is related to the specific heats, c_p and c_v, as follows:

$$c_p = \frac{\gamma R}{J(\gamma - 1)} \qquad \textbf{(Eq 3.3.8.1c)}$$

and

$$c_v = \frac{R}{J(\gamma - 1)} \qquad \textbf{(Eq 3.3.8.1d)}$$

where
γ = ratio of specific heats, Equation 3.3.3
J = conversion constant = 778 $ft\text{-}lb_f$/Btu

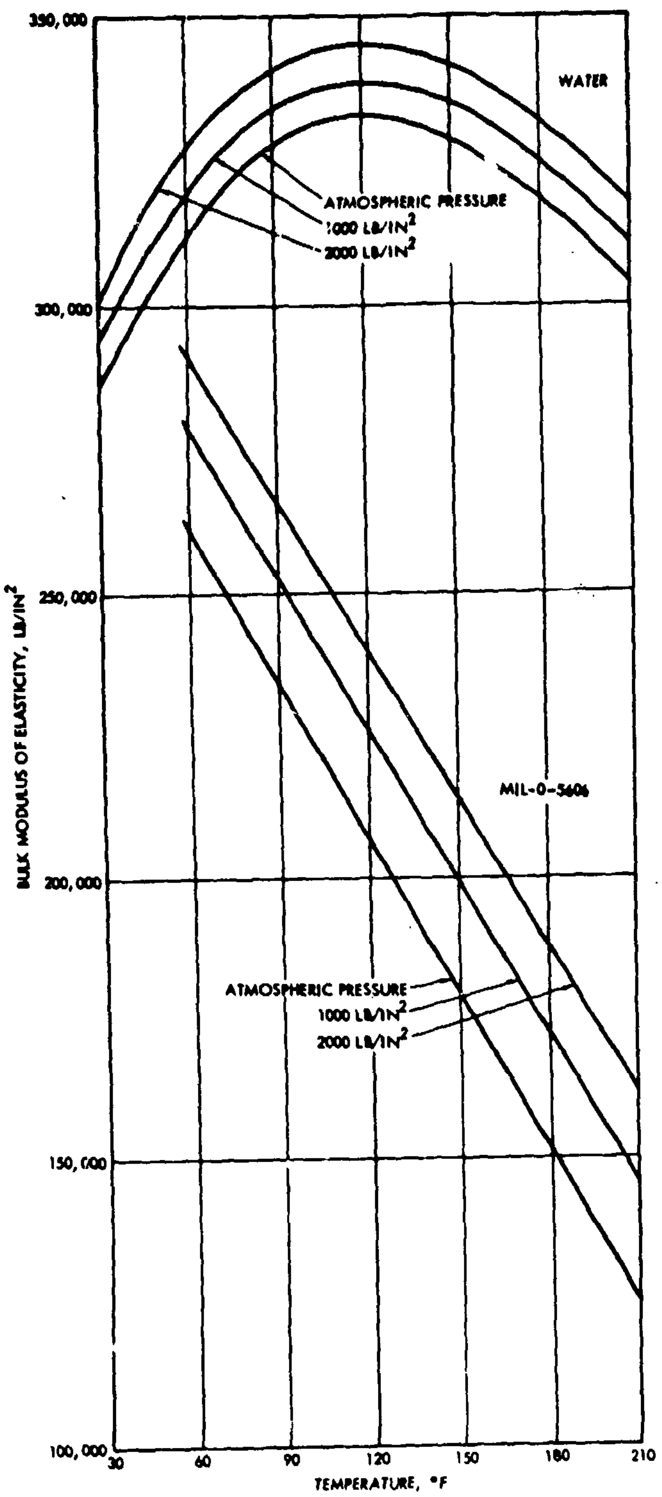

Figure 3.3.7. Bulk Moduli of Elasticity for Water and Hydraulic Oil as Influenced by Temperature and Pressure

3.3.8.2 BOYLE'S AND CHARLES' LAWS. The equation of state is an embodiment of Boyle's Law and Charles' Law. *Boyle's Law* states that at constant temperature volume varies inversely with pressure

$$P_1v_1 = P_2v_2 = \text{constant} \qquad \textbf{(Eq 3.3.8.2a)}$$

Charles' Law states that at constant volume an increase in pressure is proportional to an increase in temperature, or

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} = \text{constant} \qquad \textbf{(Eq 3.3.8.2b)}$$

3.3.8.3 PROCESS EQUATIONS. Unique solutions of the perfect gas equation can be obtained only if two of the variables (P, v, T) are known. By investigating the process by which the final conditions are achieved, one of the variables may be eliminated or considered a constant. The most general case of a polytropic process produces another fundamental equation for a perfect gas

$$P_1v_1^n = P_2v_2^n \qquad \textbf{(Eq 3.3.8.3a)}$$

The significance of (n) is illustrated in a P-v diagram, shown in Figure 3.3.8.3.

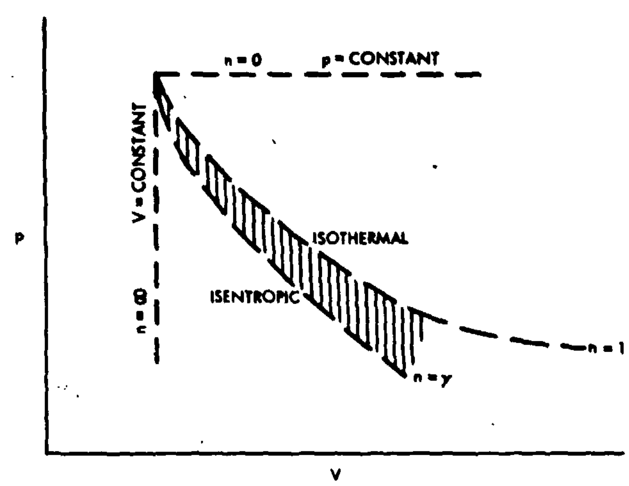

Figure 3.3.8.3. P-V Diagram for Four Processes

If $n = 1$ the equation is simply Boyle's Law for a constant temperature process as defined by Equation (3.3.8.2a).

If $n = \infty$, the equation satisfies Charles' Law for a constant volume process as defined in Equation (3.3.8.2b).

If $n = 0$, the process is a constant pressure process, and

$$\frac{T_2}{T_1} = \frac{v_2}{v_1} \qquad \textbf{(Eq 3.3.8.3b)}$$

For an isentropic process, $n = \gamma$, and Equation (3.3.8.3a) becomes

$$P_1 v_1^{\gamma} = P_2 v_2^{\gamma} \qquad \text{(Eq 3.3.8.3c)}$$

By substitution of the perfect gas law into the above equation, the following temperature relations may be obtained

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1} \qquad \text{(Eq 3.3.8.3d)}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} \qquad \text{(Eq 3.3.8.3e)}$$

A summary of perfect gas formulae are given in Table 3.3.8.3.

Isothermal and isentropic processes are, in general, special cases. Real flow processes (polytropic) show a P-v relation in the region between $n = \gamma$ and $n = 1$.

Values of γ are, in general, highest for monatomic gases and decrease as the number of atoms in the molecule increase. For monatomic gases at room temperatures, γ has values of approximately 1.66 to 1.68. For diatomic gases, including air, values of γ are in the order of 1.40. Triatomic gases, carbon dioxide and sulfur dioxide, have values of 1.28 and 1.26, respectively (Reference 141-1). For a perfect gas, γ is assumed to be constant, though in reality, γ decreases as the temperature increases.

3.3.8.4 AVOGADRO'S LAW. Avogadro's Law states that at a fixed temperature and pressure, a mole of any perfect gas occupies the same volume and contains the same number of molecules. A mole is defined as the mass of a substance numerically equal to its molecular weight. The number of molecules per gram mole, called Avogadro's number, equals 6.023×10^{23}. The number of molecules per pound mole is 2.725×10^{26}. In the metric system one gram mole of any gas at 0°C and 1 atm pressure has a volume of 22.4 liters. For example, 32 grams of O_2 (M.W. = 32) occupies 22.4 liters at 0°C and 1 atm.

In the English system, a pound mole at 32°F and 1 atm pressure occupies a volume of 359 ft^3. For example, 32 lb_m of O_2 at 32°F and 1 atm occupies 359 ft^3.

3.3.8.5 PERFECT GAS REGION. A perfect gas is an idealized case of a real gas, and its precepts may be applied only when a gas is at a temperature well above saturation. For most so-called permanent gases (gases whose evaporation temperatures are in the cryogenic region, e.g., nitrogen, oxygen, etc.), the perfect gas law is applicable with a good degree of accuracy in regions shown on p-V and p-T diagrams (Figures 3.3.8.5a and 3.3.8.5b). At pressures greater than approximately 3000 psia or at temperatures below the critical temperature, T_c, (except at pressures below 100 psia) real gas effects must be considered (Reference 45-1).

3.3.8.6 TOTAL TEMPERATURE AND PRESSURE. The total temperature of a flowing gas is that temperature which would be found if the gas were brought to rest

Table 3.3.8.3. Perfect Gas Formulae

PROCESS	CONSTANT VOLUME V = C	CONSTANT PRESSURE P = C	ISOTHERMAL T = C	ISENTROPIC s = C	POLYTROPIC $Pv^n = C$
P, v, T relations	$\frac{T_2}{T_1} = \frac{P_2}{P_1}$	$\frac{T_2}{T_1} = \frac{v_2}{v_1}$	$P_1 v_1 = P_2 v_2$	$P_1 v_1^{\gamma} = P_2 v_2^{\gamma}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma - 1}$ $= \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}$	$P_1 v_1^{n} = P_2 v_2^{n}$ $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n - 1}$ $= \left(\frac{P_2}{P_1}\right)^{\frac{n - 1}{n}}$
$\int_1^2 P\,dv$	0	$P(v_2 - v_1)$	$P_1 v_1 \ln \frac{v_2}{v_1}$	$\frac{P_2 v_2 - P_1 v_1}{1 - \gamma}$	$\frac{P_2 v_2 - P_1 v_1}{1 - n}$
$u_2 - u_1$	$c_v (T_2 - T_1)$	$c_v (T_2 - T_1)$	0	$c_v (T_2 - T_1)$	$c_v (T_2 - T_1)$
q	$c_v (T_2 - T_1)$	$c_p (T_2 - T_1)$	$\frac{P_1 v_1}{J} \ln \frac{v_2}{v_1}$	0	$c_n (T_2 - T_1)$
n	∞	0	1	γ	$-\infty$ to $+\infty$
Specific heat	c_v	c_p	∞	0	$c_n = c_v \frac{\gamma - n}{1 - n}$
$h_2 - h_1$	$c_p (T_2 - T_1)$	$c_p (T_2 - T_1)$	0	$c_p (T_2 - T_1)$	$c_p (T_2 - T_1)$
$s_2 - s_1$	$c_v \ln \frac{T_2}{T_1}$	$c_p \ln \frac{T_2}{T_1}$	$\frac{R}{J} \ln \frac{v_2}{v_1}$	0	$c_n \ln \frac{T_2}{T_1}$

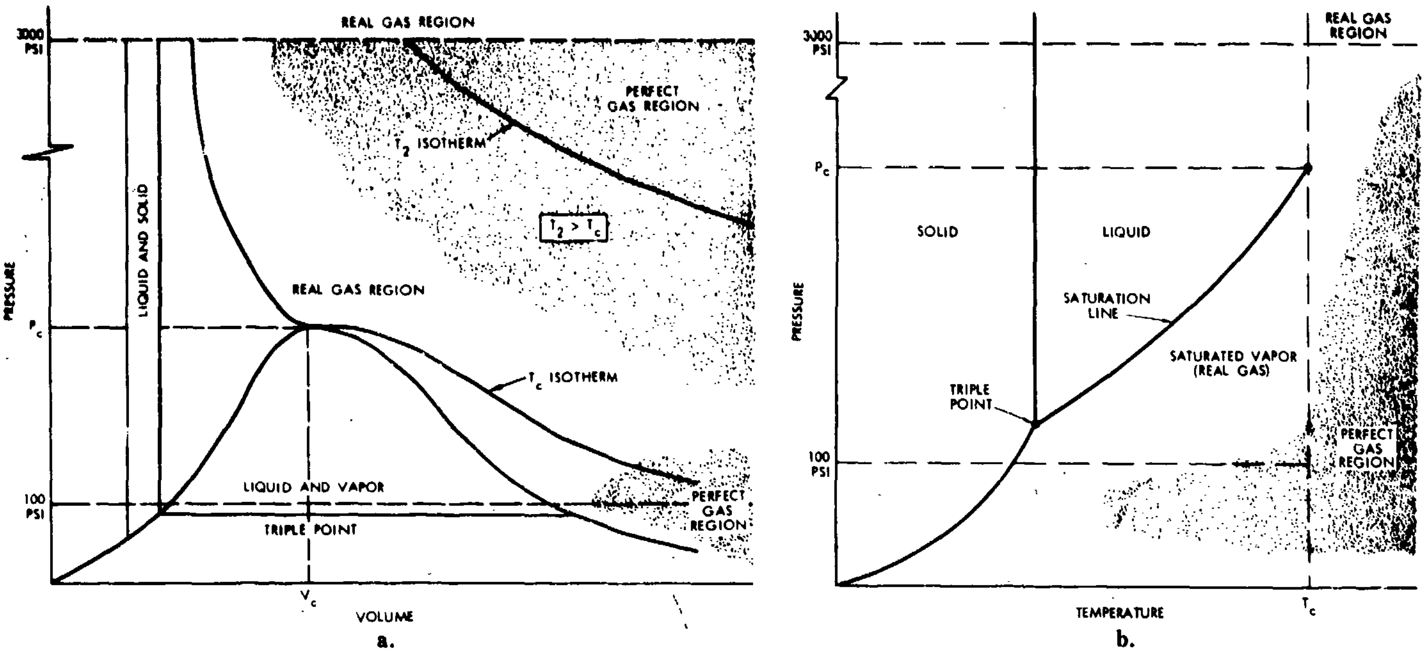

Figure 3.3.8.5a, b. Perfect Gas Regions Shown on P-V and P-T Diagrams

isentropically. The kinetic energy of the fluid is converted to thermal energy. The total temperature is sometimes called the stagnation temperature of the gas and is calculated by

(Eq 3.3.8.6a)

$$T_t = T_1\left[1 + \left(\frac{g}{g_c}\right)\left(\frac{V^2}{2g}\right)\left(\frac{\gamma - 1}{\gamma R T_1}\right)\right]$$

where T_t = total (stagnation) temperature, °R
T_1 = free stream temperature, °R

$\left(\frac{g}{g_c}\right)\left(\frac{V^2}{2g}\right)\left(\frac{\gamma - 1}{\gamma R T_1}\right)$ = kinetic energy in terms of thermal energy (°R)

Note: $\left(\frac{g}{g_c}\right)$ = 1 lb_f/lb_m when g = 32.2 ft/sec^2

The total pressure, using the perfect gas relationship, is

(Eq 3.3.8.6b)

$$P_t = P_1\left(1 + \frac{(\gamma - 1)V^2}{2\gamma g_c R T_1}\right)^{\frac{\gamma - 1}{\gamma}}$$

3.3.8.7 **SONIC VELOCITY.** The sonic velocity of a perfect gas may be expressed as

$$V_s = \sqrt{\gamma g_c R T} \qquad \textbf{(Eq 3.3.8.7a)}$$

where V_s = sonic velocity, ft/sec
γ = ratio of specific heats, Equation (3.3.3)
$g_c = 32.2 \frac{lb_m}{lb_f}$ ft/sec^2
R = gas constant, ft-lb_f/lb_m °R
T = absolute temperature, °R

For air at normal pressures the sonic velocity may be determined by the equation

$$V_s = 49.1\sqrt{T} \qquad \textbf{(Eq 3.3.8.7b)}$$

where V_s = sonic velocity, ft/sec
T = absolute temperature, °R

The sonic velocity for several common gases is given in Table 3.3.8.7.

Table 3.3.8.7. Sonic Velocities for Some Gases at 530°R

GAS	SONIC VELOCITY (ft/sec)
Helium	3310
Air	1130
Oxygen	1072
Nitrogen	1140
Hydrogen	4280

ISSUED: MARCH 1967
SUPERSEDES: MAY 1964

3.3.8.8 MACH NUMBER. The Mach number is the ratio of the fluid stream velocity to the local sonic velocity, V_s, or

$$M = \frac{V_{stream}}{\sqrt{\gamma g_c RT}} \qquad \text{(Eq 3.3.8.8)}$$

3.3.8.9 CRITICAL VALUES. In Equation (3.3.8.7a) T is called the *critical temperature,* hereafter designed T_c. This is not to be confused with the critical values discussed in Sub-Topic 3.3.9. In terms of the stagnation (total) temperature, the critical temperature may be expressed

$$T_c = \left(\frac{2}{\gamma + 1}\right)T_t \qquad \text{(Eq 3.3.8.9a)}$$

Substitution of the critical temperature in terms of stagnation temperature gives the so-called *critical velocity*

$$V_c = \sqrt{\frac{2\gamma}{\gamma + 1} g_c RT_t} \qquad \text{(Eq 3.3.8.9b)}$$

The critical velocity is the only velocity where the stream velocity equals the velocity of sound for a given stagnation temperature. For air,

$$V_c = 109.7 \sqrt{T_t} \qquad \text{(Eq 3.3.8.9c)}$$

By use of the perfect gas isentropic relationships, the *critical pressure* may be expressed as

$$P_c = P_1\left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}} \qquad \text{(Eq 3.3.8.9d)}$$

Since air has a γ value of 1.4, the critical pressure ratio, P_c/P_1, has a value of 0.529. Thus air flow in a pipe or through a constant area restriction will be sonic when the throat pressure is 52.9 per cent of the inlet pressure (Figure 3.3.8.9). Values of critical pressure ratios for several gases are listed in Table 3.3.8.9.

Table 3.3.8.9. Values of P_c/P_1 for Several Gases

GAS	γ	$\frac{P_c}{P_1}$
Air	1.40	0.5283
Nitrogen	1.40	0.5283
Hydrogen	1.40	0.5283
Oxygen	1.40	0.5283
Helium	1.66	0.4881
Methane	1.30	0.5457

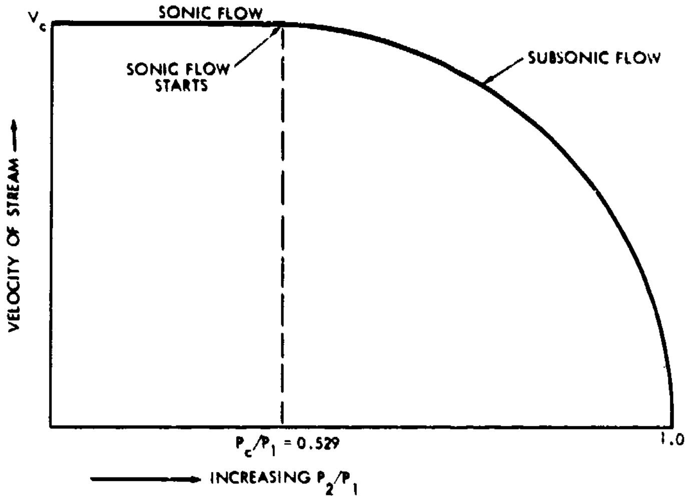

Figure 3.3.8.9. Critical Flow for Air

3.3.9 Vapor Pressure and Critical Points

A vapor is distinguished from a gas in that it can be converted to a liquid by a slight increase in pressure. The minimum pressure at a given temperature necessary for condensation is known as the vapor pressure of the liquid. Condensation is accompanied by a liberation of heat which is a return of the heat of vaporization. Unless the heat is removed, the temperature of both the liquid and vapor will rise.

Vapor pressure increases with temperature (Figure 3.3.9) until it equals the pressure of the vapor over the liquid. The temperature at which the vapor pressure of a liquid equals the pressure over the liquid is known as the *boiling point.* If the pressure over the liquid is barometric pressure, the temperature is known as the *normal boiling point.*

The *critical temperature* is the temperature above which a gas cannot be liquefied by compression alone. The *critical pressure* is the saturation pressure corresponding to the critical temperature. Table 3.3.9 lists critical points for some gases.

Table 3.3.9. Critical Pressures and Temperatures

GAS	SYMBOL	$P_{critical}$ (psia)	$T_{critical}$ (°R)
Ammonia	NH_3	1640	730
Carbon dioxide	CO_2	1072	548
Nitrogen	N_2	492	227
Oxygen	O_2	730.4	278
Helium	He	33.2	9.4
Hydrogen	H_2	188	59.8

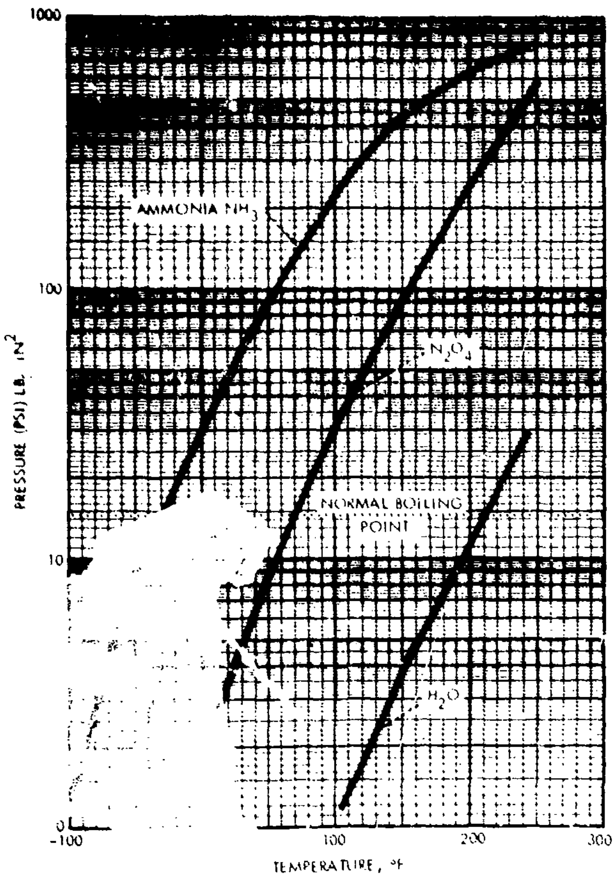

Figure 3.3.9. Vapor Pressure for Several Fluids

3.3.10 Real Gases

3.3.10.1 EQUATIONS OF STATE. Attempts have been made to treat real gas effects by means of semi-empirical equations such as those of Van der Waals and Beattie-Bridgeman. Van der Waals' equation of state is given by the relationship

$$p = \frac{RT}{v - b} - \frac{a}{v^2} \qquad \textbf{(Eq 3.3.10.1a)}$$

where
- p = pressure, atm
- R = gas constant, atm ft³/mole °R
- T = absolute temperature, °R
- v = specific volume, ft³/mole
- a, b = constants for particular gas

Van der Waals' equation correlates characteristics of real gases (p-v-T trends) even though it does not represent the values of properties with precision. The relationship holds for the liquids and vapors near the critical pressure as well as pressures above the critical.

The Beattie-Bridgeman Equation is

$$p = \frac{RT\,(1 - \alpha)\,(v + B)}{v^2} - \frac{A}{v^2} \qquad \textbf{(Eq. 3.3.10.1b)}$$

where
$$A = A_0\left(1 - \frac{a}{v^2}\right)$$
$$B = B_0\left(1 - \frac{b}{v^2}\right)$$
$$\alpha = \frac{c}{vT^3}$$

- R = gas constant, atm ft³/mole °R
- T = absolute temperature, °R
- v = specific volume, ft³/mole
- p = pressure, atm
- a, b, c = constants

The Beattie-Bridgeman Equation is applicable for pressures above the triple point.

Constants for Equations (3.3.10.1a) and (3.3.10.1b) for several gases are listed in Table 3.3.10.1.

3.3.10.2 COMPRESSIBILITY FACTORS. A simpler method for representing the behavior of real gases introduces a dimensionless correction factor, Z, into the perfect gas equation such that

$$Pv = ZRT \qquad \textbf{(Eq. 3.3.10.2a)}$$

where
- P = pressure, lb_f/ft^2
- v = specific volume, ft^3/lb_m
- R = gas constant, ft-lb_f/lb_m °R
- T = absolute temperature, °R
- Z = compressibility factor, the values of which may be obtained from Figures 3.3.10.2a and 3.3.10.2b for any gas if its critical values are known

By use of these figures, plotted at reduced values of temperature and pressure, the properties of any gas can be determined with accuracies of 1 to 2 percent. A reduced value of temperature is defined as the ratio of actual temperature to the critical temperature, thus

$$T_r = \frac{T_{actual}}{T_{critical}} \qquad \textbf{(Eq. 3.3.10.2b)}$$

and the reduced pressure is defined as

$$p_r = \frac{p_{actual}}{p_{critical}} \qquad \textbf{(Eq. 3.3.10.2c)}$$

More accurate values of (Z) for helium and hydrogen are obtained by using pseudo-critical values ($p_c{}^*$, $T_c{}^*$) as follows: for helium, $p_c{}^* = 151.5$ psi, $T_c{}^* = 8.3$°R; for hydrogen, $p_c{}^* = 306$ psi, $T_c{}^* = 41.3$°R.

When (Z) is equal to unity, Equation (3.3.10.2a) becomes the perfect gas law. The actual temperature increases as the line of reduced temperature approaches the line indicating a (Z) value of unity. As the actual pressure increases, the iso-reduced temperature line diverges from Z = 1.

Table 3.3.10.1. Gas Equations' Constants

(From "Aerospace Applied Thermodynamics Manual," Society of Automotive Engineers, Committee A-9, 1960)

GAS	VAN DER WAALS' EQUATION		BEATTIE-BRIDGEMAN EQUATION				
	a $\frac{\text{Atm ft}^6}{\text{mole}^2}$	b $\frac{\text{ft}^3}{\text{mole}}$	A_o $\frac{\text{Atm ft}^3}{\text{lb mole}^2}$	B_o $\frac{\text{ft}^3}{\text{lb mole}}$	a $\frac{\text{ft}^3}{\text{lb mole}}$	b $\frac{\text{ft}^3}{\text{lb mole}}$	$c \times 10^{-4}$ $\frac{\text{ft}^3 \cdot \text{R}^3}{\text{lb mole}}$
Air	343.8	0.585	334.1	0.739	0.309	−0.176	406
O_2	349.5	0.510	382.5	0.741	0.410	0.0674	448
N_2	346.0	0.618	344.3	0.809	0.419	0.111	391.7
CO_2	924.2	0.685	1248.9	1.678	1.143	1.159	165.0
NH_3	1076.0	0.598	613.9	0.547	2.729	3.062	44,560.0
H_2	63.2	0.427	50.57	0.336	−0.0811	−0.698	4.7
He	—	—	5.6	0.224	0.958	0.0	0.37

In addition to the so-called permanent gases, (e.g., air, nitrogen) the compressibility charts are also applicable to many organic vapors (Reference 141-1).

3.3.10.3 JOULE-KELVIN EFFECT. The Joule-Kelvin effect, also called the negative Joule-Thomson effect, is a relationship of temperature change per unit decrease in gas pressure for an irreversible, adiabatic, free expansion process. Most gases at normal temperature cool during a pressure decrease, but at temperatures or pressures above transition point the temperature may actually rise as pressure decreases. A transition point is reached when the temperature begins to increase with a decrease in pressure. The points of transition, as functions of temperature and pressure, form the so-called *inversion curve*. An inversion curve for nitrogen is shown in Figure 3.3.10.3. Some gases, notably helium, have low inversion temperatures. The inversion point for helium has been estimated to be between 55.0° and 81°R at one atmosphere pressure, and between 34.9° and 81°R at 30 atmospheres pressure.

3.4 FLUID STATICS

3.4.1 Static Pressure

Static pressure is defined as the force exerted on a surface of unit area and has dimensions of F/L^2. Static pressure refers to pressure exerted by a fluid at rest on its surroundings, or pressure exerted by a fluid in motion normal to the direction of flow. Pressures may be given as either absolute or gage, with the relationship between the two shown in Figure 3.4.1. The reference (datum) plane is set at zero pressure. Absolute pressure is measured with respect to the datum plane and is always positive. In the English system of units, it is given in dimensions of pounds force per square inch, pounds per square foot, or in multiples of standard atmospheres (14.7 psia). Gage pressure is measured relative to the local atmospheric (barometric) pressure. Positive gage pressure is com-

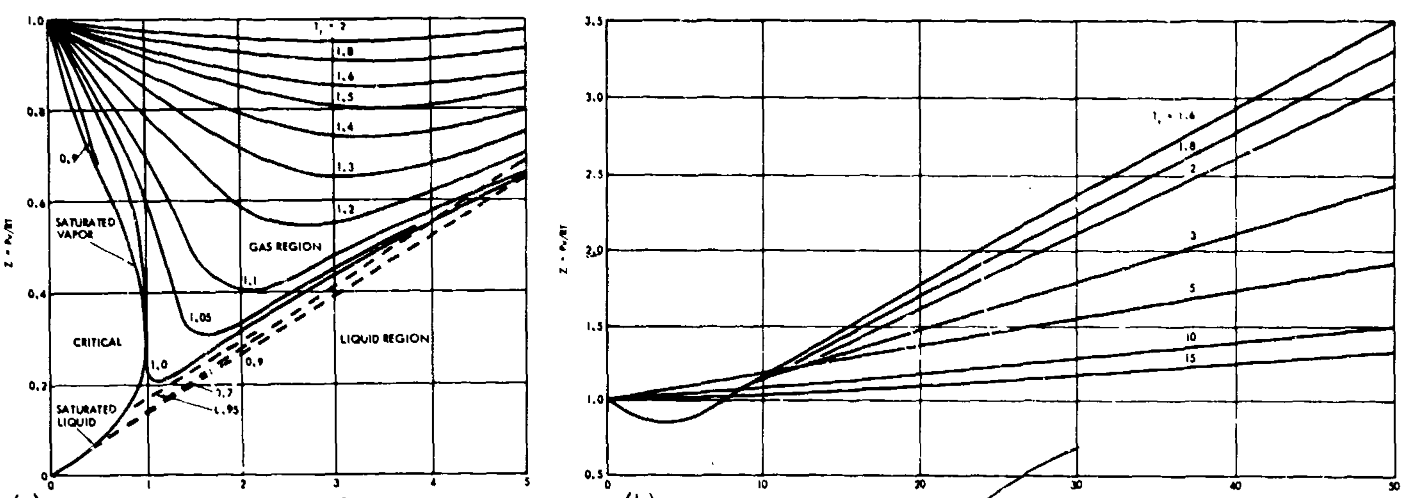

Figure 3.3.10.2a, b. Compressibility Factors

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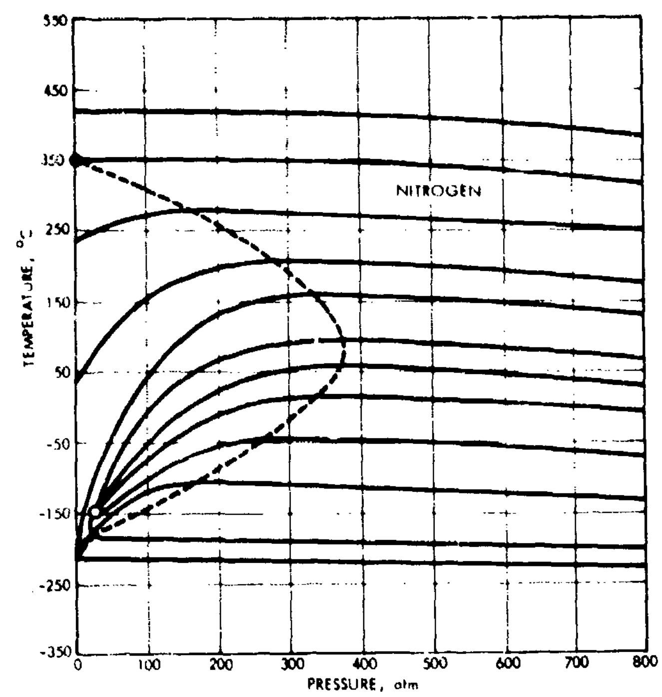


Figure 3.3.10.3. Insenthalpic Curves and Inversion Curve for Nitrogen
(Reprinted with Permission from "Heat and Thermodynamics," M. W. Zemansky, McGraw-Hill Book Company, Inc., 1951)

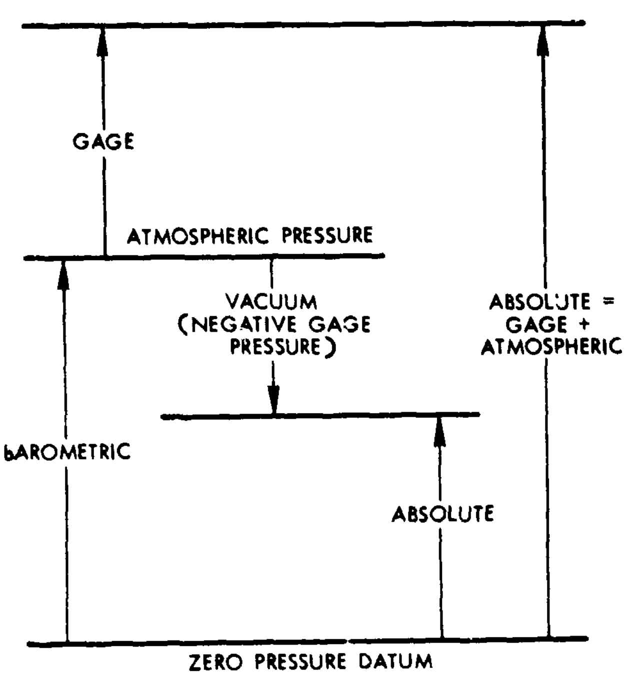


Figure 3.4.1. **Absolute and Gage Pressure Relationships**

monly expressed in pounds per square inch or pounds per square foot, while negative gage pressure, termed *vacuum*, is usually expressed in inches of mercury (Hg). Vacuum may also be expressed in terms of the absolute pressure with dimensions of millimeters of Hg, inches of Hg, or microns of Hg. In vacuum technology, one millimeter of Hg is commonly referred to as one *torr*, and 760 torrs are equivalent to one atmosphere.

3.4.2 Pressure Variation With Density and Height

When gage pressures are designated in terms of fluid height, pressure in terms of force per unit area may be calculated from the equation

$$P = \rho\left(\frac{g}{g_c}\right)h \qquad \textbf{(Eq. 3.4.2a)}$$

or

$$P = wh \qquad \textbf{(Eq. 3.4.2b)}$$

where
P = gage pressure, lb_f/ft^2
ρ = density, lb_m/ft^3
w = specific weight, lb_f/ft^3
h = fluid height, ft
$\frac{g}{g_c}$ = 1 lb_f/lb_m when g = 32.2 ft/sec^2

From Pascal's Law, all points at the same depth (normal to gravitational field) in a fluid at rest are at the same pressure. In general terms

$$\int dP = \int \rho\left(\frac{g}{g_c}\right)dh = \int w\,dh \qquad \textbf{(Eq. 3.4.2c)}$$

If density increases with depth as with a compressible fluid, e.g., air, a relation between ρ and either P or h must be obtained before Equation (3.4.2c) can be solved. For a perfect gas, pressure is related to density by the equation of state.

If the density does not vary with depth, as for an incompressible fluid, the pressure difference between two planes is

$$P_2 - P_1 = \rho\left(\frac{g}{g_c}\right)(h_2 - h_1) \qquad \textbf{(Eq. 3.4.2d)}$$

or

$$P_2 - P_1 = w\,(h_2 - h_1) \qquad \textbf{(Eq. 3.4.2e)}$$

3.4.3 Buoyant Force

The concept of buoyancy was first postulated by Archimedes and may be stated, "A body immersed in a fluid loses as much weight as the weight of the fluid the body displaces." In Figure 3.4.3 the buoyant force, F_B, shown vectorially vertically upward, is equal to $F_y' - F_y$ (weight of the volume of fluid displaced).

If the weight of the body, W, equals the buoyant force, F_B, the densities of the body and the displaced fluid are equal and the body is in equilibrium. If the body weight is greater than the buoyant force, the body will sink. If the

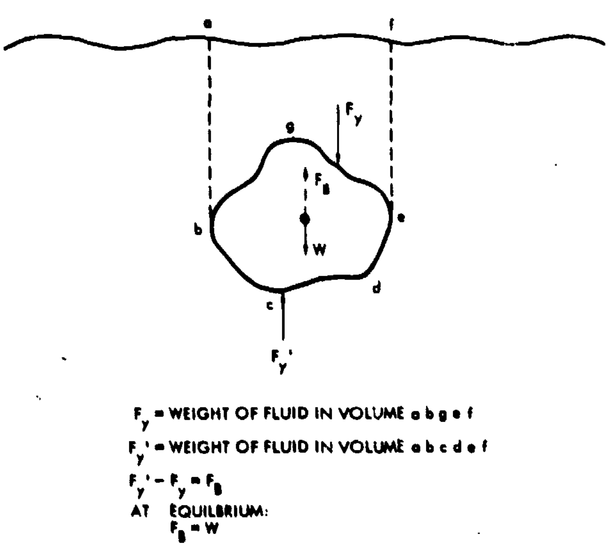


Figure 3.4.3. Buoyant Force on a Submerged Body

body is more compressible than the fluid, its density will increase more rapidly with depth than that of the fluid. As the body reduces in size, the buoyant force decreases and the body will sink to the bottom. If less compressible than the fluid, the body will sink until the two densities are equal.

If the body weight is less than the buoyant force, the body will rise until the body and fluid densities are equal. If the body is more compressible than the fluid, it will rise indefinitely; if it is less compressible, a definite equilibrium level will be reached (Reference 138-1).

3.5 FLOW REGIMES

The characteristics of fluid motion are influenced by viscous and inertial forces. *Viscous forces* resist changes to the form of flow and are dependent upon viscosity, μ_m. *Inertial forces* are represented by the product of stream velocity, a characteristic dimension of length and fluid density, $V\rho L_c$. The ratio of inertial forces to viscous forces is called *Reynolds number*

$$Re = \frac{V\rho L_c}{\mu_m} \qquad \textbf{(Eq. 3.5a)}$$

where V = velocity, ft/sec
ρ = density, lb_m/ft^3
L_c = characteristic length, ft
μ_m = absolute viscosity, lb_m/ft sec

For pipe flow, L_c equals the cross-sectional diameter of the pipe, D. When viscous forces predominate, the fluid flows in what are sometimes considered to be parallel layers, or lamina. This flow is called *laminar*. The velocity vector of each particle is tangent to a smooth (imaginary) path called a *streamline*. Without constraint, streamlines will be straight lines. If the fluid is compelled to alter its path gradually, the streamlines may turn, or tend to diverge or converge. In general, flow in pipes is considered laminar if the Reynolds number is less than 2000. Laminar flows in pipe have been attained at Reynolds numbers as high as 50,000 in laboratories, but these are exceptional cases (Reference 138-1). When inertial forces predominate, the particles assume a random lateral motion which is superimposed on the main forward motion. Particles no longer follow streamlines but may have velocity vectors at arbitrary angles to the main direction of flow. This flow is called *turbulent*. In pipes, flow is considered turbulent at Reynolds numbers greater than 4000.

Between Reynolds numbers of 2000 and 4000, flow is termed critical where it is in a state of transition varying between laminar and turbulent flows. Critical flow occurs during changes from laminar flow to turbulent flow or vice versa, and possesses characteristics common to both flows (Figure 3.5a).

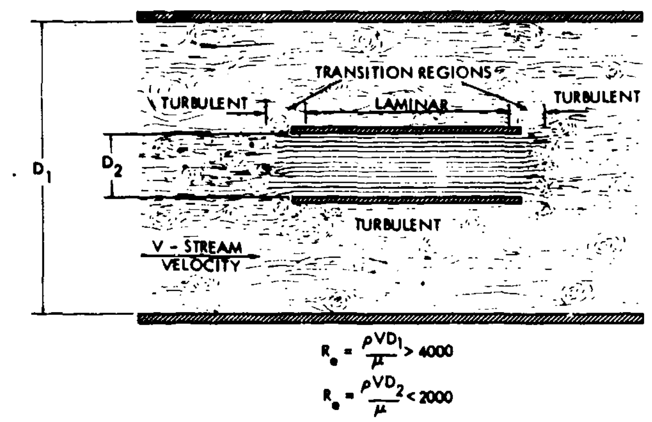


Figure 3.5a. Laminar and Turbulent Flow Regions

The flow transition values of Reynolds numbers are somewhat arbitrary. Some references use Reynolds numbers from a low of 1100 to 1600, and to a high of 3000 as defining the critical range. The values of Reynolds numbers are applicable to liquids, gases, and two-phase flow. At transition from laminar to turbulent flow, convective heat transfer increases greatly and continues to increase as fluid turbulence increases.

In two-phase liquid-gas flow four basic flow types can exist:

a) Laminar liquid and laminar vapor
b) Laminar liquid and turbulent vapor
c) Turbulent liquid and laminar vapor
d) Turbulent liquid and turbulent vapor.

The four types of flow may take on various observable forms depending upon magnitude of the gas flow rate relative to the liquid flow rate. In order of increasing gas flow rate, these flow forms for horizontal tubes are: bubble,

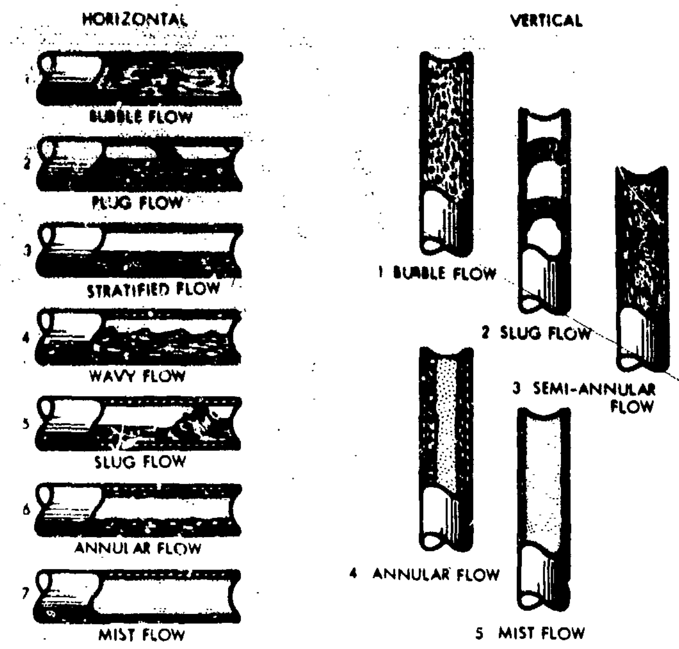


Figure 3.5b. Types of Two-Phase Flow
(Reprinted from WADC TR-55-422, Gresham, Foster, and Kyle)

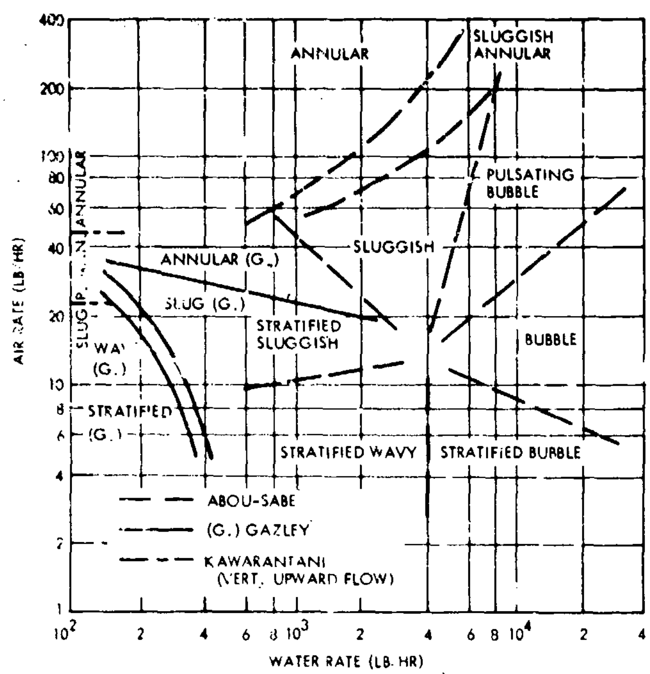


Figure 3.5c. Regions of Two-Phase Flow of an Air-Water Mixture
(Reprinted from WADC TR-55-422, Gresham, Foster and Kyle)

plug, stratified, wavy, slug, annular, and mist. These flow forms are shown in Figure 3.5b. The observed regions and types of two-phase horizontal flow of an air and water mixture are shown in Figure 3.5c.

3.6 BOUNDARY LAYER AND VELOCITY PROFILE

When a viscous fluid flows past stationary walls, (e.g., in a pipe) the particle velocity next to the wall is reduced to zero due to friction. Fluid viscous forces cause the establishment of a *velocity gradient* with velocity ranging from zero at the walls to the free stream value. The region over which this occurs is called the *boundary layer.* The viscous shearing stresses cause the boundary layer to thicken as it progresses downstream from a geometrical discontinuity.

Where viscous forces predominate (Re < 2000) the laminar boundary layer will thicken and may join the boundary layer from the opposite wall to form free stream laminar flow. The growth of the laminar velocity profile and boundary layer is shown in Figure 3.6a. Initially, each particle possesses the same velocity. The laminar boundary layer thickness as a function of length along the pipe is

$$\frac{\delta}{x} = \frac{4.91}{\sqrt{\frac{\rho V x}{\mu_m}}} \qquad (\text{For } \delta < D/20) \qquad \textbf{(Eq. 3.6a)}$$

The distance required to attain a parabolic profile is

$$L = 0.058\, D\, Re \qquad \textbf{(Eq. 3.6b)}$$

where
δ = boundary layer thickness, ft
x = distance along pipe, ft
ρ = density, lb_m/ft^3
V = velocity, ft/sec
μ_m = absolute viscosity, lb_m/ft sec
D = pipe diameter, ft
$Re = \frac{\rho V D}{\mu_m}$

If inertial forces predominate (Re > 4000), a turbulent boundary layer will form adjacent to the laminar layer at some distance downstream from the initial laminar boundary layer (Figure 3.6b). In the turbulent boundary layer no parallel streamlines are present, instead eddies are formed and the streamlines are well mixed. The turbulent boundary layer is considered to be made up of three regions: (1) a laminar sublayer in which viscous forces predominate, (2) a transition or buffer region, and (3) a turbulent region where inertial or turbulent forces predominate. In turbulent flow the effects of shearing are reduced, and the velocity profile, while rather flat near the channel center, is steeper near the wall then in laminar flow due to the laminar sublayer.

If the Reynolds number is very high, transition may occur very near the pipe entrance so that the boundary layer may be considered turbulent along the entire length.

The boundary layer thicknesses for laminar flow with transition to turbulent flow may be expressed

$$\frac{\delta_1}{x_1} = \frac{4.91}{\sqrt{\frac{\rho V x}{\mu_m}}} \qquad (\text{laminar } \delta < D/20) \qquad \textbf{(Eq. 3.6c)}$$

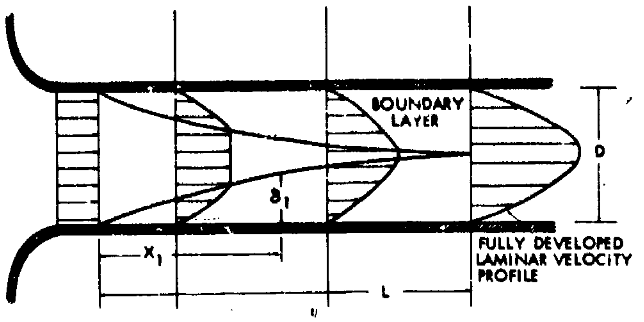


Figure 3.6a. Laminar Boundary Layer and Velocity Profile Growth

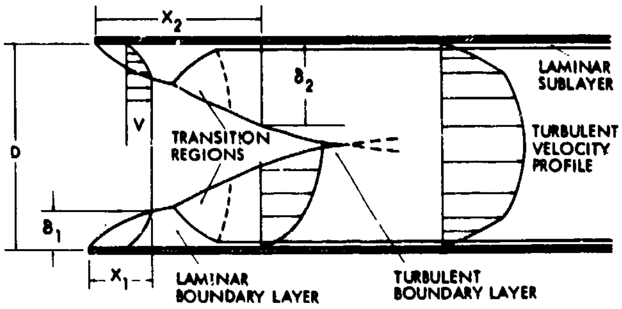


Figure 3.6b. Turbulent Boundary Layer and Velocity Profile Growth

$$\frac{\delta_2}{x_2} = \frac{0.377}{\left(\frac{\rho V x_2}{\mu_m}\right)^{1/5}} \qquad \text{(turbulent)} \qquad \textbf{(Eq. 3.6d)}$$

Subscript 1 refers to laminar flow and subscript 2 refers to turbulent flow.

For more comprehensive discussion of this topic consult Reference 141-1 and its bibliography.

3.7 FLOW EQUATIONS

3.7.1 General

Fluid mechanics is based on the same physical laws which Newton, Lagrange, and early mathematicians applied to the dynamics of rigid bodies. From such laws as the conservation of mass, conservation of momentum, and conservation of energy, the equations describing fluid flow have been derived. In general, the relationships and equations are complex. For example, due to the non-linearities of general flow equations (e.g., the Navier-Stokes Equation), no specific solution exists. However, by assuming that certain parameters within the equations are either negligible or constant, many practical and useful equations such as the Continuity Equation and Bernoulli's Equation have been derived.

Gases may be considered as either incompressible or compressible fluids, depending on the extent of density variation. Liquids are only considered as incompressible fluids (ρ = constant). Density changes are functions of pressure and temperature and are related to the stream velocity and the heat transfer to or from the system. Generally, gases in free stream flow may be considered incompressible if the Mach number is less than 0.20. The behavior analysis of gases as compressible fluids is greatly simplified in Sub-Topics 3.7.1 to 3.7.3 by assuming that the flow is steady and frictionless, that the flow process is isentropic, and that the gas follows the perfect gas law. Flow with friction and heat transfer (Raleigh and Fanno-type gas flow processes) are treated in Sub-Topic 3.7.4. An incompressible fluid compels the flow process to be steady, since the density will not change with time.

For piping systems it is convenient to consider only unidirectional flow. Furthermore, since friction effects are usually non-linear quantities, flow equations are derived for ideal liquids, and a correction factor, derived separately from empirical data, is used to estimate friction effects.

Formulation of dynamic flow equations by the classical approach for two-phase flow appears remote because of the instability and intricate geometry of the flow and the virtual impossibility of expressing the ever-changing boundary conditions (Reference 141-1). The behavior of two-phase flow is dependent on the ratio of gas to liquid and on the geometry of the transporting system, whether it be vertical, horizontal, or inclined. A compilation of theory and expressions for two-phase flow is presented in References 141-1, 188-1, and 213-1.

3.7.2 Continuity Equation

3.7.2.1 DERIVATION. The continuity equation is an expression of the conservation of mass. In general terms, the equation of continuity is

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) = 0 \qquad \textbf{(Eq. 3.7.2.1a)}$$

where $\vec{V} = V_x \vec{i} + V_y \vec{j} + V_z \vec{k}$ = velocity vector

V_x, V_y, and V_z = speeds in x, y, and z directions, respectively, and $\vec{i}, \vec{j}, \vec{k}$ are unit vectors.

For steady flow, $\frac{\partial \rho}{\partial t} = 0$ and Equation (3.7.2.1a) becomes

(Eq. 3.7.2.1b)

$$\frac{\partial}{\partial x}(\rho V_x) + \frac{\partial}{\partial y}(\rho V_y) + \frac{\partial}{\partial z}(\rho V_z) = 0$$

For unidirectional flow ($V_y = 0, V_z = 0$) and Equation (3.7.2.1b) reduces to

$$\frac{d\,(\rho V_x)}{dx} = 0 \qquad \textbf{(Eq. 3.7.2.1c)}$$

By multiplying Equation (3.7.2.1c) by a unit volume (dx dy dz) and integrating, Equation (3.7.2.1c) becomes

(Eq. 3.7.2.1d)
$$\rho_1 V_{x1} A_1 = \rho_2 V_{x2} A_2 = \text{constant}$$

The weight flow rate, $\dot{w}$, is stated

(Eq. 3.7.2.1e)
$$\dot{w} = \rho \left(\frac{g}{g_c}\right) VA = wVA = \text{constant}$$

where ρ = density, lb_m/ft^3
V = velocity, ft/sec
A = cross-sectional area, ft^2
w = specific weight, lb_f/ft^3
$\dot{w}$ = weight flow rate, lb_f/sec

Since density may change along the flow stream, the stream velocity does not necessarily change linearly with cross-sectional area.

Assuming an adiabatic flow process, the velocity and area relationship may be expressed in terms of the Mach number as

$$\frac{dA}{A} = \frac{dV}{V}(M^2-1) \qquad \text{(Eq. 3.7.2.1f)}$$

If $M < 1$, dA/A is negative for positive values of dV/V, that is, increasing velocity corresponding to decreasing area. If $M > 1$, dA/A is positive for positive values of dV/V, that is, an increase in area is accompanied by an increase in velocity (for example, in bell or De Laval nozzles). If $M^2 << 1$, (for example, $M < 0.20$) then $dA/A \approx -dV/V$, which is similar to that of frictionless, incompressible flow.

3.7.2.2 IDEALIZED EQUATIONS. The continuity equation for a fluid is

(Eq. 3.7.2.2a)
$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 = \text{constant}$$

where $(\rho_1 V_1 A_1)$ refers to inlet conditions and $(\rho_2 V_2 A_2)$ refers to downstream conditions. Velocities V_1 and V_2 are mean velocities over the cross-section. Equation (3.7.2.2a) has dimensions of lb_m/ft^3, and is defined as the mass flow rate, $\dot{m}$.

$$\dot{m} = \rho VA = \text{constant} \qquad \text{(Eq. 3.7.2.2b)}$$

The continuity equation for liquid flow is

$$V_1 A_1 = V_2 A_2 \qquad \text{(Eq. 3.7.2.2c)}$$

where $(V_1 A_1)$ refers to the inlet condition and $(V_2 A_2)$ refers to the downstream or exit conditions. Velocities V_1 and V_2 are mean (average) velocities of flow.

The product of velocity and area has dimensions of volume flow per unit time, and is defined as the *volumetric rate of flow* Q.

The *mass rate of flow* is

$$\dot{m} = \rho Q = \rho V_1 A_1 = \rho V_2 A_2 \qquad \text{(Eq. 3.7.2.2d)}$$

The *weight rate of flow* is

(Eq. 3.7.2.2e)
$$\dot{w} = wQ = w V_1 A_1 = w V_2 A_2$$

3.7.3 Energy Equations for Idealized Fluids

3.7.3.1 DERIVATION. For a Newtonian fluid the equation of motion may be written in vector form as

(Eq. 3.7.3.1a)
$$\rho\left[\frac{\partial \vec{V}}{\partial t} + \nabla \frac{V^2}{2} - \vec{V} \times (\nabla \times \vec{V})\right] = \rho \vec{F}_{body} - \nabla p$$
$$+ 4/3\, \nabla (\mu\ \nabla \cdot \vec{V}) + \nabla (\vec{V} \cdot \nabla \mu) - \vec{V}\, \nabla^2 \mu$$
$$+ \nabla \mu \times (\nabla \times \vec{V}) - (\nabla \cdot \vec{V})\, \nabla \mu$$
$$- \nabla \times (\nabla \times \mu \vec{V})$$

where $\vec{F}_{body}$ = sum of body forces (gravity, electromagnetic, etc.)

Equation (3.7.3.1a) is called the Navier-Stokes Equation.

For *one dimension*, x, Equation (3.7.3.1a) reduces to

(Eq. 3.7.3.1b)
$$\rho\left(\frac{\partial V_x}{\partial t} + V_x \frac{\partial V_x}{\partial x} + V_y \frac{\partial V_x}{\partial y} + V_z \frac{\partial V_x}{\partial z}\right)$$
$$= \rho\, (F_{body})_x - \frac{\partial p}{\partial x}$$
$$+ \frac{\mu}{3}\frac{\partial}{\partial x}(\nabla \cdot \vec{V}) + \mu\ \nabla^2 V_x + 2\frac{\partial V_x}{\partial x}\frac{\partial \mu}{\partial x}$$
$$- 2/3\, (\nabla \cdot \vec{V})\frac{\partial \mu}{\partial x}$$
$$+ \left(\frac{\partial V_y}{\partial x} + \frac{\partial V_x}{\partial y}\right)\frac{\partial \mu}{\partial y} + \left(\frac{\partial V_x}{\partial z} + \frac{\partial V_z}{\partial x}\right)\frac{\partial \mu}{\partial z}$$

Similar expressions could be developed for dimensions y and z.

For *non-viscous (frictionless) flow*, Equation (3.7.3.1b) simplifies to Euler's Equation of Motion:

(Eq. 3.7.3.1c)
$$\frac{\partial V_x}{\partial t} + V_x\frac{\partial V_x}{\partial x} + V_y\frac{\partial V_x}{\partial y} + V_z\frac{\partial V_x}{\partial z}$$
$$= (F_{body})_x - \frac{1}{\rho}\frac{dp}{dx}$$

If $(F_{body})_x$ is conservative (i.e., flow is irrotational, or has a potential), by integration along a streamline the equation becomes

(Eq. 3.7.3.1d)

$$\frac{\partial}{\partial t}\int \frac{\vec{V}\cdot d\vec{V}}{g} + \frac{V^2}{2g} + \int \frac{dp}{\rho\left(\frac{g}{g_c}\right)} + Z = \text{constant}$$

where Z = the force potential (e.g., elevation change).

For *steady flow* $\frac{\partial}{\partial t}\int \frac{\vec{V}\cdot d\vec{V}}{g} = 0$, and Equation (3.7.3.1d) becomes the general Bernoulli Equation for steady flow

(Eq. 3.7.3.1e)

$$\frac{V^2}{2g} + \int \frac{dp}{\rho\left(\frac{g}{g_c}\right)} + Z = \text{constant}$$

The generalized form of Bernoulli's Equation could also have been derived from the First Law of Thermodynamics (Reference 141-1).

3.7.3.2 BERNOULLI'S EQUATION. For a perfect gas undergoing an isentropic flow process, Bernoulli's Equation is

(Eq. 3.7.3.2a)

$$\frac{g}{g_c}\left(\frac{V_2^2 - V_1^2}{2g}\right) - \frac{\gamma}{\gamma - 1}\frac{P_1}{\rho_1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right] + \frac{g}{g_c}(Z_2 - Z_1) = 0$$

where V_1, V_2 = upstream and downstream velocities, respectively, ft/sec

γ = ratio of specific heats

P_1, P_2 = upstream and downstream pressures, respectively, lb_f/ft^2

Z_1, Z_2 = elevation above a reference datum, ft

This equation is usually called Bernoulli's Equation for gas flow, but is sometimes called St. Venant's Equation. Using the perfect gas Equations (3.3.8.3d) and (3.3.8.3e) and assuming $Z_1 = Z_2$, Equation (3.7.3.2a) may be rewritten as a function of temperature and specific volume. Thus

(Eq. 3.7.3.2b)

$$\left(\frac{g}{g_c}\right)\frac{V_2^2 - V_1^2}{2g} = \frac{\gamma}{\gamma - 1}\frac{P_1}{\rho_1}\left[1 - \frac{T_2}{T_1}\right]$$

or

(Eq. 3.7.3.2c)

$$\left(\frac{g}{g_c}\right)\frac{V_2^2 - V_1^2}{2g} = \frac{\gamma}{\gamma - 1}\frac{P_1}{\rho_1}\left[1 - \left(\frac{v_1}{v_2}\right)^{\gamma-1}\right]$$

With isentropic flow, when the initial conditions and final pressure remain unaltered, a real gas in comparison with a perfect gas will have (1) a higher final static temperature and acoustic velocity, (2) a lower density, and (3) a higher velocity but a lower Mach number (Reference 141-1).

Assuming a liquid as an incompressible fluid, Bernoulli's Equation becomes

(Eq. 3.7.3.2d)

$$\boxed{\frac{V_2^2 - V_1^2}{2g} + \frac{P_2 - P_1}{w} + Z_2 - Z_1 = 0}$$

where $\frac{V^2}{2g}$ = dynamic pressure or velocity head, ft

$\frac{P}{w}$ = static pressure head, ft

Z = elevation head, elevation above a reference plane, ft

Equation (3.7.3.2d) demonstrates that (1) the total head (total energy) at any particular point in the flow stream above an arbitrary reference plane is equal to the sum of the velocity head, pressure head, and height above the reference plane; (2) with no energy dissipation (e.g., friction) or addition (e.g., heat, pump work), the total energy will be a constant at any point in the fluid; and (3) energy can be transferred to or from the dynamic pressure head, static pressure head, or elevation head without changing the total head or energy within the system. The change in static pressure head due to geometry changes along the pipe line are illustrated by the hydraulic gradient. The change in the sum of the static, dynamic, and elevation head is reflected by the energy gradient. For frictionless flow the total energy gradient, illustrating the total head, coincides with the energy gradient.

Bernoulli's Equation is frequently expanded to include a correction for friction losses and, as necessary, shaft work such as by a pump or turbine. Bernoulli's Equation in extended form becomes

(Eq. 3.7.3.2e)

$$\boxed{\frac{V_2^2}{2g} + \frac{P_2}{w} + Z_2 = \frac{V_1^2}{2g} + \frac{P_1}{w} + Z_1 \pm \frac{g_c}{g}W_s - h_f}$$

where $\frac{V^2}{2g}$ = dynamic pressure velocity head, ft

$\frac{P}{w}$ = static pressure head, ft

Z = elevation head, ft

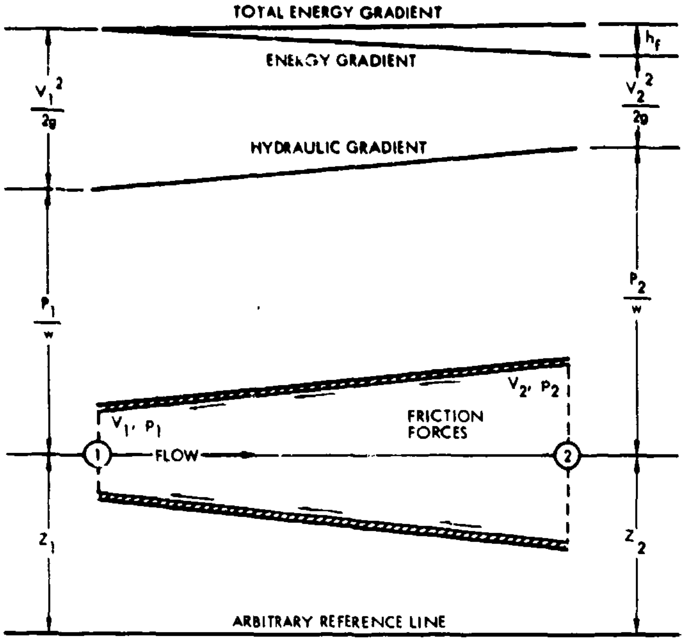
W_s = mechanical shaft work (positive for pump, negative for turbine) $ft\text{-}lb_f/lb_m$

h_f = friction losses = $\sum_{i=1}^{n} K_i \frac{V^2}{2g}$, ft. The K value is a variable dependent on the type of loss.

Sub-Section 3.9 presents losses for selected fittings, pipes, orifices, etc.

$\frac{g_c}{g} = 1\ \mathrm{lb_m/lb_f}$ when $g = 32.2\ \mathrm{ft/sec^2}$.

With friction losses, energy is taken from the system in the form of heat and the temperature of the fluid increases. Available energy (sum of dynamic, static, and elevation heads) decreases in the direction of flow and is denoted in Figure 3.7.3.2 by the energy gradient line. Ideally, if the heat dissipated from friction were reapplied to the system, the total energy at points downstream would be equal to the initial conditions. Hence, the total energy line is a constant.



$$\frac{V_2^2}{2g} + \frac{P_2}{w} = \frac{V_1^2}{2g} + \frac{P_1}{w} - h_f$$

BERNOULLI'S EQUATION WITH FRICTION, $Z_1 = Z_2$

Figure 3.7.3.2. Bernoulli's Equation with Friction

3.7.4 Equations for Gas Flow Processes

The idealized Bernoulli Equation for gas flow cannot approximate all flow processes to acceptable limits. For most gas flows more sophisticated methods are required. Such methods are described under the following Detailed Topics, reprinted with permission from "Thermodynamics of Compressible Fluids," by R. P. Benedict and W. G. Steltz, in *Electro-Technology*, February 1963, Copyright 1963 by C-M Technical Publications Corporation (Reference 111-25).

First, the concepts of continuity and the critical state will be combined to yield equations for a Generalized Flow Process; then, the various flows mentioned above will be analyzed and shown to be special cases of the Generalized Process. A Generalized Compressible-Flow Table will later be developed and presented, and a number of realistic compressible flow problems will be solved using this table.

Let us note that we are considering here the one-dimensional, steady, workless flow processes of a real gas (*i.e.*, one having viscosity and thermal conductivity) whose equation of state, however, is well represented by the ideal gas relation

$$\frac{p}{\rho} = gRT \tag{1}$$

and whose specific heat capacity at constant pressure c_p is taken as a constant.

Continuity

The conservation-of-mass concept, which relates one thermodynamic state to another for the whole gamut of possible one-dimensional flow processes, is expressed as

$$\rho VA = \text{a constant} \tag{2}$$

Since a dimensionless form of Eq (2) will be more useful, the density and velocity are next evaluated in terms of the total parameters ρ_t, p_t and T_t, which are based on isentropic stagnation† processes from given static states. Thus, the density term may be written as

$$\rho = \rho_t\left(\frac{p}{p_t}\right)^{1/\gamma} \tag{3}$$

upon recalling that p/ρ^γ is a constant for any isentropic change of state of a gas defined by Eq (1). The velocity term also can be written in terms of total and static states by combining the general energy equation (on the basis of a pound of flowing fluid)

$$\delta Q = dh + \frac{VdV}{g} + dZ \tag{4}$$

with the thermodynamic identity

$$Tds = dh - \frac{dp}{\rho g} \tag{5}$$

(Note that $\delta Q = 0$ and $ds = 0$ in the isentropic stagnation process, while dZ is negligible in any gaseous flow process.) There results

$$VdV = -\frac{dp}{\rho} \tag{6}$$

On integrating Eq (6) between static and total states, we obtain

† Adapted from the authors' paper, "A Generalized Approach to One-Dimensional Gas Dynamics," *Journal of Engineering for Power*, ASME, January 1962.

‡ By the term *isentropic*, we imply the special constant entropy process which is both adiabatic and reversible; by *stagnation*, a deceleration (actual or postulated) of the fluid to zero velocity.

$$V = \left[\frac{2\gamma}{\gamma - 1}\left(\frac{p_t}{\rho_t} - \frac{p}{\rho}\right)\right]^{\frac{1}{2}} \tag{7}$$

When Eqs (3) and (7) are combined according to Eq (2), there results

$$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left\{\left[\frac{p_1}{p_{t1}}\right]^{\frac{1}{\gamma}}\left[1 - \left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} = \left\{\left[\frac{p_2}{p_{t2}}\right]^{\frac{1}{\gamma}}\left[1 - \left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} \tag{8}$$

where Eq (8) is an entirely general dimensionless continuity expression.

The Critical State

Now, directing our attention elsewhere, we note that for every thermodynamic process there exists a critical state in which, for a differential change along the process line, $\delta F = 0$ and $dA = 0$. By this we mean that the entropy change, generally given by the Second Law of Thermodynamics as

$$ds = \frac{\delta Q}{T} + \frac{\delta F}{T} \tag{9}$$

always reduces to

$$ds^* = \frac{\delta Q}{T} \tag{10}$$

at the critical state. [The asterisk, here and henceforth, denotes variables pertaining to the critical state.]

Expressions for velocity and pressure ratio at this critical state will prove useful, and are developed next. A general differential form for the critical velocity is derived by combining general energy, Eq (4), with the First Law of Thermodynamics, generally given by

$$\delta Q + \delta F = du + p dv \tag{11}$$

on a per-pound basis. Recalling that $\delta F = 0$ at the critical state, then

$$V^* dV = -\frac{dp}{\rho} \tag{12}$$

While Eq (12) is similar in form to Eq (6), its implications are quite different. Equation (12) holds for *any* process at the critical state, while Eq (6) is restricted to a *frictionless* process. Recalling that $dA = 0$ at the critical state, Eq (2) can be written as

$$\rho dV + V^* d\rho = 0 \tag{13}$$

While Eq (13) is similar in form to a constant-area differential form of continuity, the implications are quite different. Equation (13) holds for any process at the critical state. Combining Eqs (12) and (13), we obtain

$$V^* = \left(\frac{dp}{d\rho}\right)^{\frac{1}{2}} \tag{14}$$

which, by employing the equation of state, Eq (1), and the thermodynamic identity, Eq (5), can also be written as

$$V^* = \left[\frac{p}{\rho}\left(\frac{1}{\frac{1}{\gamma} - \frac{ds}{dp}\frac{p}{c_p}}\right)\right]^{\frac{1}{2}} \tag{15}$$

Equation (15) defines the critical velocity for any flow process. By equating the expressions for general velocity and for critical velocity, Eqs (7) and (15), we obtain

$$\frac{p^*}{p_t^*} = \left(\frac{\frac{1}{\gamma} - \frac{ds}{dp}\frac{p}{c_p}}{\frac{1}{\gamma} - \frac{ds}{dp}\frac{p}{c_p} + \frac{\gamma - 1}{2\gamma}}\right)^{\frac{\gamma}{\gamma-1}} \tag{16}$$

Equation (16) defines the critical pressure ratio for any flow process.

The Gamma Function

The factors $(T_{t2}/T_{t1})^{1/2}$, p_{t1}/p_{t2}, and A_1/A_2 of Eq (8) can be considered to be arbitrary process multipliers of the inlet pressure-ratio function

$$P_1 = \left\{\left[\frac{p_1}{p_{t1}}\right]^{\frac{1}{\gamma}}\left[1 - \left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} \tag{17}$$

By referring the P_1 of Eq (17) to a similar function

$$P^* = \left\{\left[\frac{p^*}{p_t^*}\right]^{\frac{1}{\gamma}}\left[1 - \left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} \tag{18}$$

pertaining to the critical state, we can rewrite Eq (8) as

$$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2 \tag{19}$$

The symbol Γ, which equals P/P^*, represents a *generalized compressible flow function* since it embodies the concepts of continuity and the critical state. By maximizing the function P of Eq (17), we obtain the familiar relationship

$$\left.\frac{p}{p_t}\right|_{P_{max}} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}} \tag{20}$$

When Eq (18) is evaluated in terms of Eq (20), the Γ function can be uniquely defined as

$$\Gamma = \left\{\frac{\left[\frac{p}{p_t}\right]^{\frac{1}{\gamma}}\left[1 - \left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma - 1}{\gamma + 1}\right)^{\frac{1}{2}}}\right\} \tag{21}$$

where Γ varies between 0 and 1 only, for any and all flow processes. By plotting pressure ratio p/p_t vs Γ for particular values of γ, we obtain the *generalized compressible-flow chart* of Fig. 1. When the function Γ_1 (completely defined by the inlet pressure ratio p_1/p_{t1} for a specific value of γ) is multiplied by the total temperature, total pressure, and area ratios pertaining to a particular flow process, the function Γ_2 completely defining the exit conditions is obtained as indicated by Eq (19), the referred continuity expression.

Since the condition chosen to particularize the Γ function, Eq (20), need not be the critical condition of an arbitrary process, it follows that Γ^* need not always equal 1. Nevertheless, all arbitrary flow processes are precisely represented by the plot of Fig. 1.

Other parameters which advantageously can be referred to critical-state conditions are the total pressure, the total temperature, and the area. Thus,

$$\frac{p_t}{p_t^*} = \left\{ \frac{\left[\frac{p^*}{p_t^*}\right]^{\frac{1}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left[\frac{T_t}{T_t^*}\right]^{\frac{1}{2}}}{\left[\frac{p}{p_t}\right]^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left[\frac{A}{A^*}\right]} \right\} \tag{22}$$

$$\frac{T_t}{T_t^*} = \left\{ \frac{\left[\frac{p}{p_t}\right]^{\frac{2}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]}{\left[\frac{p^*}{p_t^*}\right]^{\frac{2}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]}\left[\frac{A}{A^*}\right]^2\left[\frac{p_t}{p_t^*}\right]^2 \right\} \tag{23}$$

and

$$\frac{A}{A^*} = \left\{ \frac{\left[\frac{p^*}{p_t^*}\right]^{\frac{1}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left[\frac{T_t}{T_t^*}\right]^{\frac{1}{2}}}{\left[\frac{p}{p_t}\right]^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left[\frac{p_t}{p_t^*}\right]} \right\} \tag{24}$$

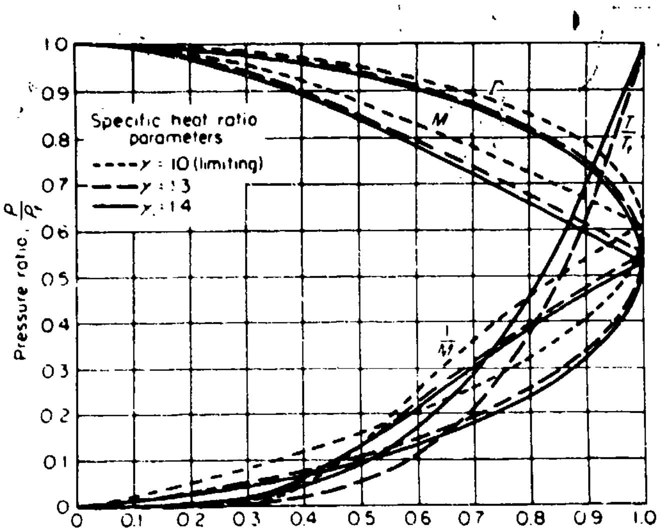


Fig. 1 Generalized compressible-flow chart.

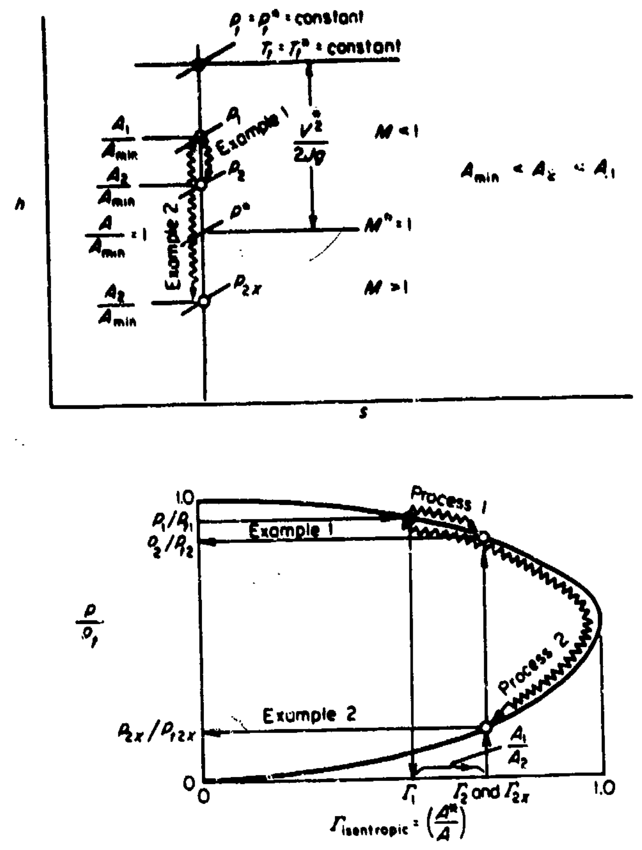


Fig. 2 --Isentropic-process curves.

are all consequences of Eq (8), the generalized continuity expression.

Finally, a relation expressing the entropy change from an arbitrary state to the critical state will prove useful, and is given as

$$s^* - s = 2c_p \ln \left\{ \frac{\left[\frac{p^*}{p_t^*}\right]^{\frac{1}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left[\frac{p}{p_t}\right]^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left[\frac{A}{A^*}\right]\left[\frac{p_t}{p_t^*}\right]^{\frac{\gamma+1}{2\gamma}}} \right\} \tag{25}$$

We have seen thus far that:

1. A dimensionless form of continuity, Eq (8), can be applied to all flow processes.

2. A thermodynamic critical state exists for all flow processes.

3. A generalized compressible flow function, Γ, is obtained by referring continuity to certain conditions at the critical state, Eq (20).

4. The Γ function so defined varies only between 0 and 1.

Application of Gamma in Specific Flow Processes

Many practical situations may be approximated by certain processes having restrictions which make possible simplified solutions. Several such flow processes of most general interest will be treated here.

• **Adiabatic Flow without Friction.** Here we are referring to the one-dimensional, steady flow of a fluid whose thermodynamic properties change solely because of area variations *ie.*, the total temperature and the total pressure remain constant throughout the process. This is called isentropic flow. Referred continuity, Eq (19), reduces to

$$\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2 \tag{26}$$

Since it is also true that

$$\left(\frac{A_1}{A_2}\right)\left(\frac{A^*}{A_1}\right) = \left(\frac{A^*}{A_2}\right) \tag{27}$$

it follows that

$$\Gamma_{\text{isentropic}} = \frac{A^*}{A} \tag{28}$$

Schematic h-s and p/p_t-Γ curves are given in Fig. 2, where subsonic and supersonic processes are indicated.

Thus, the Γ function serves well in isentropic flow processes, and the loci of all possible state points in any isentropic process are precisely represented by the single p/p_t-Γ curve. Of course, isentropic flows are quite simple to treat (note that the differential change in entropy at the critical state is zero, as it is throughout any isentropic process), and solutions of this type have been known for years. Nevertheless, even in more complex situations, this same procedure will be seen to be valid, and herein lies the generality of the Γ function.

• **Adiabatic Flow with Friction.** Here we are referring to the one-dimensional, steady flow of fluid, whose thermodynamic properties change solely because of viscous effects and area variations. In the constant-area case, this type of flow is called *Fanno flow*, after an early worker in this field. The total temperature remains constant throughout the process while the viscous effects are reflected in a decrease in total pressure. However, the more general variable-area Fanno-type flow can always be treated as well. Referred continuity, Eq (19), for Fanno-type flow becomes

$$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2 \tag{29}$$

Since it is also true that

$$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left[\left(\frac{p_t^*}{p_{t1}}\right)\left(\frac{A^*}{A_1}\right)\right] = \left[\left(\frac{p_t^*}{p_{t2}}\right)\left(\frac{A^*}{A_2}\right)\right] \tag{30}$$

it follows that

$$\Gamma_{\text{Fanno-type}} = \left(\frac{p_t^*}{p_t}\right)\left(\frac{A^*}{A}\right) \tag{31}$$

Schematic h-s and p/p_t-Γ curves are given in Fig. 3, where subsonic and supersonic processes are indicated. The numerical evaluation of $\Gamma_{\text{Fanno-type}}$ and $\Gamma_{\text{isentropic}}$ as functions of p/p_t will be discussed, and practical examples closely following these illustrative diagrams will be given later.

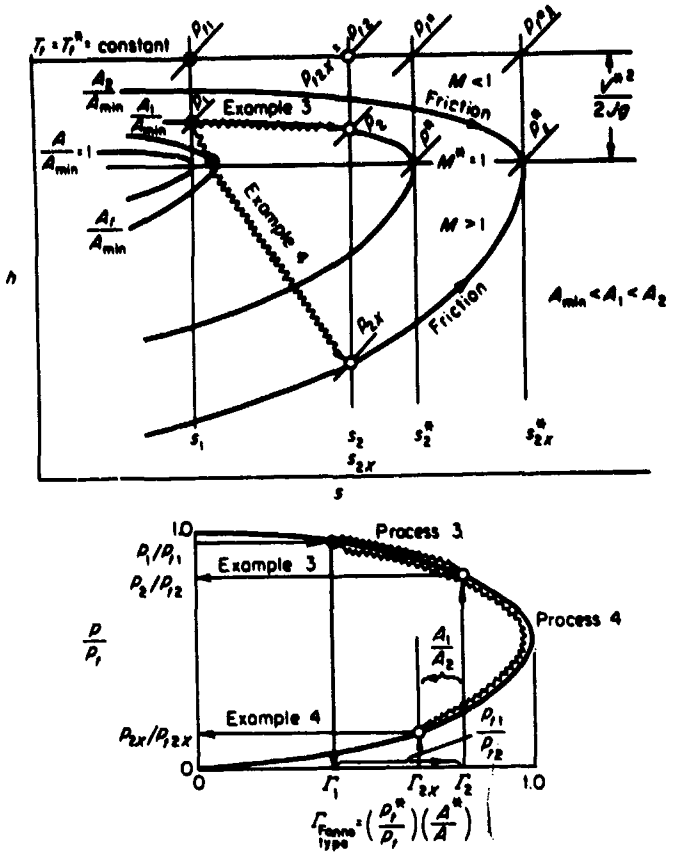


Fig. 3—Fanno-process curves.

Note that the constant-area Fanno plots on the h-s diagram can be interpreted in terms of the more general variable-area Fanno-type flow simply by considering the area effect as an isentropic change of state to a new area ratio line, and then proceeding along the new constant-area Fanno line passing through this revised initial state to the final state.

Compressible-flow Fanno lines are not straight lines on an h-s plot; thus, we note that if a frictional process were to proceed far enough, changes of state would occur more rapidly than changes of entropy until a point was reached along each Fanno line where there was no further increase in entropy (*i.e.*, $ds = 0$ at this critical state).

• **Diabatic Flow without Friction.** Here, we are referring to the one-dimensional, steady flow of a fluid whose thermodynamic properties change solely because of heat-transfer effects and area variations. In the constant-area case, this type of flow is called *Rayleigh flow*. Here, the total temperature, and therefore the total pressure, change because of the allowed heat transfer. However, by the general energy equation, Eq (4), and the First Law, Eq (11), we have as a conserved quantity $p + \rho V^2$, conventionally called the *thrust* or *impulse function*. Referred continuity, Eq (19), for Rayleigh flow must be written as

$$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2 \tag{32}$$

This constant-area Rayleigh flow can always be treated once the total temperature ratio is specified, for the total pressure ratio is a function of T_{t1}/T_{t2} only. It follows that

$$\Gamma_{\text{Rayleigh}} = \left(\frac{T_t}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_t}\right) \tag{33}$$

The more general variable-area Rayleigh-type flow may sometimes be treated, but no completely general solution is possible. While we can always specify the total temperature ratio, we do not necessarily know the total pressure ratio. In the variable-area case, p_{t1}/p_{t2} depends on the definite physical conditions under which the heat transfer occurs. Only when the heat transfer takes place at specified areas can a variable-area Rayleigh-type flow be treated. With this restriction, the problem reduces to a combination of separate variable-area isentropic processes and constant-area Rayleigh processes. Referred continuity for this semi-generalized Rayleigh-type flow is, of course,

$$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2 \tag{34}$$

Schematic h-s and p/p_t-Γ curves are given in Fig. 4, where subsonic and supersonic processes are indicated.

Note that compressible-flow Rayleigh lines are not straight lines on an h-s plot; thus, if a heating process were to proceed far enough, a point of maximum enthalpy would be reached. If the heating process were to proceed further, the change in enthalpy would actually become negative. As in the Fanno case, changes of state would then occur more rapidly than changes of entropy until a point was reached along each Rayleigh line where there was no further increase in entropy (*i.e.*, $ds = 0$ at this critical state).

• Diabatic Flow with Friction (Isothermal). Here we are referring to the one-dimensional, steady flow of a fluid whose thermodynamic properties change solely because of heat-transfer effects, viscous effects and area variations, in such a manner that the static temperature remains constant. Since neither the total temperature nor the total pressure remain constant during such a process, referred continuity for the constant-area isothermal case is the same as for Rayleigh flow *i.e.*, Eq (32). It can be shown that

$$\Gamma_{\text{isothermal}} = \left(\frac{T_t}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_t}\right)\left[\gamma^{\frac{1}{\gamma-1}}\left(\frac{\gamma+1}{3\gamma-1}\right)^{\frac{\gamma+1}{2(\gamma-1)}}\right] \tag{35}$$

No general solution is possible for variable-area isothermal flow. Remarks similar to those under Rayleigh flow apply here also.

Schematic h-s and p/p_t-Γ curves are given in Fig. 5, where subsonic and supersonic processes are indicated. The numerical evaluation of Γ_{Rayleigh} and $\Gamma_{\text{isothermal}}$ as functions of

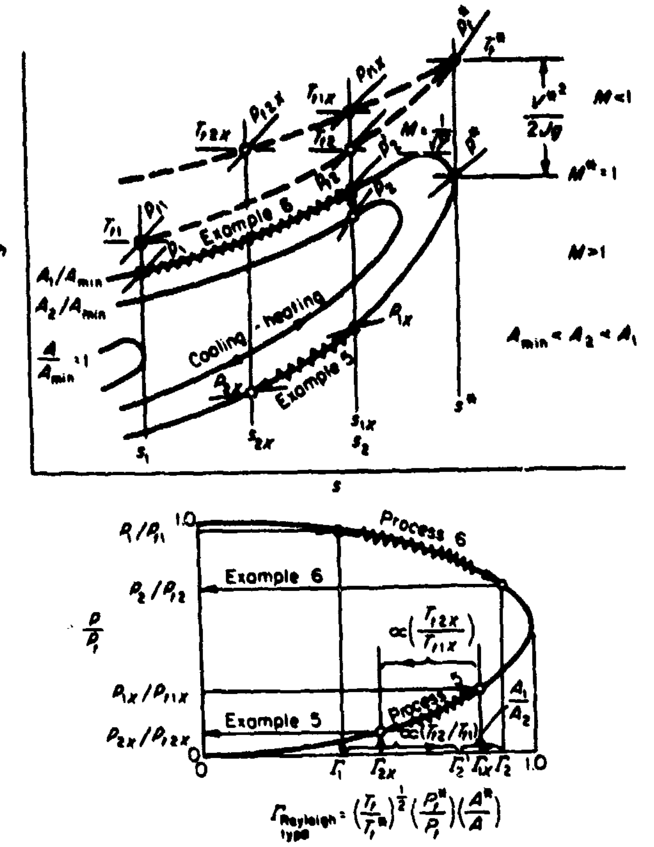


Fig. 4 Rayleigh-process curves.

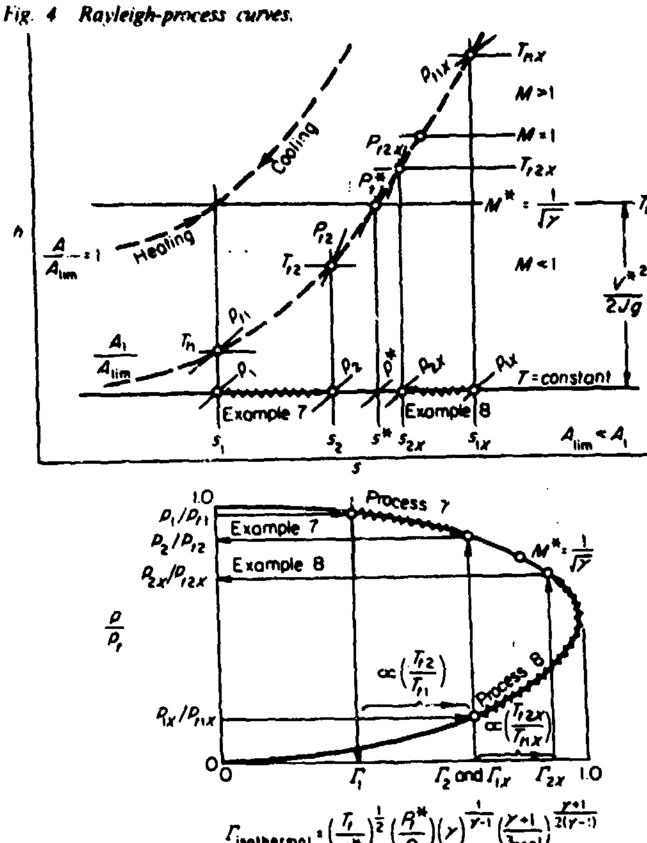


Fig. 5 -Isothermal-process curves.

p/p_t will be discussed, and practical examples closely following these illustrative diagrams will be given later.

Note that compressible-flow isothermal lines are straight lines on an h-s plot; as the combined frictional-heat-transfer process proceeds, changes of state occur in step with changes of entropy. Hence, we should not expect the differential entropy change to be zero at the thermodynamic critical state; indeed, applying the Second Law, Eq (9), and the equation of state, Eq (1), we have

$$ds^*_{\text{isothermal}} = \frac{\delta Q}{T} = -\frac{R\,dp}{p} \tag{36}$$

• **Normal Shock.** Here we are referring to the one-dimensional, steady, supersonic flow of a fluid whose thermodynamic properties change suddenly to those corresponding to subsonic flow, with no change in total temperature, area, or thrust function. The first two conditions (T_t = a constant and A = a constant) ensure that the final state will be on the Fanno locus passing through the initial state; the last two conditions (A = a constant and $p + \rho V^2$ = a constant) ensure that the final state will be on the Rayleigh locus passing through the initial state. Evidently, the final state of the normal shock must lie at the intersection of these Fanno and Rayleigh lines. But the intersection of these two flow processes always occurs at an entropy greater than the initial value. Hence, the total pressure must always decrease across the normal shock. Referred continuity for the normal shock process reduces to

$$\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2 \tag{37}$$

which is of the same form as that for Fanno flow. However, in the case of normal shock, the multiplier of Γ_1 can be defined explicitly in terms of the (supersonic) inlet pressure ratio only—*i.e.*,

$$\left(\frac{p_{t1}}{p_{t2}}\right)_{\substack{\text{normal}\\\text{shock}}} = \left\{\frac{4\gamma\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]}{(\gamma+1)(\gamma-1)\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}} - \left(\frac{\gamma-1}{\gamma+1}\right)\right\}^{\frac{1}{\gamma-1}} \left\{\frac{(\gamma-1)}{(\gamma+1)\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]}\right\}^{\frac{\gamma}{\gamma-1}} \tag{38}$$

Schematic h-s and p/p_t-Γ curves are given in Fig. 6, where a discontinuous normal-shock process is indicated. A numerical example closely following these illustrative diagrams is given later.

In Fig. 7, we compare all of these simplified flow processes, indicating accessible states along both static and total loci, as well as those states which are unattainable from the given (supersonic) inlet state.

The general equations, as well as equations for all the processes dealt with in this article, are summarized in terms of pressure ratio in Table I, and in terms of Mach number in Table II. (Both these tables are on the following pages.)

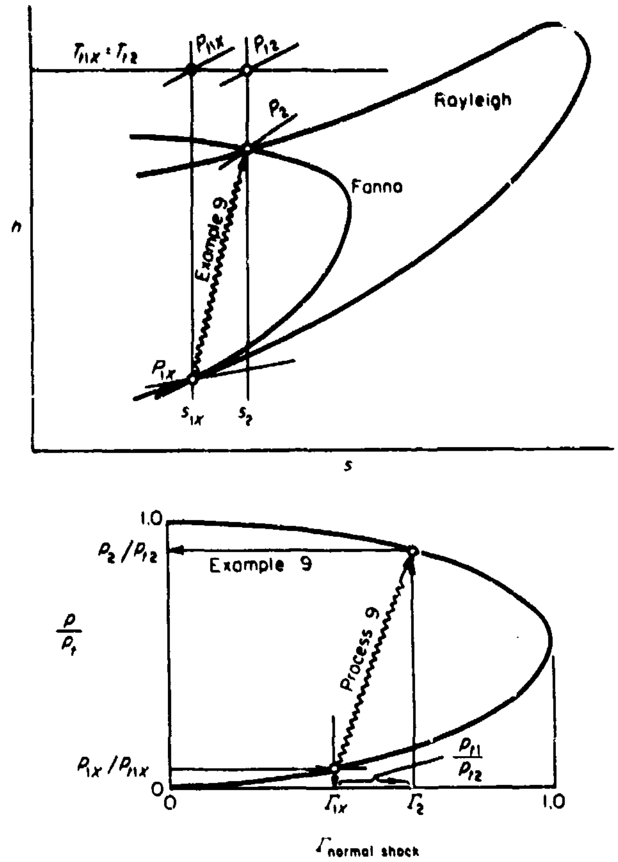


Fig. 6—Normal-shock-process curves.

The Gamma Function for Numerical Solutions

We have seen that a *generalized compressible-flow function*, Γ, may be defined and that it has significance in any arbitrary, one-dimensional, workless flow process. The individual processes of most general interest have been discussed in some detail and the role of the Γ function in describing and facilitating the solution of these typical processes has been presented. Of more practical use would be a convenient means of obtaining numerical answers to the problems at hand in a manner that would utilize the generalized compressible-flow function, Γ.

We find, in the published literature and texts concerned with the thermodynamics of moving fluids, separate and distinct analyses and problem-solution methods. For example, one finds chapters on isentropic, adiabatic and diabatic flows, as well as the normal-shock process, while the isothermal process is quite often ignored. It would therefore be quite valuable to have a means available for the solution of each of these basic processes in a similar manner, using similar procedures, and (if possible) a common reference scheme (*e.g.*, a table or plot). We shall now proceed to

Table I—Summary of General and Specific Equations in Terms of Pressure Ratio

Parameter \ Process	General	Adiabatic flow without friction (isentropic flow with variable area)
Restrictions		T_t = constant, p_t = constant
Continuity	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left\{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} = \left\{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\}$	$\left(\frac{A_1}{A_2}\right)\left\{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} = \left\{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\}$
Referred continuity	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left\{\frac{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\} = \left\{\frac{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\}$	$\left(\frac{A_1}{A_2}\right)\left\{\frac{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\} = \left\{\frac{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\}$
Referred continuity in terms of the generalized compressible flow function Γ	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2$	$\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2$
Critical velocity V^*	$\left\{\left[\frac{p}{\rho}\,\frac{1}{\frac{1}{\gamma}-\frac{ds}{dp}\,\frac{p}{c_p}}\right]\right\}^{\frac{1}{2}}$	$\left(\frac{\gamma p}{\rho}\right)^{\frac{1}{2}}$
Critical pressure ratio p^*/p_t^* (at critical velocity)	$\left[\frac{\frac{1}{\gamma}-\frac{ds}{dp}\,\frac{p}{c_p}}{\frac{1}{\gamma}-\frac{ds}{dp}\,\frac{p}{c_p}+\frac{\gamma-1}{2\gamma}}\right]^{\frac{\gamma}{\gamma-1}}$	$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$
Limiting entropy change s^*-s (to critical velocity)	$2c_p\ln\left[\frac{\left(\frac{p^*}{p_t^*}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{A}{A^*}\right)\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{p_t}{p_t^*}\right)^{\frac{\gamma+1}{2\gamma}}}\right]$	0
Referred total temperature T_t/T_t^* (at critical velocity)	$\frac{\left(\frac{p}{p_t}\right)^{\frac{2}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\left(\frac{A}{A^*}\right)^2\left(\frac{p_t}{p_t^*}\right)^2}{\left(\frac{p^*}{p_t^*}\right)^{\frac{2}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]}$	1
Referred total pressure p_t/p_t^* (at critical velocity)	$\frac{\left(\frac{p^*}{p_t^*}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{T_t}{T_t^*}\right)^{\frac{1}{2}}}{\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{A}{A^*}\right)}$	1
Referred area A/A^* (to critical velocity)	$\frac{\left(\frac{p^*}{p_t^*}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p^*}{p_t^*}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{T_t}{T_t^*}\right)^{\frac{1}{2}}}{\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{p_t}{p_t^*}\right)}$	$\frac{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}{\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}$

Adiabatic flow with friction (Fanno-type flow with variable area)	Diabatic flow without friction (Rayleigh flow constant area)	Diabatic flow with friction (isothermal flow constant area)
T_t = constant, $A^* = A_2$	$p + \rho v^2$ = constant, $A^* = A_1$	T = constant, $A^* = A_1$
$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left\{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} = \left\{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\}$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left\{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} = \left\{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\}$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left\{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\} = \left\{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\right\}$
$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left\{\frac{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\} = \left\{\frac{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\}$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left\{\frac{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\} = \left\{\frac{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\}$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left\{\frac{\left(\frac{p_1}{p_{t1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_1}{p_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\} = \left\{\frac{\left(\frac{p_2}{p_{t2}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p_2}{p_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}\right\}$
$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2$
$\left(\frac{\gamma p}{\rho}\right)^{\frac{1}{2}}$	$\left(\frac{\gamma p}{\rho}\right)^{\frac{1}{2}}$	$\left(\frac{p}{\rho}\right)^{\frac{1}{2}}$
$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$	$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$	$\left(\frac{2\gamma}{3\gamma-1}\right)^{\frac{\gamma}{\gamma-1}}$
$2c_p\ln\left[\frac{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{2\gamma}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{\gamma-1}{4\gamma}}}{\left(\frac{A}{A_2}\right)^{\frac{\gamma-1}{2\gamma}}\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{2\gamma^2}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{\gamma-1}{4\gamma}}}\right]$	$2c_p\ln\left[\frac{\left\{\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}+\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{\frac{\gamma+1}{2\gamma}}}{\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{2\gamma^2}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{2}{\gamma-1}\right)^{\frac{1}{2}}\left(\gamma+1\right)^{\frac{\gamma+1}{2\gamma}}}\right]$	$2c_p\ln\left[\frac{\left\{\left(\frac{p}{p_t}\right)^{\frac{\gamma+1}{\gamma}}\left(\frac{2\gamma}{\gamma-1}\right)\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{\frac{\gamma+1}{4\gamma}}}{\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{2\gamma}{\gamma-1}\right)^{\frac{1}{2}}}\right]$
1	$\frac{4\left(\frac{\gamma+1}{\gamma-1}\right)\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]}{\left\{\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}+\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{2}}$	$\left(\frac{2\gamma}{3\gamma-1}\right)\left(\frac{p}{p_t}\right)^{\frac{1-\gamma}{\gamma}}$
$\frac{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{\gamma-1}{\gamma+1}\right)^{\frac{1}{2}}}{\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}}\left(\frac{A}{A_2}\right)}$	$\frac{2\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}}{\left(\frac{p}{p_t}\right)^{\frac{1}{\gamma}}\left\{\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}+\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}}$	$\frac{\left(\frac{2\gamma}{3\gamma-1}\right)^{\frac{\gamma}{\gamma-1}}}{\left\{\left(\frac{p}{p_t}\right)^{\frac{\gamma+1}{\gamma}}\left(\frac{2\gamma}{\gamma-1}\right)\left[1-\left(\frac{p}{p_t}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{\frac{1}{2}}}$
$\frac{A}{A_2}$	1	1

Table II—Summary of General and Specific Equations in Terms of Mach Number

Parameter \ Process	General	Adiabatic flow without friction (isentropic flow with variable area)
Restrictions		T_t = constant, p_t = constant
Continuity	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left[\frac{M_1\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_2^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$	$\left(\frac{A_1}{A_2}\right)\left[\frac{M_1\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_2^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$
Referred Continuity	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left[\frac{M_1}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_1^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_2^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$	$\left(\frac{A_1}{A_2}\right)\left[\frac{M_1}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_1^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_2^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$
Referred continuity in terms of the generalized compressible flow function Γ	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2$	$\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2$
Critical Mach number M^*	$\left[\frac{1}{1-\frac{ds}{dp}\frac{\gamma p}{c_p}}\right]^{\frac{1}{2}}$	1
Critical pressure ratio p^*/p_t^* (at critical Mach number)	$\left[\frac{\frac{1}{\gamma}-\frac{ds}{dp}\frac{p}{c_p}}{\frac{1}{\gamma}-\frac{ds}{dp}\frac{p}{c_p}+\frac{\gamma-1}{2\gamma}}\right]^{\frac{\gamma}{\gamma-1}}$	$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$
Limiting entropy change s^*-s (to critical Mach number)	$2c_p\ln\left[\frac{M^*\left(1+\frac{\gamma-1}{2}M^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}{\left(\frac{A}{A^*}\right)M\left(1+\frac{\gamma-1}{2}M^{*2}\right)^{\frac{\gamma+1}{2(\gamma-1)}}\left(\frac{p_t}{p_t^*}\right)^{\frac{\gamma+1}{2\gamma}}}\right]$	0
Referred total temperature T_t/T_t^* (at critical Mach number)	$\frac{M^2}{M^{*2}}\left[\frac{1+\frac{\gamma-1}{2}M^{*2}}{1+\frac{\gamma-1}{2}M^2}\right]^{\frac{\gamma+1}{\gamma-1}}\left(\frac{p_t}{p_t^*}\right)^2\left(\frac{A}{A^*}\right)^2$	1
Referred total pressure p_t/p_t^* (at critical Mach number)	$\frac{M^*}{M}\left[\frac{1+\frac{\gamma-1}{2}M^2}{1+\frac{\gamma-1}{2}M^{*2}}\right]^{\frac{\gamma+1}{2(\gamma-1)}}\frac{\left(\frac{T_t}{T_t^*}\right)^{\frac{1}{2}}}{\left(\frac{A}{A^*}\right)}$	1
Referred area A/A^* (to critical Mach number)	$\frac{M^*}{M}\left[\frac{1+\frac{\gamma-1}{2}M^2}{1+\frac{\gamma-1}{2}M^{*2}}\right]^{\frac{\gamma+1}{2(\gamma-1)}}\frac{\left(\frac{T_t}{T_t^*}\right)^{\frac{1}{2}}}{\left(\frac{p_t}{p_t^*}\right)}$	$\frac{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}{M}$

Adiabatic flow with friction (Fanno-type flow with variable area)	Diabatic flow without friction (Rayleigh flow constant area)	Diabatic flow with friction (isothermal flow constant area)
T_t = constant, $A^* = A_2$	$p + \rho v^2$ = constant, $A^* = A_1$	T = constant, $A^* = A_1$
$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left[\frac{M_1\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_2^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left[\frac{M_1\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_2^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left[\frac{M_1\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2\left(\frac{\gamma-1}{2}\right)^{\frac{1}{2}}}{\left(1+\frac{\gamma-1}{2}M_2^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$
$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\left[\frac{M_1}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_1^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_2^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left[\frac{M_1}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_1^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_2^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$	$\left(\frac{T_{t1}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\left[\frac{M_1}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_1^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right] = \left[\frac{M_2}{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M_2^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}\right]$
$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2$	$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2$
1	1	$\left(\frac{1}{\gamma}\right)^{\frac{1}{2}}$
$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$	$\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$	$\left(\frac{2\gamma}{3\gamma-1}\right)^{\frac{\gamma}{\gamma-1}}$
$2c_p \ln\left[\frac{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M^2\right)\right]^{\frac{\gamma+1}{4\gamma}}}{\left(\frac{A}{A_2}\right)^{\frac{\gamma-1}{2\gamma}} M^{\frac{\gamma-1}{2\gamma}}}\right]$	$2c_p \ln\left[\frac{\left[\frac{1}{\gamma+1}(1+\gamma M^2)\right]^{\frac{\gamma+1}{2\gamma}}}{M}\right]$	$2c_p \ln\left[(\gamma M^2)^{\frac{1-\gamma}{4\gamma}}\right]$
1	$\frac{2(\gamma+1)M^2\left(1+\frac{\gamma-1}{2}M^2\right)}{(1+\gamma M^2)^2}$	$\left(\frac{2\gamma}{3\gamma-1}\right)\left(1+\frac{\gamma-1}{2}M^2\right)$
$\frac{\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}}}{M\left(\frac{A}{A_2}\right)}$	$\frac{(\gamma+1)\left[\frac{2}{\gamma+1}\left(1+\frac{\gamma-1}{2}M^2\right)\right]^{\frac{\gamma}{\gamma-1}}}{(1+\gamma M^2)}$	$\frac{\left[\frac{2\gamma}{3\gamma-1}\left(1+\frac{\gamma-1}{2}M^2\right)\right]^{\frac{\gamma}{\gamma-1}}}{M\gamma^{\frac{1}{2}}}$
$\frac{A_1}{A_2}$	1	1

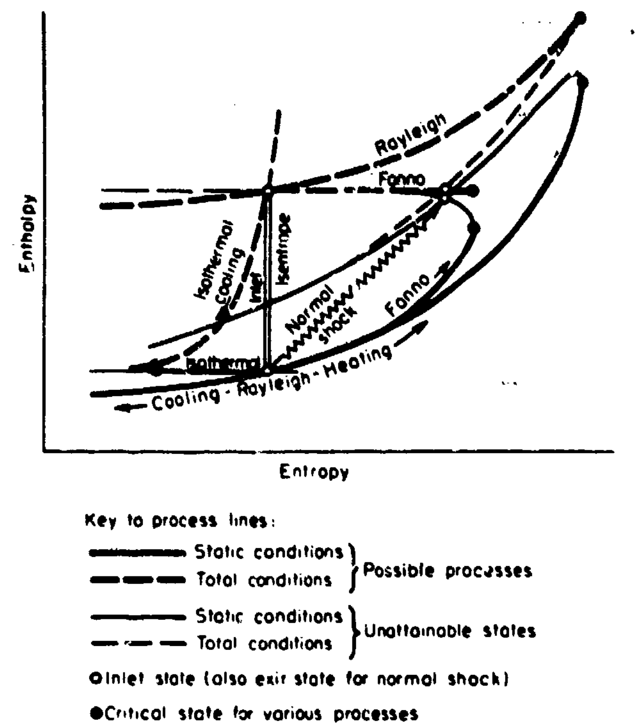


Fig. 7 - Comparison of various constant-area processes for a supersonic-inlet state.

develop a generalized compressible-flow table which provides the means to satisfy these requirements.

The Γ function has been carried through in its development as a function of pressure ratio, p/p_t; it is then just one step to convert this independent variable to Mach number, M†, as they are both state functions and intimately related. Hence, we now have a choice of two independent variables for our generalized compressible flow table. Upon further examination of the Γ function, Eq (21), it is seen to be also a function (however weak) of the ratio of specific heats, γ, and this forms one other independent variable to consider. As we can usually consider γ to be constant over the range of thermodynamic variables for a particular problem, we may compile our table for various constant values of γ, essentially reserving a particular value for a particular gas.

The question arises as to what to include in our table. Of those physical variables such as pressure, temperature and density, the first two cited are the most easily measurable and are logical as candidates for the table. We have seen that p/p_t will be a prime independent variable; thus, we shall include temperature, in the form of T/T_t, as one of our table entries. The density may easily be derived, if desired, from Eq (1), once the pressure and temperature are defined. Let us now consider the various processes and determine which parameters will be necessary to complete the generalized compressible-flow table.

† M = the ratio of directed velocity to acoustic velocity, analytically

$$M = \left\{\left[\left(\frac{p}{p_t}\right)^{\frac{1-\gamma}{\gamma}} - 1\right]\frac{2}{\gamma - 1}\right\}^{\frac{1}{2}}$$

● **Adiabatic Flow without Friction.** The flow characteristics of this process change solely because of area variations as the total temperature and total pressure remain constant; hence, referred continuity, Eq (19), reduces to

$$\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2 \tag{26}$$

It has been mentioned that $\Gamma_{\text{isentropic}} = A^*/A$; it may also be easily shown that $\Gamma = (W/\Delta t)/(W/\Delta t)^*_{\text{isentropic}}$. As our means of isentropic-flow solution requires only the Γ function, we need only this column in the generalized compressible-flow table. One simply modifies Γ_1 by the isentropic multiplier A_1/A_2 to obtain Γ_2 (see Examples 1 and 2, to follow).

● **Adiabatic Flow with Friction.** In this flow process we consider changes in fluid properties to be effected by variations in area and total pressure; total temperature remains invariant. Referred continuity, Eq (19), for the more general Fanno-type process reduces to

$$\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right)\Gamma_1 = \Gamma_2 \tag{29}$$

Hence, when developing a generalized compressible-flow table by which Fanno-type flows may be solved, we again need only the Γ column, since area ratio and total-pressure ratio may always be specified; no other special column is required. One simply modifies Γ_1 by the Fanno multiplier, $(p_{t1}/p_{t2})(A_1/A_2)$, to obtain Γ_2 (see Examples 3 and 4, to follow).

● **Diabatic Flow without Friction.** Here, we consider changes in thermodynamic properties to be effected solely by heat transfer and (under certain conditions) area variation. For the Rayleigh case (constant area), total temperature and total pressure do not remain constant and the referred-continuity relation, Eq (19), is written

$$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2 \tag{32}$$

As the total-pressure and total-temperature ratios are related by an implicit function, we need specify only one, the other then being immediately defined. Hence, when developing a generalized compressible-flow table by which Rayleigh flows may be solved, we will naturally make use of the Γ column. But, in addition, we must include a special column tabulating either T_t/T_t^* or p_t^*/p_t to define the process completely, since the two ratios making up the Rayleigh multiplier $(T_{t2}/T_{t1})^{1/2}(p_{t1}/p_{t2})$, while not independent variables, are yet not explicit functions one of the other, as previously mentioned. Choosing the referred temperature as the more significant parameter in Rayleigh flow (which is predominately a heat-transfer problem), one simply modifies $(T_{t1}/T_t^*)_{\text{Rayleigh}}$ by the multiplier T_{t2}/T_{t1} to obtain $(T_{t2}/T_t^*)_{\text{Rayleigh}}$. Since $(T_{t2}/T_t^*)_{\text{Rayleigh}}$ is a function only of the exit-pressure ratio, it must necessarily appear in the same row as the Γ_2 function in the table. Note that Γ_2/Γ_1, when divided by $(T_{t2}/T_{t1})^{1/2}$, yields the unspecified total-pressure ratio p_{t1}/p_{t2}.

The more general variable-area Rayleigh-type flow process may be treated if the heat transfer takes place at certain specified areas. The problem then reduces to a series combination of the isentropic and Rayleigh processes (see Examples 5 and 6, to follow).

• **Diabatic Flow with Friction (Isothermal).** Here, we consider changes in thermodynamic properties (except for static temperature, which is constant) to be influenced by heat transfer, viscous effects and (under certain conditions) area variations. Referred continuity for the constant-area case, Eq (19), is again written

$$\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2 \tag{32}$$

An analysis nearly identical to that described under the Rayleigh process requires another special column the parameter $(T_t/T_t^*)_{\text{isothermal}}$ to facilitate this problem solution. Again, the total-pressure and total-temperature ratios are implicit functions one of the other, and hence remarks similar to those for the Rayleigh case apply here as well (see Examples 7 and 8, to follow).

• **Normal Shock.** Referred continuity, Eq (19), for this discontinuous-flow phenomenon reduces to

$$\left(\frac{p_{t1}}{p_{t2}}\right)\Gamma_1 = \Gamma_2 \tag{37}$$

where the total pressure ratio p_{t1}/p_{t2} is an explicit function of the inlet-pressure ratio only, Eq (38). Hence, when developing a generalized compressible-flow table, to include the solution of normal-shock processes we will naturally make use of the Γ column. In addition, we will include a column tabulating $(p_{t1}/p_{t2})_{\text{normal shock}}$. One simply modifies Γ_1 by the ratio $(p_{t1}/p_{t2})_{\text{normal shock}}$, found in the same row as Γ_1, to obtain Γ_2 (see Example 9, to follow).

A Generalized Compressible-Flow Table

The generalized compressible-flow function, Γ, is tabulated against the independent parameters p/p_t and M, the state point function T/T_t, and also against the specific process functions $(T_t/T_t^*)_{\text{Rayleigh}}$, $(T_t/T_t^*)_{\text{isothermal}}$, and $(p_{t1}/p_{t2})_{\text{normal shock}}$ to give the Generalized Compressible Flow Tables III to VI presented here. Only skeleton tables are reproduced here, and these at selected values of the specific heat ratio, γ. Tables III and IV are in terms of pressure ratio as the independent variable; Tables V and VI are in terms of Mach number as the independent variable.

Numerical Examples

• **Example 1** (Isentropic Flow, Fig. 8). Consider the subsonic expansion of air (γ = 1.4) from an inlet pressure ratio of 0.980 through a duct of area ratio 1.6937. The inlet total temperature is 540 R, the inlet static pressure is 20 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit.

Solution, by Table III. From

$$p_1/p_{t1} = 0.980$$

Table III—Generalized Compressible Flow Table In Terms of Pressure Ratio (γ = 1.4000)

Pressure Ratio, p/p_t	Temperature Ratio, T/T_t	Mach Number, M	Isentropic & Fanno, Γ	Rayleigh, T_t/T_t^*	Isothermal, T_t/T_t^*	Normal Shock, p_{t1}/p_{t2}
1.00000	1.00000	0.	0.	0.	0.87500	0.
0.98000	0.99424	0.17013	0.28894	0.12907	0.88007	0.
0.96000	0.98840	0.24220	0.40412	0.24327	0.88527	0.
0.94000	0.98248	0.29863	0.48938	0.34434	0.89061	0.
0.92000	0.97646	0.34720	0.55858	0.43380	0.89610	0.
0.90000	0.97035	0.39090	0.61715	0.51294	0.90174	0.
0.88000	0.96414	0.43127	0.66789	0.58290	0.90755	0.
0.86000	0.95782	0.46922	0.71249	0.64468	0.91353	0.
0.84000	0.95141	0.50536	0.75203	0.69914	0.91969	0.
0.82000	0.94488	0.54009	0.78729	0.74706	0.92605	0.
0.80000	0.93823	0.57372	0.81880	0.78911	0.93260	0.
0.78000	0.93147	0.60650	0.84701	0.82589	0.93937	0.
0.76000	0.92458	0.63862	0.87222	0.85791	0.94637	0.
0.74000	0.91757	0.67022	0.89469	0.88565	0.95361	0.
0.72000	0.91041	0.70144	0.91464	0.90953	0.96110	0.
0.70000	0.90311	0.73239	0.93222	0.92990	0.96887	0.
0.68000	0.89566	0.76318	0.94756	0.94710	0.97693	0.
0.66000	0.88806	0.79389	0.96079	0.96142	0.98530	0.
0.64000	0.88028	0.82461	0.97199	0.97312	0.99400	0.
0.62665	0.87500	0.84515	0.97837	0.97959	1.00000	0.
0.62000	0.87234	0.85542	0.98123	0.98244	1.00305	0.
0.60000	0.86420	0.88639	0.98858	0.98959	1.01250	0.
0.58000	0.85587	0.91761	0.99409	0.99474	1.02235	0.
0.56000	0.84733	0.94914	0.99778	0.99808	1.03265	0.
0.54000	0.83857	0.98107	0.99970	0.99974	1.04344	0.
0.52828	0.83333	1.00000	1.00000	1.00000	1.05000	1.00000
0.52000	0.82958	1.01348	0.99985	0.99988	1.05475	1.00000
0.50000	0.82034	1.04645	0.99825	0.99859	1.06664	1.00012
0.48000	0.81082	1.08008	0.99490	0.99600	1.07915	1.00057
0.46000	0.80102	1.11446	0.98979	0.99219	1.09235	1.00157
0.44000	0.79091	1.14969	0.98291	0.98726	1.10631	1.00330
0.42000	0.78047	1.18591	0.97424	0.98126	1.12112	1.00596
0.40000	0.76967	1.22324	0.96375	0.97428	1.13685	1.00973
0.38000	0.75847	1.26183	0.95139	0.96636	1.15364	1.01480
0.36000	0.74684	1.30186	0.93712	0.95756	1.17160	1.02140
0.34000	0.73475	1.34353	0.92088	0.94790	1.19089	1.02977
0.32000	0.72213	1.38707	0.90258	0.93742	1.21170	1.04020
0.30000	0.70893	1.43277	0.88214	0.92614	1.23425	1.05303
0.28000	0.69510	1.48096	0.85945	0.91407	1[illegible]	1.06871
0.26000	0.68053	1.53205	0.83438	0.90121	1[illegible]	1.08777
0.24000	0.66515	1.58655	0.80677	0.88754	1[illegible]50	1.11090
0.22000	0.64882	1.64510	0.77642	0.87304	1.34861	1.13906
0.20000	0.63139	1.70853	0.74311	0.85766	1.38584	1.17348
0.18000	0.61266	1.77795	0.70652	0.84134	1.42819	1.21593
0.16000	0.59239	1.85484	0.66630	0.82397	1.47707	1.26898
0.14000	0.57021	1.94130	0.62194	0.80539	1.53451	1.33647
0.12000	0.54564	2.04046	0.57280	0.78540	1.60361	1.42457
0.10000	0.51795	2.15719	0.51795	0.76364	1.68936	1.54383
0.08000	0.48596	2.29978	0.45606	0.73958	1.80057	1.71411
0.06000	0.44761	2.48403	0.38495	0.71224	1.95482	1.97848
0.04000	0.39865	2.74634	0.30066	0.67966	2.19492	2.45413
0.02000	0.32703	3.20769	0.19386	0.63639	2.67562	3.64440
0.	0.	∞	0.	0.48980	∞	∞

Table IV—Generalized Compressible Flow Table In Terms of Pressure Ratio (γ = 1.3000)

Pressure Ratio, p/p_t	Temperature Ratio, T/T_t	Mach Number, M	Isentropic & Fanno, Γ	Rayleigh, T_t/T_t^*	Isothermal, T_t/T_t^*	Normal Shock, p_{t1}/p_{t2}
1.00000	1.00000	0.	0.	0.	0.89655	0.
0.98000	0.99535	0.17650	0.29626	0.13299	0.90074	0.
0.96000	0.99062	0.25120	0.41400	0.25026	0.90504	0.
0.94000	0.98582	0.30964	0.50091	[illegible]	0.90945	0.
0.92000	0.98094	0.35989	0.57123	[illegible]	0.91397	0.
0.90000	0.97598	0.40507	0.63056	0.52533	0.91862	0.
0.88000	0.97093	0.44676	0.68177	0.59613	0.92339	0.
0.86000	0.96579	0.48592	0.72660	0.65839	0.92831	0.
0.84000	0.96056	0.52317	0.76618	0.71304	0.93336	0.
0.82000	0.95524	0.55893	0.80130	0.76088	0.93857	0.
0.80000	0.94981	0.59354	0.83252	0.80264	0.94393	0.
0.78000	0.94428	0.62723	0.86030	0.83894	0.94946	0.
0.76000	0.93863	0.66020	0.88495	0.87033	0.95517	0.
0.74000	0.93287	0.69261	0.90675	0.89731	0.96106	0.
0.72000	0.92699	0.72460	0.92590	0.92031	0.96716	0.
0.70000	0.92099	0.75627	0.94259	0.93972	0.97347	0.
0.68000	0.91485	0.78774	0.95695	0.95588	0.98000	0.
0.66000	0.90857	0.81909	0.96911	0.96910	0.98678	0.
0.64000	0.90214	0.85041	0.97915	0.97965	0.99381	0.
0.62301	0.89655	0.87706	0.98607	0.98669	1.00000	0.
0.62000	0.89555	0.88178	0.98715	0.98776	1.00112	0.
0.60000	0.88880	0.91328	0.99318	0.99366	1.00872	0.
0.58000	0.88187	0.94498	0.99730	0.99755	1.01664	0.
0.56000	0.87476	0.97696	0.99953	0.99959	1.02491	0.
0.54573	0.86957	1.00000	1.00000	1.00000	1.03103	1.00000
0.54000	0.86745	1.00930	0.99993	0.99994	1.03355	1.00000
0.52000	0.85993	1.04207	0.99849	0.99873	1.04259	1.00009
0.50000	0.85218	1.07536	0.99525	0.99610	1.05207	1.00049
0.48000	0.84419	1.10925	0.99021	0.99214	1.06203	1.00141
0.46000	0.83594	1.14385	0.98336	0.98696	1.07251	1.00304
0.44000	0.82741	1.17925	0.97470	0.98065	1.08357	1.00556
0.42000	0.81857	1.21556	0.96421	0.97327	1.09526	1.00915
0.40000	0.80941	1.25292	0.95186	0.96489	1.10766	1.01400
0.38000	0.79988	1.29146	0.93762	0.95556	1.12085	1.02031
0.36000	0.78997	1.33136	0.92144	0.94534	1.13492	1.02831
0.34000	0.77961	1.37280	0.90328	0.93426	1.14999	1.03826
0.32000	0.76878	1.41600	0.88305	0.92234	1.16620	1.05047
0.30000	0.75742	1.46122	0.86069	0.90961	1.18369	1.06533
0.28000	0.74545	1.50878	0.83608	0.89607	1.20269	1.08330
0.26000	[illegible]	1.55906	0.80913	0.88172	1.22344	1.10498
0.24000	[illegible]	1.61254	0.77967	0.86653	1.24624	1.13115
0.22000	0.70510	1.66980	0.74755	0.85049	1.27152	1.16285
0.20000	0.68976	1.73162	0.71254	0.83352	1.29980	1.20149
0.18000	0.67319	1.79899	0.67439	0.81556	1.33179	1.24903
0.16000	0.65514	1.87330	0.63275	0.79647	1.36848	1.30836
0.14000	0.63526	1.95645	0.58721	0.77611	1.41131	1.38384
0.12000	0.61306	2.05127	0.53719	0.75420	1.46242	1.48244
0.10000	0.58780	2.16218	0.48190	0.73038	1.52526	1.61621
0.08000	0.55830	2.29659	0.42016	0.70402	1.60586	1.80792
0.06000	0.52244	2.46860	0.35016	0.67403	1.71609	2.10732
0.04000	0.47577	2.71028	0.26857	0.63817	1.88441	2.65122
0.02000	0.40545	3.12668	0.16782	0.59018	2.21127	4.03705
0.	0	∞	0.	0.40528	∞	∞

Table V—Generalized Compressible Flow Table In Terms of Mach Number (γ = 1.4000)

Mach Number, M	Pressure Ratio, p/p_t	Temperature Ratio, T/T_t	Isentropic & Fanno, Γ	Rayleigh, T_t/T_t^*	Isothermal, T_t/T_t^*	Normal Shock, p_{t1}/p_{t2}
0.	1.00000	1.00000	0.	0.	0.87500	0.
0.05000	0.99825	0.99950	0.08627	0.01192	0.87544	0.
0.10000	0.99303	0.99800	0.17177	0.04678	0.87675	0.
0.15000	0.98441	0.99552	0.25573	0.10196	0.87894	0.
0.20000	0.97250	0.99206	0.33744	0.17355	0.88200	0.
0.25000	0.95745	0.98765	0.41620	0.25684	0.88594	0.
0.30000	0.93947	0.98232	0.49138	0.34686	0.89075	0.
0.35000	0.91877	0.97609	0.56244	0.43894	0.89644	0.
0.40000	0.89561	0.96899	0.62888	0.52903	0.90300	0.
0.45000	0.87027	0.96108	0.69029	0.61393	0.91044	0.
0.50000	0.84302	0.95238	0.74636	0.69136	0.91875	0.
0.55000	0.81417	0.94295	0.79685	0.75991	0.92794	0.
0.60000	0.78400	0.93284	0.84161	0.81892	0.93800	0.
0.65000	0.75283	0.92208	0.88058	0.86833	0.94894	0.
0.70000	0.72093	0.91075	0.91377	[illegible]	0.96075	0.
0.75000	0.68857	0.89888	0.94125	0.94009	0.97344	0.
0.80000	0.65602	0.88652	0.96318	0.96395	0.98700	0.
0.84515	0.62665	0.87500	0.97837	0.97959	1.00000	0.
0.85000	0.62351	0.87374	0.97975	0.98097	1.00144	0.
0.90000	0.59126	0.86059	0.99121	0.99207	1.01675	0.
0.95000	0.55946	0.84710	0.99786	0.99814	1.03294	0.
1.00000	0.52828	0.83333	1.00000	1.00000	1.05000	1.00000
1.05000	0.49787	0.81934	0.99798	0.99838	1.06794	1.00015
1.10000	0.46835	0.80515	0.99214	0.99392	1.08675	1.00107
1.15000	0.43983	0.79083	0.98285	0.98721	1.10644	1.00332
1.20000	0.41238	0.77640	0.97046	0.97872	1.12700	1.00725
1.25000	0.38606	0.76190	0.95534	0.96886	1.14844	1.01311
1.30000	0.36091	0.74738	0.93782	0.95798	1.17075	1.02106
1.35000	0.33697	0.73287	0.91824	0.94637	1.19394	1.03121
1.40000	0.31424	0.71839	0.89692	0.93425	1.21800	1.04363
1.45000	0.25272	0.70398	0.87415	0.92184	1.24294	1.05838
1.50000	0.27240	0.68966	0.85022	0.90928	1.26875	1.07552
1.55000	0.25326	0.67545	0.82537	0.89669	1.29544	1.09506
1.60000	0.23527	0.66138	0.77985	0.88419	1.32300	1.11707
1.65000	0.21840	0.64746	0.77386	0.87184	1.35144	1.14157
1.70000	0.20259	0.63371	0.74760	0.85971	1.38075	1.16860
1.75000	0.18782	0.62016	0.72125	0.84784	1.41094	1.19823
1.80000	0.17404	0.60680	0.69494	0.83628	1.44200	1.23049
1.85000	0.16120	0.59365	0.66881	0.82504	1.47394	1.26545
1.90000	0.14924	0.58072	0.64298	0.81414	1.50675	1.30317
1.95000	0.13813	0.56802	0.61755	0.80358	1.54044	1.34373
2.00000	0.12780	0.55556	0.59259	0.79339	1.57500	1.38720
2.10000	0.10935	0.53135	0.54438	0.77406	1.64675	1.48323
2.20000	0.09352	0.50813	0.49876	0.75614	1.72200	1.59201
2.30000	0.07997	0.48591	0.45597	0.73954	1.80075	1.71440
2.40000	0.06840	0.46468	0.41613	0.72421	1.88300	1.85136
2.50000	0.05853	0.44444	0.37926	0.71006	1.96875	2.00395
2.60000	0.05012	0.42517	0.34531	0.69700	2.05800	2.17333
2.70000	0.04295	0.40684	0.31417	0.68494	2.15075	2.36077
2.80000	0.03685	0.38941	0.28570	0.67380	2.24700	2.56763
2.90000	0.03165	0.37286	0.25976	0.66350	2.34675	2.79538
3.00000	0.02722	0.35714	0.23615	0.65398	2.45000	3.04558
3.10000	0.02345	0.34223	0.21472	0.64516	2.55675	3.31992
3.20000	0.02023	0.32808	0.19528	0.63699	2.66700	3.62018
3.30000	0.01748	0.31466	0.17766	0.62941	2.78075	3.94826
3.40000	0.01512	0.30193	0.16172	0.62236	2.89800	4.30615
3.50000	0.01311	0.28986	0.14728	0.61581	3.01875	4.69598

Table VI—Generalized Compressible Flow Table In Terms of Mach Number (γ = 1.3000)

Mach Number, M	Pressure Ratio, p/p_t	Temperature Ratio, T/T_t	Isentropic & Fanno, Γ	Rayleigh, T_t/T_t^*	Isothermal, T_t/T_t^*	Normal Shock, p_{t1}/p_{t2}
0.	1.00000	1.00000	0.	0.	0.89655	0.
0.05000	0.99838	0.99963	0.08532	0.01143	0.89689	0.
0.10000	0.99353	0.99850	0.16990	0.04489	0.89790	0.
0.15000	0.98551	0.99664	0.25302	0.09803	0.89958	0.
0.20000	0.97441	0.99404	0.33400	0.16726	0.90193	0.
0.25000	0.96037	0.99071	0.41217	0.24822	0.90496	0.
0.30000	0.94355	0.98668	0.48694	0.33629	0.90866	0.
0.35000	0.92413	0.98196	0.55774	0.42702	0.91303	0.
0.40000	0.90233	0.97656	0.62410	0.51647	0.91807	0.
0.45000	0.87839	0.97052	0.68560	0.60145	0.92378	0.
0.50000	0.85255	0.96386	0.74192	0.67960	0.93017	0.
0.55000	0.82506	0.95659	0.79280	0.74937	0.93723	0.
0.60000	0.79620	0.94877	0.83805	0.80993	0.94497	0.
0.65000	0.76623	0.94040	0.87759	0.86105	0.95337	0.
0.70000	0.73540	0.93153	0.91138	0.90294	0.96245	0.
0.75000	0.70397	0.92219	0.93947	0.93614	0.97220	0.
0.80000	0.67218	0.91241	0.96196	0.96139	0.98262	0.
0.85000	0.64026	0.90222	0.97903	0.97952	0.99372	0.
0.87706	0.62301	0.89655	0.98607	0.98669	1.00000	0.
0.90000	0.60842	0.89166	0.99088	0.99143	1.00548	0.
0.95000	0.57685	0.88077	0.99777	0.99799	1.01792	0.
1.00000	0.54573	0.86957	1.00000	1.00000	1.03103	1.00000
1.05000	0.51521	0.85809	0.99788	0.99823	1.04482	1.00015
1.10000	0.48542	0.84638	0.99176	0.99334	1.05928	1.00110
1.15000	0.45649	0.83446	0.98198	0.98593	1.07441	1.00341
1.20000	0.42850	0.82237	0.96890	0.97653	1.09021	1.00748
1.25000	0.40154	0.81013	0.95288	0.96556	1.10668	1.01357
1.30000	0.37566	0.79777	0.93428	0.95342	1.12383	1.02189
1.35000	0.35091	0.78531	0.91343	0.94041	1.14165	1.03258
1.40000	0.32730	0.77280	0.89068	0.92679	1.16014	1.04573
1.45000	0.30487	0.76024	0.86634	0.91279	1.17930	1.06144
1.50000	0.28361	0.74766	0.84070	0.89858	1.19914	1.07980
1.55000	0.26352	0.73509	0.81405	0.88430	1.21965	1.10086
1.60000	0.24457	0.72254	0.78664	0.87008	1.24083	1.12472
1.65000	0.22675	0.71004	0.75870	0.85600	1.26268	1.15145
1.70000	0.21003	0.69759	0.73047	0.84215	1.28521	1.18114
1.75000	0.19436	0.68523	0.70211	0.82856	1.30840	1.21389
1.80000	0.17972	0.67295	0.67382	0.81529	1.33228	1.24979
1.85000	0.16605	0.66078	0.64573	0.80237	1.35682	1.28896
1.90000	0.15331	0.64872	0.61799	0.78982	1.38203	1.33153
1.95000	0.14147	0.63679	0.59070	0.77765	1.40792	1.37763
2.00000	0.13046	0.62500	0.56396	0.76587	1.43448	1.42741
2.10000	0.11079	0.60187	0.51244	0.74350	1.48962	1.53866
2.20000	0.09393	0.57937	0.46392	0.72269	1.54745	1.66672
2.30000	0.07955	0.55757	0.41868	0.70339	1.60796	1.81323
2.40000	0.06731	0.53648	0.37686	0.68551	1.67117	1.98005
2.50000	0.05692	0.51613	0.33847	0.66898	1.73707	2.16930
2.60000	0.04813	0.49652	0.30345	0.65370	1.80565	2.38332
2.70000	0.04070	0.47767	0.27166	0.63956	1.87693	2.62474
2.80000	0.03442	0.45956	0.24293	0.62649	1.95090	2.89647
2.90000	0.02913	0.44218	0.21705	0.61440	2.02755	3.20172
3.00000	0.02466	0.42553	0.19381	0.60320	2.10689	3.54404
3.10000	0.02090	0.40958	0.17299	0.59282	2.18893	3.92731
3.20000	0.01773	0.39432	0.15438	0.58319	2.27365	4.35581
3.30000	0.01506	0.37972	0.13777	0.57424	2.36107	4.83418
3.40000	0.01280	0.36576	0.12296	0.56592	2.45117	5.36752
3.50000	0.01090	0.35242	0.10977	0.55818	2.54396	5.96136

we obtain

$$\Gamma_1 = A^*/A_1 = 0.28894$$

But

$$\Gamma_2 = \Gamma_1(A_1/A_2) = 0.28894 \times 1.6937 = 0.48938 = A^*/A_2$$

From

$$\Gamma_2 = 0.48938$$

we obtain

$$p_2/p_{t2} = 0.940 \qquad T_2/T_{t2} = 0.98248 \qquad M_2 = 0.29863$$

whence

$$p_{t2} = p_{t1} = \frac{20\text{ psia}}{0.980} = 20.408\text{ psia}$$

$$p_2 = 0.940 \times 20.408\text{ psia} = 19.184\text{ psia}$$

$$T_{t2} = T_{t1} = 540\text{ R}$$

$$T_2 = 0.98248 \times 540\text{ R} = 530.54\text{ R}$$

$$M_2 = 0.29863$$

• **Example 2** (Isentropic Flow, Fig. 8). Consider the supersonic expansion of steam (γ = 1.3) from an inlet Mach number of 0.15 through a convergent-divergent nozzle of overall area ratio 2.99858. The inlet total temperature is 1000 R, the inlet static pressure is 100 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit.

Solution, by Table VI. From

$$M_1 = 0.15$$

we obtain

$$p_1/p_{t1} = 0.98551 \qquad \Gamma_1 = A^*/A_1 = 0.25302$$

But

$$\Gamma_{2x} = \Gamma_1\left(\frac{A_1}{A_2}\right) = 0.25302 \times 2.99858 = 0.75870 = \frac{A^*}{A_2}$$

From

$$\Gamma_{2x} = 0.75870$$

we obtain

$$p_{2x}/p_{t2x} = 0.22675 \qquad T_{2x}/T_{t2x} = 0.71004 \qquad M_{2x} = 1.65$$

whence

$$p_{t2x} = p_{t1} = \frac{100\text{ psia}}{0.98551} = 101.470\text{ psia}$$

$$p_{2x} = 0.22675 \times 101.470\text{ psia} = 23.008\text{ psia}$$

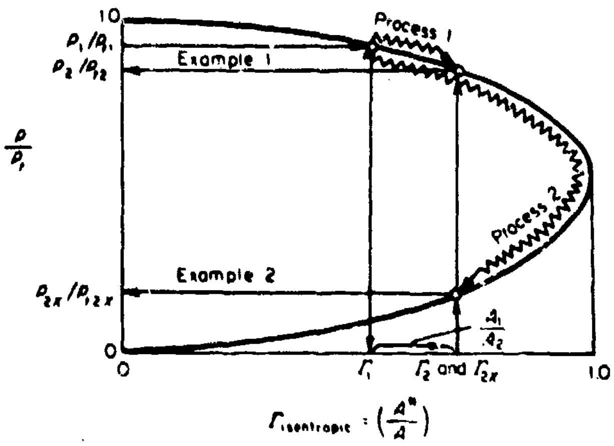


Fig. 8 Examples 1 and 2, isentropic flow.

$$T_{t2x} = T_{t1} = 1000 \text{ R}$$
$$T_{2x} = 0.71004 \times 1000 \text{ R} = 710.04 \text{ R}$$
$$M_{2x} = 1.65$$

• **Example 3** (Fanno Flow, Fig. 9). Consider the adiabatic flow of air ($\gamma = 1.4$) from an inlet pressure ratio of 0.960 through a duct of constant area. The inlet total temperature is 540 R, the inlet static pressure is 50 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit if the total pressure ratio across the duct is 1.8609.

Solution, by Table III. From

$$\frac{p_1}{p_{t1}} = 0.960$$

we obtain

$$\Gamma_1 = \left(\frac{p_t^*}{p_{t1}}\right)\left(\frac{A^*}{A_1}\right) = 0.40412$$

But

$$\Gamma_2 = \Gamma_1\left(\frac{p_{t1}}{p_{t2}}\right)\left(\frac{A_1}{A_2}\right) = 0.40412 \times 1.8609 \times 1$$
$$= 0.75203 = \left(\frac{p_t^*}{p_{t2}}\right)\left(\frac{A^*}{A_2}\right)$$

From

$$\Gamma_2 = 0.75203$$

we obtain

$$p_2/p_{t2} = 0.840 \qquad T_2/T_{t2} = 0.95141 \qquad M_2 = 0.50536$$

whence

$$p_{t1} = \frac{50 \text{ psia}}{0.960} = 52.083 \text{ psia}$$

$$p_{t2} = \frac{52.083 \text{ psia}}{1.8609} = 27.988 \text{ psia}$$
$$p_2 = 0.840 \times 27.988 \text{ psia} = 23.510 \text{ psia}$$
$$T_{t2} = T_{t1} = 540 \text{ R}$$
$$T_2 = 0.95141 \times 540 \text{ R} = 513.76 \text{ R}$$
$$M_2 = 0.50536$$

• **Example 4** (Fanno-Type Flow, Fig. 9). Consider the adiabatic flow of steam ($\gamma = 1.3$) from an inlet Mach number of 0.25 through a convergent-divergent nozzle of overall area ratio 0.73527. The inlet total temperature is 1000 R, the inlet static pressure is 100 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit if the total pressure ratio across the duct is 1.8609.

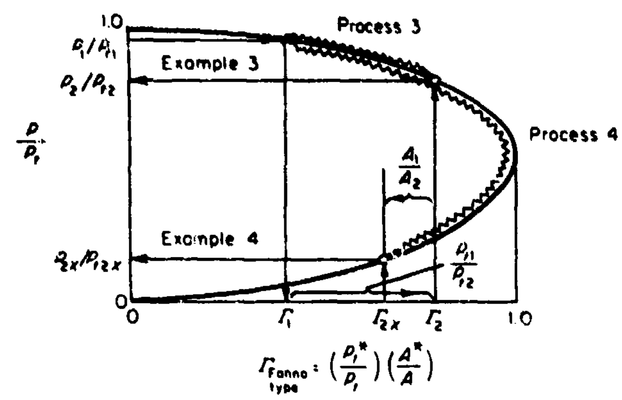


Fig. 9 Examples 3 and 4, Fanno flow.

Solution, by Table VI. From

$$M_1 = 0.25$$

we obtain

$$\frac{p_1}{p_{t1}} = 0.96037 \qquad \Gamma_1 = \left(\frac{p_t^*}{p_{t1}}\right)\left(\frac{A^*}{A_1}\right) = 0.41217$$

But

$$\Gamma_{2x} = \Gamma_1\left(\frac{p_{t1}}{p_{t2x}}\right)\left(\frac{A_1}{A_2}\right)$$
$$= 0.41217 \times 1.8609 \times 0.73527 = 0.56396$$

From

$$\Gamma_{2x} = 0.56396$$

we obtain

$$p_{2x}/p_{t2x} = 0.13046 \qquad T_{2x}/T_{t2x} = 0.62500 \qquad M_{2x} = 2.0$$

whence

$$p_{t1} = \frac{100 \text{ psia}}{0.96037} = 104.126 \text{ psia}$$

$$p_{t2x} = \frac{104.126 \text{ psia}}{1.8609} = 55.955 \text{ psia}$$
$$p_{2x} = 0.13046 \times 55.955 \text{ psia} = 7.300 \text{ psia}$$
$$T_{t2x} = T_{t1} = 1000 \text{ R}$$
$$T_{2x} = 0.6250 \times 1000 \text{ R} = 625.00 \text{ R}$$
$$M_{2x} = 2.0$$

• **Example 5** (Rayleigh Flow, Fig. 10). Consider the frictionless flow of air ($\gamma = 1.4$) from an inlet pressure ratio of 0.300 through a duct of constant area. The inlet total temperature is 540 R, the inlet static pressure is 5 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit if the total temperature ratio across the duct is 0.79856.

Solution, by Table III. From

$$p_{1x}/p_{t1x} = 0.300$$

we obtain

$$\Gamma_{1x} = \left(\frac{T_{t1x}}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t1x}}\right) = 0.88214 \qquad \frac{T_{t1x}}{T_t^*} = 0.92614$$

But

$$\frac{T_{t2x}}{T_t^*} = \left(\frac{T_{t1x}}{T_t^*}\right)\left(\frac{T_{t2x}}{T_{t1x}}\right) = 0.92614 \times 0.79856 = 0.73958$$

Corresponding to $T_{t2x}/T_t^* = 0.73958$ is

$$\Gamma_{2x} = \Gamma_{1x}\left(\frac{T_{t2x}}{T_{t1x}}\right)^{\frac{1}{2}}\left(\frac{p_{t1x}}{p_{t2x}}\right)$$
$$= 0.45606 = \left(\frac{T_{t2x}}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t2x}}\right)$$

From

$$\Gamma_{2x} = 0.45606$$

we obtain

$$p_{2x}/p_{t2x} = 0.080 \qquad T_{2x}/T_{t2x} = 0.48596 \qquad M_{2x} = 2.29978$$

whence

$$\frac{p_{t1x}}{p_{t2x}} = \left(\frac{\Gamma_{2x}}{\Gamma_{1x}}\right)\left(\frac{T_{t1x}}{T_{t2x}}\right)^{\frac{1}{2}} = \frac{0.45606}{(0.88214)(0.79856)^{1/2}} = 0.57854$$
$$p_{t1x} = \frac{5 \text{ psia}}{0.300} = 16.667 \text{ psia}$$
$$p_{t2x} = \frac{16.667 \text{ psia}}{0.57854} = 28.809 \text{ psia}$$
$$p_{2x} = 0.080 \times 28.809 \text{ psia} = 2.305 \text{ psia}$$
$$T_{t2x} = 0.79856 \times 540 \text{ R} = 431.22 \text{ R}$$
$$T_{2x} = 0.48596 \times 431.22 \text{ R} = 209.56 \text{ R}$$
$$M_{2x} = 2.29978$$

• **Example 6.** (Rayleigh-Type Flow, Fig. 10). Consider the frictionless flow of air ($\gamma = 1.4$) from an inlet Mach number of 0.25 through a duct of area ratio 1.45301. The inlet total temperature is 460 R, the inlet static pressure is 10 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit if the total temperature ratio across the duct is 2.05976 and all heat is transferred at the inlet area.

Solution, by Table V. From

$$M_1 = 0.25$$

we obtain

$$\frac{p_1}{p_{t1}} = 0.95745 \qquad \Gamma_1 = \left(\frac{T_{t1}}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t1}}\right) = 0.41620$$
$$\frac{T_{t1}}{T_t^*} = 0.25684$$

But

$$\frac{T_{t2}}{T_t^*} = \left(\frac{T_{t1}}{T_t^*}\right)\left(\frac{T_{t2}}{T_{t1}}\right) = 0.25684 \times 2.05976 = 0.52903$$

Corresponding to $T_{t2}/T_t^* = 0.52903$ is

$$\Gamma_2' = \Gamma_1\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t2}}\right) = 0.62888$$
$$= \left(\frac{T_{t2}}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t2}}\right)$$

Now $\quad \Gamma_2 = \Gamma_2'\left(\frac{A_1}{A_2}\right) = 0.62888 \times 1.45301 = 0.91377$

From $\quad \Gamma_2 = 0.91377$

we obtain

$$p_2/p_{t2} = 0.72093 \qquad T_2/T_{t2} = 0.91075 \qquad M_2 = 0.70$$

whence

$$\frac{p_{t1}}{p_{t2}} = \frac{\Gamma_2'}{\Gamma_1}\left(\frac{T_{t1}}{T_{t2}}\right)^{\frac{1}{2}} = \frac{0.62888}{(0.41620)(2.05976)^{1/2}} = 1.05282$$
$$p_{t1} = \frac{10 \text{ psia}}{0.95745} = 10.444 \text{ psia}$$
$$p_{t2} = \frac{10.444 \text{ psia}}{1.05282} = 9.920 \text{ psia}$$
$$p_2 = 0.72093 \times 9.920 \text{ psia} = 7.152 \text{ psia}$$
$$T_{t2} = 2.05976 \times 460 \text{ R} = 947.49 \text{ R}$$
$$T_2 = 0.91075 \times 947.49 \text{ R} = 862.93 \text{ R}$$
$$M_2 = 0.70$$

• **Example 7** (Isothermal Flow, Fig. 11). Consider the isothermal flow of air ($\gamma = 1.4$) from an inlet pressure ratio of 0.960 through a duct of constant area. The inlet total temperature is 540 R, the inlet static pressure is 15 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit if the total temperature ratio across the duct is 1.112994.

Solution, by Table III. From

$$p_1/p_{t1} = 0.960$$

we obtain

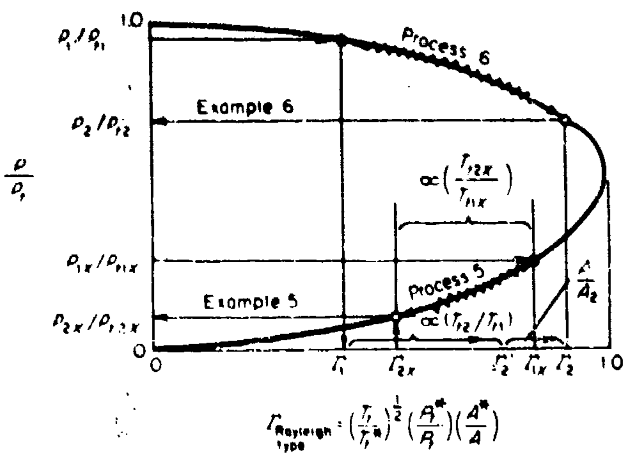


Fig. 10. Examples 5 and 6. Rayleigh flow.

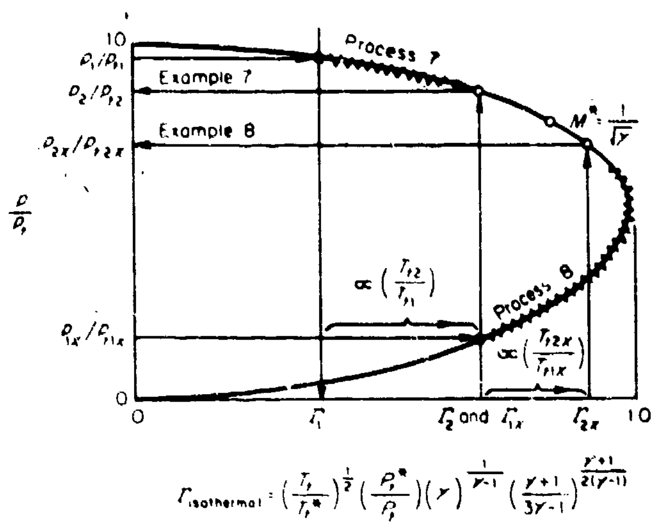


Fig. 11. Examples 7 and 8. Isothermal flow.

$$\Gamma_1 = \left(\frac{T_{t1}}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t1}}\right)(\gamma)^{\frac{1}{\gamma-1}}\left(\frac{\gamma+1}{3\gamma-1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} = 0.40412$$

$$\frac{T_{t1}}{T_t^*} = 0.88527$$

But

$$\frac{T_{t2}}{T_t^*} = \left(\frac{T_{t1}}{T_t^*}\right)\left(\frac{T_{t2}}{T_{t1}}\right) = 0.88527 \times 1.112994 = 0.98530$$

Corresponding to $T_{t2}/T_t^* = 0.98530$ is

$$\Gamma_2 = \Gamma_1\left(\frac{T_{t2}}{T_{t1}}\right)^{\frac{1}{2}}\left(\frac{p_{t1}}{p_{t2}}\right) = 0.96079$$

From

$$\Gamma_2 = 0.96079$$

we obtain

$$p_2/p_{t2} = 0.660 \qquad T_2/T_{t2} = 0.88806 \qquad M_2 = 0.79389$$

whence

$$\frac{p_{t1}}{p_{t2}} = \left(\frac{\Gamma_2}{\Gamma_1}\right)\left(\frac{T_{t1}}{T_{t2}}\right)^{\frac{1}{2}} = \frac{0.96079}{(0.40412)(1.112994)^{1/2}} = 2.25357$$

$$p_{t1} = \frac{15\text{ psia}}{0.960} = 15.625\text{ psia}$$

$$p_{t2} = \frac{15.625\text{ psia}}{2.25357} = 6.933\text{ psia}$$

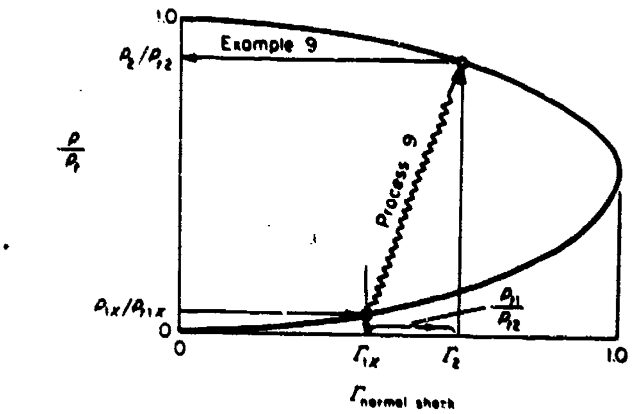


Fig. 12. Example 9, normal shock.

$$p_2 = 0.660 \times 6.933\text{ psia} = 4.576\text{ psia}$$

$$T_{t2} = 1.112994 \times 540\text{ R} = 601.02\text{ R}$$

$$T_2 = T_1 = 0.88806 \times 601.02\text{ R} = 533.74\text{ R}$$

$$M_2 = 0.79389$$

• **Example 8** (Isothermal Flow, Fig. 11). Consider the isothermal flow of air ($\gamma = 1.4$) from an inlet Mach number of 1.30 through a duct of constant area. The inlet total temperature is 700 R, the inlet static pressure is 3 psia. Find the total and static pressures, the total and static temperatures, and the Mach number at exit if the total temperature ratio across the duct is 0.86846.

Solution, by Table V. From

$$M_1 = 1.30$$

we obtain

$$\frac{p_{1x}}{p_{t1x}} = 0.36091$$

$$\Gamma_{1x} = \left(\frac{T_{t1x}}{T_t^*}\right)^{\frac{1}{2}}\left(\frac{p_t^*}{p_{t1x}}\right)(\gamma)^{\frac{1}{\gamma-1}}\left(\frac{\gamma+1}{3\gamma-1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} = 0.93782$$

$$\frac{T_{t1x}}{T_t^*} = 1.17075$$

But

$$\frac{T_{t2x}}{T_t^*} = \left(\frac{T_{t1x}}{T_t^*}\right)\left(\frac{T_{t2x}}{T_{t1x}}\right) = 1.17075 \times 0.86846 = 1.01675$$

Corresponding to $T_{t2x}/T_t^* = 1.01675$ is

$$\Gamma_{2x} = \Gamma_{1x}\left(\frac{T_{t2x}}{T_{t1x}}\right)^{\frac{1}{2}}\left(\frac{p_{t1x}}{p_{t2x}}\right) = 0.99121$$

From

$$\Gamma_{2x} = 0.99121$$

we obtain

$$p_{2x}/p_{t2x} = 0.59126 \qquad T_{2x}/T_{t2x} = 0.86059 \qquad M_{2x} = 0.90$$

whence

$$\frac{p_{t1x}}{p_{t2x}} = \left(\frac{\Gamma_{2x}}{\Gamma_{1x}}\right)\left(\frac{T_{t1x}}{T_{t2x}}\right)^{\frac{1}{2}} = \frac{0.99121}{(0.93782)(0.86846)^{1/2}} = 1.13415$$

$$p_{t1x} = \frac{3\text{ psia}}{0.36091} = 8.312\text{ psia}$$

$$p_{t2x} = \frac{8.312\text{ psia}}{1.13415} = 7.329\text{ psia}$$

$$p_{2x} = 0.59126 \times 7.329\text{ psia} = 4.333\text{ psia}$$

$$T_{t2x} = 0.86846 \times 700\text{ R} = 607.92\text{ R}$$

$$T_{2x} = T_{1x} = 0.86059 \times 607.92\text{ R} = 523.17\text{ R}$$

$$M_{2x} = 0.90$$

• **Example 9** (Normal Shock, Fig. 12). Consider air ($\gamma = 1.4$) at an initial Mach number of 3.0 to undergo a normal-shock process. The inlet total temperature is 900 R, the inlet static pressure is 5 psia. Find the total and static pressures, the total and static temperatures, and the Mach number immediately after the shock.

Solution, by Table V. From

$$M_{1x} = 3.0$$

we obtain

$$\frac{p_{1x}}{p_{t1x}} = 0.02722 \qquad \Gamma_{1x} = \left(\frac{p_t^*}{p_{t1x}}\right) = 0.23615 \qquad \frac{p_{t1x}}{p_{t2}} = 3.04558$$

But

$$\Gamma_2 = \Gamma_{1x}\left(\frac{p_{t1x}}{p_{t2}}\right) = 0.23615 \times 3.04558$$

$$= 0.71921 = \left(\frac{p_t^*}{p_{t2}}\right)$$

From

$$\Gamma_2 = 0.71921$$

we obtain, by linear interpolation,

$$p_2/p_{t2} = 0.85621 \qquad T_2/T_{t2} = 0.95659 \qquad M_2 = 0.47579$$

whence

$$p_{t1x} = \frac{5\text{ psia}}{0.02722} = 183.688\text{ psia}$$

$$p_{t2} = \frac{183.688\text{ psia}}{3.04558} = 60.313\text{ psia}$$

$$p_2 = 0.85621 \times 60.313\text{ psia} = 51.641\text{ psia}$$

$$T_{t2} = T_{t1} = 900\text{ R}$$

$$T_2 = 0.95659 \times 900\text{ R} = 860.93\text{ R}$$

$$M_2 = 0.47579$$

3.7.5 Lockhart-Martinelli Equations for Two-Phase Flow

The Lockhart-Martinelli procedure is probably the most popular method used to determine the pressure drop for two-phase flow in a pipe. The derivation can be found in Reference 166-7. The basic assumptions (Reference 213-1) are:

a) Isothermal transportation

b) No mass transfer between phases

c) Static pressure drop for the liquid phase equals the static pressure drop for the gas phase, regardless of flow pattern

d) Pressure drop per unit length is constant, thus eliminating slug flow

e) Total volume of the pipe equals the sum of the volumes of liquid phase and gas phase.

The basic Lockhart-Martinelli equation for two-phase pressure drop is stated

(Eq. 3.7.5a)

$$\left(\frac{\Delta p}{\Delta L}\right)_{TP} = \phi_l^2 \left(\frac{\Delta p}{\Delta L}\right)_l = \phi_g^2 \left(\frac{\Delta p}{\Delta L}\right)_g$$

where Δp = pressure drop, lb_f/in^2

ΔL = length increment, in.

ϕ = a function of X (Figure 3.7.5)

and

(Eq. 3.7.5b)

$$X^2 = \frac{\left(\frac{\Delta p}{\Delta L}\right)_l}{\left(\frac{\Delta p}{\Delta L}\right)_g} = \frac{(Re_g)^m\, C_l \rho_g\, \dot{w}_l^2}{(Re_l)^n\, C_g \rho_l\, \dot{w}_g^2}$$

where

$Re_{g,l}$ = Reynolds numbers for gas and liquid phases, respectively

$C_l, C_g, m, n,$ = experimentally determined constants, listed in Table 3.7.5

$\dot{w}_{g,l}$ = weight flow rates for gas and liquid, respectively, lb_f/sec

Table 3.7.5. Two-Phase Flow Constants

(From "Applied Cryogenic Engineering," R. W. Vance and W. M. Duke, Copyright 1962 by John Wiley and Sons)

FUNCTION	LAMINAR l LAMINAR g	TURBULENT l LAMINAR g	LAMINAR l TURBULENT g	TURBULENT l TURBULENT g
Re_l	< 1000	> 2000	< 1000	> 2000
Re_g	< 1000	< 1000	> 2000	> 2000
m	1.0	1.0	0.2	0.2
n	1.0	0.2	1.0	0.2
C_l	16.0	0.046*	16.0	0.046*
C_g	16.0	16.0	0.046*	0.046*

*For smooth pipes only.

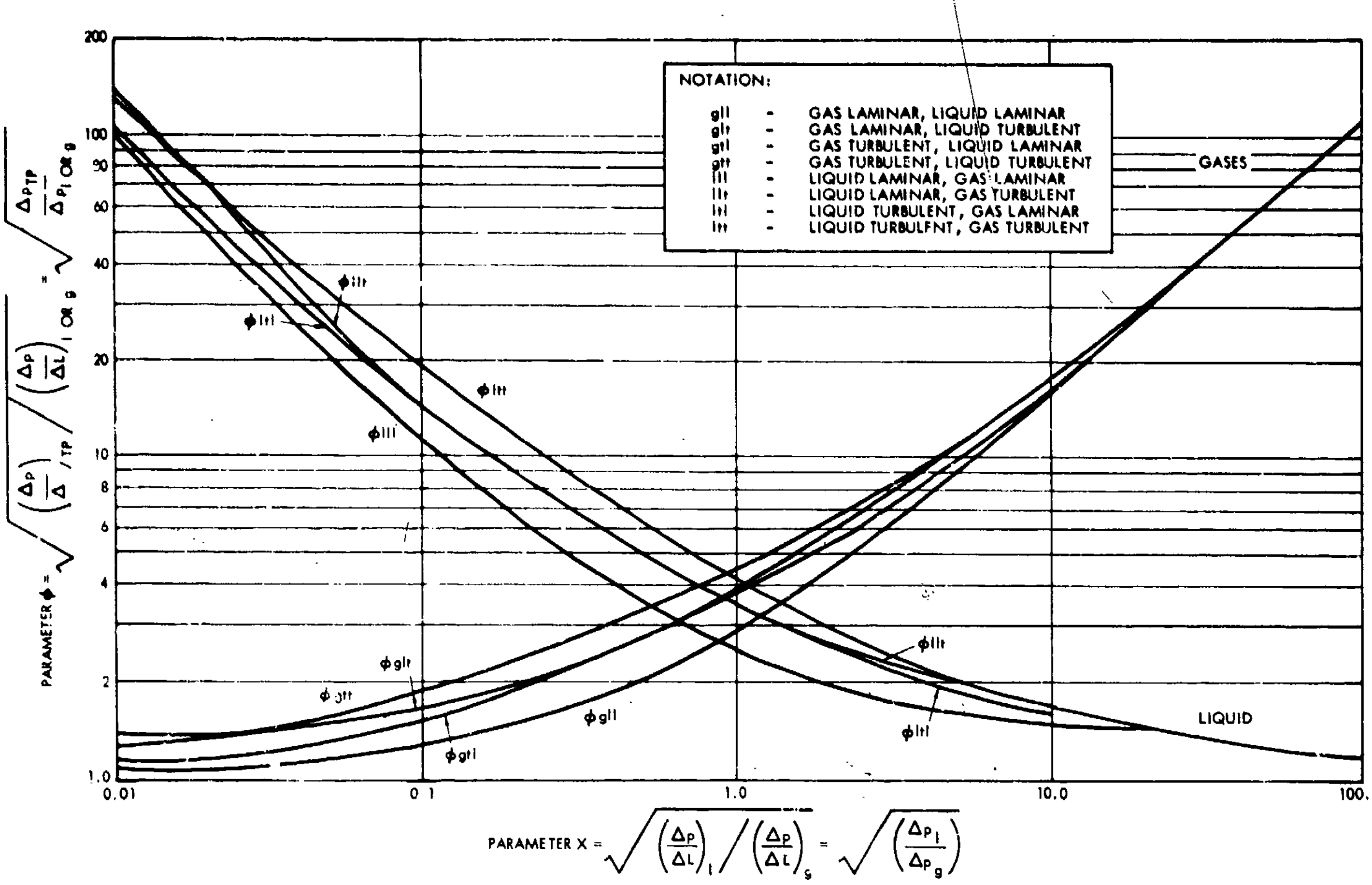


Figure 3.7.5. Parameters Ø and X for Horizontal, Two-Phase Flow

(Reprinted with permission from "Handbook of Fluid Dynamics," Streeter, Copyright 1961 by McGraw Hill Book Company)

The steps to be followed in the Lockhart-Martinelli procedure (Reference 213-1) are:

1. Calculate the Reynolds number for each phase assuming that it is the only phase flowing in the pipe.
2. From Table 3.7.5 find values of m, n, C_l, C_g and calculate X^2.
3. From Figure 3.7.5 find the corresponding ϕ and calculate the two-phase pressure loss from Equation (3.7.5a).

3.8 FLOW THROUGH METERING DEVICES AND VALVES

3.8.1 Derivation of Flow Meter Coefficients

The most versatile type of measuring device for fluid flow is the variable head flowmeter which includes orifices, flow nozzles, and venturi meters.

By Bernoulli's Equation (3.7.3.2d), the velocity head increases as the static pressure head decreases. If flow is frictionless and the velocity of approach is zero ($V_1 = 0$), typical of flow from a reservoir, the velocity of the orifice jet or the velocity at the venturi throat, V_2, may be expressed by the equation

(Eq. 3.8.1a)

$$V_2 = \sqrt{2g\left(\frac{P_1}{w} + Z_1 - \frac{P_2}{w} - Z_2\right)}$$

where P_1 and P_2 = pressures upstream and downstream, respectively lb_f/ft^2

Z_1 and Z_2 = the heights of the pressure gages in feet from a reference plane (see Figure 3.8.1a)

w = specific weight, lb_f/ft^3

If $Z_1 = Z_2$, Equation (3.8.1a) reduces to

$$V_2 = \sqrt{2g\frac{\Delta P}{w}} \quad \textbf{(Eq. 3.8.1b)}$$

If friction cannot be assumed negligible, the actual jet velocity is expressed as

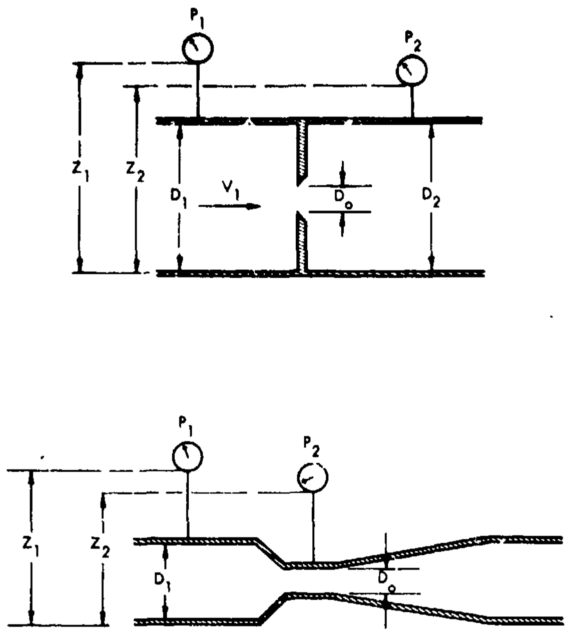


Figure 3.8.1a. Flow Measurement by Flow Meters with Gages

$$V_2 = c_v \sqrt{2g \frac{\Delta P}{w}} \qquad \textbf{(Eq. 3.8.1c)}$$

where c_v is the coefficient of velocity and is a measure of the extent to which friction retards velocity.

When streamlines converge approaching an orifice, they continue to converge beyond the orifice exit face. The point of minimum cross-sectional area is called the vena contracta (Reference 138-1). The relationship between the actual orifice area and the area of the jet at the vena contracta is

$$A_2 = c_c A_o \qquad \textbf{(Eq. 3.8.1d)}$$

where A_2 = area of the jet at the vena contracta
A_o = throat area of the orifice
c_c = contraction coefficient

Combining Equations (3.8.1c) and (3.8.1d), the volumetric flow rate may be obtained by the expression for continuity

(Eq. 3.8.1e)

$$Q = A_2 V_2 = c_c c_v A_o \sqrt{2g \frac{\Delta P}{w}}$$

Downstream pressure is henceforth constrained to be measured at the vena contracta of an orifice, or at the venturi throat.

The product $c_c c_v$ is defined as the discharge coefficient C_D, then

$$\boxed{Q = C_D A_o \sqrt{2g \frac{\Delta P}{w}}} \qquad \textbf{(Eq. 3.8.1f)}$$

If *the velocity of approach is not zero*, ($V_1 > 0$), it is necessary to multiply the discharge coefficient, C_D, presented in Equation (3.8.1f) by the approach factor

$$\frac{1}{\sqrt{1 - C_D^2 \left(\frac{D_o}{D_1}\right)^4}} \qquad \textbf{(Eq. 3.8.1g)}$$

where $\frac{D_o}{D_1}$ is a ratio of the throat diameter to the upstream diameter.

The volumetric flow rate becomes

$$Q = \frac{C_D A_o \sqrt{2g \frac{\Delta P}{w}}}{\sqrt{1 - C_D^2 \left(\frac{D_o}{D_1}\right)^4}} \qquad \textbf{(Eq. 3.8.1h)}$$

Since C_D must be determined from actual test data, it is simpler to establish one coefficient relating Q, ΔP, D_o and D_1. Calling this the modified discharge coefficient C, the volumetric flow rate is

(Eq. 3.8.1i)

$$\boxed{Q = \frac{C A_o}{\sqrt{1 - \left(\frac{D_o}{D_1}\right)^4}} \sqrt{2g \frac{\Delta P}{w}}}$$

C and C_D are very nearly the same value for small values of $\frac{D_o}{D_1}$, for instance, 0.5 or less.

In the case of an *orifice* and *flow nozzle* it is convenient to set

$$K_F = \frac{C}{\sqrt{1 - \left(\frac{D_o}{D_1}\right)^4}} \qquad \textbf{(Eq. 3.8.1j)}$$

where K_F is the flow coefficient. If $Z_1 = Z_2$ in Equation (3.8.1a), the volumetric flow rate is

$$\boxed{Q = K_F A_o \sqrt{2g \frac{\Delta P}{w}}} \qquad \textbf{(Eq. 3.8.1k)}$$

The weight flow rate is

$$\dot{w} = w K_F A_o \sqrt{2g \frac{\Delta P}{w}} \qquad \textbf{(Eq. 3.8.1l)}$$

The relation between C_D, C, and K_F as a function of $\frac{D_o}{D_1}$ for a sharp-edged orifice is shown in Figure 3.8.1b.

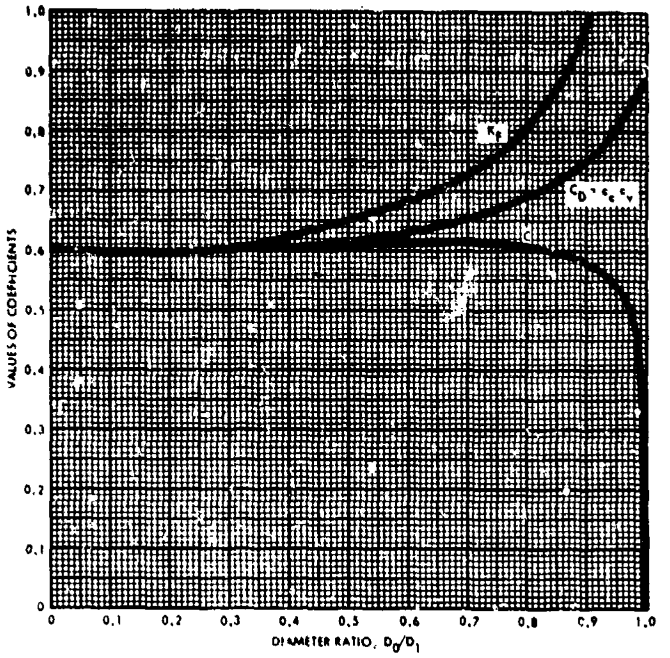


Figure 3.8.1b. Summary of Coefficients for Sharp-Edged Orifices

(Reprinted with permission from "Fluid Mechanics," Daugherty and Ingersoll, McGraw-Hill Book Company, Inc., 1954, p. 133)

The discharge coefficient C or C_D varies with orifice size, increasing slightly with decreasing orifice size. The increase is due primarily to a reduction in the velocity coming in from the side that produces jet contraction. Reduced jet contraction in smaller orifices is caused by the effects of adhesion, surface tension, capillarity, and wall friction. For the same diameter ratio, a coefficient of 0.60 for an orifice in a 15-inch pipe might be 0.61 in a 3-inch pipe.

The change in discharge coefficient with Reynolds number for an orifice is shown in Figure 3.8.1c. With low viscosity and high Reynolds number the coefficient is approximately constant. As viscosity is increased, with a corresponding decrease in Reynolds number, c_c increases from an initial low value until it equals 1.0. This discharge coefficient, therefore, increases with decreasing Reynolds number. When viscosity increases to a point where increasing friction (decreasing c_v) dominates, it causes the discharge coefficient to decrease rapidly with smaller values of Reynolds number.

3.8.2 Flow Nozzles and Orifices

3.8.2.1 GENERAL. A nozzle may be considered a formed channel in which the fluid velocity increases in the direction of flow. Although there are many flow nozzle designs, the ISA (International Standards Association) nozzle has become an acceptable standard. An orifice is a holed plate and may be used in the same manner as the flow nozzle. The best recognized orifice is the VDI or sharp-edged orifice.

Orifices and nozzles may be installed in pipe lines with flange taps, corner taps, vena contracta taps, and "1D and 1/2D taps" for pressure measurement. A discussion of orifices, nozzle design, and tap location is presented under Detailed Topic 5.17.5.2.

3.8.2.2 LIQUIDS. For an orifice discharging freely into the atmosphere from a reservoir, the velocity of approach is zero, the outlet pressure is zero psig, and the volumetric flow rate can be calculated by Equation (3.8.1f). The discharge coefficient depends upon the shape of the orifice, its size and upstream head. Figure 3.8.2.2a illustrates the effect of orifice shape for orifice sizes greater than 1 inch in diameter and with pressure heads greater than 4 feet. The discharge coefficients are constant for larger sizes and

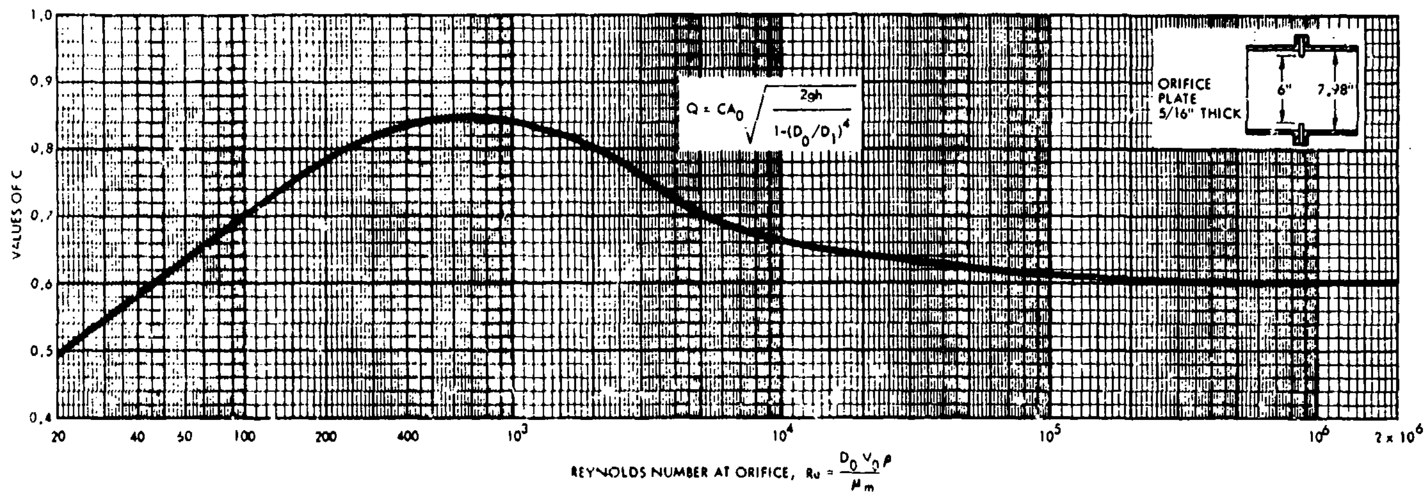
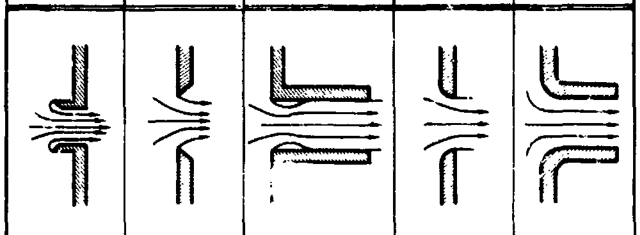


Figure 3.8.1c. Variations of Coefficient C with Reynolds Number

(Reprinted with permission from "Fluid Mechanics," Daugherty and Ingersoll, McGraw-Hill Book Company, Inc., 1954, p. 137)



	BORDA MOUTHPIECE	SHARP EDGE	SHORT TUBE (SQUARE)	ROUNDED EDGE	SHORT TUBE (ROUNDED)
c_v	0.98	0.98	0.80	0.98	0.98
c_c	0.52	0.62	1.00	1.00	1.00
C_D	0.51	0.61	0.80	0.98	0.98

Figure 3.8.2.2a. Discharge Coefficients for Several Orifice Shapes

higher pressure heads. For smaller sizes and lower heads, orifices should be calibrated in place.

The volumetric flow rate for *liquid flow*, with orifices or nozzles in pipelines, is calculated by the equation

(Eq. 3.8.2.2a)

$$Q = K_F A_o \sqrt{2g\left(\frac{P_1}{w} + Z_1 - \frac{P_2}{w} - Z_2\right)}$$

where

w = specific weight, lb_f/ft^3
Q = volumetric flow rate, ft^3/sec
P_1, P_2 = upstream and downstream pressures, respectively, lb_f/ft^2
K_F = flow coefficient
A_o = throat area, ft^2
Z_1, Z_2 = gage elevation above reference plane, ft
g = local acceleration of gravity, ft/sec^2

Pressure, P_2, is measured at the vena contracta. The expansion of the fluid from the vena contracta into a downstream pipe equal in diameter to the upstream pipe affords some static pressure recovery. The difference between the upstream pressure and the pressure downstream of the vena contracta is equal to the head loss due to friction, h_f, which appears in Equation (3.7.3.2e).

If $Z_1 = Z_2$, the equation reduces to

$$Q = K_F A_o \sqrt{2g \frac{\Delta P}{w}} \qquad \textbf{(Eq. 3.8.2.2b)}$$

Values of K_F as functions of Reynolds number and diameter ratio are presented for nozzles in Figure 3.8.2.2b; for orifices in Figures 3.8.2.2c, d. For computation of the Reynolds number, use inlet pipe parameter values rather than those of the nozzle or orifice throat.

Comparing Figures 3.8.2.2b and 3.8.2.2d, it is seen that the flow coefficients for orifices and nozzles vary differently with Reynolds number, particularly at values less than 2×10^5. The difference between orifices and nozzles is that for nozzles there is no area contraction; thus jet area is also the area of the throat and is fixed. For an orifice, the area of the jet is a variable, and generally less than the throat area. For nozzles, the discharge coefficient is practically the velocity coefficient, c_v, but for orifices the discharge coefficient is primarily affected by the contraction coefficient, c_c. Since c_v is related to friction, the flow coefficient for nozzles increases with increasing Reynolds number. For nozzles, c_v is usually between values of 0.97 and 0.99; for orifices, values may range between 0.95 and 0.994. For nozzles, the c_c is usually 1.0; for orifices, the c_c may range between 0.50 and 1.00. Thus, the discharge coefficient for orifices is usually less than that for nozzles.

Lists of discharge coefficients for various tap locations can be found in Reference 68-1.

3.8.2.3 GASES. The mass flow rate of a frictionless, perfect gas with a negligible upstream velocity flowing subsonically through an orifice or nozzle can be calculated by

(Eq 3.8.2.3a)

$$\dot{m} = A_o \rho_1 \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} \sqrt{2g_c RT_1 \frac{\gamma}{\gamma - 1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

when $\frac{P_2}{P_1} > \frac{P_c}{P_1}$

where $\dot{m}$ = mass flow rate, lb_m/sec
A_o = throat area, ft^2
ρ_1 = density, lb_m/ft^3
R = gas constant, $ft\text{-}lb_f/lb_m\ ^\circ R$
T_1 = absolute temperature, °R
γ = ratio of specific heats
P_1 = upstream pressure, lb_f/ft^2
$g_c = 32.2 \frac{lb_m}{lb_f} ft/sec^2$
P_2 = downstream pressure, lb_f/ft^2
P_c = critical pressure, lb_f/ft^2

To include friction losses and, primarily, contraction effects the geometric orifice area, A_o, should be replaced by the effective orifice area, $C_D A_o$, which is sometimes written A_{eff}. For nozzles, $C_D = 0.95$; for sharp-edged and square-edged orifices, $C_D = 0.60$ to 0.843 and 0.82, respectively. With nozzles, C_D is independent of the pressure ratio. With square-edged orifices C_D is virtually constant for orifice lengths twice the diameter, (Reference 408-1).

The effects of small pressure differences may be found from Equation (3.8.2.3a) by expansion of the pressure ratios in an infinite series of the form $(a - x)^n$.

Rewriting Equation (3.8.2.3a) in the form **(Eq. 3.8.2.3b)**

$$\dot{m} = A_o \rho_1 \sqrt{2g_c \frac{\gamma}{\gamma - 1} RT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}$$

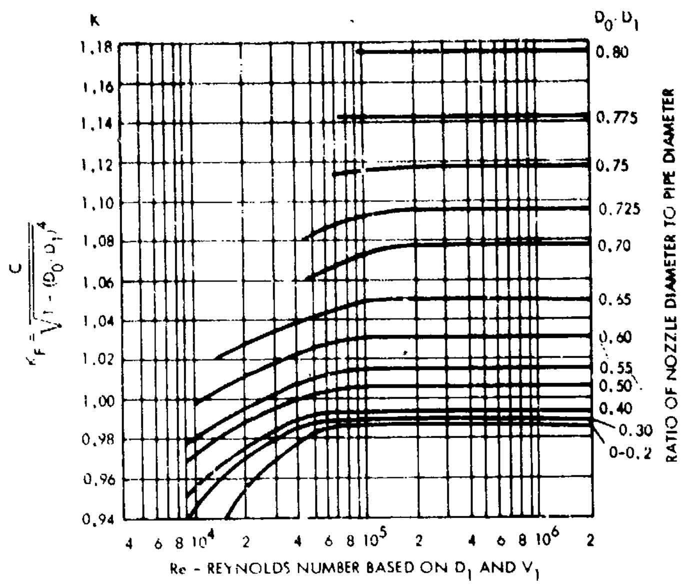


Figure 3.8.2.2b. Flow Coefficient K_F for Various Nozzle Ratios
(From Crane Co., Chicago, Illinois, Paper No. 409, 1942, and NACA Technical Memo 952.)

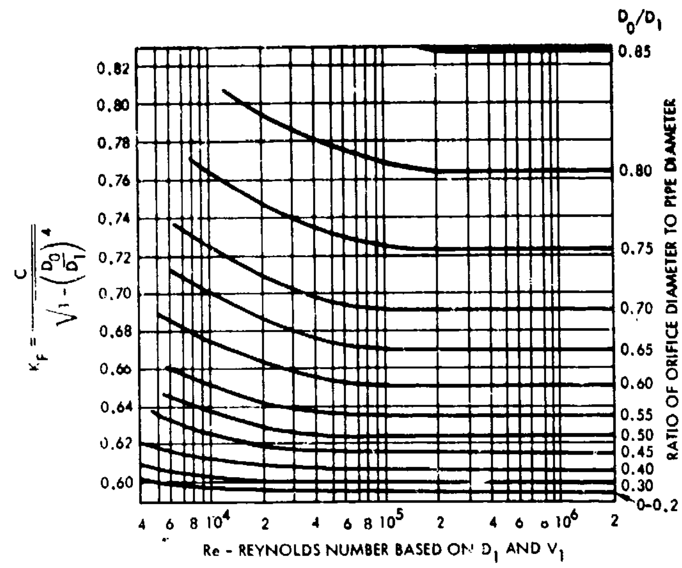


Figure 3.8.2.2d. Flow Coefficient K_F for Orifices
(Adapted from NACA Technical Memo 952, and "Instruments," November 1933, vol. 6, no. 11, G. L. Tuve and R. E. Sprenkle, Instruments Publishing Company, Illinois)

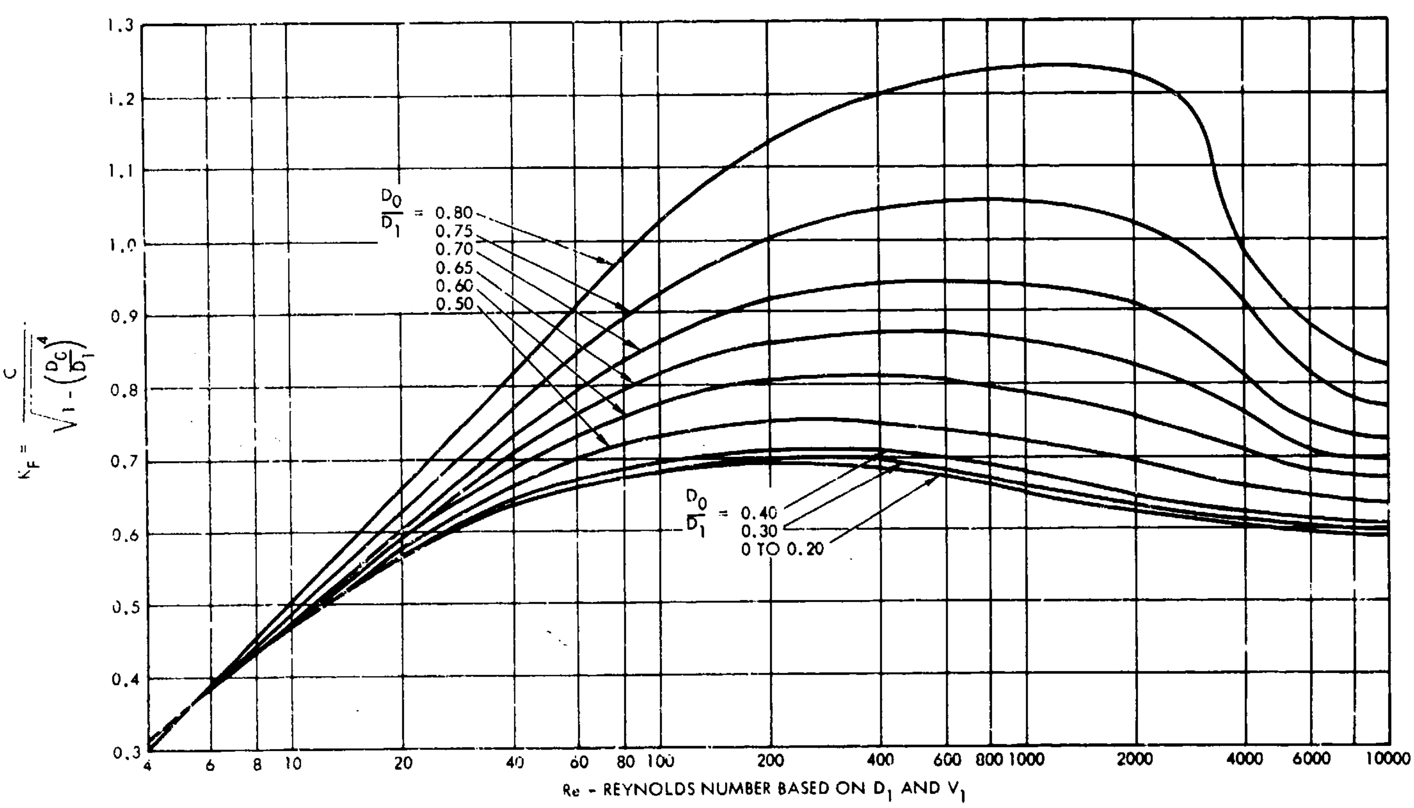


Figure 3.8.2.2c. Flow Coefficient K_F for Orifices
(Adapted from NACA Technical Memo 952, and "Instruments," November 1933, vol. 6, no. 11, G. L. Tuve and R. E. Sprenkle, Instruments Publishing Company, Illinois)

ISSUED: MARCH 1967
SUPERSEDES: MAY 1964

and letting $P_2 = P_1 - \Delta P$, the pressure ratios expanded in a series become

(Eq 3.8.2.3c)

$$\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} = 1 - \frac{2}{\gamma}\frac{\Delta P}{P_1} + \frac{2-\gamma}{\gamma^2}\frac{\Delta P^2}{P_1}$$

(Eq 3.8.2.3d)

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}} = 1 - \frac{\gamma+1}{\gamma}\frac{\Delta P}{P_1} + \frac{\gamma+1}{2\gamma^2}\frac{\Delta P^2}{P_1}$$

Neglecting terms higher than those squared, and subtracting, the mass flow rate becomes

(Eq 3.8.2.3e)

$$\dot{m} = A_o\rho_1\sqrt{2g_cRT_1\left[\frac{\Delta P}{P_1}\left(1 - \frac{3}{2\gamma}\frac{\Delta P}{P_1}\right)\right]}$$

If *V_1 is not negligible*, a correction factor similar to that used in liquid flow is used. The mass flow rate becomes

(Eq 3.8.2.3f)

$$\dot{m} = \frac{A_o\rho_1\left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}}\sqrt{2g_c\frac{\gamma}{\gamma-1}RT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}}{\sqrt{1 - \left(\frac{A_o}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}}}}$$

If $\frac{P_2}{P_1} = 1$, there is no flow. As $\frac{P_2}{P_1}$ decreases to its critical value, the flow rate will increase. Decrease of the critical pressure ratio beyond that required for sonic flow results in no increase in flow through a nozzle. Critical flow through an orifice is complicated by jet area contraction. To compensate for friction losses, Equation (3.8.2.8f) is multiplied by a correction factor called the discharge coefficient, C_D. At the *critical pressure* ratio, by substitution of critical values for pressure and temperatures from Equations (3.3.8.9a) and (3.3.8.9b) of Sub-Section 3.3 into Equation (3.8.2.5a), the mass flow rate for flow at sonic velocity becomes

(Eq 3.8.2.3g)

$$\dot{m} = A_oP_1\sqrt{\frac{g_c\gamma}{RT_1}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

$$\text{when } \frac{P_2}{P_1} \leq \frac{P_c}{P_1}$$

where $\dot{m}$ = mass flow rate, lb_m/sec
A_o = throat area, ft^2
R = gas constant, ft-lb_f/lb_m °R
P_1 = upstream pressure, lb_f/ft^2
T_1 = upstream absolute temperature, °R
γ = ratio of specific heats
$g_c = 32.2 \frac{lb_m}{lb_f}$ ft/sec²

Multiplication of $\dot{m}$ by $\frac{g}{g_c}$ results in the weight flow rate, $\dot{w}$, in lb_f/sec. Division of $\dot{m}$ by the density of the gas at standard conditions gives the volume flow rate in ft³/sec at standard conditions. Since computations of equations presented above are long and tedious, it has become common practice to use Equation (3.8.11) multiplied by an expansion factor, Y. The weight flow rate of gas for *subsonic flow* may then be expressed as

(Eq 3.8.2.3h)

$$\boxed{\dot{w} = K_FA_ow_1Y\sqrt{\frac{2g\,(P_1 - P_2)}{w_1}}}$$

where Y = expansion factor (from Figure 3.8.2.3)
$\dot{w}$ = weight flow rate, lb_f/sec
K_F = flow coefficient
w_1 = upstream specific weight, lb_f/ft^3
P_1, P_2 = upstream and downstream pressures, respectively, lb_f/ft^2
A_o = throat area, ft²

Note that in liquid flow which is assumed to be incompressible, Y = 1. Pressure P_2 is measured at the vena contracta.

Values of Y are given in Figure 3.8.2.3 as functions of γ, pressure difference, and diameter ratio. As approximations, the values of K_F for orifices are 0.60 to 0.62; for flow nozzles, 0.97.

For sonic flow, the pressure ratio in the expansion factor can never be less than the critical pressure ratio, P_c/P_1. As with the mass flow rate calculations, a correction for velocity of approach should be applied if the orifice area is more than five percent of the cross-sectional area of the approaching pipe or duct. This correction may be made by multiplying $\dot{w}$ by F_v

$$F_v = \frac{1}{\sqrt{1 - \left(\frac{A_o}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}}}} \qquad \text{(Eq 3.8.2.3i)}$$

Where F_v = approach velocity correction factor
A_o = throat area, ft²
A_1 = upstream pipe area, ft²

For sonic flow, a similar factor Y* may be used with a variation of Equation 3.8.2.3g, to yield the expression

$$\boxed{\dot{w} = K_FA_oY^*\frac{P_1}{\sqrt{RT}}} \qquad \text{(Eq 3.8.2.3j)}$$

$$\text{where } Y^* = \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\left(\frac{2g\gamma}{\gamma+1}\right)^{\frac{1}{2}} \qquad \text{(Eq 3.8.2.3k)}$$

Values of Y* are given in Table 3.8.2.3. Additional data may be found in References 49-17, 68-1, 212-2, and 410-1.

Table 3.8.2.3. Values of Y* at Critical Flow
(Reference 49-17)

γ	$\frac{P_c}{P_1}$	$\frac{Y^*}{\sqrt{2g}}$
1.66	0.488	0.496
1.40	0.528	0.480
1.30	0.546	0.470

3.8.2.4 ORIFICES IN SERIES. In a number of fluid components two orifices occur in series separated by a reservoir of limited volume, as shown in Figure 3.8.2.4a. With two orifices, four types of flow may exist:

Type 1 — Critical (sonic) flow through both orifices

Type 2 — Subcritical flow through the first orifice and critical flow through the second

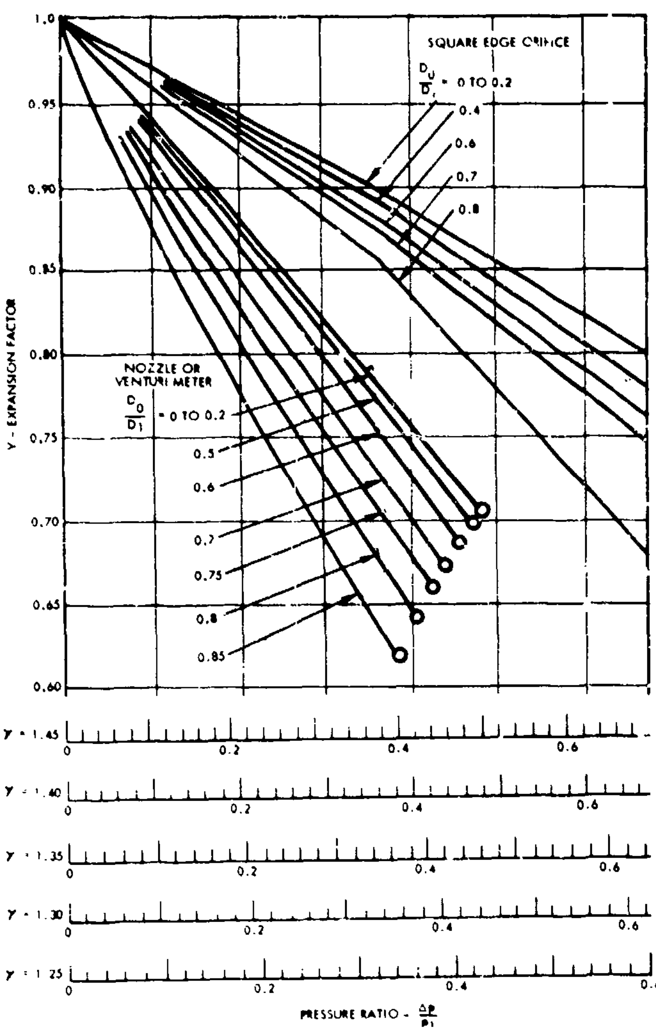


Figure 3.8.2.3. Expansion Factor Y

(Reprinted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe," Crane Co., Chicago: Technical Paper No. 410, 1957; data extracted from "Fluid Meters, Their Theory and Application," Fourth Edition, 1937, and "Orifice Meters with Supercritical Flow," by R. G. Cunningham, with permission of the publisher, The American Society of Mechanical Engineers, United Engineering Center 345 East 47th St., New York, N. Y. 10017.)

Type 3 — Critical flow through the first orifice and subcritical flow through the second

Type 4 — Subcritical flow through both orifices.

To determine the type of flow existing, it is necessary to find the ratio of the product of discharge coefficients, and the areas and ratios of upstream and downstream pressures. With these ratios as coordinates, Figure 3.8.2.4a will define the type of flow.

For Type 1 flow, the weight flow rate for the first orifice is

(Eq 3.8.2.4a)

$$\dot{w}_a = C_{D_{(a)}} A_{o_{(a)}} P_{t_{(1)}} \sqrt{\frac{\gamma g_c}{T_{t_{(1)}} R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

and for the second orifice

(Eq 3.8.2.4b)

$$\dot{w}_b = C_{D_{(b)}} A_{o_{(b)}} P_{t_{(res)}} \sqrt{\frac{\gamma g_c}{T_{t_{(res)}} R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

where
C_D = orifice discharge coefficient
A_o = throat area, ft^2
$P_{t_{(1)}}$ = upstream total pressure, lb$_f$/ft^2
γ = ratio of specific heats
R = gas constant, ft-lb$_f$/lb$_m$ °R
$T_{t_{(1)}}$ = upstream total temperature, °R
$T_{t_{(res)}}$ = reservoir total temperature, R
$P_{t_{(res)}}$ = reservoir total pressure, lb$_f$/ft^2

$$= P_t \frac{C_{D_{(a)}} A_{o_{(a)}}}{C_{D_{(b)}} A_{o_{(b)}}}$$

For Type 2 flow, the weight flow rate for the first orifice becomes

(Eq 3.8.2.4c)

$$\dot{w}_a = C_{D_{(a)}} A_{o_{(a)}} \rho_1 \left(\frac{P_{t_{(res)}}}{P_{t_{(1)}}}\right)^{\frac{1}{\gamma}} \sqrt{2g_c \frac{\gamma}{\gamma-1} RT_{t_{(1)}} \left[1 - \left(\frac{P_{t_{(res)}}}{P_{t_{(1)}}}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

and for the second orifice becomes

(Eq 3.8.2.4d)

$$\dot{w}_b = C_{D_{(b)}} A_{o_{(b)}} P_{t_{(res)}} \sqrt{\frac{g_c\gamma}{RT_{t_{(res)}}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

Values of $P_{t_{(res)}}$ may be found from Figure 3.8.2.4b.

For Type 3 flow, the weight flow rate for the first orifice becomes

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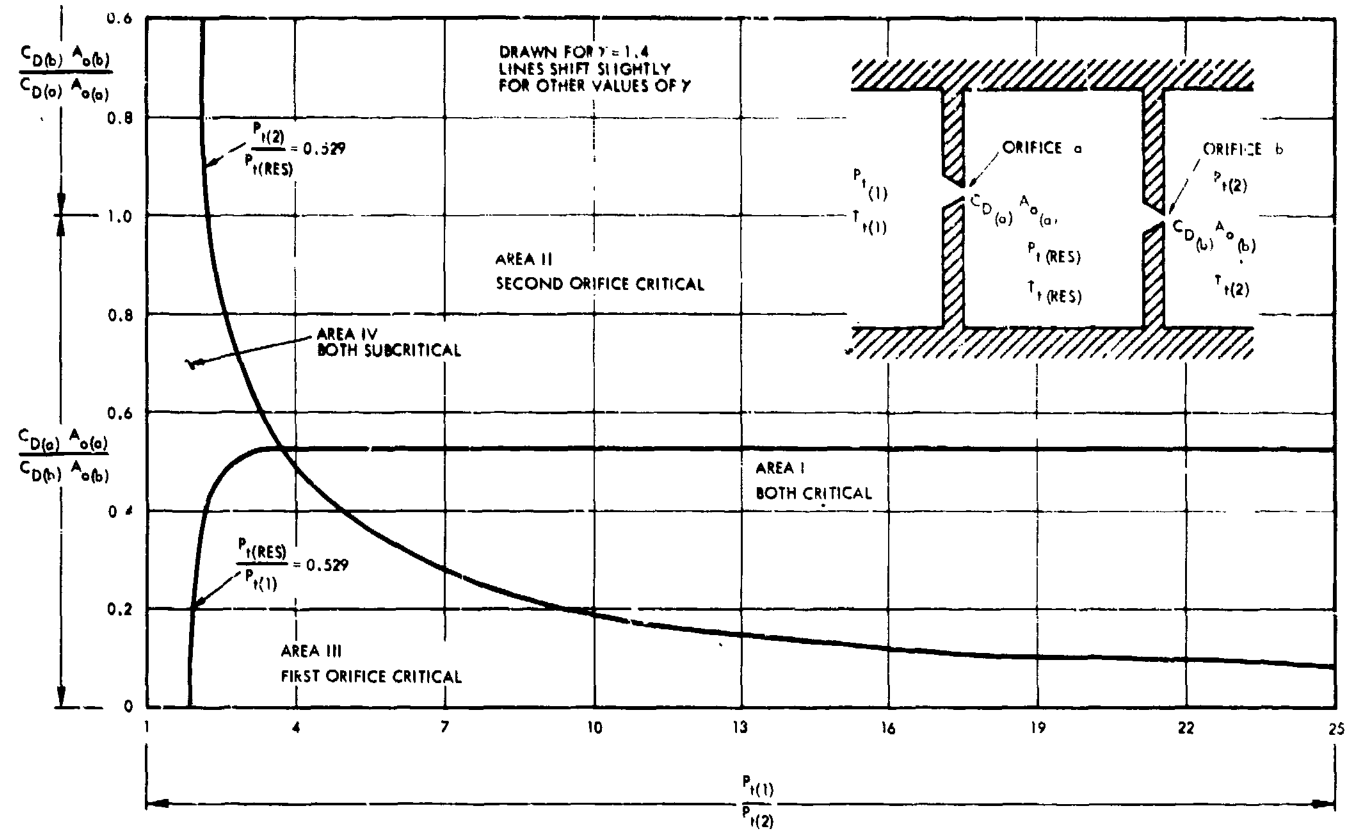

Figure 3.8.2.4a. Flow Regions for Two Orifices in Series

(Eq 3.8.2.4e)

$$\dot{w}_a = C_{D_{(a)}} A_{o_{(a)}} P_{t_{(1)}} \sqrt{\frac{\gamma g_c}{RT_{t_{(1)}}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

and for the second orifice becomes

(Eq 3.8.2.4f)

$$\dot{w}_b = C_{D_{(b)}} A_{o_{(b)}} \rho_{res} \left(\frac{P_{t_{(2)}}}{P_{t_{(res)}}}\right)^{\frac{1}{\gamma}} \sqrt{2g_c \frac{\gamma}{\gamma-1} RT_{t_{(res)}}\left[1-\left(\frac{P_{t_{(2)}}}{P_{t_{(res)}}}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

Values of $P_{t_{(res)}}$ may be determined from Figure 3.8.2.4c.

For Type 4 flow, flow rates may be determined by Equation (3.8.2.3a) using the appropriate values of upstream pressure. Reservoir pressures may be approximated by the equation

(Eq 3.8.2.4g)

$$P_{t_{(res)}} = \frac{1}{2}\left[P_{t_{(1)}} - \phi^2 P_{t_{(2)}} + \sqrt{4\phi^2 P_{t_{(2)}}^2 + (\phi P_{t_{(2)}} - P_{t_{(1)}})^2}\right]$$

where $\phi = \frac{C_{D_{(b)}} A_{o_{(b)}}}{C_{D_{(a)}} A_{o_{(a)}}}$

If the *orifices are identical,* this technique may be expanded to include a series of orifices (Reference 31-4 and Figure 3.8.2.4d). Two types of flow may exist:

Type 1 — All flow will be subcritical

Type 2 — The last orifice will have critical flow.

For Type 1 flow, the continuity equation applied to any two orifices may be expressed as

(Eq 3.8.2.4h)

$$\left(\frac{1}{r_a}\right)^2\left[r_a^{\frac{2}{\gamma}}\left(1-r_a^{\frac{\gamma-1}{\gamma}}\right)\right] = \left[r_b^{\frac{2}{\gamma}}\left(1-r_b^{\frac{\gamma-1}{\gamma}}\right)\right]$$

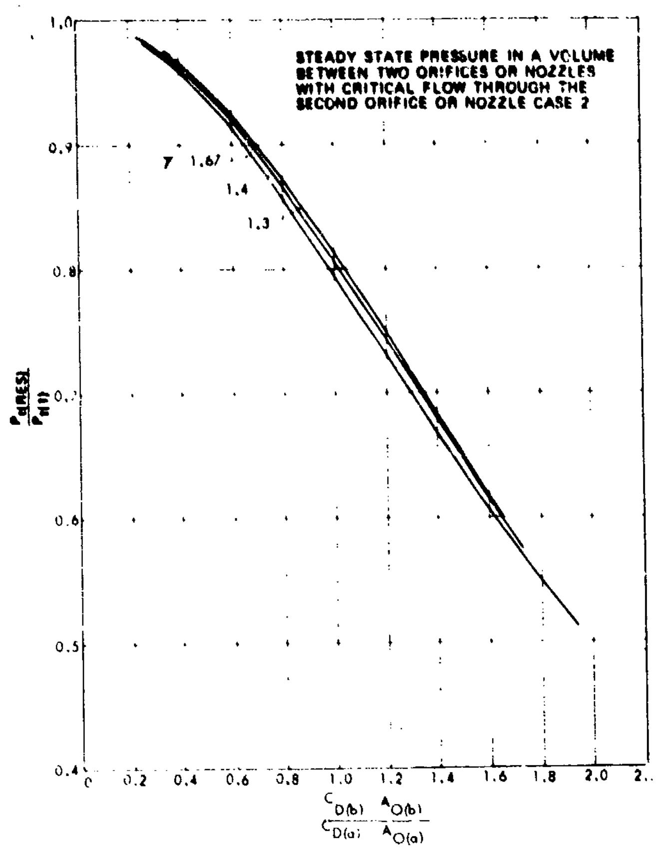

Figure 3.8.2.4b. Reservoir Volume Pressure, Type-2 Flow

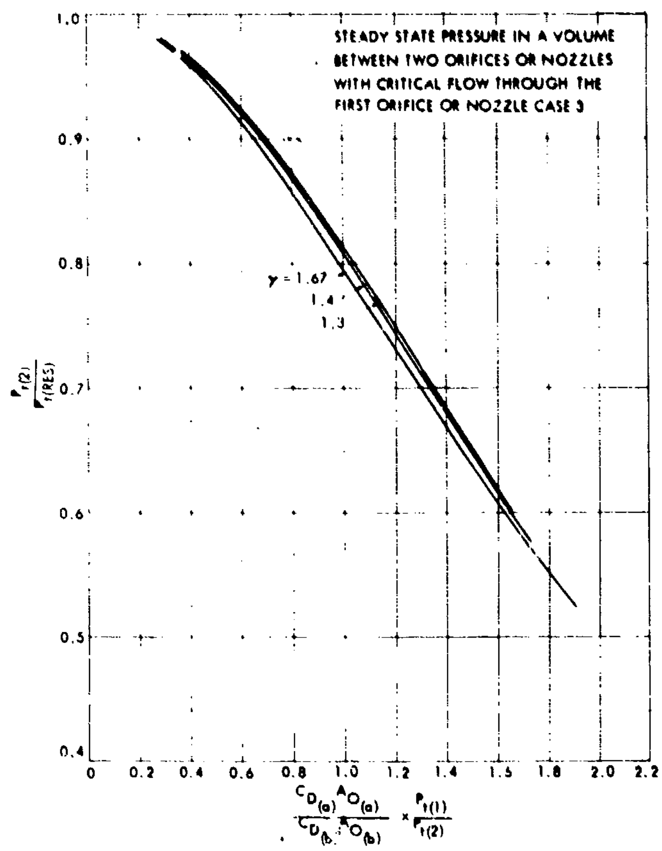

Figure 3.8.2.4c. Reservoir Volume Pressure, Type-3 Flow

where $r_a = \dfrac{P_{a(2)}}{P_{a(1)}}$

$r_b = \dfrac{P_{b(2)}}{P_{b(1)}}$

For air $\gamma = 1.40$ and the values of the equation above may be found in Figure 3.8.2.4d for any pressure ratio to the critical value. In general, the curves may be applied to all gases and even to saturated steam ($\gamma = 1.10$). Usually only the inlet and outlet pressures are known; but by trial computation a sequence of pressure ratios may be derived the product of which is the overall pressure ratio.

If the pressure ratio across the last orifice is known, the pressure ratio for the next to the last orifice may be found from Figure 3.8.2.4e. By working backwards, the pressure ratio across all the orifices may be found as a product of the individual pressure ratios.

For Type 2 flow, the pressure ratio across the last orifice is critical, thus

$$r_c = \left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{1-\gamma}} \qquad \text{(Eq 3.8.2.4i)}$$

where $r_c = \dfrac{P_{b(2)}}{P_{b(1)}}$

Just as the last orifice is insensitive to reduction in downstream pressure below that for critical flow, so is the complete orifice series. The effect of the number of orifices in series on the critical pressure ratio (P_{outlet}/P_{inlet}) is shown in Figure 3.8.2.4f (Reference 31-4).

3.8.2.5 TWO-PHASE FLOW. The flow rates and pressure drop relations for two-phase flow are at best a compromise of the single phase equations (Reference 60-13). For an *ideal orifice*, with negligible velocity of approach and $K_F = 1.0$, the weight flow rates of liquid and gas are related as follows (Reference 60-13):

(Eq 3.8.2.5a)

$$\frac{\dot{w}_l}{\sqrt{2g\,\Delta p_{TP}\,w_l}} + \frac{\dot{w}_g}{\sqrt{2g\,\Delta p_{TP}\,w_g}} = A_o$$

where $\dot{w}_l, \dot{w}_g$ = weight flow rates of the liquid and gas, respectively, lb_f/sec

A_o = throat area, ft^2

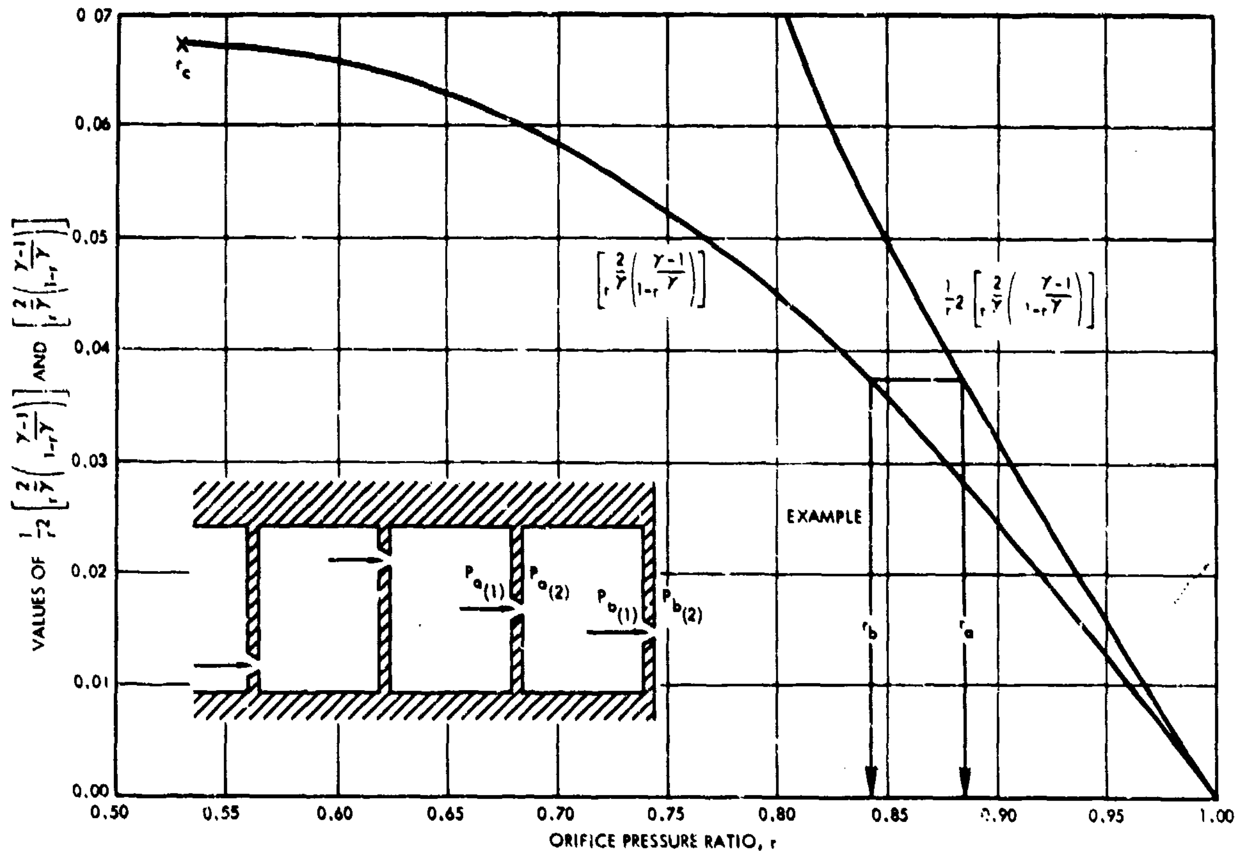

Figure 3.8.2.4d. Pressure Ratio for Successive Orifices

(Reprinted with permission from "Journal of Applied Mechanics," December 1948, vol. 15, no. 4, C. S. L. Robinson, ASME, New York)

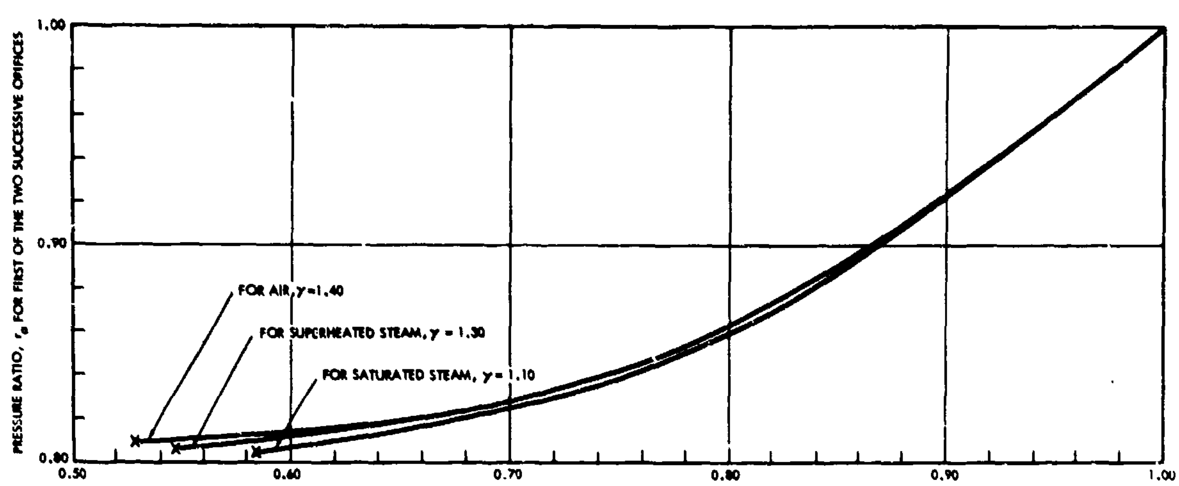

Figure 3.8.2.4e. Pressure Ratio for Any Two Consecutive Orifices

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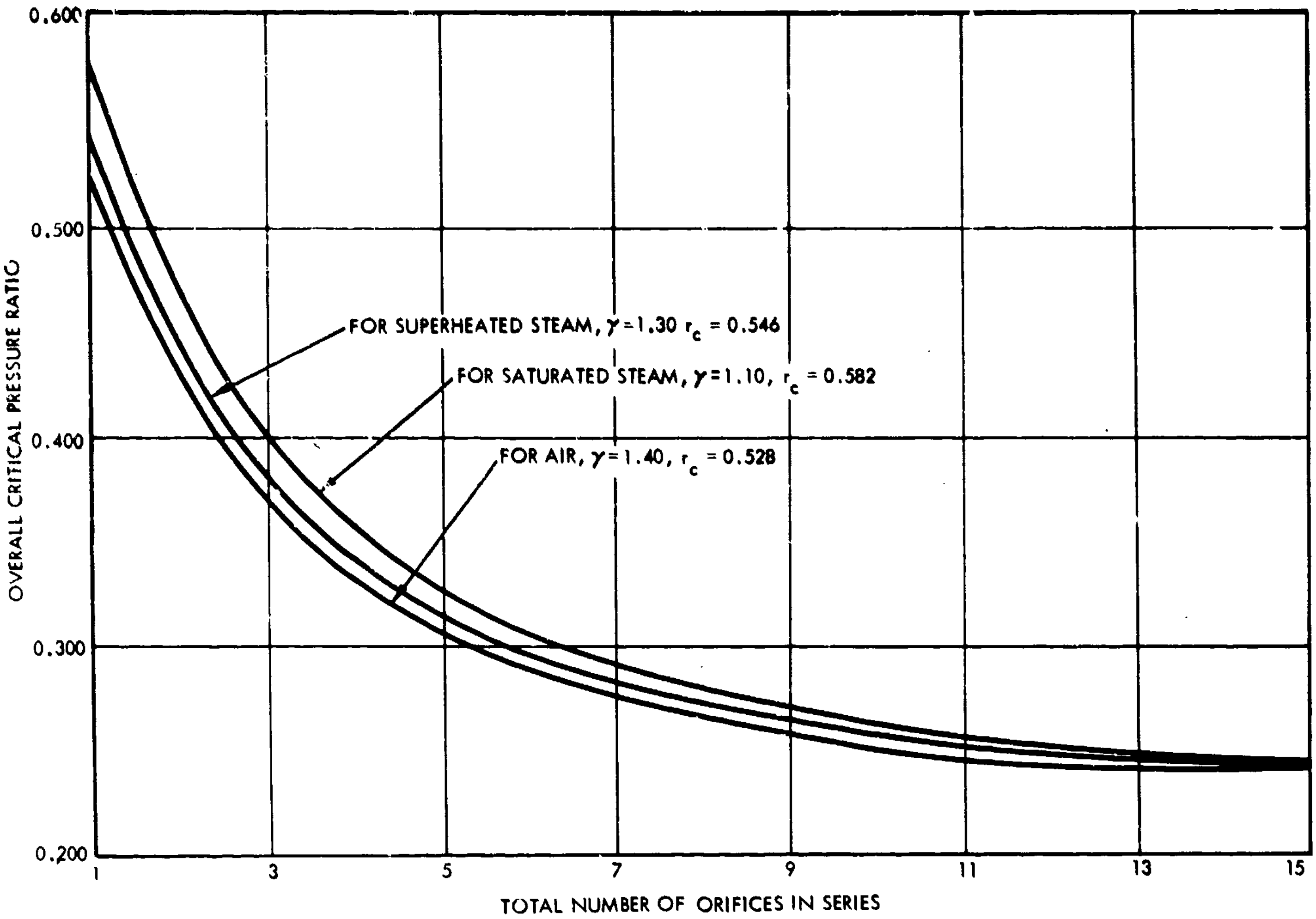

Figure 3.8.2.4f. Overall Critical Pressure Ratio
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Δp_{TP} total pressure drop due to both liquid and gas, lb_f/ft^2

w_l, w_g specific weights of liquid and gas, respectively, lb_f/ft^3

For an *actual orifice* (Reference 60-13), the total weight flow rate is

(Eq 3.8.2.5b)

$$\dot{w} = \frac{(K_{F_g} Y_g)_{TP} A_o \sqrt{2g \Delta p_{TP} w_g \left[\frac{\dot{w}_l}{\dot{w}_g} + 1\right]}}{1 + \frac{(K_{F_g} Y_g)_{TP} \dot{w}_l}{(K_{F_l})_{TP} \dot{w}_g} \sqrt{\frac{w_g}{w_l}}}$$

where $\dot{w}$ total flow rate, lb_f/sec

K_{F_l}, K_{F_g} flow coefficients of liquid and gas flows, respectively

A_o throat area, ft^2

Y_g = expansion factor (Figure 3.8.2.3)

w_l, w_g = specific weights of liquid and gas, respectively, lb_f/ft^3

Δp_{TP} = total pressure drop due to both liquid and gas, lb_f/ft^2

Values of K_{F_g} and K_{F_l} are functions of Reynolds numbers which are calculated as if each fluid (gas or liquid) were the only fluid passing through the orifice at the time. Figures 3.8.2.2c and d may be used to find the flow coefficients, K_{F_g} and K_{F_l}.

The subject of two-phase flow measurement with an orifice is well presented in References 60-13 and 68-5.

3.8.3 Venturi Meters

A venturi meter consists of a convergent section, throat, and diffuser. The throat diameter is usually one-fourth to one-half the inlet pipe diameter. For flow in one direction with minimum loss, the convergent inlet cone's included angle should be approximately 21° with a diffuser cone's included angle of 5° to 9°, assuming very smooth surfaces. For flow in either direction the inlet cone and diffuser should be identical. If the entrance cone angle and/or diffuser angle is too large, the fluid separates from the venturi boundary and a formation of eddies develops. This increased turbulence results in a change of flow energy into thermal energy which cannot be recovered in useful form. As with nozzles and orifices, the nature of flow entering the venturi affects hydraulic performance. It is important that the venturi be preceded by a straight pipe of sufficient length to permit a normal flow pattern.

The flow rate may be determined from equal elevation pressure measurements at the throat and upstream of the convergent section. The volume flow rate equation for a venturi is stated

$$Q = CA_0\sqrt{\frac{2g\,\Delta P}{w\left[1-\left(\frac{D_0}{D_1}\right)^4\right]}} \qquad \textbf{(Eq 3.8.3a)}$$

where Q = flow rate, ft^3/sec
A_0 = throat area, ft^2
D_0 = throat diameter, ft
D_1 = inlet diameter, ft
w = specific weight, lb_f/ft^3
ΔP = pressure difference, lb_f/ft^2
C = modified flow coefficient

The weight flow rate is

$$\dot{w} = \frac{C\,A_0\,w}{\sqrt{1-\left(\frac{D_0}{D_1}\right)^4}}\sqrt{2g\frac{\Delta P}{w}} \qquad \textbf{(Eq 3.8.3b)}$$

The values of C may be assumed to be about 0.99 for large venturis, and for small venturis, 0.97 to 0.98, provided the Reynolds number is greater than 10^5. For cavitating venturis, C is approximately 0.93, since the throat area is not completely available for liquid but is enveloped by a vapor. Values of C versus Reynolds number for $\frac{D_0}{D_1} = 0.5$ are shown in Figure 3.8.3. This diagram is also reasonably accurate for smaller diameter ratios.

For larger diameter ratios, C decreases slightly. For example, if $\frac{D_0}{D_1} = 0.75$, the values of C are about 1 percent less than shown in Figure 3.8.3. An analytical method of computation is presented in Reference 26-115.

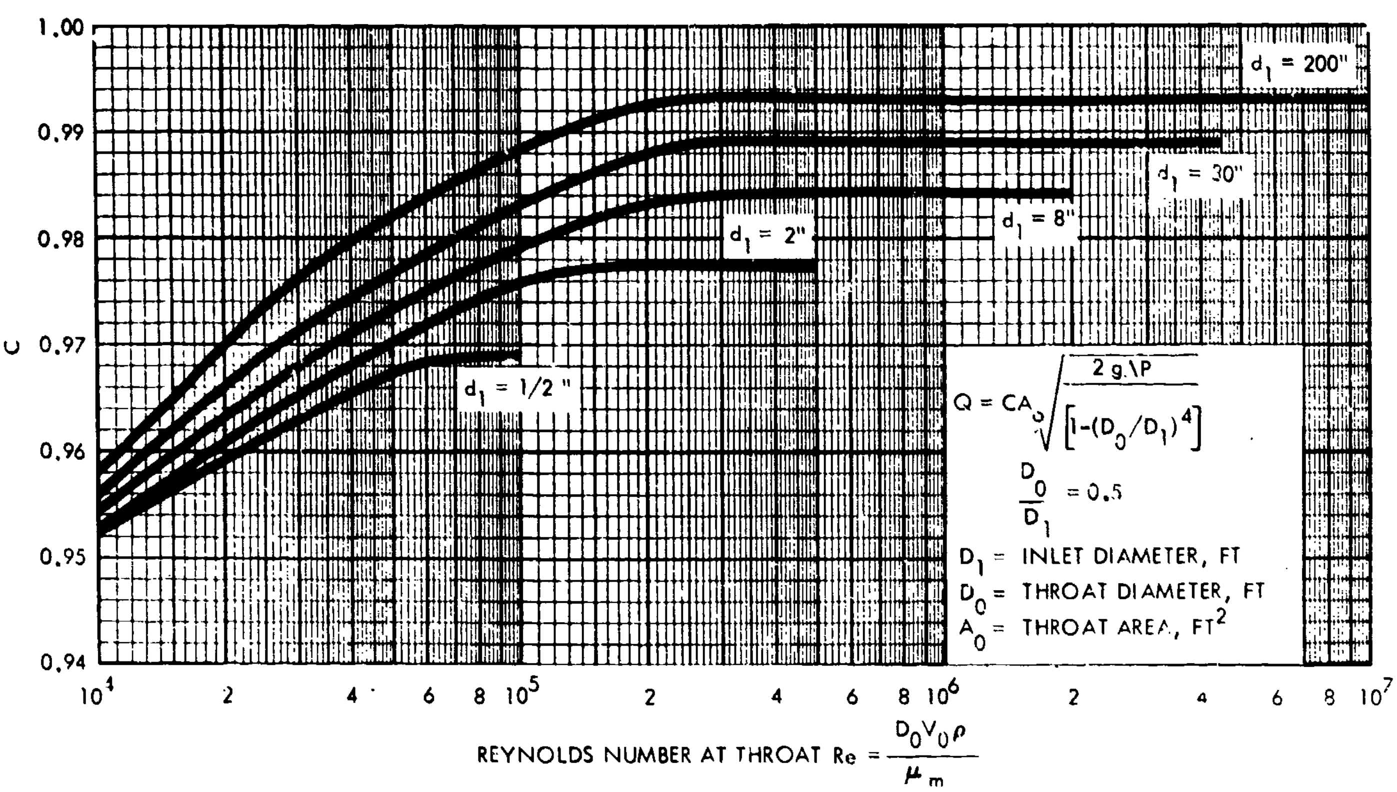

Figure 3.8.3. Coefficients for Venturi Meters
(Adapted from "Fluid Mechanics," Daugherty and Ingersoll, McGraw-Hill Book Company, Inc., 1954, p. 129)

Table 3.8.4. Conversion Factors for D_o, C_v, F, and K

(Adapted from "Product Engineering," D. Lapera and J. Yeaple, January 21, 1963, Copyright 1963 by McGraw-Hill Publishing Company, Inc.)

		D_o	C_v	F	K: $\frac{P_2}{P_1} = 1.0$	K: $\frac{P_2}{P_1} = 0.75$	K: $\frac{P_2}{P_1} = 0.5$
D_o			$= 0.236\sqrt{C_v}$	$= 0.316\sqrt{F}$	$= 1.456\frac{\sqrt{A}}{K^{1/4}}$	$= 1.521\frac{\sqrt{A}}{K^{1/4}}$	$= 1.641\frac{\sqrt{A}}{K^{1/4}}$
C_v		$= 18.0 D_o^2$		$= 1.8F$	$= 38.2\frac{A}{\sqrt{K}}$	$= 41.5\frac{A}{\sqrt{K}}$	$= 48.3\frac{A}{\sqrt{K}}$
F		$= 10 D_o^2$	$= 0.556\ C_v$		$= 21.2\frac{A}{\sqrt{K}}$	$= 23.1\frac{A}{\sqrt{K}}$	$= 26.9\frac{A}{\sqrt{K}}$
K	$\frac{P_2}{P_1} = 1.0$	$= 4.5\frac{A^2}{D_o^4}$	$= 1460\frac{A^2}{C_v^2}$	$= 450\frac{A^2}{F^2}$	NOTE: The K factor varies with P_2/P_1 and A and you must know which values the manufacturer used to derive his published K. For example, if K was derived at $P_2/P_1 = 0.75$ and valve inlet pipe area A = 0.2, in² then $F = 23.1 \times 0.2 / \sqrt{K} = 4.62 / \sqrt{K}$. Discharge coefficient, $C_o = 0.60$.		
K	$\frac{P_2}{P_1} = 0.75$	$= 5.36\frac{A^2}{D_o^4}$	$= 1725\frac{A^2}{C_v^2}$	$= 534\frac{A^2}{F^2}$			
K	$\frac{P_2}{P_1} = 0.5$	$= 7.29\frac{A^2}{D_o^4}$	$= 2330\frac{A^2}{C_v^2}$	$= 724\frac{A^2}{F^2}$			

Note: In this table, A = in² and D_o = in.

3.8.4 Valves

The flow of a liquid through a valve can be related to pressure drop by

a) The equivalent orifice method
b) The flow coefficient, C_v, method
c) The NBS flow factor method
d) The discharge coefficient method
e) The resistance coefficient method.

The C_v method of valve sizing has come into use largely because of its simplicity and limited test work required for its determination. The resistance coefficient is discussed in Detailed Topic 3.9.5.2. To convert from one method to another use the factors listed in Table 3.8.4.

3.8.4.1 EQUIVALENT ORIFICE. The pressure drop across a valve may be expressed in terms of the diameter of a sharp-edged orifice having a discharge coefficient of 0.60. The orifice equation then becomes

$$Q = 0.60\ A_o \sqrt{\frac{2g\Delta P}{w}} \qquad \textbf{(Eq 3.8.4.1a)}$$

where

- Q = flow rate, ft³/sec
- A_o = equivalent orifice area, ft² (C_o = 0.60)
- ΔP = pressure drop across valve (fitting) measured between points where upstream and downstream diameters are equal, lb$_f$/ft²
- w = specific weight, lb$_f$/ft³

Expressed in terms of equivalent orifice diameter

$$\boxed{D_o = \frac{0.514\, Q^{1/2}\, w^{1/4}}{\Delta P^{1/4}}} \qquad \textbf{(Eq 3.8.4.1b)}$$

where D_o = equivalent orifice diameter, ft

This equation provides satisfactory results for diameter ratios $\frac{D_o}{D_1} \leq 0.5$ where D_1 is the nominal diameter of the connecting pipe.

To relate D_o obtained with water to volumetric air flow, the following equation is used (Reference 19-189):

(Eq 3.8.4.1c)

$$Q = 0.55\, D_o^2 P_1 \sqrt{\frac{P_2}{P_1}\left[\left(\frac{P_2}{P_1}\right)^{0.43} - \left(\frac{P_2}{P_1}\right)^{0.71}\right]}$$

where

- P_1, P_2 = upstream and downstream pressures, respectively, lb$_f$/ft²
- Q = flow rate, ft³/sec
- D_o = equivalent orifice diameter, ft

The accuracy of D_o as a measure of capacity is indicated in Figure 3.8.4.1. The test method is described in Reference 19-189.

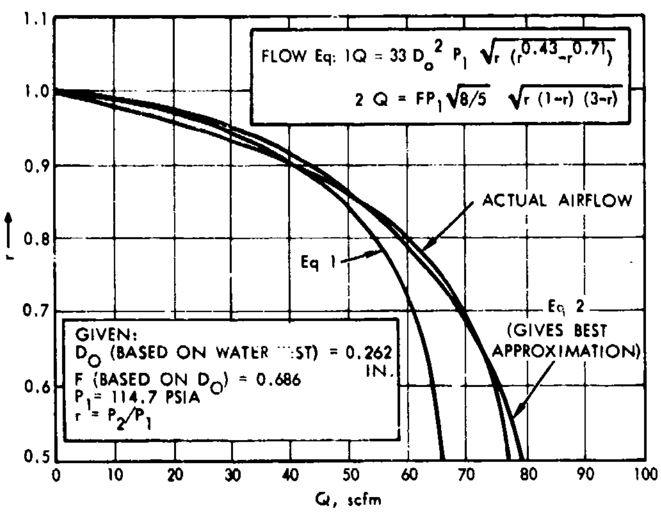

Figure 3.8.4.1. Comparison of Actual Flow Based on Equivalent Orifice and Flow Factor (tests conducted on 3/8 inch valve)
(Reprinted with permission from "Product Engineering," January 21, 1963, D. Lopera and J. Yeaple, Copyright 1963 by McGraw-Hill Publishing Company, Inc.)

3.8.4.2 **FLOW COEFFICIENT, C_v.** The flow coefficient is the volume (in gallons) of water at 60°F that will flow per minute through a valve with a pressure drop of 1 psi across the valve. At rated capacity the valve is wide open. The flow coefficient, C_v, of each valve depends upon manufacturer design and must always be determined individually by actual tests. Initially, considerable confusion resulted from the lack of specific equations for C_v computation and standards for testing. In the fall of 1950, the Fluid Controls Institute, Inc. (then known as the National Steam Specialty Club) established standards which could be applied to the testing of control valves. In 1952, formal engineering committees were established to set forth standards for valve tests. Standards for testing control and solenoid valves were formulated as a guide to manufacturers and consumers. These standards are revised periodically to meet changes in technology. Standards for rating flow and pressure drop of solenoid valves and sizing control valves are presented in References 159-1 and 159-3. On the basis of the work accomplished by the Control Valve Standards Committee of the Fluid Controls Institute (FCI) and their contributing authorities, the recommended formula for C_v determination with water is

$$C_v = Q\sqrt{\frac{1}{\Delta p}} \qquad \textbf{(Eq 3.8.4.2a)}$$

where C_v = flow coefficient

Q = flow rate in GPM of water at 60°F

Δp = pressure difference, lb_f/in^2

The formula for determining flow rate for any liquid when C_v is known is

$$\boxed{Q = C_v\sqrt{\frac{\Delta p}{S}}} \qquad \textbf{(Eq 3.8.4.2b)}$$

where S = specific gravity of the liquid.

When handling highly viscous fluids, a correction factor should be used for sizing control valves. As of May, 1962, no viscosity correction factor had been recommended by the Fluid Controls Institute. Some references have established a correction factor, based on a viscosity index, which is multiplied by the valve size determined assuming a non-viscous fluid. Correction factor versus a viscosity index are shown in Figure 3.8.4.2a. The viscosity index, θ, may be computed by the following equation (Reference 49-10):

$$\theta = \frac{379.2\,Q}{D\,\nu} \qquad \textbf{(Eq 3.8.4.2c)}$$

where θ = viscosity index

Q = flow rate, GPM

D = valve size assuming non-viscous condition, ft

ν = kinematic viscosity, stokes

C_v cannot be used in flow equations to determine overall system pressure drop; but, by substitution of appropriate values into Darcy's Equation for pipe friction, C_v and the resistance coefficient, K, are related by

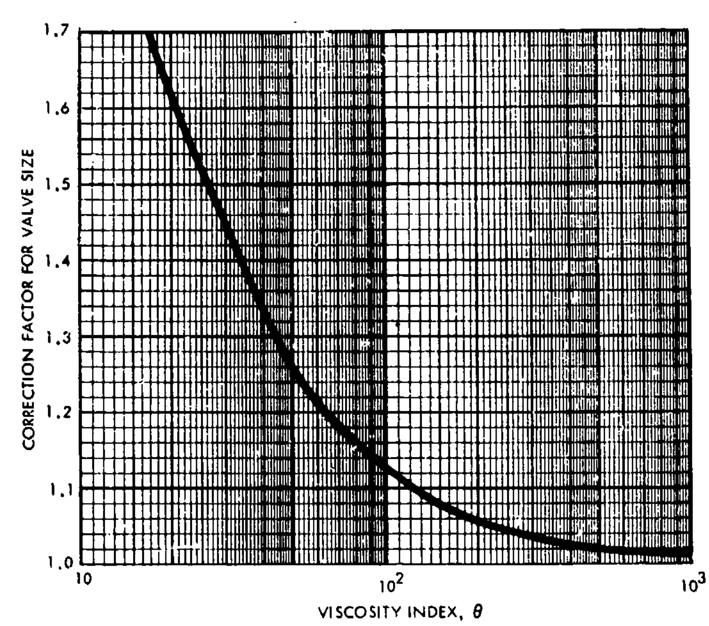

Figure 3.8.4.2a. Correction Factor for Viscosity Effects on C_v
(Adapted from "Chemical Engineering," April 1959, vol. 66, no. 8, W. G. Holzbock, Copyright 1959 by McGraw-Hill Publishing Company, Inc.)

Table 3.8.4.2a. Air-Flow Equations

(Adapted from "Product Engineering," D. Lapera and J. Yeaple, January 21, 1963, Copyright 1963 by McGraw-Hill Publishing Company, Inc.)

1. $Q = \frac{1360}{60} C_v \sqrt{\frac{\Delta p \times p_1}{ST_1}}$	C_v = flow coefficient	$= \frac{Q \times 60}{1360} \sqrt{\frac{ST_1}{\Delta p \times p_1}}$
2. $Q = \frac{1390}{60} C_v \sqrt{\frac{\Delta p \times p_2}{ST_1}}$	C_v = capacity factor	$= \frac{Q \times 60}{1390} \sqrt{\frac{ST_1}{\Delta p \times p_2}}$
3. $Q = \frac{5180}{60} C_v \sqrt{\frac{p_1^2 - p_2^2}{M.W.\ T_1}}$	C_v = valve-flow coefficient	$= \frac{Q \times 60}{5180} \sqrt{\frac{M.W.\ T_1}{p_1^2 - p_2^2}}$
4. $Q = \frac{963}{60} C_v \sqrt{\frac{p_1^2 - p_2^2}{ST_1}}$	C_v = flow coefficient	$= \frac{Q \times 60}{963} \sqrt{\frac{ST_1}{p_1^2 - p_2^2}}$
5. $Q = \frac{2.32^{0.443}}{60} C_g \frac{\Delta p^{0.443} \times p_1^{0.6}}{\sqrt{ST_1 / 520}}$	C_g = gas-flow coefficient	$= \frac{Q \times 60}{(2.32)^{0.443}} \times \frac{\sqrt{ST_1 / 520}}{\Delta p^{0.443} \times p_1^{0.6}}$

$$C_v = \frac{4310\ D^2}{\sqrt{K}} \qquad \textbf{(Eq 3.8.4.2d)}$$

where C_v = flow coefficient

D = valve nominal pipe diameter, ft

K = resistance coefficient (Detailed Topic 3.9.5.2)

Although determined with water, C_v may be used to relate gas flow rate and pressure drop with the use of an appropriate gas equation. Many formulae are being used to calculate gas flow. Reference 19-189 tabulated five different air flow equations taken from manufacturers' catalogues. These equations are shown in Table 3.8.4.2a. Most formulae are based on C_v's determined with water; but one equation is based on test data determined with air and is designated C_g rather than C_v. A comparison of a few of these coefficients with actual air flow data is illustrated in Figure 3.8.4.2b. The test method is described in Reference 19-189. In May, 1962, the Fluid Controls Institute recommended the following equation (Reference 159-2)

$$Q = \frac{1360}{60} C_v \sqrt{\frac{p_1 - p_2}{S_{av}\ T_1}} \sqrt{\frac{p_1 + p_2}{2}} \qquad \textbf{(Eq 3.8.4.2e)}$$

where Q = volume flow rate, standard ft^3/minute

p_1, p_2 = upstream and downstream pressures, respectively, psia

T_1 = absolute upstream temperature, °R

C_v = flow coefficient

S_{av} = average specific gravity (relative to air at 14.7 psia and 60°F)

Although the specific gravity is assumed to be a constant, it represents an average value based on upstream and downstream conditions.

For vapors, FCI has recommended the formula

$$\dot{w} = K_v C_v \sqrt{\Delta p} \sqrt{p_1 + p_2} \qquad \textbf{(Eq 3.8.4.2f)}$$

where $\dot{w}$ = weight flow rate, lb$_f$/hr

Δp = pressure difference = $p_1 - p_2$, psia

p_1, p_2 = upstream and downstream pressures, respectively, psia

C_v = valve flow coefficient, gpm / $\sqrt{\text{lb}_f/\text{in}^2}$

K_v = correction factor for vapors. A few values are presented in Table 3.8.4.2b and additional constants will be published by FCI when data becomes available (Reference 159-2)

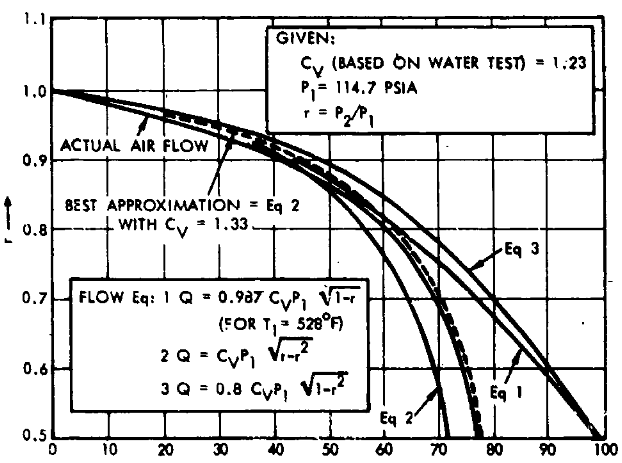

Figure 3.8.4.2b. Comparison of Actual Flow with Calculated Flow Based on C_v (tests conducted on 3/8 inch valve)

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Table 3.8.4.2b. Table of K_v Values

(From Fluid Controls Institute Bulletin 58-2, November 1958)

VAPOR	K_v
Steam	2.1
Freon 12	7.1
Freon 14	8.4
Freon 114	8.3
Dowtherm A	5.6
Ammonia	2.7

For both gases and vapors, when p_2 becomes less than the critical pressure, p_c, p_c should be used instead of p_2.
The value of C_v varies with valve size, type, and design. Change in C_v with diameter for several representative valve types is shown in Figure 3.8.4.2c. Since each type and size of valve must be tested individually, manufacturer's data should be consulted before valve size is selected. Typical values of C_v for 2-inch valves of various types are shown in Table 3.8.4.2c.

Although much has been done to standardize flow tests, there are still many factors which have an adverse effect upon the rating of control valves and valve sizing. Such factors include test procedures, end connections, conditions of flow-to-open or flow-to-close, and magnitude of pressure drop, and are discussed in the following paragraphs.

Test Procedures. The locations of pressure taps affect the magnitude of the pressure drop. At the present time there is a difference of opinion concerning the placement of pressure taps. Three methods are presented in Figure 3.8.4.2d. In Method 1, the pressure drop is measured across the valve including the fittings. FCI standards recommend locating the upstream pressure tap one-half to two and one-half pipe diameters upstream of the fitting connection and the downstream pressure tap 4 to 6 pipe diameters downstream from the outlet connection (Reference 159-1). This method includes pressure drop due to the fittings. In Method 2, this loss is avoided by measuring the pressure drop from the valve inlet to outlet. In Method 3, the Δp is measured as in Method 1, but the valve is later replaced by a spool of the same nominal size and length. Pressure readings are again taken at the same flow rates and subtracted from the first reading. Comparative test results, using the three methods discussed, are shown in Figure 3.8.4.2e. At low flow rates all three methods give the same results. At near maximum flow Method 3 exhibits a C_v 18 percent greater than by Method 1 (Reference 68-61).

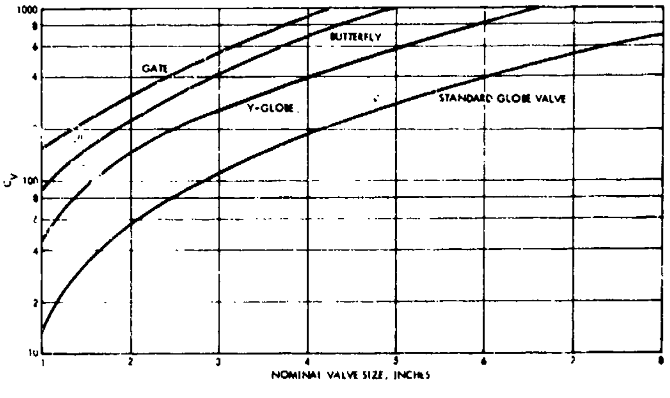

Figure 3.8.4.2c. Approximate Values of C_v Versus Valve Diameter

Table 3.8.4.2c. Typical Flow Coefficients for Two-Inch Valves

(References 138-1, 143-1, 23-38)

TYPE	C_v
Globe Valve	40-60
Angle Valve	47
Y Valves	
45° angle between stem and pipe centerline	72
60° angle between stem and pipe centerline	65
V-port Plug Valve	60-80
Butterfly Valve	
7 percent thick	333
35 percent thick	154
Conventional Gate	300-310
Pinch Valve	360
Swing Check	76
Recessed Swing Check	123
Ball Valve (reduced trim)	131
Ball Valve (full port)	440

End Connections. The type of valve connector affects the value of C_v. Generally, valves with flanged connections have lower values of C_v than valves with screwed connections. The degree of variation depends on the fitting geometry. To

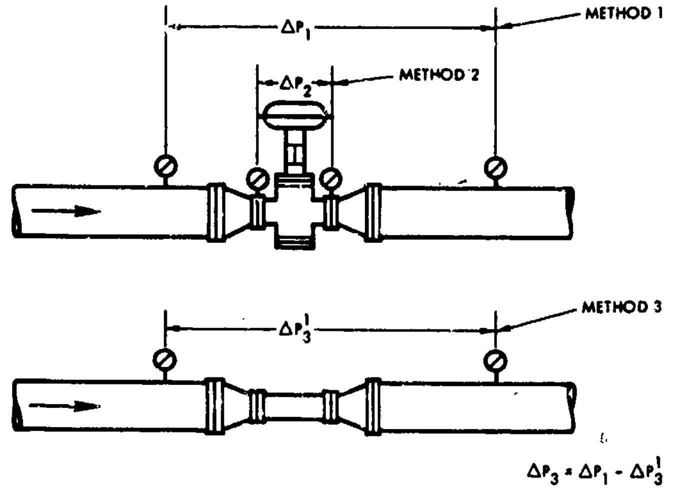

Figure 3.8.4.2d. Pressure Tap Locations for C_v Tests

(Adapted from ASME Paper 55A-152, by Johnson and Fallis, ASME, New York, November 1955)

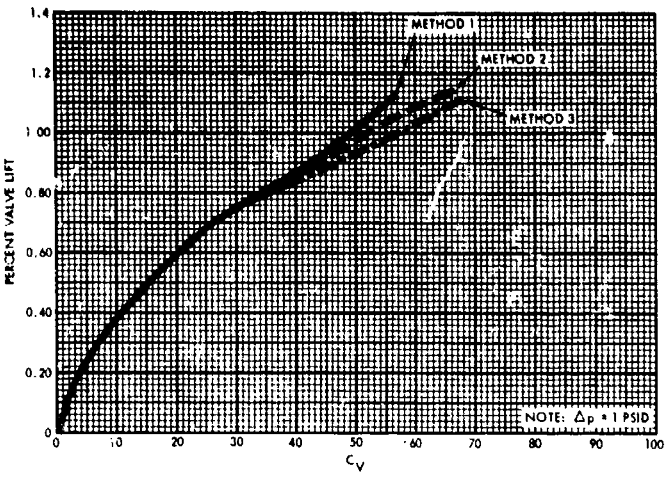

Figure 3.8.4.2e. Effect of Pressure Tap Location on C_v Values
(Reprinted with permission from ASME Paper 55A-152, by Johnson and Fallis, ASME, New York, November 1955)

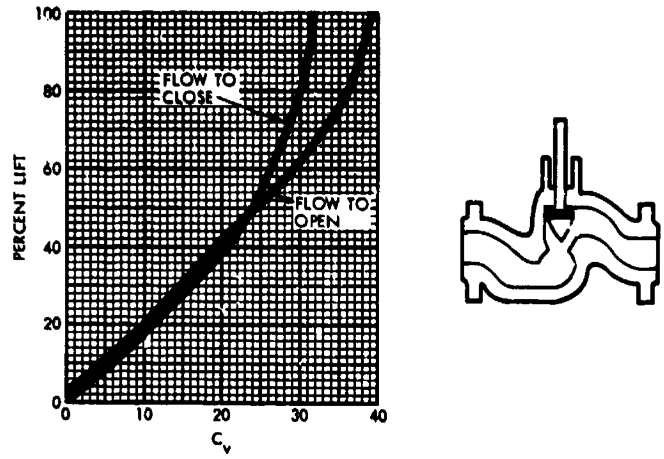

Figure 3.8.4.2f. C_v Versus Lift for Single-Seated Globe Valve (Flow-to-Open)
(Reprinted with permission from "ISA Journal," September 1960, vol. 7, no. 9, P. Wing, Jr., Instrument Society of America)

avoid undue complexity of sizing data, many manufacturers are presenting ratings based on threaded connections for sizes 2 inches or less and on flanged connections for larger sizes.

Conditions of Flow-to-Open or Flow-to-Close. Values of C_v for the same valve configuration usually change if the direction of flow is reversed. This change is commonly a consequence of difference in pressure recovery. For example, if the flow through the valve shown in Figure 3.8.4.2f tends to open the valve, there is an annular divergent path downstream of the restrictor, posed by the plug and body, which promises some pressure recovery. In the flow-to-close direction the level of the seat is too steep for this effect. When the valve reaches a stroke value of 50 percent or smaller, the divergent angle downstream of the plug is so large that little pressure recovery is possible in either flow direction (Reference 52-21).

For a high pressure angle valve shown in Figure 3.8.4.2g, a higher C_v is obtained in the flow-to-close direction where the divergent cone of the seat ring affords pressure recovery. Different internal geometries will alter these graphs.

With globe valves using air (Figure 3.8.4.2h), flow in the direction to close the valve results in a higher C_v than in the direction to open the valve. Furthermore, pressure drop has negligible effect on flow-to-open, but reduces the C_v as Δp increases. However, if p_2/p_1 is critical, C_v versus percent lift is identical for both flow-to-open and flow-to-close (Reference 52-21).

The fluid mechanic principles of pressure recovery within valves are the same as those pertaining to contraction and expansion losses in venturi design. For identical pressure drops across the valve, it follows that if pressure recovery can be attained within the valve, the C_v rating will be a larger value and a higher flow rate will be possible. Pressure recovery can be attained by valve body geometry, but more importantly by the valve trim or seat design. This is

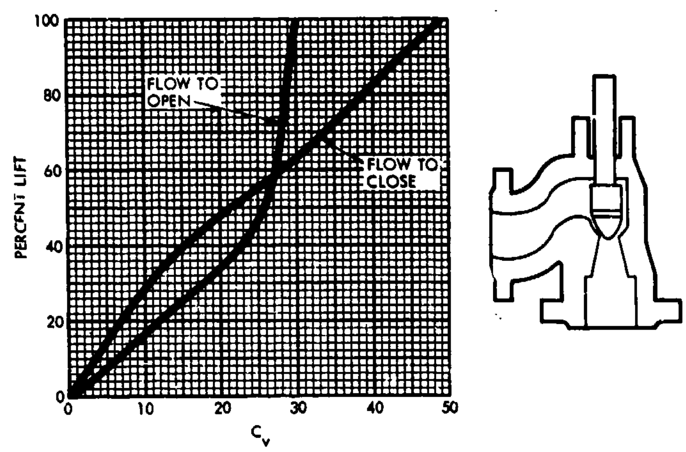

Figure 3.8.4.2g. C_v Versus Lift for Angle Valve
(Reprinted with permission from "ISA Journal," September 1960, vol. 7, no. 9, P. Wing, Jr., Instrument Society of America)

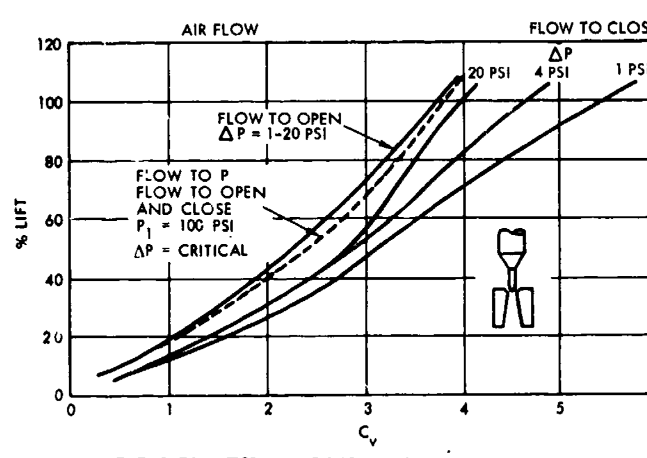

Figure 3.8.4.2h. Effect of Lift and Pressure Drop on C_v
(Adapted from "ISA Journal," September 1960, vol. 7, no. 9, P. Wing, Jr., Instrument Society of America)

illustrated in Figure 3.8.4.2i by comparing trims. As an example, using the original needle plug, the seat ring bored out as a sharp-edged orifice is the least efficient, reducing the standard trim C_v value from nearly 6 to 3 at 100 percent lift. By designing the plug taper and seat so that the annular flow area at the top of the seat is equal to the total area at the bottom of the seat ring, a C_v of nearly 7 was obtained at 100 percent lift (Reference 52-21).

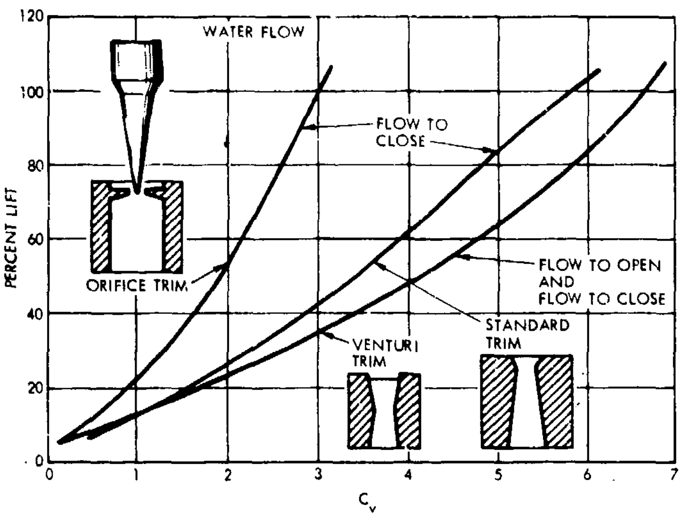

Figure 3.8.4.2i. Effect of Trim on C_v
(Adapted from "ISA Journal," September 1960, vol. 7, no. 9, P. Wing, Jr., Instrument Society of America)

Pressure Drop. Valve flow characteristics are dependent on pressure-drop conditions. Test results conducted at pressure drops of 80 and 5 psi are shown in Figure 3.8.4.2j.

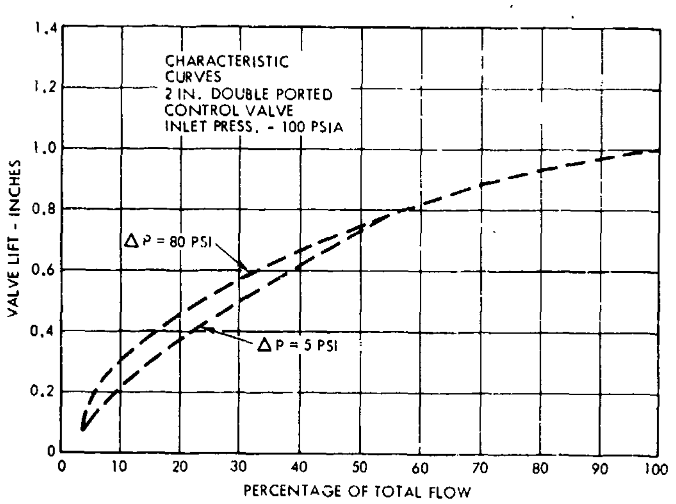

Figure 3.8.4.2j. Effect of Pressure Drop on C_v
(Reprinted with permission from ASME Paper 55A-152, by Johnson and Fallis, ASME, New York, November 1955)

The deviation of C_v shown is usually not detected when tests are conducted to determine C_v, since in most test procedures pressure drops of 5 to 10 psi are used (Reference 68-61).

3.8.4.3 NBS FLOW FACTOR. A measure of gas flow through orifices and valves is the NBS flow factor (F). The flow factor is defined as the ratio of air flow in SCFM to the upstream pressure in psia when $p_2/p_1 = 0.5$, or

$$F = \frac{Q_{(max)}}{p_1} \qquad \textbf{(Eq 3.8.4.3a)}$$

where F = flow factor, SCFM/psia
$Q_{(max)}$ = air flow in SCFM at $p_2/p_1 = 0.5$
p_1 = upstream pressure, psia

The flow factor relationship is derived from typical test data, converted to standard conditions as shown in Figure 3.8.4.3a, b, c. Dividing by the inlet pressure reduces the data to a single curve, (Figure 3.8.4.3b). The abscissa is normalized by dividing by the flow factor so that at $p_2/p_1 = 0.5$ the abscissa value is unity. The resulting curve is shown in Figure 3.8.4.3c. The equation of the normalized curve is

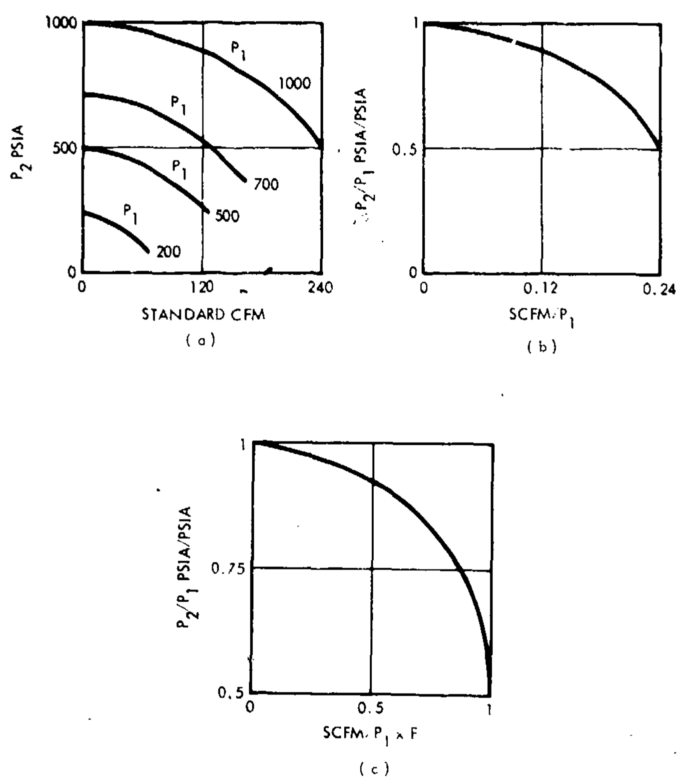

Figure 3.8.4.3a, b, c. Flow Factor Derivation
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(Eq 3.8.4.3b)

$$Q = p_1 F \sqrt{\frac{4}{3}} \times \sqrt{1 - \left(\frac{p_2}{p_1}\right)^2}$$

where Q = air flow in SCFM at $p_2/p_1 \geqq 0.5$.

The original report of the NBS flow factor can be found in Reference 82-6.

References 19-189, and 19-75 use

(Eq 3.8.4.3c)

$$Q = p_1 F \sqrt{\frac{8}{5}} \times \sqrt{\left(\frac{p_2}{p_1}\right)\left(1 - \frac{p_2}{p_1}\right)\left(3 - \frac{p_2}{p_1}\right)}$$

Solution of Equation (3.8.4.3b) may be simplified by using the nomograph in Figure 3.8.4.3d. The accuracy of the flow factor in Equation (3.8.4.3c) is illustrated in Figure 3.8.4.1. The use of flow factors for evaluating pneumatic systems is presented in Reference 19-165. Valve flow factors are determined in Reference 1-144 by relating equivalent lengths of the bends, enlargements, orifices, etc., within a valve to a so-called basic flow factor.

The flow factor equations, although based on air flow, are applicable to all gases through the use of appropriate corrections for differences in γ and R.

$$Q_g = Q_a \frac{Z_g}{Z_a} \sqrt{\frac{R_g T_g}{R_a T_a}} \qquad \textbf{(Eq 3.8.4.3d)}$$

where Q = flow in SCFM
R = gas constant, ft-lb_f/lb_m °R

$$Z = \left\{\frac{\gamma}{\gamma - 1}\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{\frac{1}{2}}$$

Subscripts a and g refer, respectively, to air and any other gas.

The product $\sqrt{\frac{R_g}{R_a}\frac{Z_g}{Z_a}}$ calculated for several common gases at $\frac{p_2}{p_1} = 0.5$ is given as a correction factor in Table 3.8.4.3.

Table 3.8.4.3. Correction Factors for Flow Factor, F

GAS	FACTOR
Helium	2.830
Oxygen	0.951
Nitrogen	1.020
Hydrogen	3.800
Carbon dioxide	0.784

When flow factors for individual components have been established, the system flow factor, F_s, can be calculated from the following equations (Reference V-56):

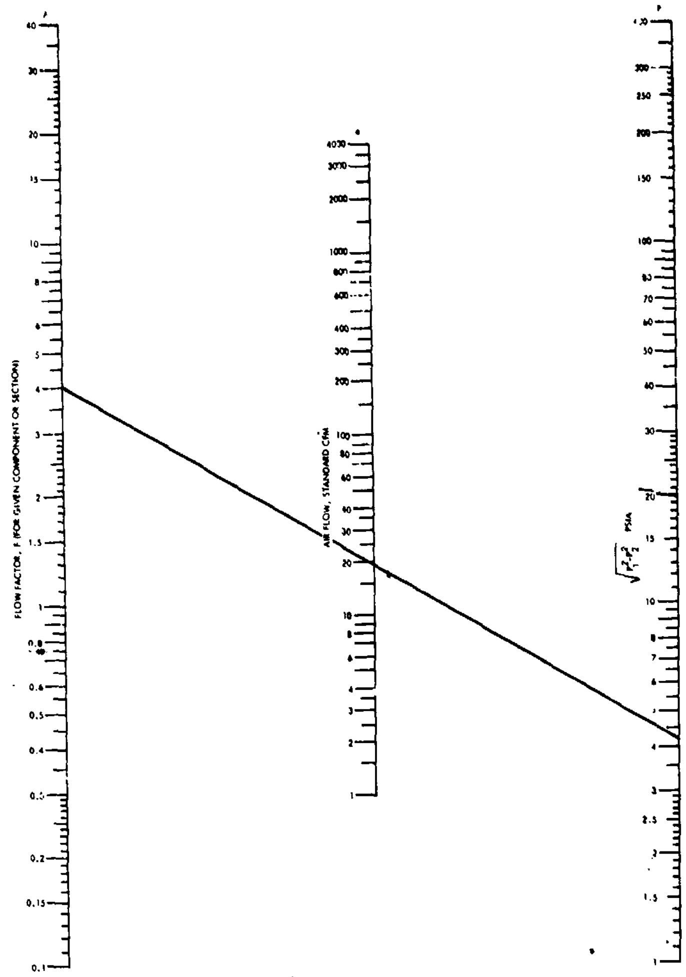

Figure 3.8.4.3d. Nomograph for Calculating Air Flow Rates by Flow Factors

(Adapted from "Product Engineering," October 1959, G. Riske, Copyright 1959 by McGraw-Hill Publishing Company, Inc.)

For two components

$$F_s = \frac{F_1 \times F_2}{\sqrt{(F_1)^2 + (F_2)^2}} \qquad \textbf{(Eq 3.8.4.3e)}$$

For three components

(Eq 3.8.4.3f)

$$F_s = \frac{F_1 \times F_2 \times F_3}{\sqrt{[F_1 F_2]^2 + [F_2 F_3]^2 + [F_3 F_1]^2}}$$

For systems with more than three components these equations can be expanded in a similar manner, but become progressively more complicated and difficult to handle. A simplified method for calculating the pressure drop of components in a series uses unit flow factors which are related to equivalent lengths of Schedule 40 pipe. This method is discussed in Reference 19-165.

The flow factor for a system, F_s, is defined in the same way as the flow factor for a single component. Once the F_s is known, the whole system can be treated as if it were a single component.

3.8.4.4 DISCHARGE COEFFICIENT, C_D. The discharge coefficient for valves is not the same as an orifice discharge coefficient. The downstream pressure in the orifice flow, Equation (3.8.1) is measured at the vena contracta; thus the C_D for an orifice can be defined in terms of actual orifice flow area to effective flow area at the vena contracta. The downstream pressure in a valve is measured downstream of minimum flow restriction, therefore the C_D for a valve is an empirical constant relating Q, A, and P, expressed as follows:

$$Q = C_D A \sqrt{\frac{2g\Delta P}{w}} \qquad \text{(Eq. 3.8.4.4)}$$

where
Q = volume flow rate, ft^3/sec
C_D = valve discharge coefficient
A = minimum valve geometric flow area, ft^2
ΔP = valve upstream pressure-valve downstream pressure, lb_f/ft^2
w = specific weight, lb_f/ft^3

For poppet and slide valves, C_D values between 0.80 and 0.90 are typical; for butterfly valves, 0.87 is typical (Reference 408-1).

3.9 PRESSURE LOSSES

3.9.1 General

Fluid flow through systems is always accompanied by the friction of particles moving relative to one another or against the stream-confining walls. As discussed in Sub-Section 3.6, friction viscous forces cause a decreasing energy gradient with corresponding pressure drop in the direction of flow. In the expanded Bernoulli Equation the loss due to friction is related to the number of velocity heads by the equation

$$h_f = \sum_{i=1}^{n} K_i \frac{V_i^2}{2g} \qquad \text{(Eq 3.9.1)}$$

The term K_i is the resistance coefficient of each element composing the system; V_i is the fluid velocity through the element. The value of K varies with element configuration, surface condition, and Reynolds number. The following Sub-Topics provide K-values or expressions to calculate K for various elements and conditions of fluid flow.

The head loss, h_f can also be determined from the difference in static pressure between any two points, provided the dynamic heads and elevations at the two points are equal.

3.9.2 Pipe Losses

3.9.2.1 STRAIGHT PIPE. Frictional drag due to fluid viscosity causes a pressure loss in the direction of flow. This pressure loss through a horizontal straight pipe with no change in kinetic energy may be calculated from Darcy's Equation

$$\Delta P = f_D \left(\frac{L}{D}\right) w \frac{V^2}{2g} \qquad \text{(Eq 3.9.2.1a)}$$

or in terms of the frictional head loss

$$h_f = f_D \left(\frac{L}{D}\right) \frac{V^2}{2g} \qquad \text{(Eq 3.9.2.1b)}$$

where
ΔP = pressure loss, lb_f/ft^2
f_D = Darcy friction factor
L = pipe length, ft
D = pipe diameter, ft
V = stream velocity, ft/sec
w = specific weight, lb_f/ft^3
h_f = head loss, ft

The resistance coefficient, K, is represented by $f_D\left(\frac{L}{D}\right)$.

An equation similar to Equation (3.9.2.1a) is the Fanning Equation for pressure drop derived from the concept of the hydraulic radius, namely

$$\Delta P = f_F \left(\frac{L}{R_H}\right) w \frac{V^2}{2g} \qquad \text{(Eq 3.9.2.1c)}$$

The hydraulic radius, R_H, is the ratio of the area to the wetted perimeter

$$R_H = \frac{A}{P_w} \qquad \text{(Eq 3.9.2.1d)}$$

For circular pipe R_H equals D/4, which, if substituted into Equation (3.9.2.1c), formulates the Fanning Equation for circular pipe

$$\Delta P = 4f_F \left(\frac{L}{D}\right) w \frac{V^2}{2g} \qquad \text{(Eq 3.9.2.1e)}$$

By comparing the Darcy and Fanning Equations it is apparent that the Fanning friction factor has a value one-fourth the value of the Darcy friction factor. This difference in value can be a source of error since pipe friction data is available for both equations; but, occasionally, the type of friction factor may not be defined. For high velocity gas flow ($> 0.2V_s$) density changes become significant and Darcy's and Fanning's Equations are no longer adequate. Many formulae, developed on an empirical or semi-empirical basis, are presented in Reference 143-1, page 1-8 for calculating natural gas flow, gas flow in long pipe lines, and gas flow with isothermal transportation.

For *laminar flow* ($Re < 2000$), $f_D = \frac{64}{Re}$. Substitution into

Darcy's Equation yields the Hagen-Poiseuille Law for laminar flow

$$\Delta P = \frac{32\, V L \mu_m}{g_c\, D^2} \qquad \textbf{(Eq 3.9.2.1f)}$$

where ΔP = pressure difference, lb_f/ft^2

V = velocity, ft/sec

D = diameter, ft

L = length, ft

μ_m = absolute viscosity, lb_m/ft sec

For *turbulent flow* ($Re > 4000$), f_D depends not only on Reynolds number but also the relative surface roughness of the pipe, $\frac{\epsilon}{D}$. Because of the complexity of the interrelationship of friction factor and surface roughness, no general theoretical equation is suitable for all cases. However, two empirical formulae have been developed. Blaius found that for smooth pipes and values of Reynolds number from 4×10^3 to 10^5, the friction factor may be expressed as

$$f_D = \frac{0.316}{\sqrt[4]{Re}} \qquad \textbf{(Eq 3.9.2.1g)}$$

For values of Reynolds number greater than 10^5, the relationship between f_D and Reynolds number for smooth pipes may be expressed by the Karman-Prandtl equation

(Eq 3.9.2.1h)

$$\frac{1}{\sqrt{f_D}} = -0.8 + 2 \log_{10} Re \sqrt{f_D}$$

This equation is solved easily by plotting values of Re versus f_D.

The most useful and widely accepted data of friction factors versus Reynolds number were prepared by L. F. Moody (Reference 26-95) and are shown in Figure 3.9.2.1a. Val-

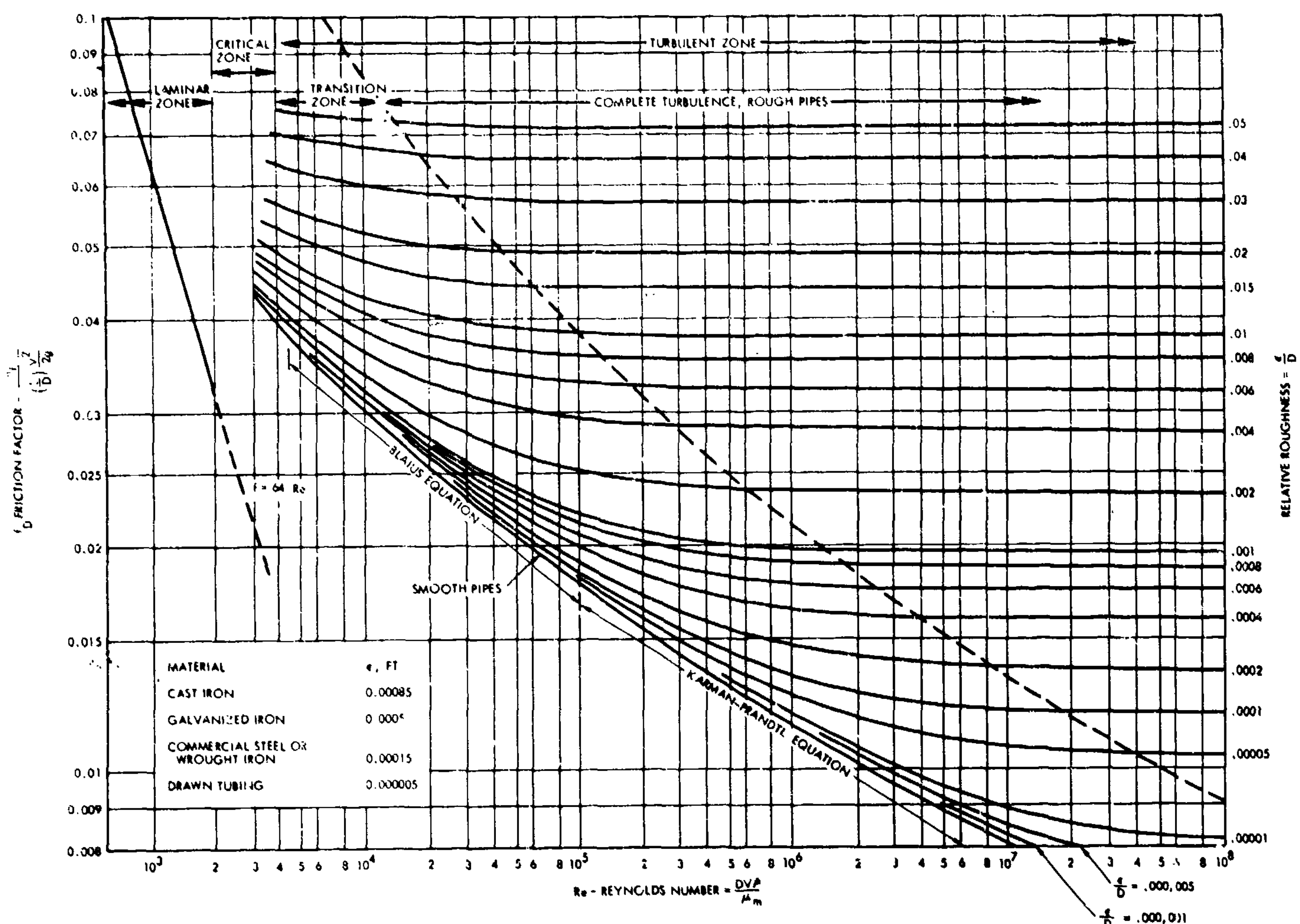

Figure 3.9.2.1a. Friction Factors for Straight Pipe

(Adapted from "ASME Transactions—Journal of Basic Engineering," November 1944, vol. 66, no. 8, L. F. Moody, New York)

ues of absolute roughness, ϵ, for several grades of pipe are shown in the figure. Values of relative roughness, $\frac{\epsilon}{D}$, may be computed by dividing the absolute roughness, ϵ, by the internal diameter, D, measured in feet.

The pressure drop of *two-phase flow* through a pipe can be determined approximately by the equation (Reference 141-1):

$$\Delta P = 2f' \frac{L_\rho V^2}{Dg_c} \qquad \text{(Eq 3.9.2.1i)}$$

where D = pipe I.D., ft

ΔP = pressure drop, lb_f/ft^2

V = average velocity, ft/sec

L = pipe length, ft

ρ = density, lb_m/ft^3

f' = two-phase friction factor as a function of a two-phase Reynolds number, Re'

The Reynolds number for two-phase flow is (Reference 141-1):

$$Re' = \left(\frac{D\rho_g V_g}{\mu_g}\right)^a \left(\frac{D\rho_l V_l}{\mu_l}\right)^b \qquad \text{(Eq 3.9.2.1j)}$$

where $a = \frac{y}{y+1}$

$b = e^{-0.1y}$

y = mass flow ratio between gas and liquid phase

Values of f' versus Re' are presented in Figure 3.9.2.1b for different gas-liquid weight ratios.

Practical engineering experience has provided a range of velocities which give reasonable pressure drops, the magnitudes of which vary from 1 to 5 psi for every 100 feet equivalent length of pipe. These velocities are found in Table 3.9.2.1.

For rocket systems, somewhat higher velocities are considered to be consistent with good design practice. Pro-

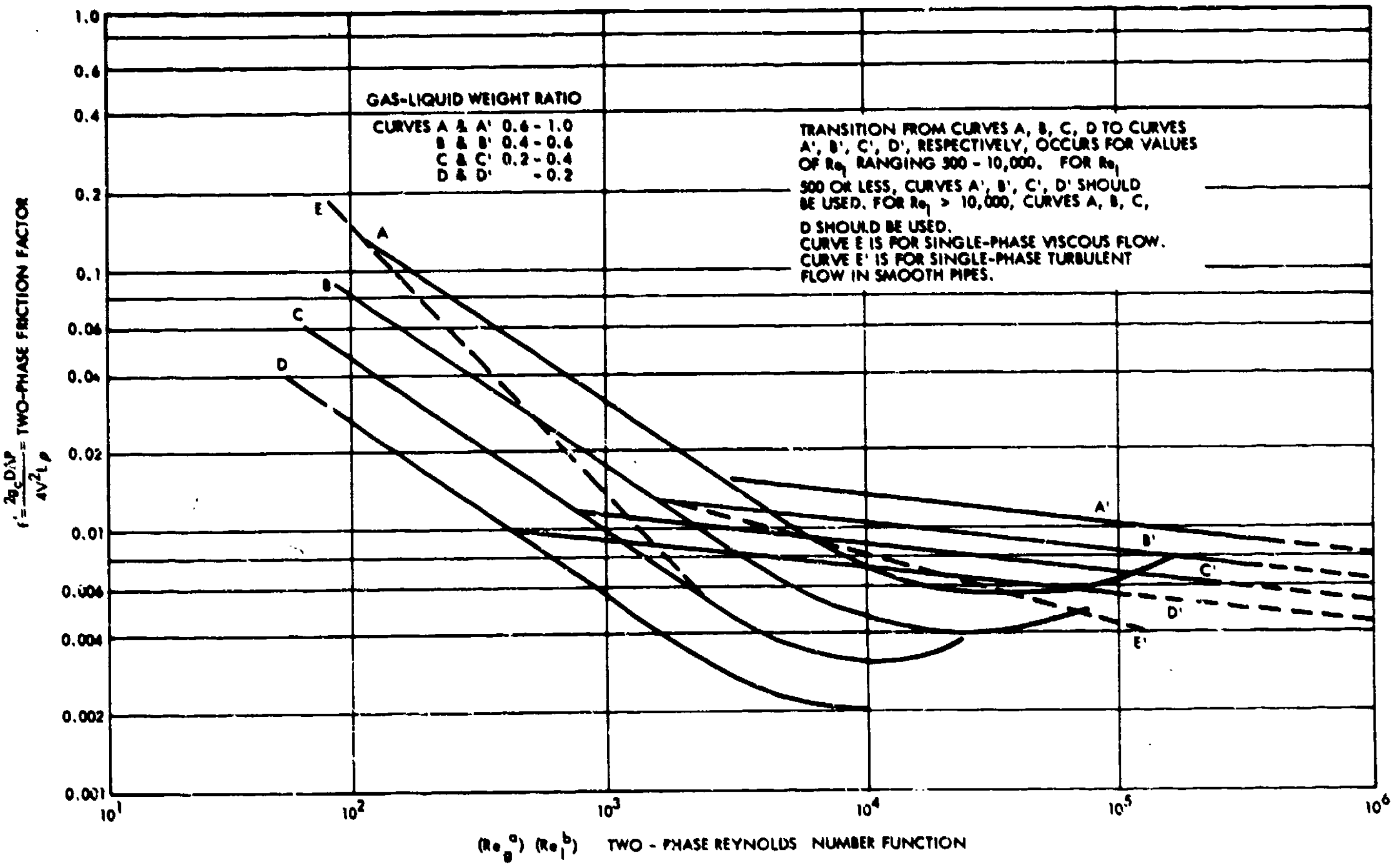

Figure 3.9.2.1b. Two-Phase Friction Factor

(Reprinted with permission from "Transactions AIME," vol. 207, American Institute of Mining, Metallurgical, and Petroleum Engineers, 1956)

Table 3.9.2.1. Velocity of Fluids in Circular Pipes
(Reference 49-7)

FLUID	VELOCITY (ft/sec)
Water	3 to 20
Non-viscous organic fluids	2 to 10
Viscous liquids	1 to 3
Gases or vapors	30 to 300

pellant velocities of up to 50 ft/sec are common for pump discharge ducts, and velocities considerably higher are used in thrust chamber coolant passages.

3.9.2.2 FLEXIBLE METAL HOSE. Pressure loss through corrugated flexible hose may be several times greater than that for pipe of the same size. The high pressure drops encountered in the flexible metal tubing can be attributed to hose convolutions acting as a very rough wall; however, they do not follow an $\frac{\epsilon}{D}$ relationship for pipe roughness. The effect of convolutions is not known, but from tests conducted on the same size pipe, helical-type hose has a lower pressure drop than annular-type, possibly due to extra flow area (Reference 19-170). Pressure losses for helical and annular metal hoses are shown in Figure 3.9.2.2.

3.9.2.3 BENDS. When the direction of flow is changed, work is done by the fluid and the total fluid energy decreases. The loss of head due to direction change is accounted for by including an additional loss coefficient, K_B, in the pressure drop equation for straight pipe. The flow loss in a bend is computed by the equation (Reference 23-38):

$$\Delta P = \left(f_D \frac{L}{D} + cK_B\right)\frac{wV^2}{2g} \quad \textbf{(Eq 3.9.2.3)}$$

Values of K_B for 90° bends are shown in Figure 3.9.2.3a. For computing bends other than 90°, multiply the value of $K_{B_{90°}}$ by the correction factor c, shown in Figure 3.9.2.3b. The minimum pressure loss occurs at $\frac{R}{D} = 4.0$.

Values of K_B for compound "U" bends, offset bends, and "Z" bends are shown, respectively, in Figures 3.9.2.3c, 3.9.2.3d, and 3.9.2.3e. K_B for mitre bends are shown in Figure 3.9.2.3f.

Loss coefficients for bends in square and rectangular ducts may be found in Section 1 of Reference 23-38, pages A-25 to A-30.

Friction factors for turbulent flow in curved pipe may be found in Reference 23-67.

3.9.2.4 BRANCHED DUCTS. The total pressure drop across a branched duct may be approximated by considering both the pipe friction and pressure losses resulting from the turbulent mixing or separation of fluid. The

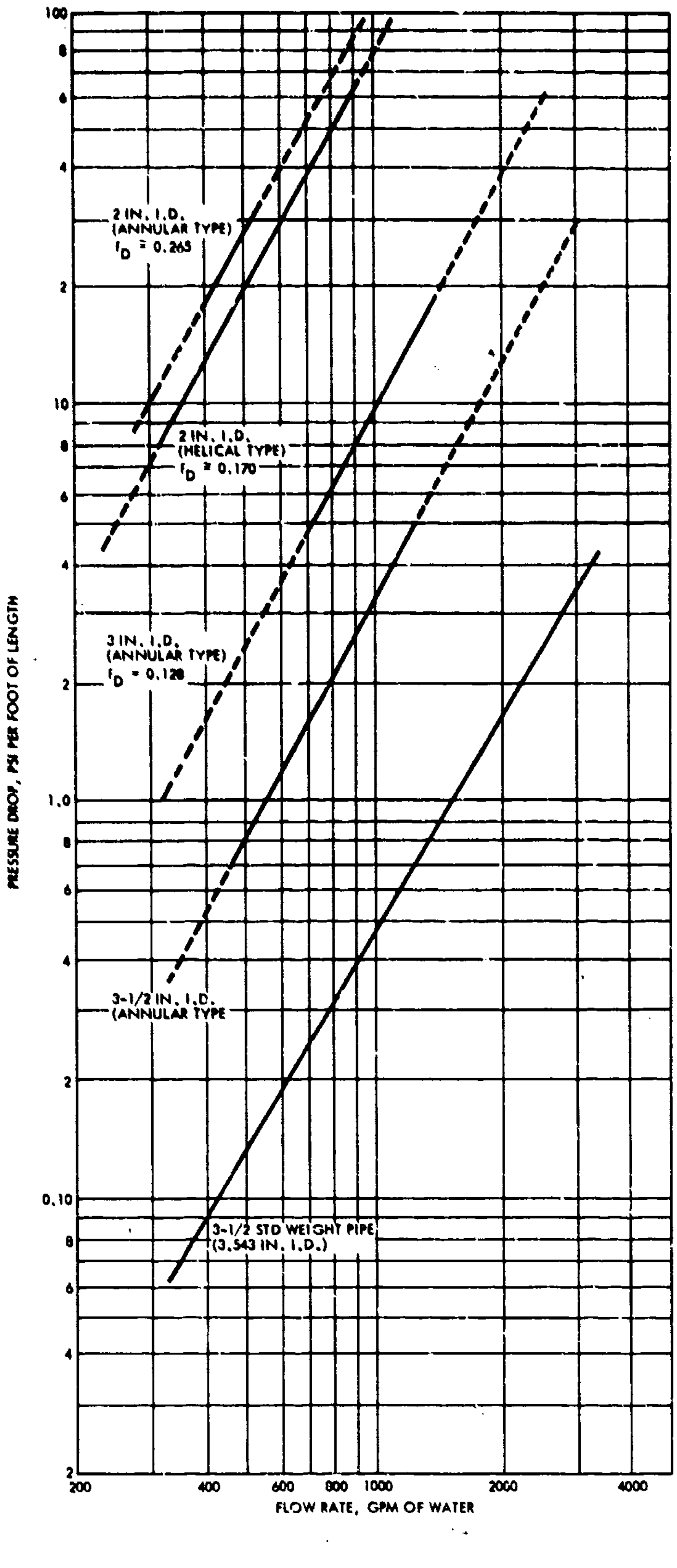

Figure 3.9.2.2. Pressure Drop for Flexible Hose
(Adapted from "Product Engineering," April 1956, vol. 27, no. 4, Daniels, Copyright 1956 by McGraw-Hill Publishing Company, Inc.)

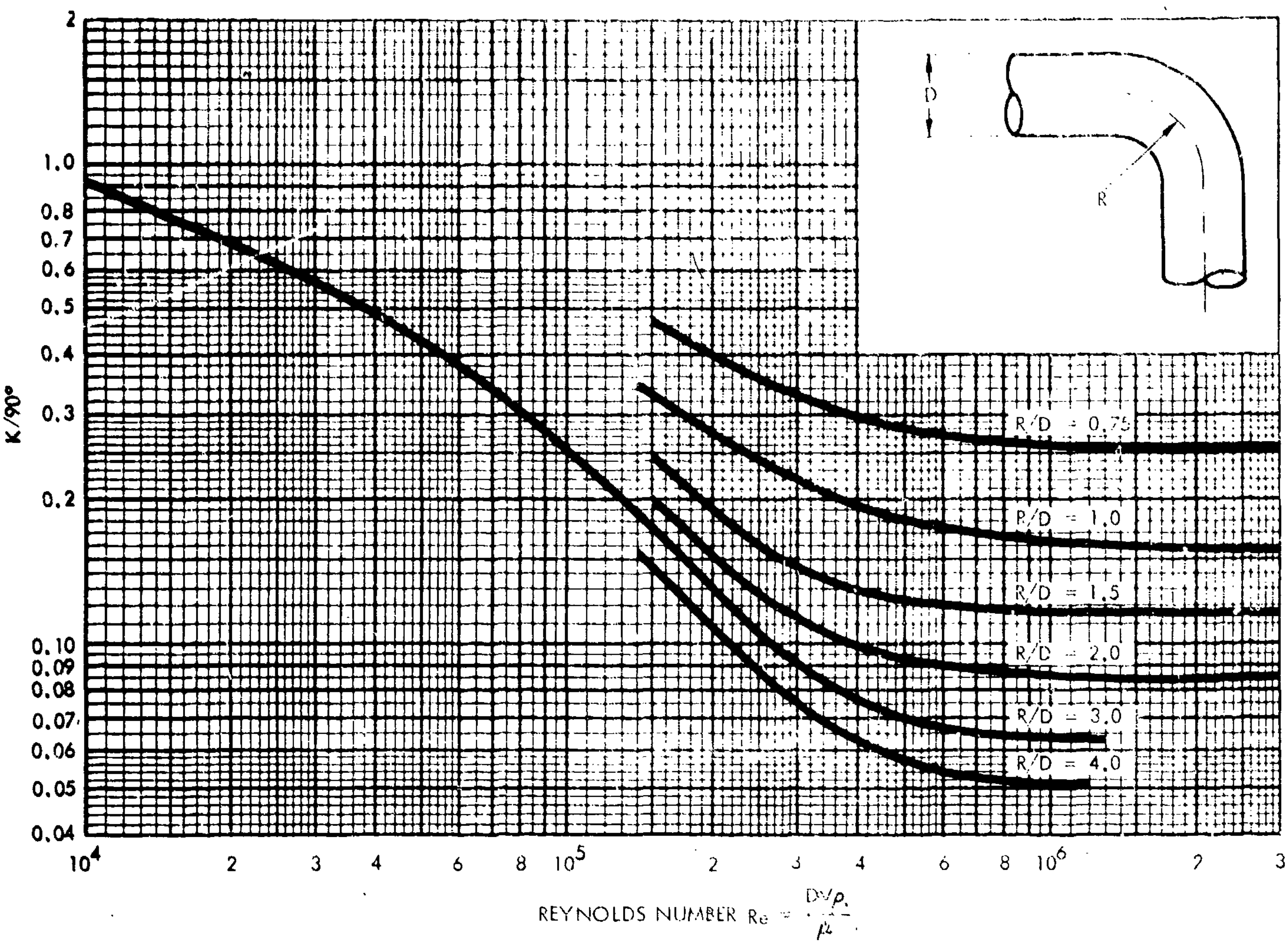

Figure 3.9.2.3a. Resistance Coefficients K for 90° Bends
(Adapted from NACA Wartime Report L 208, J. Henry; and "ASME Transactions," November 1957, vol. 79, Kittredge and Rawley)

total pressure drop may be estimated by the equation (Reference 23-38):

$$\Delta P = \left(f\,\frac{L}{D} + {}_xK_Y\right) w\,\frac{V^2}{2g} \quad \textbf{(Eq 3.9.2.4a)}$$

For a *converging branched duct*, the resistance coefficient ${}_xK_Y$ between 1 and 3 shown in Figure 3.9.2.4a may be calculated from the equation (Reference 23-38):

(Eq 3.9.2.4b)

$$ {}_1K_3 = \lambda + \left(\frac{w_3V_3}{w_1V_1}\right)^2 - 2\,\frac{A_1}{A_3}\cos\beta' - \left(\frac{w_2V_2}{w_1V_1}\right)^2 \times 2\,\frac{A_2}{A_3}\cos\delta' $$

Values of λ and the terms $(2\,\frac{A_1}{A_3}\cos\beta')$ and $(2\,\frac{A_2}{A_3}\cos\delta')$ may be found in Figure 3.9.2.4a as a function of β, δ, and the area ratios.

For a *diverging branched duct*, values of ${}_xK_Y$ between points 1 and 2 may be found in Figure 3.9.2.4b.

Diverging flow is subject to eddy formation and separation with a substantial increase of turbulence. Consequently, diverging flow is less efficient than converging flow and results in a higher resistance coefficient. Converging flow results in a more uniform velocity across the section while diverging flow increases the velocity variation.

3.9.3 Entrance and Exit Losses

Fluid entering a pipe from a reservoir exhibits a converging of streamlines and an increase in turbulence and

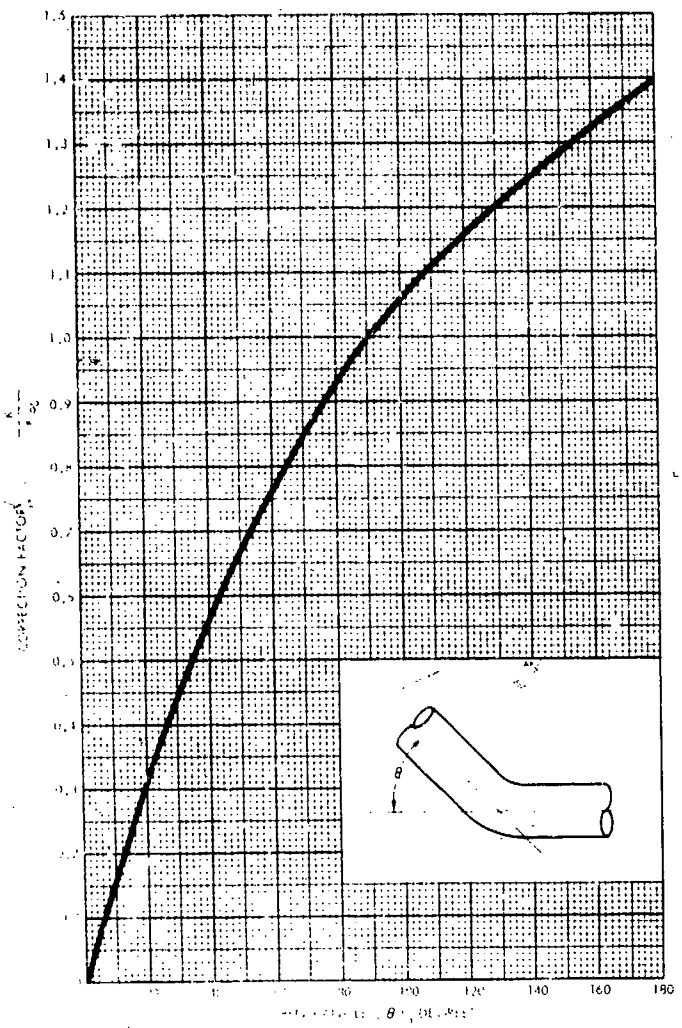

Figure 3.9.2.3b. Correction Factors for Bends Other than 90°
(Adapted from SAE Information Report No. 23, October 1951.)

vortex motion which may be observed several diameters downstream. Consequently, the head losses can be greater than in a corresponding length with normal flow. A portion of this loss can be attributed to pipe friction, the remaining losses to entrance conditions. The head loss due to entrance conditions may be expressed as

$$h_f = \frac{\Delta P}{w} = K_e \frac{V^2}{2g} \qquad \textbf{(Eq 3.9.3)}$$

Values of K_e are equal to $1/c_v^2 - 1$, where c_v represents values from Figure 3.8.2.2a. Average values of K_e for various entrance conditions are shown in Table 3.9.3.

A fluid discharging from a pipe into a reservoir which is larger than the pipe absorbs the total velocity head of the fluid. Consequently, the *resistance coefficient is 1.0* for all cases including rounded and sharp-edged exits. It should be noted that as contrasted to entrance losses, dis-

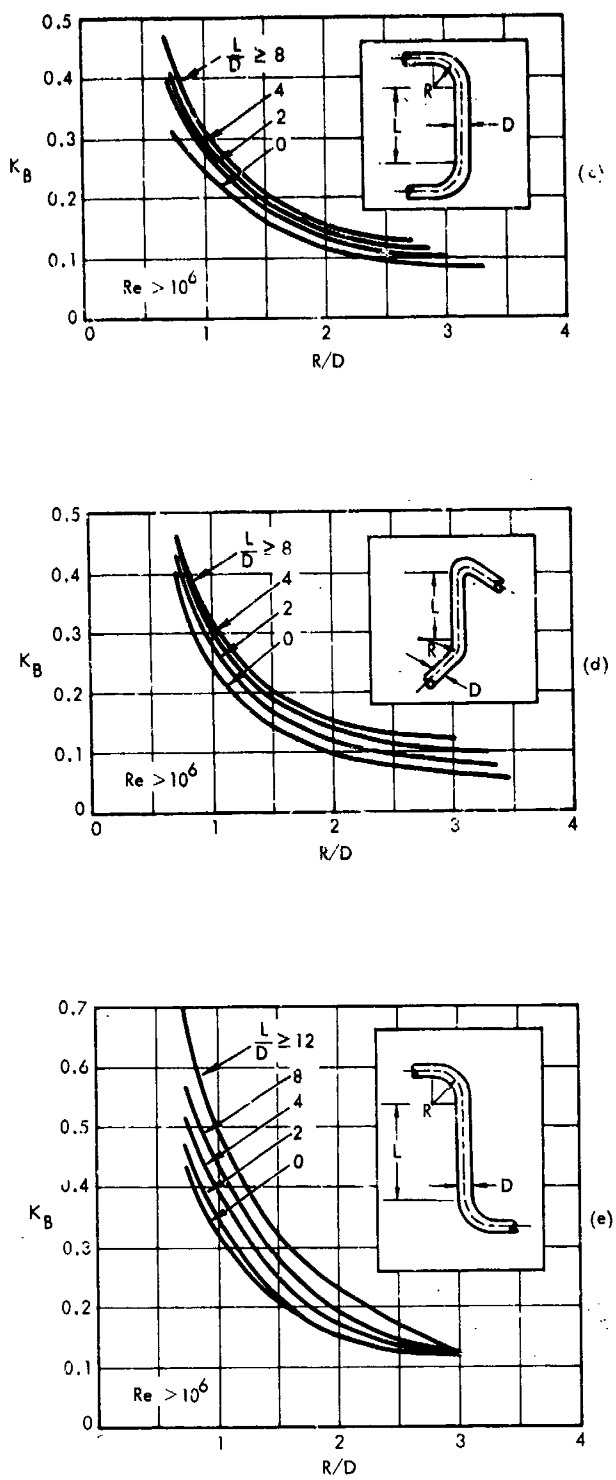

Figure 3.9.2.3c, d, e. Resistance Coefficients for Various Bend Configurations
(From WADC TR 56-187, prepared by Fluidyne, September 1956, and the NACA Wartime Report W-39, February 1943)

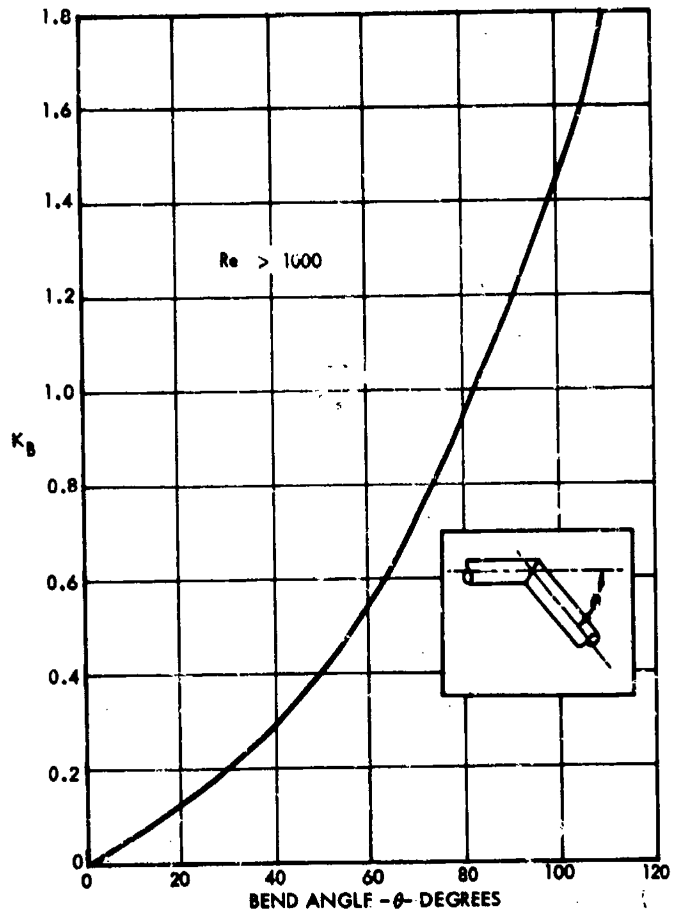

Figure 3.9.2.3f. Resistance Coefficient for Mitre Bends in Circular Ducts

(From SAE Information Report No. 23, October 1951, and Transactions ASME, Kittredge and Rowley, vol. 79, 1957.)

charge losses occur in the reservoir after the fluid leaves the pipe (Reference 138-1).

3.9.4 Enlargement and Contraction Losses

Flow from a small diameter pipe into a large diameter pipe is accompanied by an increase in static pressure due to the decrease in velocity of fluid. Conversely, flow from a large diameter pipe to a small diameter pipe is marked by a decrease in static pressure due to increase in fluid velocity. In both cases, the total energy of the system is decreased due to the turbulent conditions occurring downstream of the interface. The magnitude of the energy loss is dependent on the type of change, from smaller to larger pipe or vice versa.

Head loss due to *sudden enlargement* is stated

$$h_f = \frac{\Delta P}{w} = K_{en}\left(1 - \frac{A_1}{A_2}\right)^2 \frac{V_1^2}{2g} \qquad \text{(Eq 3.9.4a)}$$

$K_{en} = 1.0$ for sudden enlargement.

Table 3.9.3. Entrance Losses

TYPE	K_e
Well Rounded	0.04
Slightly Rounded	0.23
Sharp edged	0.485 — 0.56
Inward Projecting Pipe	0.62 — 0.93

where V_1 is the mean velocity in the *inlet* pipe of cross-sectional area A_1. Area A_2 is the cross-sectional area of the larger downstream pipe. Values of K_{en} are shown in Figure 3.9.4a.

For *gradual enlargement*, K_{en} varies with the included angle of the diffuser cone. Minimum losses occur at diffuser angles between 5° and 9°. Values of K_{en} for diffusers with very smooth surfaces as functions of cone angle are shown in Figure 3.9.4b.

The pressure loss may be calculated from Equation (3.9.4a) with appropriate resistance coefficients.

For *sudden contraction*, the head loss is computed by

$$h_f = \frac{\Delta P}{w} = K_c \frac{V_2^2}{2g} \qquad \text{(Eq 3.9.4b)}$$

where V_2 is the mean velocity in the *discharge* pipe. Values of K_c are shown in Figure 3.9.4a.

For *gradual contraction*, K_c is approximately 0.04 for all cases. The head loss is computed by

$$h_f = \frac{\Delta P}{w} = 0.04 \frac{V_2^2}{2g} \qquad \text{(Eq 3.9.4c)}$$

where V_2 is the mean velocity in the *discharge* pipe.

3.9.5 Fittings and Valve Losses

The resistance of fittings and valves to liquid flow may be expressed by two methods:

a) Equivalent length in pipe diameters
b) Resistance coefficient, K

The two methods are interrelated, with the choice dependent only on the method each engineer wishes to use in order to compute the overall system pressure drop.

3.9.5.1 EQUIVALENT LENGTH IN PIPE DIAMETERS. In Darcy's equation for pipe losses the resistance coefficient, K, is the product of the friction factor, f_D, and the equivalent length of pipe in pipe diameters, $\frac{L}{D}$. By rearrangement of terms, the equivalent length in pipe diameters may be expressed

$$ {}_1K_3 = \lambda + \left(\frac{w_3V_3}{w_1V_1}\right)^2 - 2\frac{A_1}{A_3}\cos\beta' - \left(\frac{w_2V_2}{w_1V_1}\right)\times 2\frac{A_2}{A_3}\cos\delta' $$

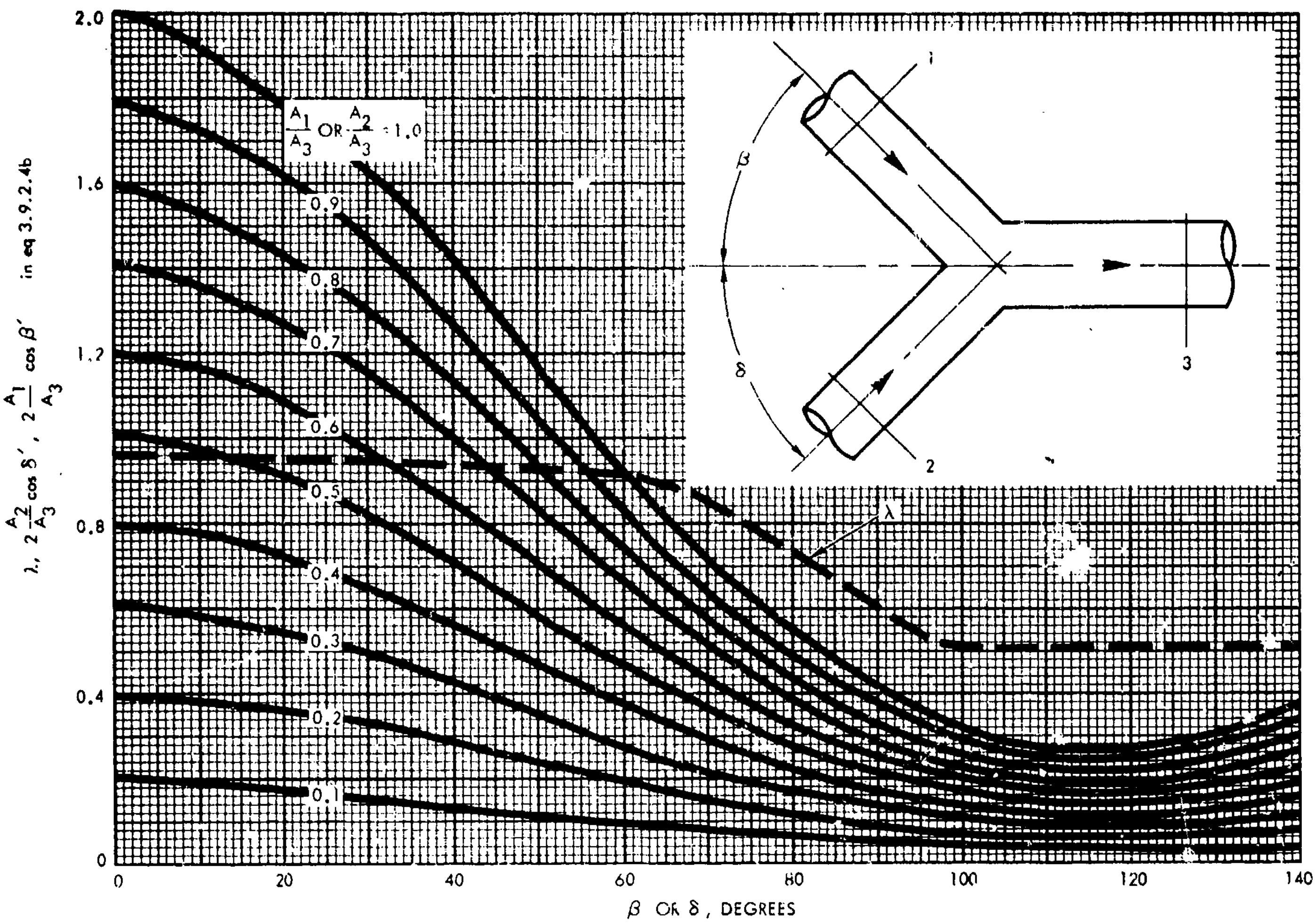

Figure 3.9.2.4a. Functions for Calculating Resistance Coefficient K for Converging Branches

(Reference 23-38)

$$ L_e = \frac{L}{D} = \frac{K}{f_D} \qquad \textbf{(Eq 3.9.5.1a)} $$

where
- D = internal diameter of pipe
- L = pipe length
- L_e = equivalent length in pipe diameters
- K = resistance coefficient
- f_D Darcy friction factor

Use of the equivalent length principle may simplify pipe system calculations since, if the friction factor can be approximated to that of the piping itself, the equivalent length of fittings and valves can be added to the pipe equivalent length. For example

$$ \Delta P_{total} = \Delta P_{pipe} + \Sigma\Delta P_{fittings\ and\ valves} \qquad \textbf{(Eq 3.9.5.1b)} $$

$$ \Delta P_{total} = f_D\left(\frac{L}{D} + \Sigma L_e\right) w \frac{V^2}{2g} \qquad \textbf{(Eq 3.9.5.1c)} $$

where V = fluid velocity in the pipe
w = specific weight of fluid, lb_f/ft^3

Because of the large number of valves and fittings on the market, each one possessing different internal passages and wall roughnesses, the designer should rely primarily on the data supplied by the manufacturer. In many cases the manufacturer does not provide this data unless requested. When this data is not available, values of equivalent length for Re > 1000 may be approximated from Table 3.9.5.1.

3.9.5.2 RESISTANCE COEFFICIENT, K. The resistance coefficient is used to correct for friction losses in fittings

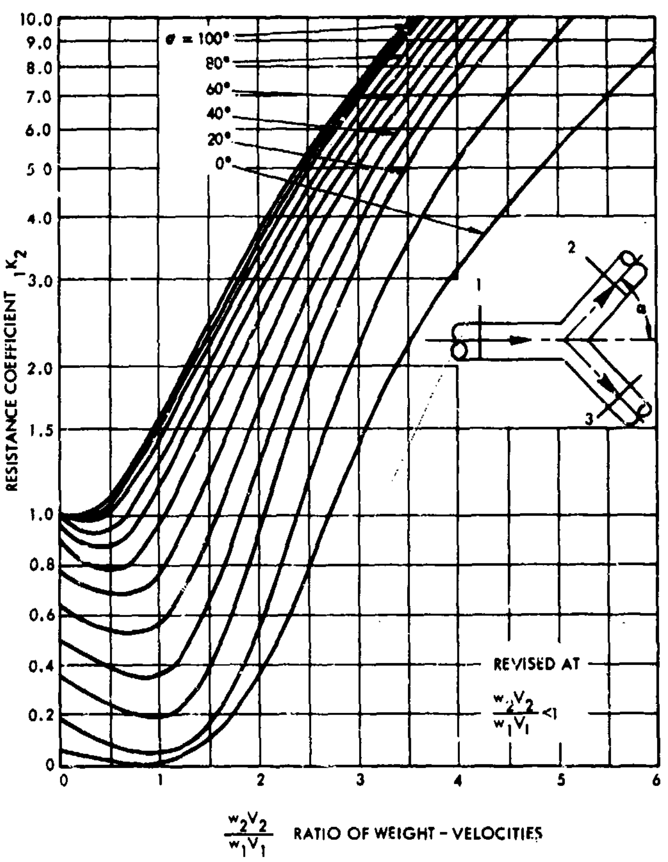

Figure 3.9.2.4b Resistance Coefficients K for Diverging Branches

(Adapted from "ASME Transactions," vol. 66, pp. 177-183, 1944, Vazsonyi)

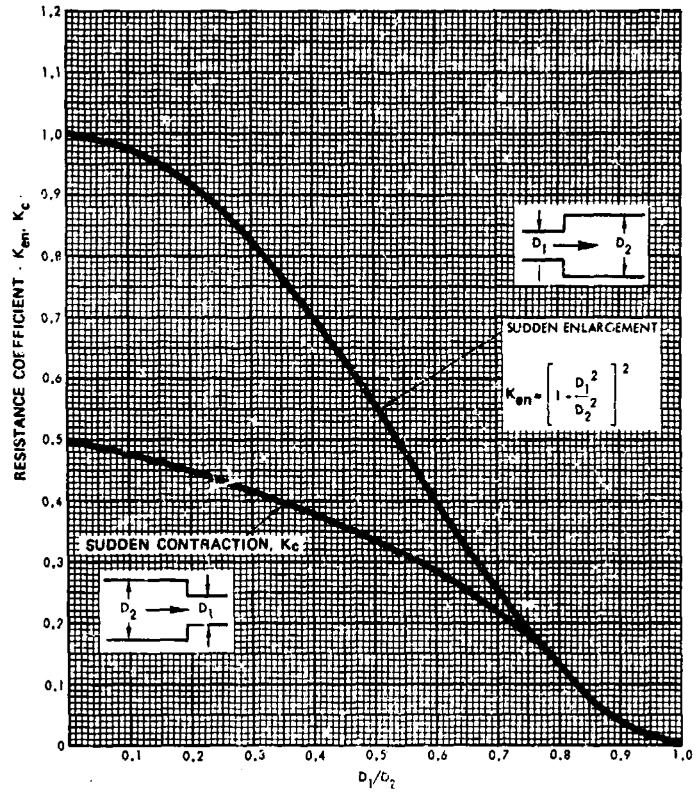

Figure 3.9.4a. Resistance Coefficients K for Sudden Enlargements and Contractions

(Reprinted from "Applied Fluid Mechanics;" O'Brien and Hickox, McGraw-Hill Book Company, Inc., 1937)

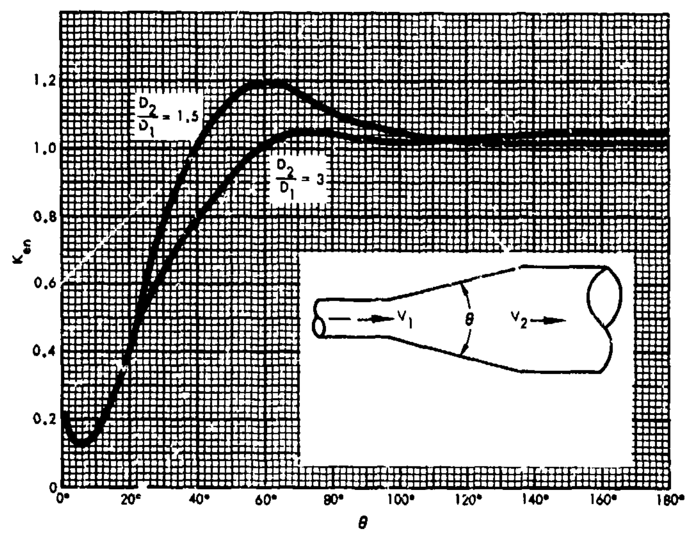

Figure 3.9.4b. Resistance Coefficients for Gradual Enlargements

(Reprinted with permission from "Handbook of Engineering Fundamentals," Eshbach, John Wiley and Sons, 1952)

and valves in the same sense as for change in pipe geometry, namely

$$\Delta P = K w \frac{V^2}{2g} \qquad \textbf{(Eq 3.9.5.2a)}$$

where ΔP = total pressure drop across valve (fitting) where upstream and downstream diameters are equal, lb_f/ft^2

w = specific weight, lb_f/ft^3

V = fluid velocity in pipe of nominal size of valve or fitting, ft/sec

K = resistance coefficient

Values of K may vary with surface finish, body shape, and internal geometry. Since each manufacturer's fitting or valve is different from his competitor's, the values of K for standard items (e.g., 45° elbow, tees) may be different. Whenever possible, manufacturers' data should be consulted first. If data are unavailable, typical K-values shown in Table 3.9.5.2 may be used to estimate system

Table 3.9.5.1. Representative Equivalent Lengths in Pipe Diameters (L/D) of Various Valves and Fittings

(References 143-1 and 23-38)

DESCRIPTION OF VALVE			EQUIVALENT LENGTH IN PIPE DIAMETERS (L/D)
Globe Valves	Conventional (no obstruction in seat)	Fully open	340
	Conventional	Fully open	450
	Y-Pattern (stem 60° from pipe centerline)	Fully open	175
	(stem 45° from pipe centerline)	Fully open	145
Angle Valves	Conventional (no obstruction in seat)	Fully open	145
	Conventional	Fully open	200
Gate Valves	Conventional Wedge, Disc, Double Disc, or Plug Disc	Fully open	13
		Three-quarter open	35
		One-half open	160
		One-quarter open	900
Check Valves	Conventional Swing	Fully open	135
	Clearway Swing	Fully open	50
	Globe Lift or Stop	Fully open	Same as Globe
	Angle Lift or Stop	Fully open	Same as Angle
	In-Line Ball	Fully open	150
Butterfly Valves (6-inch and larger)		Fully open 90°	20
		open 60°	225
		open 45°	850
		open 30°	5000
Fittings	90° Standard Elbow		30
	45° Standard Elbow		16
	90° Long Radius Elbow		20
	90° Street Elbow		50
	45° Street Elbow		26
	Square Corner Elbow		57
	Standard Tee (with flow through run)		20
	Standard Tee (with flow through branch)		60
	Close Pattern Return Bend		50

NOTE: D = Pipe inside diameter (nominal) — L/D values based on valve designs having seat diameters approximately equal to D.

Table 3.9.5.2. Resistance Coefficients for Various Valves and Fittings

(References 138-1, 143-1, and 23-38)

TYPE	K
Globe Valve (fully open)	10.0
Angle Valve (fully open)	5.0
Swing Check Valve (fully open)	2.5
Recessed Swing Check (fully open)	0.95
Ball Check Valve (fully open)	1.0
Gate Valve (fully open)	0.19
(half open)	2.0
(quarter open)	17.0
Butterfly Valve (fully open)	0.4
(half open)	13.0
(quarter open)	400.0
Plug Valve (fully open)	0.77
180° Bend	2.2
Mitered 90° Elbow	1.4
Short Radius 90° Elbow	0.9
Medium Radius 90° Elbow	0.75
Long Radius 90° Elbow	0.60
45° Elbow	0.42
Vaned Elbow	0.2

pressure drop. The K-values for valves are based on designs having seat diameters approximately equal to the internal diameter of the connecting pipe.

The value of K is also dependent on the amount of opening. K-values increase from the fully opened value as the valve closes. Table 3.9.5.2 shows the variations of K for typical valves and fittings. The relative variations of K compared with those of L/D in Table 3.9.5.1 are indicative of the variations to be found between various designs of a given component type.

For branch-off fittings (e.g., tees), values of K are dependent on the division of flow from the main stream to the branch and the angle of the branch to main stream. Various configurations of branch-off fittings with corresponding K values are shown in Figure 3.9.5.2. The total pressure loss is

$$\Delta P = K\,(\Delta P_{\text{main branch}}) \qquad \textbf{(Eq 3.9.5.2b)}$$

$$\Delta P = K\left(\frac{wV^2}{2g}_{\text{main branch}}\right) \qquad \textbf{(Eq 3.9.5.2c)}$$

3.10 FLUID TRANSIENTS

3.10.1 General

Many dynamic problems are associated with an increase of stored energy in the fluid as its pressure increases. A typical example is the pressure surge due to valve closing or opening. Liquid cannot be considered incompressible, and the pipe in which it flows is usually con-

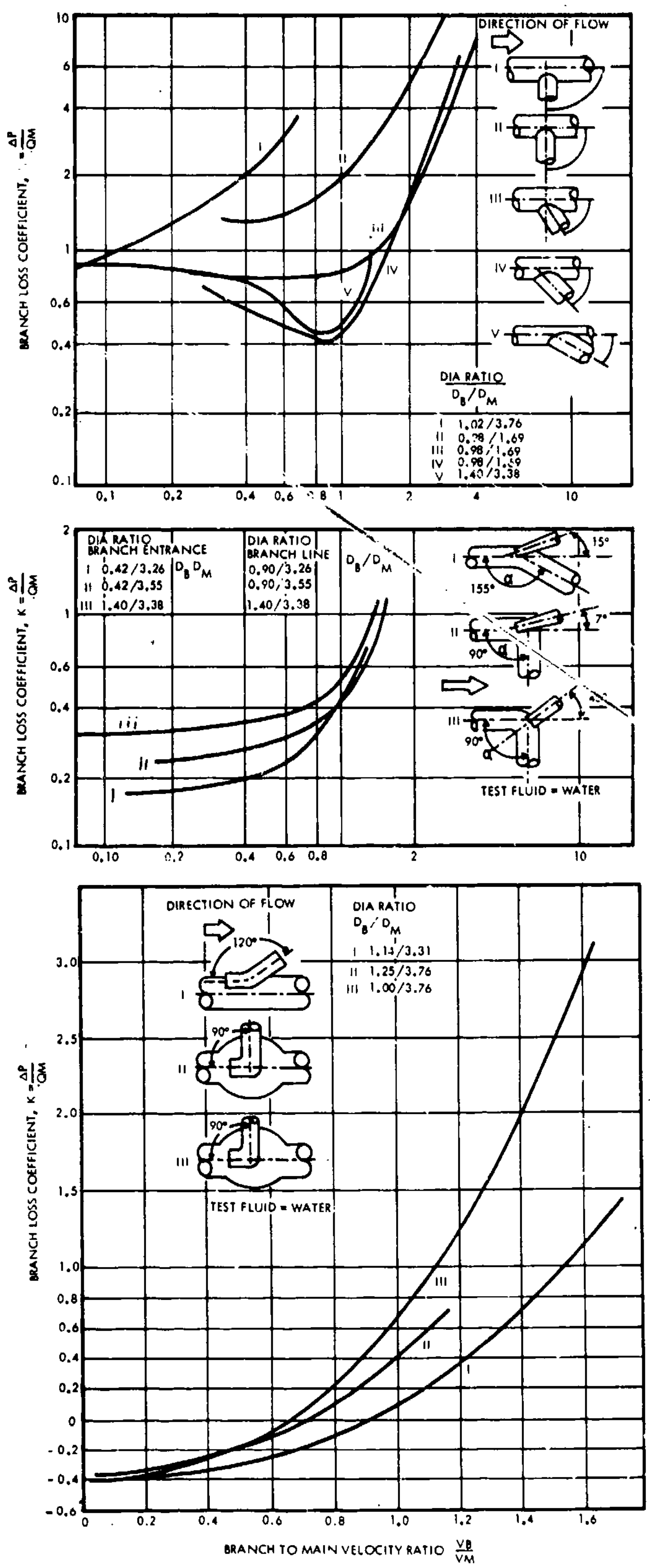

Figure 3.9.5.2. Losses Due to Branch-Off Fittings
(Adapted with permission from "Product Engineering," July 20, 1959, Daniels and Pelton, Copyright 1959 by McGraw-Hill Publishing Company, Inc.)

sidered elastic with a varying diameter due to a pressure change. A steady-state condition no longer exists.

The design of pneumatic systems and components often involves the behavior of gases under unsteady flow conditions. Many pneumatic problems may be reduced to venting or charging a volume or series of volumes through a constant area restriction to or from an infinite reservoir. Typical examples are: venting or charging of valve actuator cavities, discharge through a vent valve, and pressurization of gas storage bottles. The important parameters are the times to charge to, or vent from, a particular pressure, and the pressures after a given time of charging or venting.

The flow of a gas to or from a container may be either critical or subcritical, while gas expansion or compression in the fixed volume container may be either adiabatic or isothermal. The assumption of an adiabatic expansion (or compression) is valid for a well-insulated system, or where the pressure change is rapid. The assumption of isothermal expansion (or compression) is more accurate for slow pressure changes. Only perfect gases are considered.

3.10.2 Unsteady Flow in a Pipe

For a *frictionless* fluid flowing in an *elastic pipe*, the unsteady flow equations (sometimes called the wave equations) are

Continuity

$$-\frac{\partial V}{\partial x} = \left[\frac{1}{\beta} + \frac{d}{Eb}\right]\frac{\partial p}{\partial t} \qquad \text{(Eq 3.10.2a)}$$

Momentum

$$-\frac{\partial p}{\partial x} = \frac{\rho}{144\, g_c}\,\frac{\partial V}{\partial t} \qquad \text{(Eq 3.10.2b)}$$

where β = bulk modulus of fluid, lb_f/in^2

d = pipe diameter, in.

E = modulus of elasticity of pipe, lb_f/in^2

b = pipe thickness, in.

ρ = density, lb_m/ft^3

p = pressure, lb_f/in^2

V = velocity, ft/sec

x = directional distance, ft

Solution of these simultaneous equations may be expressed as a set of differential equations with hyperbolic cosine and sine functions, or as a set of time-difference equations. Solutions valid for small pressure variations are presented in Reference 45-1, page 86.

Unsteady flow with friction in a uniform elastic pipe may be treated by assuming that the friction loss is linearly proportional to the conduit velocity. The flow equations become

Continuity

$$-\frac{\partial V}{\partial x} = \left[\frac{1}{\beta} + \frac{d}{Eb}\right]\frac{\partial p}{\partial t} \qquad \text{(Eq 3.10.2c)}$$

Momentum

$$-\frac{\partial p}{\partial x} = \left[\frac{1}{X} + \frac{f}{D}\right] V + \frac{\rho}{144\, g_c}\,\frac{\partial V}{\partial t} \qquad \text{(Eq 3.10.2d)}$$

Solution of these simultaneous equations, using the Laplace-Mellin transformation, result in pressure and velocity relationships in terms of Bessel functions of form $I_0(\theta)$ and $I_1(\theta)$. Solutions are presented in Reference 60-4, page 363.

3.10.3 Valve Closing*

Pressure surges are a phenomenon of unsteady flow resulting from either acceleration or deceleration of a liquid. The more dense or incompressible the liquid, the greater the magnitude of pressure surges.

When a valve in an elastic pipeline is suddenly closed, the liquid is brought to rest progressively along the pipe with a "wave action" called *water hammer*. If a fluid is flowing with a constant velocity through a pipe and a downstream valve is closed instantaneously, the following sequence of events takes place. The fluid adjacent to the valve is stopped, and the unconsumed kinetic energy in the fluid is converted to a pressure head which compresses the fluid and expands the pipe walls. The velocities of adjacent particles are successively brought to rest, transforming kinetic energy to pressure energy. When the fluid is compressed due to the sudden stop, a pressure wave at sonic velocity, V_s, travels upstream to the end of the pipe. If the pipe length is L, then after L/V_s seconds the fluid will be at rest. At this time, the fluid density, pressure and pipe diameter will be greater than normal (Figure 3.10.3a). The pressure wave, after reaching the upstream end of the pipe, is reflected downstream as a negative pressure wave and returns to the valve at $2L/V_s$ seconds after valve closure. The hydrostatic pressure is reduced to its normal value and the pipe returns to its original diameter.

Due to fluid inertia, the fluid velocity is in the reverse direction separating the fluid from the downstream end of the pipe (Figure 3.10.3b). The pressure drops to subnormal values, the pipe contracts, and the fluid expands.

* *Method adapted from References 46-6 and 19-151.*

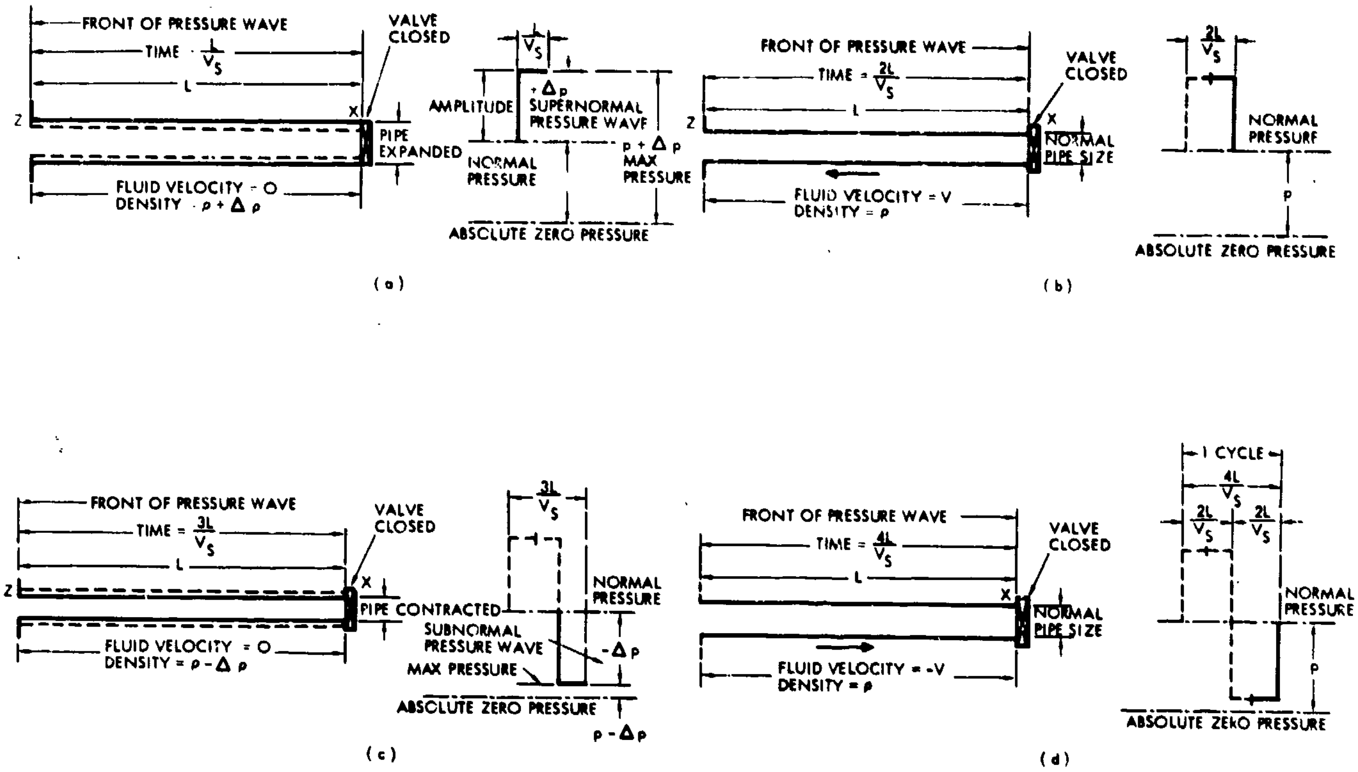

Figure 3.10.3a, b, c, d. Pressure-Time History for Instantaneous Valve Closure
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A wave of subnormal pressure travels upstream and after $3L/V_s$ seconds reaches the upstream end of the pipe (Figure 3.10.3c). After $4L/V_s$ seconds the pressure wave reaches the valve again, the liquid moves downstream, and the pressure and density return to normal values (Figure 3.10.3d).

If the fluid were ideal there would be no resistance to flow, and the surge phenomenon would continue to repeat indefinitely. With friction and imperfect pipe elasticity the pressure amplitudes successively diminish until the original kinetic energy is absorbed.

The explanation of water hammer presented in the previous paragraphs was made assuming instantaneous closure ($t = 0$), but valve closure may also be rapid or slow. In real cases, valves require a finite time to close. If closure takes place in time $0 < t < 2L/V_s$, closure is termed *rapid*, and the incremental pressure rise is

$$\Delta P = \frac{68.094\, w\, V}{g\sqrt{w\left(\frac{1}{\beta} + \frac{d}{Eb}\right)}} \qquad \textbf{(Eq 3.10.3a)}$$

where ΔP = pressure rise, lb_f/ft^2

w = specific weight, lb_f/ft^3

V = fluid velocity, ft/sec

g = 32.2 ft/sec^2

β = bulk modulus of fluid, lb_f/in^2

E = modulus of elasticity of pipe, lb_f/in^2

b = pipe thickness, in.

d = pipe diameter, in.

A typical pressure trace is shown in Figure 3.10.3e. The maximum pressure is

$$P_{max} = \Delta P + P_s \qquad \textbf{(Eq 3.10.3b)}$$

where P_s = the normal static pressure, lb_f/ft^2

ΔP = incremental pressure rise, lb_f/ft^2

The velocity of the pressure wave in the elastic pipe may be expressed as

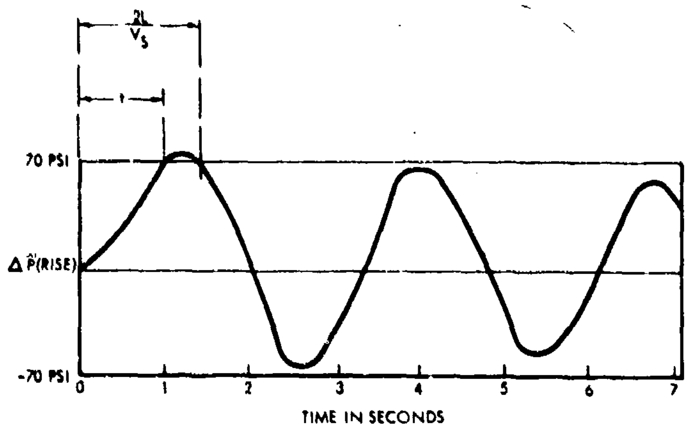

Figure 3.10.3e. Pressure-Time History for Rapid Valve Closure

$$V_s = \frac{68{,}094}{\sqrt{w\left[\frac{1}{\beta} + \frac{d}{Eb}\right]}} \qquad \textbf{(Eq 3.10.3c)}$$

For normal pipe dimensions, the pressure wave velocity for water ranges between 2000 and 4500 ft/sec, but is always less than 4720 ft/sec. For liquid oxygen and liquid hydrogen, values of V_s commonly used are 3370 ft/sec and 4010 ft/sec, respectively. (See Sub-Topic 12.2.2 for data.)

The fluid velocity is related to the expected incremental pressure rise by the equation

$$V = \frac{g \Delta P}{w V_s} \qquad \textbf{(Eq 3.10.3d)}$$

where V = fluid velocity, ft/sec

V_s = velocity of pressure wave, ft/sec

g = 32.2 ft/sec^2

ΔP = pressure rise, lb$_f$/ft^2

w = specific weight, lb$_f$/ft^3

The magnitude of the pressure rise as a function of water velocity may be found using Figure 3.10.3f.

If valve closing time $t > 2L/V_s$, the pressure rise will be substantially less than for rapid close. Since the unloading pressure wave returns before valve closure is complete, any further increase will be avoided. This closure is called *slow closure*, and a typical pressure history is shown in Figure 3.10.3g. The number of pressure wave intervals, N, during valve closure is

$$N = \frac{t}{\frac{2L}{V_s}} \qquad \textbf{(Eq 3.10.3e)}$$

where t = valve closure time, sec

L = pipe length, ft

V_s = velocity of pressure wave, ft/sec

The amplitude of the pressure wave during any interval is proportional to the decrease in velocity during the interval, and the total pressure change is the sum of the positive and negative pressure amplitudes produced. Thus the *pressure rise* may be given by the expression

(Eq 3.10.3f)

$$\sum_{i=n}^{i=1} \Delta P_i = \Delta P_n - \Delta P_{n-1} + \Delta P_{n-2} \cdots \pm \Delta P_1$$

where
$$\Delta P_n = \frac{w V_s}{g} \Delta V_n \qquad \textbf{(Eq 3.10.3g)}$$

The total upstream pressure is

$$P = P_{initial} + \sum_{i=n}^{i=1} \Delta P_i \qquad \textbf{(Eq 3.10.3h)}$$

where n refers to the number of intervals taken until n = N.

The velocity decrease in n-intervals is

(Eq 3.10.3i)

$$\sum_{i=1}^{i=n} \Delta V_i = \Delta V_1 + \Delta V_2 + \Delta V_3 \cdots + \Delta V_n$$

The velocity of the fluid after the nth interval is

$$V_n = V_{initial} - \sum_{i=1}^{i=n} \Delta V_i \qquad \textbf{(Eq 3.10.3j)}$$

The fluid velocity initially may be denoted in terms of valve port size

(Eq 3.10.3k)

$$V = \sqrt{\frac{2g}{w}}\left(\frac{A_o}{A}\right) C_D \sqrt{P_1 - P_2}$$

or

$$V = \phi_o \sqrt{P_1 - P_2} \qquad \textbf{(Eq 3.10.3l)}$$

At the nth interval

$$V_n = \phi_n \sqrt{P_{n_1} - P_{n_2}} \qquad \textbf{(Eq 3.10.3m)}$$

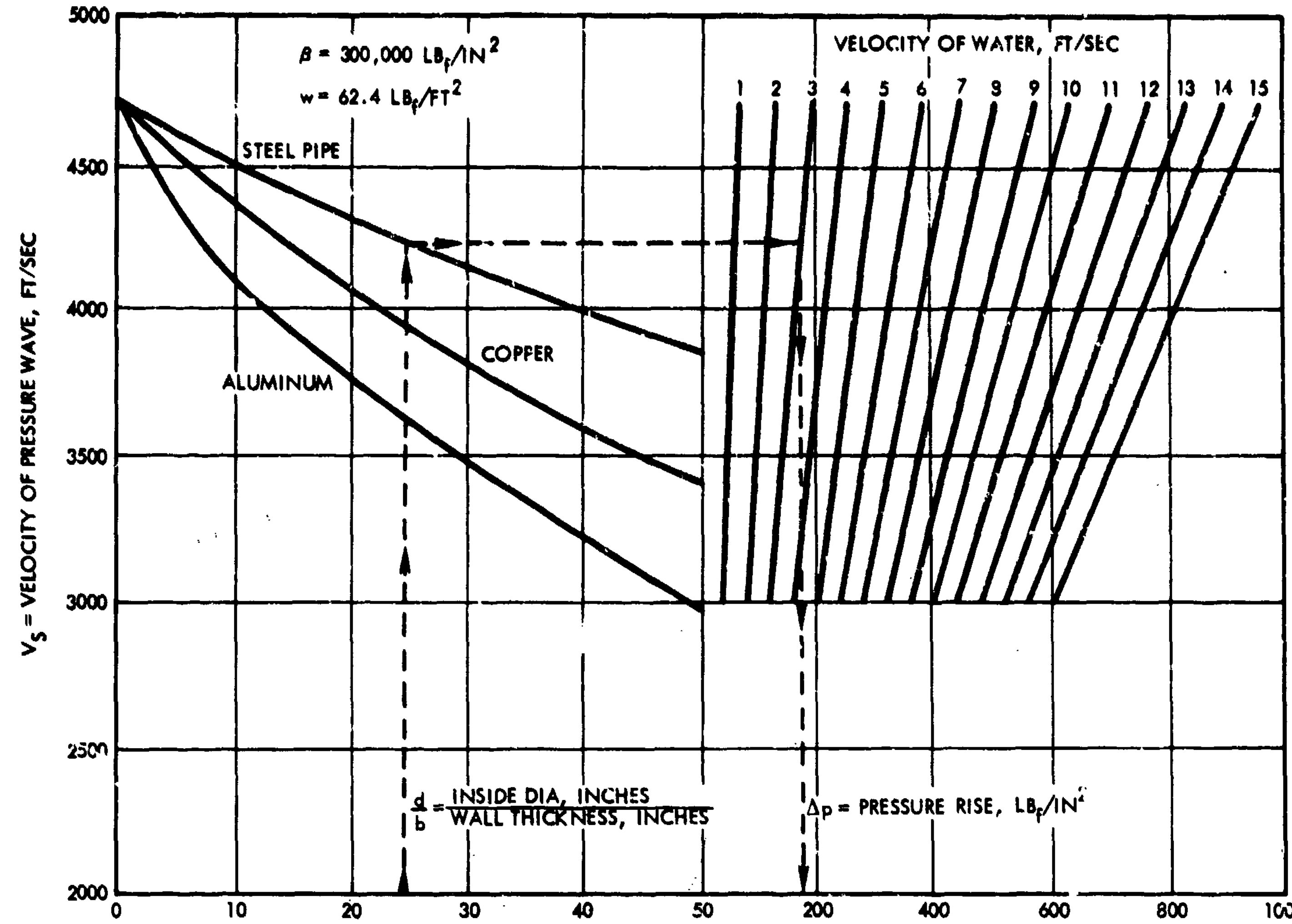

Figure 3.10.3f. Graph for Calculation of Pressure Surges in Water Systems
(Adapted from "Valve World," 1957, vol. 48, no. 3, E. C. Petrie and E. P. DeCraene, Crane Company)

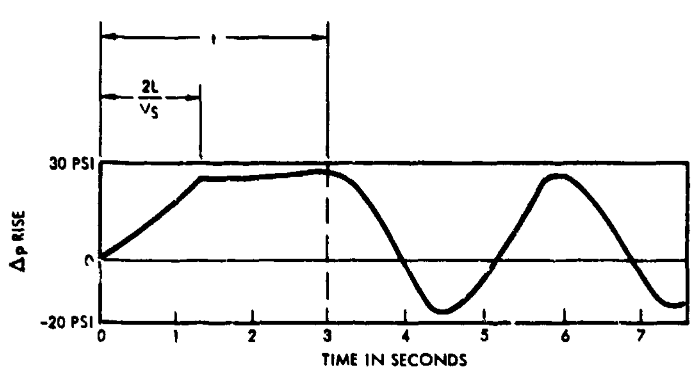

Figure 3.10.3g. Pressure-Time History for Slow Valve Closure

Substituting values of ΔP_i from Equation (3.10.3h) the velocity equation becomes

(Eq 3.10.3n)

$$V_n = \phi_n \sqrt{\left[P_{initial} + \sum_{i=n}^{i=1} \Delta P_i\right] - P_2}$$

If the rate of decrease in orifice area (valve closing) is uniform and C_D is assumed constant, the value of ϕ_n may be expressed as

$$\phi_n = \left[1 - n\left(\frac{2L}{V_s t}\right)\right]\phi_o \quad \textbf{(Eq 3.10.3o)}$$

or

$$\phi_n = [1 - n/N]\,\phi_o \qquad \textbf{(Eq 3.10.3p)}$$

where t refers to valve closure time.

Example. To demonstrate these equations, consider a valve with a closing time of 0.10 seconds. If the fluid is water flowing in a 50-foot stainless steel pipe with d/b = 40, the velocity of the pressure wave is approximately 4000 ft/sec. The number of pressure wave intervals, N, from Equation (3.10.3e) is 4. If the original fluid velocity equals 8 ft/sec and static pressure upstream and downstream are respectively 500 psia and 14.7 psia, then by Equation (3.10.3l)

$$\phi_o = \frac{8}{\sqrt{500\text{-}14.7}} = 0.362$$

For *first interval,* Equation (3.10.3p) gives

$$\phi_1 = (1 - 1/4)\,(0.362) = 0.272$$

From Equations (3.10.3g), (3.10.3j) and (3.10.3n)

$$\Delta p_1 = \frac{(62.4)\,(4000)}{(144)\,(32.2)}\,\Delta V_1 = 53.9\,\Delta V_1$$

$$V_1 = 8 - \Delta V_1$$

$$V_1 = 0.272\sqrt{(500 + \Delta p_1) - 14.7}$$

Solving the above equations simultaneously

$$\Delta V_1 = 1.5 \text{ ft/sec}$$

$$\Delta p_1 = (53.9)\,(1.5) = 81 \text{ psi}$$

The fluid velocity for the first interval becomes

$$V_1 = V - \Delta V_1 = 8 - 1.5 = 6.5 \text{ ft/sec}$$

$$p_1 = p_{initial} + \Delta p_1 = 500 + 81 = 581 \text{ psi}$$

For the second interval

$$\phi_2 = (1 - 1/2)\,(0.362) = 0.181$$

$$\Delta p_2 = 53.9\,\Delta V_2$$

$$V_2 = 8 - (1.5 + \Delta V_2)$$

$$V_2 = 0.181\sqrt{500 + (\Delta p_2 - 81) - 14.7}$$

Solving the above equations

$$V_2 = 2.38 \text{ ft/sec}$$

$$\Delta p_2 = 112 \text{ psi}$$

Therefore

$$V_2 = V_{initial} - (\Delta V_1 + \Delta V_2)$$

$$= 8 - (1.5 + 2.38) = 4.12 \text{ ft/sec}$$

$$p_2 = p_{initial} - \Delta p_1 + \Delta p_2$$

$$= 500 - 81 + 112 = 531 \text{ psi}$$

The third and fourth intervals can be computed in a similar manner.

Where water hammer will occur, pressure lines should be supported at short intervals to prevent vibrational damage to fittings, flanges, and seals. In addition to surge damage, noise can occur, even at low absolute pressures. Noise depends on the time rate of change of pressure and can be reduced by adding capacitive volume or elasticity to absorb the pressure waves. Common methods include use of vibration isolators, mufflers, surge chambers, suppressors, and accumulators.

Noise can also be reduced by modifying the valve to smooth out flow during closing. Examples include tapering valve spools to throttle flow, using dashpots to soften end-of-stroke shock, and controlling valve actuation time to reduce pressure rise. A brief discussion of typical pulsation dampeners is presented in Reference 19-188.

3.10.4 Valve Opening*

When a downstream valve is opened rapidly, pressure waves similar to those produced in valve closure travel up and down the pipe until damping develops a steady flow. Analytical treatment is similar to that employed in water hammer analysis except that the pressure change is limited to the difference between the initial pressures upstream and downstream of the valve.

As with valve closing, water hammer effects vary with the rate of valve opening with one important exception. When the time of valve opening is either instantaneous or slow ($t_{open} >> 2L/V_s$), the effect of the fluid's elastic property is negligible and no pressure build-up occurs (Figure 3.10.4). However, if valve opening time is intermediate between these two extremes, pressure surges can occur.

Rapid opening times may be examined by considering a valve which is partially opened instantaneously and held constant thereafter. Upon opening, the fluid behind the valve experiences a pressure drop, Δp, and propagates a compression wave upstream. The magnitude of the pres-

**Adapted from Reference 46-6.*

sure drop is, of course, dependent on the amount of valve opening. When the compression wave returns, only part of the wave is transmitted because the valve is only partially open. The rest of the wave is reflected with an intensity

$$\Delta p(\text{rise}) = \Delta p\left(\frac{1-\phi}{1+\phi}\right) \qquad \textbf{(Eq 3.10.4a)}$$

where ϕ = percentage area of valve opening.

The maximum pressure in the [illegible] is

$$p_{max} = p_{initial} + \Delta p\left(\frac{1-\phi}{1+\phi}\right) \qquad \textbf{(Eq 3.10.4b)}$$

The pressure rise is always less than $p_{initial}$ in the pipeline before opening.

If the valve opening is moderate the velocity expression is

$$V = C_D\,\phi\sqrt{2g\frac{\Delta P}{w}} \qquad \textbf{(Eq 3.10.4c)}$$

Example. Consider a pipeline carrying water at 200 psig when a valve ($C_D = 0.8$) opens rapidly to an area one-fourth the total area. ($\phi = 1/4$). If the pipe is stainless steel with a d/b ratio of 40, then from Figure 3.10.3f, the pressure wave velocity is 4000 ft/sec.

Since the initial velocity is zero, Equation (3.10.3d) reduces to

$$\Delta p = 53.9\ V$$

and from Equation (3.10.4c)

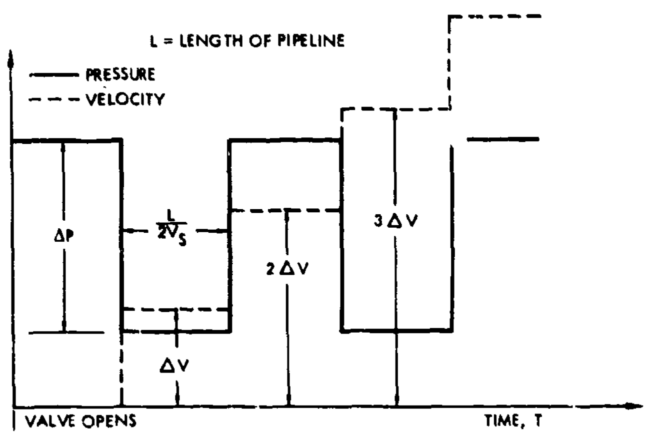

Figure 3.10.4. Pressure-Time History for Instantaneous Valve Opening

(Reprinted with permission from General Electric Quarterly Report No. 2, Contract NAS 8-4012, October 11, 1962)

$$V = (0.8)\ (1/4)\sqrt{\frac{(200-\Delta p)\ (144)\ (2g)}{62.4}}$$

Solving the above equations simultaneously

$$V = 3.72\ \text{ft/sec}$$

and

$$\Delta p = 199\ \text{psi}$$

The maximum pressure in the pipeline is

$$p_{max} = p_o + \Delta p\left(\frac{1-\phi}{1+\phi}\right)$$

$$p_{max} = 200 + 199\left(\frac{1-1/4}{1+1/4}\right) = 319\ \text{psi}$$

3.10.5 Venting or Charging Gas from a Single Volume*

Time solutions for adiabatic and isothermal venting and charging may be found in Table 3.10.5a. Both subcritical and critical flows are considered. For critical flows the solutions are exact and can be used directly; for subcritical flow the equations cannot be solved as easily but can be evaluated by graphical integration.

Times for combined subcritical and critical flows may be found by adding the subcritical and critical times. A general form of the pressure-response equations may be written as

$$t = C\ (\tau_f - \tau_i) \qquad \textbf{(Eq 3.10.5)}$$

where
t = time
C = constant for any process and gas (see Table 3.10.5b)
$\tau_f - \tau_i$ = difference of the sums of subcritical and critical flow integrals

Equation (3.10.5) may be used to determine all times required if values of C are computed and Figures 3.10.5a, b and c are used to evaluate τ.

To demonstrate the usefulness and generality of the equations in Table 3.10.5a the following sample problem is given:

Example. Find the time required to vent a cylinder of helium from 5000 psia to 500 psia. The data are as follows:

Tank volume, V	= 20 in³
Effective Flow Area, $C_D A_o$	= 0.08 in²
Downstream pressure, p_2	= 40 psia
Initial temperature, T_i	= 530 °R

Step 1. Assume adiabatic venting

*Much of the Sub-Topic was adapted from Reference 33-2.

Table 3.10.5a. Time-Pressure Equations
(Reference 33-2)

Adiabatic Charging:
Subcritical

$$t_{SCR} = \frac{V}{\gamma\, C_1\, C_D\, A_O\, R \sqrt{T_1}} \int_{\frac{P_1}{P_1}}^{\frac{P_f}{P_1}} \frac{1}{a} \frac{d\left(\frac{P}{P_1}\right)}{\sqrt{\left(\frac{P}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P}{P_1}\right)^{\frac{\gamma+1}{\gamma}}}}$$

Adiabatic Charging:
Critical

$$t_{CR} = \frac{V}{\gamma\, C_1\, C_D\, A_O\, R \sqrt{T_1}} \left(\frac{P_f}{P_1} - \frac{P_1}{P_1}\right)$$

Isothermal Charging:
Subcritical

$$t_{SCR} = \frac{V \sqrt{T_1}}{C_1\, C_D\, A_O\, R\, T_1} \int_{\frac{P_1}{P_1}}^{\frac{P_f}{P_1}} \frac{1}{a} \frac{d\left(\frac{P}{P_1}\right)}{\sqrt{\left(\frac{P}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P}{P_1}\right)^{\frac{\gamma+1}{\gamma}}}}$$

Isothermal Charging:
Critical

$$t_{CR} = \frac{V \sqrt{T_1}}{C_1\, C_D\, A_O\, R\, T_1} \left(\frac{P_f}{P_1} - \frac{P_1}{P_1}\right)$$

Adiabatic Venting:
Subcritical

$$t_{SCR} = \frac{V}{\gamma\, C_1\, C_D\, A_O\, R \sqrt{T_1} \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{2\gamma}}} \int_{\frac{P_2}{P_1}}^{\frac{P_2}{P_f}} \frac{1}{a} \frac{d\left(\frac{P_2}{P}\right)}{\sqrt{\left(\frac{P_2}{P}\right)^{\frac{\gamma+3}{\gamma}} - \left(\frac{P_2}{P}\right)^{\frac{2(\gamma+1)}{\gamma}}}}$$

Adiabatic Venting:
Critical

$$t_{CR} = \frac{V}{\gamma\, C_1\, C_D\, A_O\, R \sqrt{T_1} \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{2\gamma}}} \left[\frac{2\gamma}{\gamma-1} \left\{\left(\frac{P_2}{P_f}\right)^{\frac{\gamma-1}{2\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{2\gamma}}\right\}\right]$$

Isothermal Venting:
Subcritical

$$t_{SCR} = \frac{V}{C_1\, C_D\, A_O\, R \sqrt{T}} \int_{\frac{P_2}{P_1}}^{\frac{P_2}{P_f}} \frac{1}{a} \frac{d\left(\frac{P_2}{P}\right)}{\frac{P_2}{P}\sqrt{\left(\frac{P_2}{P}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P}\right)^{\frac{\gamma+1}{\gamma}}}}$$

Isothermal Venting:
Critical

$$t_{CR} = \frac{V}{C_1\, C_D\, A_O\, R \sqrt{T}} \left[\ln \frac{P_2}{P_f} - \ln \frac{P_2}{P_1}\right]$$

Table 3.10.5b. Definition of Constant, C

(Reference 33-2)

PROCESS	C (SEC)
Adiabatic Charging	$\dfrac{V}{C_1 C_D A_0 \gamma R \sqrt{T_1}}$
Isothermal Charging	$\dfrac{V\sqrt{T_1}}{C_1 C_D A_0 R T_1}$
Adiabatic Venting	$\dfrac{V}{\gamma C_1 C_D A_0 R \sqrt{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{2\gamma}}}$
Isothermal Venting	$\dfrac{V}{C_1 C_D A_0 R \sqrt{T}}$

Notation for Tables 3.10.5a and 3.10.5b

$$a = \frac{1}{C_1}\sqrt{\frac{2 g \gamma}{\frac{g_c}{g} R (\gamma - 1)}}$$

C_D = discharge coefficient

$$C_1 = \sqrt{\frac{\gamma g}{R\left(\frac{g_c}{g}\right)}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}, \ \frac{\sqrt{{}^\circ R}}{sec}$$

T = tank temperature, °R

P = tank pressure, lb_f/ft^2

V = tank volume, ft^3

R = gas constant, ft-lb_f/lb_m °R

A_o = throat area, ft^2

Subscripts:

f = final

i = initial

1 = upstream of nozzle

2 = downstream of nozzle

Step 2. (From Table 3.10.5b)

$$C = \frac{V}{\gamma C_1 C_D A_o R \sqrt{T_i}\left(\frac{p_2}{p_i}\right)^{\frac{\gamma-1}{2\gamma}}}$$

$C = 1.76 \times 10^{-2}$ sec

Step 3.

$$\frac{p_2}{p_i} = \frac{40}{5000} = 0.008$$

$$\frac{p_2}{p_f} = \frac{40}{500} = 0.08$$

Step 4. (From Figure 3.10.5b)

$\tau_f = \tau_{0.08} = 3.08$

$\tau_i = \tau_{0.008} = 2.1$

Step 5. Substituting into Equation (3.10.5)

$t = C\,(\tau_f - \tau_i)$

$t = 1.76 \times 10^{-2}\ (0.98)$

$t = 1.72 \times 10^{-2}$ sec

3.10.6 Venting or Charging Two Volumes in Series

Time solutions of isothermal venting or charging of two volumes interconnected by three constant area restrictions (Figure 3.10.6a) may be found by the simultaneous solution of the following equations

(Eq 3.10.6a)

$$\frac{d}{dt}\left(\frac{P_1}{P_0}\right) = K_1\,\mathcal{F}\left(\frac{P_1}{P_0}\right) - K_2\frac{P_1}{P_0}\,\mathcal{F}\left(\frac{P_2}{P_1}\right)$$

(Eq 3.10.6b)

$$\frac{d}{dt}\left(\frac{P_2}{P_0}\right) = K_3\frac{P_1}{P_0}\,\mathcal{F}\left(\frac{P_2}{P_1}\right) - K_4\frac{P_2}{P_0}\,\mathcal{F}\left(\frac{P_3}{P_2}\right)$$

where

$$\mathcal{F}\left(\frac{P_1}{P_0}\right) = \frac{1}{C_1}\sqrt{\frac{2g\gamma}{(\gamma-1)\left(\frac{g_c}{g}\right)(R)}\left[\left(\frac{P_1}{P_0}\right)^{\frac{2}{\gamma}} - \left(\frac{P_1}{P_0}\right)^{\frac{\gamma+1}{\gamma}}\right]}$$

and $$C_1 = \sqrt{\frac{\gamma g}{\left(\frac{g_c}{g}\right)R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

Similar functions may be written by substituting proper pressure ratios into Equations (3.10.6a) and (3.10.6b). Coefficients K_n are as follows:

$$K_1 = \frac{C_1 R\sqrt{T}\,(C_D A)_{10}}{V_1}$$

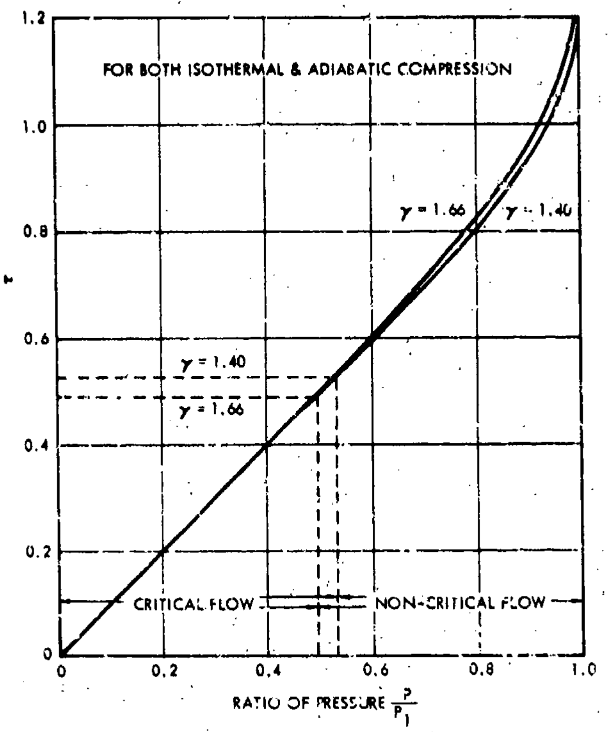

Figure 3.10.5a. Charging a Constant Volume Reservoir
(Adapted from AFBMD-TR 60-72, II. Robertshaw-Fulton Controls Company, Aeronautical and Instrument Division)

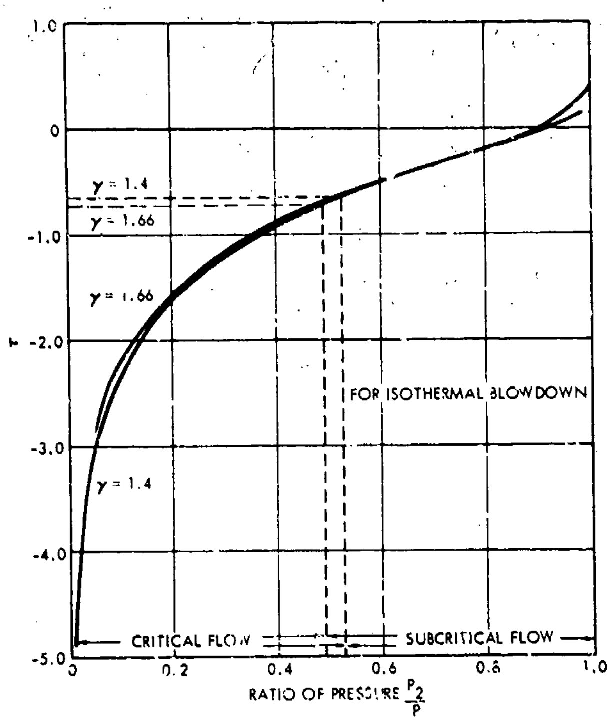

Figure 3.10.5c. Venting a Constant Volume Reservoir (Isothermal)
(Adapted from AFBMD-TR 60-72, II, Robertshaw-Fulton Controls Company, Aeronautical and Instrument Division)

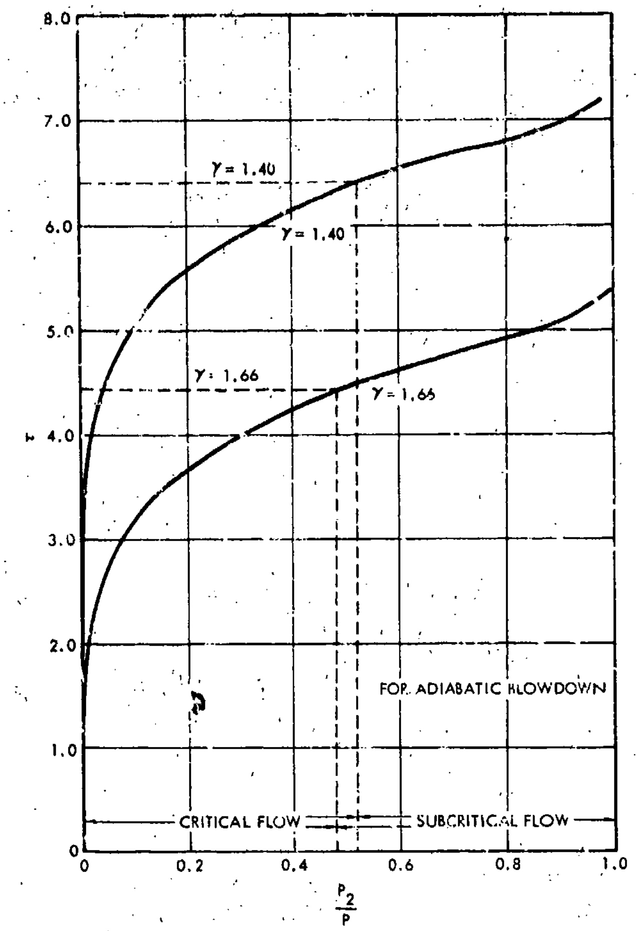

Figure 3.10.5b. Venting a Constant Volume Reservoir (Adiabatic)
(Adapted from AFBMD-TR-60-72, II, Robertshaw-Fulton Controls Company, Aeronautical and Instrument Division)

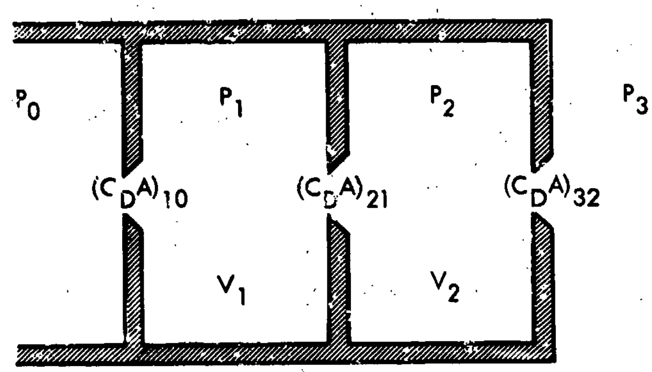

Figure 3.10.6a. System of Two Volumes in Series

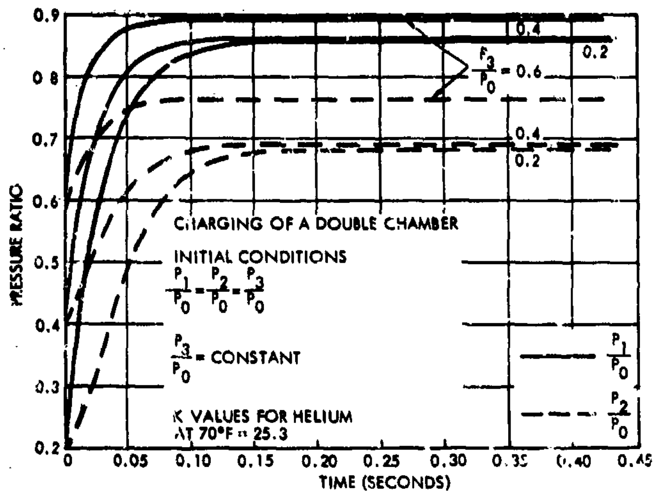

Figure 3.10.6b. Charging of a Double Chamber (K = 25.3 sec⁻¹)

(Adapted from AFBMD-TR-60-72, II, Robertshaw-Fulton Controls Company, Aeronautical and Instrument Division)

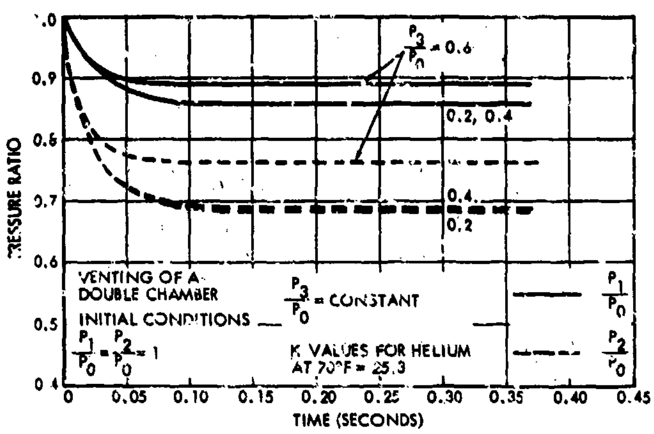

Figure 3.10.6c. Venting of a Double Chamber (K = 25.3 sec⁻¹)

(Adapted from AFBMD-TR-60-72, II, Robertshaw-Fulton Controls Company, Aeronautical and Instrument Division)

$$K_2 = \frac{C_1 R \sqrt{T}\,(C_D A)_{21}}{V_1}$$

$$K_3 = \frac{C_1 R \sqrt{T}\,(C_D A)_{21}}{V_2}$$

$$K_4 = \frac{C_1 R \sqrt{T}\,(C_D A)_{32}}{V_2}$$

where V = tank volume, ft³

T = absolute temperature, °R

R = gas constant, ft-lb$_f$/lb$_m$ °R

$C_D A$ = effective orifice area, ft²

P = pressure, lb$_f$/ft²

Pressure-time plots may be constructed based on solutions of the above equations. Figures 3.10.6b and 3.10.6c represent typical plots for identical K values equal to 25.3 sec⁻¹. The graphs are plotted for helium but are sufficiently accurate for nitrogen or air. For different K values, the time coordinate may be corrected by the relation

$$t\,K = t'\,K' \qquad \text{(Eq 3.10.6c)}$$

where primed values refer to the desired values, provided all K′ values are equal. If one of the K′ values is different from the other three, the graphs cannot be used. Additional graphs, varying individual K values by factors of ten, are given in Reference 33-2.

3.11 FLUID FLOW THROUGH NARROW PASSAGES

3.11.1 General

Liquid flow through narrow passages is characteristic of capillary flow in tubes, leakage past seals, and even flow through porous materials. The amount of flow depends on the passage configuration and size which may vary from a few millimeters to less than a micron. Four passage configurations will be considered:

a) smooth circular conduits (capillarity)

b) stationary flat plates

c) annuli

d) tortuous passages.

In each of the passage configurations discussed, only steady flow will be considered. Reynolds number will usually be less than 2000 and laminar flow will prevail. Although fully developed laminar flow may appear macroscopic relative to molecular flow, leakage rates as low as 10⁻⁸ lb$_m$/hr can be calculated in this manner (Reference 46-6).

3.11.2 Circular Tubes

The steady flow of a fluid through a smooth conduit may be treated by the Hagen-Poiseuille Law for laminar flow (Sub-Topic 3.9.1). The volumetric flow rate, determined by integration of velocity profile (Figure 3.11.2), is

$$q = \frac{\pi d^4}{128\,\mu_f l}\,(p_1 - p_2) \qquad \text{(Eq 3.11.2a)}$$

The mass flow rate is

$$\dot{m} = \frac{\rho \pi d^4}{128\,\mu_f l}\,(p_1 - p_2) \qquad \text{(Eq 3.11.2b)}$$

where q = volumetric flow rate, in³/sec

$\dot{m}$ = mass flow rate lb$_m$/sec

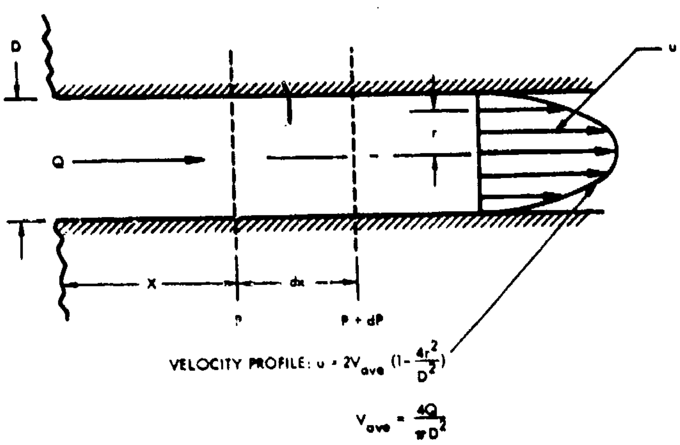

Figure 3.11.2. Flow Through Small Circular Tubes
(Reprinted with permission from "Fluid Power Control," Blackburn, Reethof, and Shearer, MIT Press, Cambridge, Massachusetts, 1960, p. 58)

ρ = density, lb_m/in^3
d = tube diameter, in.
l = length, in.
μ_f = absolute viscosity, lb_f sec/in²
p_1, p_2 = upstream and downstream pressures, respectively, lb_f/in^2

The above equation assumes zero velocity at the wall, μ_f is a constant, and the velocity profile is identical along the tube.

As the tube diameter becomes very small, molecular flow becomes more predominant. With a capillary radius of 10^{-3} cm the flow of air at 20°C is approximately 90 percent laminar. As the capillary radius decreases to 10^{-4} cm and then to 10^{-5} cm, the flow of air becomes 50 percent laminar and 90 percent molecular, respectively. The relative flow rates compared to air for different gases is shown in Table 3.11.2 as a function of capillary radius. At *extremely low pressures* ($p \leq 10^{-4}$psi) molecular flow exists entirely (Reference 164-1).

Table 3.11.2. Relative Leak Rates

(Reference 164-1)

GAS	CAPILLARY RADIUS 10^{-3}cm	10^{-4}cm	10^{-5}cm
Air	1.0	1.0	1.0
Argon	0.83	0.84	0.85
Carbon dioxide	1.20	1.04	0.85
Helium	1.09	1.77	2.51
Hydrogen	2.22	2.90	3.62

Assuming that molecules never collide with other molecules but only with the capillary wall, the volumetric flow rate may be computed by

(Eq 3.11.2c)

$$Q = \left(\frac{\pi}{2}\right)^{3/2} \frac{D^3}{8L} (g_c RT)^{1/2} (P_1 - P_2) \frac{1}{\bar{P}}$$

where
Q = flow rate, ft³/sec
D = tube radius, ft
L = tube length, ft
$g_c = 32.2 \frac{lb_m}{lb_f}$ ft/sec²
R = gas constant, ft-lb_f/lb_m °R
T = absolute temperature, °R
P_1, P_2 = pressure across capillary, lb_f/ft^2
$\bar{P}$ = average pressure, lb_f/ft^2

3.11.3 Parallel Flat Plates

The laminar flow rate of a liquid past parallel plates (Figure 3.11.3) is calculated by the expression

$$q = \frac{b\,h^3}{12\,\mu_f l} (p_1 - p_2) \qquad \textbf{(Eq 3.11.3a)}$$

where
q = volumetric flow rate, in³/sec
b = passage width, in.
h = passage height, in.
μ_f = absolute viscosity, lb_f sec/in²
l = path length, in.
p_1, p_2 = upstream and downstream pressures, respectively, lb_f/in^2

Leakage past flanges of liquid oxygen, RP-1, and liquid hydrogen using the above equation was calculated in Reference 46-6.

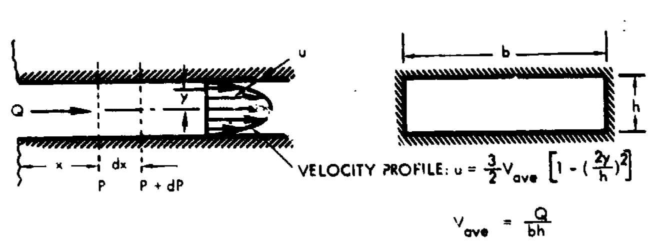

Figure 3.11.3. Flow Past Stationary Flat Plates
(Reprinted with permission from "Fluid Power Control," Blackburn, Reethof, and Shearer, MIT Press, Cambridge, Massachusetts, 1960, p. 58)

For laminar flow of gas past stationary flat surfaces, the mass flow rate is

$$\dot{m}_L = \frac{bh^3}{12\,RT\,\mu_f l}\,(p_1^2 - p_2^2) \qquad \text{(Eq 3.11.3b)}$$

where $\dot{m}_L$ = mass flow rate, lb_m/sec
b = passage width, in.
h = passage height, in.
l = passage length, in.
μ_f = absolute viscosity, $lb_f\ sec/in^2$
p_1, p_2 = upstream and downstream pressures, respectively, lb_f/in^2
T = temperature, °R
R = gas constant, in-$lb_f/(lb_m)$ °R

For the molecular flow of a gas (Reference 46-7) :

$$\dot{m}_M = \frac{4}{3}\sqrt{\frac{2}{\pi}}\,\frac{bh^2}{l}\,\sqrt{\frac{\rho\,g_c}{\bar{p}}}\,(p_1 - p_2) \qquad \text{(Eq 3.11.3c)}$$

where $\dot{m}_M$ = mass flow rate, lb_m/sec
b = passage width, in.
h = passage height, in.
l = passage length, in.
ρ = density, lb_m/in^3
$\bar{p}$ = average pressure, lb_f/in^2
p_1, p_2 = upstream and downstream pressures, respectively, lb_f/in^2

In terms of viscosity and mean free path of the molecules, the mass flow rate for molecular flows is (Reference 46-7) :

$$\dot{m}_M = \frac{16\,b\,h^3\,(p_1 - p_2)\,\bar{p}\,\lambda}{9\,\pi\,\mu_f\,RTl\,h} \qquad \text{(Eq 3.11.3d)}$$

where $\dot{m}_M$ = mass flow rate lb_m/sec
b = passage width, in.
h = passage height, in.
l = passage length, in.
p_1, p_2 = upstream and downstream pressures, respectively, lb_f/in^2
$\bar{p}$ = average pressure, lb_f/in^2
μ_f = absolute viscosity, $lb_f\ sec/in^2$
R = gas constant, in-lb_f/lb_m °R
T = absolute temperature, °R
λ = mean free path, in.
$\frac{\lambda}{h}$ = Knudsen's number

For the transition region of flow between laminar and molecular flow, a relationship of the form

$$\dot{m} = \dot{m}_L + E\,\dot{m}_M \qquad \text{(Eq 3.11.3e)}$$

is used, where E is an experimentally determined constant with the values of approximately 0.9 for single gases and 0.66 for mixtures of gases.

If $E = 0.9$, then

$$\dot{m} = \dot{m}_L\left(1 + 6.1\,\frac{\lambda}{h}\right) \qquad \text{(Eq 3.11.3f)}$$

If $E = 0.66$, then

$$\dot{m} = \dot{m}_L\left(1 + 4.5\,\frac{\lambda}{h}\right) \qquad \text{(Eq 3.11.3g)}$$

The flow rates calculated for helium with laminar and partially molecular flows are presented in Reference 46-6.

3.11.4 Annuli And Annular Orifices

For steady, laminar flow through an annular path (Figure 3.11.4a) the volumetric flow rate is

$$q = \frac{\pi d_2 h^3}{12\mu_f l}\left[1 + 1.5\left(\frac{\epsilon}{h}\right)^3\right][p_1 - p_2] \qquad \text{(Eq 3.11.4a)}$$

where ϵ = eccentricity, in
h = clearance, $\frac{d_2 - d_1}{2}$, in
μ_f = absolute viscosity, $lb_f\ sec/in^2$
d_1 = minor annular diameter, in
d_2 = major annular diameter, in
l = passage length, in
q = flow rate, in^3/sec
p_1, p_2 = upstream and downstream pressures, respectively, lb_f/in^2

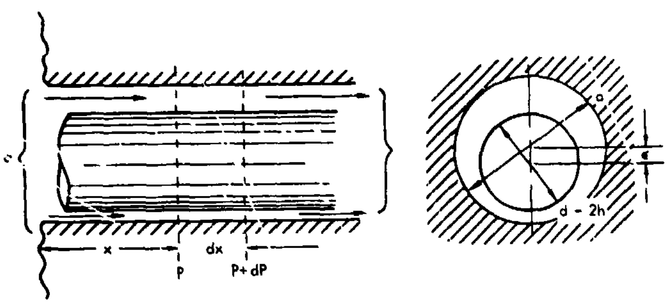

Figure 3.11.4a. Flow Through Annuli
(Reprinted with permission from "Fluid Power Control," Blackburn, Reethof, and Shearer, MIT Press, Cambridge, Massachusetts, 1960, p. 58)

Attempts have been made to correlate flows through annuli in terms of a friction factor plot similar to the Moody diagram. A chart based on a series of test data is shown in Reference 66-2. Friction factors are defined in terms of an effective diameter.

Where the inner wall length is very small, an *annular orifice* is formed. The characteristics of annular orifice flow change with geometry variation — sharp-edged or thick, concentric or tangent. A tangent orifice is an annular orifice with maximum eccentricity.

For laminar flow the flow coefficients for substitution in Equation (3.8.1k) for orifices are (Reference 26-84):

For a concentric orifice

(Eq 3.11.4b)

$$K_F = \left[\frac{64}{Re} + \frac{48Z}{Re} + K'\right]^{-1/2}$$

For a tangent, sharp-edged orifice

$$K_F \cong \left[\frac{128}{3Re} + 1\right]^{-1/2} \qquad \textbf{(Eq 3.11.4c)}$$

For a tangent, thick orifice

(Eq 3.11.4d)

$$K_F \cong \left[\frac{128}{3Re} + \frac{96}{5}\frac{Z}{Re} + K'\right]^{-1/2}$$

where Re = Reynolds number

$$Z = \frac{2l}{d_2 - d_1}$$

l = disc thickness

d_2 = major annular diameter

d_1 = minor annular diameter

K' = kinetic variable which may be found in Figure 3.11.4b.

For coefficients for turbulent flow with concentric and tangent orifices see Reference 26-84, pages 597-598.

A summary of concentric and tangent annular orifice coefficients is presented in Figures 3.11.4c and 3.11.4d. Data was compiled from tests on orifices with d_2/d_1 ratios of 0.95 to 0.996 and orifice length-to-width ratios from 0.118 to 33.3.

3.11.5 Tortuous Passages

The term "tortuous passages" refers to paths taken by a homogeneous fluid flowing through a porous medium. The most useful fluid-flow property is the permeability of a porous medium — permeability being a measure of the ease with which a fluid will flow through the medium. The higher the permeability, the higher the flow rate.

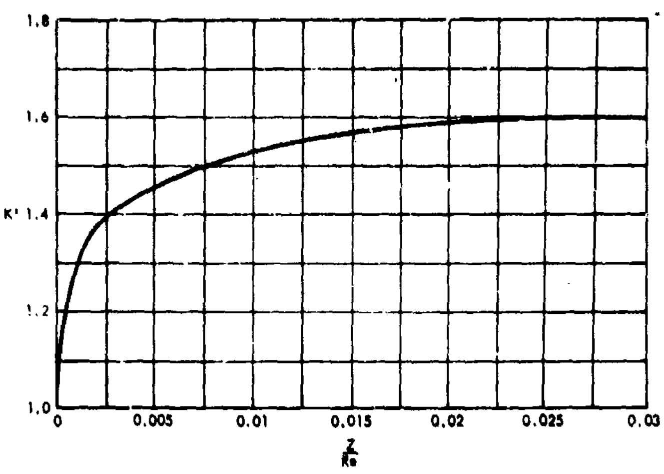

Figure 3.11.4b. Kinetic Term for Laminar Flow
(Reprinted with permission from "ASME Transactions — Journal of Basic Engineering," April 1957, vol. 79, no. 4, Bell and Bergelin, New York)

Permeation, the process whereby gases diffuse through solids, is considered the overall steady-state process, although the initial flow rate may be as much as ten times the final amount. If a pressure differential exists, the gas on the high pressure side is adsorbed and dissolves in the external surface layer of the solid. The adsorption may be physical, where Van der Waal's forces are prominent; or it may be chemical, where the solid surface provides binding sites for the atoms. The extent to which a gas is likely to be adsorbed can sometimes be determined from the critical temperature. Generally, the higher the critical temperature the greater the adsorption. Once adsorbed, the gas diffuses through the solid, forced by a concentration gradient. When it reaches the low pressure side of the solid, the gas transforms to an adsorbed state and is desorbed to the surroundings (Reference 46-6).

Laminar flow of a single fluid through a porous medium can be calculated from Darcy's Law of Permeability, thus

(Eq 3.11.5a)

$$Q = -\frac{k_p A}{\mu_f}\left(\frac{\partial P}{\partial s} + \rho\frac{g}{g_c}\sin\alpha\right)$$

For liquid flow, assumed to be incompressible

(Eq 3.11.5b)

$$Q = \frac{k_p A}{\mu_f}\left(\frac{P_1 - P_2}{L} - \rho\frac{g}{g_c}\sin\alpha\right)$$

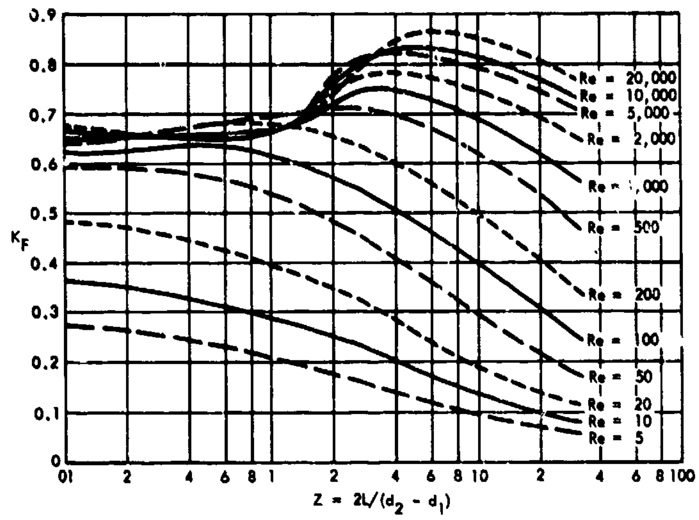

Figure 3.11.4c. Summary of Concentric Orifice Coefficients
(Adapted from "ASME Transactions – Journal of Basic Engineering," April 1957, vol. 79, no. 4, Bell and Bergelin, New York)

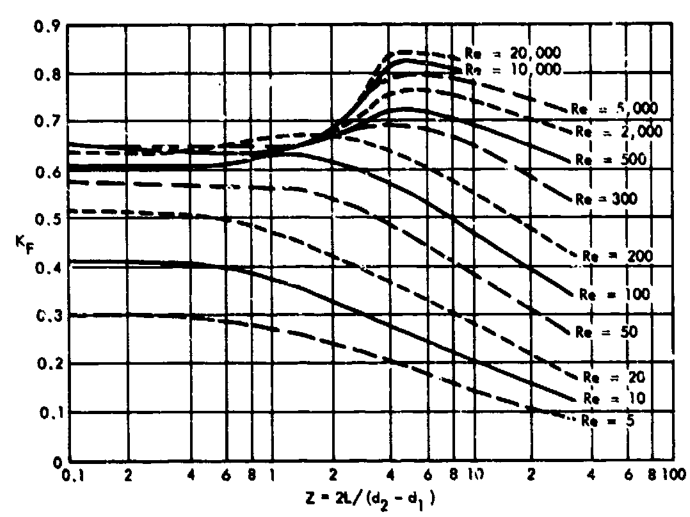

Figure 3.11.4d. Summary of Tangential Orifice Coefficients
(Adapted from "ASME Transactions – Journal of Basic Engineering," April 1957, vol. 79, no. 4, Bell and Bergelin, New York)

where Q = flow rate, ft^3/sec

k_p = permeability, ft^2

A = cross-sectional area, ft^2

μ_f = absolute viscosity, $lb_f\ sec/ft^2$

ρ = density, lb_m/ft^3

α = angle from horizontal, degrees

$\frac{\partial P}{\partial s}$ = pressure gradient, lb_f/ft^3

L = path length, ft

The permeation rate, P, is defined as

$$P = \frac{k_p}{\mu_f} \qquad \text{(Eq 3.11.5c)}$$

where k_p = permeability

μ_f = absolute viscosity

As velocity increases, losses due to fluid inertia become progressively more important. For horizontal flow, Equation (3.11.5a) rewritten and expanded to include inertia, becomes

(Eq 3.11.5d)

$$Q = A\,\frac{\partial P}{\partial x}\left[\frac{1}{\frac{\mu_f}{k_p} + \frac{\rho V}{g_c k_i}}\right]$$

where $\frac{1}{k_i}$ is a measure of the inertial resistance due to the tortuosity of the flow channels.

Both k_p and k_i are independent of fluid properties, and are characterized only by the porosity of the material. Values of $\frac{1}{k_p}$ and $\frac{1}{k_i}$ are presented in Figure 3.11.5.

At least four sets of units have been used for permeation rates:

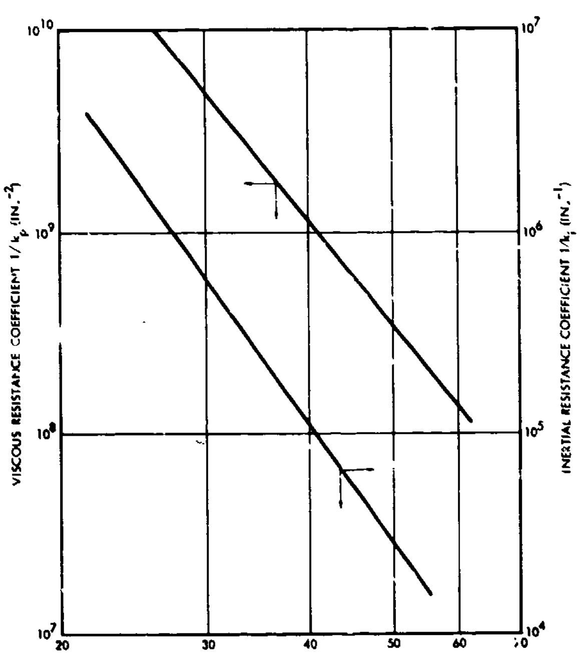

Figure 3.11.5. Viscous and Inertial Resistance Coefficients
(Adapted from "Journal of Applied Mechanics," March 1951, vol. 18, no. 1, Green and Duwez)

a) $\frac{cm^3 - mm}{sec - cm^2 - torr}$

b) $\frac{cm^3 - cm}{sec - cm^2 - atm}$

c) $\frac{cm^3 - mm}{sec - cm^2 - atm}$

d) $\frac{cm^3 - mm}{sec - cm^2 - cm\ Hg}$

Equation (3.11.5a) is valid only for molecular diffusion through polymers. If the diffusion process is atomic in nature, as with diatomic gases permeating metal, Equation (3.11.5a) is not valid.

Some generalizations may be made about gas permeation through metals and [illegible] rence 46-6).

Metals:

a) No rare [illegible] any metal

b) H[illegible] meation

[illegible] pecially Fe

[illegible] e square root of the [illegible]

Polymers:

a) All gases permeate polymers

b) H_2 rate is probably high

c) Permeation rates vary with the pressure

d) Permeation rate varies exponentially with temperature.

Table 3.11.5a lists a table of some permeability coefficients.

Table 3.11.5a. Permeation Rate, $P \times 10^7$ where P has units of $\frac{cm^3 - cm}{sec \cdot cm^2 \cdot atm}$

(Reference 459-1)

FILM	N_2	O_2	CO_2	H_2
Saran	0.0005	0.0026	0.011	—
PVC	0.0092	0.069	—	—
Kel F	0.025	0.0035	0.056	—
Teflon FEP	0.68	1.49	1.44	—
Mylar	0.0041	0.026	0.08	—
Butyl Rubber	0.05	0.34	4.16	—
Natural Rubber	6.1	17.7	99.6	23.7

The following table shows the permeability of various gases through rubber relative to hydrogen.

Table 3.11.5b. Relative Permeability of Gases Through Rubber

GAS	MOLECULAR WEIGHT	RELATIVE PERMEABILITY $H_2 = 1$
H_2	2	1.0
N_2	28	0.16
Air	29	0.22
A	39.9	0.26
O_2	32	0.45
He	4	0.65
CO_2	44	2.9
NH_3	17	8.0

3.12 CAVITATION

When an inline restriction (e.g., a valve, venturi, or orifice) is subjected to constant inlet pressure, but decreasing downstream pressure, the flow rate increases. However, a point is eventually reached where further decreases result in no increase in flow. Defined as *the critical flow*, this is the point where cavitation begins.

Cavitation may be defined as a two step phenomenon involving the formation of bubbles or cavities in a flowing stream with subsequent bubble collapse downstream. From Bernoulli's Equation, the pressure head decreases as the velocity head increases. As fluid flows through a restriction, the pressure at the vena contracta may decrease to a value equal to the vapor pressure of the liquid. The liquid will boil and cavities or vapor bubbles will form. As the fluid moves downstream the velocity decreases due to the increase in flow area (fluid filling the pipe), and the static pressure increases collapsing the bubbles entrapped in the liqiud. A pressure and velocity history is shown in Figure 3.12.

Sudden collapse of the bubbles downstream produces high dynamic pressure upon any adjacent material and such concurrent effects as noise, vibration, loss of power and efficiency. If cavitation continues at high frequencies, the material will be damaged. Several theories have been proposed to explain the cause of damage. One theory, probably the most popular, concludes that damage is caused by the high pressure shock waves produced by bubble collapse. The shock waves act as "hammer blows" on the surface of the material which, by fatigue, tear away minute pieces of material (Reference 141-1). A second theory is that the high pressure drives the fluid into the crystalline structure of metals, and at relief of pressure tears away pieces of the metal.

Cavitation damage may be substantially reduced or eliminated by a proper choice of materials, or by installing

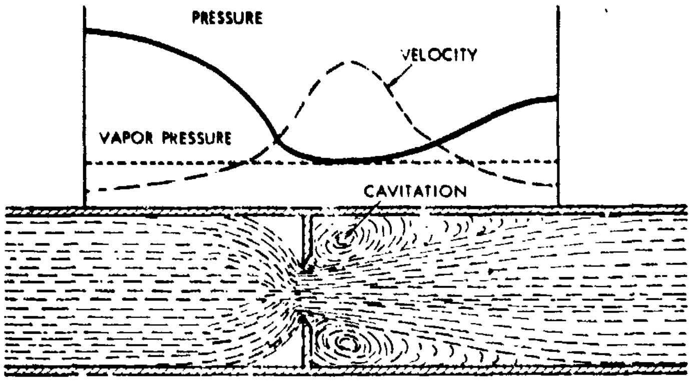

Figure 3.12. Pressure Gradient with Cavitation

valves in series such that the pressure drop across any one is less than that for cavitation to occur.

The point where cavitation may occur for any given component can be determined from a dimensionless factor, called the cavitation number or index of incipient cavitation, K_{cv}.

$$K_{cv} = \frac{p_1 - p_2}{p_1 - p_v} = \frac{\Delta p_{max}}{p_1 - p_v} \qquad \textbf{(Eq. 3.12)}$$

where

K_{cv} = index of cavitation
Δp_{max} = maximum pressure differential which may be taken across the valve before cavitation begins
p_1 = inlet pressure
p_2 = outlet pressure
p_v = vapor pressure of the liquid

The index K_{cv} is determined experimentally for each valve configuration and is strongly influenced by the design of the valve plug and outlet configuration. The outlet configuration determines the area of the vena contracta and, consequently, the static pressure at that point.

A perfect valve would have a K_{cv} equal to unity, and the downstream pressure could be reduced to the vapor pressure before cavitation would take place. In reality this is not the case. Results of tests (Reference V-33) on several types of control valves indicate a wide variation of K_{cv} values. For contour valve plugs installed in the flow-to-open direction in either globe or angle bodies, K_{cv} values are approximately 0.65; for plugs installed in the flow-to-close direction, K_{cv} values are approximately 0.20. The values indicate the desirability to have flow from the flow-to-open direction whenever possible. Furthermore, cavitation damage was small when pressure differentials were less than 400 psi. Damage to control valve plugs when installed in the flow-to-open direction almost always occurred to the plug and sometimes to the orifice. For flows in the flow-to-close direction, the reverse was generally true (Reference 74-2).

Treatment of cavitation in butterfly valves requires a slightly different technique than with control valves. In butterfly valves cavitation inception is a function of butterfly angle as well as pressure and velocity. Analysis of cavitation for steady flows is well presented in Reference 68-4, and is verified by experimental values.

A discussion of cavitating venturis as flow control devices is presented in Sub-Section 5.3, "Control Valves," and Sub-Section 6.2, "Valving Units."

If p_2 is less than p_v, K_{cv} is no longer applicable because the fluid will not return to the liquid phase. Cavitation will not take place, but some of the liquid will flash, forming a liquid-vapor mixture. If the downstream pressure is sufficiently low, all the liquid will vaporize. These conditions are presented in Sub-Section 3.13, "Flashing Liquids."

A theoretical treatment of cavitation is presented in Reference 141-1.

3.13 FLASHING LIQUIDS*

In cavitation, the pressure downstream of the vena contracta formed by an in-line restriction is greater than the vapor pressure of the liquid. Vapor formed in the vena contracta condenses downstream. If pressure downstream is equal to or less than the vapor pressure, a portion of the liquid will flash to a vapor which will not return to the liquid phase, forming a liquid-vapor mixture downstream. A similar effect will also take place in a liqud near its boiling point (e.g., a cryogenic fluid) as it passes through an in-line restriction.

To control the flow of such a liquid by valves, an accurate and economical determination of valve size may be made by using the so-called lower-density technique, based on the fact that after flashing, the density of the vapor-liquid mixture is lower. Valve sizing is determined by selection of an appropriate C_v.

If the liquid flashes as it passes through the valve, a relationship must be made to the specific gravity term due to the decrease in density. Based on steady-flow, the specific gravity of a flashed vapor-liquid mixture may be expressed as

(Eq. 3.13a)

$$S_{lg} = \frac{S_l}{1 + 62.4\, S_l\, v\, \dfrac{(h_1 - h_2)}{q_{lg}}}$$

where

S_{lg} = specific gravity of the flashed mixture
S_l = specific gravity of the liquid before flashing
v = specific volume of vapor formed (ft^3/lb_m)
h_1 = enthalpy of liquid before flashing (Btu/lb_m)
h_2 = enthalpy of liquid after flashing (Btu/lb_m)
q_{lg} = heat of vaporization (Btu/lb_m)

**Adapted from References 27-37 and 52-14*

Values for substitution into the above equation may be found in thermodynamic charts and tables for the liquid considered. Assuming a saturated liquid with constant inlet pressure, and solving for the specific gravity of the liquid-vapor mixture for different downstream pressures, values of S_{lg} may be plotted against outlet pressure, as shown in Figure 3.13a.

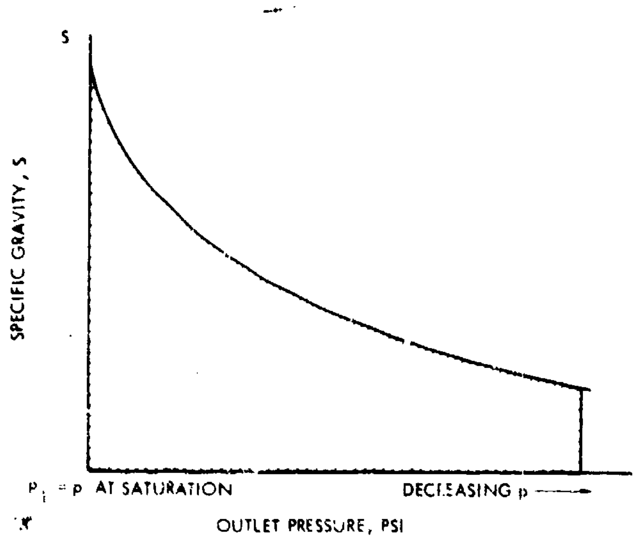

Figure 3.13a. Change in Specific Gravity with Pressure Drop
(Reprinted with permission from "ISA Journal," March 1957, J. G. Ziegler, Instrument Society of America)

By multiplying the standard C_v equation by S_{lg}, the equation becomes

$$S_{lg}Q = C_v\sqrt{\Delta p S_{lg}} \qquad \textbf{(Eq. 3.13b)}$$

Values of the quantity ($\Delta p S_{lg}$) may be obtained by integrating the area under the curve plotted above. For example, considering water at 50 psia and 281 F (boiling point at that pressure) flowing through a valve with an over-all pressure drop of 15 psi, the value of $\Delta p S_{lg}$ is equal to 3.1 psi (Figure 3.13b). In other words, although there is a pressure drop of 15 psi across the valve, the amount of flow through the valve is equivalent to that amount of water at 60 F with a 3.1 psi pressure drop. The value of C_v required may be determined by dividing the product of the specific gravity of water at 281 F and the desired flow rate, Q, by $\sqrt{\Delta p S_{lg}}$.

By successive integration of the areas between various saturation conditions and any outlet pressure, products of $\Delta p S_{lg}$ may be plotted against stated pressure drops. A typical graph is shown in Figure 3.13c for saturated water. Graphs for some additional fluids are given in Reference 27-37.

If the upstream pressure is greater than the pressure necessary for flashing, but the pressure drop is still sufficient to produce flashing, an additional ($\Delta p S_{lg}$) product must be

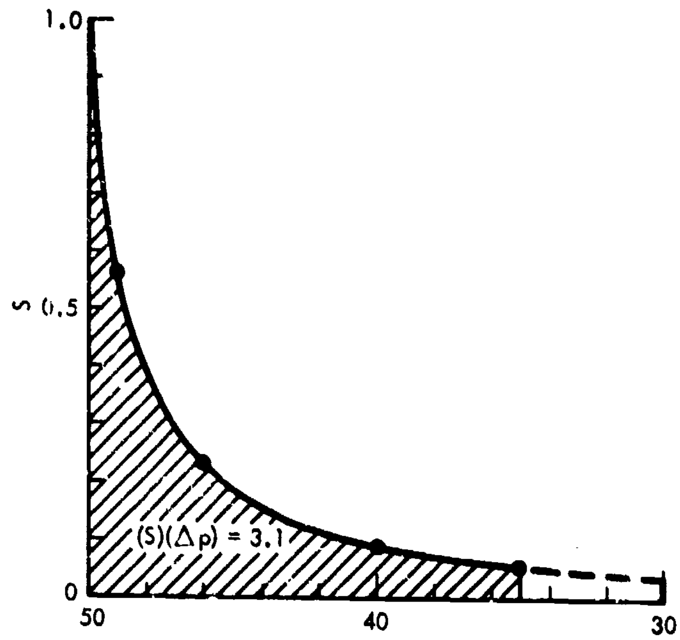

Figure 3.13b. Changes in Specific Gravity with Pressure Drop (Water)
(Reprinted with permission from "ISA Journal," March 1957, J. G. Ziegler, Instrument Society of America)

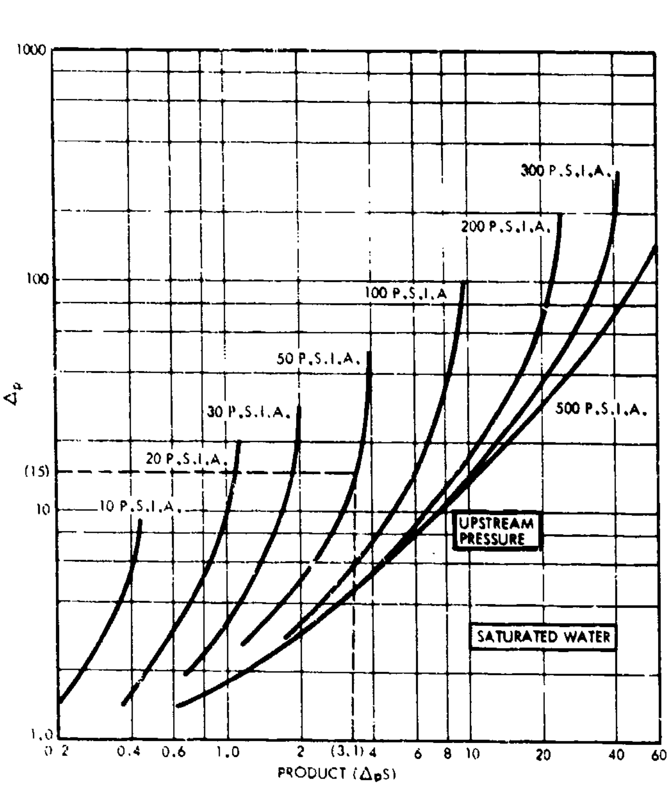

Figure 3.13c. Product ($\Delta p S_{lg}$) for Water
(Reprinted with permission from "Control Engineering," February 1961, vol. 8, no. 2, A. Hanssen, Copyright 1961 by McGraw-Hill Publishing Company, Inc., all rights reserved)

added to that of liquid at saturation determined above. The effect of the additional term is to move the original graph to the right, as shown in Figure 3.13d. The total area under the curve has increased to 8.1 psi, an increase of 5 psi from the original. To incorporate this change, the original C_v equation is modified to the expression

$$S_{lg}Q = C_v\sqrt{S_l\,(p_l - p_{sat})} + C_v\sqrt{\Delta p S_{lg}} \qquad \textbf{(Eq. 3.13c)}$$

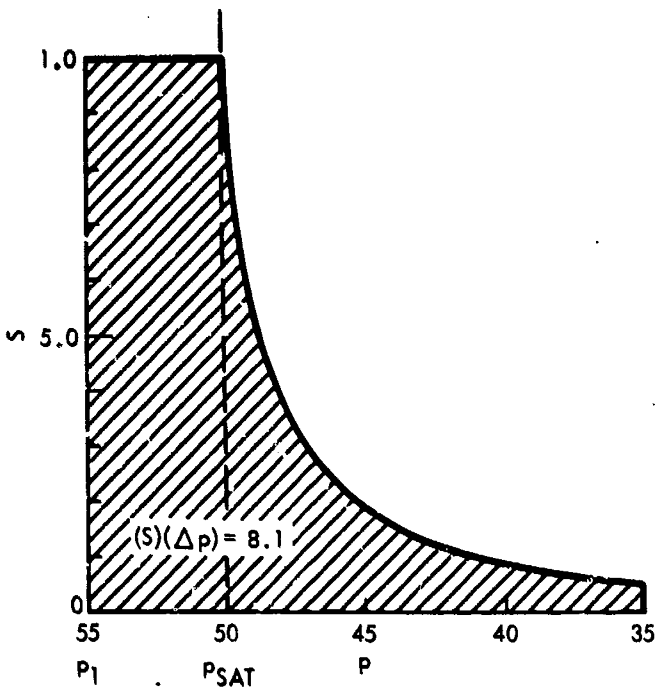

Figure 3.13d. Change in Specific Gravity with Higher Upstream Pressure
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3.14 DYNAMIC FORCES

When a fluid's velocity is compelled to change magnitude, direction, or both, a force is exerted against the object which induces the change. The magnitude of the force can be found from Newton's Second Law which may be expressed mathematically as

$$\vec{F} = \frac{d}{dt}(m\,\vec{V}) \qquad \textbf{(Eq. 3.14a)}$$

where

$\vec{F}$ = force exerted by the fluid stream in any given direction
m = mass of fluid flowing in a given time
$\vec{V}$ = velocity of the fluid

This equation states that a dynamic force is exerted by a fluid equal to the time rate of change of the momentum of the fluid.

As an example, consider a curved conduit as shown in Figure 3.14 enclosed by a control volume (imaginary volume of fluid held static at a given time).

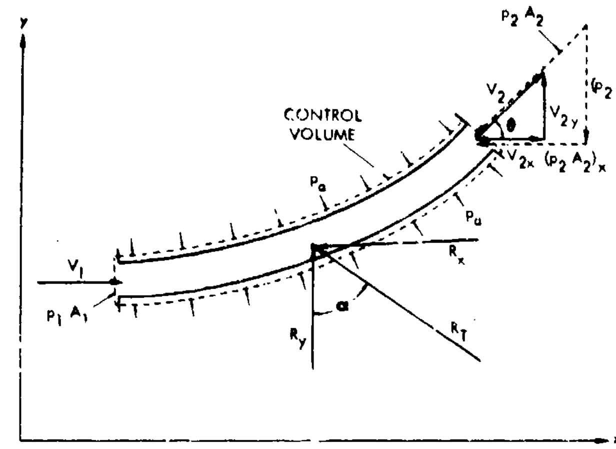

Figure 3.14. Reaction Forces Due to Fluid Flow

For an incompressible fluid and steady-flow process, the mass rate of flow into the control volume equals the mass rate of flow out ($\dot{m}_1 = \dot{m}_2$), and Newton's equation may be expressed as

$$F_x = \frac{1}{g_c}\left(\dot{m}_1 V_{1_x} - \dot{m}_2 V_{2_x}\right) \qquad \textbf{(Eq. 3.14b)}$$

$$F_y = \frac{1}{g_c}\left(\dot{m}_1 V_{1_y} - \dot{m}_2 V_{2_y}\right) \qquad \textbf{(Eq. 3.14c)}$$

Since $\dot{m}_1 = \dot{m}_2 = \rho VA = \rho Q$, the equations for the *dynamic forces* may be rewritten as

$$F_x = \frac{\rho}{g_c} Q\,(V_1 - V_2\cos\theta) \qquad \textbf{(Eq. 3.14d)}$$

$$F_y = -\frac{\rho}{g_c} Q\,(V_2\sin\theta) \qquad \textbf{(Eq. 3.14e)}$$

where

F_x, F_y = force exerted by the fluid in the x-direction and y-direction, respectively, ft

V_1, V_2 = fluid velocities into and out of the control volume, respectively, ft/sec

θ = angle of velocity vector from the horizontal axis

ρ = density, lb_m/ft^3

Q = volumetric flow rate, ft^3/sec

$g_c = 32.2 \frac{lb_m}{lb_f} ft/sec^2$

The total *reaction force* exerted by the conduit is the sum of the dynamic forces exerted by the flow stream, plus the static forces (e.g., pressure) over the control volume (Figure 3.14). The reaction forces become

(Eq. 3.14f)

$$R_x = \frac{\rho}{g_c} Q\,(V_1 - V_2\cos\theta) + P_1A_1 - P_2A_2\cos\theta$$

and

$$R_y = -\frac{\rho}{g_c} Q\,(V_2\sin\theta) + P_2A_2\sin\theta \qquad \text{(Eq. 3.14g)}$$

The total reaction force is

$$R_T = \sqrt{(R_x)^2 + (R_y)^2} \qquad \text{(Eq. 3.14h)}$$

The angle between R_x and R_y is

$$\alpha = \tan^{-1}\frac{R_x}{R_y} \qquad \text{(Eq. 3.14i)}$$

The change in static pressure from points 1 to 2 may be determined from Bernoulli's Equation (see Sub-Topic 3.7.3).

3.15 SIMILITUDE AND DIMENSIONAL ANALYSIS

3.15.1 Scope

This Sub-Section presents descriptions of the principles of similitude and dimensional analysis as applied to fluid flow and heat transfer problems. Under the subject of Similitude geometric similarity, dynamic similarity, and thermal similarity are discussed. Under the subject of Dimensional Analysis, Lord Rayleigh's Method and the Buckingham Pi Theorem are considered.

3.15.2 Similitude

3.15.2.1 DEFINITION. Similitude is the utilization of dimensionless parameter ratios in designing models and relating model performance to full scale prototype performance.

3.15.2.2 GEOMETRIC SIMILARITY. Geometric similarity necessitates a similarity of configuration between the model and the prototype. The fundamental dimension is length, where the length ratio, L_r, establishes the scale of the model.

$$L_r = L_m/L_p \qquad \text{(Eq. 3.15.2.2)}$$

where L_r = length ratio

L_m = any length on model

L_p = corresponding length on prototype

Geometric similarity between the model and the prototype is achieved when all dimensions on the model are related to the corresponding dimensions on the prototype by the appropriate length ratio, L_r.

An example of geometric similarity is a flow tube 20 feet long and 5 inches inside diameter (prototype) to be represented by a model with an inside diameter of 1/2 inch. The scale ratio, L_r, is 1/10 (fixed by the diameter ratio 1/2 inch to 5 inches), therefore, for geometric similarity the model must be 20 feet divided by 10, or 2 feet long.

3.15.2.3 DYNAMIC SIMILARITY. Dynamic similarity requires a similarity of shapes, paths, times, and forces. The fundamental unit used in dynamic similarity is force, which is related to the length, density, viscosity, and acceleration ratios as follows:

(Eq. 3.15.2.3a)

$$F_r \text{ (terrestrial gravitation governing)} = \rho_r L_r^3 g_r$$

(Eq. 3.15.2.3b)

$$F_r \text{ (viscous fluid forces governing)} = \rho_r \nu_r^2$$

where F_r = ratio of any model force to the corresponding prototype force.

Useful dimensionless ratios used in deriving or converting Equations (3.15.2.3a and b) are as follows:

Time ratio = $t_r = t_m/t_p$

Velocity ratio = $v_r = v_m/v_p = L_r/t_r$

Acceleration ratio = $a_r = a_m/a_p = L_r/t_r^2$

Terrestrial gravitation ratio = $g_r = g_m/g_p$

Mass ratio = $m_r = m_m/m_p = \rho_r L_r^3$

Density ratio = $\rho_r = \rho_m/\rho_p$

Kinematic viscosity ratio = $\nu_r = \nu_m/\nu_p = L_r^2/t_r$

Absolute viscosity ratio = $\mu_r = \mu_m/\mu_p = \rho_r L_r^2/t_r$

The derivation of Equation (3.15.2.3a) for systems where *terrestrial gravitation* is the governing physical force (e.g., gravity flow from a storage tank through an orifice) is presented below.

According to Newton's Second Law of Motion, $F = \frac{1}{g_c} ma$, or $F_r = m_r a_r$.

Since terrestrial gravitation, g, is the only physical force acting, then

$$F_r = m_r g_r \qquad \textbf{(Eq. 3.15.2.3c)}$$

Combining the Mass ratio with Equation (3.15.2.3c)

$$F_r = \rho_r L_r^3 g_r \qquad \textbf{(Eq. 3.15.2.3d)}$$

The ratios of other parameters where terrestrial gravitation is the governing physical force acting are listed in Table 3.15a with their relation to g_r, L_r, and ρ_r.

The derivation of Equation (3.15.2.3b) for situations where *viscous forces* are the governing physical forces (e.g., fluid flow through tubing) is presented below.

Viscous force due to fluid shear on area, A, is

$$F = \mu A \left(\frac{dv}{dx}\right) \qquad \textbf{(Eq. 3.15.2.3e)}$$

where μ = absolute viscosity of fluid

$\frac{dv}{dx}$ = velocity profile

(Eq. 3.15.2.3f)

and since $\mu_r = \nu_r \rho_r$ and $\nu_r = L_r^2/t_r$, $F_r = \rho_r \nu_r^2$

$$F_r = \mu_r A_r v_r / x_r$$

$$F_r = \mu_r L_r^2 (L_r/t_r)/L_r$$

$$F_r = \mu_r L_r^2/t_r$$

The ratios of other parameters where viscous fluid forces are the governing physical forces are listed in Table 3.15b with their relation to ν_r, L_r, and ρ_r.

An example of a dynamically similar model is the geometrically similar model discussed under Sub-Topic 3.15.2.2, "Geometric Similarity," with the added factors of fluid velocity, V_p, density, ρ_p, and absolute viscosity, μ_p, for the prototype. If the fluid in the model has a viscosity, μ_m, and density, ρ_m, dynamic similarity will be achieved if the model fluid velocity, V_m, is $V_m = V_p (\nu_r/L_r)$ (see Table 3.15b). The same result will be obtained by applying Reynolds' Law, which states that in a geometrically similar model where fluid friction is the only physical force acting, dynamic similarity will be achieved if the Reynolds numbers in the model and prototype have the same value.

3.15.2.4 THERMAL SIMILARITY. Thermal similarity is achieved in a model by maintaining constant dimensionless ratios, determined by analysis and experiment, between the model and prototype. These dimensionless ratios may be determined by Lord Rayleigh's method or the Buckingham Pi Theorem discussed under Sub-Topic 3.15.3, Dimensional Analysis.

An example of the application of thermal similarity is the problem of obtaining heat transfer data for hydrogen peroxide flowing (turbulent) in straight tubing heated by still air. For economy and safety reasons water has been substituted for the hydrogen peroxide for conducting these tests. In Reference 134-1, page 219, for Prandtl numbers from 0.7 to 120 and Reynolds numbers from 10,000 to 120,000, a dimensionless equation has been developed which applies to the above problem, having the form

$$Nu = 0.023\,(Re)^{0.8}\,(Pr)^{0.4}$$

where Nu = Nusselt number = hD/k

Re = Reynolds number = $VD\rho/\mu_m$

Pr = Prandtl number = $c_p\mu_m/k$

Using the subscripts (m) and (p) for model and prototype, respectively, and subscript (r) for ratio of model to prototype, the prediction equation can be written

$$\frac{(Nu)_m}{(Nu)_p} = \frac{(Re)_m^{0.8}\,(Pr)_m^{0.4}}{(Re)_p^{0.8}\,(Pr)_p^{0.4}}$$

$$(Nu)_r = (Re)_r^{0.8}\,(Pr)_r^{0.4} = h_r\frac{D_r}{k_r} = \frac{h_m}{h_p}\frac{D_r}{k_r}$$

$$h_p = \frac{h_m D_r}{k_r}(Re)_r^{-0.8}\,(Pr)_r^{0.4}$$

The value of h_m can be determined from test, and D_r and $(Re)_r$ can be chosen arbitrarily as 1. With the values of $(Pr)_r$ and k_r known, the heat transfer coefficient, h_p, can be computed. By changing variables such as flow velocity, temperature, etc., in the model, the effects of the change on h_p can be studied.

3.15.3 Dimensional Analysis

3.15.3.1 DEFINITION. Dimensional analysis is a tool used to develop the interrelationship between the significant variables affecting a physical event. The key to developing these interrelationships is the application of dimensional homogeneity to the problem under consideration.

3.15.3.2 APPLICATIONS. Dimensional analysis has numerous applications in engineering analysis, some of which are noted below.

a) It provides a guide for planning experimental programs, and for analyzing and correlating the observed data.

b) It simplifies the conversion of variables from metric to English units, or the reverse by the use of dimensionless ratios.

c) Prediction equations can be developed and the physical constants determined experimentally. These equations can then be used for solving *similar* problems where the sizes, materials, forces and time dependent variables are different.

d) It trains a person to be cognizant of the dimensional homogeneity of a given empirical equation. Ordinarily, equations which are not dimensionally homogeneous

Table 3.15a. Dynamic Similarity Relationships for Various Quantities When Terrestrial Gravitation is the Only Acting Physical Force

(From "Dimensional Analysis and the Principle of Similitude," K. C. Reynolds, University of Southern California, Los Angeles)

QUANTITY	DIMENSIONLESS RELATIONSHIP	
Ratio = $\frac{\text{model}}{\text{prototype}}$		when $g_p = g_m$, $\rho_p = \rho_m$
Angular Velocities, ω_r	$\sqrt{g_r/L_r}$	$\sqrt{1/L_r}$
Accelerations, a_r	g_r	1
Times, t_r	$\sqrt{L_r/g_r}$	$\sqrt{L_r}$
Velocities, V_r	$\sqrt{L_r/g_r}$	$\sqrt{L_r}$
Linear dimensions, velocity heads, friction heads, L_r	L_r	L_r
Pressures, p_r	$L_r\rho_r g_r$	L_r
Areas, A_r	L_r^2	L_r^2
Volumetric Flow Rates, Q_r	$L_r^{2.5}\sqrt{g_r}$	$L_r^{2.5}$
Volumes, V_r	L_r^3	L_r^3
Masses, m_r	$L_r^3\rho_r$	L_r^3
Forces, F_r	$L_r^3\rho_r g_r$	L_r^3
Power, P_r	$L_r^{3.5}\rho_r g_r^{1.5}$	$L_r^{3.5}$
Impulses, momentums, I_r	$L_r^{3.5}\rho_r g_r^{0.5}$	$L_r^{3.5}$
Work, energy, E_r	$L_r^4\rho_r g_r$	L_r^4

Table 3.15b. Dynamic Similarity Relationships for Various Quantities When Viscous Fluid Force is the Only Acting Physical Force

(From "Dimensional Analysis and the Principle of Similitude," K. C. Reynolds, University of Southern California, Los Angeles)

QUANTITY	DIMENSIONLESS RELATIONSHIP	
Ratio = $\frac{\text{model}}{\text{prototype}}$		When $\rho_p = \rho_m$ and $\nu_p = \nu_m$
Accelerations, a_r	ν_r^2/L_r^3	$1/L_r^3$
Angular Velocities, ω_r	ν_r/L_r^2	$1/L_r^2$
Pressures, p_r	$\rho_r\nu_r^2/L_r^2$	$1/L_r^2$
Power, P_r	$\rho_r\nu_r^3/L_r$	$1/L_r$
Velocities, V_r	ν_r/L_r	$1/L_r$
Forces, F_r	$\rho_r\nu_r^2$	1
Lengths, L_r	L_r	L_r
Volumetric Flow Rates, Q_r	$\nu_r L_r$	L_r
Work, energy, W_r	$\rho_r\nu_r^2 L_r$	L_r
Areas, A_r	L_r^2	L_r
Times, t_r	L_r^2/ν_r	L_r^2

(balanced units) are developed empirically and give dependable results only in one system of units with a narrow range of variables.

3.15.3.3 LORD RAYLEIGH'S METHOD. Lord Rayleigh's Method is suitable for determining the interrelationships of the variables affecting a physical event for a limited number of dimensionless ratios. In more complex problems the Buckingham Pi Theorem is better suited. Lord Rayleigh's Method depends on the reasonable assumption that an equation describing a physical event must be dimensionally homogeneous to be correct, and consists of the following steps:

1. List all physical quantities (variables) which influence the performance of a physical event such as velocity, density, viscosity, diameter, etc. In addition, tabulate the fundamental dimensions (M, L, t, F, T) for each quantity with its proper exponent (e.g., Q, volumetric flow rate, L^3/t).

2. Write an equation of the variable to be studied in terms of the other variables involved, such as:

$$Z = K(Z_1^a Z_2^b Z_3^c Z_4^d Z_5^e) \qquad \textbf{(Eq 3.15.3.3)}$$

where Z = dependent variable

K = constant

Z_1, Z_2, etc. = independent variables

3. Substitute into Equation (3.15.3.3) the fundamental dimensions of the variables.

4. Select one fundamental dimension such as mass, M, and write an expression equating the exponents of all M terms. Repeat this for the remaining fundamental dimensions, and then solve the equations (the number of equations = number of fundamental dimensions) to determine the values of the exponents. Often the number of unknown exponents exceeds the number of equations, and this procedure results merely in the reduction of the number of unknown exponents by a factor equal to the number of fundamental dimensions.

5. Substitute into Equation (3.15.3.3) the values of the exponents determined in Step (4).

6. Determine the values of the unknown exponents and the constant K by experiment. The unknown exponents and the constant K are termed *experimental constants.*

Application of Lord Rayleigh's Method to a common engineering problem: It is desired to determine experimentally the output power of a propellant feed pump design in terms of the dependent variables. What dimensionless ratios should be employed in obtaining the experimental data and how are they related to the pump power output?

Step 1. (Using the fundamental dimensions M,L,t.)

PHYSICAL QUANTITY		FUNDAMENTAL DIMENSIONS
P	power output	ML^2/t^3
Q	volumetric flow rate	L^3/t
N	angular velocity of rotor	$1/t$
H	pressure head rise	L
ρ	propellant density	M/L^3
μ_m	propellant viscosity	M/Lt
D	impeller diameter	L
W	impeller blade width	L

Step 2.

$$P = K(Q^a N^b D^c \rho^d \mu_m^e H^f W^g)$$

Step 3.

$$\frac{ML^2}{t^3} = \left(\frac{L^3}{t}\right)^a \left(\frac{1}{t}\right)^b (L)^c \left(\frac{M}{L^3}\right)^d \left(\frac{M}{Lt}\right)^e (L)^f (L)^g$$

Step 4.

M: $1 = d + e$

L: $2 = 3a + c - 3d - e + f + g$

t: $-3 = -a - b - e$

Solving and collecting terms:

$$e = 1 - d$$

$$a = 3 - b - e = 3 - b - 1 + d = 2 - b + d$$

$$c = 2 - 3a + 3d + e - f - g$$

$$c = 2 - 6 + 3b - 3d + 3d + 1 - d - f - g$$

$$c = -3 + 3b - d - f - g$$

Step 5.

$$P = KQ^{2-b+d} N^b D^{-3+3b-d-f-g} \rho^d \mu_m^{1-d} H^f W^g$$

$$P = \frac{KQ^2\mu_m}{D^3}\left(\frac{ND^3}{Q}\right)^b \left(\frac{Q\rho}{D\mu_m}\right)^d \left(\frac{H}{D}\right)^f \left(\frac{W}{D}\right)^g$$

Step 6.

The experimental constants are K, b, d, f, and g.

3.15.3.4 THE BUCKINGHAM PI THEOREM. The Pi Theorem serves the same purpose as Lord Rayleigh's Method for deriving equations expressing one variable in terms of its dependent variables. The Pi Theorem is preferred where it is apparent that the number of dimensionless ratios will exceed two. The steps involved in using the Pi Theorem are described as follows:

1. List all physical quantities (variables) which influence the performance of a physical event. In addition, tabulate the fundamental dimensions (M, L, t, F, T, H) for each variable with its proper exponent.

2. Determine n, the number of physical quantities involved, k the number of fundamental dimensions, and n′ the number of quantities with like fundamental dimensions such as length (L) and diameter (L).

3. Determine the number of r terms; terms which are ratios of the n′ terms. The number of r terms (r) is one less than the number of n′ terms. Select one n′ term, to set up r ratios $r_1, r_2, \ldots r_{n'-1}$ by dividing each remaining n′ term by the selected term.

4. Determine the number of dimensionless π terms = n − k − r.

5. Write the general equation

$$\pi_1 = K (\pi_2)^a (\pi_3)^b (\ldots \pi_{n-k-r})^c (r_1)^d (r_2)^e (\ldots r_{n'-1})^f$$

6. Construct the π terms as follows:

a) Each π term shall consist of k + 1 number of physical quantities, not to exceed 5.

b) All fundamental dimensions are to be included, plus a vital linear dimension, a kinematic or dynamic term, and a fluid property term. One physical quantity, Q, will have an exponent of 1. All other exponents will be unknown, i.e., a, b, c, etc.

c) Arbitrarily set the exponent of one π term such as π_1 equal to 1.

7. Solve for the unknown exponents of the physical quantities in π_1 as follows:

$$\pi_1 = Q_1^a Q_2^b Q_3^c Q_4$$

Substitute into each term above its fundamental dimensions and equate the exponents of like fundamental dimensions. Solve the equations simultaneously to determine the value of the exponents. Repeat this procedure for the remainder of the π terms.

8. Substitute into the general equation in Step (5) the values of the 3 terms from Step (3) and the π terms from Step (7).

Application of the Pi Theorem to the problem solved by Lord Rayleigh's Method. Using the step-by-step procedure detailed above, application is presented in the following paragraphs. It is interesting to note that the solutions by the two methods are similar, although not identical. In fact, two persons using the same procedure may arrive at different but sound solutions.

Step 1.

PHYSICAL QUANTITY		FUNDAMENTAL DIMENSIONS
P	power output	ML^2/t^3
Q	volumetric flow rate	L^3/t
N	angular velocity of rotor	$1/t$
H	pressure head use	L
ρ	propellant density	M/L^3
μ_m	propellant viscosity	M/Lt
D	impeller diameter	L
W	impeller blade width	L

Step 2.

The number of physical quantities, n, is 8.

The number of fundamental dimensions, k, is 3 (M, L, t). The number of physical quantities with like fundamental dimensions, n′, is 3 (H, D, and W are all length terms).

Step 3.

The number of r terms is r = n′ − 1 = 2. Selecting D, set up the r terms: $r_1 = H/D$, $r_2 = W/D$.

Step 4.

The number of π terms is 3. (n−k−r = 8−3−2 = 3)

Step 5.

$$\pi_1 = f(\pi_2, \pi_3, r_1, r_2)$$

Step 6.

$$\pi_1 = H^a P^b N^c \rho$$

$$\pi_2 = W^a Q^b \rho^c \mu_m$$

$$\pi_3 = Q^a H^b \rho^c N$$

Step 7.

$$\pi_1 = H^a P^b N^c \rho$$

$$M^0 L^0 t^0 = L^a \left(\frac{M^b L^{2b}}{t^{3b}}\right) \left(\frac{1}{t^c}\right) \left(\frac{M}{L^3}\right)$$

M: $0 = b + 1;\ b = -1$

L: $0 = a + 2b - 3 = a - 2 - 3;\ a = 5$

t: $0 = -3b - c = 3 - c;\ c = 3$

$$\pi_1 = \frac{H^5 N^3 \rho}{P}$$

$$\pi_2 = W^a Q^b \rho^c \mu_m$$

$$M^0 L^0 t^0 = L^a \left(\frac{L^{3b}}{t^b}\right) \left(\frac{M^c}{L^{3c}}\right) \left(\frac{M}{Lt}\right)$$

M: $0 = c + 1;\ c = -1$

L: $0 = a + 3b - 3c - 1 = a + 3b + 2;\ a = 1$

t: $0 = -b - 1;\ b = -1$

$$\pi_2 = \frac{W \mu_m}{Q \rho}$$

$$\pi_3 = Q^a H^b \rho^c N$$

$$M^0 L^0 t^0 = \frac{L^{3a}}{t^a} L^b \frac{M^c}{L^{3c}} \frac{1}{t}$$

M: $0 = c$

t: $0 = -a - 1; a = -1$

L: $0 = 3a + b; b = 3$

$$\pi_3 = \frac{H^3N}{Q}$$

Step 8.

$$P = KH^5N^3\rho\left(\frac{Q\rho}{W\mu_m}\right)^e\left(\frac{H^3N}{Q}\right)^f\left(\frac{H}{D}\right)^g\left(\frac{W}{D}\right)^i$$

There are five experimental constants to be determined from tests: K, e, f, g, and i.

3.15.4 Dimensionless Numbers

3.15.4.1 GENERAL. Applications of dimensional analysis to engineering problems has resulted in the repeated appearance of certain useful dimensionless ratios which are worthy of noting. In Table 3.15.4.1 dimensionless quantities useful in fluid flow and heat transfer studies are tabulated, with their formulas, relationships, and applications.

3.16 ELECTROHYDRAULIC ANALOGIES**

3.16.1 Summary of Analogous Parameters

The corresponding electrical and hydraulic parameters are presented in Table 3.16.1.

**Adapted from Reference 6-204.*

Table 3.15.4.1. Dimensionless Quantities

NAME	SYMBOL	FORMULA	RELATIONSHIP	APPLICATION
Reynolds number	Re	$= \rho L_c V/\mu_m$	inertial forces/ viscous forces	friction factor, type of flow, boundary layer
Froude number	Fr	$= V^2/Lg$	inertial forces/gravity forces	stream flow through an orifice
Prandtl number	Pr	$= \mu_m c_p/k$	viscosity/heat conduction	heat transfer
Mach number	M	$= V/V_s$	stream velocity/sonic velocity	compressible flow and flow through nozzles
Cavitation number	K_{cv}	$= \frac{(p_f - p_v)}{\rho V^2/2}$	pressure head/velocity head	cavitation
Weber number	W	$= V^2L\rho/\sigma$	inertial forces/cohesive forces	liquid flow over flat surfaces
Strouhal number	S	$= fd/V$	pulsating frequency/velocity profile	pulsating flow
Nusselt number	Nu	$= hL/k$	$Nu = f(Re, Pr)$	heat transfer across an interface

Table 3.16.1. Electrical and Hydraulic Parameters

(From Hydraulics and Pneumatics, R. E. Raymond, March 1961, Copyright 1961 by The Industrial Publishing Corporation)

ELECTRICAL PARAMETERS					HYDRAULIC PARAMETERS				
TERM	SYMBOL	UNITS	DIFFERENTIAL	FREQUENCY	TERM	SYMBOL	UNITS	DIFFERENTIAL	FREQUENCY
Voltage	E	Volts	$E = iR$	$E = iR$	Pressure drop	Δp	psid	$\Delta p = R_h q$	$\Delta p = R_h q$
Current	i	Amperes	$i = \frac{E}{R}$	$i = \frac{E}{R}$	Flow rate	q	in³/sec	$q = \frac{\Delta p}{R_h}$	$q = \frac{\Delta p}{R_h}$
Resistance	R	Ohms	$R = \frac{E}{i}$	$R = \frac{E}{i}$	Hydraulic resistance	R_h	psi sec/in⁵	$R_h = \frac{\Delta p}{q}$	$R_h = \frac{\Delta p}{q}$
Inductance	L	Henrys	$L = \frac{E}{\frac{di}{dt}}$	$(j\omega L) = \frac{E}{i}$	Hydraulic inductance	L_h	psi sec²/in⁵	$L_h = \frac{\Delta p}{\frac{dq}{dt}}$	$(j\omega L_h) = \frac{\Delta p}{q}$
Capacitance	C	Farads	$C = \frac{i}{\frac{dE}{dt}}$	$\frac{(-j)}{(\omega C)} = \frac{E}{i}$	Hydraulic capacitance	C_h	in⁵/psi	$C_h = \frac{\Delta q}{\frac{dp}{dt}}$	$\frac{(-j)}{(\omega C_h)} = \frac{p}{q}$

3.16.2 Calculation of Hydraulic Parameters

Resistance, R_h, for laminar flow from Poiseuille's Law is stated

$$R_h = \frac{\Delta p}{q} = \frac{8 \pi \mu_f l}{a^2} \quad \text{(Eq. 3.16.2a)}$$

For turbulent flow, with the fluid velocity assumed constant

$$R_h = \frac{\Delta p}{q} = \frac{f(L/D)\, w\, V}{(1728)\, a2g} \quad \text{(Eq 3.16.2b)}$$

Inductance, L_h, can be determined experimentally by measuring fluid acceleration in a line when a given pressure differential is applied to the ends. The inductance, or sometimes called inertia, is stated

$$L_h = \frac{5\, w\, l}{qa} \times 10^{-5} \quad \text{(Eq 3.16.2c)}$$

Capacitance, C_h, can be measured by determining the weight of fluid that must be introduced slowly into the line to produce a unit pressure rise. The capacitance for a spring accumulator is stated

$$C_h = \frac{a^2}{k} \quad \text{(Eq 3.16.2d)}$$

for liquid accumulator

$$C_h = \frac{V_t}{\beta} \quad \text{(Eq 3.16.2e)}$$

for a gas accumulator

$$C_h = \frac{(V_t - V_l)^2}{p_t V_t} \quad \text{(Eq 3.16.2f)}$$

where V_t = total accumulator volume

V_l = volume of liquid stored

p_t = precharge pressure

3.16.3 Systems

In the analysis of a hydraulic circuit or system, the parameters may be either lumped or distributed. To simplify analysis a lumped parameter system is commonly assumed. In a lumped parameter system each parameter is dealt with individually and any interreaction effects are neglected.

3.16.4 Impedance Law

The general impedance law in complex form is stated

$$\Delta p = Z_h q \quad \text{(Eq 3.16.4a)}$$

where

$Z_h = R_h + jX_h$

X_h = reactance, inductive (ωL_h), or capacitive $\left(-\frac{1}{\omega C_h}\right)$

The magnitude of the impedance vector is stated

$$|Z_h| = \sqrt{R_h^2 + X_h^2} \quad \text{(Eq 3.16.4b)}$$

The phase angle is stated

$$\theta = \tan^{-1}\left(\frac{X_h}{R_h}\right) \quad \text{(Eq 3.16.4c)}$$

3.16.5 Equivalent Circuit Equations

An elementary hydraulic system is shown in Figure 3.16.5a.

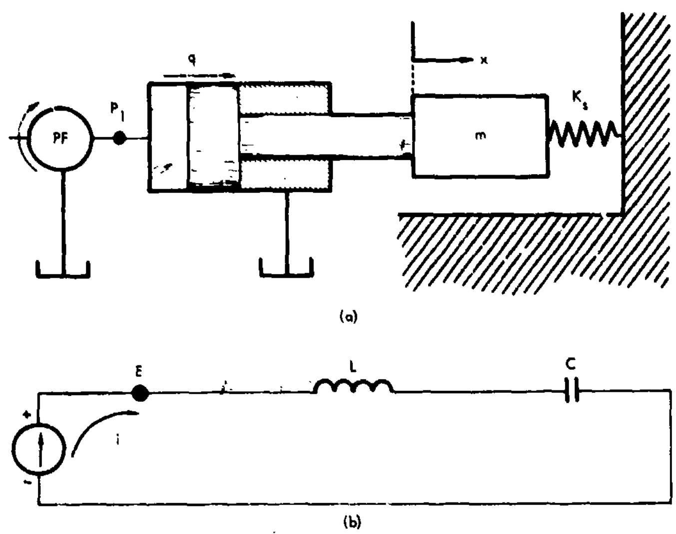

Figure 3.16.5a, b. Hydraulic System and Equivalent Circuit
(Reprinted with permission from "Hydraulics and Pneumatics," March 1961, R. E. Raymond, Copyright 1961 by the Industrial Publishing Corporation)

The force balance equation for the piston is stated

$$F_1 = m \frac{d^2x}{dt^2} + k_s x \quad \text{(Eq 3.16.5a)}$$

or

$$p_1 = \frac{m}{A^2}\frac{dq}{dt} + \frac{k_3}{A^2}\int q\, dt \quad \text{(Eq 3.16.5b)}$$

if, $\frac{m}{A^2} = L_h$ and $\frac{k_1}{A^2} = \frac{1}{C_h}$

Then

$$p_1 = L_h \frac{dq}{dt} + \frac{1}{C_h}\int q\,dt \qquad \text{(Eq 3.16.5c)}$$

In frequency form

$$\frac{p_1}{q} = j\left[\frac{\omega^2 L_h C_h - 1}{\omega C_h}\right] \qquad \text{(Eq 3.16.5d)}$$

The equivalent electrical circuit uses a current generator to supply current to a series inductive and capacitance circuit, shown in Figure 3.16.5b. The equations are

$$E = L\frac{di}{dt} + \frac{1}{C}\int i\,dt \qquad \text{(Eq. 3.16.5e)}$$

or frequency form

$$\frac{E}{i} = j\left[\frac{\omega^2 LC - 1}{\omega C}\right] \qquad \text{(Eq 3.16.5f)}$$

REFERENCES

*References added March 1967

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SYMBOL	QUANTITY	UNIT	DIMENSION
a	Acceleration	ft/sec^2	L/t^2
a	Constant for Beattie-Bridgeman Equation	ft^3	L^3
a	Constant for Van der Waals' Equation	Atm ft^6/mole^2	FL^4/M^2
a	Cross-sectional area	in^2	L^2
A	Cross-sectional area	ft^2	L^2
A_0	Constant for Beattie-Bridgeman Equation	Atm ft^3/lb-mole	L
b	Constant for Beattie-Bridgeman Equation	ft^3/lb-mole	L^3/F
b	Constant for Van der Waals' Equation	ft^3/mole	L^3/M
b	Width	in	L
B_0	Constant for Beattie-Bridgeman Equation	ft^3/lb-mole	L^3/F
c	Constant for Beattie-Bridgeman Equation	ft^3 °R^3/lb-mole	L^3T^3/F
c_c	Coefficient of contraction	dimensionless	
c_p	Specific heat at constant pressure	Btu/lb$_m$ °F	FL/MT
c_v	Coefficient of velocity	dimensionless	
c_v	Specific heat at constant volume	Btu/lb$_m$ °F	FL/MT
C_D	Discharge coefficient	dimensionless	
C_h	Hydraulic capacitance	in^3/psi	L^3F
C	Modified discharge coefficients for orifices, nozzles, and venturis	dimensionless	
C_v	Flow coefficients for valves	gpm/$\sqrt{\text{lb}_f/\text{in}^2}$	$L^{3/2}/tF^{1/2}$
d	Diameter	in.	L
D	Diameter	ft	L
E	Modulus of elasticity	lb$_f$/in^2	F/L^2
f_D	Darcy friction factor	dimensionless	
f_F	Fanning friction factor	dimensionless	
F	Flow factor	ft^3/psi	L^5/F

SYMBOL	QUANTITY	UNIT	DIMENSION
F	Force	lb$_f$	F
F_b	Buoyant force	lb$_f$	F
F_r	Froude number	dimensionless	
g	Local acceleration of gravity (32.2 ft/sec^2 at sea level on earth)	ft/sec^2	L/t^2
g_c	Conversion constant = 32.2 in $F = \frac{1}{g_c} ma$	$\frac{\text{lb}_m}{\text{lb}_f}$ ft/sec^2	ML/Ft^2
h	Enthalpy per pound-mass	Btu/lb$_m$	FL/M
h	Height	in.	L
h_f	Head loss due to friction	ft	L
H	Total head in feet of fluid	ft	L
J	Conversion constant = 778	ft-lb$_f$/Btu	*none*
k	Compressibility	in^2/lb$_f$	L^2/F
k	Thermal conductivity	Btu/hr ft^2 (°F/ft)	F/tF
k_i	Inertia resistance	ft	L
k_p	Permeability	ft^2	L^2
K	Resistance coefficient *subscript:* B Bend resistance c Contraction en Enlargement e Entrance	dimensionless	
K_F	Flow coefficient for fluid meters (equals the modified discharge coefficient C divided by the area factor) $K_F = \frac{C}{\sqrt{1-\left(\frac{D_0}{D_1}\right)^4}}$	dimensionless	
K_{cv}	Cavitation number	dimensionless	
K_v	C_v-correction for vapors	dimensionless	
l	Length	in.	L
L	Length	ft	L
L_c	Characteristic length	ft	L
L_e	Equivalent length in pipe diameters	dimensionless	

SYMBOLS,

SYMBOL	QU
L_h	Hydraul inductan
m	Mass
$\dot{m}$	Mass flo
M	Mach nu
M.W.	Molecula
n	Any exp
p	Pressur
p_r	Pressur
P	Pressur
P	Permeat
P_w	Wetted
q	Heat pe mass
q	Volume
q_{lg}	Heat of
Q	Volume
R	Gas con
R	Radius
Re	Reynol
R_h	Hydra
R_0	Univer consta
s	Specifi
S	Specifi
t	Time
T	Tempe absolut
T_r	Tempe reduce
u	Intern pound-
v	Specifi
V_x	Velocit directi
V_y	Velocit directi
V_z	Velocit directi

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SUPERSEDES: OCTOBER 1965

SYMBOLS, UNITS, AND DIMENSIONS

QUANTITY	UNIT	DIMENSION
Force	lb_f	F
Buoyant force	lb_f	F
Froude number	dimensionless	
Local acceleration of gravity (32.2 ft/sec² at sea level on earth)	ft/sec^2	L/t^2
Conversion constant = 32.2 in $F = \frac{1}{g_c} ma$	$\frac{lb_m}{lb_f}$ ft/sec²	ML/Ft^2
Enthalpy per pound-mass	Btu/lb_m	FL/M
Height	in.	L
Head loss due to friction	ft	L
Total head in feet of fluid	ft	L
Conversion constant = 778	ft-lb_f/Btu	none
Compressibility	in²/lb_f	L^2/F
Thermal conductivity	Btu/hr ft²(°F/ft)	F/tF
Inertia resistance	ft	L
Permeability	ft²	L^2
Resistance coefficient	dimensionless	
subscript:		
B Bend resistance		
c Contraction		
en Enlargement		
e Entrance		
Flow coefficient for fluid meters (equals the modified discharge coefficient C divided by the area factor)	dimensionless	
K = $\frac{C}{\sqrt{1-\left(\frac{D}{D}\right)^4}}$		
Cavitation number	dimensionless	
C correction for vapors	dimensionless	
Length	in	L
Length	ft	L
Characteristic length	ft	L
Equivalent length in pipe diameters	dimensionless	

SYMBOL	QUANTITY	UNIT	DIMENSION
L_h	Hydraulic inductance	psi sec²/in³	Ft^2/L^5
m	Mass	lb_m	M
$\dot{m}$	Mass flow rate	lb_m/sec	M/t
M	Mach number	dimensionless	
M.W.	Molecular weight	dimensionless	
n	Any exponent	dimensionless	
p	Pressure, absolute	lb_f/in^2	F/L^2
p_r	Pressure, reduced	dimensionless	
P	Pressure, absolute	lb_f/ft^2	F/L^2
P	Permeation rate	ft/lb_f sec	L^3/Ft
P_w	Wetted perimeter	ft	L
q	Heat per pound-mass	Btu/lb_m	FL/M
q	Volumetric flow rate	in³/sec	L^3/t
q_{fg}	Heat of vaporization	Btu/lb_m	FL/M
Q	Volumetric flow rate	ft³/sec	L^3/t
R	Gas constant $\frac{1544}{M.W.}$	ft-lb_f/lb_m °R	LF/MT
R	Radius	ft	L
Re	Reynolds number	dimensionless	
R_h	Hydraulic resistance	psi sec/in³	Ft/L^5
R_u	Universal gas constant 1544	ft-lb_f/lb-mole °R	LF/MT
s	Specific entropy	Btu/lb_m °R	FL/MT
S	Specific gravity	dimensionless	
t	Time	sec	t
T	Temperature, absolute	°R	T
T_r	Temperature, reduced	dimensionless	
u	Internal energy per pound-mass	Btu/lb_m	FL/M
v	Specific volume	ft³/lb_m	L^3/M
V_x	Velocity in x direction	ft/sec	L/t
V_y	Velocity in y direction	ft/sec	L/t
V_z	Velocity in z direction	ft/sec	L/t

SYMBOL	QUANTITY	UNIT
V	Mean velocity	ft/sec
V	Volume	ft³, in³
V_s	Sonic velocity	ft/sec
w	Specific weight	lb_f/ft^3
$\dot{w}$	Weight flow rate	lb_f/sec
W	Weber number	dimensionless
W	Body weight	lb_f
W_s	Shaft work	ft-lb_f/lb_m
x	Distance in x direction	ft
y	Ratio of gas to liquid mass flow rates	dimensionless
Y	Expansion factor	dimensionless
Z	Compressibility factor	dimensionless
Z	Elevation above arbitrary datum plane	ft
α	Constant in Beattie-Bridgeman Equation	dimensionless
β	Bulk modulus of elasticity	lb_f/in^2
β	Mean bulk modulus of elasticity	lb_f/in^2
γ	Ratio of specific heats	dimensionless
δ	Thickness of boundary layer	ft
ϵ	Eccentricity	dimensionless
ϵ	Height of surface roughness	ft
Θ	Viscosity index	dimensionless
λ	Mean free path	in.
μ_f	Absolute viscosity	lb_f sec/ft², lb_f sec/in²
μ_m	Absolute viscosity	lb_m/ft sec
ν	Kinematic viscosity	ft²/sec
π	3.14159	dimensionless
π	Dimensionless ratio used in Buckingham π Theorem	dimensionless
ρ	Density	lb_m/ft^3, lb_m/in^3
σ	Surface Tension	lb_f/ft
Σ	Summation	dimensionless
τ	Shear stress	lb_f/ft^2

B

SYMBOL	QUANTITY	UNIT	DIMENSION
V	Mean velocity	ft/sec	L/t
V	Volume	ft^3, in^3	L^3
V_s	Sonic velocity	ft/sec	L/t
w	Specific weight	lb_f/ft^3	F/L^3
$\dot{w}$	Weight flow rate	lb_f/sec	F/t
W	Weber number	dimensionless	
W	Body weight	lb_f	F
W_s	Shaft work	$ft\text{-}lb_f/lb_m$	LF/M
x	Distance in x direction	ft	L
y	Ratio of gas to liquid mass flow rates	dimensionless	
Y	Expansion factor	dimensionless	
Z	Compressibility factor	dimensionless	
Z	Elevation above arbitrary datum plane	ft	L
α	Constant in Beattie-Bridgeman Equation	dimensionless	
β	Bulk modulus of elasticity	lb_f/in^2	F/L^2
β	Mean bulk modulus of elasticity	lb_f/in^2	F/L^2
γ	Ratio of specific heats	dimensionless	
δ	Thickness of boundary layer	ft	L
ϵ	Eccentricity	dimensionless	
ϵ	Height of surface roughness	ft	L
Θ	Viscosity index	dimensionless	
λ	Mean free path	in.	L
μ_f	Absolute viscosity	$lb_f\ sec/ft^2$, $lb_f\ sec/in^2$	Ft/L^2
μ_m	Absolute viscosity	$lb_m/ft\ sec$	M/Lt
ν	Kinematic viscosity	ft^2/sec	L^2/t
π	3.14159	dimensionless	
π	Dimensionless ratio used in Buckingham π Theorem	dimensionless	
ρ	Density	lb_m/ft^3, lb_m/in^3	M/L^3
σ	Surface Tension	lb_f/ft	F/L
Σ	Summation	dimensionless	
τ	Shear stress	lb_f/ft^2	

Abbreviations

atm	Atmospheres
Btu	British thermal units
cal	Calories
cfm	Cubic feet per minute
cfs	Cubic feet per second
fpm	Feet per minute
fps	Feet per second
gpm	Gallons per minute
pcf	Pounds per cubic foot
psf	Pounds per square foot
psfa	Pounds per square foot absolute
psi	Pounds per square inch
psia	Pounds per square inch absolute
psig	Pounds per square inch gage
scch	Standard cubic centimeters per hour
sccs	Standard cubic centimeters per second
scfh	Standard cubic feet per hour
scfm	Standard cubic feet per minute
scfs	Standard cubic feet per second
scim	Standard cubic inches per minute
scis	Standard cubic inches per second

SUBSCRIPTS

0	Throat condition
1	Upstream or initial conditions
2	Downstream or final conditions
3	Downstream branch
a	Atmosphere
c	Critical
f	Final
g	Gas
i	Initial or reference conditions
l	Liquid laminar
m	Model
n	Any integer
p	Prototype
s	Adiabatic
t	Total
v	Vapor
M	Molecular
T	Isothermal
TP	Two-phase
res	Reservoir

Dimensions

F	*Force*
M	*Mass*
L	*Length*
t	*Time*
T	*Temperature*

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TABLE OF CONTENTS

ILLUSTRATIONS

4.1 INTRODUCTION

The purpose of the Fluid Systems Section is to describe the function of the major fluid components used in typical aerospace fluid system applications. The aerospace systems covered are classified as propellant feed systems, attitude control systems, secondary injection systems, and fluid power systems. The operation of some typical system configurations is described with emphasis on the function and requirements of the fluid components making up the system. The Fluid Power Sub-Section presents a discussion of the comparative advantages and disadvantages of hydraulics and pneumatics for fluid power applications.

4.2 FLUID POWER SYSTEMS

4.2.1 Introduction

A fluid power system is a system that transmits and controls power through the use of a pressurized fluid. This Sub-Section presents a brief description of some typical fluid power systems and discusses the comparative characteristics of hydraulic and pneumatic systems.

4.2.2 Typical Fluid Power System

Hydraulic and pneumatic power systems are used extensively to provide the muscles in aerospace applications such as moving gimballed rocket motors and aerodynamic control surfaces (air rudder, jet vanes, and jetavators) and actuating control valves for gas generator systems, propellant feed systems, propellant utilization systems and head suppression systems. The following major components are common to all fluid power systems:

a) Fluid: liquid or gas
b) Power source: pump, compressor, accumulator, high pressure storage vessel, or solid propellant gas generator
c) Control valve: servovalve, solenoid valve, shuttle valve, or sequence valve
d) Actuator: linear and rotary actuators
e) Relief valves
f) Check valves
g) Filters
h) Low pressure sink: reservoir for hydraulic systems (closed loop), atmosphere for pneumatic systems (open loop).

A typical closed loop hydraulic system used for actuating a gimballed thrust chamber is shown schematically in Figure 4.2.2. The hydraulic system illustrated is composed of a pump driven either by an electric motor or a power take-off from the main turbopump or other auxiliary power source, flow control servo valves, actuators, reservoir, accumulator, relief valves, check valves, filters, and disconnects. Hydraulic pressure in the order of 3000 psi operating through servo valves is transmitted to double-acting linear actuators to gimbal the main rocket engine in two perpendicular planes for directional control of a rocket vehicle. A similar actuator is used to direct a jet nozzle for roll control.

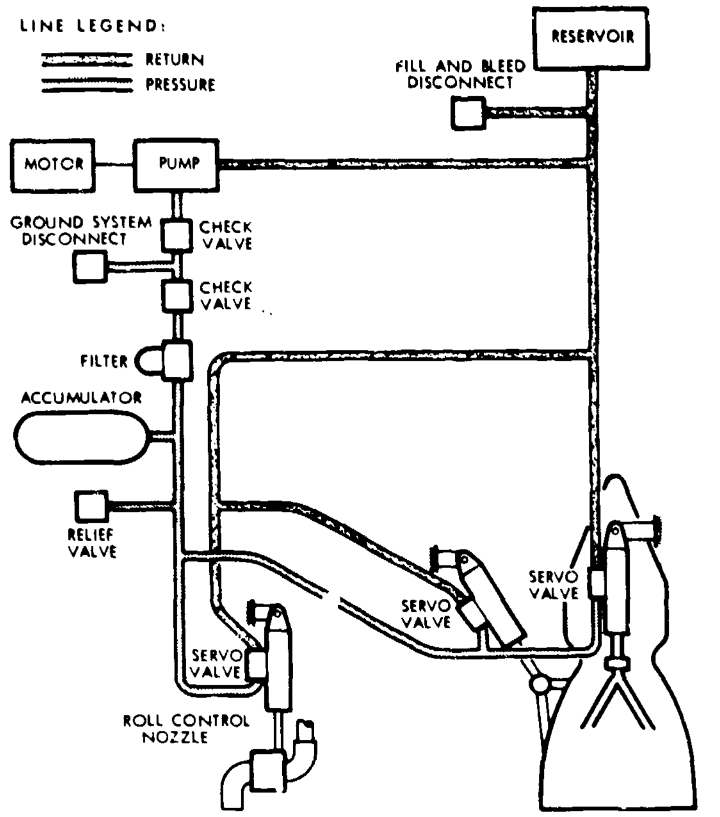

Figure 4.2.2. Typical Missile Hydraulic System

In a pneumatic fluid power system, instead of a pump the high pressure power source would typically be a compressor, hot gas generator and relief valve, or high pressure accumulator and regulator. Pneumatic systems are typically "one-pass," (open loop) systems.

4.2.3 Comparison Factors Betw... Hydraulic and Pneumatic Fluid Power Systems

Most fluid power requirements could be met by using either a hydraulic or a pneumatic system. The selection of a system using either fluid medium involves overlapping advantages and disadvantages requiring great care in weighing all the significant factors prior to selecting a particular system. In general, the following represent some of the major considerations involved.

4.2.3.1 TEMPERATURE. Based on aircraft experience, hydraulic systems have had a successful history in the temperature range of -65 to approximately 300°F. More recently, speed and environmental requirements have increased operating temperature conditions to 400°F. Under extremely limited time durations, operation is possible at temperatures of 550°F. Based on present knowledge, it appears that a temperature limitation for organic fluids is approximately 900°F.

One of the characteristics of fluid flow in hydraulic systems is the temperature increase experienced during pressure drops through restrictions and line flow. The recirculating nature of most hydraulic systems tends to result in a temperature buildup, thus affecting not only the viscosity of the fluid medium itself, but also other system components as well.

Low fluid viscosity, poor lubricity, and low bulk modulus have been the primary restrictions limiting the use of pneumatic systems operating in the normal temperature ranges. The possibility of entrained moisture in the gas being subjected to low temperature presents an icing problem. Although air has a low viscosity at normal temperatures, at temperatures above 100 F its viscosity is greater than that of high temperature hydraulic fluids (Reference 27-3). Over temperature ranges which could be imposed by a space flight environment, the viscosity of air may vary by a factor of three, while the viscosity of hydraulic fluids may change by a factor of 20,000. The bulk modulus of hydraulic fluid also varies greatly with temperature, while the bulk modulus of air is relatively independent of temperature changes. As a result of these advantages, high temperature pneumatic systems upwards of 1000 F are being investigated. Solid propellant gas generators provide a convenient source of pneumatic power for flight systems.

4.2.3.2 FLUID MEDIA. Within the broad classification of pneumatic and hydraulic systems, a wide choice among fluid media is available for various applications. Some of the more common types of hydraulic and pneumatic fluids are described in the following paragraphs.

Hydraulic. Depending upon the environment and the desired performance and safety consideration characteristics, almost any liquid can be used as a hydraulic fluid

Hydraulic fluids which combine the best properties of low viscosity, high bulk modulus, good lubricity, and have minimum problems of corrosion and safety are petroleum based fluids such as MIL-O-5606 and silicone fluids such as Oronite 8515.

Although having less desirable characteristics, liquid propellants are sometimes used as hydraulic fluids in rocket systems for such applications as the actuation of shutoff valves

Since the parameters such as bulk modulus, viscosity, and lubricity are temperature dependent and directly affect system operation, hydraulic systems have been classified into the following temperature categories:

Type I: use fluids conforming to MIL-O-5606 almost exclusively with operating temperature limits from -65 to 160 F.

Type II: operate from -65 to 275°F. The petroleum-based MIL-O-5606 fluids can also operate in this temperature range, provided an airless system is used.

Type III: operate from -65°F up to 400°F utilize silicone fluids such as Oronite 8515.

Liquid metals are being considered as hydraulic fluids for extreme high temperature service, NaK_{77}, a sodium-potassium eutectic alloy showing promise for 1000°F service.

Pneumatic. Five types are considered:

Air: commonly used in aircraft applications due to its accessibility by way of compressor bleed air, storage bottles, or other auxiliary gas generators.

Nitrogen: widely used in space applications, particularly in ground checkout equipment, due to its inertness and relatively low cost.

Helium: generally used in flight systems for tank pressurization, actuation of components, attitude control, etc., due to its lightweight feature.

Liquified Gases: application of heat required for vaporation to attain sufficient levels of pressurization. Liquid nitrogen and anhydrous ammonia are commonly used.

Hot Gas: high temperature gas usually generated by various combinations of propellants such as hypergolic (hydrazine and nitrogen tetroxide), monopropellant (hydrogen peroxide), and solid propellant gas generators.

4.2.3.3 FLUID DENSITY. The relative density difference between hydraulic and pneumatic media is a significant factor in system selection and design.

Hydraulic fluid weighs approximately 60 pounds per cubic foot. Dry air weighs about 0.075 pounds per cubic foot at atmospheric pressure. Therefore, pneumatic circuits are usually lighter than comparable hydraulic systems operating at the same pressure.

For a given actuation force, the lower mass pneumatic medium which travels at a much higher velocity than hydraulic fluids provides a much higher response, particularly where large actuator strokes are required.

Water hammer which can cause severe damage in fluid systems is a function of fluid density and the square of the velocity, hence particular care must be given to the design of high speed hydraulic systems with long fluid runs.

4.2.3.4 BULK MODULUS. Bulk modulus is a measure of the compressibility of fluids, and is expressed as the reciprocal of the compressibility factor. It is an important factor in servomechanisms, where the natural frequency of the system is directly proportional to the square root of the bulk modulus of the fluid used.

Liquids are almost incompressible, making them ideal for accurate actuator speed control and positive response in servo systems. Actually, at 1700 psi hydraulic oil is compressed about 1 percent. Compressibility of hydraulic fluids is of special interest when working at very high pressures or, with devices the operational characteristics of which depend on the rigidity of the control and power fluids. Air entrainment and high temperatures tend to reduce the bulk modulus.

Many of the arguments favoring hydraulics over pneumatics rest on the high bulk modulus of hydraulic systems. Although many innovations have developed in pneumatic servomechanisms to improve the stiffness of control systems, all these techniques result in increased bulk, higher weight, lower reliability, and other undesirable effects.

The compressibility of pneumatic fluids provides for self-damping characteristics making pneumatic systems less susceptible to conditions of shock resulting from rapid acceleration and deceleration.

4.2.3.5 LUBRICITY. Gases do not have inherent lubricating qualities, nor are they able to carry lubricants under pressure for extended distances in contrast to the inherent lubricating characteristics of hydraulic oils and the relative ease of suspending or mixing lubricants with hydraulic fluids. Lubrication is definitely a problem in pneumatic circuits wherever wear of mating surfaces can occur.

4.2.3.6 LEAKAGE. Traditionally, leakage constitutes one of the main trouble spots for hydraulic systems. Hydraulic leakage is easily noticeable by the obvious traces it leaves in the area surrounding the system and by the general appearance of its components Leakage can be controlled by careful manufacturing and maintenance procedures. This means close fits, good finishes, adequate processing techniques, and proper selection of components. At higher pressures, some parts such as spool type valves are no longer practical because the clearances involved induce excessive leakage. Most satisfactory valves in this respect have been the poppet type with lapped seats. Difficulties can also be traced to lodging of minute particles in valve seats which, in turn, points out the necessity for good filtration.

Hydraulic leaks can easily be detected by visual means whereas pneumatic leaks are difficult to locate. The greatest cause for pressure loss in pneumatic systems is in the large number of fitting joints and line-to-unit junctions throughout the system. To minimize leakage, all threaded connections should be properly sealed by means of some resilient material in the joint itself.

4.2.3.7 CONTROLS. The design and selection of controls is also affected by media considerations. Directional control valves used in conjunction with pneumatic circuits are generally quick acting. Hydraulic valves are generally slower in action than pneumatic valves. This is necessary if the shock loads imposed on the system during shift are to be kept to a minimum. Hydraulic valves are generally also of the lapped-spool type where the clearance flow of the hydraulic fluid can be expected to lubricate and continuously flush the working surfaces. The difference in the media also helps to determine the materials used in construction of controls.

Pressure pulses can be transmitted at very high speeds in pneumatic systems without the need for special provisions to accommodate the high flow velocities, (i.e., prevent water hammer). A pneumatic medium also provides smoother operation. For these reasons, pneumatic circuits are widely adopted for high speed control applications.

4.2.3.8 RADIATION. Nuclear radiation has no effect on common pneumatic media. However, most hydraulic fluids, especially hydraulic oils, undergo a chemical and physical change in the presence of nuclear radiation. Also, strong consideration must be given to the compatibility of seals, and gaskets with hydraulic fluids under radiation conditions.

4.2.3.9 SAFETY. Assuming that a hydraulic system is bled of trapped air, the energy content of a compressed liquid is relatively low because of its inherent high bulk modulus. Thus, the rupturing of a line or the loss of a fitting should not result in an explosive loss of the liquid. However, in the case of high pressure pneumatic systems, a structural failure could result in a large and destructive explosion.

One of the greatest hazards attributed to high temperature hydraulic systems using organic fluids lies in the potential danger of spontaneous combustion in the event of leakage. Also, toxicity problems may be produced as a result of the thermal breakdown products of some of the fluids. In all cases, it is evident that all high temperature and high pressure fluid systems must be treated with extreme caution.

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	27-7	

4.3 ROCKET PROPELLANT FEED SYSTEMS

4.3.1 Introduction

The primary objective of rocket propellant feed systems is to provide liquid propellant(s) to the rocket motor at the desired pressure, flow rate, and mixture ratio. The purpose of this Sub-Section is to describe typical rocket propellant feed systems, with emphasis on the function of fluid components making up the system.

4.3.2 Description

Liquid propellant rocket propulsion systems can be broadly divided into pressure fed systems and pump fed systems. Pressure fed systems are sometimes referred to as gas pressurized systems. Each of these two basic systems can be subdivided into three major subsystems: (1) the pressurization system, (2) the propellant storage system, and (3) the engine system. A fourth propulsion subsystem employed in some rocket vehicles is a propellant utilization system. The engine system may be further subdivided into assemblies which include turbopump assembly, the flow control assembly, and the rocket motor consisting of injector, thrust chamber, and nozzle. Figure 4.3.2 illustrates these subassemblies integrated into a typical pump fed system. By removing the turbopump assembly and connecting points (B) and (C), the illustration can be made to represent a typical pressure fed system.

4.3.3 Comparison of Pressure Fed and Turbopump Liquid Rocket Systems

Of the two systems, the pressure fed system is by far the simplest and is usually the preferred system for low thrust vehicles with short mission durations. In terms of component requirements, the principle difference between the two systems is the presence or absence of a turbopump assembly. Whereas in the pressure fed system stored gas provides the energy for expelling the propellants, in the pump fed system the turbopump provides the energy. Although a pressurization system is not used with a turbopump system to expel propellants, it is usually required to keep the tank pressure above the minimum value necessary for tank rigidity (a partial vacuum could collapse the tanks as propellants are withdrawn) and to provide adequate suction head to prevent pump cavitation.

Although the weight of tanks and plumbing is greater for a pressure fed system than for a comparable pump fed system, the pump fed system's total weight for low thrust, short duration missions is more than for a pressure fed system. The tradeoff point above which the turbopump system is lighter of the two varies with mission duration, propellant performance characteristics, and propellant density. In general, this point is found to occur between 8000 and 20,000 pounds of total propellant weight. Below this range, the pressure fed systems are found to be most practical. Furthermore, the relative simplicity of the pressure fed system makes it an attractive alternative in spite of a weight disadvantage of the larger sizes.

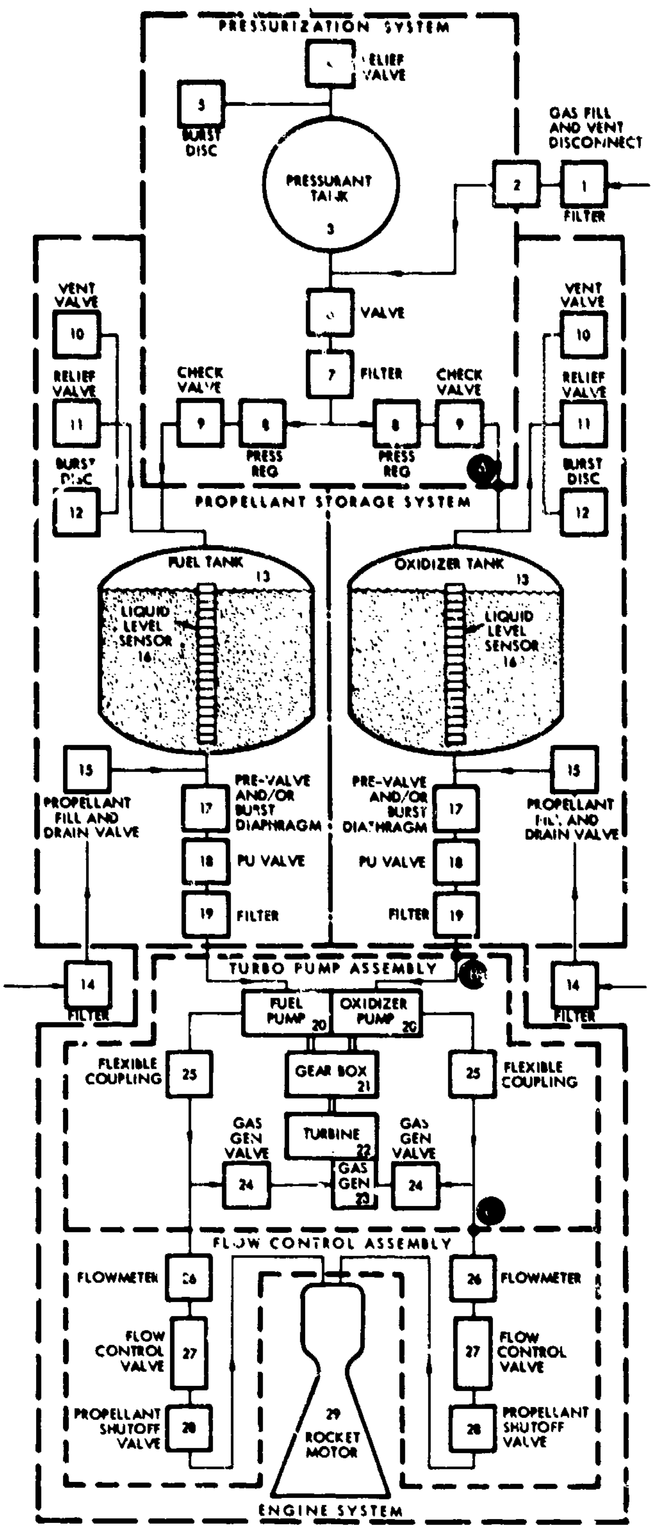

Figure 4.3.2. Typical Bipropellant Rocket Propulsion System

4.3.4 Turbopump Systems

Figure 4.3.2 illustrates a typical bipropellant turbopump system. The same figure can be used to illustrate a monopropellant turbopump system by removing the oxidizer system. In contrast to the pressure fed system, the propellant tank and feed lines upstream of the pump are maintained at relatively low pressure, usually in the range of 10 to 50 psi. A gas generator, either solid propellant, bipropellant, or monopropellant is usually used to power the turbopump. When a liquid propellant gas generator is used, it is common practice to feed the gas generator with propellants supplied by the main propellant feed system. Such a system is commonly known as a bootstrap system. A separate turbopump start system is required to develop sufficient pump pressure to feed the gas generator. At this point, bootstrap operation can take over and the system sustains itself with no external source of power. The start system for spinning up the turbine can be fed by a separate pressure fed propellant system (a so-called start system), a liquid gas generator, a solid propellant gas generator, or an auxiliary pneumatic system. Another turbopump system which to date has seen only limited use is the *tap off turbine* which is driven by combustion chamber gas, thus eliminating the need for a separate gas generator.

Referring to Figure 4.3.2, a typical turbopump propellant feed system functions in the following manner: the pressurant tank (3) is charged with a high pressure inert gas (N_2 or He) through a ground filter (1) and the fill and vent disconnect (2). When rated pressure has been reached, pressurization is discontinued and fill and vent valve disconnected. Inadvertent overpressure of the pressurant tank during filling or post fill thermal induced pressure rise is relieved by the relief valve (4). Potentially catastrophic over-pressures beyond the capability of the relief valve are vented through the burst disc (5) which is set for some pressure above the relief valve cracking pressure. The propellant tanks (13) are loaded through the propellant fill and drain valves (15) with the propellant passing through a ground filter (14) to prevent potentially harmful contaminants from being transferred from the loading system to the flight system. The vent valve (10) is opened during the filling operation to vent excess gases and vapors from the propellant tank ullage volume. The relief valve (11) and burst disc (12) provide the same safety function both during fill and any time during system operation as was described for the pressurant tank. The quantity of liquid loaded into the tanks is indicated remotely by the liquid level sensor (16).

The system is pressurized by opening the gas shutoff valve (6), often called a start valve. High pressure gas then flows through the start valve to the pressure regulator (8). The filter (7) removes particulate contaminant from the tank, valve, and plumbing which could damage the regulator. The pressure regulator reduces the pressure down to the desired tank pressure level. In the schematic shown, pressure is reduced through a single regulator. In some turbopump systems, due to the large pressure differential between maximum upstream pressure and regulated pressure, regulation is done in stages through two regulators in series. Check valves (9) are usually installed between the regulator and the tank to prevent back flow of propellant vapors to the regulator. Two reasons for preventing this back flow are: (1) to keep corrosive vapors out of the regulator itself and, (2) to prevent possible mixing of propellant vapors with resulting fire and explosion hazards. Typically, pressure regulators in pump fed rocket systems are required to regulate over a pressure range from approximately 100 to 1,300 psi down to [illegible].

After the main tanks have been pressurized, propellant flow is initiated by opening the prevalves and/or rupturing the burst diaphragms (17). A burst diaphragm or rupture type prevalve is used if long term propellant storage is required. A filter (19) is shown in the propellant feed system to protect the turbopump, flow control assembly, and injector from contaimination collected from the tank and intermediate valving and pumps. The pumps (20) must be brought up to speed before propellants are introduced into the thrust chamber. The turbine (22) which drives the pumps through a gear box (21) is started either by means of a separate gas generator tank system, by a gas generator, or by a pneumatic start system. After the pump has built up sufficient pressure, the turbopump system goes into bootstrap operation by opening the gas generator valves (24), thus introducing propellants from the high pressure side of the pump to the gas generator (23). The main propellant shutoff valves (28) are also opened after the pump has been brought up to speed. These valves are often pressure actuated, opening at a preset system pressure. This is usually accomplished by pressure actuated shuttle valve used as a pilot to control actuation of the main propellant valve. The propellant shutoff valve is used to stop as well as start flow. The shutdown characteristics of the engine (thrust termination) are to a large extent dependent on the shutoff valve characteristics. Flow control valves (27) are required in a system where throttling of the engine is required. Control valves are also commonly used in fixed thrust rocket systems in the gas generator feed system. The gas generator control valve becomes part of a closed loop system which responds to changes in thrust, opening or closing as required to maintain the thrust at the desired level. The rocket motor (29) in many systems is required to gimbal for thrust vectoring. Flexible couplings (25) in the feed lines provide for the relative motion between the engine (motor plus control assembly) and the rest of the system. In a system where propellant outage is critical, i.e., minimum residual propellants are required, a propellant utilization system consisting of tank liquid level sensors (16) or flowmeters (26) for gaging, a computer system, and a propellant utilization (PU) valve (18), is utilized to adjust the mixture ratio throughout the mission duration such that both propellants are exhausted simultaneously. The gages (level sensor or flowmeter) provide data to a computer which integrates total flow to the engine and determines the required mixture ratio which is controlled by the PU valve.

4.3.5 Pressure Fed Systems

Removal of the turbopump assembly from the system (Figure 4.3.2) by connecting points (B) and (C) converts the schematic to a pressure fed system. This system then is entirely dependent upon pneumatic pressure for its operation. The propellants are forced out of the tanks under pressure which controls the propellant flow rate. By virtue of the increased pressure, the weight of tanks and plumbing is by necessity higher in the pressure fed system than in the turbopump system. With the exception of the operation of the turbopump assembly, system operation and component functions in a pressure fed system are similar to those described for the turbopump system.

Chamber pressure for pressure fed systems commonly ranges between 100 and 300 psi in contrast to chamber pressure as high as 1000 psi encountered in turbopump systems. Pressure fed systems can be monopropellant as well as bipropellant, a monopropellant system consisting of essentially one-half the system illustrated.

4.3.6 Pressurization Systems

4.3.6.1 GENERAL. A wide range of propellant tank pressurization systems can be selected for both pressure fed and turbopump rocket systems. The following types of systems are practical for main propellant tank pressurization:

a) Stored cold gas
b) Heated gas systems (turbopump only)
c) Hot gas extraction systems (turbopump only)
d) Solid propellant gas generation
e) Auto pressurization.

4.3.6.2 STORED COLD GAS. Stored cold gas is the most common pressurization system used for both pressure fed and turbopump rocket systems. Chemically inert stored gases are virtually limited to helium and nitrogen. Neon is lighter than nitrogen but is prohibitively expensive.

The cold gas pressurization system (Figure 4.3.2) represents the simplest possible configuration with minimum components and having low activation time.

Selection of components for this type of pressurization system is based primarily on flow and pressure conditions. The pressurants are chemically compatible with most structural materials and require little consideration during component design.

4.3.6.3 HEATED GAS SYSTEM. A heated gas system (Figure 4.3.6.3a) is similar to the cold gas system except for the addition of a heat exchanger. Heating the gas provides a greater available energy from a given volume of pressurant. The heated gas system is practical only in conjunction with a turbopump system using the turbine exhaust gas as the high temperature fluid. Gas flowing from the storage bottle is heated before entering the propellant tanks. The schematic shows a high-pressure heat exchanger located upstream of the regulator. The unit could be low-pressure rated if it was located downstream of the regulator. The choice of heat exchanger rating depends on

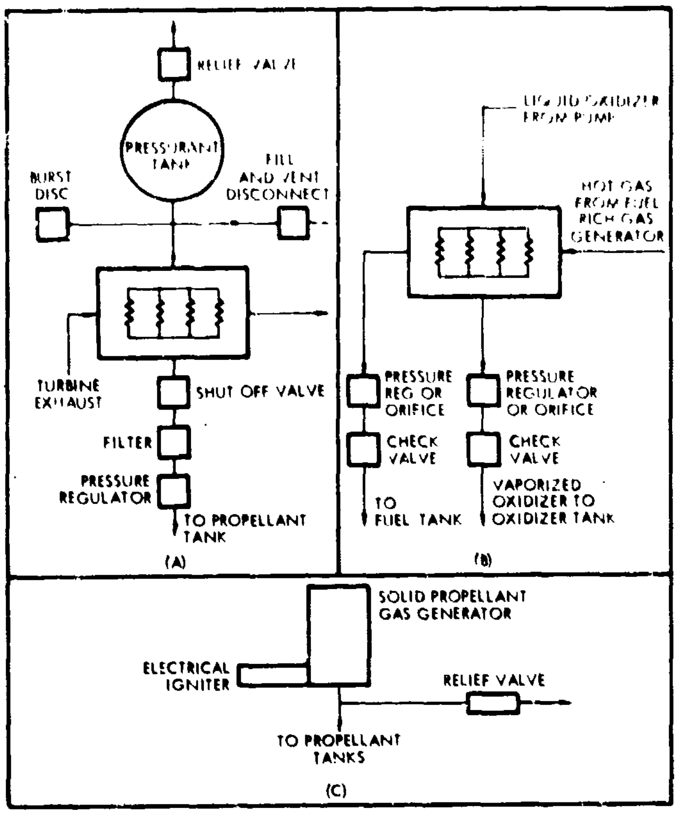

Figure 4.3.6.3a, b, c. (a) Heated Gas Pressurization System, (b) Hot Gas Extraction System, and (c) Solid Propellant Pressurization System

propulsion system size and pressure rating and has little influence on component performance and reliability.

Heat exchangers may be located at various places on the vehicle particularly where adequate heat can be extracted. In a turbopump engine it may be conventionally located in the turbine exhaust. For pressure fed systems, the heat exchanger may be located on the nozzle usually at the exit.

A heated gas system which eliminates the heat exchanger utilizes a one-shot cartridge gas generator located inside the gas storage tank.

4.3.6.4 HOT GAS EXTRACTION SYSTEM. The hot gas extraction system, also known as the autogeneous system, is used exclusively with turbopump systems. In the hot gas extraction system, the pressurant is provided by the propellants themselves, thus obviating the requirement for a separate pressurant and pressurant tank (Figure 4.3.6.3b).

Heat is generated by a fuel-rich gas generator, with the fuel-rich reaction products used directly to pressurize the fuel tank. Before entering the fuel tank the hot gas (800 to 1500°F) passes through a heat exchanger which vaporizes the oxidizer from the high pressure side of the pump which then becomes the pressurant for the oxidizer tank.

Components in this system must not only be able to withstand the high temperatures, but the corrosive propellant vapors as well.

4.3.6.5 **SOLID PROPELLANT GAS GENERATION.** Solid propellant gas generation systems (Figure 4.3.6.3c) are used in both turbopump and pressure fed systems. Solid propellant pressurization systems for liquid rockets are attractive from the standpoint of simplicity, lightweight, and compactness. The system consists of an electrical igniter, a solid propellant gas generator, and a pressure relief valve or back pressure regulator.

This type of system is useful only for single start operations. Once the generator is ignited pressurization rate and duration are fixed.

4.3.6.6 **AUTO PRESSURIZATION.** The simplest means of pressurization is an auto pressurization system which uses the vapor pressure of the propellants as the expulsion pressure. Auto pressurization is feasible with certain propellants having reasonably high vapor pressures such as ammonia, propane, oxygen, and mixed oxides of nitrogen.

REFERENCES

2-41
2-42
471-1
331-1
472-1

4.4 ATTITUDE CONTROL ROCKET SYSTEMS

4.4.1 Introduction

This Sub-Section describes fluid component functions in typical attitude control rocket systems. Attitude control systems are used for manned and unmanned spacecraft to provide vehicle reorientation prior to velocity vector correction or for maintaining a fixed orientation on a point in space (station keeping), outside of the atmosphere where aerodynamic controls are ineffective. Attitude corrections are necessary to compensate for the forces which act on a space vehicle such as solar radiation pressure, magnetic fields, change in internal momentum within the vehicle, booster stage firing or separation, meteroid impact, aerodynamic forces, and gravitation gradients.

Attitude sensing devices and electronic supporting equipment are not included within the scope of this sub-section.

4.4.2 General Description of Attitude Control Rocket Systems

Attitude control rocket systems are reaction control devices which orient the vehicle about the roll, pitch, and or yaw axes. Electrical signals from the spacecraft guidance system actuate flow control valves which meter gas or propellant flow and thereby control rocket thrust and correct vehicle attitude. Gas or propellant flow in attitude control systems are regulated either by an ON-OFF control valve with pulse width modulation or pulse frequency modulation or by a proportional flow control valve. Systems with pulse width modulation vary the pulse duration of a fixed thrust while those with pulse frequency modulation vary the frequency of a fixed thrust of fixed duration. In systems with proportional valves, the flow rate and thrust is proportional to the magnitude of the guidance signal to the valve. During periods of inactivity for systems containing an ON-OFF type start valve, the system is shut down by closing the start valve. When a squib actuated burst type valve is used for system activation, shutoff is achieved by regulator lockup and by the closed control valves. A low pressure non-reactive* relief valve located between the flow control valves and pressurization system serves to prevent over-pressurization of the control system which could occur in the event the pressure regulator leaks during lockup, i.e., when no attitude correction is called for and all flow control valves are closed. Selection of the type of flow control (thrust control) depends upon the accuracy, response, and leakage requirements necessary for a given mission. Error-sensing devices continue attitude correction until the desired vehicle orientation is achieved. The number and magnitude of attitude corrections necessary to orient the vehicle dictate the total impulse requirement for the space vehicle. Hence, when sizing attitude control systems, accurate assessment of the size and number of attitude corrections required is important.

The basic distinctions between the various attitude control systems are the state of the fluid and the feed system necessary to supply the fluid. Attitude control rocket systems are classified in this handbook according to the state of the fluid passing through the flow control valves. These include:

a) Cold gas
    Stored gas
    Stored liquefied gases

b) Monopropellant

c) Bipropellant

d) Hot gas
    Solid propellant hot gas generation
    Monopropellant hot gas generation
    Bipropellant hot gas generation

Schematics of these systems are illustrated in Figures 4.4.3.1, 4.4.3.2, 4.4.4, 4.4.5, 4.4.6, 4.4.7, and 4.4.8.

4.4.3 Cold Gas Systems

4.4.3.1 **STORED GAS ATTITUDE CONTROL ROCKET SYSTEM.** A stored gas attitude control rocket system is

*Discharge is equally divided between two opposing nozzles to eliminate any reactive thrust from the relief exhaust.

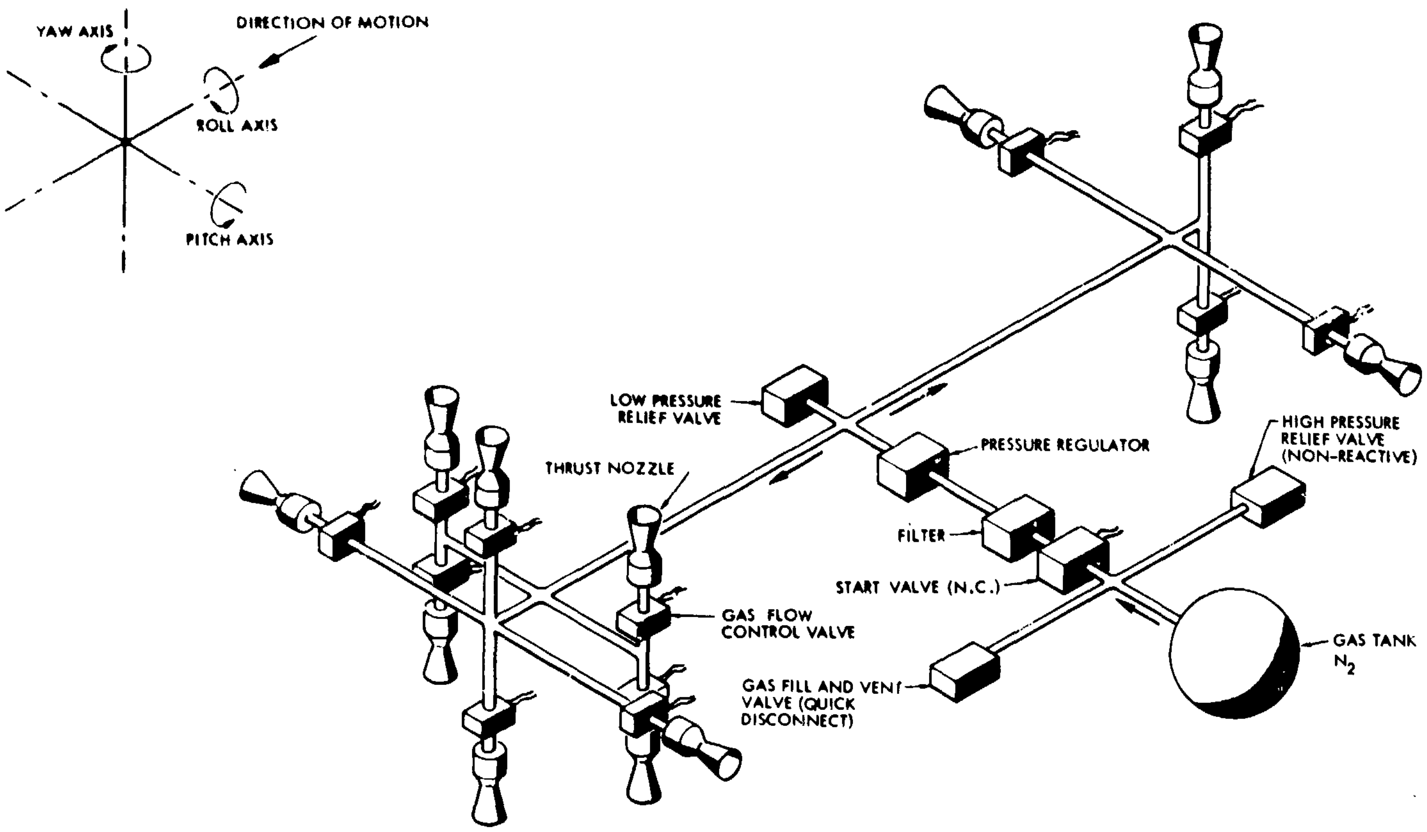

Figure 4.4.3.1. Stored Gas Attitude Control Rocket System

shown schematically in Figure 4.4.3.1. The gas tank is charged to rated pressure with gaseous nitrogen, helium, or propane which enters the tank through a gas fill and vent disconnect. The disconnect usually has an integral poppet on the accumulator side for shutoff when disconnected. If the disconnect does not have a poppet, a separate shutoff valve, squib or solenoid-actuated, is located on the accumulator side of the disconnect. After rated pressure is obtained, pressurization is discontinued and the gas fill and vent valve disconnected. Inadvertent over-pressure of the gas tank during filling or by post-fill pressure rise due to heating is relieved by a non-reactive, high pressure relief valve. In some systems the relief valve is either replaced by, or used in conjunction with, a burst disc. The burst disc is usually set above the relief valve cracking pressure. It serves to prevent hazards to personnel during pre-launch operations in the event of potentially catastrophic tank over-pressurization resulting from a closed failure of the relief or inadequate relief valve capacity in the event of an extremely rapid pressure buildup in the tank.

The attitude control system is activated by energizing the start valve (squib or solenoid-actuated, normally-closed shutoff valve) which permits gas to flow from the accumulator to the pressure regulator. A filter upstream of the regulator removes potentially harmful particulate contaminants emanating from the start valve or the tank and upstream plumbing. Gas pressure is reduced by the regulator to a preset value determined by thrust requirements of the system. After the system has been energized, electrical signals from the spacecraft guidance system actuate the flow control valves metering gas to the appropriate control rocket for attitude correction thrust. Flow regulation for attitude control systems is described in Sub-Topic 4.4.2.

4.4.3.2 LIQUEFIED GAS ATTITUDE CONTROL ROCKET SYSTEM. The liquefied gas* (e.g., anhydrous ammonia) attitude control system shown in Figure 4.4.3.2 resembles the stored gas system, differing principally in the design of the storage tank and the addition of a vent valve and gas accumulator. Modification of the tank includes:

a) Addition of tank insulation to control propellant vaporization

b) Addition of a bladder with a check valve to separate the liquid and vapor

c) Reduction of tank wall thickness permitted by reduction of storage pressure

d) Addition of heating equipment, when required to accelerate or control propellant vaporization.

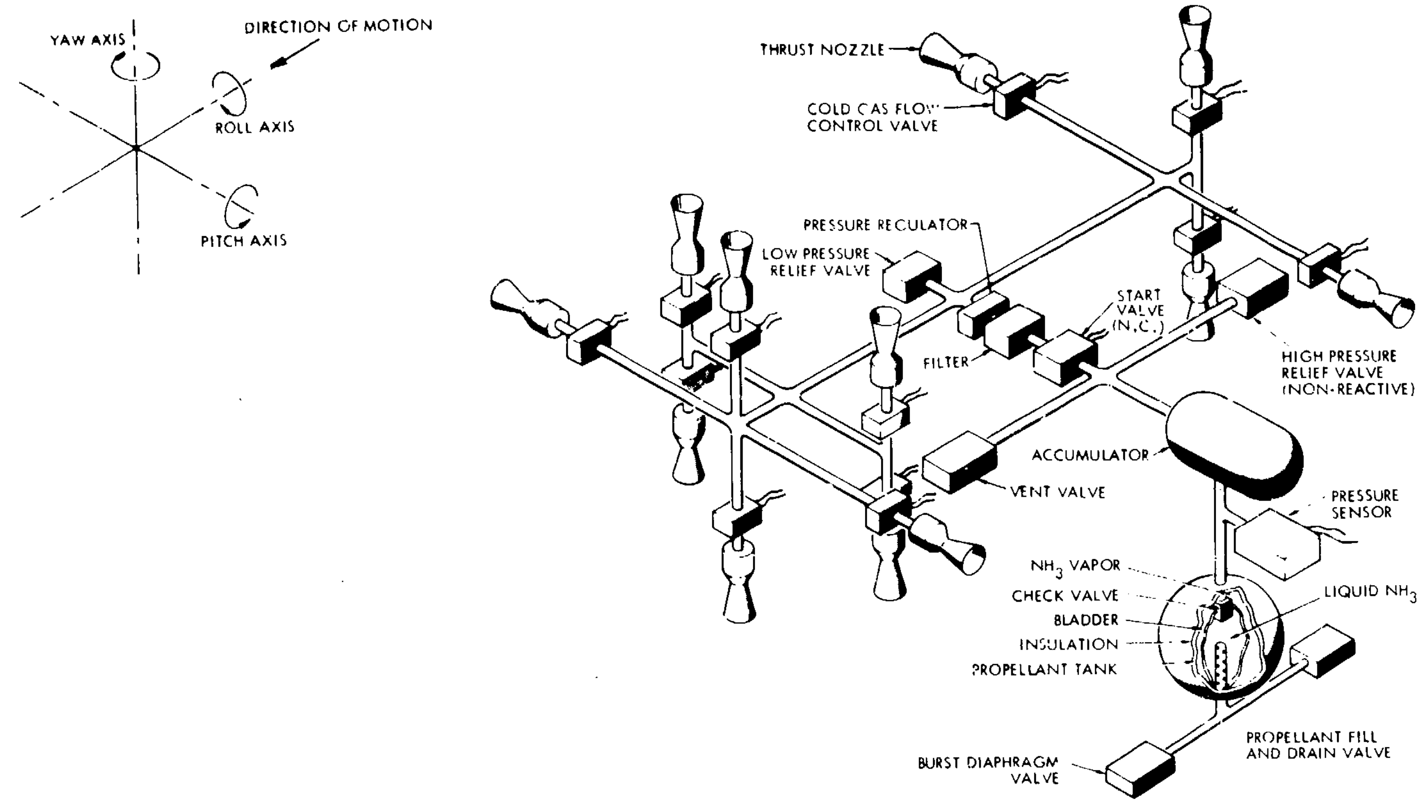

Figure 4.4.3.2. Liquefied Gas Attitude Control System

The propellant is loaded through the fill and drain valve, while the gases and vapors in the tank are vented through the vent valve. When propellant loading is complete, the fill and vent valves are disconnected and capped, and the system is ready for operation. Heat from the surrounding structure and/or from an external heat source vaporize the propellant, forcing the vapors (at the propellant vapor pressure corresponding to the temperature of the heated fluid) to pass through the bladder check valve and pressurize the accumulator. The system is initiated and controlled in the same manner as the cold gas system. Detailed Topic 4.4.3.1.

4.4.4 Monopropellant Systems

A typical monopropellant attitude control rocket system is illustrated in Figure 4.4.4. The propellant system is evacuated prior to loading through the propellant fill and drain valve. After the gases and vapors are removed, propellant (e.g., hydrogen peroxide) is pressure fed into the system through the same valve, and gases trapped between the tank bladder and start valve are vented through the gas vent valve. The gas pressurization system is charged in the same manner as described in Detailed Topic 4.4.3.1, and the system is armed by energizing the start valve (squib or solenoid-actuated). This results in regulated pressurization of the propellant tank and propellant flow control valves. Upon command from the guidance system, the appropriate flow control valves are energized, allowing liquid propellant to pass through a decomposition catalyst bed located at the head of the thrust nozzle. The hot gases which are generated by the exothermic decomposition process expand through the thrust nozzles providing reactive thrust.

*The use of liquefied gases includes, but is not limited to, cryogenic fluids. (Cryogenic fluids are defined as fluids which boil at or below –150°C (–238°F) under a pressure of one atmosphere.

4.4.5 Hypergolic Bipropellant Systems

A typical hypergolic bipropellant attitude control rocket system is illustrated in Figure 4.4.5. The pressurant gas (e.g., helium) and propellants (e.g., a hydrazine mixture and nitrogen tetroxide) are loaded in the same manner as described respectively in Detailed Topics 4.4.3.1 and 4.4.4. The system is armed by energizing the start valve, resulting in regulated pressurization of the expulsion tanks and propellant lines up to the bipropellant flow control valves.

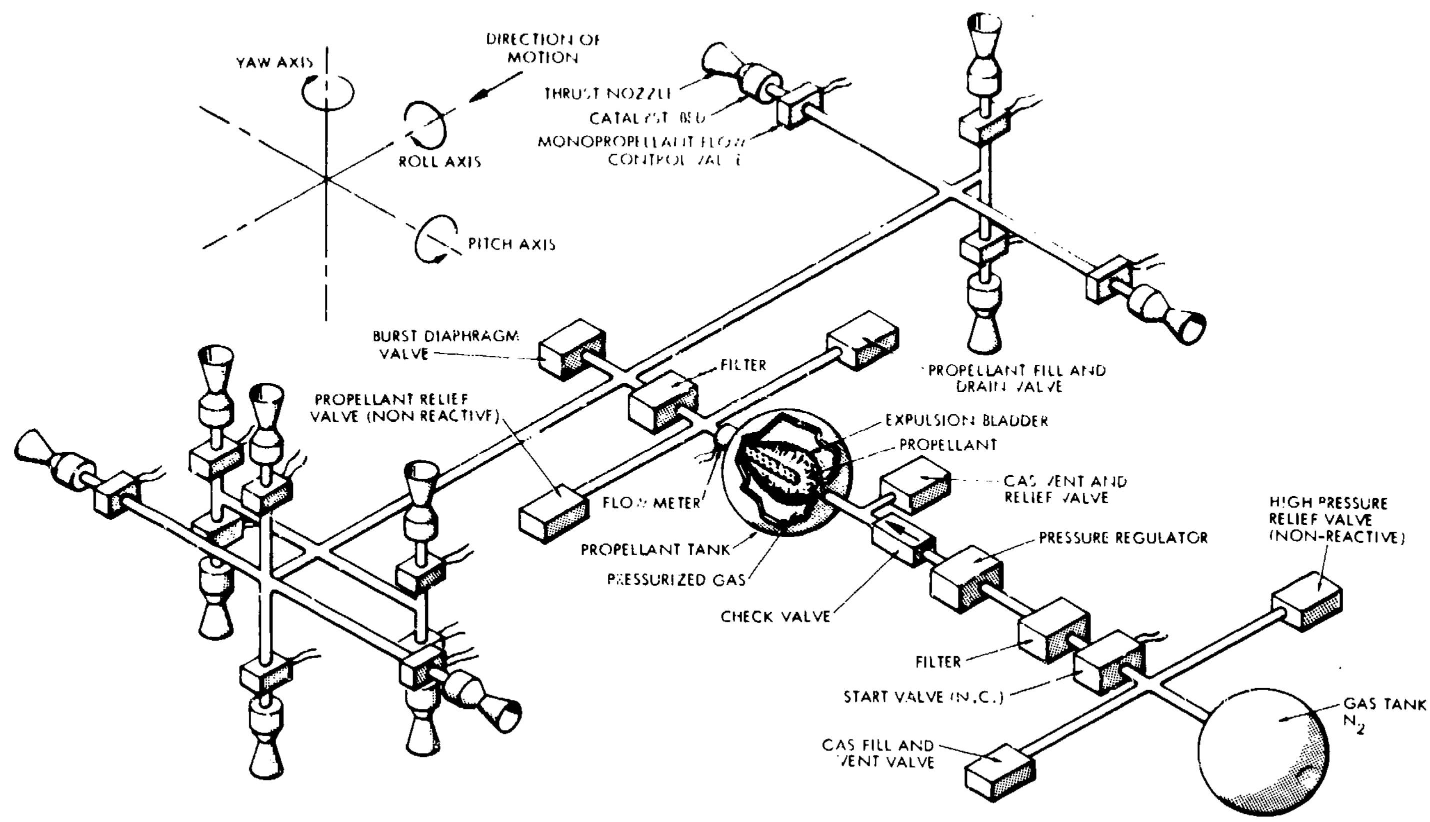

Figure 4.4.4. Monopropellant Attitude Control Rocket System

When the appropriate flow control valves are energized, the propellants are mixed by an injector causing hypergolic ignition. The gaseous combustion products expand in the nozzles causing reactive thrust.

The bipropellant system illustrated utilizes one gas pressure regulator for pressurizing both the fuel and oxidizer tanks. Another possible method of regulating tank pressure is to use two regulators, one for each tank, which would permit the use of sensing lines between the tanks and their corresponding regulators. The system would be heavier than a single regulator system, but would provide closer pressure control.

4.4.6 Solid Propellant Systems

A typical solid propellant attitude control rocket system is illustrated in Figure 4.4.6. The system is activated by an electrical impulse to the igniter which initiates combustion of the solid propellant. Increasing gas pressure caused by the burning propellant ruptures the burst diaphragm, pressurizing the system to the flow control valves. The thrust duration of this system is limited to the burning time of the solid propellant, and problems with extinguishing and reigniting solid propellants limit the use to one shot, short duration applications.

4.4.7 Monopropellant Hot Gas Systems

The monopropellant hot gas attitude control system employs a single hot gas generator to supply gas for each thrust nozzle (Figure 4.4.7). Propellant (e.g., hydrazine) and stored gas (e.g., helium) are loaded in the same manner as described in Detailed Topic 4.4.3.1. System arming is accomplished by energizing the start valve, resulting in pressurization up to the gas generator igniter valve.

Upon command from the guidance system, the gas generator igniter valve is opened supplying propellant to the decomposition chamber. The squib igniter initiates decomposition of the propellant forming hot gases which pressurize the accumulator and the hot gas flow control valves. After the accumulator pressure reaches a predetermined maximum, the gas generator igniter valve is automatically closed and decomposition stopped. Attitude correction is executed by energizing the appropriate hot gas flow control valves causing reactive thrust at the nozzles. When the accumulator pressure falls below a preset minimum (corresponding to self-ignition temperature), the gas generator igniter valve is energized causing propellant flow and hot gas generation, repeating the cycle.

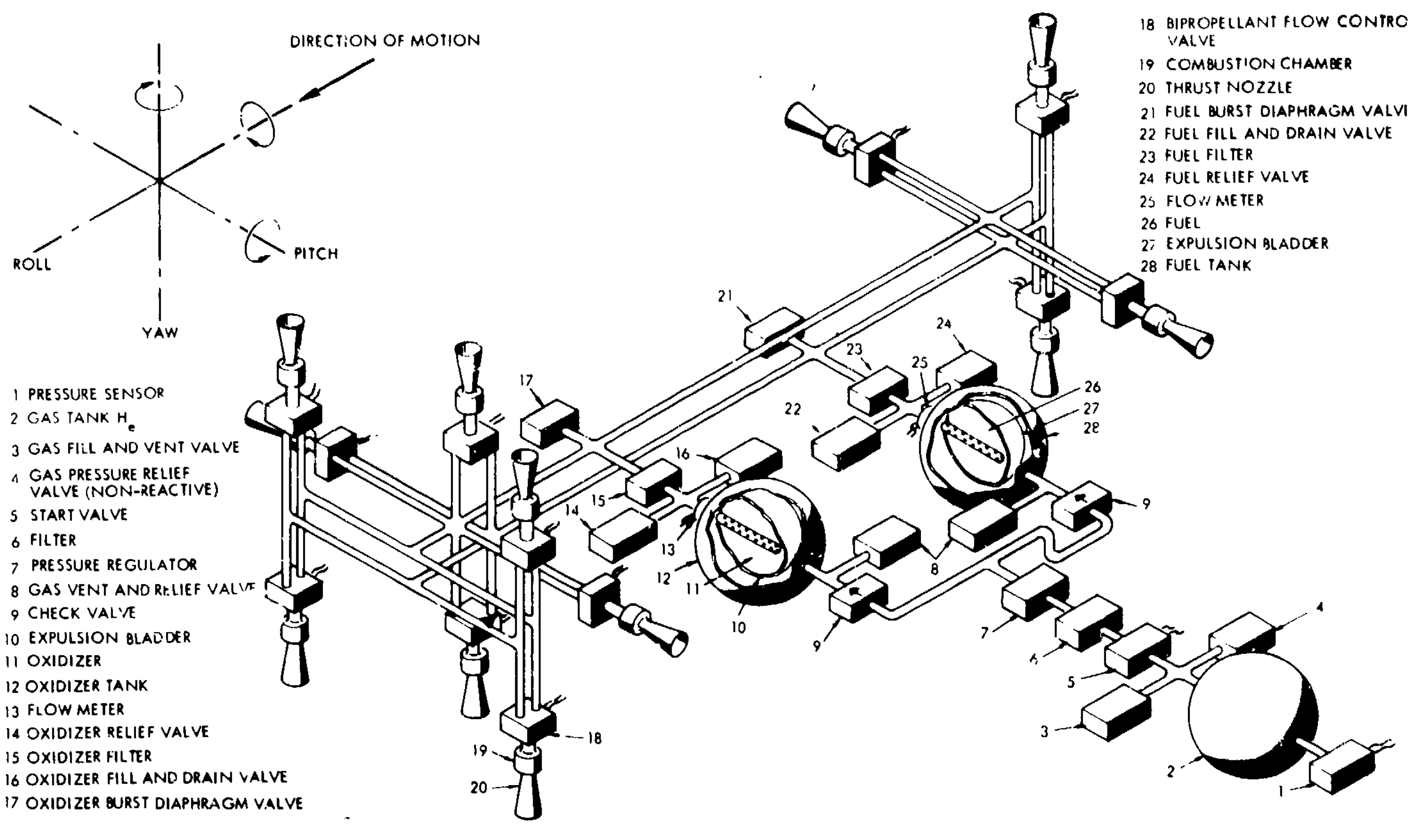

Figure 4.4.5. Hypergolic Bipropellant Attitude Control Rocket System

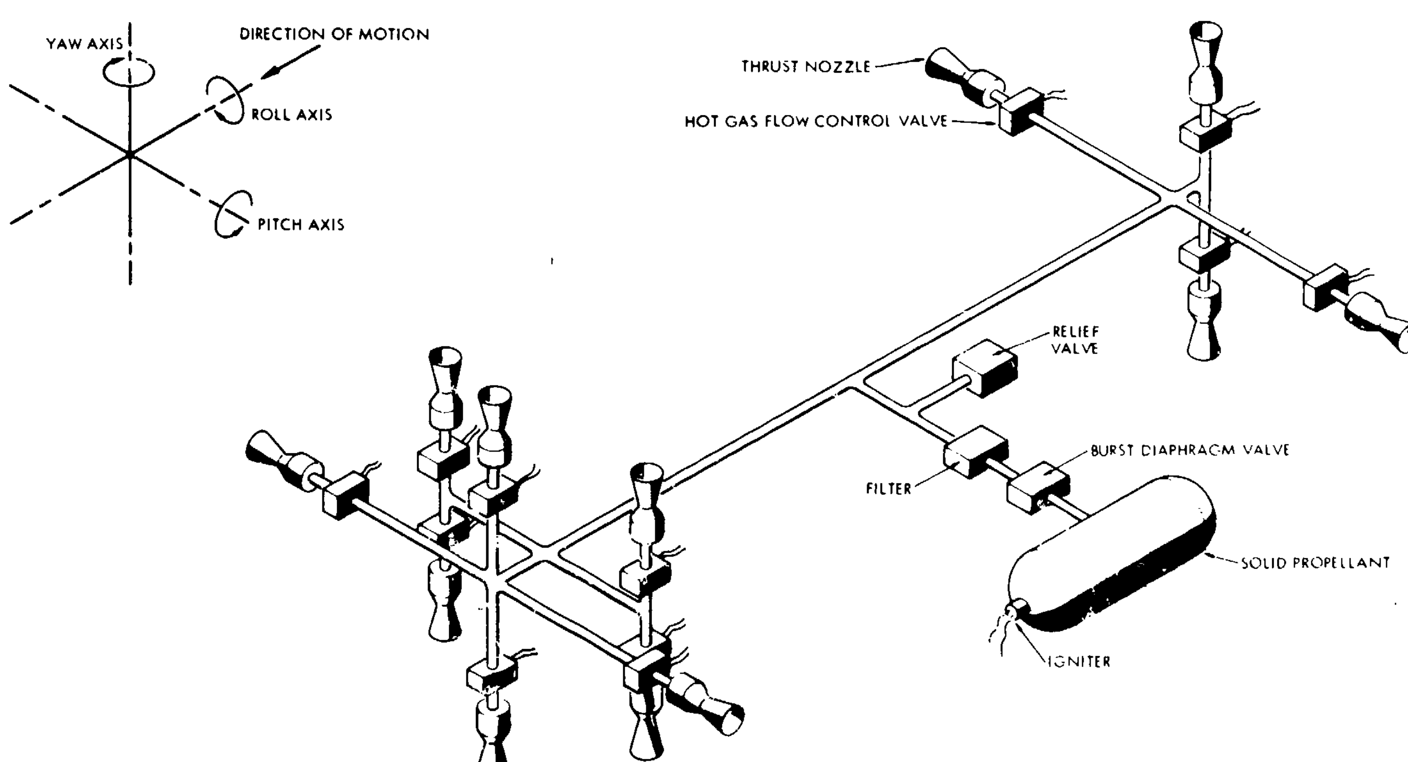

Figure 4.4.6. Solid Propellant Attitude Control Rocket System

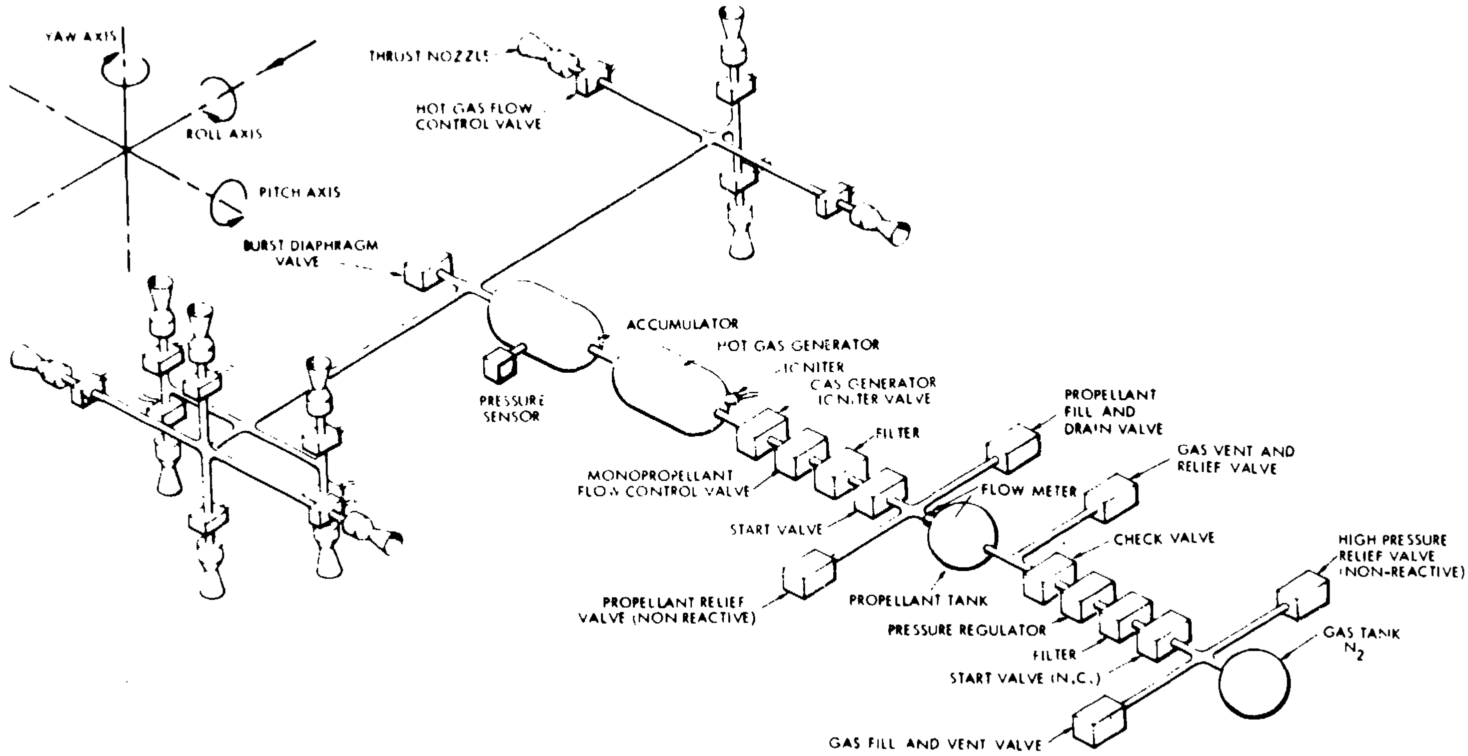

Figure 4.4.7. Monopropellant Hot Gas Attitude Control System

4.4.8 Hypergolic Bipropellant Hot Gas Systems

The hypergolic bipropellant hot gas attitude control rocket system utilizes a single hot gas generator (Figure 4.4.8). Propellant (e.g., N_2O_4 and UDMH) and gas loading and system arming are executed in the same manner as in the bipropellant system described in Sub-Topic 4.4.5.

The bipropellant flow control valve is opened permitting flow to the gas generator. The hypergolic ignition of the propellants results in hot gas generation which pressurizes the system to the hot gas flow control valves. Hot gas generation continues until a predetermined maximum pressure is attained, at which a pressure sensor directs closure of the bipropellant flow control valve.

A pressure sensor in the hot gas accumulator detects when the pressure drops below a predetermined value, whereupon the bipropellant flow control valve is energized permitting bipropellant flow and hot gas generation, repeating the cycle.

REFERENCES

Cold Gas Systems
2-20, 2-60, 27-13, V-56

Monopropellant Systems
2-20, 2-60, 27-13, 47-15, V-56

Bipropellant Systems
2-20, 2-60, V-56

Solid Propellant Hot Gas Systems
27-13, V-56

Monopropellant Hot Gas Systems
2-20, 6-82, 27-13, V-56

Bipropellant Hot Gas Systems
2-20, V-56

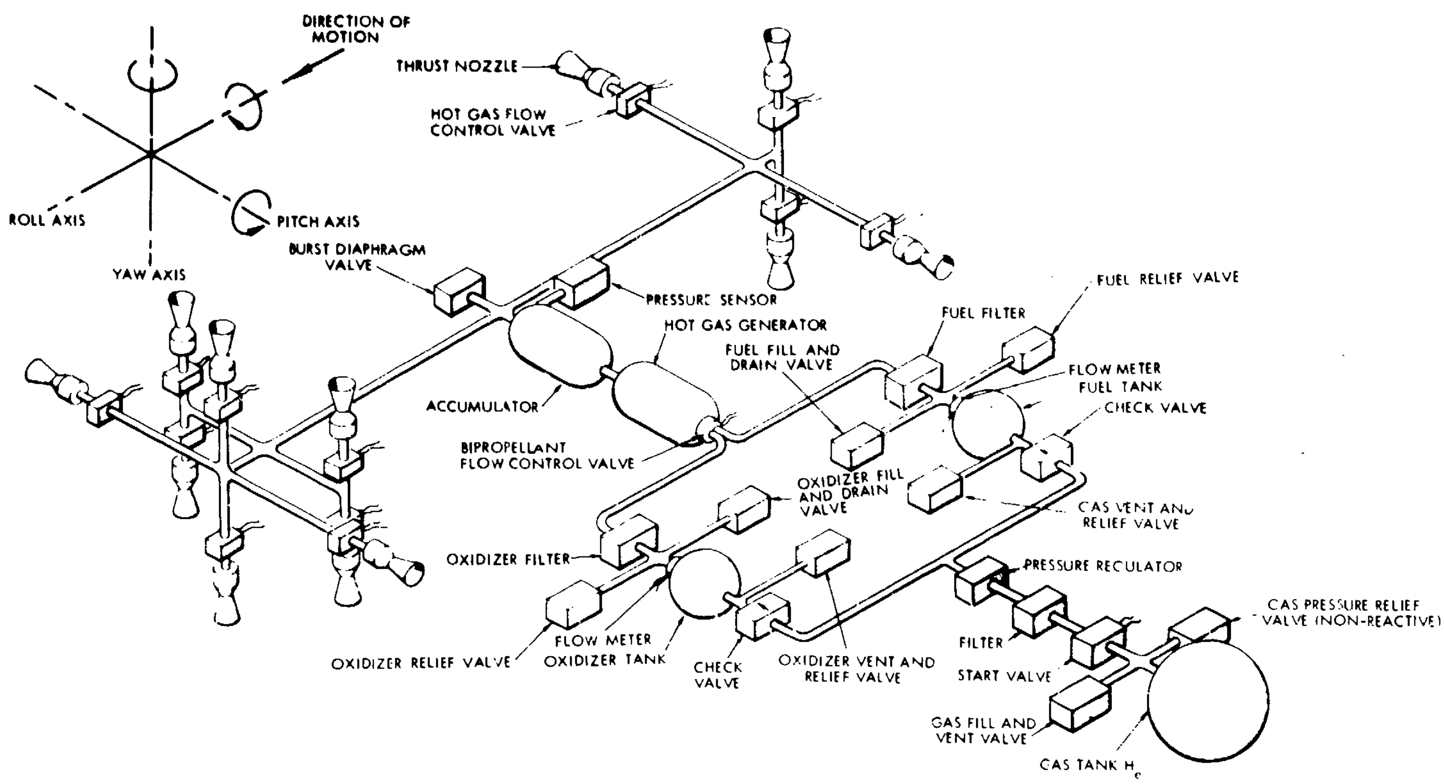

Figure 4.4.8. Hypergolic Bipropellant Hot Gas Attitude Control Rocket System

4.5 SECONDARY INJECTION THRUST VECTOR CONTROL SYSTEMS

4.5.1 Introduction

The purpose of this Sub-Section is to describe typical secondary injection thrust vector control systems for solid and liquid propellant rocket engines in terms of the fluid system operation and the functions of fluid components used in these systems.

4.5.2 General Description of Secondary Injection Thrust Vector Control Systems

A means of thrust vector control is required for rocket engines to provide pitch and yaw control for stabilizing and steering the vehicle during powered flight. Techniques for thrust vectoring include moveable vanes or jetavators located in the rocket exhaust, gimballing of the entire thrust chamber, swivelling the nozzle, and injection of a secondary fluid into the exhaust thereby deflecting the exhaust stream. Secondary injection is playing an increasingly more important role for large rocket engines and solid propellant motors.

Secondary fluid injection thrust vectoring is accomplished by injecting either a gas or liquid into the main exhaust stream as illustrated in Figure 4.5.2, forming a fluid wedge at the point of injection. An oblique shock wave is generated at the point of injection which changes the direction of the exhaust stream, producing a steering force. For a single thrust chamber rocket system a minimum of four injectant points, 90° apart, are required around the periphery of the nozzle. The net vectoring angle is determined by the resultant flow from any two adjacent injectant points.

Injectant flow into the rocket engine nozzle is regulated by the injector flow control valves. By controlling the flow of injectant, the angle and strength of the shock wave is controlled, thus providing a means of regulating the steering force. The injectant flow is controlled using pulse width modulation, pulse frequency modulation, or pulse ratio modulation (a combination of the former two) of an ON-OFF valve, or variable flow control using a proportional flow control valve. The selection of the type of injector flow control (steering force control) depends upon the accuracy and response required to maintain a specific powered flight path. The direction of the steering force is controlled by activating the appropriate injectant flow control valve or valves located around the periphery of the nozzle.

Secondary injection systems are classified according to the state or type of fluid passing through the flow control valve as listed as follows:

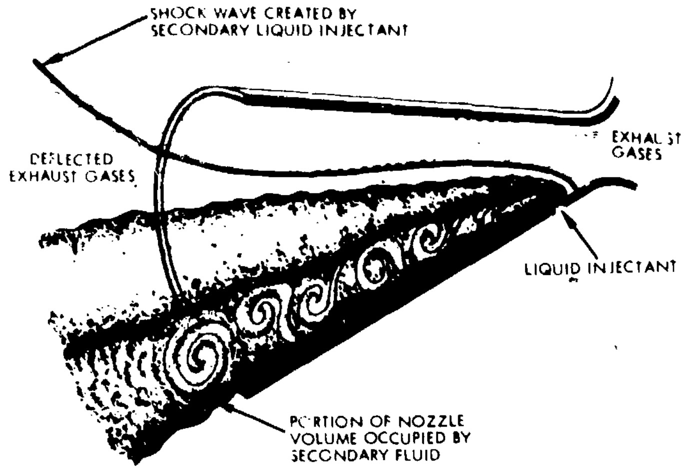

Figure 4.5.2. Rocket Nozzle Exhaust Deflection by Means of Secondary Injection

Gas Injection:

By-pass gas injection
Cold gas injection
Hot gas injection from gas generator

Liquid Injection:

Inert or reactive liquid injection
Reactive bipropellant injection

4.5.3 Gas Injection

4.5.3.1 BY-PASS GAS INJECTION. The by-pass gas injection system (sometimes called thrust chamber bleed) shown in Figure 4.5.3.1 utilizes hot gas from the combustion chamber of the rocket engine. Command signals from the guidance system energize the appropriate injectant flow control valve resulting in injection of hot gas into the exhaust nozzle. The hot gas by-pass system is advantageous in that it is lightweight, highly responsive, and consists of few components. The principal disadvantage of this system is that the hot gas by-pass valve is subjected to gas flow temperatures above 6000°F.

4.5.3.2 COLD GAS INJECTION. A cold gas injection thrust vector control system is illustrated in Figure 4.5.3.2. The gas tank is charged with high pressure gas through the gas fill valve until rated pressure is attained. Pressurization is then discontinued, and the fill valve disconnected.

A signal from the guidance system energizes the start valve (explosive or solenoid actuated), resulting in regulated pressurization of the gas injector valve. The injector valves are then energized on command by the guidance system and inject gas into the main exhaust stream of the rocket engine. As the component requirements of the cold gas injection system are basically the same as those of the stored gas attitude control system, the reader is referred to Detailed Topic 4.4.3.1 for additional discussions of component requirements in this system.

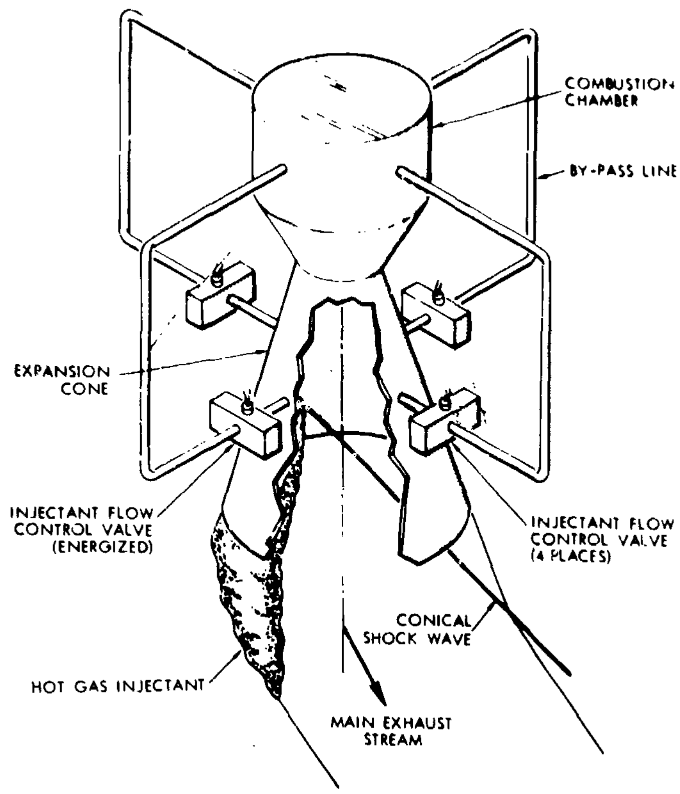

Figure 4.5.3.1. Bypass Gas Injection System (Thrust Chamber Bleed)

4.5.3.3 HOT GAS INJECTION FROM GAS GENERATOR. A hot gas injection system utilizing a gas generator is illustrated in Figure 4.5.3.3. The hot gases generated are the products of chemical reactions of monopropellants, bipropellants, or solid propellants, and the gas temperature may range between 1000 and 6000°F. System arming is accomplished by energizing the start valve in the monopropellant or bipropellant systems or by energizing the pyrotechnic igniter in the solid propellant gas generator.

Upon command from the guidance system, the appropriate hot gas injector valves are energized, resulting in gas injection into the main exhaust stream of the rocket engine. The filter downstream of the gas generator prevents particulate matter either from the gas generator or the start valve from impairing the operation of the relief valve or injectant control valves. The relief valve keeps system pressure from exceeding a preset value.

4.5.4 Liquid Injection

4.5.4.1 INERT OR REACTIVE LIQUID INJECTION. A secondary injection system utilizing an inert or reactive liquid (e.g., Freon, N_2O_4) is illustrated in Figure 4.5.4.1.

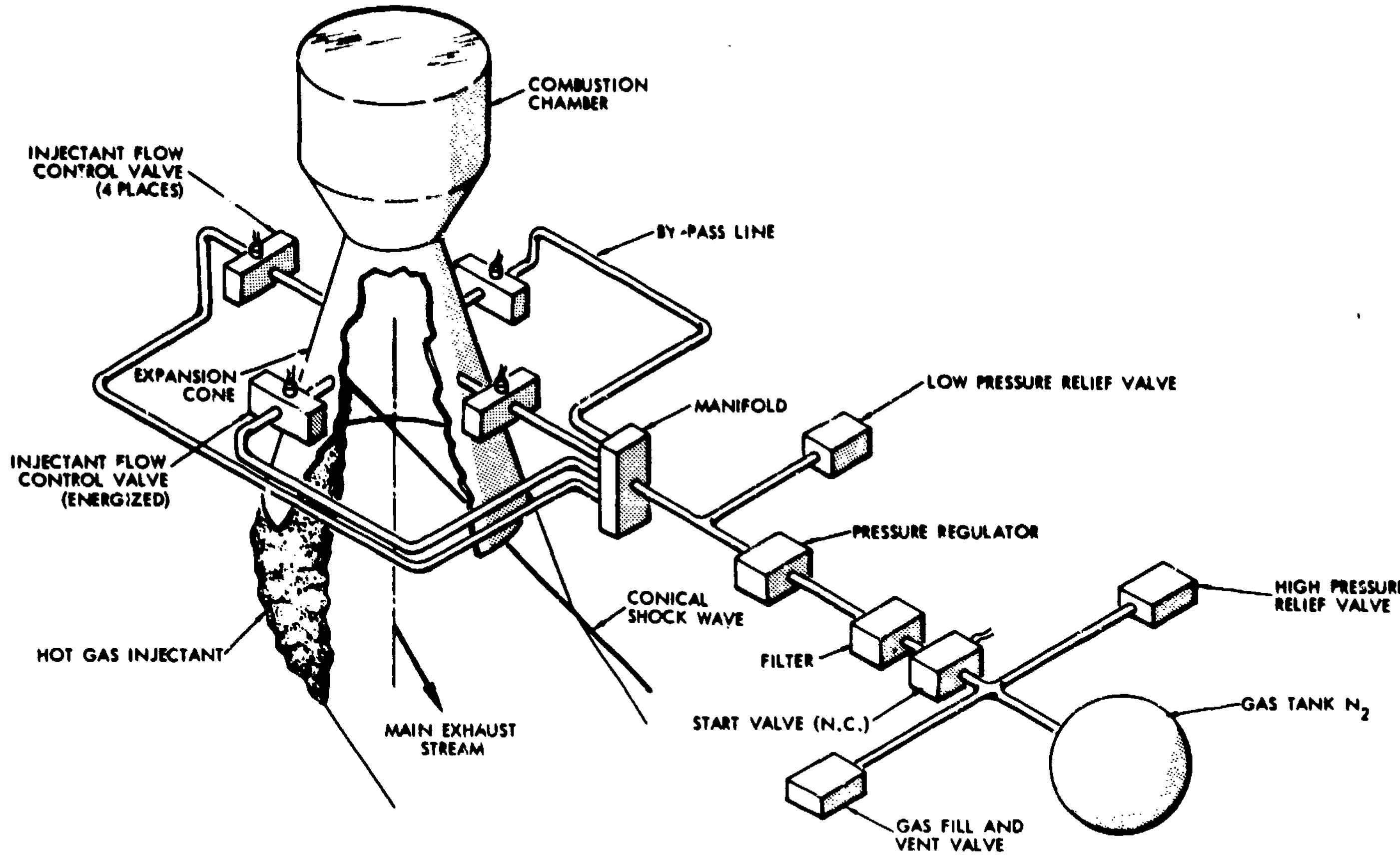

Figure 4.5.3.2. Cold Gas Injection Thrust Vector Control System

The injectant is loaded through the fill and drain valve, while the gases and vapors in the injectant storage tank are vented through the vent valve. When an expulsion bladder is used in the injectant tank, the system must be evacuated prior to filling the injectant tank. Gases trapped between the tank bladder and start valve are vented through the gas vent valve, the gas pressurization system is charged, and the system is ready for operation. The start valve (squib or solenoid-actuated) is energized, and the pressure regulator provides a constant pressure to the injectant tank for injectant expulsion.

The injector valves are then energized on command by the guidance system, resulting in liquid injection into the main exhaust stream of the rocket engine. Reactive liquid injectants react chemically with the combustion gases and the reaction products expand through the nozzle. Inert injectants entering the combustion gas stream expand from the heat of the combustion gases and form a shock wave. A method using reactive liquid with liquid propellant engines is to bypass oxidizer from the engine supply lines to the injector valve. This complicates propellant utilization control, but permits a weight savings by elimination of the secondary system.

4.5.4.2 REACTIVE BIPROPELLANT INJECTION. A reactive bipropellant liquid injection system is represented in the schematic shown in Figure 4.5.4.2. Reactive propellants (e.g., N_2O_4 and UDMH) are loaded through the fill and drain valves, while gases and vapors are vented through the vent and relief valves. Pressurizing gas (e.g., He, N_2) is loaded in the gas tank through the gas fill and vent valve. Inadvertent overpressure in the gas tank either during filling or resulting from post fill thermal expansion is relieved by the gas pressure relief valve. The system is armed by energizing the start valve. A constant pressure is provided by the pressure regulator for expelling propellants.

Signals from the guidance system energize the appropriate bipropellant injector valves, injecting fuel and oxidizer into the main exhaust stream. Filters are located upstream of the pressure regulator and the flow control valves to prevent particulate contamination from entering these components.

Liquid propellant engines can be adapted for reactive bipropellant liquid injection by diverting fuel and oxidizer from the engine supply lines to the bipropellant injector

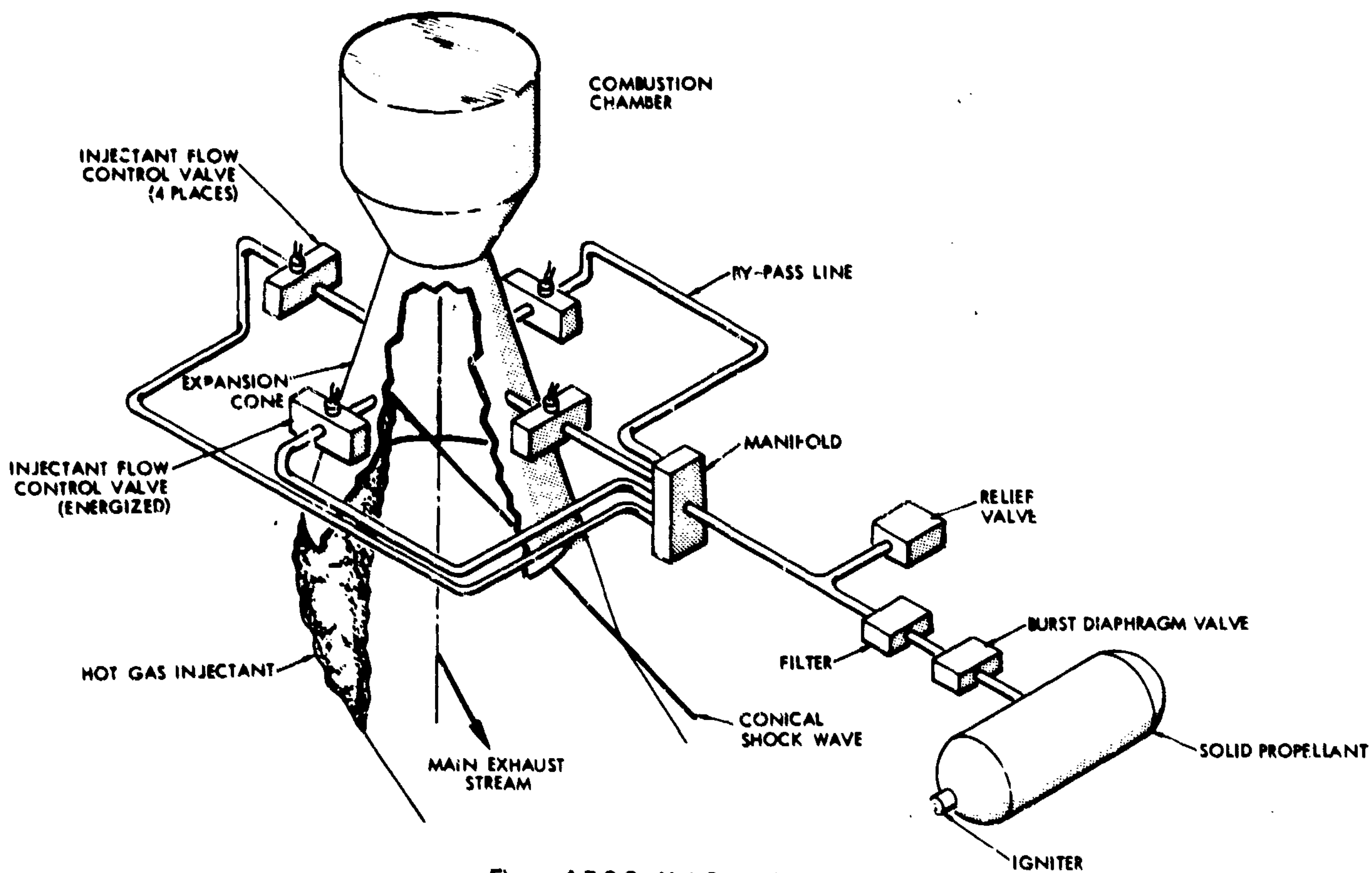

Figure 4.5.3.3. Hot Gas Injection System

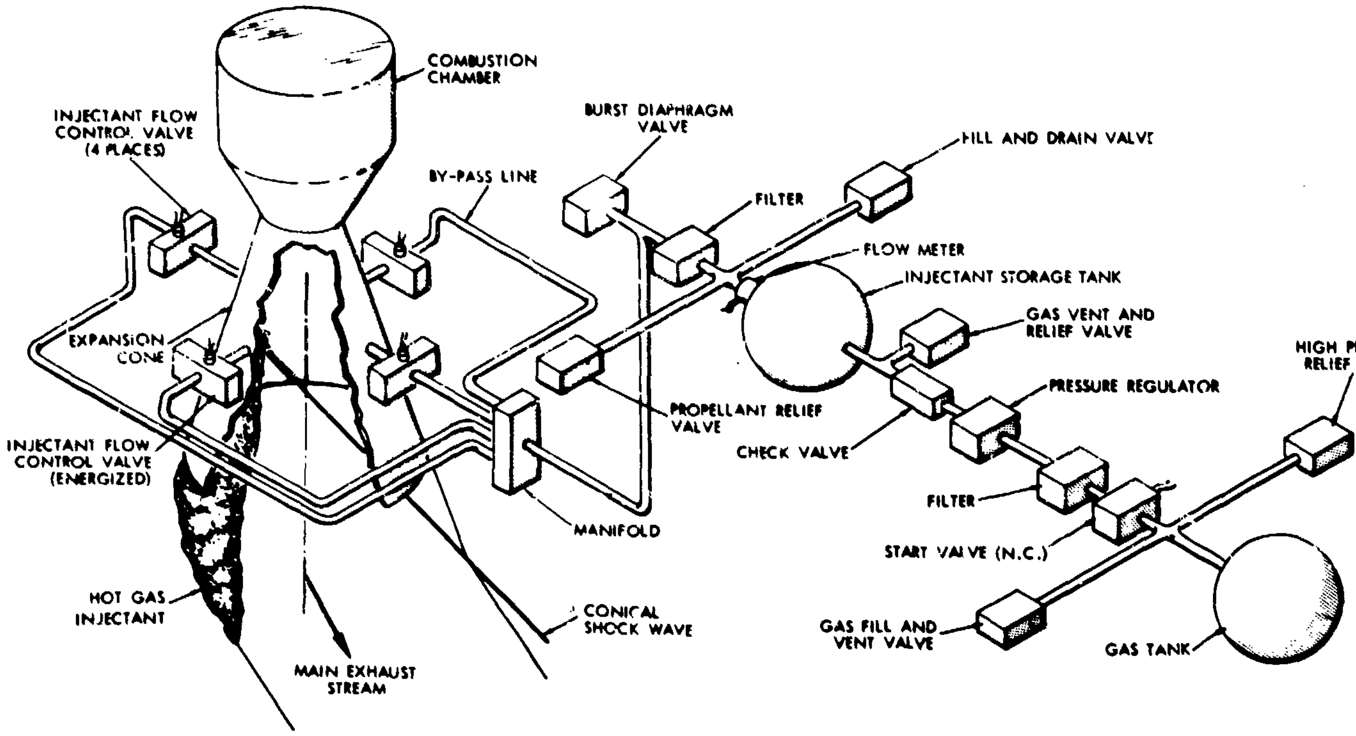

Figure 4.5.4.1. Secondary Injection Syst[illegible] U[illegible]izing An Inert Liquid

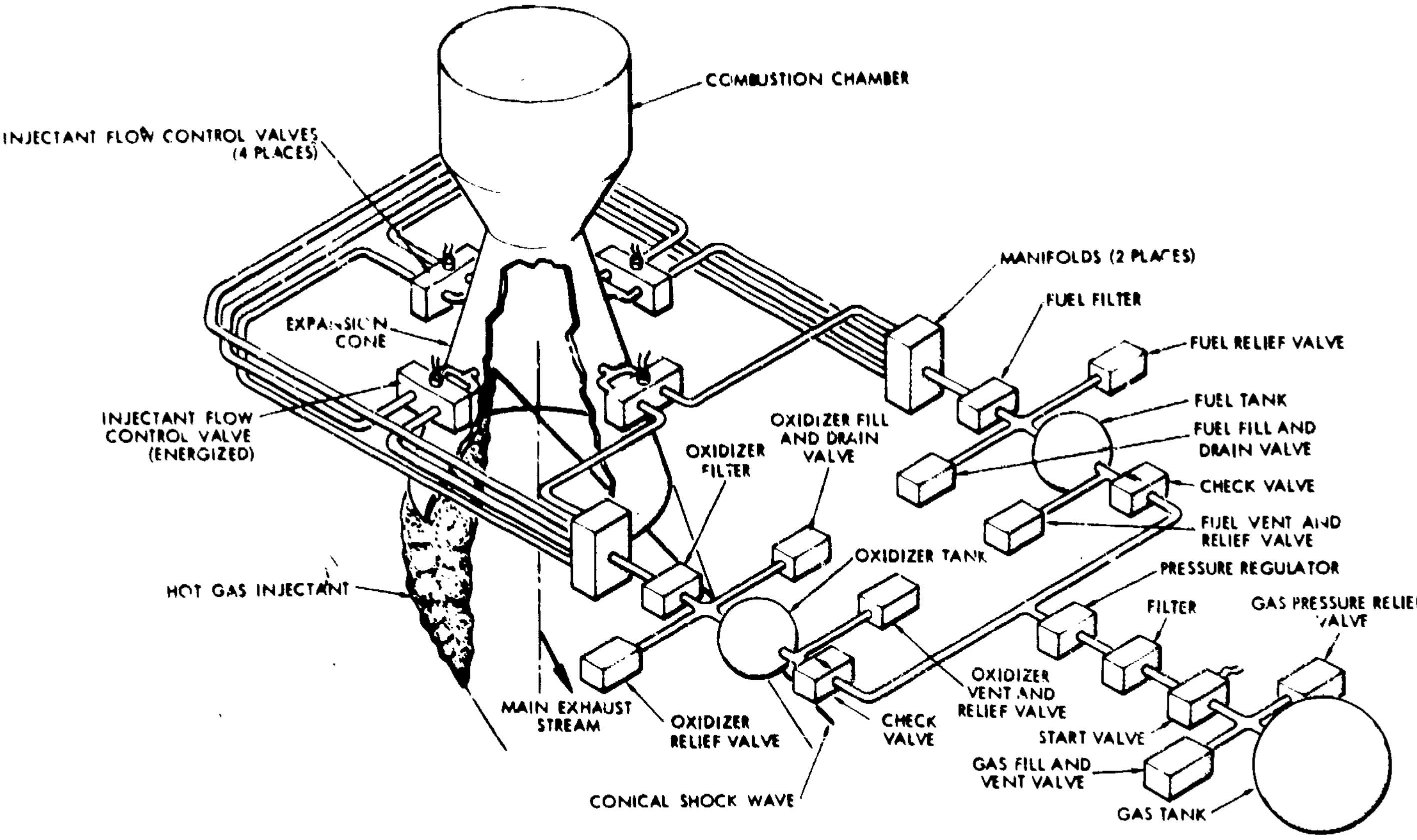

Figure 4.5.4.2. Reactive Bipropellant Liquid Injection System

valves, thus eliminating a complete separate secondary injection system.

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Hot Gas By-pass Systems
6-20, 30-1, 47-16, 308-1

Cold Gas Systems
8-3, 47-16

Hot Gas Injection from Gas Generator Systems
8-3, 30-1, 47-1

Inert or Reactive Liquid Injection
6-20, 8-3, 30-1, 47-16, 308-1

Reactive Bipropellant Injection
47-13, 47-16, 308-1

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ILLUSTRATIONS (Continued)

ILLUSTRATIONS (Continued)

ILLUSTRATIONS (Continued)

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Table

5.1 INTRODUCTION

The purpose of this section is to discuss basic fluid components in terms of component function, design types, performance characteristics, applications, and limitations. The fluid components which will be covered can be divided into *valves, fluid couplings and connectors, filters,* and *sensing devices.* The range of fluid system applications for which components are considered includes ground support, flight, hydraulic, pneumatic, cryogenic, and vacuum systems.

Valves, which are the fluid components given major emphasis in the handbook, are defined as any device which stops, starts, or otherwise regulates the flow of fluid by means of a movable valving element which opens or obstructs a flow passage. *Valve* types are discussed by function under the following categories: shutoff valves, control valves, pressure regulators, relief valves, check valves, multiple passage valves, servo valves, and explosive valves. Each functional category is further categorized by design type as — in the case of shutoff valves, for instance — poppet, butterfly, gate, spool, etc. *Fluid couplings and connectors* discussed are fittings, flexible and rotary joints, and quick disconnects. *Sensing devices* include flowmeters, and pressure switches, as well as pressure, temperature level, and position-sensing instruments. A glossary of fluid component terms is given at the end of Section 5.18.

5.2 SHUTOFF VALVES

5.2.1 Introduction

This Sub-Section describes general design and selection considerations for shutoff valves and discusses the operation, performance characteristics, and applications of specific shutoff valve types.

The term *shutoff valve* applies to valves that are always in either the ON or OFF position, in contrast to control valves which take intermediate positions to modulate flow between no flow and full flow. Several shutoff valve types can be used to modulate flow, but these valves are more properly called control valves, and are discussed in Sub-Section 5.3. The term *shutoff valve* is used synonymously with *stop valve* and *block valve*. Shutoff valves are commonly designated by service or system functions such as: fill valves, drain valves, dump valves, vent valves, prevalves, and start valves. Shutoff valves are furnished in many configurations of valving elements (closures), body styles, and actuators.

5.2.2 Design and Selection Parameters

Parameters that must be considered in the selection of a particular shutoff valve configuration are discussed in the following Detailed Topics.

5.2.2.1 FLOW MEDIUM. The choice of a shutoff valve may depend on whether the fluid medium is a liquid or gas or both (two-phase flow). Certain types of seals cannot be used interchangeably in liquids or gases; the relatively high velocities of gas flow may require special seal design to avoid exposing easily damaged seal materials to the erosive effect of the gas stream. Special sealing problems may be encountered when handling the lighter gases such as helium and hydrogen, particularly at high pressures.

Compatibility of the seals, metals, plating, and other materials with the fluid medium must be assured. If the medium has a relatively high freezing point or will be used in an area where freezing may occur, precautions must be taken to prevent damage to the valve either from expansion of the fluid during the phase change, or from structural damage resulting from high loads incurred in shearing iced surfaces.

5.2.2.2 PRESSURE DROP AND FLOW CAPACITY. Pressure drop and flow capacity are interdependent, in that when one parameter is defined, the other is also fixed. When specifying flow capacity or pressure drop, it is important that the actual operating conditions of pressure, temperature, flow medium, and inlet and outlet conditions of the system be accurately delineated. Ideally, a shutoff valve would have zero pressure drop when open, in order to reduce system losses to a minimum. There are several conventions commonly used in defining flow capacity, which are described in Sub-Topic 3.8.4. Comparative flow characteristics of various shutoff valves are given in Tables 3.8.4.2c and 3.9.5.2 of "Fluid Mechanics," Section 3.0.

5.2.2.3 OPERATING TEMPERATURE. Considerations of operating temperature must include the temperature of the flowing medium as well as the ambient temperature. In some cases, the medium temperature will govern and, in other cases, the ambient temperature will govern. Materials must be selected to meet the worst conditions. Considerations should also be given to the duty cycle, since a valve which has significant mass and corresponding heat sink capability is capable of operating for short periods of time with a flowing medium which may be either extremely cold or extremely hot. Temperature gradients should be considered in terms of possible thermal distortion and differential expansion.

Choice of lubricants and seals must be carefully considered with respect to temperature. Excessively high temperatures can cause loss of the lubricant and failure of the seals. Excessively low temperatures can cause the lubricant to congeal to a point where operation is sluggish or impossible. Dynamic seals, especially O-rings, may experience severe damage when subjected to temperatures beyond their design limit.

5.2.2.4 OPERATING PRESSURE. Obviously the valve housing must be capable of withstanding the maximum operating pressures to which it will be subjected. Flow transients (water hammer) can cause line pressure to rise several times in excess of normal operating pressure. In cases where critical tolerances exist, care must be taken to assure that the strain resulting from the pressure does not alter the tolerances to a degree that would affect the operation. In high pressure applications, attention must be paid to fittings and connectors as well as packings and various gaskets. If the valve is to be subjected to many cycles of pressure applications, consideration must be given to possible fatigue failure.

5.2.2.5 LEAKAGE REQUIREMENTS. Valve leakage is considered as both internal leakage (leakage in the flow path direction) and external leakage (leakage to the external environment in a direction other than the normal direction of flow.) It is extremely important that realistic leakage requirements for the application be determined. Excessively stringent requirements will increase the cost of the valve unnecessarily, because of the added cost in the design and fabrication of seals and closure. Four factors affecting leakage requirements for a valve are:

a) Loss of pressure or propellants. Loss of fluids through the valve must be limited to a value that will prevent system failure due to premature depletion of fluid.
b) Damage to the system. Leakage must be limited to that which will preclude system damage, as severe corrosion or fire are possible results of propellant leakage.
c) Danger to personnel. If the fluids are toxic, protection against inhalation and exposure must be given to personnel.
d) Interference with experiments. A special consideration for spacecraft is the problem of pressurants or propellants enveloping the vehicles with a gas cloud which could interfere with sampling of planetary atmospheres.

Proper evaluation of the above factors will assist in determining a reasonable leakage requirement for a particular valve application.

5.2.2.6 POWER REQUIREMENTS. Shutoff valves may be manually operated or may be operated with any of the conventional actuators, such as solenoids, motors, and hydraulic or pneumatic piston-cylinder combinations. The power required depends upon the frequency of operation, the forces involved, the response required of the valve and, in the case of solenoids and motors, the temperature. With solenoids and motors it is wise to specify the actual operating temperature range as closely as possible, because a smaller solenoid can be constructed for a given force requirement if the operating temperature is low. Power consumption may be reduced by employing such devices as latching solenoids, which require only a brief impulse to stroke the valve and no additional power to hold it open. Available pressurized fluids, either hydraulic or gaseous, that may be overboarded without penalty to the over-all performance of the system should be considered as a source or actuation power.

5.2.2.7 ACTUATORS AND ACTUATION FORCES. Shutoff valves may be actuated by handwheels, levers, solenoids, motors, diaphragms, or cylinders. The final choice is dependent upon many variables such as the forces involved, power available, remote operating requirements, response, available space, and weight. The forces to be overcome include pressure forces, flow forces, friction forces, and inertial forces of moving parts. Small valves, especially if pressure balanced, are easily and conveniently operated directly by a solenoid. As the size of the valve increases, the power requirements and the weight of the solenoid increase very rapidly. The stroke will also increase and, again, a trade-off is reached where a decision must be made to use either a different type of actuator entirely, or to design a pilot into the valve. Actuators are discussed in detail in Sub-Section 6.9.

5.2.2.8 ACTUATION TIME (RESPONSE). The actuation time of a valve is affected by the following variables: mass of the moving parts, pressure differential, friction forces, travel of valving element, and force available from the actuator.

Improvement in response characteristics can be achieved by either increasing the actuator force or decreasing the actuator load. Important tradeoffs exist between response and valve weight, and cost and operating life. High response increases cost and weight, due to increased actuator size and power requirements. A valve having a high response and, consequently, a high poppet velocity will incur large seat stresses which tend to reduce the life expectancy of the unit.

5.2.2.9 CONTAMINATION SENSITIVITY. Cleanliness of the system and of the flowing medium can have a direct effect upon the operation and life of a valve. Fluids containing excessive contamination can clog small control orifices and cause excessive leakage, especially in metal-to-metal seated valves. Particulate matter in gaseous media, especially in the lighter gases such as hydrogen or helium, can be extremely destructive to internal parts, particularly seats, because of the very high velocity that can be attained under sonic conditions. If a valve with small orifices is to be used in a system where it is known that the flowing medium is relatively dirty, consideration should be given to installing an integral filter upstream of the orifices. Water in the fluid media can cause malfunctioning in components subjected to freezing temperatures. Ice and solid CO_2 particles in cryogenic fluids may pose problems that are difficult to diagnose, especially in the case of CO_2, since the gas will disappear without a trace after the system has been warmed up and dismantled for inspection. It should also be recognized that valves in cryogenic service that are vented to atmosphere (e.g., vent valves) may accumulate large quantities of ice on the downstream side of the valve. If such an accumulation could be troublesome, preventive measures such as a small downstream volume purged with dry gas, may be required.

5.2.2.10 MAINTENANCE. In considering maintenance, requirements for special tools should be avoided, if at all possible. Critical subcomponents such as seats and packing should be accessible and easily replaced, preferably without removing the valve from the line. If it should be necessary to service a valve in the field, great care should be exercised to insure that contamination from the work area is not introduced into the valve or system. Requirements for lubrication and adjustments should be minimized.

5.2.2.11 AVAILABILITY. If a valve is to be purchased, consideration must be given as to the availability of the particular configuration chosen. Common patterns and types such as globe valves, butterfly valves, and solenoid valves are readily available. Ball valves are relatively less available, and the supply of other types, such as pinch or diaphragm valves, may be severely limited.

5.2.2.12 WEIGHT AND SIZE. Weight and size are prime considerations for all airborne and space components, and often will govern the type of component selected for a given application. For example, a full flow ball valve in line sizes of one inch or less will have better flow characteristics than a butterfly valve of the same size. However, in large sizes, (10 inches or more) the flow characteristics are similar, but the weight and size of the ball valve will greatly exceed that of the butterfly. Similarly, the actuator choice must be evaluated individually, as the choice of type will vary with line size. Solenoids are suitable for small size components, but become relatively heavy as the size increases, making pneumatic or hydraulic actuation a more attractive choice.

The extent to which weight can be reduced on a given component is related in part to the allowable cost of the unit. If weight is of extreme importance, extra machining steps and use of high strength-to-weight ratio materials such as titanium or beryllium may be justified.

5.2.2.13 COST. A great many factors affect the final cost of a valve. Some of these factors to be considered are: operating pressure, temperature, flow rate, differential pressure, response, leakage, flow medium, weight, type of actuation, reliability, life requirements, test and inspection requirements, and availability. At one extreme is the small, hand-operated shutoff valve for use on a low air pressure ground system, available as an off-the-shelf item from numerous vendors. At the other extreme is a flight-weight, remote-operated shutoff valve controlling a toxic, corrosive fluid on a space vehicle. While cost should be considered secondary to performance on missile and spacecraft systems, judgement must be used in the selection of components to avoid waste through extreme over or under-design.

5.2.2.14. OPERATING LIFE. Operating life can vary from one cycle for a rupture diaphragm valve to many hundreds of thousands of cycles for a well-designed solenoid valve. Factors and combinations of factors affecting the life expectancy of a valve are: operating temperature, operating pressure, rate of cycling, number of cycles, loads on members, materials of construction, type of lubricant, and exposure to space environments.

Multiple cycling under high pressure operating temperature decreases the life of the valve. The rate of cycling may be important if the temperature rise, as a result of the operation, becomes significant. As with any piece of mechanical equipment, the higher the structural loads the shorter the life.

Exposure to space environments will tend to shorten the life of valves because of the deleterious effects of vacuum and radiation on seals and lubricants. The degree to which this occurs depends upon the design of the valve and the location and length of time in the environment.

5.2.3 Ball Valves

5.2.3.1 DESCRIPTION AND OPERATION. The ball valve is essentially a ported sphere positioned in a housing such that 90 rotation of the sphere (ball) changes the valve from open to closed positions. Two common ball valve design configurations are the fixed ball design and the floating ball design. In the fixed ball design (Figure 5.2.3.1a) the ball is supported in fixed bearings and the seal is spring-loaded against the ball. The seal is mounted on the upstream side of the ball and is usually designed to seal in one direction only. Bi-directional sealing can be obtained by spring-loading a similar seal on the other half of the ball. In the floating ball design (Figure 5.2.3.1b) the ball is supported by fixed seals and the seating force is provided by fluid pressure forcing the ball against the seal. In this type of design, with seals both upstream and downstream of the ball, the valve seals equally well with flow in either direction. The floating ball, which is the simplest design, is restricted to lower pressure applications, particularly in large sizes due to the high seal loading and resulting high actuation forces. In addition to the usual metals, ball valve body materials include plastics such as unplasticized polyvinylchloride (UPVC). The ball material may be plastic or metallic. In metallic balls, usually a highly polished, hard chrome surface is used. In most ball valves, the ball conforms to a true sphere within 0.0005 inch.

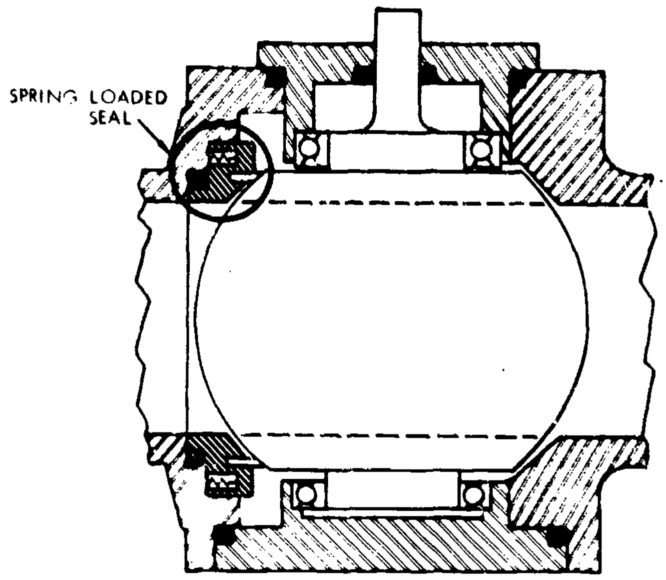

Figure 5.2.3.1a. Full Flow Ball Valve With Fixed Ball

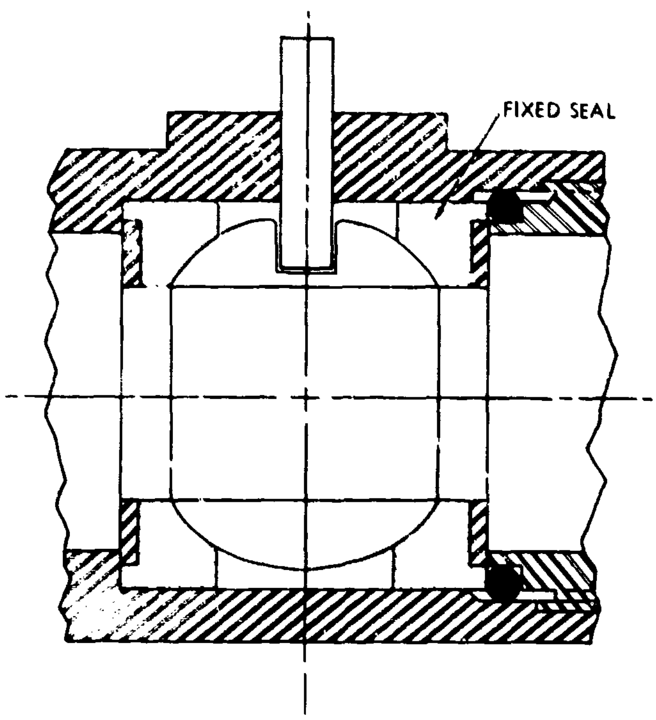

Figure 5.2.3.1b. Reduced Port Ball Valve With Floating Ball

Seal materials in ball valves are usually resilient materials, either plastics or elastomers, although hard seats made of graphite and metal are produced. Teflon, due to its high corrosion resistance and extremely low friction, is the most commonly used seal material. When hard seals are used, the valve is usually designed with cam action which lifts the ball off the seal during the first few degrees of rotation, to reduce friction and wear. Ball valves may be obtained with top entry, which permits disassembly for maintenance without removing the valve from the line.

5.2.3.2 PERFORMANCE CHARACTERISTICS. The outstanding characteristic of ball valves is the low pressure drop, which is equal to the drop experienced in an equivalent length of line provided that the valve is of the full flow design, i.e., having flow area equal to the connecting line inside diameter (Figure 5.2.3.1a). Reduced port ball valves are in wide use, although the advantage of low pressure drop with this type of valve is not fully achieved (Figure 5.2.3.1b). Leakage control in a ball valve is excellent, and the valve can be designed to seal equally well with the flow in either direction.

Since the ball valve cannot be pressure balanced, actuating forces are relatively high because of seal and/or bearing friction. The actuating torque characteristics for ball valves vary with angular position in a manner similar to butterfly valves. A fluid dynamic torque acts in a direction to close the valve, increasing as the valve is opened until a maximum torque value is reached between 60° and 80° open, then decreasing rapidly to zero as the full 90° open position is reached. Torque characteristics of butterfly valves are discussed in Sub-Section 5.3, "Control Valves." As 90° rotation is required to actuate the valve, actuators with relatively long stroke capabilities are required; however, when used as a hand valve, the ball valve has the advantage of rapid opening (90° turn) compared to a screw thread operated valve such as a gate or globe valve. It should be noted that in certain liquid system applications, rapid closing of a hand valve could create a water hammer problem.

If the valve is to be used for throttling, seals must be of such a design that they will not be washed out or extruded under the conditions of high pressure drop that occur in the near-closed position. Throttling can result in damage to the seal and to the polished surface of the ball if the flow medium is contaminated, especially if the medium is a gas with a high sonic velocity, such as helium or hydrogen. In the fully open or fully closed position, neither the ball's seating surface nor the seat are exposed to the flowing media, thus protecting these critical surfaces from high velocity flowing media. Another advantage of the ball valve used as a shutoff valve is the insensitivity to contamination resulting from the self-wiping action that prevents solid matter from wedging between the ball and seats.

5.2.3.3 APPLICATIONS AND LIMITATIONS. The popularity of ball valves has been increasing sharply in recent years as new refinements are added and as the valve becomes more publicized. It is ideally suited to applications where high pressure, low pressure drop, and small size are of prime importance. However, it should be noted that other requirements, such as high response, could negate the choice, since the ball valve is inherently slow in response because of the high seat friction and the large travel required. In double-sealed ball valves, consideration must be given to the possibility of trapping fluid in the ball cavity when the ball is moved to the closed position. Severe damage could result if venting provisions are not incorporated, due to pressure buildup from vaporized cryogenic fluids, from expansion forces that would result from freezing liquids such as water, or from actual explosion that might result when thermally unstable fluids are used. Venting is commonly provided in ball valves by designing the upstream seal such that it relieves if pressurized in the reverse direction.

For large line sizes, the ball valve becomes relatively heavy and requires large actuation forces; hence, its use in sizes above three inches is limited for airborne or space applications. Lightweight modifications of ball valve designs which utilize a portion of a ball as the closure, have been used successfully for flight systems in sizes up to nearly 12 inches. One such design is known as a *visor valve*.

Normally ball valves are not used in throttling service because of the possible damage to either the seal or the ball. Teflon is a widely used seal material but has a marked tendency to cold flow under pressure. For this reason it is often undesirable to leave a ball valve in an intermediate position, as the seal may be damaged when motion of the ball is later effected. If it is possible for moisture to be trapped in the cavity of the ball when closed, sufficient actuation forces must be available to open the valve if the moisture is allowed to freeze. The smooth, unobstructed flow path of the ball valve makes it ideally suited for the handling of very viscous fluids, slurries, gels, and granular solids.

5.2.4 Butterfly Valves

5.2.4.1 DESCRIPTION AND OPERATION. The butterfly valve consists essentially of a disc supported on a shaft such that it can be rotated in a housing. The disc can be attached to the shaft and mounted in the housing in several ways. The simpliest butterfly valve is a flat, circular disc which is mounted to the surface of a shaft perpendicular to the duct axis and is offset from the center of rotation. This design is not suited to shutoff applications, since complete sealing cannot be achieved (the disc diameter must be smaller than the duct diameter to permit rotation). However, for certain control valve applications where tight shutoff is not a requirement, this simple design is useful. A refinement of this design is illustrated in Figure 5.2.4.1a, where the disc pivots about an axis through its center. Because of its characteristic shape, this type of disc is sometimes referred to as a clam shell Although not shown in the illustration, it is common to design butterfly valves such that the disc is not positioned perpendicular to the duct in the closed position. The disc must therefore be elliptical, with a minor axis equal to the duct diameter and the major axis greater than the duct diameter by the relationship:

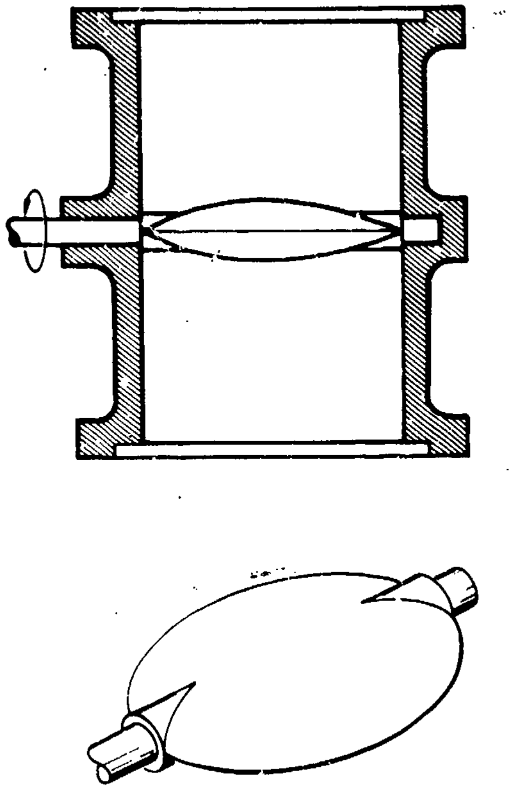

Figure 5.2.4.1a. Butterfly Valve (Clam Shell Design)

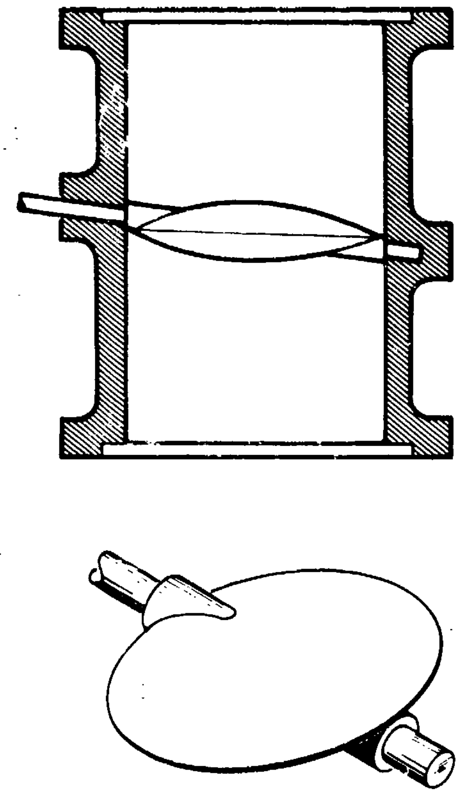

Figure 5.2.4.1b. Butterfly Valve (Canted Shaft)

$$\text{major axis} = D/\cos \alpha_o \qquad \text{(Eq. 5.2.4.1a)}$$

where D = duct diameter

α_o = disc angle from a plane perpendicular to the duct.

Values of α_o can range from 0 to 30°. A limitation of the design configuration is the discontinuity in the peripheral seal which presents a potential leakage path around the shaft. In addition to this, portions of the seal near the shaft are in rubbing contact during most of the disc rotation. A design which avoids the problem of internal leakage around the shaft incorporates a canted shaft through the disc. This design permits continuous sealing around the entire periphery (Figure 5.2.4.1b). The seal can be attached either to the housing or around the periphery of the disc. It should be noted that in this configuration the angle of rotation of the shaft will be greater than the resulting rotation of the disc, the extent of the difference depending upon the offset angle of the shaft. As an example, for a butterfly valve having a shaft offset of 15° from the plane of the disc, a rotation of the shaft of approximately 94° will be required to effect a rotation of 90° of the disc. The exact relationship is given by the following equation:

$$\alpha = 90° + \sin^{-1} \tan^2 \beta \qquad \text{(Eq 5.2.4.1b)}$$

where α = shaft rotation required to change disc position from a plane perpendicular to the duct axis, to a plane parallel to the duct axis

β = shaft offset angle

In a third configuration, the disc is a section of a sphere and is mounted such that it pivots about an axis displaced from the plane of the disc (Figure 5.2.4.1c). This arrangement permits the use of a thin disc and provides for a continuous seal. In the design shown, the valve seal is in the body section. Erosion of the seal from fluid flow is minimized since it is located in the area of lowest velocity. In addition to being offset from the plane of the disc, the axis of rotation is offset from the center line of the bore in the body. The result of this arrangement is an eccentric rotation of the spherical section such that no contact is made with the seal except for the last few degrees of rotation. The effect is similar to a translatory action on the disc at this point, but the effect is achieved without the use of cams or an extra linkage. The downstream bore of the valve body is slightly relieved to permit the initial opening of the disc.

Care must be taken in a butterfly valve design to insure that the valve will not overtravel in going from the open to closed position. Common techniques for preventing disc rotation beyond the full closed position utilize either a positive stop on the disc or shaft, or a stop in the actuator assembly.

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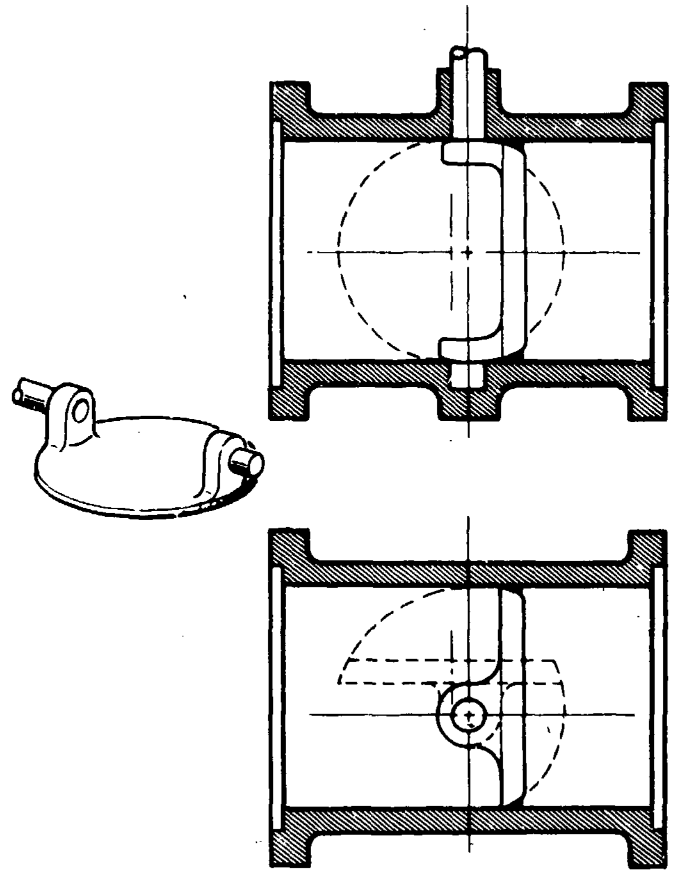

Figure 5.2.4.1c. Butterfly Valve (Eccentric Shaft)
(Courtesy of B. H. Hadley, Inc., Pomona, California)

5.2.4.2 **PERFORMANCE CHARACTERISTICS.** Butterfly valves have extremely good flow characteristics, particularly in large line sizes (> 2 inches) with flow resistance in the fully open position amounting to five to ten percent of the values for conventional globe valves having the same line size. In smaller line sizes, the disc and shaft occupy a larger percentage of the cross-sectional area available for flow and pressure drop increases.

Leakage characteristics of butterfly valves are relatively poor compared to most other valve types, due to problems of disc and bore concentricity, seal design problems, and the fact that in numerous designs the seal is exposed to damaging high velocity flow. However, high performance butterfly valves are available with soft seals (Kel-F, Teflon, or elastomers) having extremely good leakage characteristics and comparing favorably with poppet type designs.

The butterfly is an unbalanced closure as pressure drop and fluid momentum forces produce a net force on the disc in the direction of flow. This unbalanced force can result in high friction forces in the butterfly valve bearings, forces which in some designs can amount to approximately 50 percent of the total maximum actuator torque required. In all positions between closed and fully open, Bernoulli forces on the disc produce a torque tending to close the valve. The torque increases as the valve is opened, reaching a maximum between 60° and 80° of rotation, then decreasing rapidly to 0 at the full 90° open position. A more complete discussion of butterfly torque characteristics is given in Sub-Section 5.3, "Control Valves." The maximum operating position of conventional butterfly valves is achieved at some angle less than 90°. It often does not exceed 60°. The maximum duct opening is usually achieved prior to 90° rotation, because the disc disappears into the shadow of the disc shaft and further rotation of the disc does not provide additional flow area. In addition, when the valve is nearly 60° open, the disc creates a venturi action which results in good pressure recovery characteristics downstream. Any possible gain in the flow area by further opening of the disc is offset by the loss of the venturi action. Finally, rapid changes in torque characteristics which occur between 60° and 90° of disc opening can result in disc flutter and valve damage. A maximum operating point of approximately 60° is commonly dictated.

5.2.4.3 **APPLICATIONS AND LIMITATIONS.** Butterfly valves have low flow resistance, low face to face dimensions, and a relatively low initial cost. In addition, they are lightweight and usually inexpensive to maintain. The length of butterfly valves (exclusive of actuator mechanism dimensions) is the shortest of any valve type with the exception of the blade valve. The weight of the butterfly valve is characteristically low, and normally requires no support other than the line itself. The length, weight, and flow characteristics of butterfly valves make them well suited for low face-to-face pressure drop airborne applications. They normally are not used in high pressure systems, especially in the larger sizes, and are seldom used in line sizes less than one inch, regardless of pressure. Butterfly valves require at least one external shaft seal which provides a possible path for external leakage. The actuator mechanism for a butterfly valve will have a much longer travel than required for a poppet valve, because the disc must be rotated usually between 60° and 90° to achieve full flow. Actuator force requirements for butterfly valves are often higher than for other valve types, due primarily to flow force unbalance and bearing friction. As linear actuators are the most common type of valve actuators, butterfly valves require a linkage to convert linear to rotary motion. Although butterfly valves are often selected primarily for low pressure drop at a sacrifice of leakage control, high performance designs are available which have excellent leakage characteristics, if soft seats mounted either in the body or in the disc can be used in a given application. Useful temperatures for butterfly valves employing soft seats range from the cryogenic to approximately 500°F. Valves for high temperature service employing piston rings on the discs are available for temperatures to 1200°F. Valves made of refractory materials have been constructed with high temperature alloys and are usable up to 1800°F.

5.2.5 Poppet Valves

5.2.5.1 **DESCRIPTION AND OPERATION.** The term *poppet valve* refers to those valves in which the valving element travels perpendicular to a plane through the seating surface. This term is used synonymously with *globe valve* (not to be confused with globe body pattern). In its broadest sense, a globe valve is any valve using a poppet type closure. The construction of three poppet configurations — the flat disc, the sphere, and the cone — is shown in Figures 5.2.5.1a, b,c. Poppet designs which have a contoured control surface inserted into the control port are often referred to as plugs (Figure 5.2.5.1d). The terms plug and poppet are sometimes used interchangeably. As plugs are used primarily for their throttling characteristics, they are associated more with control valves than with shutoff valves. Poppet or globe type valves are usually referred to by body type as Y, angle, in-line, or globe body (Figure 5.2.5.1e,f,g). Poppet valves actuated by solenoids are commonly called solenoid valves (Figures 5.2.5.1h and 5.2.5.1i).

5.2.5.2 PERFORMANCE CHARACTERISTICS. A relatively large flow area is provided with short travel of the poppet. This characteristic simplifies the actuator requirements and permits use of actuators such as solenoids and diaphragms, which are characteristically short stroke devices.

Excellent leakage control may be achieved, using either hard or soft seats. If hard seats are used, however, great care must be exercised to eliminate contamination from the fluid media and from the system upstream of the valve.

Although unbalanced pressure forces on a poppet are often utilized to affect tight sealing, these forces can result in undesirable actuator force requirements. A poppet valve may be designed to eliminate the effects of unbalance from static and dynamic pressure, a feature not possible in many other valves. A common technique for balancing poppet valves is the use of two poppets on the same stem, seating on separate seats. The same pressure on the top of one poppet provides a counter balancing force on the bottom of the second poppet. Providing the poppet stem with a pressure balancing piston area, or making the poppet stem diameter equal to the seating diameter, are other techniques for minimizing unbalanced pressure forces. Complete pressure balance under both shutoff and flow conditions is virtually impossible to achieve. Pressure balancing usually results in compromising sealing characteristics, particularly in double poppet designs. Pressure balancing is less prevalent in shutoff valves than in control valves.

A poppet valve will be lighter than other types for many applications because of the small actuator stroke. For example, ball and butterfly valves require 90° rotation for full stroke, and the associated mechanism to accomplish this will be relatively large.

Flow characteristics can be well defined and closely controlled by contouring the poppet.

5.2.5.3 APPLICATIONS AND LIMITATIONS. A poppet type valve is used when rapid opening to full flow with a short travel of the valving element is desired. Of the various poppet valve body designs, the globe pattern creates the highest pressure drop. The flow path is improved in the Y pattern valve and it also has the advantage of a lower over-all valve height. The angle valve, which also has a much improved flow path over the globe body valve, is useful when it can replace an elbow, thus eliminating one turn in a line. The straight through, or in-line poppet valve, has the best poppet valve flow characteristics.

A poppet valve body may be cast in one piece (Figures 5.2.5.1e and f), or may be of the split body design in which the body halves are joined by bolts (Figure 5.2.5.1g).

5.2.6 Solenoid Valves

5.2.6.1 DESCRIPTION AND OPERATION. Strictly speaking, the term *solenoid valve* is not definitive in that it could mean any valve fitted with a solenoid actuator. However, by usage solenoid valve has come to mean a poppet or spool type valve operated by an integrally mounted solenoid. Because of the wide acceptance of the term and the extensive use of this type of valve, a discussion of it is included in this section.

Solenoid actuators are ideally suited for two-position applications, as required for shutoff valves. Solenoid valves may be simple, two-port, ON-OFF valves, or may be multipassage valves in which the flow can be directed to three or four ports as required. (See Sub-Section 5.8, "Multiple Passage Valves.") Solenoid shutoff valves are of a normally open or normally closed design, the normal position referring to the position of the closure with respect to upstream pressure with the solenoid de-energized. A simple, direct-acting, normally closed solenoid valve is illustrated in Figure 5.2.5.1h. A coaxial direct acting solenoid valve is illustrated in Figure 5.2.5.1i. For high pressures and large poppet sizes, where actuation forces might exceed those which can be conveniently provided by a solenoid, balanced and/or piloted designs may be used. In a piloted design, the solenoid is required to control a very small flow of fluid which permits the line pressure or pressure from an external source to actuate the main poppet. Figure 5.2.5.1j illustrates a typical, normally closed pilot design. The solenoid opens the pilot valve which dumps fluid from behind the main poppet through the pilot port. Since the pilot port is sized larger than the bleed port, the resulting pressure unbalance forces the main poppet open. Note that a differential pressure must be available to operate this type of valve. An in-line piloted solenoid valve is illustrated in Figure 5.2.5.1k. Figure 5.2.5.1l shows the approximate limits for direct actuation as a function of pressure and orifice diameter for several typical solenoid valves. The limits are based on requirements for effective sealing and long life for continuous duty applications. Direct operation above these curves can be exceeded where leakage requirements are not stringent, where continuous duty operation is not required or only a limited operation life is required, and where solenoid temperature is allowed to rise. A detailed discussion of solenoid actuators is included in Sub-Section 6.9.

5.2.6.2 PERFORMANCE CHARACTERISTICS. The outstanding feature of direct acting solenoid valves is the high response times that may be obtained—in the range of

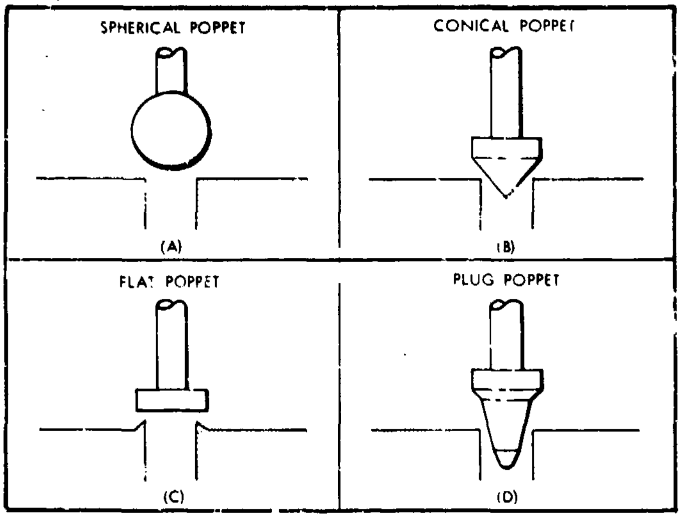

Figure 5.2.5.1a,b,c,d. Poppet Configurations

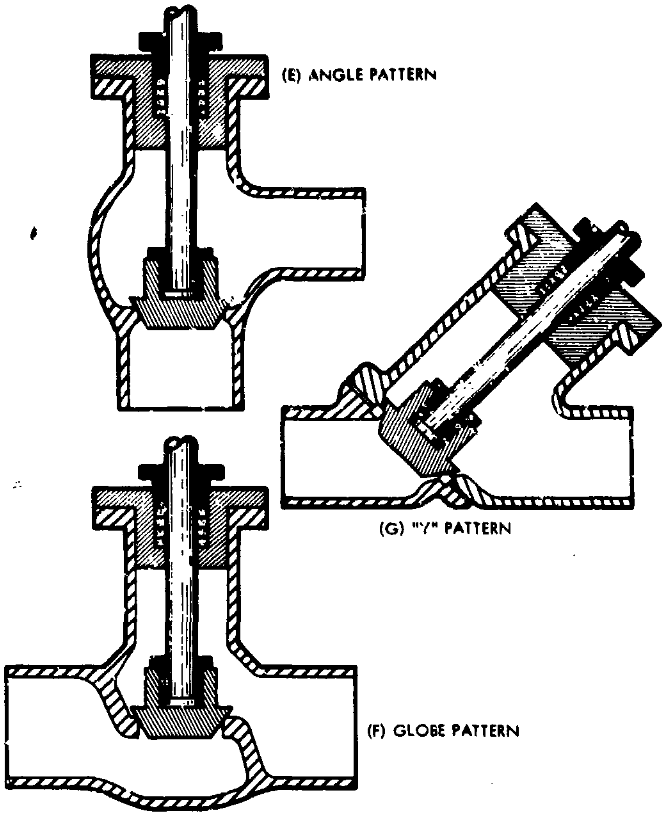

Figure 5.2.5.1e,f,g. Poppet Valve Body Types

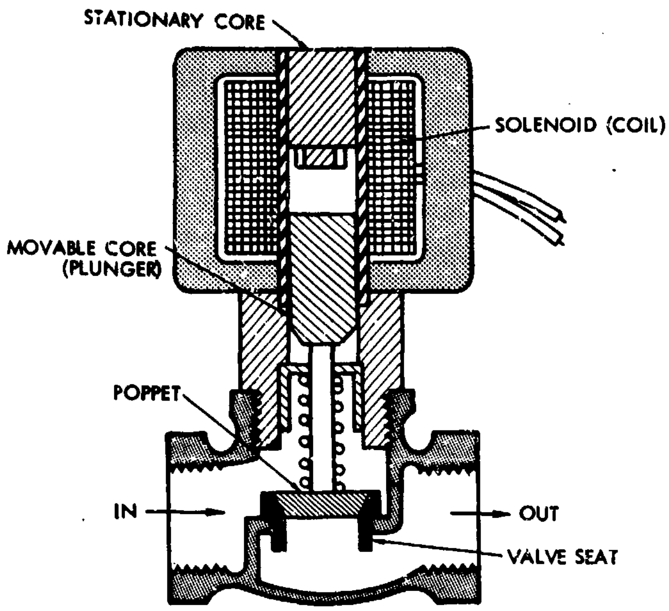

Figure 5.2.5.1h. Normally-Closed Direct-Acting Solenoid Valve

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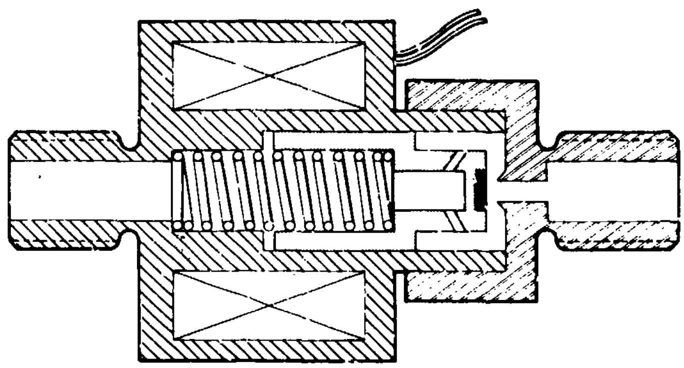

Figure 5.2.5.1i. Coaxial Direct-Acting Solenoid Valve

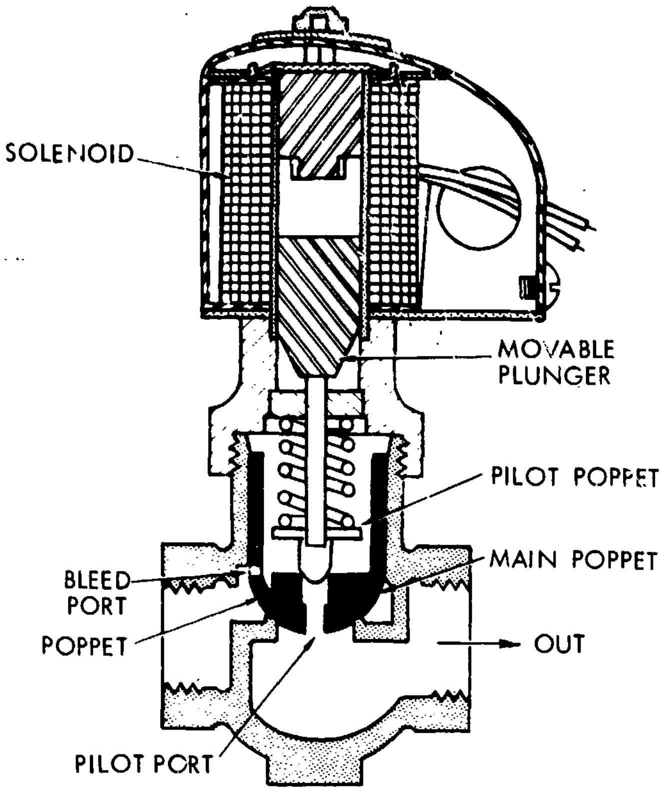

Figure 5.2.5.1j. Piloted, Normally-Closed Solenoid Valve
(Reprinted from "Instruments and Automation," September 1956, vol. 29, no. 9, F. E. Reeves, Copyright 1956, Instruments Publishing Company, Inc., Pittsburgh, Pennsylvania)

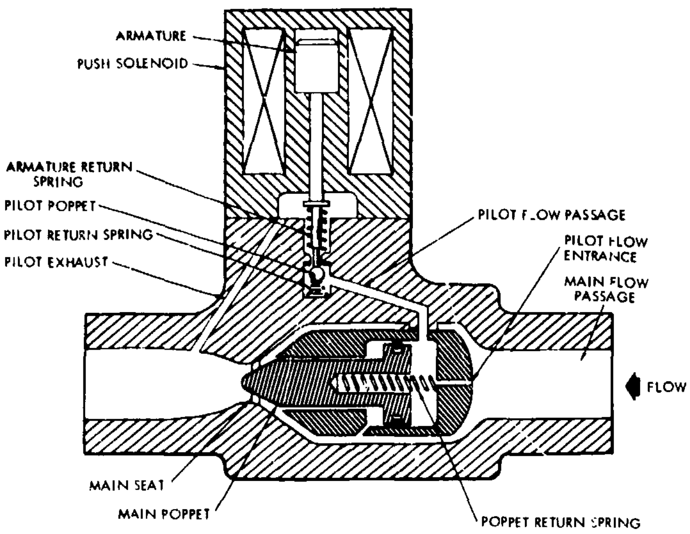

Figure 5.2.5.1k. Piloted, In-Line Solenoid Valve

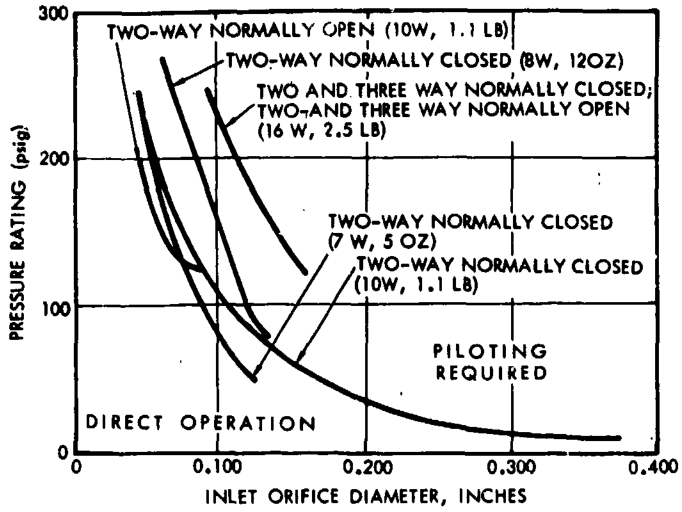

Figure 5.2.5.1i. Approximate Size Limitations for Direct-Acting Solenoid Valve
(Reprinted from "Machine Design," January 17, 1963, vol. 32, no. 2, J. E. Ellison, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

5 to 50 milliseconds. This high response is due to the fact that a relatively large opening can be effected through a short stroke of the spool or poppet. The current required to operate the valve depends upon the size of the valve, the differential pressure, and the response required. Leakage control is normally very good with this type of valve, especially if a soft seat is used. Where hard seats are used, good leakage control is obtained by careful lapping of the seat parts. Since solenoid valves require no external dynamic seals, external leakage problems are minimal. The life expectancy of this type of valve is excellent, with many hundreds of thousands of cycles being possible without degradation of performance.

5.2.6.3 APPLICATIONS AND LIMITATIONS. Solenoid valves are used for ON-OFF functions only, since the solenoid is a two-position actuator. In airborne applications, the line sizes for direct acting valves are usually one-half inch and less. Larger sizes are generally impractical because of the excessive weight of the solenoid required for operation.

Solenoid valves are commonly used to act as pilot valves for larger fluid actuated valves. In this application the solenoid valve acts as a relay which controls a high actuation force by a small electrical input signal.

5.2.7 Gate and Blade Valves

5.2.7.1 DESCRIPTION AND OPERATION. Blade and gate valves consist essentially of a sliding member that moves perpendicular to the flow stream, cutting through the fluid to accomplish shutoff. Two typical configurations

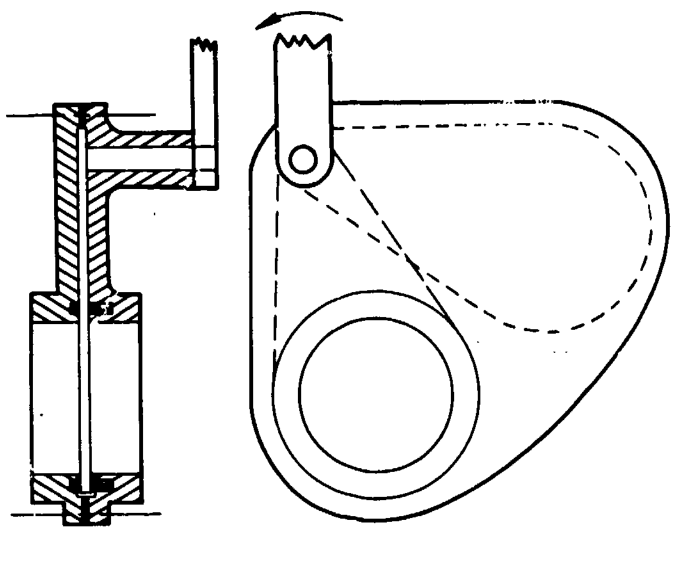

Figure 5.2.7.1a. Blade Valve

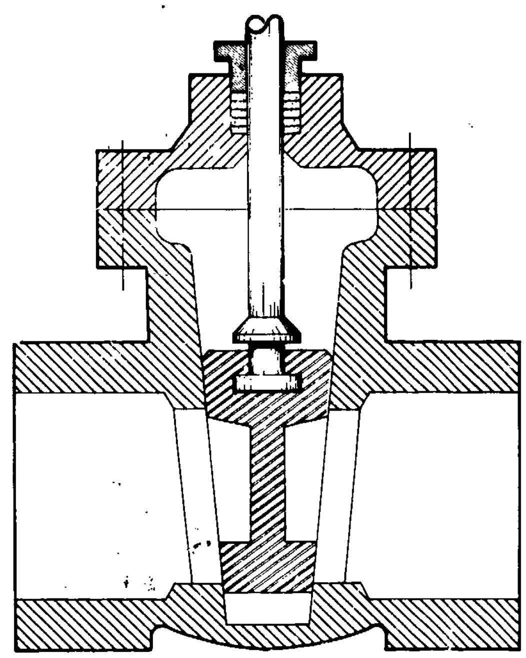

Figure 5.2.7.1b. Gate Valve (Solid Gate)

are shown in Figures 5.2.7.1a and 5.2.7.1b. The valve in Figure 5.2.7.1a is generally called a blade valve, while the valve in Figure 5.2.7.1b is more commonly called a gate valve. The gate valve usually seals on both sides of the gate by a wedging action, while the blade valve usually seals on one side of the blade with fluid pressure, providing all or part of the sealing force. The seal may be metal-to-metal but more often will consist of an elastomeric seal installed in the body. The valve may be designed to seal in either direction, but because of the application normally found for this valve, the seal is often more effective in one direction than in the other.

Blade and gate valves may be operated by any of the conventional actuators available. Quick return mechanisms have been used where it was desired to open or close the valve at different rates. Squib actuators have been used to close blade valves where very rapid, single action shutoff was required. The blade valve has the shortest face-to-face dimensions of all valves, but this advantage may be offset by the space required for the actuator. The blade can either rotate about a pivot [illegible] linearly.

5.2.7.2 PERFORMANCE CHARACTERISTICS. Blade and gate valves are normally used as ON-OFF valves exclusively, since their throttling characteristics are poor and valve closure and seat erosion occur rapidly in the near shutoff position. In the wide open position, the pressure drop characteristics are essentially equal to those of a ball valve, although some turbulence is generated in the slot, or gap, between the body halves when the blade is withdrawn. The response of the valve is inherently slow due to large travel and high actuation forces resulting from seal friction. Leakage control is excellent. The valve is unbalanced and large actuation forces limit the operating pressure range.

5.2.7.3 APPLICATIONS AND LIMITATIONS. Blade valves are ideally suited for use as pre-valves in low pressure systems such as propellant feed systems on aircraft and missiles. In these systems, pressures are nominal and response is not critical. Another example of a blade valve application is found in vacuum equipment, where very low pressure drop and tight shutoff are extremely important.

In ground support systems, gate valves find applications as shutoffs or block valves in moderate pressure systems. They are normally hand operated, and may use either rising or non-rising stem design. The rising stem gives a visual indication of the gate position. However, if overhead space is limited, a non-rising stem design can be used.

5.2.8 Diaphragm Valves

5.2.8.1 DESCRIPTION AND OPERATION. A diaphragm valve consists of three major elements: body, diaphragm, and bonnet assembly. Figure 5.2.8.1a illustrates a typical diaphragm valve known as the Saunders Patent Valve. Closure of this valve is effected by pressing the flexible diaphragm against a transverse ridge which is cast into the valve body. In this valve, the flexible diaphragm acts both as the valving element and as the external seal. A modification of the diaphragm valve, which has a rubber plug molded integrally with the diaphragm, is shown in Figure 5.2.8.1b. Actuation of diaphragm valves may be manual — by any of the usual mechanisms such as screw thread or quick acting cam, or automatic — by any of the conventionally-powered actuators.

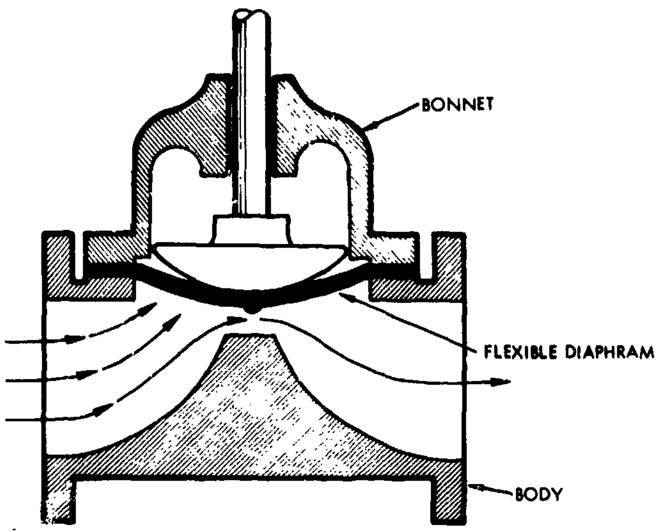

Figure 5.2.8.1a. Diaphragm Valve (Saunders Patent)

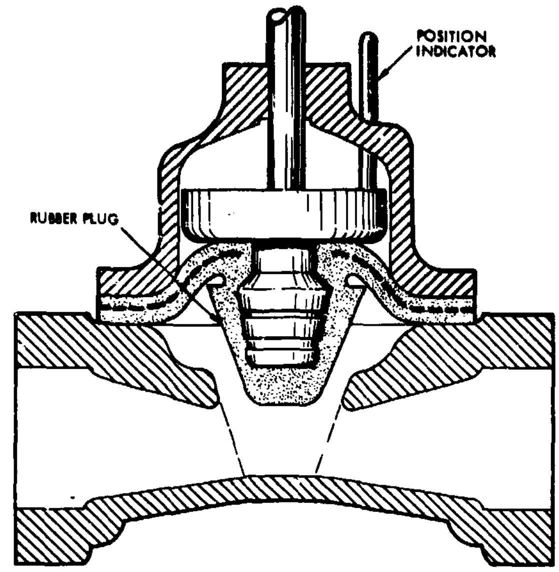

Figure 5.2.8.1b. Dia-Plug Valve
(Courtesy of Cryogenics Corporation, Meadville, Pennsylvania)

5.2.8.2 PERFORMANCE CHARACTERISTICS. The flow characteristics of diaphragm valves are considerably better than for globe valves. Bubble-tight shutoff is easily obtained and, since the valve requires no packing, external leakage is essentially zero. It should be noted that although diaphragm valves do not require external packing, due to the possibility of rupturing the diaphragm, some models are available with an external packing seal or a bellows seal for safety purposes where highly toxic or corrosive fluids are involved. The valve is very tolerant of contamination, since the diaphragm is pliable enough to envelope trapped contaminants. Diaphragm valves may be used for throttling service, as discussed in Detail Topic 5.3.5.4. Operating temperature limits are determined by the diaphragm material, normally ranging from 0°F to 300°F. The temperature range can be expanded from cryogenic temperatures to approximately 500°F by using fluorinated polymers such as Teflon, however sealing properties are poor. Operating pressure ranges for this valve are rather low, and depend upon the diaphragm material and line size. The diaphragm is accessible from the top and is easily replaced.

Table 5.2.8.2 shows maximum allowable working pressures for the various sizes of valves available. Because of the size of the diaphragm area, considerable force is required to operate a diaphragm valve, hence relatively large actuators are required, even on the smallest sizes.

5.2.8.3 APPLICATIONS AND LIMITATIONS. The diaphragm valve has a wide application in systems where the temperature and pressure limits of the diaphragm will not be exceeded. The body may be fabricated from many combinations of materials, selection of which would depend upon the flowing medium. Linear streamlined flow through the valve produces a self cleaning action that makes it ideal for slurries and viscous fluids. Because of the internal configuration of the valve, it is well suited to uses where a self draining feature is important. Because of the resilience of the diaphragm, hydraulic shocks in the system are readily absorbed if the pressure surges do not exceed the strength limitations of the diaphragm material. The diaphragm valve has the advantage of simplicity, having no internal trim except the seat and diaphragm. A disadvantage is that periodic inspection is necessary to guard against failure of the diaphragm.

Table 5.2.8.2. Maximum Pressure for Several Valve Sizes

VALVE SIZE (in.)	PRESSURE (psig)
½ – 4	150
5 – 6	125
8	100
10 – 12	65
14 – 16	50

5.2.9 Flexibe Tube Valves

5.2.9.1 DESCRIPTION AND OPERATION. The flexible tube valve closure incorporates a flexible elastomeric tube as the closure device. The simplest form of flexible closure valve is a pinch valve which consists of a collapsible rubber tube squeezed shut by a clamp. A familiar example is the tube clamp common in chemical laboratories. Several modifications of this concept have been made, illustrated in Figures 5.2.9.1a and 5.2.9.1b. Figure 5.2.9.1a shows a simple pinch valve, consisting of a flexible tube surrounded by a housing fitted with an operator which mechanically collapses the tube. Figure 5.2.9.1b shows a valve in which ap-

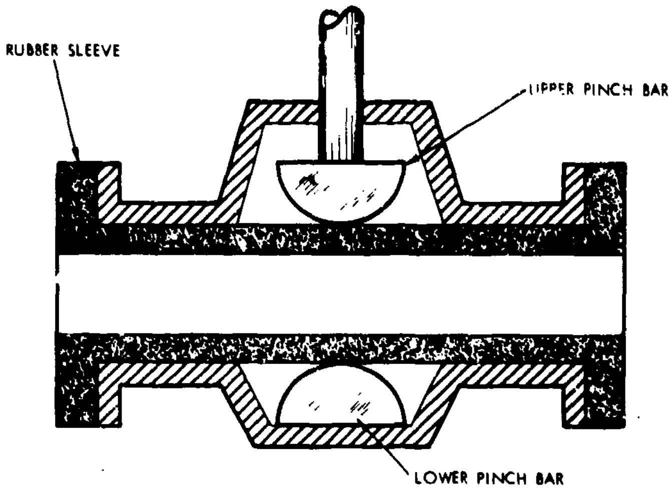

Figure 5.2.9.1a. Pinch Valve, Mechanical Actuation

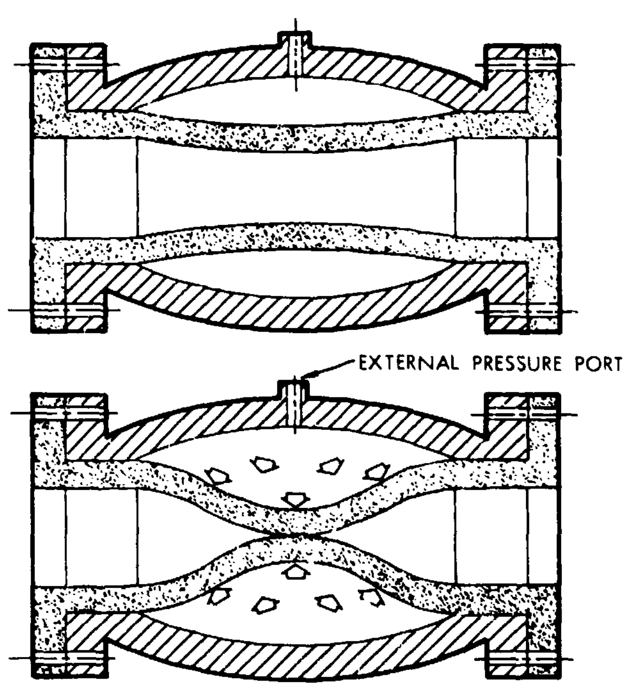

Figure 5.2.9.1b. Pinch Valve, Pressure Actuation
(Courtesy of Red Jacket Company, Inc., Carnegie, Pennsylvania)

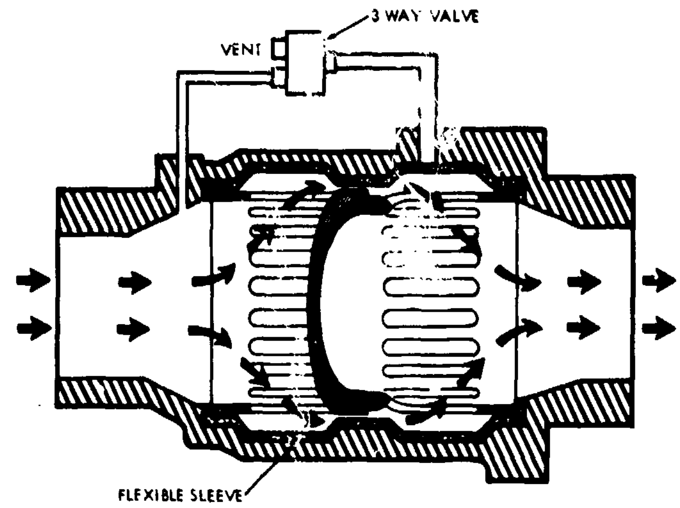

Figure 5.2.9.1c. Flexible Sleeve Valve
(Courtesy of Grove Valve and Regulator Company, Oakland, California)

plication of external pressure to the housing, in excess of the internal pressure, causes the tube to collapse, thus stopping the flow of the fluid. A third flexible tube valve configuration is the expansible tube valve shown in Figure 5.2.9.1c. The valve consists essentially of a housing — a slotted core over which is fitted an expansible elastomeric tube. When external pressure applied to the tube exceeds internal tube pressure, the valve moves to the closed position. Reversing the relative pressures by venting external pressure causes the tube to expand permitting flow through the metal core.

5.2.9.2 PERFORMANCE CHARACTERISTICS. Flexible tube valves have excellent shutoff characteristics, because they are relatively insensitive to contamination. Pressure drops are small, since the full line diameter is available for flow when the valve is in the open position. The useful temperature range of flexible tube valves is limited to approximately 0°F to +150°F. Pinch valves are normally limited to a maximum operating pressure of 50 to 100 psi, which is more restrictive than for diaphragm valves. The expansible tube valve illustrated in Figure 5.2.9.1c, on the other hand, can be used for pressures as high as 1200 psi. Operating characteristics of these valves when used as control valves are discussed in Detailed Topic 5.3.5.4.

5.2.9.3 APPLICATIONS AND LIMITATIONS. Flexible tube valves are ideally suited to handling contaminated fluids which would cause accelerated wear in other types of valves. Shutoff is possible as long as the rubber tube is able to envelope particles without damage to the tube. All applications for this type of valve would be under very nominal temperature conditions. The inherent reliability of these valves is high, due to the simplicity, lack of metal-to-metal seals, bearing surfaces, and stem packing.

5.2.10 Hermetic Seal Valves

Hermetic seal valves are a class of shutoff devices used to provide positive sealing of rocket propellants and/or pressurants over extended time periods prior to system activation. Hermetic seal valves are most often used as one shot devices which have no reclosure capability after the system is actuated. A wide variety of hermetic seal design configurations are possible. The following discussion describes a simple rupture disc, a cutter-type valve, a shear disc butterfly valve, and a fusion seal valve. Squib actuated hermetic seal valves, commonly called squib or explosive valves, are discussed in Sub-Section 5.7.

5.2.10.1 RUPTURE DISC, LINE PRESSURE ACTUATED

Description and Operation. The rupture disc consists essentially of a diaphragm clamped between two body halves. The disc is designed to break when a predetermined system pressure has been reached. Control of the breaking point is achieved in various ways. One way is by pre-scoring the disc in a pattern that will cause rupture to occur at the proper pressure and in a manner that will prevent particles from going downstream. When a pie-shaped scoring is used, the leaves or petals fold back against the walls of the tube in a uniform manner. In another configuration the disc is scored so that the rupture occurs over approximately 300 of the periphery. The remaining portion of the disc acts as a hinge, and allows the disc to swing downstream and come to rest against the wall of the tube. Laminated discs are sometimes used at high pressures, since it is easier to achieve repeatable characteristics with laminations than with a single material thickness.

Performance Characteristics. Line pressure actuated rupture discs are simple, lightweight, and cause minimum pressure drop, but cannot be reclosed. In order to insure that particles from the disc are not carried downstream, a screen or petal catcher is often located downstream of the rupture disc. Rupture discs are subject to damage from pressure reversals which may cause fatigue of the disc and result in premature opening. Backup plates should be used to prevent reversals of the diaphragm.

Applications and Limitations. Rupture discs are used where light weight and simplicity are combined with requirements for zero leakage over long periods of time and minimum pressure drop after actuation. Rupture discs must be carefully designed to avoid the introduction of particles into the system. Corrosion and fatigue resulting from pressure fluctuations can result in premature actuation.

5.2.10.2 CUTTER VALVE, MECHANICALLY ACTUATED

Description and Operation. The mechanically operated rupture valve differs from the system pressure operated rupture disc in that a positive means of breaking the disc is provided. In Figure 5.2.10.2, a design is illustrated where a cutter is forced into the diaphragm, thus initiating flow. The cutter may be either squib or pressure actuated.

Performance Characteristics. The cutter type valve is more complex and heavier than the simple rupture disc; and, because the cutter mechanism is in the stream, the valve will have a slightly higher pressure drop. However, the rupture of the disc is precisely controlled, and the manufacturing tolerances of the diaphragm are less critical.

Applications and Limitations. In general, applications of the cutter valve are similar to those for the pressure operated rupture disc described above. In systems where the difference between standby pressure and operating pressure is small, the cutter type would be superior, since the critical requirements for the breaking point would be eliminated.

5.2.10.3 SHEAR DISC BUTTERFLY VALVE

Description and Operation. The hermetically sealed, shear disc butterfly valve is a conventional butterfly valve with a positive seal between the blade and the housing. This feature can be achieved in several ways. For example, the disc may be cast as an integral part of the housing, after

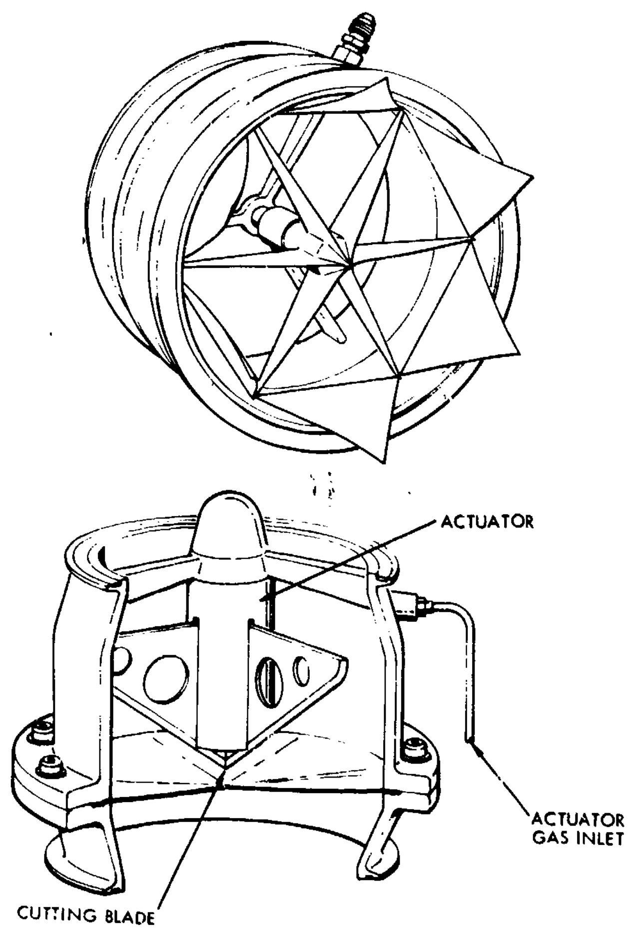

Figure 5.2.10.2. Diaphragm Cutter Valve
(Courtesy of TRW/Space Technology Laboratories, Redondo Beach, California)

which it is scored at the junction of the housing and the disc and permits the disc to shear at the housing upon command. Another configuration consists of a conventional butterfly disc incorporating a thin metal diaphragm which is clamped between the body and butterfly halves. Rotation of the butterfly shears the diaphragm. Initial rotation of the butterfly may be accomplished by a squib or high pressure pneumatic actuator. The shear disc butterfly valve can either be designed for one shot applications or can be designed with an actuator mechanism for reclosing the valve. On reclosure, sealing is accomplished in a manner similar to conventional butterfly valves.

Performance Characteristics. The shear disc butterfly valve has a pressure drop slightly higher than that of the pressure operated rupture valve, and approximately equal to that of the cutter type valve. Leakage control on reclosure is poor for some designs.

Applications and Limitations. The application of this valve would be primarily in a system where a reclosable feature is required in addition to the usual attributes of a hermetically sealed valve. Tight shut OFF-ON reclosure is often not a requirement.

5.2.10.4 FUSION SEAL VALVE

Description and Operation. The fusion seal valve shown in Figure 5.2.10.4 accomplishes a positive seal by fusing the movable portion of the closure to the stationary part by the use of a low melting point alloy, or solder. The alloy is raised to the melting point by an integral heating element, after which the movable plug of the valve may be shifted to the open position. The solder is not exposed to the flowing fluid and will not be swept away, permitting a resealing capability. Heat is again applied to fuse the valve plug and seat together when reclosure is desired.

Performance Characteristics. The fusion seal valve may be opened and closed repeatedly, and still achieve zero leakage after each resealing. In the design illustrated in Figure 5.2.10.4 the actuation time is relatively slow compared to other valves, in that several seconds are required for opening and closing. However, by the use of an impact type actuator the valve could be sheared to the open position, greatly improving the response characteristics.

Applications and Limitations. The fusion seal valve is suited for ON-OFF application in spacecraft systems where long coast periods between valve actuations would require absolutely tight shutoff. Because of propellant compatibility problems with the solder, the fusion seal valve is best suited for applications in pneumatic systems where gases such as helium or nitrogen are employed.

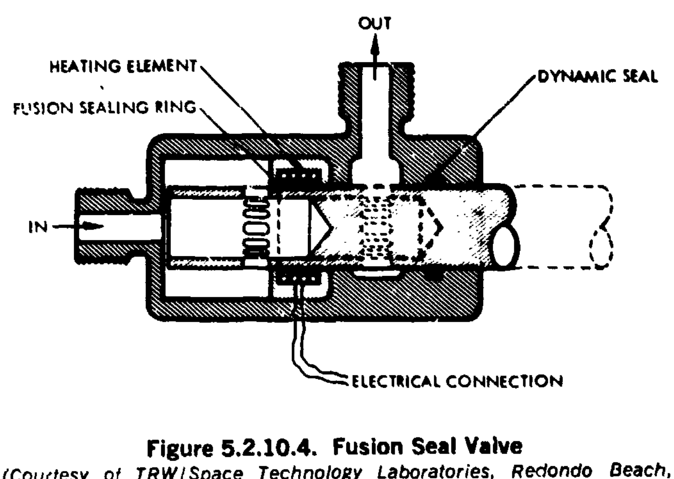

Figure 5.2.10.4. Fusion Seal Valve
(Courtesy of TRW/Space Technology Laboratories, Redondo Beach, California)

5.2.11 Rotary Plug Valves

There are three basic types of plug valves: simple cock, lubricated plugs, and non-lubricated plug valves.

5.2.11.1 SIMPLE COCK

Description and Operation. The simple cock is the first type of valve known to man. In the cock, a rotating tapered plug extends completely through the valve body. Internal and external leakage are controlled by a close fit between the tapered plug and the body. A lubricant is used between the plug and the body to make operation easier and to help maintain a tight seal. These valves are low in cost, simply constructed, and require very little headroom. Actuation is accomplished by a quarter-turn of the plug to go from full open to full close. Figure 5.2.11.1 shows the straight-

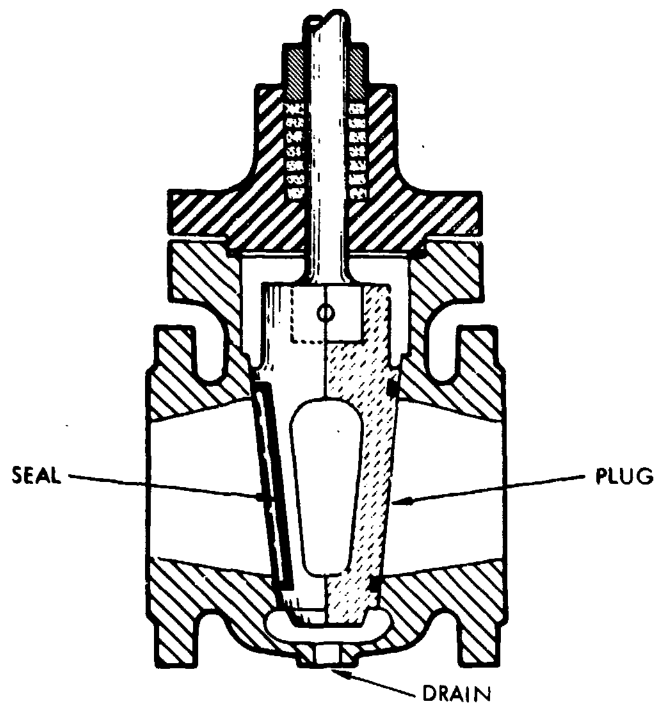

Figure 5.2.11.1. Straight-Through Plug Cock Valve

through plug cock of the simple design described. In addition to two-way designs, cocks are available for three- and four-way operation (see Sub-Section 5.8).

Performance Characteristics. Simple cocks provide tight shutoff, have low pressure drop, and are relatively long lived. Because they are unbalanced valves, the required actuation forces will increase in proportion to the pressure. External leakage may be a problem when the valve is rotated, if the conical plug is permitted to rise in the body.

Applications and Limitations. Two-way cocks are used for simple ON-OFF operations, and are usually limited to manual operation. Cocks are used for both liquid and gas systems. They should not be used for high temperature service, because the heat will dissipate the lubricant and cause the plug to bind and gall.

5.2.11.2 NON-LUBRICATED PLUG VALVE

Description and Operation. Non-lubricated plug valves are similar in design to the simple cock, but include modifications to permit use of the valve without lubrication and the use of stem packing to prevent external leakage. Friction is reduced by choice of non-galling materials, and binding under high temperature conditions is eliminated by mechanical design. This is usually accomplished by a cam mechanism which lifts the plug slightly off the seat with the first few degrees of rotation. The plugs are then reseated when the full open position is reached. Teflon inserts in the plug are used for sealing and reduction of friction.

Performance Characteristics. Performance characteristics of the non-lubricated plug valve are similar to those of the simple cock, with the exceptions that seals are provided to prevent external leakage, and actuation forces are minimized by using low friction materials for plug seals and by lifting the plug prior to rotation.

Applications and Limitations. Applications for non-lubricated plug valves are similar to those for which cocks are used. However, the range of temperature and pressure that are permissible is greatly extended. Valves have been produced for use at pressures as high as 3000 psig and temperatures to 1100 F. Plug valves can be used with remotely controlled operators as well as the more common manual operators.

5.2.11.3 LUBRICATED PLUG VALVE

Description and Operation. The lubricated plug valve differs from the simple cock and the non-lubricated plug valve in that provision is made for injection of lubricant to the plug under pressure. The lubricant serves the dual purpose of providing a seal and raising the plug from the body by hydraulic action to circumvent the problems of binding. Plug valves are made from a variety of metals and alloys including bronze, carbon steel, alloy steel, and stainless steel (hard faced or chrome plated).

Performance Characteristics. Performance characteristics of the lubricated plug valve are similar to those of the non-lubricated plug valve. The operating temperature range is limited to that for which the lubricant is suited.

Applications and Limitations. The lubricated plug valve may be used for ON-OFF or throttling service within the limitations of the lubricant, with respect to operating temperature and fluid compatibility. Lubricated plugs require periodic lubrication to minimize leakage and to prevent binding of the plug. Plug valves have been made in sizes up to 30 inches and for pressures as high as 15,000 psi.

5.2.12 Rotary Slide Valves

5.2.12.1 DESCRIPTION AND OPERATION. The rotary slide valve consists basically of a ported rotor or disc which is held closely against a seating surface having matching ports. Variations in basic rotary valve construction involve sealing techniques, construction features to balance forces in high pressure valves, and various designs for bearings and seats.

Figure 5.2.12.1a is a cross-section through a pressure balanced rotary slide valve which is manufactured in many sizes for pneumatic and hydraulic service to 6000 psi. The valve has optically flat, hardened, stainless steel seals which are spring and pressure loaded against the upper and lower rotor surfaces. These seals are backed at each port with O-rings which seal passages from each other when the valve is actuated. The bottom seal prevents leakage between ports as do the top seals, which also provide a balancing force on top of the rotor and eliminate the need for thrust bearings. Because the rotor disc is in balance, the torque required to actuate the valve remains low, regardless of operating pressure. Washer type springs under the seals provide the initial force necessary for effective sealing of the rotor and for compensation for wear of the sealing surface. The seals wipe dirt off the valve rotor during shifting, virtually eliminating scoring and leakage.

Another type of rotary valve capable of moderately high pressure service is illustrated in Figure 5.2.12.1b. This valve is made for pressures to 3,000 psi in smaller sizes only (up to one-quarter inch NPT). This valve uses single pressure loaded seals between the rotor and housing, which provide the same wiping action as in the valve illustrated in Figure 5.2.12.1a. Pressure on the valve rotor is not balanced in this design and, consequently, thrust forces must be taken by a large thrust ball bearing between the upper surface of the rotor and the valve housing. The rotor stem and housing are sealed with O-rings to prevent leakage.

5.2.12.2 PERFORMANCE CHARACTERISTICS. Excellent leakage control is achieved by rotary slide valves because of the exceptionally fine finish that may be achieved on flat lapped plates. Operating torque will be fairly high on valves that are not pressure balanced or that do not use anti-friction bearings in the moving parts. Pressure drop is relatively high because of the several turns that the fluid must make in passing through the valve. The valve may be used for throttling, and because of the close fit of the mating

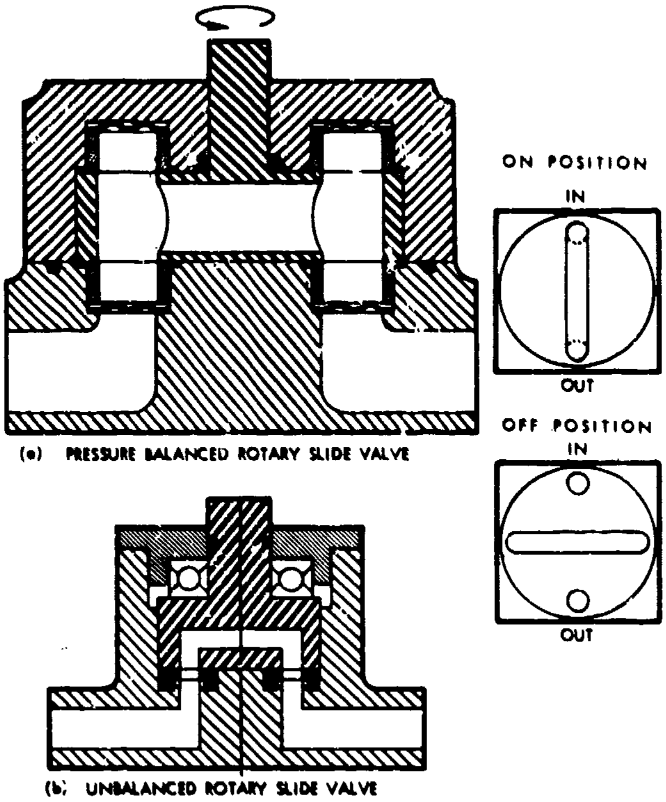

Figure 5.2.12.1a,b. Two-Way Rotary Slide Valves
(a) Pressure Balanced
(b) Unbalanced

parts, the valve is relatively insensitive to contamination. The valve is long lived and normally may be refurbished by a simple seat replacement which may be accomplished without removal from the line.

5.2.12.3 APPLICATIONS AND LIMITATIONS. Rotary slide valves should be used on pneumatic or hydraulic systems when good shutoff characteristics are required, pressure drop is not a prime consideration, and position indication and quick actuation are desired.

REFERENCES

General
2-57, 19-158, 160-26, 184-5, 185-1, 193-5

Ball
49-42, 74-3, 74-22

Butterfly
1-42, 112-5, 165-3, 193-24, 195-7, 251-1, 282-1, V-192

Poppet
6-86, 6-88, 19-66, 232-10

Solenoid
1-245, 160-5, 185-1

Gate and Blade
64-13, 193-17, 195-2

Diaphragm
6-94, 195-8, V-286, V-161

Hermetic Seal
V-283, V-192

Rotary Plug
193-23, 195-1

Rotary Slide
V-75

5.3 CONTROL VALVES

5.3.1 Introduction

A flow control valve may be defined as a valve which regulates or otherwise controls the volumetric flow rate of any gas, liquid, or fluidized solid material by means of a variable area flow restriction. A flow control valve may have in infinite number of operating positions in contrast to a shutoff valve (Sub-Section 5.2) which is either fully open or fully closed. The throttling function of a control valve is an entirely separate requirement from the shutoff function of a shutoff valve. Although the shutoff function can be added to a control valve, since control is the prime goal, the secondary importance of shutoff in many control valve designs results in a valve having poor shutoff characteristics. Therefore, it is common practice to keep the functions separated by incorporating both a shutoff valve and a control valve in a system. Pressure regulators (Sub-Section 5.4) are also flow control devices, but are distinguished from control valves in that they are self-operating controllers, whereas a control valve is the final control element in a control system which requires a source of operating power other than from the controlled fluid media. Control valve actuators are discussed under Sub-Section 6.9. It should be noted that the term *control valve* is used synonymously with *throttle valve*, and that small manual control valves are also called *metering valves*. It should also be noted that there are applications such as attitude control rocket systems where valves which are basically shutoff valves (ON-OFF valves) are called *flow control valves*, control being achieved by varying the duration of constant flow rate pulses.

5.3.2 Control Valve Terminology

5.3.2.1 FLOW CAPACITY. The flow capacity of a control valve is the maximum rate of flow through the valve at specified conditions of pressure drop and temperature.

5.3.2.2 FLOW COEFFICIENT. The flow coefficient or C_v factor of a control valve is the number of gallons of water per minute at 60°F that will flow through a valve at maximum opening with a pressure drop of 1.0 psi. For a more detailed discussion of flow coefficient, see Sub-Topic 3.8.4.

5.3.2.3 LINEARITY. Linearity of a flow control valve is a measure of the degree to which there is a straight line relationship between the valve output (volumetric flow rate) and the input (valving element position or stroke) at a constant value of pressure drop.

5.3.2.4 HYSTERESIS. Hysteresis may be defined as the difference in flow through a control valve for a given setting between the case where the set point has been attained from a lower flow position and the case where the set point has been attained from a higher flow condition. Hysteresis in a control valve is usually the result of non-linearities caused by slack or play in the valve actuator.

5.3.2.5 DEADBAND. The deadband of a control valve is the span through which the input signal can change without initiating a movement of the valving element. Deadband and hysteresis are directly related.

5.3.2.6 VALVE GAIN (VALVE SENSITIVITY). Valve gain, or sensitivity, is the ratio of change in flow, ΔQ, to the corresponding change in stroke, ΔX, with a constant pressure drop across the valve. Valve gain is constant for a linear valve.

5.3.2.7 RANGEABILITY (THROTTLE RANGE). The rangeability or throttle range of a flow control valve expresses the useable portion of the total valve travel or flow range of the valve. Rangeability may be defined as the ratio of maximum to minimum controllable flow, or the flow range through which a particular inherent flow characteristic is maintained within prescribed tolerance limits. For example, if a control valve can be used between 10 and 100 percent of its stem travel, the rangeability factor is ten. The rangeability of a control valve is limited by the problem of accurately controlling the flow with the valve in a nearly closed position. The definition of rangeability is somewhat vague in that the minimum controllable flow is not clearly defined.

5.3.2.8 UNIT SENSITIVITY. Unit sensitivity (not to be confused with valve sensitivity, or gain) is defined as the percentage change in flow rate through a valve resulting from a one percent change in the valve stroke. The percentage change in flow rate is based on the flow rate just before the change. Unit sensitivity is constant for an equal percentage valve. Unit sensitivity for a linear valve varies from a maximum value at low flow to a minimum value at full flow.

5.3.3 Control Valve Flow Characteristics

One of the most important criteria in the selection of a flow control valve is its flow characteristic. A control valve flow characteristic is specified in terms of its inherent flow characteristic (Sub-Topic 5.3.4, "Inherent Versus Effective Flow Characteristics"). The inherent flow characteristic describes the flow rate through the valve as a function of control element position (stem travel) under conditions of constant temperature and constant pressure drop across the valve. Common flow characteristic curves are linear, modified linear, equal percentage, and square root (quick opening). A graphical comparison of these characteristics appears in Figure 5.3.3. Almost any flow characteristic curve can be obtained by proper shaping of the valving element or by manipulating the valve opening and closing movement. In shaping a valving element to achieve a desired inherent flow characteristic, the designer must take into consideration the influence of valve entrance, exit, and housing pressure losses on flow through the valve. Ideally, the entire control valve pressure drop should occur across the control element. However, although every attempt should be made to minimize other pressure losses by avoiding rapid changes in flow path geometry, such losses can never be completely eliminated, particularly at higher flow rates. Therefore, it is often necessary that the designer compensate for such losses by modifying the flow area versus stroke relationship for the valving element from the theoretical in order that the entire valve have the desired flow characteristic. Some of the more common types of flow characteristics are described in the following paragraphs. Variations and/or combinations of the specific characteristics mentioned here can be engineered into a given valve design to fulfill special requirements.

5.3.3.1 LINEAR FLOW CHARACTERISTIC. A linear flow characteristic produces a flow rate which varies linearly with valve position. The general equation expressing flow as a function of valve opening for a linear control valve is

$$Q = ay \qquad \textbf{(Eq 5.3.3.1)}$$

where Q = flow rate

y = valve opening

a = arbitrary constant

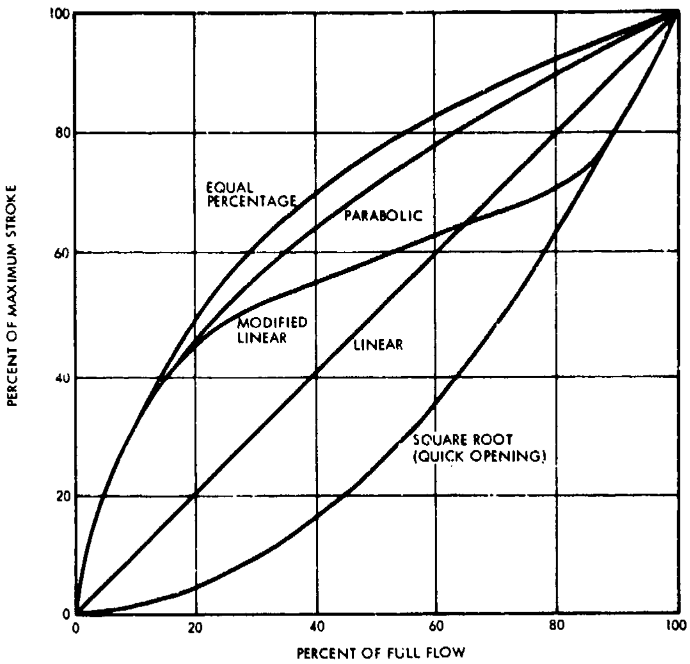

Figure 5.3.3. A Comparison of Control Valve Flow Characteristics

A plot of flow versus valve opening for a linear characteristic is a straight line on rectangular coordinate graph paper.

5.3.3.2 PARABOLIC FLOW CHARACTERISTIC. A parabolic flow characteristic is one in which flow varies with the square of the control element position. The general equation expressing flow as a function of valve opening for a parabolic control valve is

$$Q = ay^2 \qquad \textbf{(Eq 5.3.3.2)}$$

where Q = flow rate

y = valve opening

a = arbitrary constant

A parabolic flow characteristic provides a constant rate of change in flow with changing valve opening. When plotted on log-log coordinates, the parabolic characteristic becomes a straight line. A more general mathematical representation of a parabolic flow characteristic is $Q = ay^n$, where n is any real number.

5.3.3.3 MODIFIED LINEAR CHARACTERISTIC. A flow control characteristic which combines a parabolic characteristic with the linear characteristic is termed modified linear. The modified linear characteristic is graphically a parabolic characteristic for approximately the first 30 percent of the valve opening, whereupon it assumes a linear characteristic up to approximately 80 or 90 percent of flow. At that point the modified linear characteristic behaves like the square root characteristic up to full open position.

5.3.3.4 EQUAL PERCENTAGE FLOW CHARACTERISTIC. An equal percentage flow characteristic, alternatively called exponential or logarithmic, is one in which an equal percentage change in flow is produced as a result of unit change in lift (opening). For this type of control characteristic a 10 percent change in opening between 10 and 20 percent open would have the same percentage effect on flow as a 10 percent change in opening between 60 and 70 percent open. The general equation expressing flow as a function of valve opening for an equal percentage control valve is

$$Q = ae^{by} \qquad \textbf{(Eq 5.3.3.4)}$$

where
- Q = flow rate
- y = valve opening
- a, b = arbitrary constants
- e = base of natural logarithms

A plot of flow versus valve opening for an equal percentage characteristic is a straight line on semi-log graph paper with flow as the log ordinant, and is an exponential curve on rectangular coordinate graph paper.

5.3.3.5 SQUARE ROOT (QUICK OPENING) FLOW CHARACTERISTIC. A square root flow characteristic is one in which flow varies with the square root of the control element position. The general equation expressing flow as a function of valve opening is

$$Q = ay^{\frac{1}{2}} \qquad \textbf{(Eq 5.3.3.5)}$$

where
- Q = flow rate
- y = valve opening
- a = arbitrary constant

This type of characteristic is more commonly referred to as quick opening.

5.3.4 Inherent Versus Effective Flow Characteristics

Control valves are sometimes installed in systems with little thought as to the effect adjacent plumbing will have on the valve performance. It is the effective flow characteristic of a control valve installed in a system rather than its inherent characteristic determined under conditions of constant ΔP which is of ultimate importance in a system. Under system conditions when valve pressure differential does not remain constant over the throttling range, the effective flow characteristic curve will vary from the inherent valve flow characteristic. The effective flow characteristic of a valve, therefore, is dependent upon the flow pressure drop characteristics of the system in which it is installed. In order that a control valve provide a good flow regulating characteristic, full open pressure drop must be compromised since the best throttling characteristic is achieved by a valve having a resistance coefficient, K, over the larger portion of its travel higher than that of the system.

The case of a fixed system restriction is series with an inherently linear control valve serves to illustrate this phenomenon. In this case, a linear system characteristic will be obtained only when 100 percent of the system pressure drop is absorbed by the control valve.

As the total percentage pressure drop decreases across the inherently linear valve, the effective flow characteristic diverges markedly from being linear. Figures 5.3.4a,b show the inherent and effective flow characteristics of the linear and equal percentage type control valves in series with a fixed restriction. In general, as system pressure drop in series with the valve increases, the effective flow characteristic of a linear valve approaches that of a quick-opening valve, and the effective flow characteristic of an equal percentage valve approaches that of a linear valve (Reference 206-1).

Additional references treating the effect of system pressure drop on valve flow characteristics are 23-38, 27-7, 27-19, 160-20, 165-22, and 206-1.

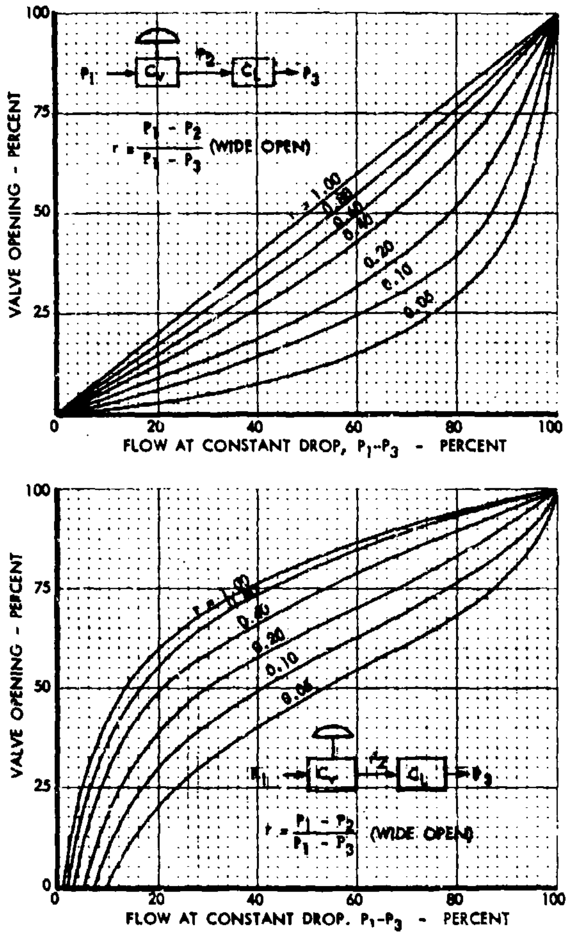

Figure 5.3.4a,b. Effective Flow Characteristics (a) Linear Valve (b) Equal Percentage Valve

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5.3.5 Types of Control Valves

A variety of geometric configurations exist for control valves, depending upon the desired flow characteristic, system configuration, etc. In general, there are seven basic control valve categories into which control valves of major importance can be grouped. Following is a listing of the major control valve types with descriptions of their more important features.

5.3.5.1 GLOBE AND ANGLE VALVES. The most commonly used control valves fall into the globe body and angle valve category. The valving element, seat, and bonnet are generally identical in both globe and angle body types. The internal configurations of typical globe body and angle body valves are shown, respectively, in Figures 5.2.5.1f and 5.2.5.1e.

The characteristics of globe and angle valves generally fall under one of the following types:

a) square root or quick opening

b) linear

c) modified linear

d) equal percentage.

Quick-Opening Valve. Quick-opening globe valves are often poppet valves, generally having either a conical plug or flat poppet-shaped valving element as shown in Figure 5.3.5.1a. Quick-opening valves reach full flow capacity with relatively short stem travel because the annular space between the seat ring and plug becomes equal to the area of the entire port with a travel of approximately one-quarter of the seat diameter. The difference in flow characteristics between two configurations of quick-opening poppets is shown in Figure 5.3.5.1a, which shows the linear characteristics of the flat poppet over practically its full range of travel, and the linear characteristic for the conical plug over only about 70 percent of its useful travel.

Quick-opening valves are used extensively for pressure regulation. They are also adaptable for ON-OFF control or for systems with a constant pressure which do not require frequent valve position changes.

Linear Characteristic Valve. A linear flow characteristic in a globe valve can be obtained by using a parabolic plug* or a rectangular port plug as shown in Figure 5.3.5.1b. This type of design is used when a large portion of the pressure drop in a fluid system is to be taken across the control valve. For maximum C_v, the parabolic plug is recommended, because for a given size of valve and seat ring the full open flow of the shaped plug is greater than that of the rectangular port design.

Modified Linear Characteristic Valve. As discussed in Detailed Topic 5.3.3.2, a modified linear flow characteristic is a characteristic which initially follows a parabolic curve, then changes to a linear flow versus stroke curve. The modified linear flow characteristic can be achieved either by modifying a linear plug (paraboloid) or a V-port plug. The modified linear characteristic is illustrated in Figure 5.3.5.1c.

Both V-port and parabolic plug shapes may be used in valves with widely varying stem travel. The most valuable feature of the modified linear characteristic type of valve is that at low flow rates the stem must travel a relatively

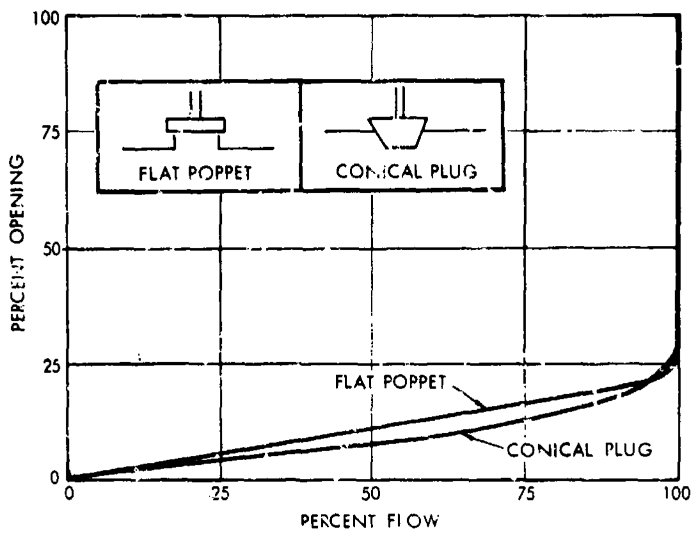

Figure 5.3.5.1a. Flow Characteristics of Quick-Opening Poppet Valves

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

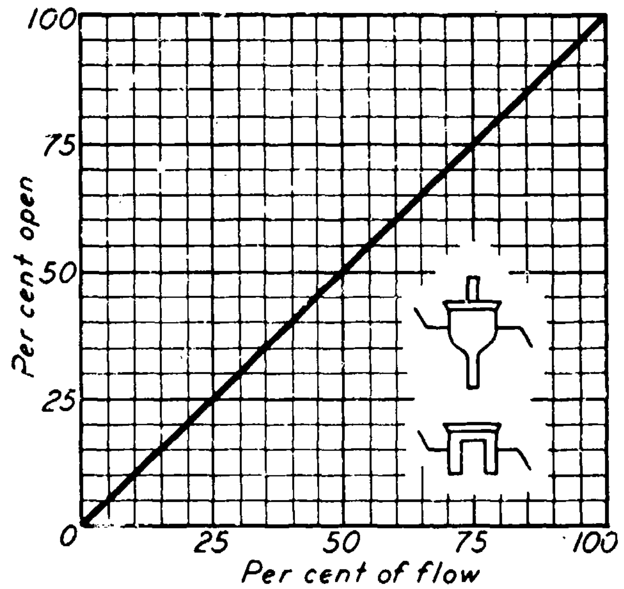

Figure 5.3.5.1b. Linear Flow Characteristics of a Rectangular Port and Parabolic Plug

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

*It should be noted that although the parabolic plug, in the strictest sense, is a plug having the geometry of a paraboloid, the term is often used to describe any plug having the general appearance of a paraboloid.

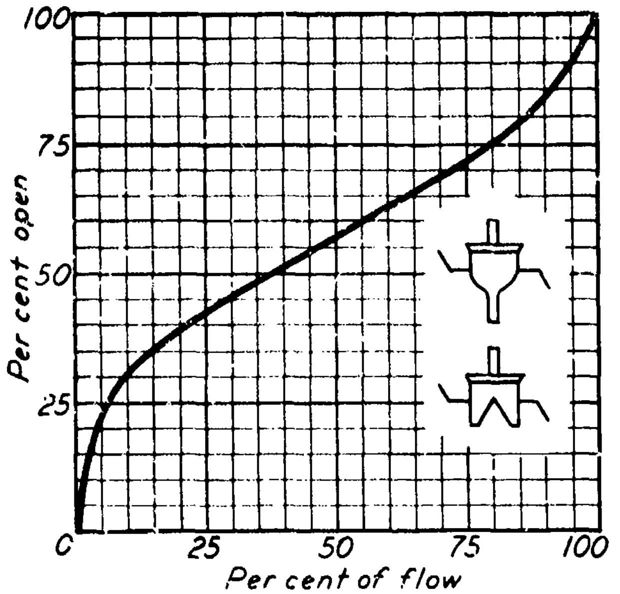

Figure 5.3.5.1c. Modified Linear Flow Characteristics of a Modified Linear Plug and Modified V-Port Plug

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

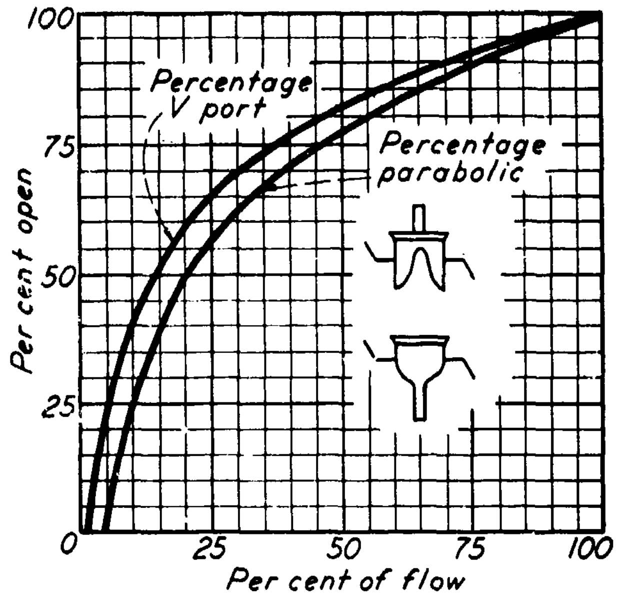

Figure 5.3.5.1d. Flow Characteristic of an Equal Percentage Valve

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

large distance to obtain a small change in flow; consequently, accuracy of flow control in the low range is possible. For example, 30 percent stem travel is required for just 10 percent of flow control in a typical modified linear valve. At higher flow rates, however, the valve assumes a linear characteristic which allows for larger changes of flow for each change of stem movement. This characteristic is useful in that it allows for closing down to low flows quickly. Valves with modified linear characteristics can be designed also for relatively low lift, although the rangeability of such valves is limited. Referring to Figures 5.3.3 and 5.3.5.1c, it should be noted that control sensitivity of the modified linear characteristic is decreased at the high end of flow as well as at the low end.

This is a disadvantage at high flow rates where it may be desirable to have as high a gain or sensitivity as at midrange. To overcome this disadvantage, another type of valve plug configuration has been designed, called the equal percentage characteristic valve.

Equal Percentage Characteristic Valve. The equal percentage characteristic valve is the most commonly used valving element in commercial globe type control valve applications. Equal sensitivity is provided over the working range of the valve, i.e., a given change in valve stroke always results in the same percentage change of flow rate.

The equal percentage valve plug is a further modification of the V-port and parabolic plugs, offering still another variety of flow characteristics as illustrated in Figure 5.3.5.1d. This configuration, which is also termed a logarithmic plug, was developed in an effort to obtain good control over a wide range of flow and pressure changes. In many flow systems, variables in the system fluctuate in a nonlinear manner. For example, non-linearities occur when line pressure decreases considerably with flow changes from low flow to maximum demand. Under these conditions, the pressure drop across the valve may be a small percentage of the system drop at maximum flow, but a large percentage at small flows. The equal percentage valve makes it theoretically possible to control over a wide range of flow changes with one size valve.

5.3.5.2 NEEDLE VALVES AND SPLINE PLUG VALVES. Needle and spline plug valves are variatons of standard globe type valves. The control orifice is smaller in size and the valving element is long and slender to permit close throttling. When it is desired to control flows of a very low rate it is difficult to obtain accuracy with the common designs of plug and poppet control valves, since these designs are seldom available with a C_v of less than one. To fill the need for accurately controlling low flows, the designer usually turns to needle valves or their equivalent. Figure 5.3.5.2 illustrates a typical needle valve, showing the characteristic long slender tapered pintle from which its name is derived. Needle valves are used for such things as controlling the flow of small quantities of chemicals in laboratory analysis work, for careful blending of fluids and gases, and for speed control of pumps and actuators. Needle valves are chosen for the purpose of close control because the shape of the valve plug gives fine flow control for a relatively long stem travel.

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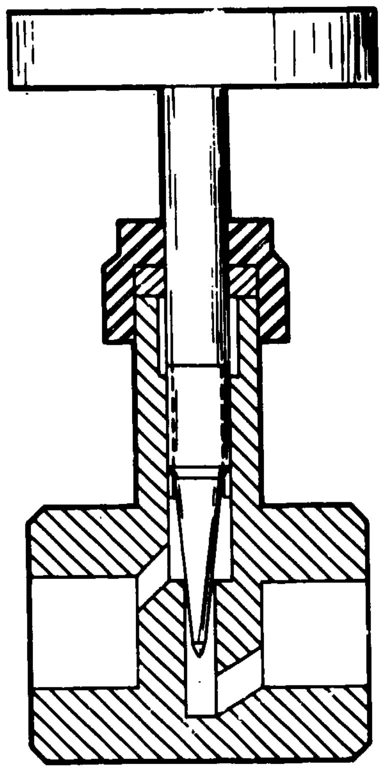

Figure 5.3.5.2. Needle Valve

A close relative of the needle valve is the spline plug valve. This valve can be made with flow capacities even lower than needle valves of comparable sizes. They can be made having equal percentage characteristics or linear characteristics.

The construction feature which allows for such fine flow control is the plug, which moves inside the seat ring and has long tapered slots milled and ground into its surface. This plug remains inside the seat ring at all times, even in the wide-open position, so the flow through the valve is closely controlled by the relative position of the long plug and gradually tapering plug grooves.

Because the flow forces can be relatively high in spline and fluted plug valves, the effects of wire drawing and erosion increase. To minimize these effects, materials of suitable hardness must be specified. In most cases, needle valves and their spline and fluted plug relatives are manually operated; and, because of the relatively small seat diameters, tight shutoff can be achieved. Care must be taken not to overtighten these valves, however, as seat galling and plug deformation will result.

Tight shutoff is usually as important as throttling where a needle valve is required. The degree of seat tightness in metal-to-metal valves seats (commonly found in needle valves) depends on the concentricity between the valving element (plug) and the seat (usually integral with the body) at the point of closure. A good finish on the seat and plug is equally important in achieving good shutoff. For repeated actuation, the best assurance of good sealing is a rubber or plastic disc closing against a metal seat. If a non-metal seal is out of the question due to extreme temperature requirements or the handling of a corrosive fluid, either one or both of the following techniques will aid in achieving good sealing (Reference 54-1):

a) Design a non-rotating plug for closing against the seat.

b) Use a device that limits the closing force applied to the stem to that required for tight shutoff and no more.

Accessories available with needle valves include locking devices to secure the needle in any selected position, and vernier dials that permit accurate repositioning of the stem if required. Needle valves are usually used in line sizes of less than one inch with one-quarter inch being a typical size. These valves are used in temperature ranges from cryogenic to over 1000°F, and pressure ranging from vacuum to over 30,000 psi.

5.3.5.3 CAVITATING VENTURI CONTROL VALVES. One type of control valve which is finding increasing use, particularly in the rocket propulsion field, is the cavitating venturi valve. A cross-section through a valve of this type used to control the flow of propellants to a rocket engine is shown in Figure 5.3.5.3. The cavitating venturi valve con-

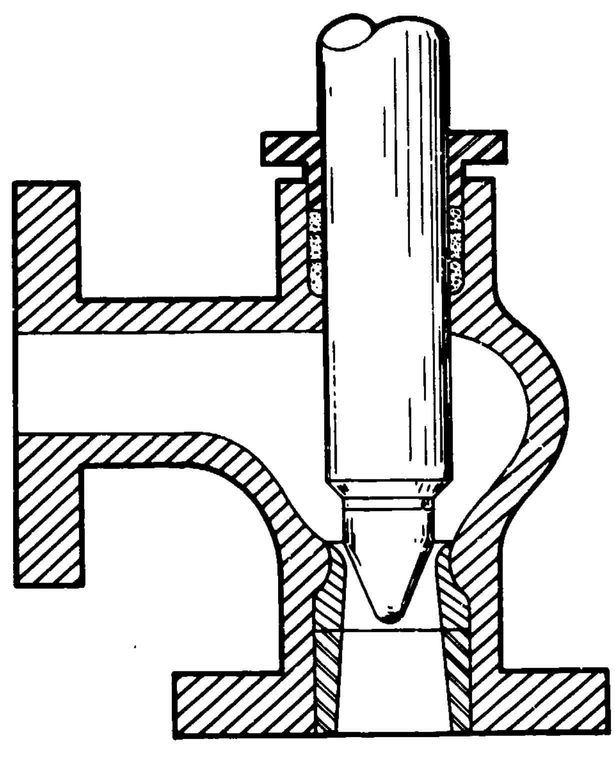

Figure 5.3.5.3. Cavitating Venturi Valve

trols flow as a function of upstream pressure and throat area, with flow being independent of downstream pressure when the fluid is cavitating. Cavitation occurs by increasing the pressure drop across the venturi section until a point is reached where substantially all of the upstream pressure head is converted into velocity head in the venturi throat. When this occurs, the only static head remaining in the throat is that of the fluid vapor pressure; and, if the upstream pressure is maintained constant, a further drop in downstream pressure will not result in increased flow, since vaporization (cavitation) at the throat maintains the pressure at the fluid vapor pressure. In effect, the flow through the valve will remain at a constant rate for a given throat area and upstream pressure regardless of the downstream pressure, providing that the downstream pressure is not increased above the point where cavitation ceases. As a general rule, the downstream pressure must be less than 85 percent of the upstream pressure for cavitation to occur. This value depends on the diffuser efficiency, decreasing with decreasing efficiency. Cavitating venturis are useful in rocket propulsion systems, where it is desirable to prevent fluctuations in flow rate caused by fluctuating back pressure on the control valve. Compensation for changes in system pressure is simplified with cavitating venturi valves as corrections are required for upstream pressure changes only. In the design of a cavitating venturi valve which is to be used for flow control purposes, it is first necessary to know the total pressure available and the vapor pressure of the liquid to be controlled. The difference between these two pressures will be the total static head available for conversion in the venturi valve to velocity head. This differential head can be used to find the theoretical velocity of the fluid stream at the venturi

$$P - P_v = \frac{wV_o^2}{2g} \qquad \text{(Eq 5.3.5.3a)}$$

where P = total pressure, lb_f/ft^2

P_v = fluid vapor pressure, lb_f/ft^2

w = specific weight, lb_f/ft^3

V_o = theoretical throat velocity, ft/sec

g = gravitational constant, ft/sec^2

The actual flow rate through the venturi can be determined by assuming a discharge coefficient, C_D, of 0.93. The flow rate equation is

(Eq 5.3.5.3b)

$$Q = C_D A_o V_o = C_D A_o \sqrt{\frac{2g}{w}(P - P_v)}$$

where A_o = throat area, ft^2

Q = flow rate, ft^3/sec

As determined experimentally (Reference 2-4), best pressure recovery through a cavitating venturi is obtained with a divergence angle in the range of 5 to 6 degrees. If the design is space limited, it is better to make the convergence angle more abrupt and retain a maximum diffuser divergence angle of 6 degrees. If the diffuser section must be shortened, it is best to terminate it abruptly at its downstream extremity rather than to use a wider divergence angle.

5.3.5.4 DIAPHRAGM AND PINCH CONTROL VALVES. Diaphragm and pinch valves which are described under "Shutoff Valves" in Sub-Topic 5.2.8, used as throttle valves provide high accuracy and repeatability. Throttling is accomplished by gradually lifting the diaphragm, making a variable flow area which effectively controls the flow rate. Figure 5.3.5.4 shows the flow characteristics of a typical pinch valve. Diaphragm valve flow characteristics are very similar to pinch valve characteristics.

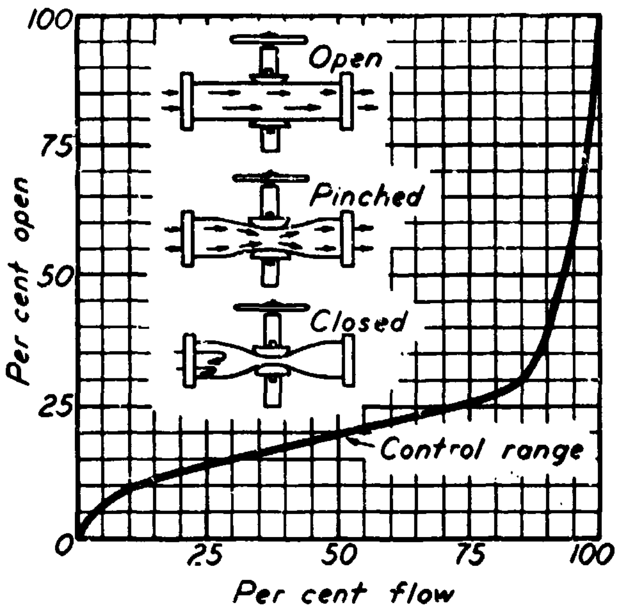

Figure 5.3.5.4. Pinch Valve Flow Characteristic
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

5.3.5.5 BUTTERFLY VALVES. The use of butterfly valves as control valves has evolved from devices known as dampers which were originally used to control flow of combustion air into the firebox of a boiler, or to regulate air flow in ducts and tunnels produced by fans and blowers. They are still used in this capacity; but in recent years refinements have been made, resulting in high precision capacity butterfly control valves. Butterfly valves range in size from 2 to 36 inches and larger. Sub-Topic 5.2.4 gives a detailed discussion of the construction of butterfly valves. Therefore, this section will be devoted to the discussion of butterfly valve flow control characteristics and the associated operating torque characteristics.

Butterfly Valve Flow Characteristics. A typical flow characteristic curve for a butterfly valve is shown in Figure

5.3.5.5a. Typically, a butterfly valve has an equal percentage characteristic for approximately the first 50 percent of opening which controls approximately 20 percent of the flow, and a linear characteristic for the remainder of the opening. In most control valve applications, the 100 percent flow condition is taken at approximately 60 degrees of the butterfly disc, since the percentage increase in flow beyond this point is usually small. The contour or shape of the butterfly disc has little influence on the flow characteristic for less than 60 degrees of valve opening, as illustrated in Figure 5.3.5.5b. This is a plot of resistance coefficient as a function of butterfly angle, showing the effects of varying butterfly disc thickness. It should be noted that the curve is plotted for a valve which is fully closed when $\alpha_o = 0$. For valves which close at some angle $\alpha_o > 0$, the corrected butterfly angle can be estimated from the following relationship for $\alpha_o < 20°$:

(Eq 5.3.5.5a)

$$\alpha = \cos^{-1}[\cos(\alpha_o + \alpha_1)/\cos\alpha_o]$$

where α_o = closing angle

α_1 = valve opening from the closed position.

In addition to flow considerations, the 60-degree open position is usually a quite stable position for the disc, whereas opening beyond 60 degrees puts the disc in an area where rapid changes in fluid dynamic torque could result in valve instability.

Butterfly Valve Torque Characteristics. The operating torque required to position the disc of a butterfly valve results from a combination of seal friction, bearing friction, non-symmetrical static pressure loading around the butterfly axis, and disc fluid dynamic torque. Seal friction is usually maximum during the first few degrees of disc opening, dropping rapidly as the disc starts to move. Bearing friction is a direct function of the total unbalanced pressure force across the butterfly disc. This unbalanced pressure force is a function of the Δp across the disc and the projected area of the disc in the flow path. Since both Δp and disc projected area decrease with valve opening, bearing friction varies from a maximum to a minimum as the valve rotates from closed position to full open position.

Non-symmetrical static pressure loading results from a disc having a non-symmetrically located axis of rotation, i.e., an axis of rotation which does not equally divide the disc frontal area. This condition can be intentionally designed into the valve by offsetting the disc axis from the disc centerline. It can also result from manufacturing tolerances or it can be caused by a floating disc seal which, by sliding toward the trailing edge of the disc, can result in a larger effective area downstream of the axis than on the upstream surface. It is desirable to have the torque resulting from static pressure unbalance act in a direction to close the valve; however, this is not always possible, particularly when the unbalanced force is caused by a floating seal.

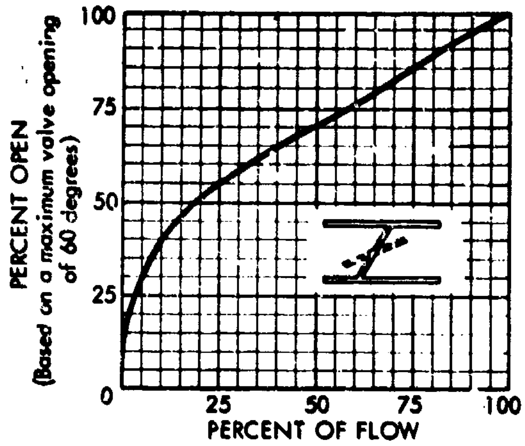

Figure 5.3.5.5a. Butterfly Valve Flow Characteristics
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

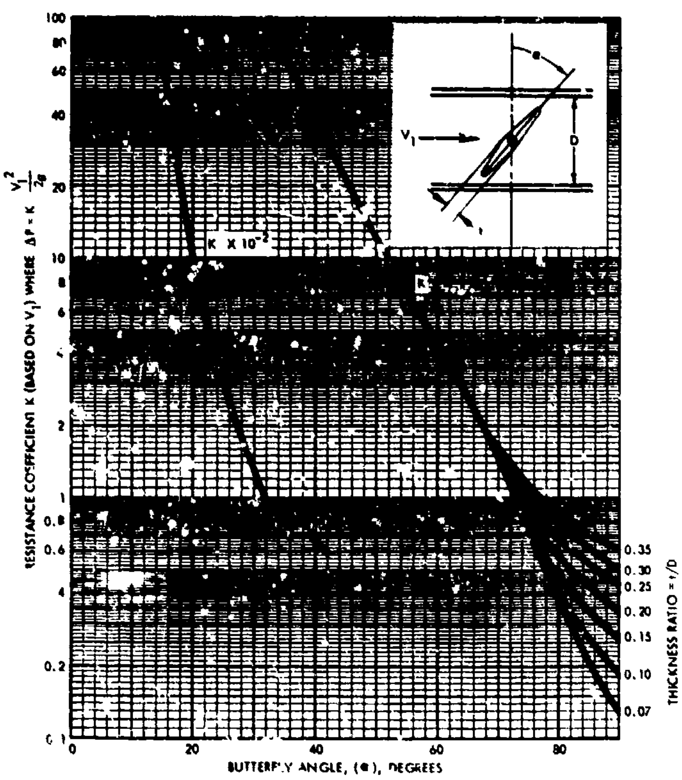

Figure 5.3.5.5b. Butterfly Valve Resistance Coefficient as a Function of Butterfly Angle
(Adapted from "Machine Design," December 1958, vol. 30, no. 26, D. Dahle, Copyright 1958, Penton Publishing Company, Cleveland, Ohio)

The undesirable feature of having this torque act in a direction to open the valve is that it opposes the fluid dynamic torque which always acts in a direction to close the valve. With these two torques opposing one another, a torque reversal can occur during the first few degrees of opening which can result in undesirable actuation characteristics. The fluid dynamic torque results from the fact that fluid velocity over the disc surface is always higher on the trailing edge of the disc than on the leading edge, as illustrated in Figure 5.3.5.5c. This velocity distribution results in a Bernoulli effect, where the local static pressures are lower across the downstream disc surface than on the upstream surface where the fluid velocity is lower. The resulting torque always acts in a direction to close the valve. Fluid dynamic torque for disc angles up to approximately 60 degrees is given by the following empirical relationship:

$$T = K \Delta P D^3 \qquad \text{(Eq 5.3.5.5b)}$$
$$\alpha \leq 60$$

where T ...

K ... y constant

ΔP ... ure differential across disc (for ... le flow when $P_2 < P_c$, $\Delta P =$ ... ere P_c = downstream pressure ... low)

D ... r

α ... tion from closed position

A range of measured fluid dynamic torque values for several butterfly valve types plotted as a function of α is given in Figure 5.3.5.5d. A representative fluid dynamic torque curve illustrated by the middle curve is shown in Figure 5.3.5.5e. The upper curve indicates the total torque required to open a symmetrically mounted butterfly disc (the sum of the fluid dynamic torque and bearing friction torque). The effects of seal friction or an unsymmetrical static pressure loading are not shown in the illustration. The lower curve shows the required closing torque which is the difference between the fluid dynamic torque and the bearing friction resisting closing. If there were no bearing friction, the closing torque measured at any butterfly angle would be a resisting torque equal to the fluid dynamic torque. As the bearing friction torque opposes the fluid dynamic torque, a point is reached in rotating from the open-to-closed position where the bearing friction torque just balances the fluid dynamic opening torque. In the illustration, this torque reversal occurs at approximately 33 degrees. This torque reversal on closing is a typical butterfly valve characteristic which can cause instability of the valve at the position where the torque reversal occurs, because at this point the direction of applied actuator torque must be completely reversed. For a valve having a static pressure unbalance acting to open the disc, it is possible to have a torque characteristic with two positions of torque reversal. Such a condition could result where, first, a positive opening torque is required to overcome seal and bearing friction. Then, when the valve opens, a torque reversal occurs due to pressure unbalance which tends to swing the valve open. As the

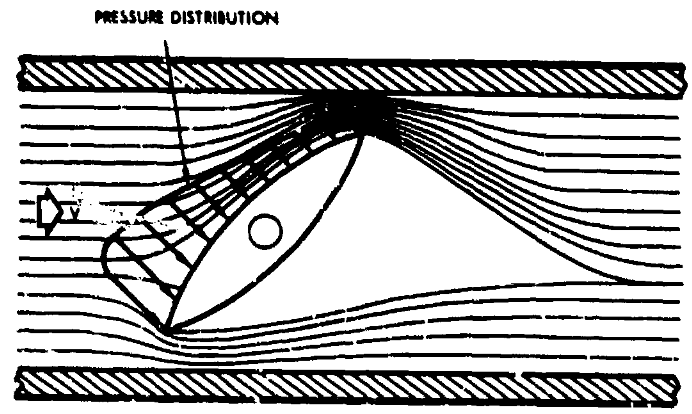

Figure 5.3.5.5c. Butterfly Flow and Pressure Distribution

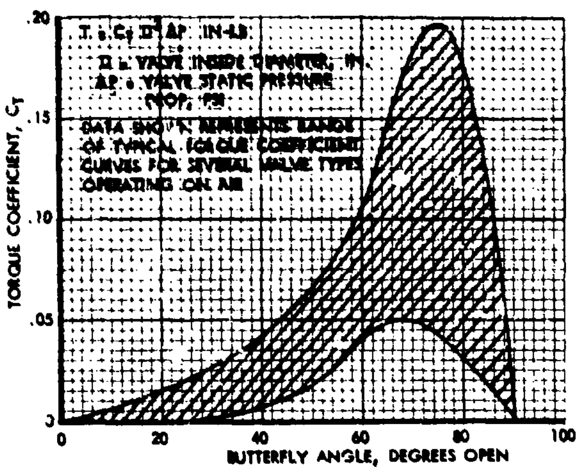

Figure 5.3.5.5d. Butterfly Valve Aerodynamic Torque Characteristic

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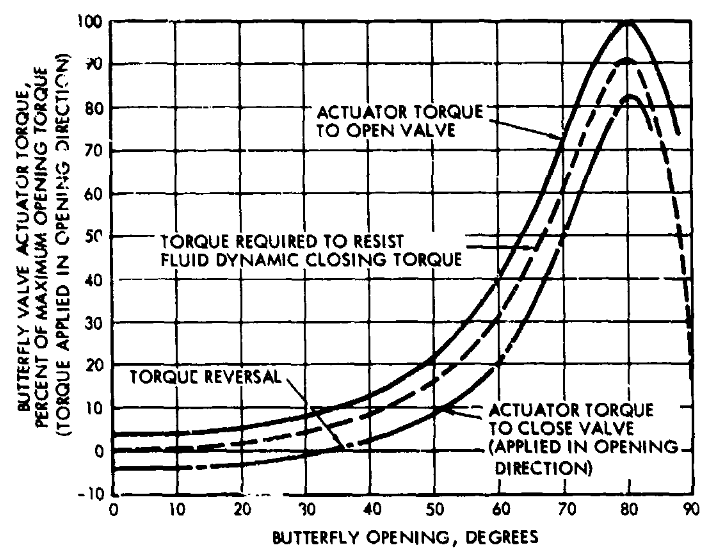

Figure 5.3.5.5e. Fluid Dynamic Torque Curve for a Typical Butterfly Valve

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

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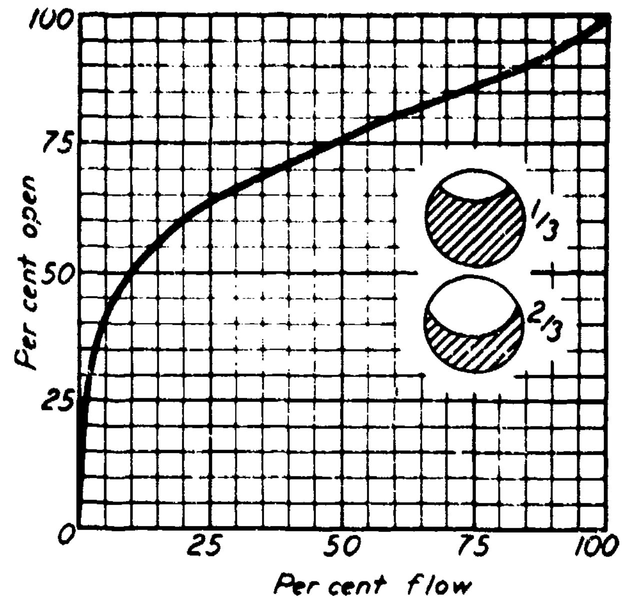
valve continues to open, the effect of pressure unbalance diminishes and a counteracting fluid dynamic torque eventually causes a positive actuator torque.

Due to the problems of instability resulting from torque reversals in the first 20 to 30 degrees of opening, certain butterfly valve designs have a limited low-flow throttling capability. It is possible to eliminate torque reversals by designing the butterfly disc to seal at some angle less than that angle which is perpendicular to the duct centerline. The seating angle required to avoid torque reversal depends on the shape of the disc, and usually approximates 30 degrees, leaving a total valve stroke of only 60 degrees for control purposes. Another technique for avoiding torque reversal is to design the disc with a larger area on the upstream side of the axis of rotation than on the downstream side (Figure 5.2.4.1c). If the resulting closing torque is great enough to overcome bearing friction and closing seal friction, the valve will experience no torque reversal. The entire question of butterfly valve actuating torque is discussed in detail in Detailed Topic 6.2.3.1.

5.3.5.6 SPOOL, PISTON, AND SLEEVE CONTROL VALVES (Cylindrical Slide Valves). Spool and piston control valves consist of a movable piston or spool (a piston with more than one land) within a cylinder. In sleeve valves, the solid piston or spool is replaced by a hollow cylinder with either the inner or outer cylinder serving as the valving element (the moving part). The great majority of cylindrical slide valves utilize axial motion of the valving element, although some design for special applications have used rotating pistons or sleeves. With cylindrical slide valves any type of flow characteristic is obtainable, depending upon the configuration of the ports and/or the valving element. A primary advantage of these control elements is the feasibility of obtaining a pressure-balanced design, especially with sleeve or spool configurations. An inherent disadvantage of cylindrical slide valves is leakage, a problem which can only be controlled by close machining or reliable dynamic sealing techniques. Spool valves, for example, are widely used in fluid power applications wherein perfect internal sealing is not required. Such multiple passage control valve applications are covered in detail in Sub-Section 5.6, "Servo Valves," and in Sub-Section 5.8, "Multiple Passage Valves." More detailed discussions of these valve types may be found in Sub-Section 6.2, "Valving Units."

5.3.5.7 ROTARY PLUG CONTROL VALVES. Rotary plug valves consist basically of a body into which a plug is inserted. Figure 5.2.11.1 of Sub-Section 5.2 illustrates a V-ported plug valve. The plug, which is the valving element, has a hole through it so that a 90-degree rotation of the plug will either open or close the valve. The hole in the plug can take various shapes, but the two most commonly used are circular and V-shaped. The circular and V-ported plugs have characteristics as illustrated in Figure 5.3.5.7a,b. Rectangular ported plugs have been used, but have high entrance and exit losses to and from the rectangular port in a circular line.



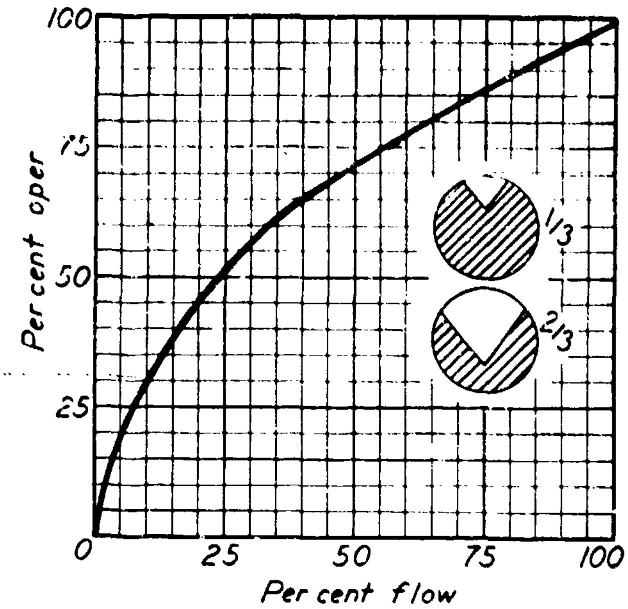


Figure 5.3.5.7a,b. Characteristics of Rotary Plug Control Valve

Plug Cock (a) has port area with circular hole in plug which varies as two converging circles. Flow characteristic is linear over last third of lift, and controls three-fourths of the flow in this linear region. Plug cock (b) with V-port has characteristic that approaches that of the equal-percentage type, providing better control of flow and high flows than plug cock with circular port.

Rotary plug valves generally have positive shutoff capability, and those with straight-through circular ports have a very low pressure drop in the full open position. The major disadvantage of the rotary plug is that considerable torque may be required for positioning because of a tendency for the plug to become too tightly seated in the valve bore. This disadvantage can be minimized by careful selection of materials and proper care and servicing, along with sufficient actuator power. Rotary plug valving units are discussed in detail in Detailed Topic 6.2.3.3.

5.3.6 Control, Actuation, and Positioning

A flow control valve requires an intelligence circuit to govern the valve output. In general, this intelligence circuit is comprised of a sensing element, a controller, an actuator, and a positioner. The sensing element and controller can include a variety of combinations, such as temperature probes and amplifiers, level sensors and amplifiers, bridge circuits, pneumatic regulators, etc. In a control or throttle valve used in a variable thrust rocket system, the sensing element is an accelerometer and the controller is the guidance system. The actuator is basically some mechanism that operates the control valve valving element. The valve actuator is normally external to the valve element and, in some instances, separated from the valve body. Actuator types include electrical, pneumatic, and hydraulic as well as combinations of these. A detailed discussion of actuators can be found in Sub-Section 6.9. One element in the intelligence loop that is inherent to the control valve is the positioner. A detailed discussion of this element follows.

5.3.6.1 CONTROL VALVE POSITIONERS. The control valve positioner is the element that imparts sensitivity to the control valve and achieves the valve positioning required by the control signal. Inherent in a valve positioner is a feedback loop between the control signal to the valve and the valve stem position.

Control valve positioners whether electrical, pneumatic, or hydraulic consist basically of an actuator which moves the valve stem in response to a control signal, and a stem position indicator which is fed back to the actuator input where it is compared with the control signal. When the control and position signals agree, an equilibrium position for the control valve has been established.

The pneumatic positioner basically assumes two forms, (1) position-balance (or deflection balance) type and (2) force-balance type. Figures 5.3.6.1a and 5.3.6.1b illustrate these types. In Figure 5.3.6.1a there are three basic elements: (a) the bellows, receiving the control signal; (b) the beam, one end fixed to the bellows; the other end fixed to the valve stem; and (c) the pilot valve which is connected to the beam In the force-balance type of positioner, a fixed pivot and a calibrated spring are added.

In the position-balance system a control signal causes a bellows deflection which in turn activates the pilot valve. The pilot either bleeds air from or admits air to the valve actuator, causing valve stem motion which in turn repositions the pilot to a neutral position.

In the force-balance system, the spring produces a force proportional to valve stem position. When the bellows receives a control signal, the pilot is activated as before and is returned to equilibrium (neutral position) when the spring balances the force produced by the bellows.

Additional information on valve positioners may be found in the following references:

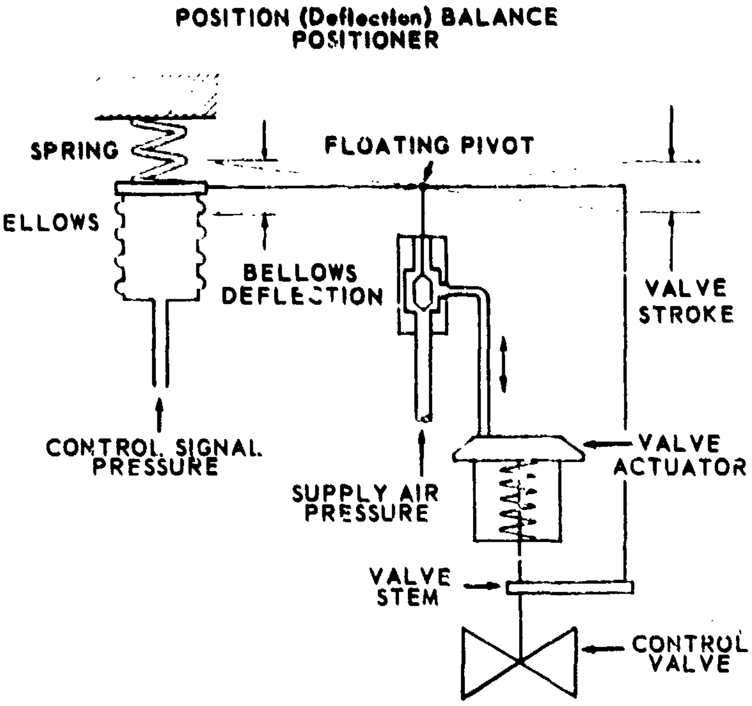


Figure 5.3.6.1a. Position Balance Pneumatic Positioner
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

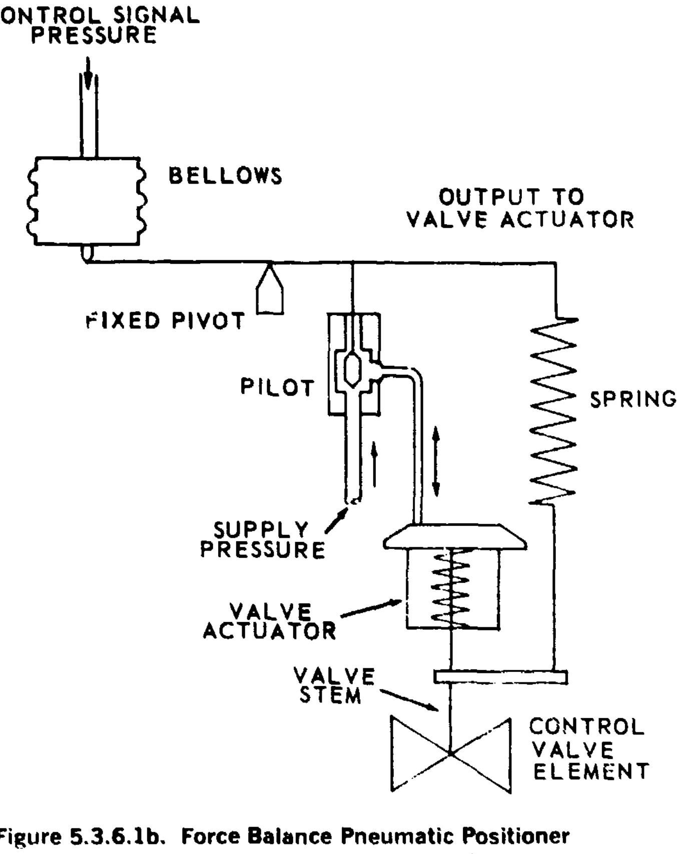


Figure 5.3.6.1b. Force Balance Pneumatic Positioner
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

General Information—References 160-52 and V-217
Pneumatic Positioners—Reference 52-5
Positioners for Diaphragm Actuators—Reference 160-6
Positioners for Cylinder Actuators—Reference 160-4.

5.3.7 Control Valve Pressure Compensation

In certain applications it is desirable that a controlled flow rate be fixed by control valve closure position independent of system pressure variations. Such an application is a control valve (throttle valve) for a variable thrust rocket motor where it is important that the rocket thrust, hence propellant flow rate, remain constant for any given throttle position in spite of possible pressure variations occurring both upstream and downstream of the control valve. Figure 5.3.7 illustrates a technique for providing control valve pressure compensation. Changes in upstream and/or downstream pressure are sensed by a spring loaded piston valve. This variable upstream restriction (the piston valve) varies the pressure upstream of the control valve plug such that the pressure drop across the control element remains constant. Another technique used for pressure compensation is through the use of an actuator controller which senses Δp across the control valve and makes corresponding corrections by changing the flow area through repositioning the actuator. The latter technique is also used to provide density compensation where it is desired to maintain a constant mass flow rate. Density compensation is accomplished by sensing fluid temperature variations which become an actuator controller input.

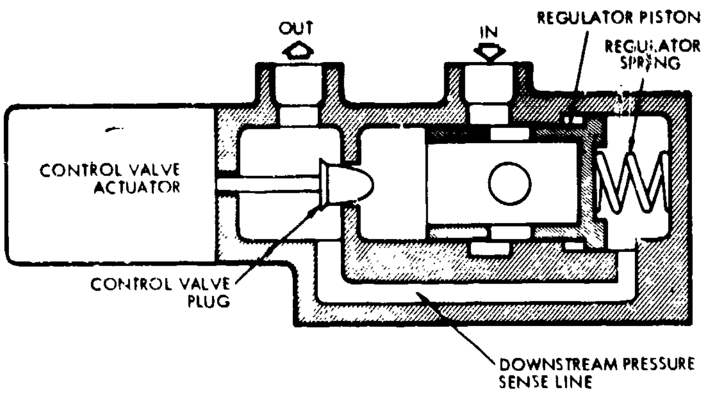


Figure 5.3.7. Pressure Compensated Flow Control Valve

REFERENCES

1-42	68-4	165-22
23-38	112-5	193-24
26-122	144-1	195-7
27-6	160-4	201-1
27-19	160-6	206-1
27-36	160-18	297-1
52-5	160-20	408-1
54-1	160-52	V-217
66-7	165-3	

5.4 PRESSURE REGULATORS

5.4.1 Introduction

A pressure regulator is a control valve that uses no auxiliary source of power during operation (Reference 144-2). It controls pressure by varying flow as a function of the sensed difference between the actual and the desired value of pressure. Any unbalanced force, resulting from the pressure difference, moves a metering element, increasing or decreasing fluid flow to nullify the pressure error.

The name *regulator* is not universally used, such terms as *pressure-reducing valve* and *self-operating controller* being used synonymously. In the space field, the term *pressure regulator* is generally accepted as meaning any device which maintains a predetermined upstream, downstream, or differential pressure by means of a pressure reducing control element.

Pressure regulators are used in a wide variety of industries and applications. In domestic use, regulators reduce line pressure to safe limits for household appliances. In manufacturing industries, regulators perform such functions as pump control, pressure reduction, and liquid level control. In aerospace systems, one of the most common applications of pressure regulators is to provide a constant pneumatic pressure in a liquid storage tank. This pressure is used either for direct expulsion of the liquid at a desired flow rate, or in a pump fed system to maintain a positive gage pressure in the tank for purposes of safeguarding the tank structure or to maintain pump suction head requirements.

5.4.2 Types of Regulators

a) Pressure-reducing regulators: reduce upstream pressure to a predetermined downstream pressure regardless of upstream pressure variations. These regulators are the most common type used in airborne systems, and the major portion of the regulator Sub-Section is devoted to them. A special valve of this type is the zero governor which regulates gas at a pressure of 0 psig.

b) Back pressure regulators: measure and regulate upstream pressure regardless of the variation in outlet pressure. Those regulators designed to regulate pressure below atmospheric, such as in vessels being evacuated, are called vacuum regulators. Back-pressure regulators are relief devices, and are discussed in Sub-Topic 5.5.10.

c) Differential pressure regulators: maintain a constant differential pressure across a constant area restrictor (e.g., an orifice) for maintaining a constant pre-set flow rate. When used with liquids these regulators are often called flow regulators, and are discussed in Sub-Topic 5.4.15.

5.4.3 Pressure Reducing Regulators

5.4.3.1 GENERAL REQUIREMENTS FOR AEROSPACE REGULATORS. A survey of regulator applications and requirements for rocket systems is presented in Table 5.4.3.1. All are pressure-reducing regulators.

Table 5.4.3.1. Regul

System	Application	Regulating Stages	Set Point (psig)	Flowrate (lb_m/sec)	Orifice Size (in^2)	Accuracy (psi)	Accuracy (%)	Inlet Pressure Maximum (psig)	Inlet Pressure Minimum (psig)
Jupiter	Guidance	-	28 to 45	0.024	0.0086	± 0.2	± 0.45	3000	200
Jupiter	Roll Control	-	350	0.3	0.052	± 5.0	± 1.4	3000	500
Thor	Fuel Tank Pressurization	-	25	0.07	0.026	± 1.5	± 6	3500	700
Atlas	Fuel Tank Pressurization	-	59.9	0.6	-	± 1.5	± 2.5	3000	75
Atlas	Oxidizer Tank Pressurization	-	24.7	0.6	-	± 0.6	± 2.5	3000	75
Titan I	First Stage Fuel Tank Pressurization	-	12.0	0.1	0.045	± 0.5	± 4.2	3250	200
Titan I	First Stage Oxidizer Tank Pressurization	-	34.0	0.11	-	± 0.5	± 1.5	3250	200
Saturn S-IVB	Fuel Tank Pressurization	-	2[illegible]0	0.18	-	± 2	± 0.8	3100	400
Apollo Service Module	Reaction Control System	-	173	0.13 0.20	-	± 2	± 1.2	4500	250
Apollo Command Module	Reaction Control System (Primary) (Secondary)	Series Redundant	291 295	0.0050	0.0006	± 4.0	± 1.4	4500	400
Saturn S-IVB	Aux. Power System Tank Pressurization (Primary) (Secondary)	Series Redundant	196 200	0.0033	0.0017	± 3.0	± 1.5	3200	350
Nimbus	Attitude Control	Single	35	0-0.0063	0.12	± 2.0	± 5.7	3900	200
	Attitude Control	Series Redundant	380	5.1	0.35	± 20	± 5.3	4800	500
Surveyor	Fuel and Oxidizer Pressurization	Single	725	0.009	0.07	± 20	± 2.6	5200	820
	Fuel Cell	Single	1.0	6.13×10^{-7}	0.006	± 0.2	± 20	30,000	25
Mercury Gemini Apollo	Life Support	Single	80	0.004	0.08	± 10	± 12.5	7500	200
NASA Re-Entry	Electronics Pressurization	Single	1.50	0.00061	0.027	± 0.075	± 5	5000	200
Apollo	O_2 Regulator	Single	3.45	0.0007	0.035	± 0.2	± 5.8	7500	200
Ranger	Cold Gas Attitude Control	Single	15	0.0015	-	± 1.0	± 6.7	4000	150
Mariner	Cold Gas Attitude Control	Single	15	0.0015	-	± 1.0	± 6.7	4000	150
Mariner '69	Fuel Tank Pressurization	Single	308	0.006	0.0781	± 3.0	± 1.0	3100	400
OGO	Cold Gas Attitude Control	Single	50	0.003	-	± 1.5	± 3	4000	150
Lunar Orbiter	Cold Gas Attitude Control	Single	19.5	0.0027	-	± 1.0	± 5.1	4000	150
Pioneer	Cold Gas Attitude Control	Single	50	0.003	-	± 1.5	± 3	4000	150
Vela	Cold Gas Attitude Control	Single Leakage Redundant	50	0.0022	-	± 2.5	± 5	4000	200
Bio-Satellite	Cold Gas Attitude Control	Single	35	0.003	-	± 1.5	± 4.3	4000	150
Lunar Orbiter	Velocity Control Engine, Propellant Tank Pressurization	Single	180-200	0-0.0071	-	± 1.0	± 0.5	3820	650

Pressure Minimum (psig)	Inlet Temperature Maximum (°F)	Inlet Temperature Minimum (°F)	Ambient Temperature Maximum (°F)	Ambient Temperature Minimum (°F)	Minimum Ullage (ft^3)	Fluid	Sensing	Reference
200	125	-65	100	85	Very Small	Nitrogen	Internal	35-1
500	165	-65	165	-65	Very Small	Nitrogen	Internal	35-1
700	160	-100	160	-65	5	Helium	Remote by probe entering tank through regulator outlet line	35-1
75	450	-100	125	-65	8	Helium	External with 35 ft 1/4" sense line	35-1
75	450	-100	125	-65	-	-	External with 60 ft 1/4" sense line	35-1
200	290	-260	[illegible]	-35	25	Helium	External with 3 ft line	35-1
200	290	-260	160	-35		Helium	-	35-1
400	160	-423	160	-65	-	Helium	Internal	147-15
250	160	-65	160	-65	2	Helium	-	147-16
400	110	-65	150	+30	0.058	Helium	Internal	V-206
350	110	-65	160	-30	0.116	Helium	Internal	V-200
200	200	-60	200	-60	0.0012	Nitrogen	Internal	V-422
500	170	-100	170	-60	0.021	Freon No. 4	Internal	V-422
820	115	-150	115	-20	0.04	Helium	Internal	V-422
25	160	-100	160	-65	0.0001	Hydrogen	Internal	V-422
200	200	-80	200	-50	0.001	Oxygen	Internal	V-422
200	165	-80	165	-60	0.001	Helium	Internal	V-422
200	200	-80	200	-50	0.001	Oxygen	Internal	V-422
150	160	-10	160	-10	0.0029	Nitrogen	Internal	V-425
150	160	-10	160	-10	0.0029	Nitrogen	Internal	V-425
[illegible]	[illegible]	[illegible]	[illegible]	14	-	Nitrogen	Internal	131-36
150	140	-10	140	-10	0.0029	Nitrogen	Internal	V-425
150	160	-15	160	-15	0.0029	Nitrogen	Internal	V-425
150	160	-15	160	-15	0.0012	Nitrogen	Internal	V-425
200	140	-10	140	-10	0.0014	Nitrogen	Internal	V-425
150	160	-20	160	-20	0.0029	Nitrogen	Internal	V-425
[illegible]	[illegible]	[illegible]	85	[illegible]	-	Nitrogen	Internal	V-191

5.4.3.2 TYPES OF PRESSURE-REDUCING REGULATORS. Pressure-reducing regulators can be broadly classified as modulating or non-modulating. Within each of these categories, regulators can be classified as direct-acting or piloted. Other characteristics used in identifying regulators include loading method and control action. A list of basic types of regulators is presented in Table 5.4.3.2. The terms used in regulator classification are defined as follows:

Modulation describes the ability of a regulator to achieve any steady-state flow rate necessary to maintain a constant regulated pressure. Most pressure regulators are designed for modulating flow control. A non-modulating regulator is simply an ON-OFF device as discussed in Sub-Topic 5.4.13.

Piloting is the use of a small flow control device operated by a small actuation force to control indirectly a large flow requiring a large actuation force.

Loading is the method of achieving a reference force to establish the regulator set point. The reference force is applied to the metering element such that it is opposed by the regulated pressure acting on an attached sensing element.

Regulator control action is defined as the relation between a change in metering valve position for a given change in the pressure being controlled. Control action may be two-position (ON-OFF), proportional, reset (integral), or proportional plus reset (Sub-Topic 5.4.13).

5.4.3.3 THE THERMODYNAMIC PROCESS OF GAS FLOW THROUGH A REGULATOR.* Figure 5.4.3.3.a shows a schematic of a regulator operating between two reservoirs, 1 and 7. Reservoir 1 would be the gas storage tank and Reservoir 7 the propellant tank. Figure 5.4.3.3b is a temperature-entropy diagram of the thermodynamic processes involved for the conditions of a pressure ratio (P_{o5}/P_{o1}) involving supersonic flow. Gas at reservoir pressure, P_{o1}, expands isentropically to Station 2, which is the isentrope 1-2 on the graph. From Station 2 to 3 is a constant area duct with friction and resultant pressure loss, following the Fanno line 2-3, representing the process (Sub-Topic 3.7.4). Mach number, M, has increased from point 2 to 3. Gas enters the regulator at 3 and expands isentropically through the orifice 4 down to some pressure p_9. The line 3-4-9 is, therefore, an isentrope passing through Mach 1 to a supersonic condition 9. From here the gas returns through shock waves and turbulence, to pressure p_5. The process 9-5 passes through non-equilibrium states, and can be represented by a wavy line indicating an unknown path. The pressure measured and maintained by the regulator is P_{o5}, which in turn affects p_5. If the pipe 5-6 were the same size as 2-3, points 5 and 6 should fall on the same Fanno line as 2 and 3, starting at the intersection of 2-3 with the wavy line. However, the area of the pipe 5-6 is greater than 2-3, so the process continues on another Fanno line. The final stage of the process is the compression to the tank at 7, where a new reservoir pressure is reached. In a typical application, there would also be liquid flowout through 8, corresponding to propellant flow from a tank. If the pressure ratio, P_{o5}/P_{o3}, is such as to involve only subsonic flow, the process may be represented by the station points 1-2-3-9′-5′-5-6-7.

*Reference 186-1

Table 5.4.3.2. Types of Pressure Reducing Regulators

MODULATING	NON-MODULATING
Direct acting, weight loaded, reset action	Pressure switch and solenoid valve, ON-OFF action
Direct acting, spring loaded, proportional action	Piloted, spring loaded, ON-OFF action
Direct acting, preset pressure loaded, proportional action	
Direct acting, constant pressure loaded, proportional plus reset action	
Piloted, variable pressure loaded, proportional plus reset action	

5.4.3.4 PERFORMANCE CHARACTERISTICS. Regulator performance characteristics can be divided into two groups—static and dynamic. Typical static performance characteristics are illustrated in Figure 5.4.3.4a. Dynamic performance is illustrated in Figure 5.4.3.4b. Figure notation is described and typical values are noted in the following paragraphs. The effect of system conditions on performance (e.g., ullage, pressurizing gas, sense line length) are discussed in later paragraphs.

Deadband. The variation of regulated pressure about its desired value (called the set point) is known as deadband. Deadband denotes regulator accuracy and is expressed in percent of regulated pressure or in psi deviation from the set point. For example, a regulator having a set point of 100 psig and a deadband ranging from 95 to 105 psig has a deadband of ±5 percent, or ±5 psig. Simple regulators, those regulators without special compensators, normally have accuracies ranging from ±5 to ±10 percent of regulated pressure, while the more complex regulators may have accuracies from ±1 to ±2 percent, depending upon the severity of system and environmental conditions.

Accuracies generally decline if the upstream pressure is substantially higher than regulated pressure, or if regulated pressure is set close to 0 psig. For example, if the regulated pressure is to be 100 psig, raising the upstream pressure from 300 psig to 3000 psig may change the dead band from ±3 percent (±3 psi) to ±5 percent (±5 psi).

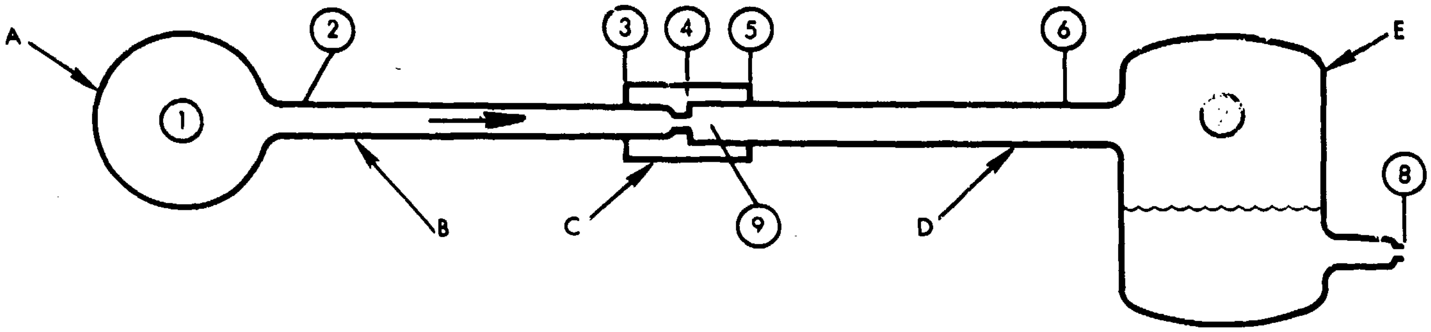


Figure 5.4.3.3a. Pressurization System Schematic
(Reference 186-1)

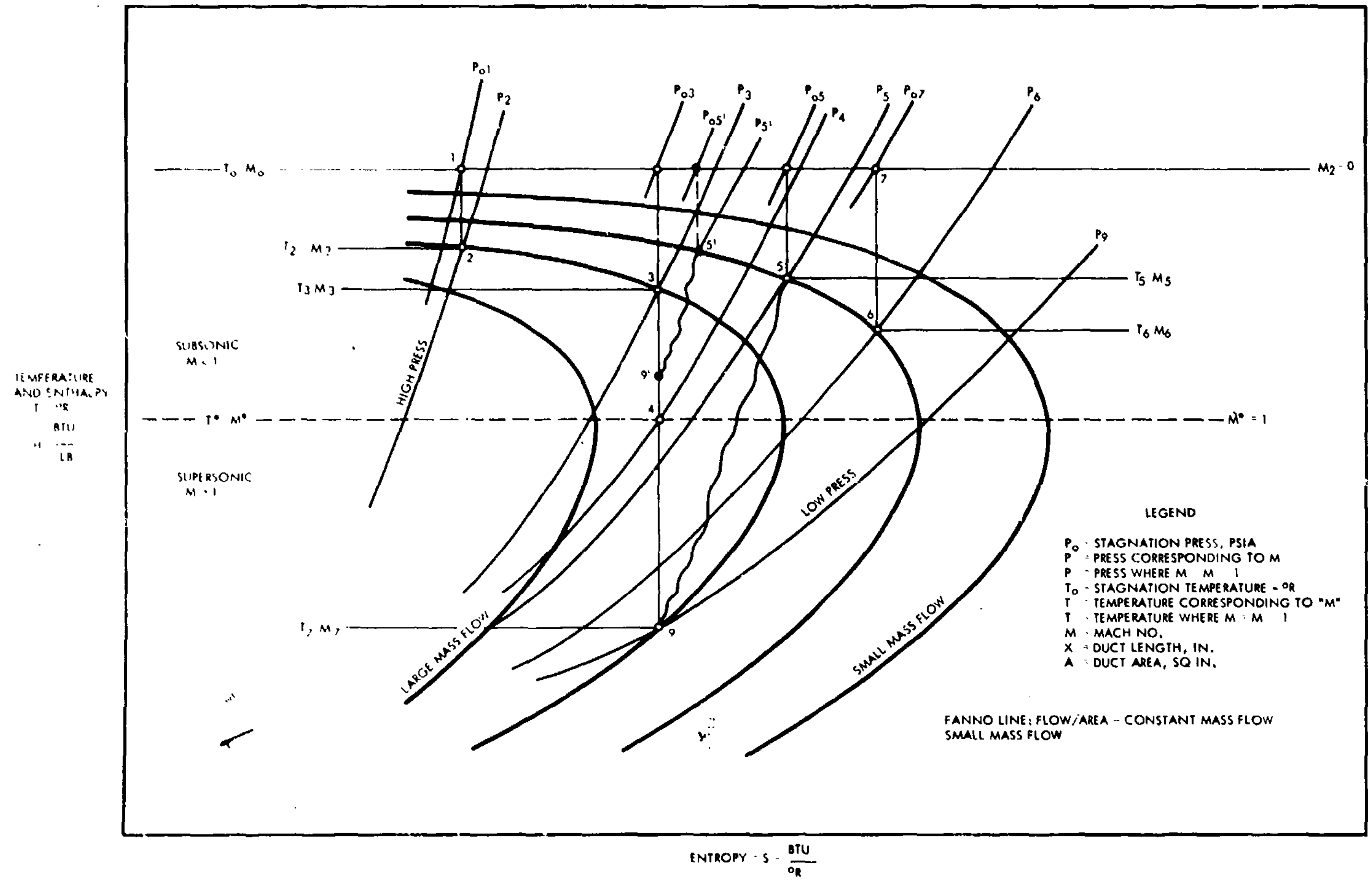


Figure 5.4.3.3b. Temperature-Entropy Diagram for a Regulator Flow Process
(Reference 186-1)

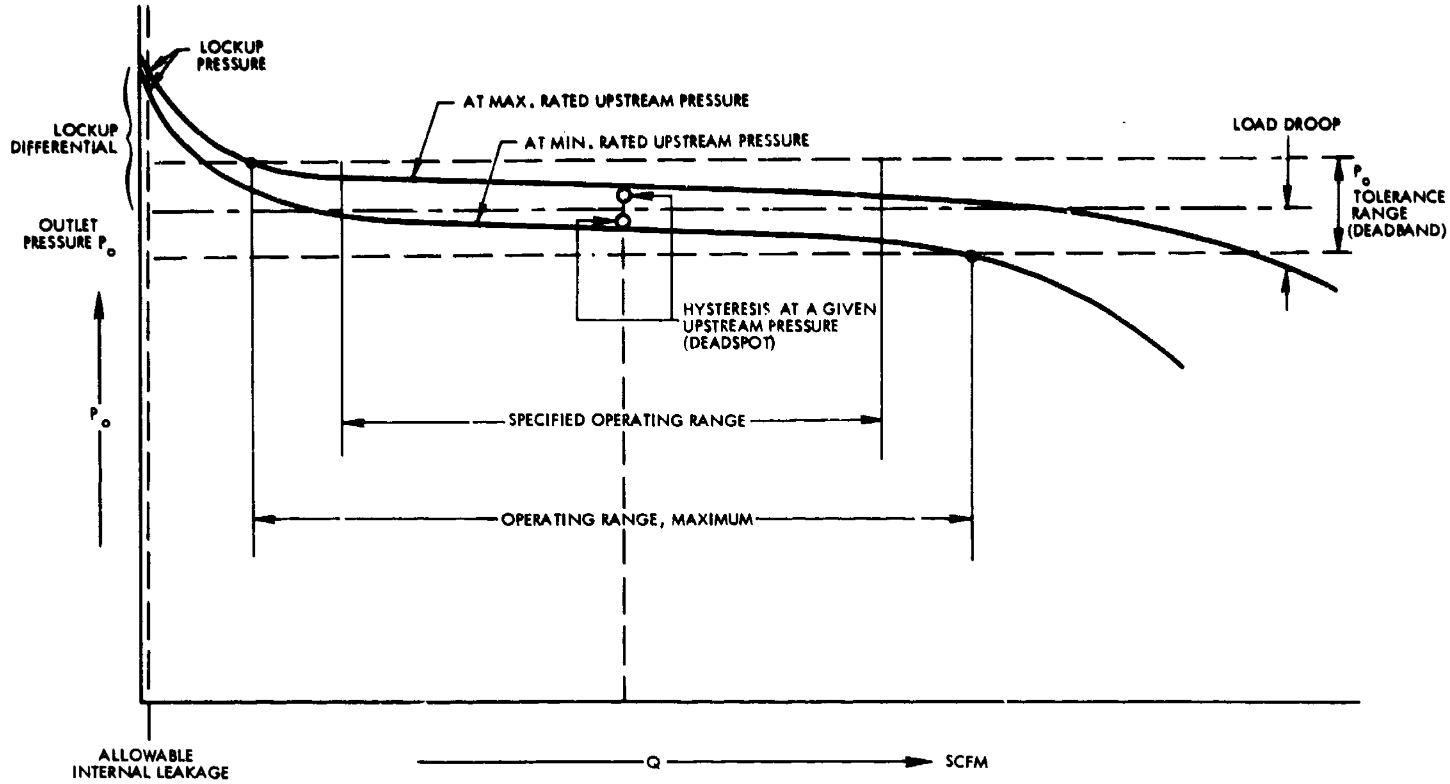


Figure 5.4.3.4a. Static Performance of a Pressure Regulator
(Adapted from Reference 89-3)

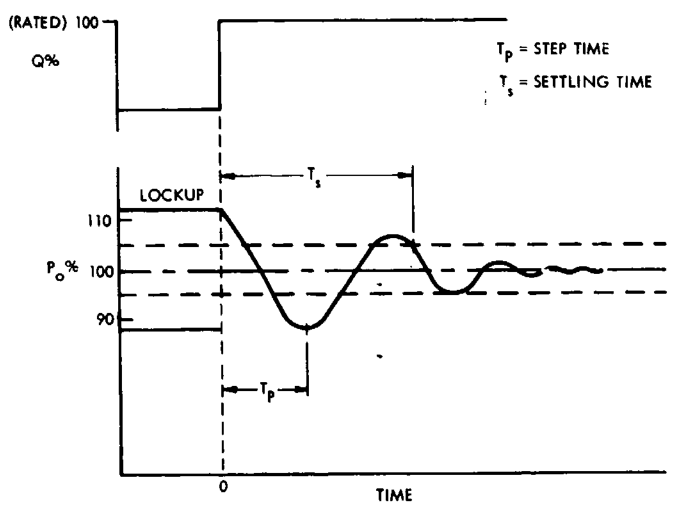


Figure 5.4.3.4b. Dynamic Performance of a Pressure Regulator
(Adapted from Reference 89-3)

If the regulated pressure is 1 psig, the minimum deadband with even a low upstream pressure of 10 psig may be approximately ±10 percent (±0.1 psi).

Lockup. Lockup is the condition reached by all pressure-reducing regulators when the downstream pressure is increased to the point where the regulator closes completely. The lockup pressure is the value of downstream pressure when the regulator is closed and the flow is equal to the allowable internal leakage. Lockup pressure is always higher than the maximum regulated pressure. The difference between lockup pressure and the set point is called the lockup differential. The lockup differential is also expressed as a percentage of the regulated pressure (set point).

Allowable Internal Leakage. Internal leakage is the amount of gas flow through the metering valve at lockup. The amount of internal leakage is usually specified in standard cubic centimeters per minute and is dependent upon the upstream pressure, nature of the gas, valving element configuration, and force available to seal. Typical helium leakage rates range from 5 scc/min to 50 scc/min; however, leakage rates from 10 to 25 scc/hr have been achieved.

Load Droop. Load droop is the decrease in regulated pressure caused by a decrease in the load (reference force) as the metering valve opens from its closed to full flow. Load droop will exist in only spring or locked-in pressure-loaded regulators. If accurate pressure regulation at only one flow rate is desired, the load can be established and the system trimmed for the desired valving element position such that load droop is eliminated. For varying flow rates, load droop may be reduced by placing a booster restriction such as a simple sharp-edged orifice just upstream of the sense line tap, as shown in Figure 5.4.3.4c. As the flow increases, the static pressure sensed at the minimum restriction (venturi throat or vena contracta) decreases, tending to further open the regulator, and thus providing compensation for load droop tendency at high flow. If placement of the sense line at the minimum restriction results in over compensation, the sense line must be located further downstream of the restriction where the velocity is lower due to pressure recovery. The effect of a booster on a droop is shown in Figure 5.4.3.4d. For a particular inlet and outlet pressure it is possible to design the restriction in such a way that remarkably close regulation is obtained.

Step (Response) Time. If a tank partially filled with liquid is under lockup pressure, any increase in liquid outflow will decrease tank pressure. If the outflow change is sufficiently rapid, pressure may decrease to a value below the regulator set point before the regulator can respond. The time required for the regulator to halt the pressure decrease with sufficient gas flow is called the *step time*. The step time is specified for a given maximum liquid flow rate, changing from zero to maximum flow in a stipulated number of milliseconds.

Settling times depend upon amount of dampening incorporated into the regulator.

Minimum Rated Upstream Pressure. Regulated pressure can be maintained at the set point only if the upstream pressure is greater or equal to the sum of the regulated pressure and regulator-plus-line-drop. Friction imposes a built-in pressure drop through the regulator and lines upstream to the gas storage tank and downstream to the tank. For example, with a 75 psi regulator-plus-line-drop, the minimum pressure to maintain a downstream regulated pressure of 100 psig must be at least 175 psig plus any pressure loss through the regulator full open.

Deadspot. Deadspot is the narrow band of outlet pressures at which the regulator becomes insensitive to small pressure fluctuations because of friction and stiction in the metering valve assembly.

Hysteresis. Hysteresis is the inability of the valving element to achieve the same relative position from the seat for the same sensed outlet pressure because of inefficiencies such as friction and inelastic deformation in the metering valve assembly.

Operating Range. The operating range is the range of gas flow in standard cubic feet per minute. It is determined on the low flow end by the maximum inlet pressure and on the high flow end by the minimum inlet pressure such that under all flow rate and upstream pressure conditions the regulated pressure remains within the deadband. A more conservative range of flow rates is the *specified operating range*, which allows a finite pressure difference between the minimum rated outlet pressure and the actual minimum outlet pressure occurring with the minimum rated inlet pressure.

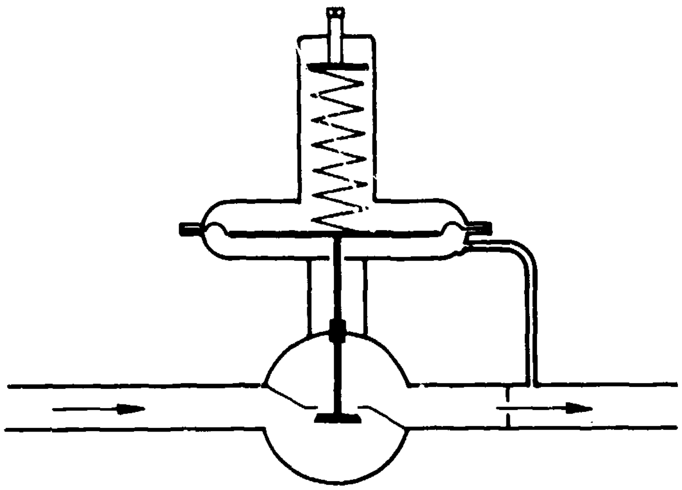


Figure 5.4.3.4c. Load Droop Compensation by Use of a Booster Restriction

(From Technical Data of Mason-Neilan, Division of Worthington Corporation, Norwood, Massachusetts)

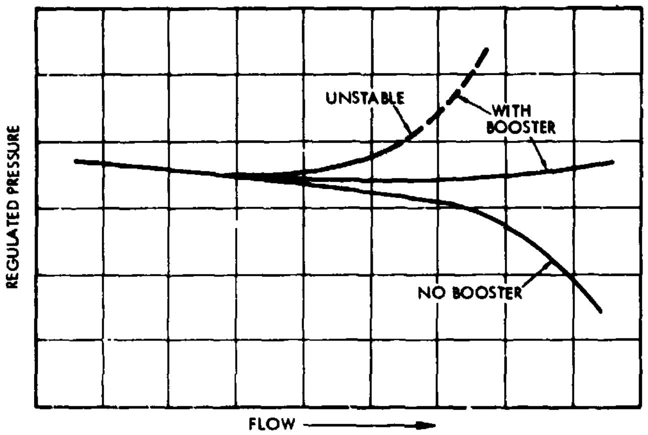


Figure 5.4.3.4d. Effect of Booster Restriction on Regulated Pressure

(From Technical Data of Mason-Neilan, Division of Worthington Corporation, Norwood, Massachusetts)

5.4.4 Direct-Acting Regulators

Direct-acting regulators have three basic elements: a sensing element (such as a diaphragm), a valving unit (valving element and seat), and a reference load. The combination of sensing element and valving unit together with their interconnecting stem is sometimes referred to as the *metering valve*. The reference force input is often called the *load*. Direct acting regulators can be weight loaded, spring loaded, or pre-set or constant pressure loaded. The loading methods are discussed in Detailed Topic 5.4.8.4. A direct-acting, spring-loaded regulator is shown in Figure 5.4.4. The sensing element integrates actual regulated pressure over its effective area and compresses a spring which is the reference force (load) used to establish the desired pressure setting. The resulting force unbalances the metering valve, thereby altering flow through the regulator.

The valving element of the regulator may or may not be pressure balanced. In unbalanced direct-acting regulators, inlet pressure acts across the main seat, effecting a change in the force equilibrium within the valve as a function of the inlet pressure. The loss of pressure force across the main seat results in an increase in outlet pressure. Pressure balancing of the main valving element eliminates the outlet pressure variation resulting from inlet pressure changes, but requires a dynamic seal of some type to pressure balance the main inlet flow seat (Reference V-56).

The set point of the regulator can be easily adjusted by compressing or extending the spring. The lower the main reference spring rate, the greater the main valving element motion that can be achieved. For extremely accurate regulation with very small outlet pressure changes, large sensing areas are required to achieve high driving forces. This results in high reference spring forces. Since a low reference spring rate is desired, reference springs having a considerable number of coils are used. This increases the regulator envelope and weight to the point where the simple configuration no longer represents a minimum weight and envelope regulating device (Reference V-56).

Similarly, for a low regulated pressure the valve is lightweight, but becomes heavier with higher regulated pressures because of the large spring force needed to oppose the pressure.

The primary disadvantage of a spring is that any change in its length during the regulation cycle will change the load and the regulator set point, resulting in load droop. The amount of load droop can be decreased by (1) lowering the spring rate, or (2) decreasing the valving element stroke.

5.4.5 Pilot-Operated Regulators

A single-stage, pilot-operated regulator is used when close tolerance regulation is required; a small pilot regulator is used to control the main metering valve pneumatically. Since adequate forces are available through piloting to drive the metering valve element, the need for valving element pressure balancing is eliminated. Considerable gain can be achieved by using this principle, reducing the errors resulting from high frictional forces and sealing forces on the valving element seat. Since pilot regulators have small closure seats, the effect of varying inlet pressure results in a negligible force when compared with the sensing area's outlet pressure force. Pilot regulator valving element balancing, with its associated dynamic seal problems, is therefore unnecessary. Pilot regulators are usually of low flow capacity and, as a result, the valving element motion and sensing area motions are small. This allows an inorganic sealing technique such as a metal diaphragm to be used in the control pilot (Reference V-56).

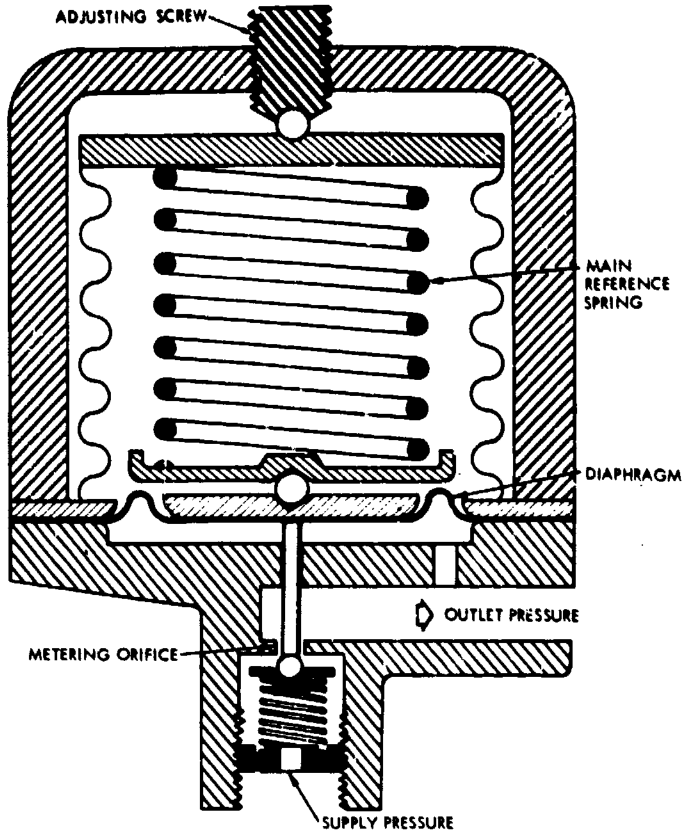


Figure 5.4.4. Direct-Acting Spring-Loaded Regulator
(Reference 186-1)

A pilot-operated regulator is shown in Figure 5.4.5*. The regulator consists of three basic parts: a bleed regulator (also called a step-down regulator), a controller (sometimes called a control pilot), and a metering valve. These three parts are also discussed in Sub-Topic 5.4.8. The bleed regulator, which is a common spring-loaded, pressure regulator, supplies an approximately constant pre-set pressure to the controller. This pressure is greater than tank pressure. The controller establishes the tank pressure set point.

*Description of this pressure regulator is adapted from Reference 35-1.

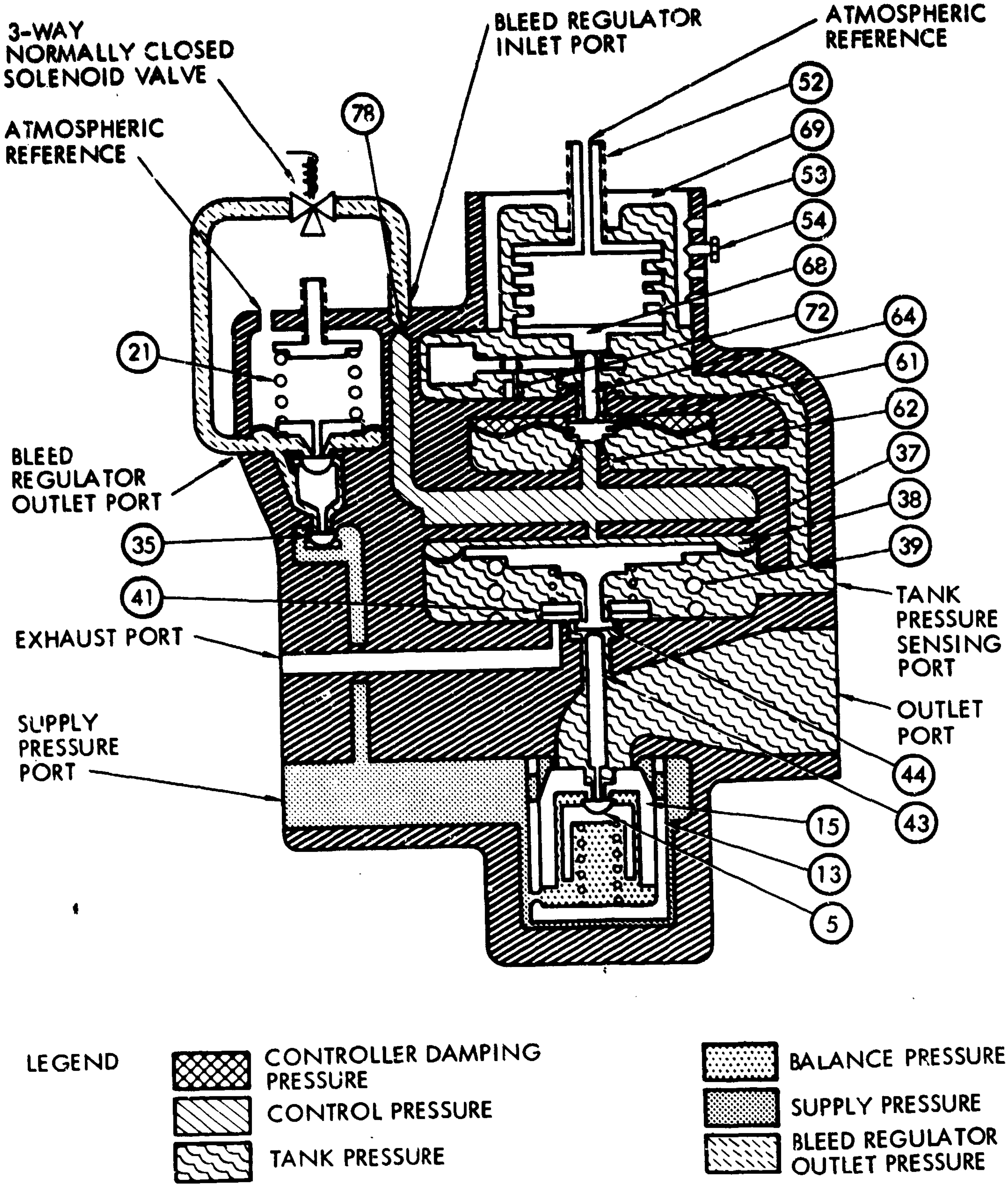


Figure 5.4.5. Pilot-Operated Regulator
(Reference 35-1)

Small deviations from the set point are magnified as an error signal by means of a pilot valve, causing an increase or decrease in the control pressure which operates the metering valve actuator. Movement of the actuator diaphragm causes the metering element (main poppet) to open or close, as required, to maintain the set point. In regulator operation the following sequence of events occurs:
Gas enters the regulator through a supply pressure port, but is prevented from entering the tank by the closed main poppet (15).[*] Some of the supply gas goes to the step-down regulator metering valve (35), whose pressure is reduced through the orifice to a pressure determined by the pre-set reference spring (21) force. The gas at the reduced pressure then flows through a restrictor (78), where it is further reduced in pressure as it enters into a control pressure chamber. The gas flows out of the control chamber through the pilot nozzle (62) into a tank pressure chamber, and from there out through the relief valve (41) to the atmosphere. To adjust the regulator, the bellows adjusting screw (52) is turned clockwise until the pilot valve (61) is seated on the nozzle (62) with a force which is transmitted from the bellows (68) through a push pin (64). With the nozzle (62) shutoff, control pressure builds up in the control pressure chamber. The force exerted by the control pressure on the actuator diaphragm (38) overcomes the pre-load on the actuator bias spring (39) and causes the actuator diaphragm (38) to move toward the main poppet (15) until the relief valve (41) closes. Additional travel of the diaphragm brings the actuator collar (44) and push pin (43) in contact with main poppet pilot valve (5). A further decrease and control pressure overcomes the seating force and opens the main poppet pilot valve.

Until this moment the balance pressure has been equal to the supply pressure, and the net pressure force has been holding the inlet valve closed. With the main poppet pilot valve (5) now open, balance pressure begins to drop as gas flows from the balance chamber into the tank pressure section. Flow of supply pressure gas into the balance chamber is restricted by an orifice in the sleeve (13), and balance pressure continues to drop until the force it exerts tending to hold the main poppet closed is equal to the supply pressure force tending to open the poppet. A further decrease in balance chamber pressure causes the main poppet (15) to open, permitting gas to flow through the outlet port into the tank. As the pressure in the tank begins to build up, it is fed back to the controller section through the tank pressure sensing port, where it is applied to the underside of the actuator diaphragm and the outside of the bellows. Tank pressure continues to increase until the force it exerts on the bellows begins to overcome the force exerted by the bellows on the pilot valve. Further increase in tank pressure causes the pilot valve to move open, allowing gas to flow out of the control chamber with a resultant decrease in control pressure. As control pressure begins to drop, the actuator diaphragm starts to move away from the main poppet. The main poppet pilot valve gradually closes, building up balance pressure and causing the main poppet to move to the closed position. When the main poppet closes, gas flow to the tank ceases and tank pressure stabilizes.

With the tank pressure and bellows force in equilibrium, the pilot poppet is positioned a sufficient distance off the nozzle seat to maintain a steady-state control pressure. The steady-state control pressure is always greater than nominal tank pressure by an amount determined by the actuator spring pre-load. Under these steady-state conditions, gas continues to flow from the bleed regulator through the controller at a rate determined by the restrictor (78), and out to the atmosphere through the relief valve, which is now positioned just off its seat by the actuator. Additional clockwise adjustment of the adjusting screw (52) further compresses the bellows, and the above sequence of events is repeated. In this manner the desired set point of the regulator is achieved.

When forward flow is demanded from the regulator, pressure starts to drop below the set point. The drop in pressure is detected by the bellows. This upsets the steady-state force balance and allows the bellows to extend slightly. As the bellows extends, it moves the pilot poppet closer to the nozzle, which further restricts the flow and results in an increase in control pressure. The rise in control pressure causes the actuator to move such that the relief valve closes, and the main poppet opens to admit additional gas to the tank and return the pressure to the set point.

As pressure returns to the set point, the bellows continues to detect the change and returns to a point near its original position. Control pressure stabilizes at a value whereby the actuator is holding the main poppet pilot valve open to a degree sufficient to maintain the flow out of the balance chamber equal to the flow into the chamber through the sleeve orifice, thus creating a steady-state balance pressure at some value lower than supply pressure. The net force exerted by supply, regulated, and balance pressures positions the main poppet at an opening sufficient to maintain the flow through the valve equal to the flow demand downstream. The force balance between the three pressures holds the main poppet at this position. Should tank pressure increase above the set point, the bellows will be compressed an additional amount, which opens the pilot valve further and causes control pressure to drop below the steady-state value. The decrease in control pressure causes the actuator to move away from the main poppet pilot and main poppet until both close. Should tank pressure continue to rise, control pressure will continue to drop, and the actuator will move until the collar (44) contacts the disc shaped relief valve poppet and lifts it off its seat. The open relief valve allows gas from the tank to escape to atmosphere through the exhaust port until the tank pressure returns to the set point, and equilibrium conditions are restored.

A pneumatic dashpot is used to obtain damping of pilot valve motion. The pilot poppet (61) is attached to a corrugated metal diaphragm, and the damping pressure chamber is above the diaphragm. Movement of the diaphragm results in gas flow between the damping chamber and the

[*] Numbers correspond to numbers appearing in Figure 5.4.5.

region of tank pressure via the restriction formed by the clearance around the push pin (64), to produce the desired damping effect. Similarly, an orifice plate (37) in association with the actuator diaphragm constitutes a pneumatic dashpot to damp actuator motion.

Errors in regulated pressure are introduced by valve spring forces, pressure forces exerted on the valve, friction, vibration, sustained acceleration, and thermal effects. The controller, as described above, adjusts the control pressure up or down to minimize the effect of the errors of the actuator on tank pressure. The controller also contains devices to compensate the bellows for the effects of vibration, sustained acceleration, and temperature. Compensators are discussed in Sub-Topic 5.4.11, Sub-Topic 13.3.3, and Detailed Topic 13.5.2.3.

5.4.6 Staging of Modulating Regulators

When two regulators are installed in series they are said to be staged. Generally, both direct-acting and piloted-operated regulators can be staged with other direct-acting regulators. Staging will not offer significant performance advantages over a single-stage regulator which has been properly designed and tuned to system conditions. The purposes of staging are:

a) To provide accurate pressure control with less sophisticated regulator designs.

b) To provide minimum pressure control in the event one regulator fails. Both regulators are designed to fail open. If the downstream regulator fails, the regulated pressure will be that of the upstream regulator less the pressure loss across the downstream regulator acting as a constant area restriction. If the first stage fails, performance accuracy is reduced due to the larger pressure forces against the regulator control element.

c) To eliminate the need to inlet pressure balance the metering valve.

d) To reduce the individual loads required to control the regulated pressures.

The upstream regulator is commonly called the *roughing regulator;* the downstream regulator, the *precision regulator.*

Staged regulators may be separate units or integrated into one unit.

5.4.6.1 STAGED REGULATORS, INTEGRATED. Staging by integration of two spring-loaded regulators is shown in Figure 5.4.6.1. Pre-set pressure or constant pressure-loaded regulators could be used as well. Variations of regulated pressure are sensed by the precision regulator. If regulated pressure decreases, the precision regulator opens, decreasing pressure behind the piston of the roughing regulator which in turn opens to allow gas flow downstream.

This type of staged regulator is sensitive to small pressure variations and exercises good control for high flows. On

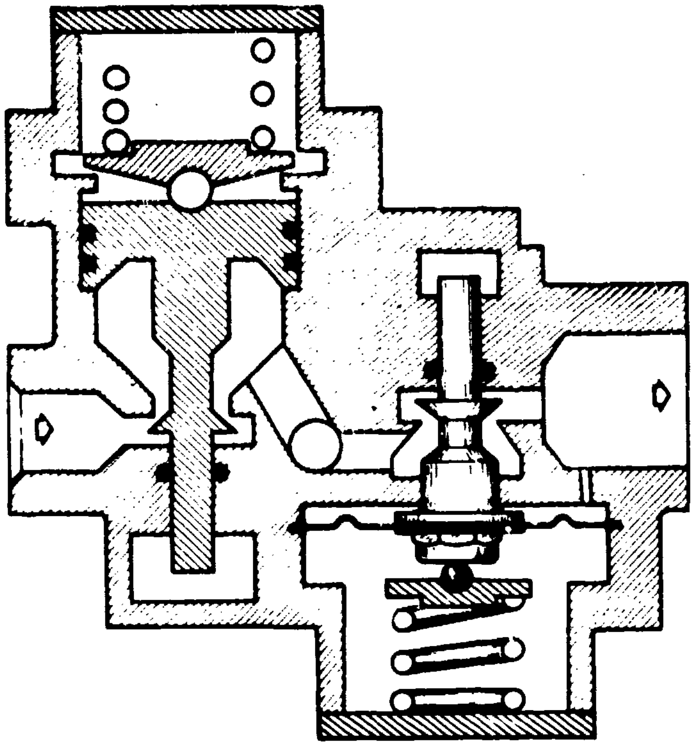


Figure 5.4.6.1. Integrated Two-Stage Regulator
(Courtesy of Futurecraft Corporation, City of Industry, California)

the other hand, the minimum pressure for regulation is increased, because an additional pressure drop is established across the roughing regulator even when it is wide open. Furthermore, the precision regulator, being spring-loaded, creates a load droop effect on regulated pressure. The roughing regulator will not contribute to load droop, the precision regulator merely having a reduced upstream pressure. A major problem with staged regulators is the effect called *coupling.* Coupling may be defined as the unstable, out-of-phase relationship between the regulators' control elements, caused by large variations in response times and rapid fluctuations in regulated pressure. For example, if regulated pressure decreases, the precision regulator will open and a short time later the roughing regulator will open. As regulated pressure increases, the precision regulator starts to close, followed by the closing of the roughing regulator. If the flow downstream changes abruptly, the roughing regulator may still be closing due to its inertia, as the precision regulator once again is starting to open. As a cumulative effect, one regulator may be open while the other is closed. Coupling is reduced if regulator response times are nearly the same, or if a secondary ullage volume is installed between the regulators. The latter alternative is discussed in Detailed Topic 5.4.6.2.

5.4.6.2 STAGED REGULATORS, SEPARATE UNITS. A typical system of staged regulators is shown in Figure 5.4.6.2. A secondary accumulator volume is shown installed between the regulators. The line connecting the two regulators may act as an accumulator if its length and diameter are large. The accumulator decouples the regulators, providing a source of pressurized gas for the precision regulator until the roughing regulator has time to respond to changes in downstream pressure. Accumulator size is determined from the total response time required for the roughing regulator. The total response time is the time required for the roughing regulator to receive the pressure signal generated by the opening of the precision regulator, plus the time for the regulator to open against its own inertia.

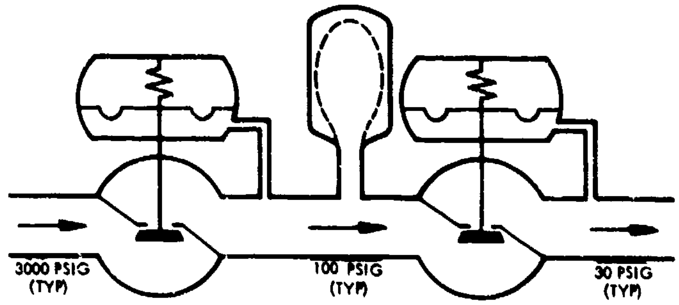


Figure 5.4.6.2. Staging of Two Spring-Loaded Regulators

5.4.7 Non-Modulating Regulators

A non-modulating regulator is a two-position device, either open or closed. The most elementary non-modulating regulator is the combination of a pressure switch and solenoid valve. Although not truly a regulator, this device is often referred to as a "bang-bang" regulator. As tank pressure decreases, the pressure switch sensing the lower pressure actuates a solenoid valve which releases gas from a high pressure accumulator. As tank pressure increases, the pressure switch opens the circuit and the solenoid closes. A pressure-valve position-time plot is shown in Figure 5.4.7a.

Solenoid valves are usually small in comparison with conventional regulators, and have a lower capacity rating. Increased capacity may be obtained by using a "gang" of solenoid valves actuated by the pressure switch. Total weight increases, however, which may offset other advantages. A "bang-bang" regulator is usually limited to small flow demands. As flight hardware, such regulators are lightweight, remain accurate under extreme environments, provide positive lockup, and have a fast response time to small pressure changes. However, long life cycle performance is required. The development of this concept is discussed in Reference 21-5 and test results of a "bang-bang"

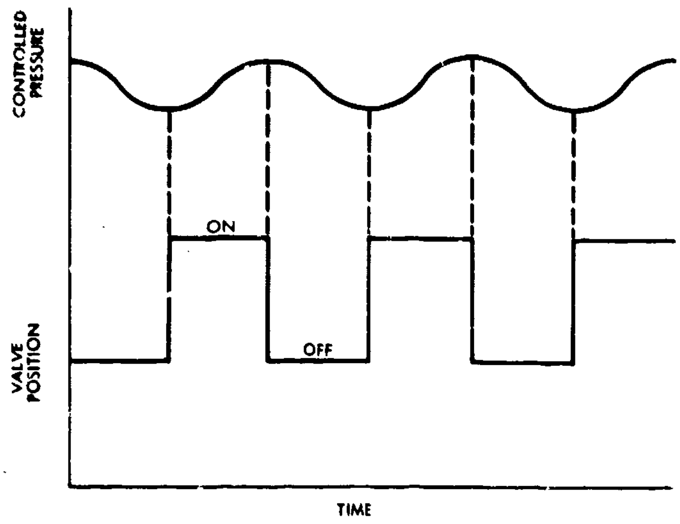


Figure 5.4.7a. ON-OFF Control Action

regulator are presented in Reference 70-5. Another example of a non-modulating regulator is the device shown in Figure 5.4.7b developed by the Frebank Company under AFBSD Contract AF04(647)-429. Design of the regulator was an outgrowth of development of a pressure switch using Belleville spring loading. (See Sub-Section 6.5 for description of the Belleville spring.) The regulator designed is a single-stage, pilot-operated unit intended to control the pressure in a missile tank. The metering valve is closed until the tank pressure falls below the lower limit set point. At this point the regulator metering valve opens to full flow until the tank pressure reaches the upper set point, at which time the valve closes.

The deadband for a non-modulating regulator is defined as the pressure range between the upper and lower set points. These set points are determined in the regulator illustrated in Figure 5.4.7b* by the pre-deflection to post deflection points, which limits the travel of the Belleville loading springs. The pre-deflection point is determined by the adjustment ring (6),** and the post deflection point by the stop (5). The regulated pressure exerts force on the sensing diaphragm (12) through the sensing port. Force on the diaphragm is transmitted by the diaphragm assembly to the belleville springs (11). With no pressure there is no force against the diaphragm. The actuator assembly, consisting of the diaphragm and belleville springs, then holds the head of the pin (8) against the pre-deflection travel stop (represented by the raised arc under the pin head in the schematic diagram) and causes the pilot valve (14) to be full open. The adjustment ring (6) provides the pre-deflection adjustment.

High pressure gas introduced into the inlet port flows through an orifice (A), through the open pilot valve (14),

*Description of the pressure regulator is taken from Reference 21-5.
**Numbers correspond to Figure 5.4.7b.

and into the sense chamber (C) through flow passage (B). With the pilot valve open, the pressure drop across the orifice (A) causes the pressure in the chamber behind the main valve poppet (24) to decrease, creating a pressure differential across the main poppet. The pressure differential creates a force unbalance, and the main poppet opens until it bottoms on the shank of the pilot valve seat (14). The inlet gas now flows directly through the main valve into the ullage.

When the pressure in the tank reaches the value corresponding to the pre-deflection setting of the belleville springs, the actuator assembly snaps to the post deflection stop (5). The pilot valve, no longer being held open by the actuator assembly, is now closed by the spring between the poppets. When the pilot valve is closed, the pressure increases in the chamber between the poppets and reaches a value where the main valve poppet closes. The pressure in the chamber continues to rise until it equals the inlet pressure, and both poppets are then held on their seats by the differential between the inlet and outlet pressures, plus the poppet spring force. With the actuator assembly at the post deflection stop and both poppets closed, the regulator is in lockup. A lockup condition is shown in schematic diagram. With the regulator in lockup, any demand on the system will decrease the pressure in the ullage. As the pressure drops to a value of the post-deflection setting, the actuator assembly snaps to the pre-deflection point. The pin moves the pilot poppet to full open, and the chamber between the poppets evacuates to create the pressure differential to cause the main valve poppet to open. The cycle is repeated to hold the regulated pressure in the deadband.

5.4.8 Modules of a Pressure-Reducing Regulator

5.4.8.1 **THE ACTUATOR.** An actuator, which is common to all regulators regardless of classification, is the assembly connected to the valving element which controls its position. The actuator consists of the sensing element and the load, which are discussed separately in Detailed Topics 5.4.8.4 and 5.4.8.5, respectively.

5.4.8.2 **THE METERING VALVE.** The metering valve consists of the actuator plus the main valving unit.

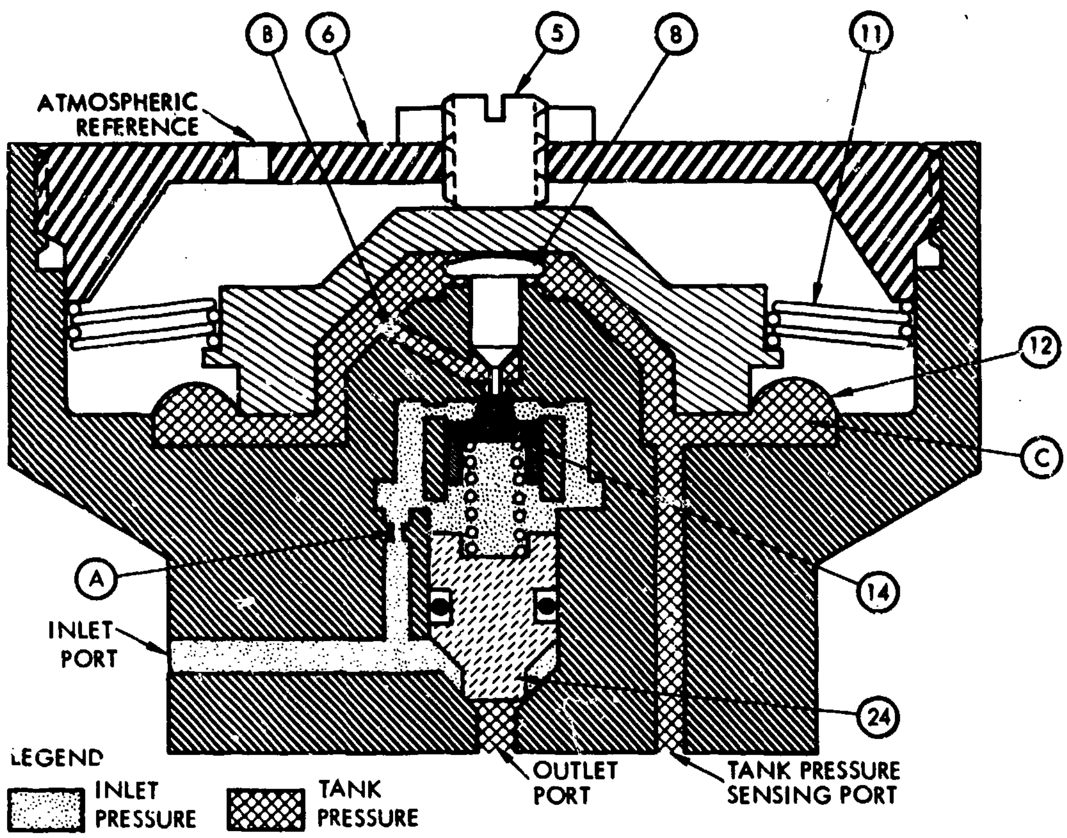


Figure 5.4.7b. Non-Modulating Pressure Regulator
(Reference 21-5)

5.4.8.3 THE VALVING UNIT. The valving unit consists of a valve seat and a valving element such as a poppet, plug, butterfly disc, piston, or sleeve. Valving elements are selected on the performance characteristics desired for the regulator. The flat and conical poppets are common choices, since large variations in flow can be obtained with small stem travel; however, piston and sleeve elements have the advantage of minimizing unbalanced pressure forces. While it is difficult to achieve good sealing with piston and sleeve elements, sealing of the poppet-type valving element may be achieved by using a well-lapped plug and seat or by using a soft elastomeric seal.

A typical example of a piston-type metering valve is shown in Figure 5.4.8.3a. Flow is regulated as the piston covers or uncovers holes located circumferentially around the cylinder. Hole size and pattern are designed to give the desired flow versus stroke characteristics. The only unbalanced pressure force results from the regulated pressure acting on an area equivalent to the stem cross-sectional area. However, lockup with low leakage is difficult to achieve, because the piston provides no positive seal but depends primarily on close tolerances. A sleeve-type metering element is illustrated in Figure 5.4.15a.

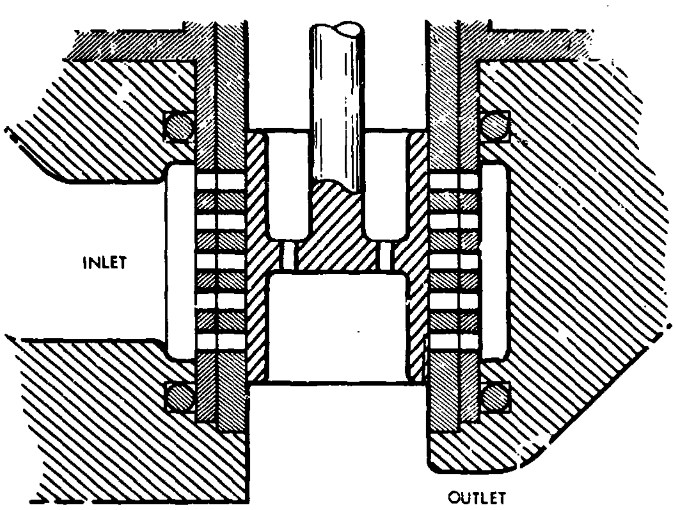


Figure 5.4.8.3a. Piston Metering Valve for Regulator
(Adapted from Technical Data of B. H. Hadley, Inc., Pomona, California)

Valving unit geometry determines capacity versus pressure drop, denoted by the quantity C_v. In some valve designs, C_v is greater in the flow-to-open direction, while in others the converse may be true. Similar considerations were discussed in Sub-Topic 3.8.4. Regulators are specified as either fail-open or fail-closed. In airborne systems, regulators are usually fail-open because gas pressure is absolutely necessary, but the consequential over-pressurization can be avoided by system relief valves. Causes of failure may include loss of reference force, diaphragm rupture, and blockage by contamination.

The position of the valving element in relation to the seat controls the regulator set point. For example, if the set point is to be controlled by a metering valve almost full open, any decrease in downstream pressure may require more than full flow. The load will force the valve open to full flow, which still may be insufficient to compensate for the pressure decrease, and control will be lost. At the other extreme, with the plug almost closed an increase in downstream pressure will quickly close the regulator. Subsequent reopening will initiate a pressure surge which may again close the regulator or, at best, induce high amplitude pressure oscillations.

A pilot poppet, shown in Figure 5.4.8.3b, uses upstream pressure to hold a conical poppet on its seat. The piloted poppet consists of a main poppet and a main poppet pilot valve. The main poppet pilot valve is connected to the actuator stem and seats against the main poppet. When the pilot valve is closed, upstream pressure in the cavity behind the main poppet keeps it closed. As the actuator stem moves down, the pilot poppet moves away from the main poppet and vents the cavity behind it to the downstream side faster than gas can fill the cavity through a

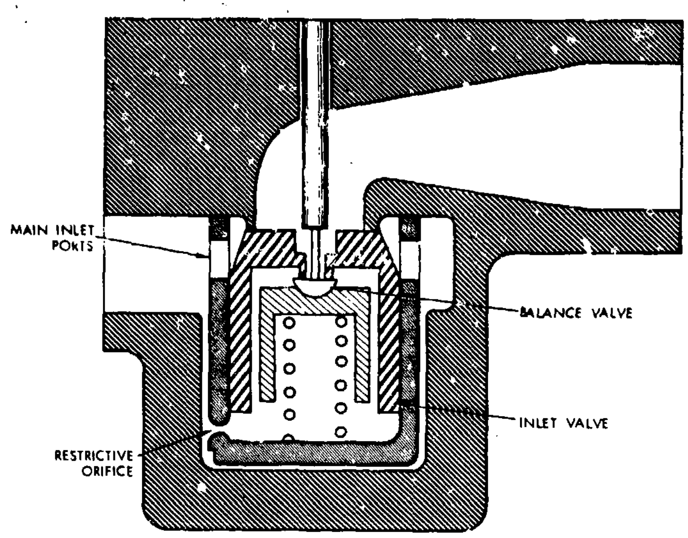


Figure 5.4.8.3b. Piloted Metering Valve

constant area restriction. Upstream pressure against the main poppet forces it open, allowing upstream gas to flow downstream.

5.4.8.4 SENSING ELEMENTS. Common sensing elements are diaphragms, pistons, and bellows. Diaphragms are usually fabric-elastomer laminates, whereas pistons and bellows are metallic. A comparison of sensing elements for regulators is presented in Table 5.4.8.4. Designs of bellows and diaphragms are presented in Sub-Sections 6.6 and 6.7, respectively.

Table 5.4.8.4. Comparison of Sensing Elements For Regulator Use

	LIGHT-WEIGHT	SMALL ENVELOPE	GOOD SEALING	LOW RESISTANCE TO MOTION	USE WITH HIGH PRESSURE	LONG STROKE	USE UNDER VIBRATION	USE WITH HIGH TEMPERA-TURES
Diaphragm	1	1	1(a)	1	2	3(d)	1	3
Piston	2	2	2	3(b)	1	1	3	1
Bellows	1	2	1	2(c)	2	2	2	2

(1) Best alternative
(2) and (3) Less desirable alternatives
(a) Non-metallic are permeable
(b) Friction (sliding)
(c) Spring rate added
(d) May be comparable to bellows if rolling diaphragms are used

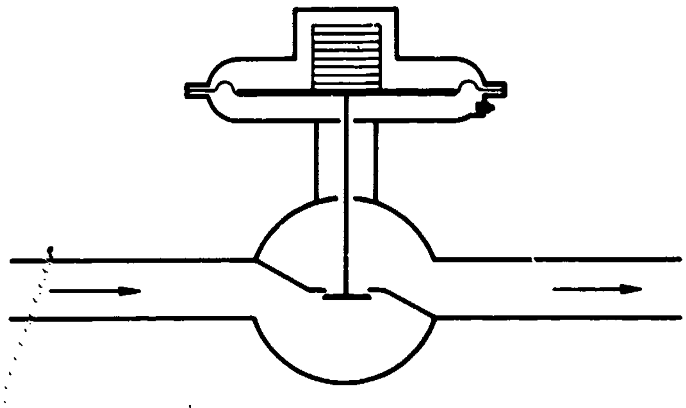


Figure 5.4.8.5a. Weight-Loaded Regulator

5.4.8.5 LOADING METHODS. The load performs two vital functions in the regulator. First, it establishes a reference force, the magnitude of which is proportional to the desired regulator set point. Second, it determines the control mode that the regulator will have. The load, or reference input force, is established in modulating regulators by weights, springs, or by pressurizing the dome above the sensing element with gas. In non-modulating regulators, Belleville springs are normally used. Weight-loaded and spring-loaded modulating regulators are shown in Figures 5.4.8.5a and 5.4.4, respectively. Weight-loaded regulators cannot be used for airborne applications; but, in some industrial applications it is a convenient method because the reference force can be changed easily by merely adding or taking away weights. The primary disadvantage of weight-loaded regulators is that large weights are required for even small regulated pressures (Reference 144-2). The operation of a spring-loaded regulator is presented in Sub-Topic 5.4.4.

Pneumatic pressure is used in three ways:

a) The dome is pre-set to a pre-determined pressure, which is locked in the dome.

b) Dome pressure is maintained at a constant pressure by a separate regulator.

c) Dome pressure is controlled by a control pilot which montiors regulated pressure.

A comparison of the loading methods is shown in Table 5.4.8.5.

Pre-set Pressure Loading. In the pre-set pressure-load regulator, dome pressure increases or decreases as the sensing element deflects, altering the set point of the regulator. The amount of set point deviation (offset) decreases as the capacity of the loading dome increases and the length of metering valve stroke decreases. In particular, this loading method is highly sensitive to environmental temperature variations which may either increase or decrease loading pressure.

The advantage of using gas is that the loading force can be changed from a remote location, thus incorporating into the regulator a flexibility of the regulated pressure range without having to depressurize the system for mechanical adjustment.

The addition of a separation plate with an orifice in the main dome acts as a damper. This reduces the tendency of the actuator to overshoot, and the valving element to chatter at the nearly closed position. Referring to Figure 5.4.8.5b, movement of the main diaphragm toward the separation plate produces a momentary increase in pressure in the small volume between the two elements. Gas at higher pressure will bleed to the dome's larger volume through the orifice. Similarly, movement of the diaphragm away from the separation plate, caused by a decrease in regulated pressure, produces a decrease in pressure behind the diaphragm, retarding the metering element stroke until the pressures are equalized.

Constant Pressure Loading. By using a small step-down or so-called bleed regulator, the dome loading pressure can be maintained constant as the sensing element changes position. A regulator of this type is shown in Figure 5.4.8.5c. Unlike the preset pressure-loaded regulator, the initial set point is retained and a constant reference force is established throughout metering valve travel. The step-down regulator is usually a spring-loaded regulator, similar in design to that shown in Figure 5.4.4. The main diaphragm is balanced between the regulated pressure on one side and the loading pressure on the other. Regulated pressure

Table 5.4.8.5. Comparison of Loading Methods

	CONTROL ACTION	LOAD DROOP EXISTS	SIMPLICITY	NARROW DEADBAND	HIGH ACCURACY	HIGH REGULATED PRESSURE	LIGHT-WEIGHT	LOW SENSI-TIVITY TO ACCELER-ATION AND VIBRATION	LOW SENSI-TIVITY TO TEMPER-ATURE
Weight loading	Reset	No	1	4	3	3	5	5	1
Spring loading	Proportional position	Yes	1	2	2	2	2	2	2
Preset pressure loading	Proportional position	Yes	1	2	2[a]	1	1	1	5
Constant pressure loading	Reset	No	2	3	2	1	3	3	4
Variable pressure loading	Proportional plus reset	No	3	1	1	1	4	4[b]	3[c]

(1) Best alternative
(2)-(5) Less desirable alternatives
(a) Temperature sensitive
(b) Unless compensator added
(c) If compensator added

against the actuator diaphragm tends to close the metering valve, while the constant loading pressure, together with the weight of the parts and pressure differential unbalance across the poppet, tend to open the valve. To eliminate the effects of the weights of internal parts, a small equalizing or bias spring is sometimes used. The spring will usually increase stability and reduce pulsation.

Variable Pressure Loading. By adding a control pilot to a constant pressure loaded regulator, the loading pressure can be made responsive to changes in regulated pressure. The reference force is said to be established by a variable pressure. Such a regulator is discussed in Sub-Topic 5.4.5.

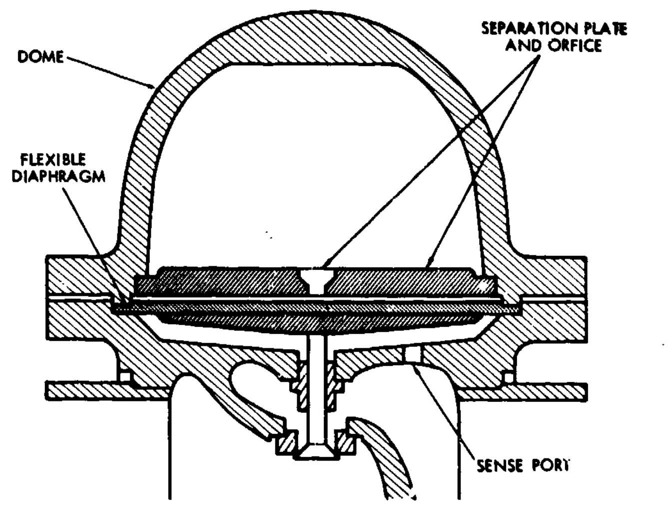


Figure 5.4.8.5b. Separation Plate and Orifice for Preset Pressure Regulator
(Courtesy of Grove Valve and Regulator Company, Oakland, California)

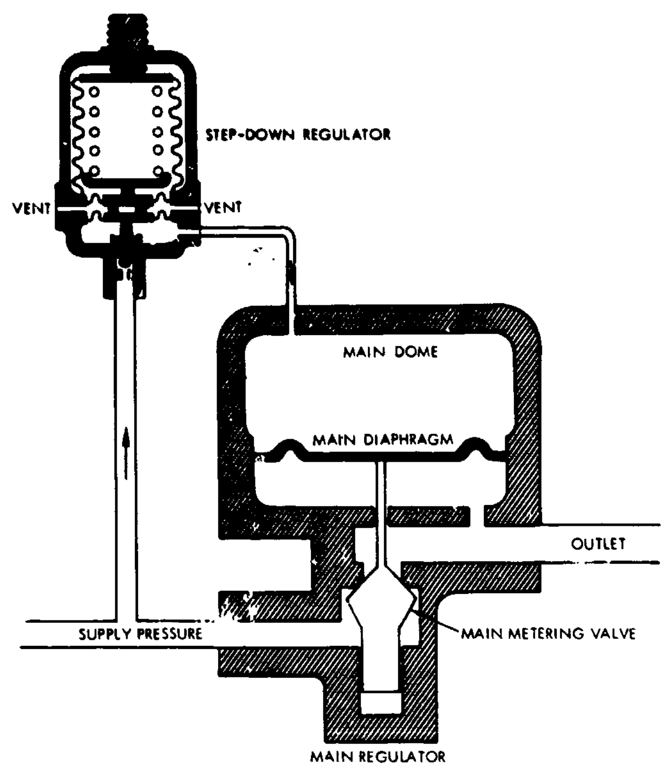


Figure 5.4.8.5c. Constant Pressure Loaded Regulator
(Reference 186-1)

A more basic model of a piloted regulator is shown in Figure 5.4.8.5d. The regulator consists of an actuator which is loaded at constant pressure by a step-down regulator. A control pilot is installed between the step-down regulator and the main dome of the regulator. (The control pilot is sometimes referred to as a controller.) The operation of the regulator is as follows:*

*Explanation of regular operation was adapted from Reference 186-1, pp. 12-14.

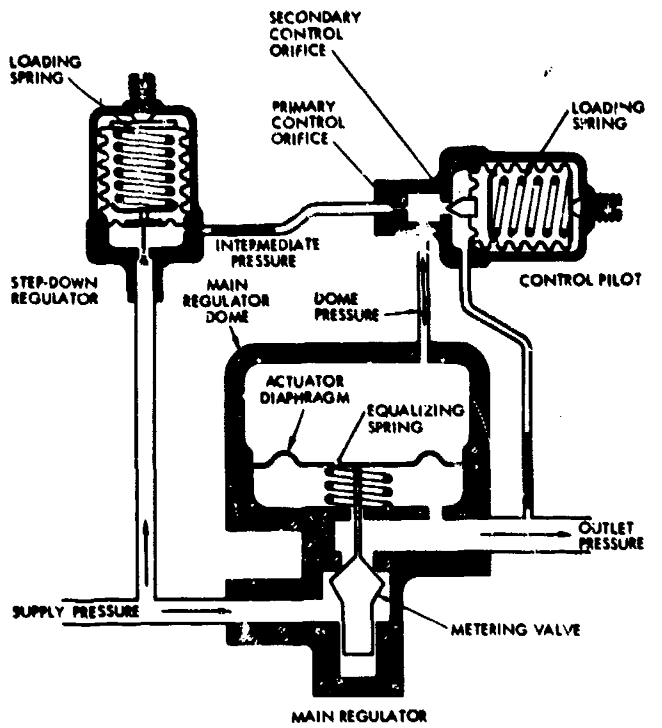


Figure 5.4.8.5d. Variable Pressure Loaded Regulator
(Reference 186-1)

Consider a system in which an imaginary hand valve is located and closed upstream of the regulator, the step-down regulator is wide open, the main regulator metering valve is closed, and the control pilot is held closed by the loading spring. If the upstream hand valve is now opened, the main regulator metering valve remains closed and prevents the gas from passing downstream. At the same time, however, gas flows through the step-down regulator until the pressure in the intermediate pressure line reaches the pressure for which the step-down regulator was set. The control pilot operating as a back pressure regulator then adjusts the pressure downstream of the primary control orifice to maintain the dome pressure at the proper pre-determined level. The pressure in the dome acting against the equalizing spring moves the diaphragm, opening the main metering valve. The valve remains open until the regulated pressure, plus the equalizing spring force, equals the pressure force above the diaphragm.

There exist in the regulator four values of pressure: inlet pressure, intermediate pressure, dome pressure, and outlet pressure. As long as the outlet (regulated) pressure is less than the set point, the control pilot is closed and the dome pressure is greater than outlet pressure. As downstream pressure increases to a value near the set point, the control pilot starts to open, reducing dome pressure to a value only slightly higher than outlet pressure.

5.4.9 Adding a Relief Function

To dampen large variations in regulated pressure, or to prevent dome overpressure, a relief or bleed is commonly provided. Provision for relief can be a separate relief valve integrated or attached to the regulator, or can be incorporated into the regulator as part of the main actuator, step-down regulator, or control pilot.

5.4.9.1 RELIEF AS PART OF THE ACTUATOR. In spring-loaded regulators, relief of regulated pressure can be accomplished by either an orificed or a hollow diaphragm seated against a plug which is made part of the actuator valve stem. A regulator with an orificed diaphragm is shown in Figure 5.4.9.1a. During the regulation cycle, the diaphragm and actuator stem move as one element. If regulated pressure becomes too high due to flow changes downstream or failure of the regulator control element to close, the diaphragm lifts off the plug and gas is vented to the atmosphere through the spring cavity. The hollow diaphragm, shown in Figure 5.4.9.1b, operates in the same manner as the orificed diaphragm, the only difference being that gas is vented to the atmosphere through the diaphragm rather than through the spring cavity.

Relief can also be made part of a disjointed metering valve stem. Referring to Figure 5.4.9.1c, the actuator diaphragm and its connecting plate are in contact with a spring-loaded stem during normal operation. If regulated pressure increases to a value above lockup, continued diaphragm deflection removes the disc poppet from its relief port, permitting gas to discharge to atmosphere.

5.4.9.2 RELIEF AS PART OF THE STEP-DOWN REGULATOR. To prevent dome overpressure in a constant pressure-loaded regulator, relief can be provided in the step-down regulator in the same manner as described in the preceding section. The relief discharges to the atmosphere whenever high pressure occurs in the dome.

Another method to prevent dome overpressure is the use of a continuous bleed from the step-down regulator. By connecting the main regulator dome and a bleed line in parallel to the step-down regulator, a constant pressure can be established in the dome. As the diaphragm deflects, causing a higher dome pressure, excess gas can discharge through the bleed line to atmosphere; or if the amount of pressurizing gas is limited, into the downstream line. However, when bleeding to the main line, the loading pressure must be greater than the downstream pressure.

5.4.9.3 RELIEF AS PART OF THE CONTROL PILOT. Relief as part of the control pilot is depicted in Figure 5.4.9.3, which is part of the regulator shown in Figure 5.4.5. Overpressure in the dome forces the poppet open, allowing dome loading gas to discharge downstream. The gas is contained within the system, utilizing the full amount of stored upstream gas for pressurization of downstream tankage.

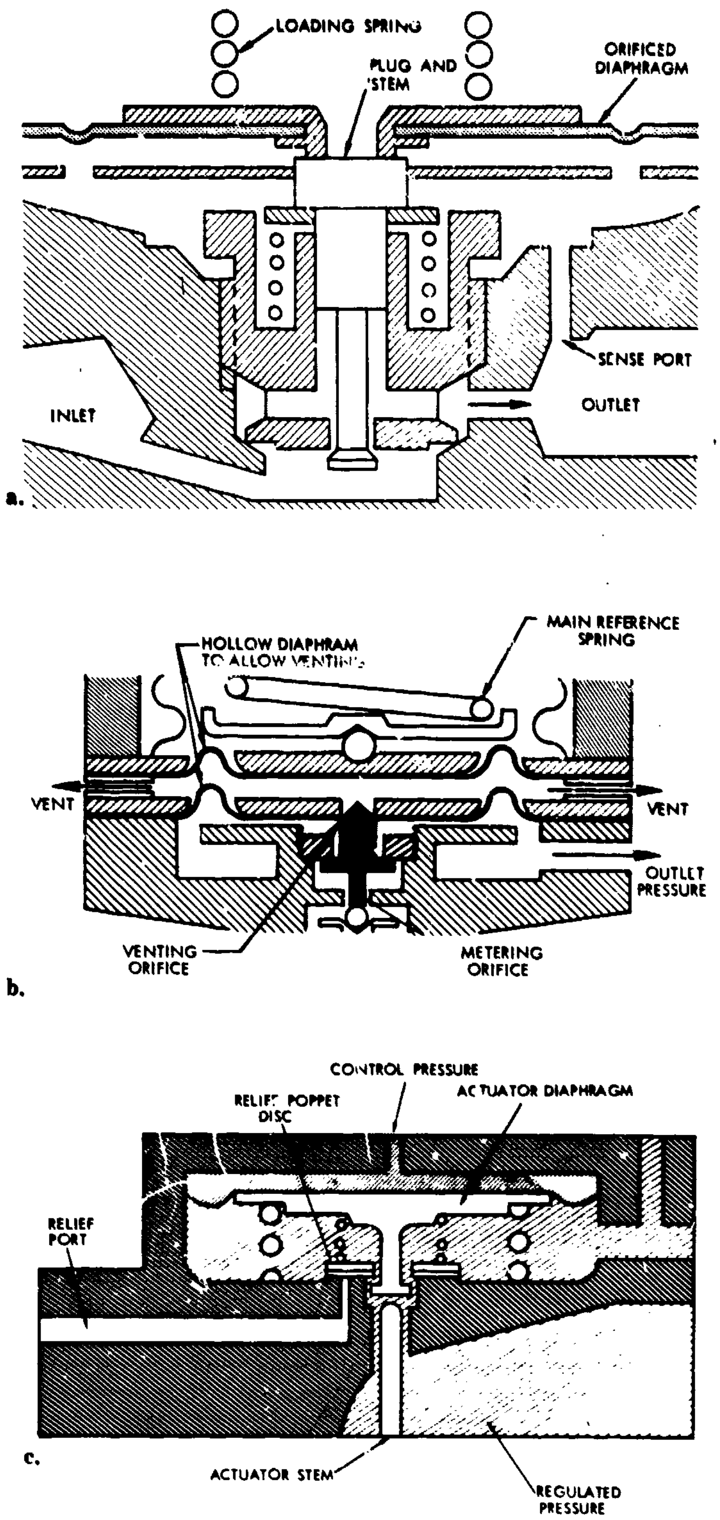


Figure 5.4.9.1a,b,c. Ways of Adding a Relief Function: (a) Orificed Diaphragm and Plug (b) Hollow Diaphragm (c) Disjointed Actuator Stem

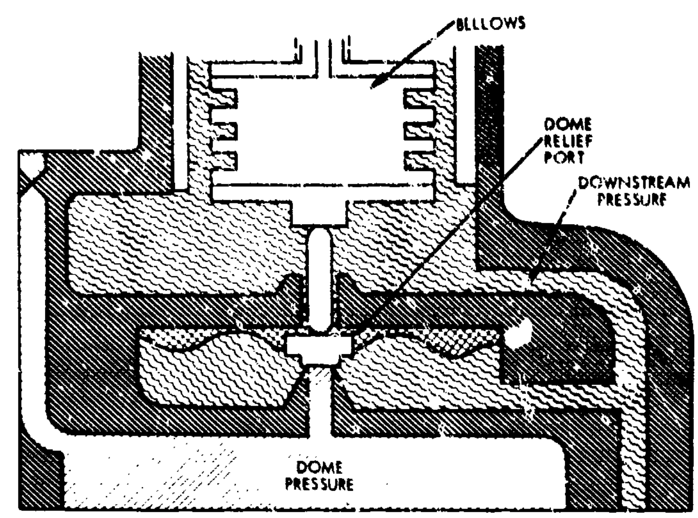


Figure 5.4.9.3. Use of Control Pilot for a Relief Function
(Reference 35-1)

5.4.10 Adding a Shutoff Function

A regulator is usually closed by its own control element when regulated pressure under the diaphragm overcomes the loading forces. This shutoff closure is called lockup. In weight- and spring-loaded regulators, the regulated pressure must be higher than the set point before the regulator will close. To effect shutoff at any regulated pressure below lockup pressure, a mechanical override must be used. With constant pressure or variable pressure-loaded regulators, shutoff may be accomplished at any pressure below that which causes regulator lockup by relieving loading pressure through a 3-way valve inserted between the step-down regulator and the main dome. Referring to Figure 5.4.5, actuation of the solenoid closes the shutoff valve and vents the main dome downstream, eliminating the load pressure. The step-down regulator is then closed by increased pressure in its downstream line. The main regulator is closed by downstream pressure under its sensing element.

5.4.11 Compensators

5.4.11.1 TEMPERATURE. The low temperatures commonly experienced in airborne applications change the regulator set point by thermal contraction. In spring-loaded regulators, compensation for contraction can be done by insulating the spring from the temperature or by housing the spring in a cage. In each case it is a rather difficult problem to determine with accuracy the thermal mass, heat transfer, and relative exposure to changes in temperature. Several types of thermal compensators are described in Detailed Topic 13.5.2.3.

5.4.11.2 VIBRATION AND ACCELERATION. Vibration poses one of the most difficult problems to assess before fabrication and to correct after valve failure. The vibration specification may require regulation at acceleration levels as high as 25 g's at frequencies from 7 to 2000 cps along any axis.

Particularly susceptible to vibration are poppets, springs, and bellows. Dampening elements, such as pneumatic dashpots, are added at critical points. Increases in coulomb friction and spring rate have been tried with varied degrees of success.

Acceleration requirements, specified for airborne regulators, are commonly 10 to 15 g's along any axis. To compensate for g loads, a counterbalance method has been used with reasonable success. Inasmuch as it is the regulator set point that is to be maintained constant, the compensator is commonly built into the control pilot. The counterbalance compensator is simply a flexure pivoted counterweight that exerts a force on the sensing element in the direction of the acceleration vector. A compensator of this design is shown in Figure 13.3.3b, where the force required to accelerate the masses of the spring element, pilot push pin, and dashpot diaphragm is balanced by the compensator beam and weight.

In tests conducted on the regulator shown in Figure 5.4.5, the maximum average deviation of regulated pressure at accelerations of 15 g's was 0.34 psi. Without acceleration compensation, the regulated pressure error was estimated to be 0.75 psi.

5.4.12 Flow Limiters

A flow limiter is usually a redundant control pilot incorporated into a regulator to prevent the flow rate from exceeding a predetermined value, usually fixed by the capacity of the system. It is used primarily as a safety device in the event dome loading pressure becomes large due to failure of the bleed regulator or control pilot. As the loading pressure increases, the regulated pressure set point and flow through the regulator increase proportionately.

A flow limiter operates by continually sensing the pressure changes (total, differential, or static) as a function of flow rate through a constant area restriction. The flow limiter setting is made for a flow rate at some nominal value above that normally controlled by the control pilot, but below the capacity of the system. During the normal regulation cycle it will be inoperative. However, as the flow rate through the regulator increases to the setting of the flow limiter, the pressure change, sensed at the restriction, causes the flow limiter to discharge excess dome gas to the atmosphere. The loading pressure is reduced to a value sufficiently low so that the regulator is able to start closing due to the downstream pressure force under the main sensing element. A regulator with a flow limiter is presented in Reference 33-1.

In some instances, a flow regulator (Detailed Topic 5.4.15) is installed as a flow limiter upstream of the main regulator. If the main regulator fails, the flow rate will not become greater than system capacity.

5.4.13 Control Modes of Modulating Regulators

Three types of control action or modes are found in modulating regulators. These are proportional-position, reset, and proportional plus reset.

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The proportional-position control mode occurs with spring-loaded and pre-set pressure-loaded regulators. The reset mode occurs with weight-loaded and constant pressure-loaded regulators. Proportional plus reset action occurs with piloted, variable pressure-loaded regulators. Each mode is explained in the following Detailed Topics.

5.4.13.1 PROPORTIONAL-POSITION ACTION. Proportional-position action causes variation in control element position to be proportional to the deviation of regulated pressure from the set point.

To illustrate the concept of proportional-position action, consider a spring-loaded regulator controlling pressure in a tank partially filled with liquid. Tank pressure is transmitted by a sense line to the underside of the diaphragm, the motion of which is opposed by the spring force. If initially the system is in equilibrium, the flow of gas through the regulator is sufficient to maintain a constant pressure in the tank for a given liquid flow demand out of the tank. If the flow demand out of the tank increases instantaneously, flow-out exceeds the flow of gas into the tank and tank pressure decreases. The spring force is now greater than the pressure force under the diaphragm. The diaphragm deflects and the spring is extended, opening the metering valve further. Since the spring is extended, the spring force is reduced and the regulator set point is lowered. The decrease of regulated pressure from the set point is called offset and exists for any flow other than the flow rate for which the setting was made. Tank pressure will decrease until flow-in equals flow-out.

If liquid flow demand returns to the original value, tank pressure will rise until the original set point is reached. Further decrease in liquid flow will cause the tank pressure to rise, tending to close the regulator. If liquid flow stops, the regulator will move to the fully closed or lockup position. These relationships are illustrated in Figure 5.4.13.1.

5.4.13.2 RESET ACTION. Reset action causes a change in the control element position proportional to the time integral of the error (deviation from set point). In other words, the control element changes position at a rate proportional to the deviation from the regulated pressure, and the total change in the control element position is a function of the rate of travel and the lapsed time of the deviation from the set point.

The reset action continues as long as the regulated pressure is off the set point, reversing only when the pressure reaches the set point. Reset action may best be described by considering a weight or constant pressure-loaded regulator initially closed with downstream tank pressure above the set point. If flow from the tank increases to a constant value instantaneously, tank pressure will decay. When tank pressure decreases to the point where it no longer supports the constant force load (weight), the regulator opens fully, permitting gas to enter the tank. When the tank pressure summed over the diaphragm area increases to a value just above that required to support the weight, the regulator starts to close. Tank pressure continues to increase until the regulator closes. With a continuing flow demand from the tank, tank pressure will then start to decrease and the cycle will repeat. The tank pressure will oscillate about the regulator set point until dampening within the regulator establishes a uniform flow. These relationships are depicted in Figure 5.4.13.2.

5.4.13.3 PROPORTIONAL PLUS RESET. The inherent disadvantages of offset and load-droop of spring loaded or pre-set pressure-loaded regulators, and the oscillations of undamped, weight-loaded regulators and constant pressure-loaded regulators are reduced by variable pressure loading which combines reset action and the proportional-position action.

For an example of proportional plus reset action, consider the following: As regulated pressure in a tank partially filled with liquid decreases following an increase in liquid out flow, the gas flow starts to increase and continues to increase until such time as the proportional action calling for a decrease in gas flow is greater than the reset action still calling for a flow increase. Since at this time the regulated pressure is increased (gas flow-in being greater than liquid flow-out), a decrease in gas flow is desirable, the objective being to match flow-in with flow-out at the time regulated pressure reaches the set point. Typical flow and pressure diagrams are shown in Figure 5.4.13.3.

5.4.14 Absolute Pressure Regulators

In an absolute pressure regulator, the reference load is independent of changes in the barometric pressure. This is accomplished by providing a hermetic seal in the loading chamber. Such regulators are useful in closed systems, for example in hydraulic systems, where changes in differential pressure across tanks or reservoirs because of changes in altitude would be inconsequential. In tank pressurization systems, however, an atmospheric reference for the regulator load is always used, permitting the reference load to vary with the barometric pressure. In this manner, the differential pressure across the tank for a given regulator setting is constant, regardless of the altitude.

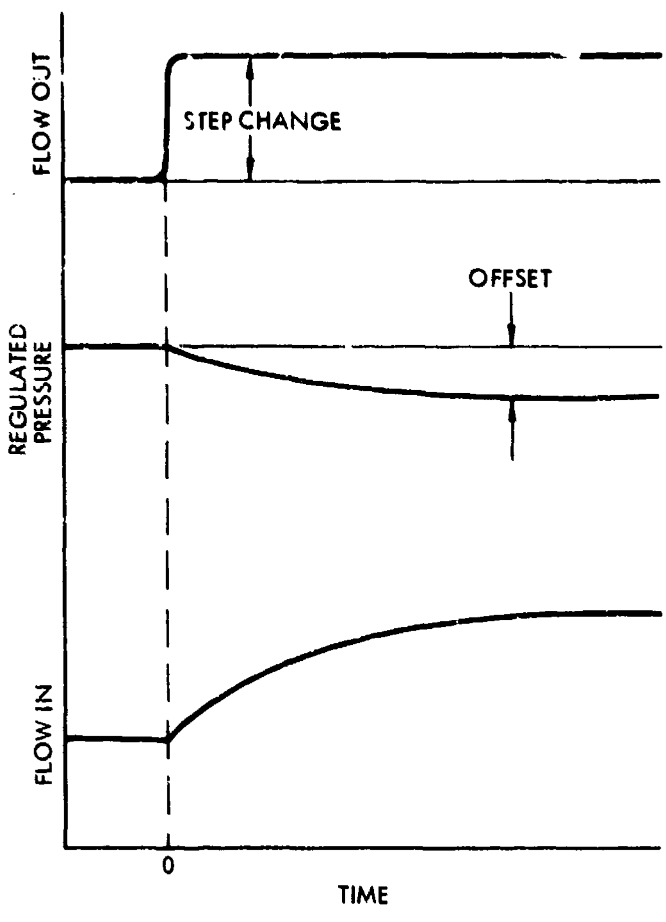

Figure 5.4.13.1. Pressure and Flow Rate Variations with Proportional Position Action

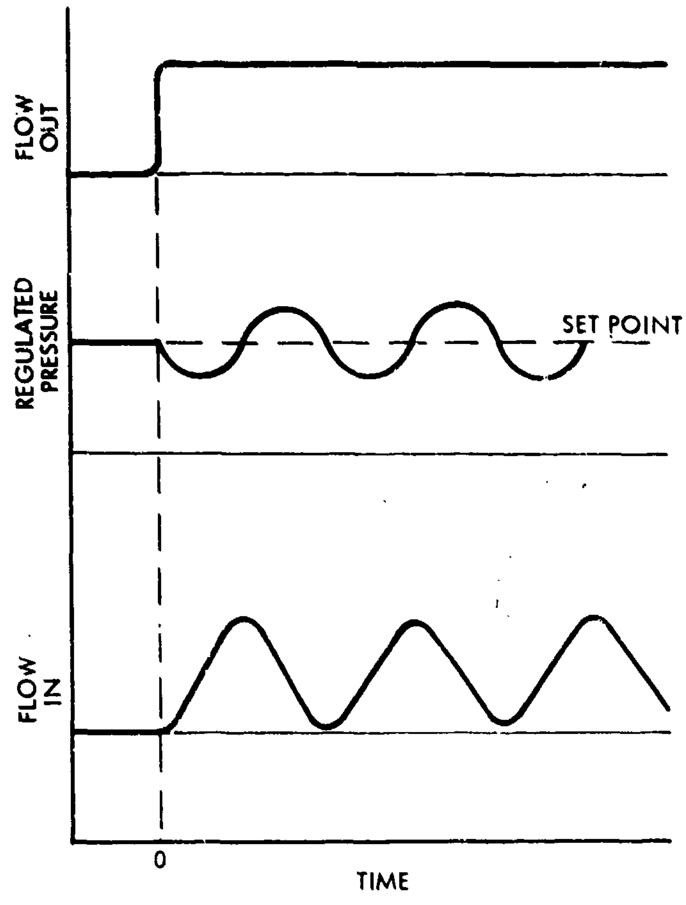

Figure 5.4.13.2. Pressure and Flow Rate Variations with Reset Action

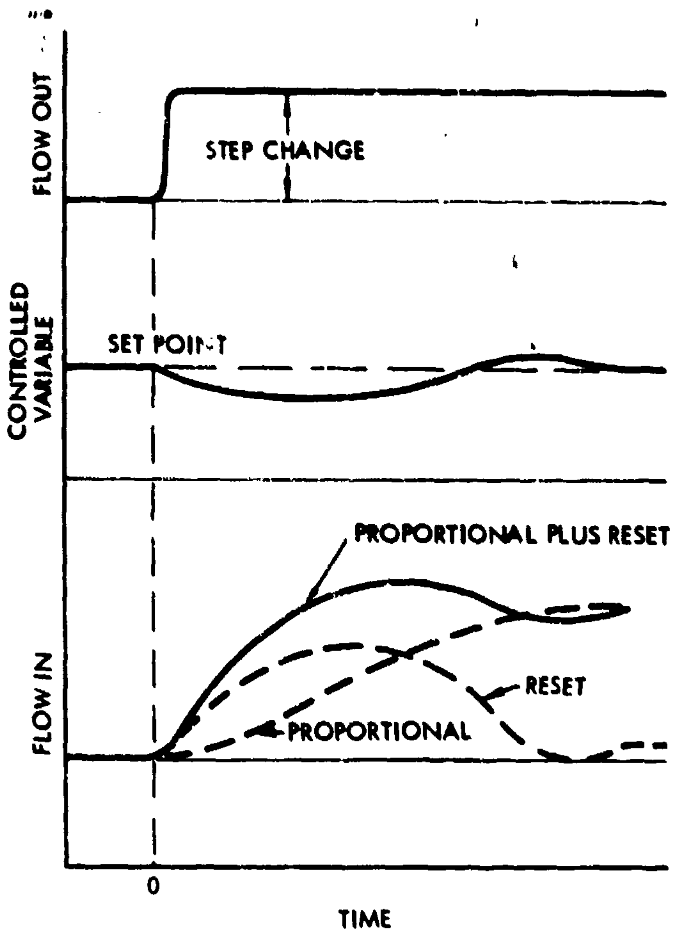

Figure 5.4.13.3. Variations of Pressure and Flow with Proportional Plus Reset Action

5.4.15 Differential Pressure Regulators

A differential pressure regulator measures the volumetric flow rate of a liquid in a line and maintains it at a constant pre-determined value.

Flow measurement is performed by a variable head flowmeter (e.g., an orifice), installed in the main line, with static pressure taps located at two points in the flow stream. Flow rate through a variable head flowmeter is a function of pressure drop. If the pressure drop across the meter can be maintained at a constant value, the flow rate will be constant.

Maintenance of constant flow is performed by a pressure-reducing regulator with the static pressures from the flow meter impressed on opposite sides of the regulator sensing element. An adjustable effect is created by a spring on one side of the diaphragm. The spring force, properly adjusted, will balance the low value of static pressure against the high value.

A direct-acting flow regulator, shown schematically in Figure 5.4.15a, maintains a constant flow rate by keeping a constant pressure drop across a pre-set orifice, regardless of variations in either supply or discharge pressure. A constant pressure drop across the orifice is maintained by a pressure balanced control valve which acts as a variable orifice downstream of the pre-set orifice. The pressure drop across the pre-set orifice is impressed across a sensing element of the control valve. Any variation of pressure drop across the pre-set orifice will cause the control valve to change the variable orifice opening, as required, so as to restore the pre-set pressure drop.

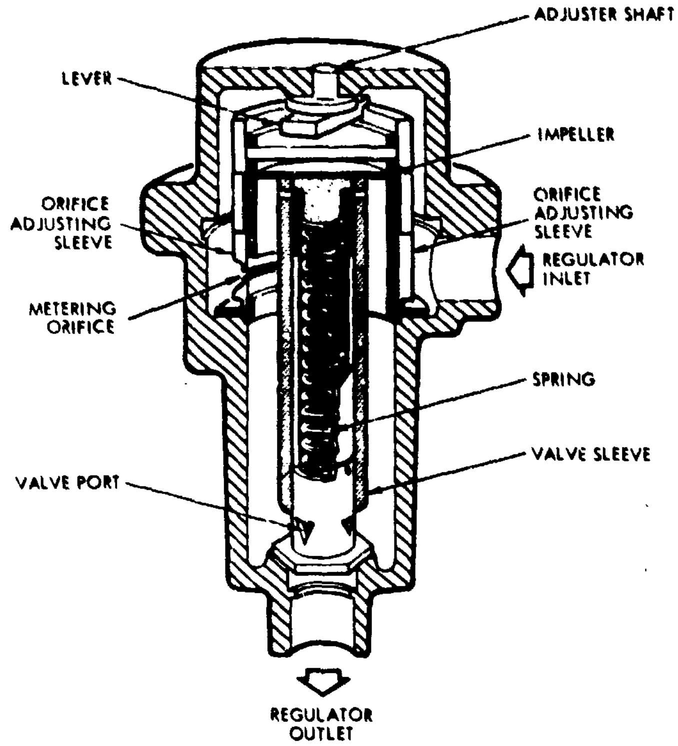

Figure 5.4.15a. Direct-Acting Flow Regulator
(Courtesy of the W. A. Kates Company, Deerfield, Illinois)

A pressure-reducing regulator can be used as a pilot valve, as shown in Figure 5.4.15b. Differential pressure across an orifice plate is sensed by the spring-loaded pilot regulator. The control element is a flexible tube valve diametrically pinched by pressure, controlled by the pilot regulator.

A slight increase in pressure differential due to an increase in flow will force the pilot valve open against its spring load; this allows upstream pressure to pinch the main valve control tube, increasing the pressure drop across the main valve during flow. For a decrease in pressure differential, the converse is true.

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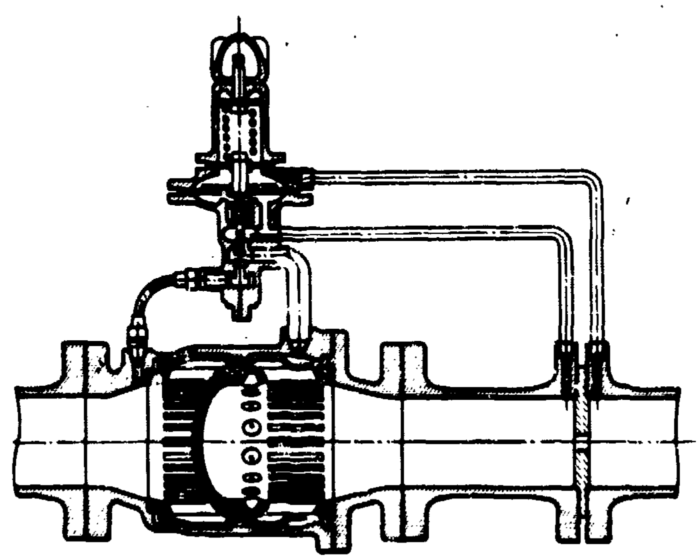


Figure 5.4.15b. Piloted Differential Pressure Regulator
(Courtesy of Grove Valve and Regulator Company, Oakland, California)

5.4.16 Effect of Upstream Pressure on the Metering Valve

Gas flowing past a flat or conical poppet metering valve establishes an unbalanced force, which becomes a function of the pressure difference across the control element as it moves during regulator operation. The unbalanced force tends to open or close the valve, depending upon the physical location of the control element and inlet port relative to the valve seat. The position alternatives are listed in Table 5.4.16.

Table 5.4.16. Valve Position Alternatives

FLOW ACTION	CONTROL ELEMENT POSITION	INLET PORT
1. Close	Under seat	Under seat
2. Close	Over seat	Over seat
3. Open	Under seat	Over seat
4. Open	Over seat	Under seat

Valves with flow action tending to close (1, 2) are called direct-acting; valves tending to open (3, 4) are called reverse-acting.

The higher the upstream gas pressure, the larger the unbalanced force across the regulator control element tending to open or close the valve. The idealized force equation for regulator equilibrium is

$$F_L = p_2 A_{se} \pm A_{sl} (p_1 - p_2) \quad \text{(Eq 5.4.16)}$$

where F_L = load (reference force)
p_1 = upstream pressure
p_2 = regulated pressure
A_{se} = sensing element area
A_{sl} = seat land area

The last term is positive for direct-acting valves and negative for reverse-acting valves. Equation (5.4.16) indicates that, first, if the pressure differential, p_1-p_2, across the seat is large and the load is constant, the unbalanced force across a large poppet could be comparable to the regulated pressure force against the sensing element area. Second, for a constant load, as the upstream pressure decreases the regulated pressure will change, increasing for the direct-acting valves and decreasing for reverse-acting valves. Spring pre-set pressure and variable pressure-loaded regulators compensate by varying the load. Regulator staging will reduce the extent of load variation. Third, to maintain a constant regulated pressure, a larger load (reference force) is required for direct-acting valves than for reverse-acting valves. To nullify the unbalanced force with single poppets, dual poppets and seats are used, called double-seated valves. By design, double-seated valves incorporate flow actions 1 and 3, or 2 and 4, listed in Table 5.4.16. The flow actions are opposed, cancelling 90 to 95 percent of any unbalanced force. The net force generated is the difference in areas of the two poppet seating surfaces. The areas are not always equal, since assembly requires passage of one poppet through the seat of another. A marked disadvantage of these valves is that simultaneous seating is difficult to achieve.

5.4.17 Effect of Ullage Volume

The volume of gas above the liquid in the tank is called *ullage*. Ullage provides a sufficient amount of gas under lockup pressure to maintain a positive pressure in the tank until the regulator has time to open and take control once flow starts. In specifications the minimum ullage is defined, since this is the least amount of gas available to maintain positive pressure. The amount of ullage is based on expected initial flow from the tank and maximum step time required by the regulator. Minimum ullage volumes for missile tank pressurization systems vary over a wide range from 2 to 43 cubic feet.

5.4.18 Effect of Pressure-Sensing Location

Depending upon system geometry, regulated pressure may be sensed internally or externally. Internal sensing is performed at the outlet of the regulator, measuring either static or total pressure in the downstream line. The static pressure tap is a passage connecting the downstream line to the underside of the diaphragm. Total pressure measurement presents a much closer approximation of tank pressure than static pressure measurement and is measured by various configurations of pitot tubes. Internal sensing is adequate only when the regulator is located in close proximity to the tank where pressure is to be controlled. A pressure drop will occur through the downstream line, causing a lower pressure in the tank than that which is sensed by the regulator. To compensate for this expected loss, a variable restriction can be installed in the downstream line which trims the line for an exact pressure drop for a given rate of flow. The regulator is said to be tuned for the one expected flow demand from the system. For any other flow conditions, regulated pressure will be different from the desired value.

For most flight applications, sensing should be done externally. External sensing uses a pressure return line from the tank to the underside of the regulator diaphragm, and/or to the control pilot. Pressure can be assumed to be its static value unless the diaphragm and valve stroke are long, requiring a flow of gas in the sense line from the tank to fill the void left by stroking.

Increasing the length of the transmission line increases the process lag time and, in addition, decreases the stability of the closed-loop system. A rigorous mathematical analysis is not given here; References 45-1 and 141-1 give a generalized approach to the subject.

REFERENCES

21-5	141-1	186-1
33-1	144-2	V-56
35-1	147-15	V-192
45-1	147-16	V-232
52-74	160-42	V-234
70-5	160-44	V-282
89-3	160-46	V-289

5.5 RELIEF VALVES

5.5.1 Introduction

This Sub-Section discusses relief valves — valves which control fluid pressure in a tank or system by discharging excess fluid to a lower pressure reservoir.

A relief valve is a pressure-relieving device which opens automatically when a pre-determined pressure is reached. Relief valves may have a full opening "pop" action or may open in proportion to the over-pressure. Those valves opening rapidly to full flow are generally referred to as *safety valves* or *"pop" valves* and are considered a special form of relief valve. A relief valve can be contrasted to a vent valve, which is an externally operated pressure-relieving device and is characteristically a shutoff valve. The term *vent* describes the application of the shutoff valve. (Sub-Section 5.2 discusses shutoff valves.)

5.5.2 Applications and Specifications

For industrial applications and ground systems, codes have been established to regulate the use of relief valves and safety valves. The ASME Power Boiler Code and the ASME Unfired Pressure Vessel Code are perhaps the most familiar to engineers. In addition, each state has published its separate code, patterned to a large degree after the ASME codes. As a further supplement, special regulations have been enacted suitable to specific installations, e.g., the U.S. Coast Guard Marine Engineering Regulations. These codes define orifice size, rated set pressure, flow rate capacity, etc. According to most codes, relief valves, as distinguished from safety valves, are specified primarily for hydraulic systems. Excess pressure is usually relieved by diverting the liquid to a reservoir of lower pressure. Safety valves are used to relieve excess pressure in pneumatic systems by discharging the gas (or vapor) to the atmosphere.

In aerospace airborne systems, relief valves are used for both liquid and gas service and are differentiated by prefacing the fluid to be used, e.g., LO_2 relief valve, helium relief, etc. Pressure is relieved by discharging gas or liquid overboard. Because of the exacting requirements for pressure control and fluid utilization in airborne systems, full opening safety ("pop") valves are seldom used.

For most airborne systems, safety and relief valve design requirements are stipulated in military and prime contractor specifications. It is not unusual to find such requirements more stringent than those defined by a code, because of the necessity for optimum fluid utilization. A typical specification for relief valves is presented in Section 9.0, "Specifications."

5.5.3 Description of Valve Operation

A relief valve consists of a valve body, a reference load, and a closure, the closure being a control element and seat. The reference load is linked to the closure and opposes the pressure build-up in the tank or system. The magnitude of the load determines the relief pressure setting. As the internal system pressure increases near the relief pressure necessary to balance the reference load, leakage usually begins. When the internal pressure reaches the relief pressure, the valve opens and discharges the upstream fluid to a lower pressure reservoir. As internal pressure decreases below the set pressure, the reference load opposing the pressure force closes the valve.

The operation of a relief valve may be conveniently illustrated by a so-called regulation or hysteresis curve shown in Figure 5.5.3. A relief valve is considered to have good operating characteristics if the pressure for rated flow, p_f, and reseat, p_r, closely approach the cracking pressure, p_c. The cracking pressure is the relief pressure setting of the valve, defined as the pressure where leakage flow reaches some specified value. The cracking pressure is always set below the allowable working pressure of the tank or system, and is commonly not more than 110 percent of normal operating pressure. The rated capacity is usually established for flows at pressures 10 percent greater than the pressure setting of the relief valve. The reseat pressure is some value below the cracking pressure, depending upon the closure configuration. A reseat pressure of 95 percent of cracking pressure is not uncommon.

Relief valves may be either *direct-acting* or *piloted*. Direct-acting relief valves can be either the *conventional* type where the control element moves relative to the seat, or the *inverted* type, where the seat moves relative to the control element.

Although not called relief valves by name, safety rupture discs which have no reclosure capability, and vent valves actuated with pressure switches perform a relief function and are included in the broad classification of relief valves. A pressure-reducing regulator can be used as a relief valve, and in this capacity is called a *back-pressure regulator*.

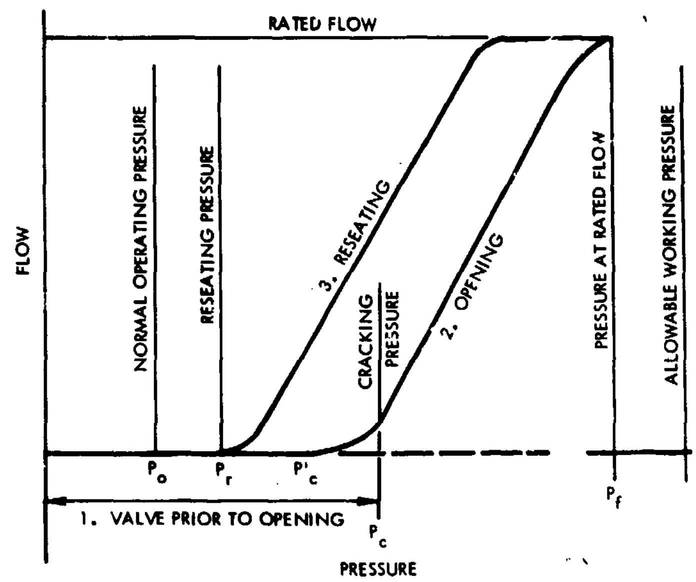


Figure 5.5.3. Hysteresis Curve for a Typical Relief Valve

5.5.4 Relief Valve Body

For airborne applications, the relief valve body is designed for minimum weight, consistent with its pressure rating and for passage of high flows with minimum pressure loss. Lightweight construction materials such as aluminum are used extensively to achieve minimum weight. To achieve minimum pressure drop, some manufacturers use a venturi design in the discharge side, while others enlarge the outlet port even to the extent of using larger connections.

5.5.5 The Reference Load

The reference load is the force opposing any pressure buildup until relief pressure is reached. The most common element used to establish the load is a compression spring. Weights could accomplish the same purpose, but are seldom used. Only spring-loaded relief valves will be considered in this Sub-Section.

5.5.6 The Valving Unit

The valving unit is composed of a seat and control element. The seat may be flat, spherical, or conical in shape, the seat configuration determining to a large degree the sealing and opening characteristics of the relief valve. These characteristics are discussed in Detailed Topic 5.5.7.1.

Four commonly used control element designs are shown in Figures 5.5.6.a through 5.5.6f. Each control element is discussed briefly here, and a more extensive treatment is presented in Sub-Section 6.2, "Valving Units."

5.5.6.1 THE BALL. The ball control element, shown in Figure 5.5.6a, is used in both quick-opening and proportional relief valves. It is used extensively because of its simplicity, low manufacturing costs, and inherent self-aligning capability as it reseats. When used in a quick-opening relief valve (safety valve), the ball tends to chatter and to be noisy when discharging fluid. It is limited to small valve sizes and has short life cycle performance.

5.5.6.2 THE CONICAL POPPET. Like ball control elements, the conical poppet may be used in both quick-opening and proportional relief valves. Conical poppets lend themselves to larger port sizes but require closer tolerances on the sealing surfaces of the poppet and seat. The control element stem must be guided to obtain alignment between the poppet and seat. Three design variations of the conical poppet with guides are shown in Figures 5.5.6b, c, and d. In Figure 5.5.6b, the small diameter stem is guided by a sleeve, and the loading spring is external to the stem guide. In Figure 5.5.6c, the stem has the same diameter as the poppet and the guide is made part of the body. The spring is restrained inside the poppet cavity. The control element (a) compared to (b) is usually lighter weight and can accommodate a larger spring. The control element in (b) possesses a better reseating capability than (a) since the stem is more rigid and is less likely to deflect under high pressures and flows. The control element in Figure 5.5.6d compromises some of the benefits of (a) and (b). When the

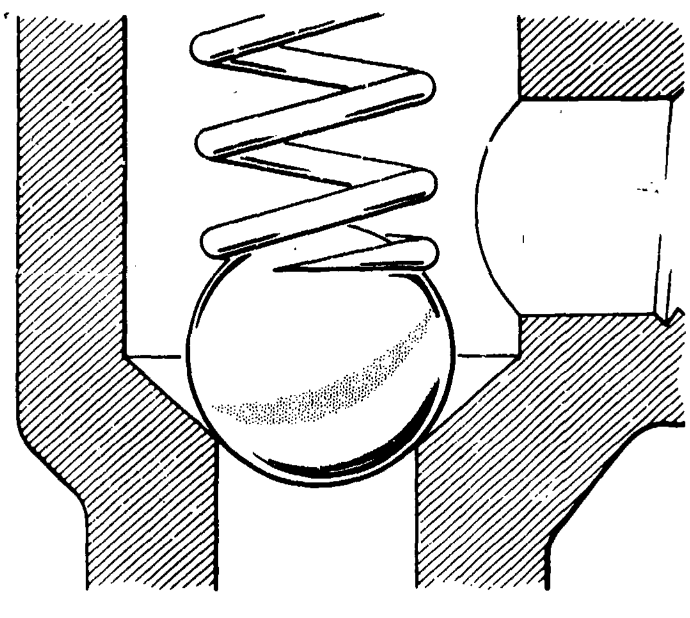


Figure 5.5.6a. Ball Closure

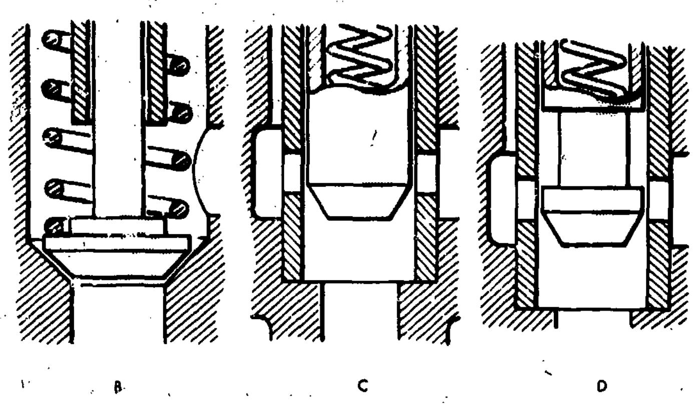


Figure 5.5.6b,c,d. Conical Poppet Control Elements

valve is open, however, the inside surface of the guide is exposed to the fluid. If gas is discharged, the cooling during expansion may result in an ice buildup on the guide which will prohibit the valve from closing. All three control elements shown are quieter in operation than the ball, due to frictional damping induced by the guide.

5.5.6.3 THE V-POPPET. The control element, shown in Figure 5.5.6e, is used only in the safety valve. As soon as the valve starts to open, the fluid by changing momentum due to the V-design exerts a greater force against the poppet, causing it to pop open to full flow. The poppet uses only its inner cone for a sealing surface. Like the conical plug, the poppet stem is guided. Precision machining is required to obtain accurate poppet-seat concentricity and alignment.

5.5.6.4 THE PISTON. A piston, shown in Figure 5.5.6f, is sometimes used in relief valves for closed hydraulic systems. The piston offers no positive seating surface to prevent leakage, but depends primarily on closed tolerances. Valve opening is proportional to the overpressure. Pistons are more commonly used as the second stage in pilot-operated relief valves than in single-stage valves.

5.5.7 Conventional Relief Valves

The operation of a relief valve can be divided into three distinct phases, illustrated in Figure 5.5.3:

1) Valve Prior to Opening
2) Opening Characteristics
3) Reseating Characteristics

Each phase is discussed separately in the following Detailed Topics.

5.5.7.1 VALVE PRIOR TO OPENING. At pressures below the cracking pressure, it is desired that the relief valve maintain a tight seal to prevent premature leakage, commonly called dribble. Sealing requires a minimum seating pressure (seating force per unit area) between the control element and seat. The seating force is the spring load less

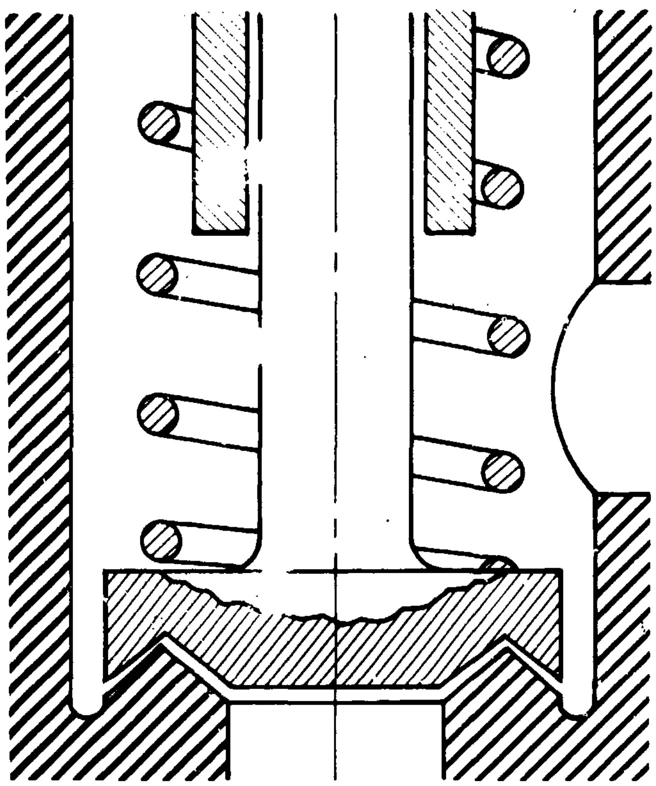


Figure 5.5.6e. V-Poppet Closure

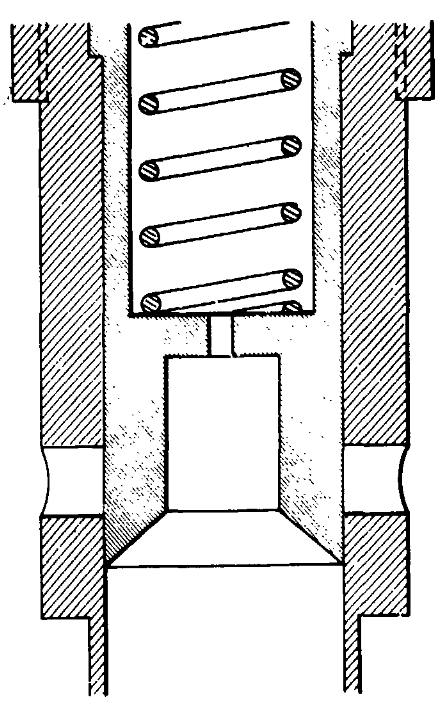


Figure 5.5.6f. Piston Control Element

any opposing tank pressure force. At zero tank pressure, the seating pressure is a maximum. As tank pressure increases to the cracking pressure, the seat pressure diminishes. When the seat pressure decreases below the minimum required to seal, leakage starts. This concept is illustrated in Figure 5.5.7.1. The difference between the cracking pressure, p_c, and the pressure where leakage starts, p_c', is called the dribble range. Parameter p_c' is the inlet pressure which causes the seat pressure to decrease to the minimum value to seal. (See also Figure 6.2.3.11u.)

To prevent leakage at pressures near relief, the seat area must be minimized to maintain an adequate seating pressure as the seating force is reduced. The minimum seat area is limited by the seat stress (bearing stress) at zero tank pressure that can be tolerated by the seat material. The seat area may be varied from essentially circumferential point contact, as with a control element seating against a

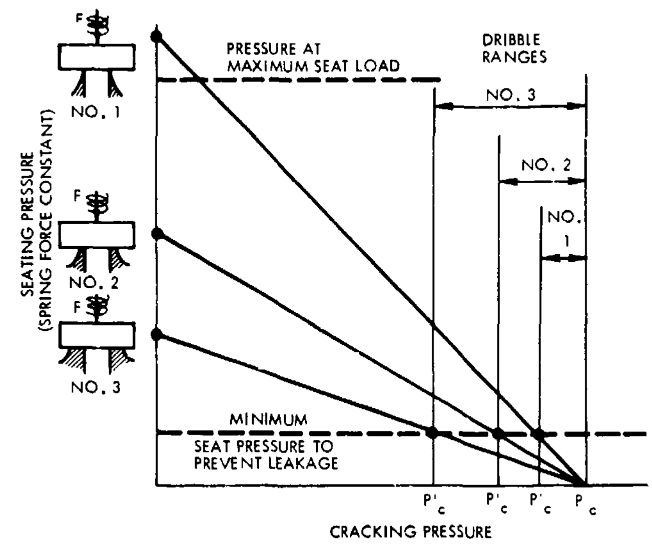


Figure 5.5.7.1. Effect of Seat Area on "Dribble Range"

sharp-edged orifice, to a finite interfacial area, resulting from a conical control element seating against a matched conical seat. For the same initial seat load, as the seat area increases, leakage will occur at lower tank pressures.

5.5.7.2 OPENING CHARACTERISTICS. When the tank pressure force increases to a value equal to the reference load, the control element is in equilibrium and the force balance can be represented by the equation

$$F = p_1 A_o \qquad \text{(Eq 5.5.7.2a)}$$

where F = spring force

p_1 = tank pressure

A_o = orifice area of seat

When the pressure force exceeds the spring force, the valve will open. As flow starts through the valve, a pressure drop is established across the seat width and increases the effective pressure force acting on the control element. The force balance equation then becomes (Reference 51-3)

$$F = p_t A_o + p_m (A_1 - A_o) \quad \textbf{(Eq 5.5.7.2b)}$$

where F = spring force

p_t = tank pressure

p_m = mean effective pressure

A_o = orifice area of seat

A_1 = poppet area, based on the seat outside diameter

Depending upon closure configuration, advantage may be made of the additional force to increase poppet lift. Contrasting designs are shown in Figure 5.5.7.2a and b. In (a) the seat area is small and the control element lifts a distance nearly proportional to the amount of overpressure. In (b) the value of (A-A_o) is large, and the additional force increases the poppet lift. For safety valves, the area difference is made large enough to open the valve quickly to its maximum flow. As the valve opens, an additional force is impressed against the control element, attributed to fluid frictional drag. Frictional drag is a function of Reynolds number and varies for different control element configurations. The additional force increases control element lift. A comparison of regulation curves for opening of the valve designs in Figures 5.5.7.2a, b is illustrated in Figure 5.5.7.2c.

As the valve opens further due to frictional forces imposed by the fluid, the pressure drop across the seat decreases momentarily, decreasing the pressure force opening the valve. The spring force having increased at valve opening tends to close the valve. The pressure drop across the seat is increased and the additional pressure force reopens the valve and the cycle repeats. This cyclic action of the control element is called *chattering*, and can be described as the hammering of the poppet against the seat at high frequency. Adverse effects of chattering are pitting, wear, and abrasion.

Chattering may be reduced by:

a) Diminishing the control element seat area difference, A-A_o.

b) Adding a dashpot effect in the spring cavity. This method is used extensively for relief valves in hydraulic systems. An example is shown in Figure 5.5.7.2d.

c) Using a pilot valve to initiate relief of the main valve. See Sub-Topic 5.5.9 for a discussion of piloted relief valves.

d) Using loading springs with negative spring rates to get snap action opening (See Detailed Topic 6.5.3.7, Belleville Springs.")

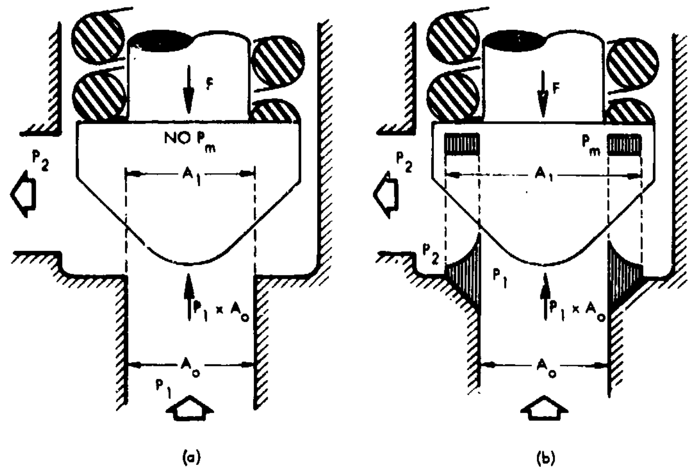


Figure 5.5.7.2a,b. Seat Configurations: (a) Sharp-Edged Orifice (b) Conical Seating Surface
(From "Missile Design and Development," June 1960, vol. 5, no. 6, S. Kowalski, Copyright Missiles and Space, Manhasset, New York)

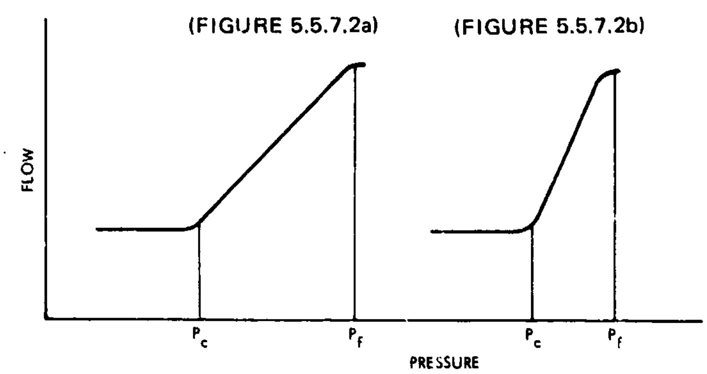


Figure 5.5.7.2c. Relief Valve Opening Characteristics

5.5.7.3 RESEATING CHARACTERISTICS. As tank pressure decays, the opening force (force opposing the reference load) decreases and the relief valve starts to close. Ideally, if the opening stroke were proportional to the overpressure, the control element stroke during the reseat phase will be proportional to the pressure decrease. However, frictional effects will create hysteresis, causing reseat pressures to be lower than the true cracking pressure. The magnitude of frictional effects can only be judged for each individual design. For closing the quick-opening relief valve, the tank pressure must decay below that required for proportional closing in order to nullify the increased opening force. This

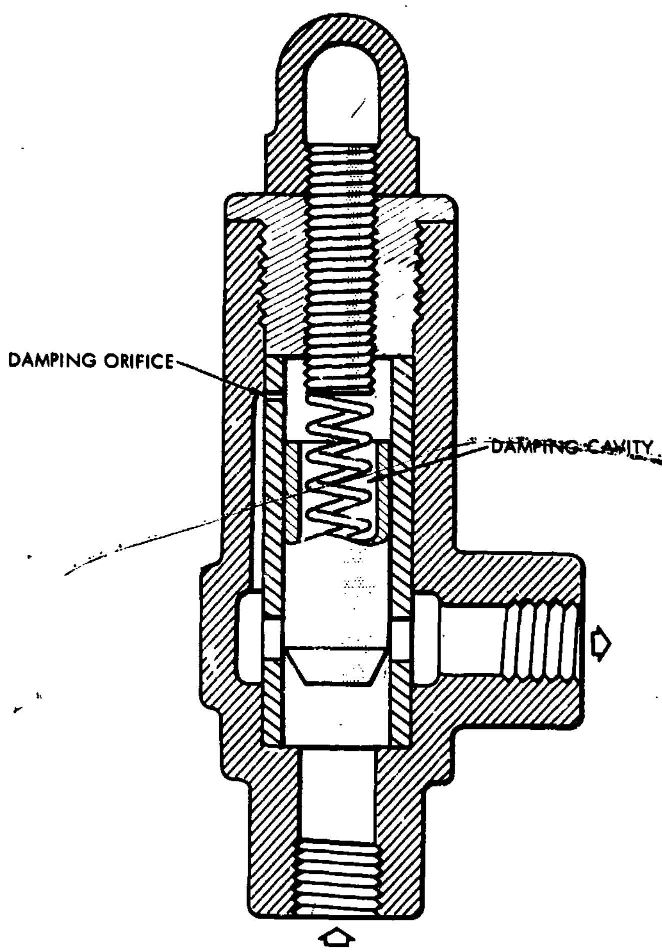


Figure 5.5.7.2d. Relief Valve with Damping Orifice and Cavity

(From "Machine Design," April 1948, vol. 20, no. 4, L. S. Linderoth, Jr., Copyright 1948, Penton Publishing Company, Cleveland, Ohio)

pressure offset plus frictional effects creates a broader hysteresis curve than with relief valve with proportional opening. A comparison of typical hysteresis curves are shown in Figure 5.5.7.3.

5.5.8 Inverted Relief Valve*

5.5.8.1 DESCRIPTION OF OPERATION. This valve inverts the control element and seat of the conventional relief valve such that the seat is spring-loaded closed against the control element (Figure 5.5.8.1a.) Sometimes both the control element and seat of the inverted valve are loaded by springs, but the seat-loading spring establishes the reference force which reacts against the increase in internal

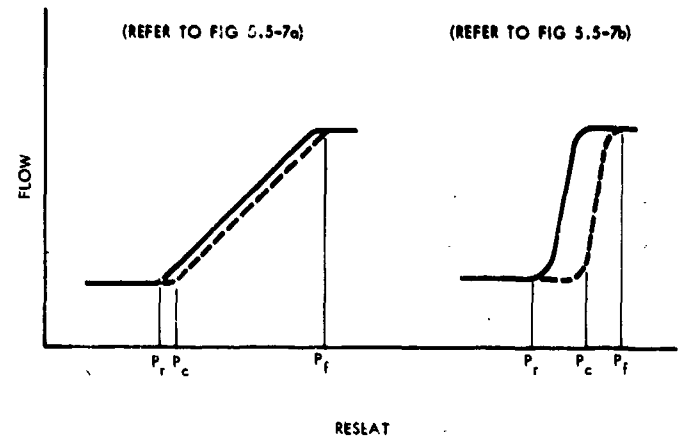


Figure 5.5.7.3. Relief Valve Reseating Characteristics

pressure. The seat and its accompanying control element move together, compressing the seat spring. As the inlet pressure rises, the ball is forced against the seat, increasing the seating pressure until the ball strikes the stop rod. Inlet pressure acting upon the annular area between the piston outside diameter and the seat outside diameter ($A_2 - A_1$) moves the seat away from the now stationary ball, causing the seating pressure to decrease rapidly until the valve opens. Valve operation from contact with the stop rod is similar to that of the conventional type relief valve. Seating pressure versus inlet pressure for an inverted relief valve is shown in Figure 5.5.8.1b.

As in conventional relief valves, the inverted type requires a positive seat load to prevent leakage. Below the minimum

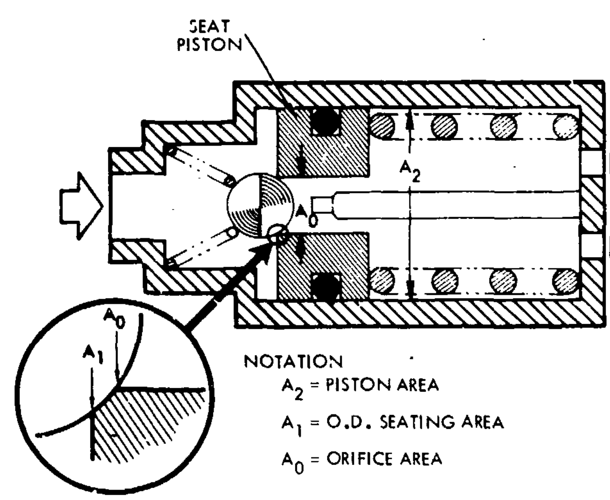


Figure 5.5.8.1a. Inverted Relief Valve

(From "Missile Design and Development," June 1960, vol. 6, no. 6, S. Kowalski, Copyright Missiles and Space, Manhasset, New York)

*Adapted from Reference 51-3

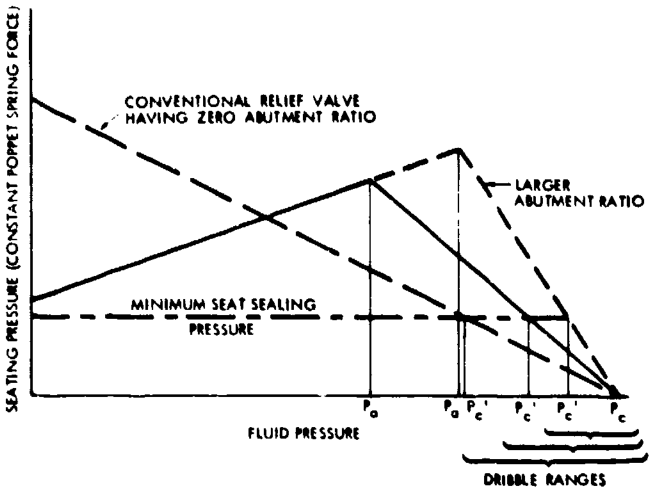


Figure 5.5.8.1b. Operation of Inverted Relief Valve and Effect of Abutment Ratio

(Adapted from "Missile Design and Development," June 1960, vol. 6, no. 6, S. Kowalski, Copyright Missiles and Space, Manhasset, New York)

level required for sealing, the valve starts to leak.

With the inverted relief valve, the actual value of cracking pressure is much closer to the ideal cracking pressure than with the conventional valve. When the ball contacts the stop rod, the ratio of the inlet pressure to the cracking pressure is directly proportional to the ratio of the annular area to the full piston area. Thus

$$\frac{p_a}{p_c} = \frac{A_2 - A_1}{A_2} \qquad \text{(Eq 5.5.8.1)}$$

where p_a = inlet pressure at contact
p_c = cracking pressure
A_2 = piston area
A_1 = seat outside diameter

The ratio $\frac{A_2 - A_1}{A_2}$ is called the *abutment ratio*. For minimum dribble range this ratio should be as large as possible. Typical design values may be as high as 85 to 95 percent. The effect of abutment ratio on dribble range is shown in Figure 5.5.8.1b. P_c' indicates the inlet pressure above which leakage exists, due to the fact that the seating pressure has dropped to the minimum required for sealing.

5.5.8.2 PERFORMANCE CHARACTERISTICS. Since the seating surface, $A_1 - A_0$, is exceedingly small in comparison with the seat-piston annular diameter, $A_2 - A_1$, the additional pressure force tending to open the valve is negligible. Consequently, the valve will not open more than necessary to pass a required flow, and will reseat close to the cracking pressure.

The additional force term in Equation (5.5.7.2b) is small, rendering the inverted relief valve virtually chatter free. Until the ball reaches the stop rod, the seat piston breathes as an accumulator, smoothing out minor pressure fluctuations.

5.5.9 Pilot-Operated Relief

5.5.9.1 DESCRIPTION OF OPERATION. As the capacity requirements for direct-acting relief valves become large, the inherent problems of dribble and hysteresis become intolerable. As a result a small, direct-acting relief valve (hereafter called a pilot) is used to control a larger valve closure. The closure for the main valve is commonly a conical plug, although for hydraulic systems a spool is sometimes used. The closure for the pilot may be any of those discussed in Sub-Topic 5.5.7.

A pilot-operated relief valve is illustrated in Figure 5.5.9.1. The main valving element is ported, allowing fluid from the upstream tank to pass to the spring cavity. The main closure is maintained in the closed position by slight bias from a low rate spring. When upstream pressure increases to relief pressure, the pilot opens and vents fluid from the main cavity. Fluid is expelled faster from the spring cavity than it can be replaced. Relief of pressure in the main valve spring cavity causes a force unbalance, causing upstream pressure to open the valve. Relief then occurs in two stages:

1) relief of the pressure behind the main valve cavity through the pilot
2) relief of the tank pressure through the main valve.

For slight overpressure, relief in a piloted relief valve can sometimes be accomplished by the pilot action alone.

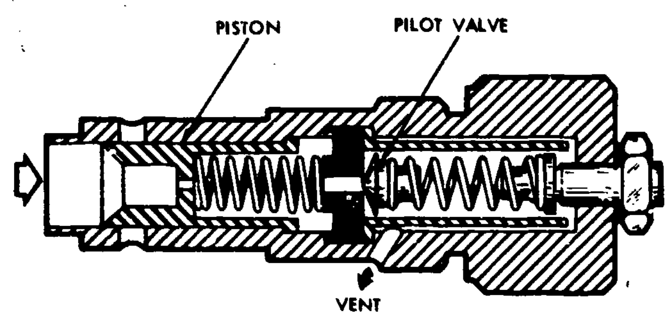


Figure 5.5.9.1. Piloted Relief Valve

(From "Machine Design," March 29, 1962, vol. 34, no. 8, A. M. Bower, Copyright 1962, Penton Publishing Company, Cleveland, Ohio)

5.5.9.2 PERFORMANCE CHARACTERISTICS. With the exception of response time, the pilot-operated relief valve possesses the same characteristics of sealing, opening, and reseat as the pilot if it were used in a single-stage valve. Dribble will occur through the pilot, but inlet pressure may

Table 5.5.10.1. Relief Valve Comparisons

	DESIGN FOR PROPORTIONAL OPENING	DESIGN FOR "POP" ACTION	NARROW DEADBAND	RELATIVE SIMPLICITY	HIGH SEALING FORCE	HIGH CAPACITY	LOW LEAKAGE	VIBRATION INSENSITIVITY	TEMPERATURE INSENSITIVITY	LONG LIFE
Conventional	3	1	3	1	3	2	2	2	1	3(a)
Inverted	2	2(b)	2	2	2	3	2(d)	1	3(c)	2
Piloted	1	2(b)	1	3	1	1	1(d)	1	2	1

(1) Best alternative
(2) and (3) Less desirable alternatives
(a) If chattering occurs
(b) Normally not used
(c) Dynamic seals used
(d) Chance of leakage great because of two leakage paths

actually hold the main valving element tighter against its seat. If pilot opening and reseat are proportional to the upstream overpressure, the main valve will open and close in a similar manner. If the pilot opens to full flow, the main valve will do likewise.

The main disadvantage of a pilot relief valve is its longer response time, attributed to the two-stage relief function. For this reason, pilot-operated relief valves are not adequate for protecting against rapid pressure increases, surges, or shock.

5.5.10 Performance Characteristics

5.5.10.1 TABLE OF COMPARISONS. Table 5.5.10.1 compares the characteristics of conventional, inverted, and piloted relief valves.

5.5.11 Back-Pressure Regulator

The back-pressure regulator is a pressure-reducing regulator used in a relief capacity. As a relief valve, it prevents the overpressurization of the system by discharging upstream fluid to atmosphere or a low pressure reservoir. As a regulator, it maintains upstream pressure at a constant value, limited only by its deadband.

In contrast to a relief valve, the back-pressure regulator is set to maintain upstream pressure at its desired operating pressure, whereas a relief valve is set to open at some pressure above the operating pressure. In contrast to a pressure-reducing regulator, the back-pressure regulator opens with an increase in upstream pressure, where a pressure-reducing regulator closes with an increase in downstream pressure.

The regulator consists of a sensing element, an actuator, and a reference load. Each element is defined and discussed in Sub-Section 5.4, "Pressure Regulators". Spring loading and variable pressure loading are used primarily. The spring-loaded back-pressure regulator functions in a similar manner to its pressure-reducing regulator counterpart, explained in Detailed Topic 5.4.6.2. The one exception is that upstream pressure rather than downstream pressure is sensed. The variable pressure-loaded, back-pressure regulator differs from its counterpart by the fact that it does not require a step-down regulator to reduce upstream pressure for loading. Upstream pressure is bled directly from the main line or tank to the loading dome of the regulator and to the diaphragm of a control pilot. The control pilot is a spring-loaded relief valve with a sensing element (usually a diaphragm) added. The reference spring in the pilot is set to the upstream pressure requirements. When upstream pressure increases above its desired value, the control pilot opens in proportion to the overpressure and vents the loading dome to atmosphere. Upstream pressure against the main valve closure opens the regulator and relieves upstream pressure. As upstream pressure decreases, the pilot tends to close, decreasing flow from the dome cavity. The dome pressure increases and starts to close the main closure against the upstream pressure.

A special type of back-pressure regulator is a combination of pressure switch and vent valve. It is perhaps more aptly called a non-modulating back-pressure regulator.

The combination of pressure switch and shutoff valve, used as a pressure-reducing regulator, is discussed in Detailed Topic 5.4.7.1. The same principles and concepts apply to its use as a back-pressure regulator. The only difference between the two is that as a back-pressure regulator the pressure switch senses the upstream pressure rather than downstream regulated pressure. At a pre-determined upstream pressure (either normal operating pressure or a desired percentage above operating pressure), the pressure switch closes an electrical circuit which actuates a solenoid and opens the vent valve.

This valve produces zero leakage capability, can be lightweight in relation to the vent port size, and remains accurate under extreme environments. However, an electrical power source and a long life cycle performance are required.

5.5.12 Safety Disc

The simplest type of relief device is the safety disc, sometimes called a rupture or frangible disc or burst disc. This is a thin metallic sheet designed to burst when pressure exceeds a pre-set value. A safety disc is a "one shot" device which vents down the tank or system to atmospheric pressure. Safety discs can be classified into shear bursters and flower bursters. In the shear burster, the disc is flat and fails in shear as it deflects against a sharp-edged holder. This type is used with pressures greater than 1000 psig. For lower pressures, the flower burster is commonly used. The disc is contoured and tears from the center to the holder, creating a flower configuration.

Safety discs are commonly used in parallel with a relief valve, and are usually set for a burst pressure higher than the relief valve's full flow pressure. The safety disc acts as the last resort to prevent a hazardous pressure increase, should the relief valve fail to open. Occasionally, shear discs are used in series with the relief valve to prevent corrosion or leakage through the relief valve. In a series arrangement, the burst pressure setting of the disc must be substantially lower than that of the relief valve. An additional consideration is economy, for with an added relief valve the entire load of the tank is not lost.

The desirable qualities of safety discs are accuracy and repeatability of burst pressure. Each quality is affected by the material and the conditions of the disc holder.

Selection of material is based on the following conditions:

1) High ratio of yield point to ultimate strength
2) Homogeneity
3) Corrosion resistance.

The ratio of yield point to ultimate strength determines the ratio of system operating pressure to code working pressure. System operating pressure must be equivalent to the disc strength at the yield point, and the code working pressure must be at least equivalent to the disc material's ultimate strength. The best materials permit an operating-to-working pressure ratio of 0.5 to 0.65. Such materials include aluminum, copper, iron, nickel, silver, gold, and platinum (Reference 160-79).

Material homogeneity is necessary to insure that physical characteristics are constant over the diaphragm in order to obtain reproducible burst pressures. Corrosion weakens the rupture disc, causing burst below the desired pressure. To provent corrosion, metallic discs are either coated with a corrosion-resistant material such as an elastomer, or protection discs of some low strength material are used under the main disc. Such protective discs are usually made of zinc, lead, or tin, and are never used as the main discs.

The disc is usually held between companion flanges. In the shear burster, the inside edges of the flange must be knife-edged and are resharpened after burst of the disc. In the flower burster discs, the edges of the flange in contact with the disc must be smooth and beveled. Any sharp edge or nick will produce tear bursts at pressures much less than those desired.

When installed in a system, the following precautions should be taken to insure safety and proper burst settings:

1) Protection of the disc from mechanical damage or disc reversal, which could arise from windblown debris or probing by personnel.
2) Protection of the disc from reversals or damage due to vacuums being pulled upstream. Vacuum supports are commonly used.
3) Provision against the reaction of the jet by using a Tee as a discharge port.

5.5.13 Relief with Overrides

For many rocket system applications it is desirable to have a valve which will perform the relief function during one phase, but which can be either locked out of the system as a closed shutoff valve, or opened remotely to vent pressure regardless of pressure intensity during another phase. A typical example of a valve that will perform in this manner is the *boiloff valve* shown in Figure 5.5.13a. During tanking, topping, and countdown of a cryogenic rocket system, the valve is in the relief mode and maintains a positive tank pressure. At launch the valve is locked tight.

The basic valve is a piloted relief valve. The pilot valve poppet and loading spring are enclosed by a bellows which is ported to one leg of a three-way solenoid valve. The solenoid valve, on command, directs upstream gas to either the pilot valve bellows or to the atmosphere (vent). The sequence of operation is as follows:

The valve is shown in the shutoff mode in Figure 5.5.13a. Upstream pressure is bled through the solenoid valve to the pilot valve. Pressure is equalized across the pilot valve poppet. For the relief mode, the relief solenoid is energized, closing the pressure line and venting the cavity behind the pilot poppet. A Belleville disc designed to snap through holds the solenoid valve in the relief mode. When tank pres-

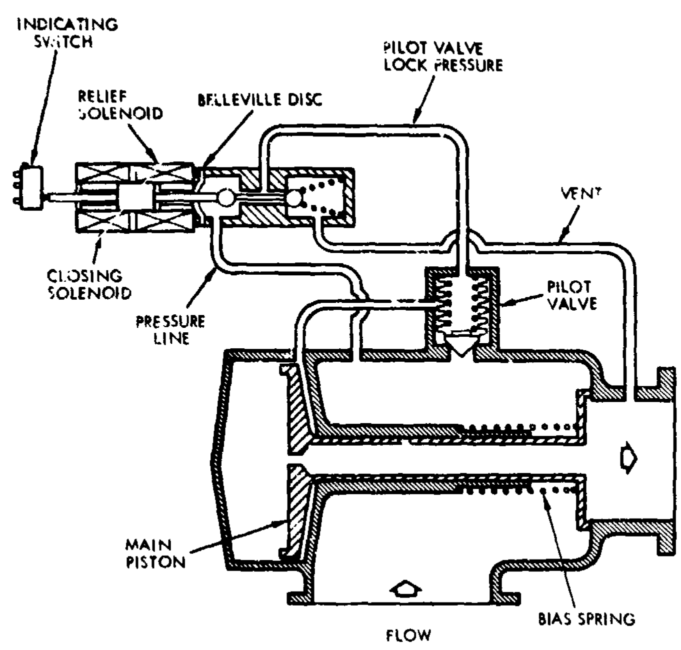


Figure 5.5.13a. Boiloff Valve
(Courtesy of Calmec Manufacturing Corporation, Los Angeles, California)

sure overcomes the pilot valve spring load, the pilot opens and upstream pressure enters under the main piston. The piston area is larger than the main poppet area and the force against the piston is sufficient to overcome the pressure force against the poppet and the load of the bias spring. To return to the shutoff mode, the closing solenoid is actuated, allowing tank pressure to close the pilot relief valve. Pressure under the main piston is relieved through the clearance around the piston and out to the atmosphere through the poppet stem. Internal pressure and the bias spring close the main poppet.

A typical example of a valve that performs a relief function which can be overridden to the vent phase is the *vent and relief valve*. A vent and relief valve is a combination of vent and relief functions in one valve body. The valve's ports and closure are made common to both functions; but each function retains its independent identity through separate and distinct methods of actuation.

A variety of vent and relief valves can be made up from the different types of vent valves and relief valves. Two valve designs are shown here to illustrate the combination of functions. The first valve configuration, shown in Figure 5.5.13b, adds a pilot relief valve to a pneumatically actuated vent valve. In the relief mode, as upstream pressure increases to the relief pressure, the pilot relief valve opens permitting gas to enter under the poppet piston. Since the area under the poppet piston is greater than the poppet area, the force difference opens the main valve. When the pressure decreases and the pilot relief closes, pressure under the poppet piston is metered to the downstream line past the piston and through the main poppet. Upstream pressure plus a small bias spring closes the main poppet. To vent the system at pressure below relief pressure, the solenoid is actuated permitting gas at high pressure to enter under the valve opening piston. This in turn pulls the main poppet open against upstream pressure. Deactivation of the solenoid closes off the high pressure source and vents the chamber under the piston. Upstream pressure against the main poppet closes the valve.

A second valve configuration is shown in Figure 5.5.13c. The main valve is a spring and pressure-loaded poppet. The loading pressure is controlled by a two-stage pilot which references tank pressure against an evacuated bellows. Gas at tank pressure is applied to the bellows through Port A. As pressure increases to cracking pressure, the bellows is compressed and vents the pilot's second stage loading pressure to ambient (Cavity A) through Port B. The pilot's second stage opens and vents Cavity B behind the main poppet. The inlet pressure in Cavity C causes the main valve to stroke. As tank pressure decreases, the bellows expands, the pilot vent port is closed, and gas at tank pressure fills Cavity B, closing the main valve.

For venting operation, pneumatic pressure from an outside source is applied to Cavity D, forcing the main valve open against the pressure in Cavity B and the spring force. To close the valve, actuating pressure is vented by energizing solenoid A. For safety conditions a second solenoid valve

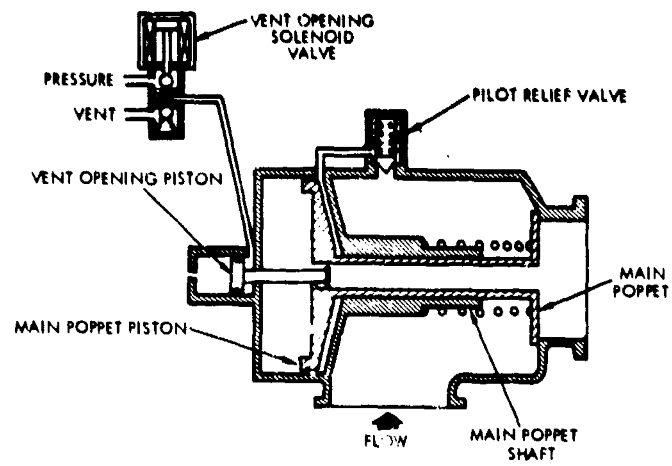


Figure 5.5.13b. Vent and Relief Valve
(Courtesy of Calmec Manufacturing Corporation, Los Angeles, California)

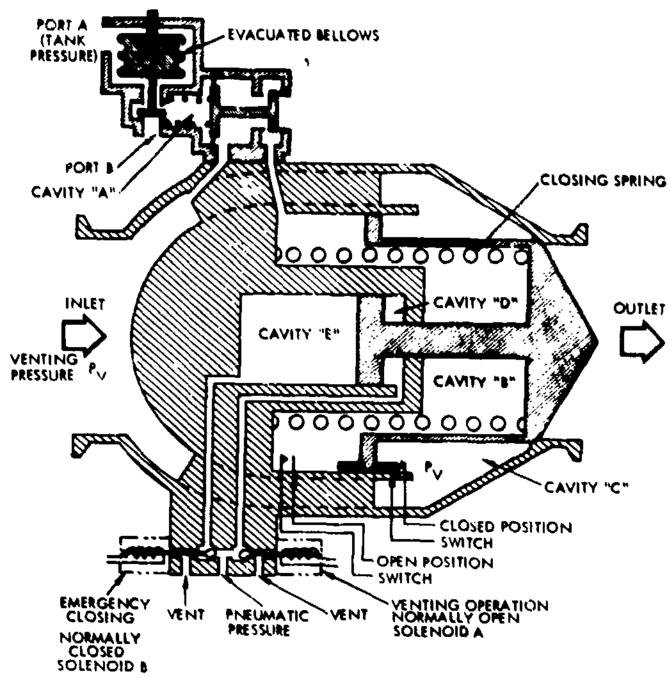


Figure 5.5.13c. Vent and Relief Valve
(Courtesy of Rocketdyne, Division of North American Aviation, Inc., Canoga Park, California)

is used. Actuation of solenoid valve B permits pressure from an outside source to enter Cavity E, opposing the pressure in Cavity D. The valve closes due to the pressure in Cavity E and the spring force.

REFERENCES

1-37	51-3
1-194	160-71
6-129	160-79
10-3	V-127
	V-267

5.6 SERVO VALVES

5.6.1 Introduction

A servo valve is a valve normally used in a servomechanism, having a predictable relationship between hydraulic or pneumatic output (flow, pressure) and an electrical, fluid pressure or mechanical input. Servo valves are error actuated, and provide power amplification and proportional control of the working fluid. A servo valve output may be either flow or pressure controlled. In a flow control servo valve the flow (with constant pressure drop across the servo valve) is proportional to the input signal. Whereas in the presure controlled servo valve, the pressure difference across the load is proportional to the input signal.

The purpose of this section is to discuss servo valves in terms of function, general performance characteristics, and various design types.

5.6.2 General Description of a Servo Valve

A typical two-stage servo valve is illustrated in Figure 5.6.2. An electrical input signal is translated into mechanical motion by means of an electromagnetic device (torque motor). The torque motor produces movement of the flapper, proportional in magnitude and direction to the input signal. The flapper is located between two opposing nozzles so that movement in either direction will restrict fluid flow through one or the other of the nozzles. With no applied signal, the flapper is centrally located between the two nozzles (null position), but a signal which moves it toward either nozzle results in a pressure unbalance across the spool. The torque motor-nozzle-flapper configuration is usually referred to as the first stage, or the pilot stage, of a servo valve. All servo valve designs utilize some technique for positioning the output spool proportional to the input (or first stage output). This is accomplished either by spool position feedback or spring-loading of the spool, which produces a movement of the spool proportional to the pressure unbalance created by the first stage. In the servo valve illustrated in Figure 5.6.2, the spool is centered by helical coil springs on each side. These serve to position it in proportion to the pressure differential across the spool, provided by the first stage flapper position. This, in turn, is determined by the input signal to the torque motor. The flapper nozzle arrangement is essentially a hydraulic bridge, and is used in an analogous manner to an electrical bridge. The direction and magnitude of the spool movement, which is proportional to the magnitude and direction of the input signal, controls the amount and direction of flow to the actuator through the cylinder pressure ports. A signal which would shift the flapper to the right would restrict the flow from the right hand nozzle, thereby increasing the pressure upstream of the right nozzle circuit while decreasing the pressure upstream of the left nozzle circuit. This pressure unbalance would cause the spool to shift to the left until the net pressure force upon it was counterbalanced by the net spring force. The shifting of the spool to the left would provide a flow path from the pressure port out through cylinder port C_1, while the returning fluid through cylinder port C_2 would flow through the return line to the reservoir. In an actual servo valve, the two return ports and the first stage drain port would be manifolded together internal to the valve. As shown in the figure, integral filters are employed to remove small particles of contamination which can build up and plug orifices and nozzles. Although not indicated in Figure 5.6.2, integral filters are commonly used to protect the spool from contaminants.

Servo valves for aircraft and rocket system applications are generally small in size, typically 3" × 2½" × 2", light in weight (½ to 1 pound) and having low power requirements (approximately 0.1 watt).

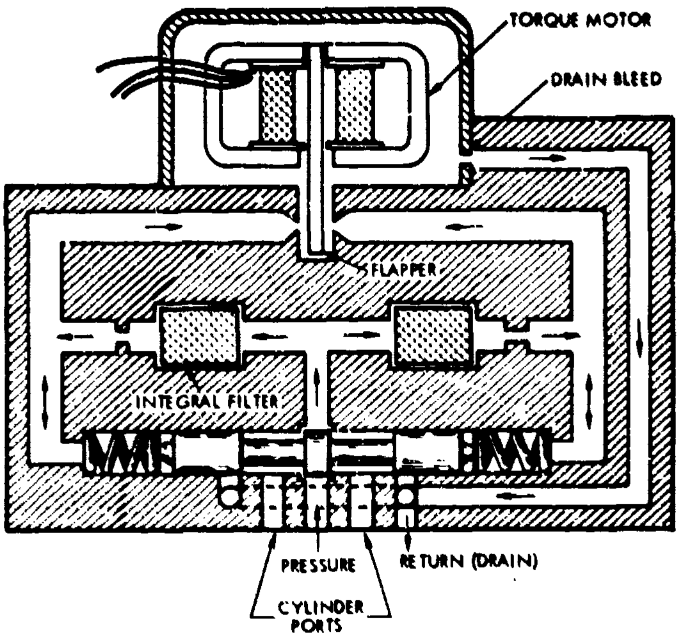


Figure 5.6.2. Typical Two-Stage Servo Valve

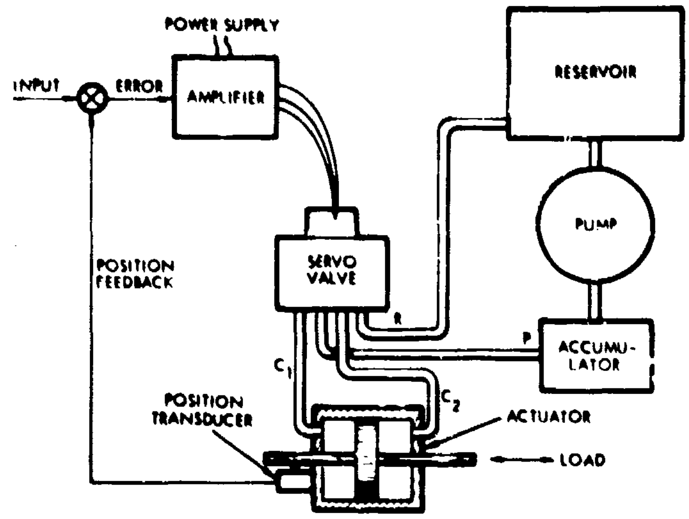


Figure 5.6.3. Typical Hydraulic Servo Positioning Loop

5.6.3 Servo Valve Function

The function of a servo valve can best be described by showing a typical servo valve application in a servo positioning loop. Figure 5.6.3 shows a typical servo loop with the servo valve acting as a flow control device, allowing hydraulic fluid to enter either side of the actuator piston while relieving the opposite side. In operation, an input command is algebraically added to the position feedback to produce what is called the *error signal*. The error signal, therefore, represents the difference between the desired actuator position and the actual actuator position at any given instant in time. This error signal has a polarity with respect to the position of the load. The servo valve delivers fluid flow at a flow rate proportional to the error signal, with a polarity such that the actuator moves the load in a direction towards the commanded position tending to reduce the error to zero. The error signal, which is in volts, is then fed through an amplifier which produces an output of ΔI milliamperes proportional to the error signal. The ΔI milliampere current has a polarity with respect to the servo valve such that the servo valve will cause fluid to flow into the actuator, moving the actuator piston toward the position commanded by the input. The load position is fed back to the error summing point. As the actuator moves in a direction which will reduce the error signal to the amplifier, the net result will be a continually smaller ΔI signal to the servo valve, causing the output of the servo valve to be reduced to zero. At zero input signal, the servo valve returns to its null position, thereby reducing the load flow to zero and causing the load to stop at the position commanded by the input.

5.6.4 Servo Valve Terminology and Performance Characteristics

The following is a generally accepted list of terms, definitions, performance characteristics, and design criteria for servo valves. The material has been adapted from References V-167 and V-273. The given values, which are typical of servo valves presently available, are presented for illustrative purposes only.

5.6.4.1 CENTERING. Centering is the act of adjusting the valve so that the hydraulic null corresponds to the electrical null (zero flow for zero input signal with hysteresis removed).

5.6.4.2 COIL IMPEDANCE. Coil impedance is the vector ratio of the AC voltage across the valve coil to the AC current through the coil. Coil impedance will vary with frequency, but can be approximated by the DC coil resistance and the apparent coil inductance measured at a signal frequency (usually below 100 cps). If the valve coils are connected in a push-pull circuit, mutual inductance will cause the plate-to-plate inductance to be approximately three times the inductance per coil.

5.6.4.3 CURRENT—RATED. Rated current is the specified input current of either polarity required to produce rated flow. It does not include null bias current (Detailed Topic 5.6.4.17). As coils may be operated with differential, series, or parallel coil connections (Figure 5.6.4.3), the rated current must be specified for the particular coil connection utilized. With series coil connection, full valve output will be achieved with one-half the current for differential or parallel connection. Typically, servo valve input currents can range from 5 to 50 ma.

5.6.4.4 CURRENT—QUIESCENT. The quiescent current is a DC current flowing through differentially connected

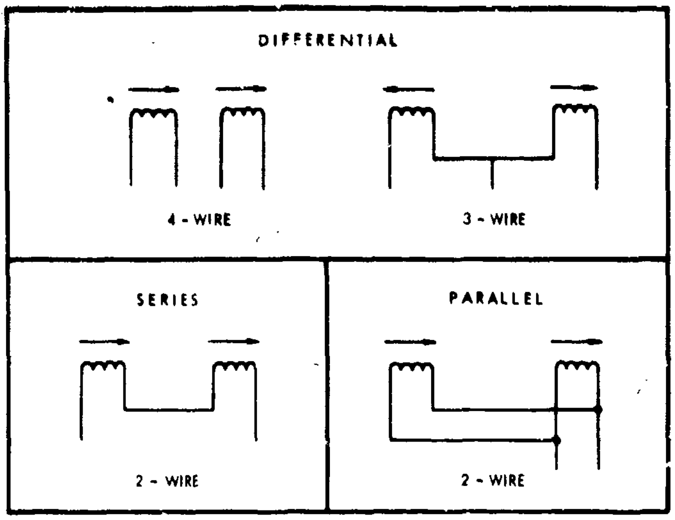


Figure 5.6.4.3. Various Servo Valve Coil Connections

valve coils as a result of a torque motor design requirement or an amplifier requirement. The current, which is usually one-half maximum current, has opposite polarity in each coil such that no electrical control power exists. The control currents vary up and down from this quiescent value.

5.6.4.5 DEADBAND (DEADZONE) OR THRESHOLD AT NULL. Deadband is the value of differential current required to reverse the differential load pressures when the load parts are blocked with pressure gages. It is usually expressed as a maximum current value, depending upon the application, and is a function of the overlap or underlap condition of the main spool. (For a discussion of overlap and underlap see Detailed Topic 5.6.7.2, "Spool Lands.")

5.6.4.6 DITHER. Dither is a low amplitude, high frequency signal superimposed upon the input control signal in order to minimize the effect of stiction or hysteresis, deadband, and threshold. (When used it is normally 60 to 400 cycle current with the lowest amplitude that gives satisfactory results.)

5.6.4.7 ELECTROHYDRAULIC FLOW CONTROL VALVE. This is a servo valve that produces hydraulic flow output proportional to input current.

5.6.4.8 ELECTROPNEUMATIC FLOW CONTROL VALVE. This is a servo valve that produces pneumatic flow output proportional to input current.

5.6.4.9 FLOW. Rated flow and load flow are defined as follows:

Rated Flow (No Load). Rated flow is the control flow developed by taking the full supply pressure drop across the valve metering orifices (no load) with rated current at a specified inlet fluid temperature (normally 100 ± 5°F). (Rated flow at other supply pressures will be proportional to the square root of the pressure drop.)

Load Flow. Load flow is the flow developed under a given load at rated current. The flow developed for any load pressure with full rated current can be calculated by the following:

(Eq 5.6.4.9)

$$Q_l = Q\sqrt{\frac{p_s - p_l}{p_s}}$$

where Q = Rated flow, gpm
Q_l = Load flow, gpm
p_s = Effective supply pressure, psi, (supply pressure minus return pressure)
p_l = Load pressure, psi

5.6.4.10 FOUR-WAY VALVE. A four-way valve is a valve which has a supply, return, and two cylinder ports. When the main spool is activated, it simultaneously opens one cylinder port to supply and the second cylinder port to return. Likewise, reverse action is obtained if the main spool is moved in the opposite direction.

5.6.4.11 FREQUENCY RESPONSE. The sinusoidal frequency response of a valve defines the dynamic characteristics as a function of frequency. It is usually expressed graphically on a plot of amplitude ratio in decibels, and phase angle in degrees versus frequency (Figure 5.6.4.11).

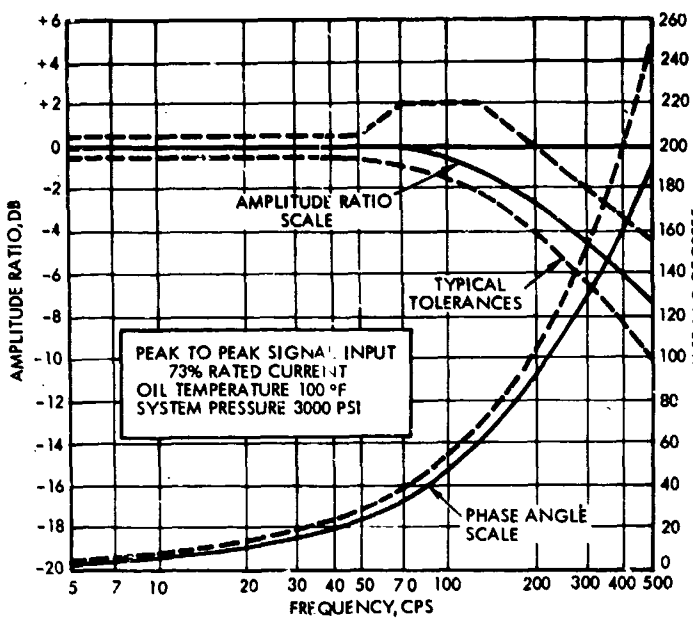


Figure 5.6.4.11. Typical Servo Valve Frequency Response and Tolerance
(Courtesy of Moog Servocontrols, Inc., East Aurora, New York)

Amplitude Ratio. The output amplitude is the average output flow rate at zero-load pressure drop, measured with a sinusoidally varying input current over a range of frequencies with the peak current some fixed percentage of rated current. It is measured with a constant rms input signal.

The amplitude ratio is expressed in decibels from the following formula:

(Eq 5.6.4.11)

$$db = 20 \log_{10} \frac{\text{Output amplitude at frequency f}}{\text{Output amplitude at base frequency}}$$

The base, or reference frequency, is usually 5 cps.

Phase Lag. Phase la is the time differential in degrees (for a given frequency) between the sinusoidal flow output and the sinusoidal current input.

5.6.4.12 GAIN — FLOW. Flow gain is the change in output flow per unit current input, gpm/ma. (Incremental flow gain will vary with the magnitude of the signal input and other variables). Tolerances are usually specified by an envelope on a plot of load flow versus input current. Figure 5.6.4.12 shows normal flow tolerances. Flow gain at null is determined by the relationship of the spool and sleeve metering edges and may vary somewhat from valve to valve. With standard production tolerances, flow gain in the region of ±5 percent rated current input from null may range from −50 percent to +200 percent of the normal flow gain.

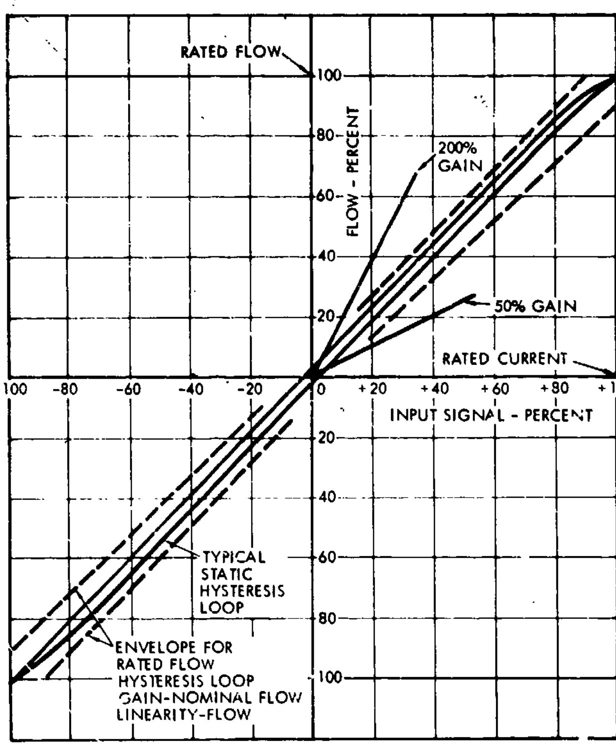


Figure 5.6.4.12. Typical No-Load Flow Curve Illustrating Flow Gain, Hysteresis Loop and Flow Tolerance

(Courtesy of Vickers, Inc., Division of Sperry Rand Corporation, Detroit, Michigan)

5.6.4.13 GAIN — PRESSURE. Pressure gain is the average slope of a plot through null of the cylinder port differential pressure at zero-load flow expressed in psi/ma (Figure 5.6.4.13). Pressure gain is an indication of the response and stability characteristics of a servo valve.

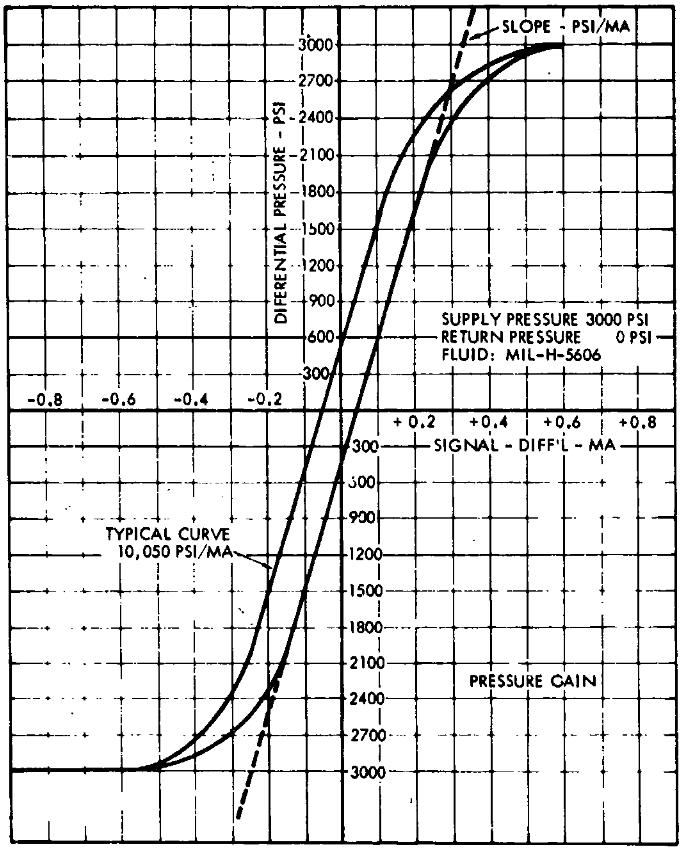


Figure 5.6.4.13. Typical Plot of Pressure Versus Input Current Illustrating Pressure Gain

(Courtesy of Vickers, Inc., Division of Sperry Rand Corporation, Detroit, Michigan)

5.6.4.14 HYSTERESIS. Valve hysteresis is measured as the maximum separation, parallel to the current axis, of the flow curves obtained from one complete cycle of rated input currents (Figure 5.6.4.12). At least one loop is run prior to taking measurement. Normally expressed in ma, it can be expressed as a percentage of rated current or percentage of full differential current. A typical value is 5 percent of rated current.

5.6.4.15 LINEARITY — FLOW. Flow linearity is the consistency of valve flow gain throughout the full range of current input with other operational variables held constant. It is usually measured graphically by fitting into a specified envelope on a plot of valve flow versus current input. It can be expressed in percentage, as the maximum deviation of flow from a straight line established by connecting the two 90 percent flow points. The percent deviation is expressed as a ratio of the maximum deviation of the actual flow curve from this straight line to the sum of the flow values at rated current.

(Eq 5.6.4.15)

$$\text{Percent Deviation} = \frac{\Delta Q}{Q_1 + Q_2}$$

5.6.4.16 LOAD FLOW — PRESSURE CHARACTERISTICS. Load flow — pressure characteristics closely approximate the theoretical square root orifice relationship. Typical load flow versus pressure drop curves from various spool positions are illustrated in Figure 5.6.4.16.

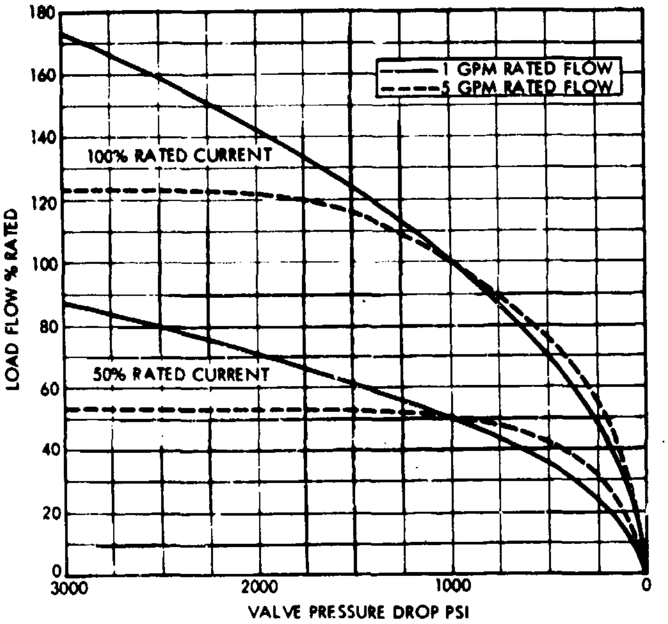


Figure 5.6.4.16. Typical Plot of Load Flow Versus Pressure Drop
(Courtesy of Moog Servocontrols, Inc., East Aurora, New York)

5.6.4.17 NULL BIAS CURRENT. Null bias current is the amplifier compensation bias current to achieve hydraulic null. It is a function of valve hysteresis, temperature, applied acceleration, and supply and return pressure. It is expressed in milliamperes, ma.

5.6.4.18 NULL—ELECTRICAL. Electrical null is the operating point where there is zero signal current.

5.6.4.19 NULL—HYDRAULIC. Hydraulic null is the operating point (ma) where the pressure at the two blocked load ports are equal and the load flow is zero.

5.6.4.20 NULL PRESSURE GAIN. Null pressure gain is the slope at null of a plot of load differential pressure versus input current with zero load flow. It is expressed as psi/ma (Figure 5.6.4.13). Null pressure normally exceeds 30 percent of supply pressure for 1 percent of rated current, and can be as high as 80 percent under the null pressure gain test conditions. Load pressures at null are normally held between one-third and two-thirds of system pressure, but can be specified otherwise.

5.6.4.21 NULL LEAKAGE. Null leakage is the sum of first stage and second stage internal leakages, measured at null position with zero load flow. Normal first stage flows are less than 0.35 in^3/sec. Increased first stage flow permits higher frequency response. Second stage null leakage flow is related to the maximum valve flow rates at system pressure and is normally maintained less than 3 percent of this flow. (See Detailed Topic 5.6.4.24, "Quiescent Flow").

5.6.4.22 NULL SHIFT. Null shift is the change in current required to return the main spool to hydraulic null. It is normally a function of the following:

a) Temperature — measured in ma per 100°F change in temperature.
b) Supply Pressure — measured in ma between 50 percent rated supply pressure and full supply pressure.
c) Return Pressure — measured in ma between 0 and maximum return pressure anticipated.

5.6.4.23 POWER INPUT. Power input is the electrical power dissipated within the valve coils.

5.6.4.24 QUIESCENT FLOW (LEAKAGE FLOW). Quiescent flow is the internal valve flow or leakage from supply-to-return with zero load flow. Quiescent flow (Figure 5.6.4.24) will vary with input current (spool position) and is a maximum at valve null (null leakage).

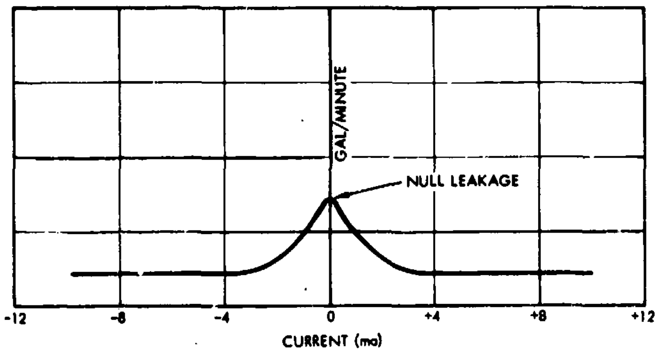


Figure 5.6.4.24. Typical Internal Leakage Curve

5.6.4.25 RESOLUTION. Resolution is defined as the maximum increment of input current required to produce a change in the valve output flow. It is expressed in percent of rated current. Normally less than 2 percent of rated current throughout the operating range without input dither, it can be held as low as 1 percent. If dither is required to improve system resolution, peak-to-peak amplitudes less than 20 percent of rated current are recommended.

5.6.4.26 SATURATION — FLOW. Flow saturation occurs where an incremental change in input current fails to produce a corresponding increase in output flow. It is considered saturated when there is no change in output flow as a result of a change in input current.

5.6.4.27 **STICTION.** A static friction phenomenon which requires a force to break the spool loose and start it moving.

5.6.4.28 **SYMMETRY — FLOWS.** Flow symmetry is the degree to which the flow characteristic with one polarity of signal corresponds to the flow characteristics with the reverse polarity of signal. It should fall within the tolerances given for rated flow (Figure 5.6.2).

5.6.4.29 **THREE-WAY VALVE.** A three-way valve is a valve which has a supply, return, and one-cylinder port. When the main spool is activated in one direction, it opens the cylinder port to supply; when activated in the other direction, it opens the cylinder port to return.

5.6.4.30 **THRESHOLD.** (See Deadband.)

5.6.4.31 **TRANSIENT RESPONSE.** Transient response is the response of the output flow to a step input signal of a given magnitude as a function of time; it is usually given by a curve. (The time required to obtain full no-load flow should be between 0.003 and 0.005 seconds for high response valves).

5.6.4.32 **VALVE PRESSURE DROP.** Valve pressure drop is the sum of the pressure drop from supply to cylinder one, and from cylinder two to return. It is equal to the effective supply pressure (supply minus return pressure) minus the load pressure drop.

5.6.5 Servo Valve Types

5.6.5.1 **SINGLE-STAGE.** In the single-stage servo valve, the main spool is actuated directly by the input or the input transducer. Single-stage servo valves provide high speed of response and low leakage rates but are limited to control of low flow rates, usually less than 5 gpm at pressure from 2000 to 3000 psi. The spool is driven either by two electromagnets (solenoids) or by a torque motor and is resisted by a spring so that the flow area will be proportional to the current input. A typical single-stage flow control servo valve is shown in Figure 5.6.5.1.

5.6.5.2 **TWO-STAGE.** Two-stage servo valves (also called pilot-operated servo valves) provide fluid force and stroke amplification between the input transducer and the output spool. A typical two-stage servo valve is shown in Figure 5.6.2. Two-stage valves are used primarily for higher flow rates; however they are also used at low flows where high response and low leakage are not critical, since the higher spool driving forces provide for added reliability in overcoming the effects of contamination. Proportional control is maintained by having the second-stage spool restrained by a spring or by either mechanical or hydraulic feedback.

5.6.5.3 **HYDRAULIC.** Servo valves may be used to control a number of liquids, including hydraulic oils and liquid propellants. MIL-H-5606 is the most commonly used hydraulic fluid for normal ambient temperature applications. Special high temperature and fire resistant fluids are used

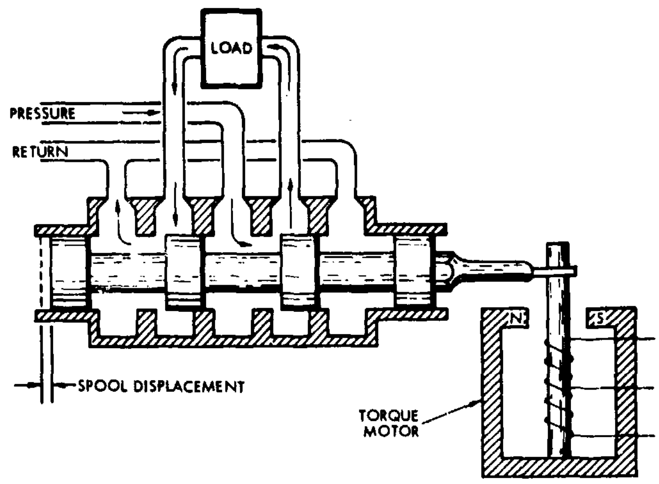


Figure 5.6.5.1. Single-Stage Servo Valve with Torque Motor Drive

for special high temperature requirements. Hydraulic fluids provide a stiff (hence, responsive) media (high bulk modulus) for power transmissions, provide lubrication, and can be readily sealed.

5.6.5.4 **PNEUMATIC.** Pneumatics, particularly high temperature gas systems, are playing an increasingly important role in servo control systems. Hydraulic servo valve-type designs are used with some success in pneumatic systems; however, tolerance requirements, lubrication problems, and the low viscosity of gases present special problems in gas systems. Since gases are less viscous than liquid by a factor of roughly 1,000, leakage is a far more serious problem in pneumatic valves than in hydraulic valves. As a result, clearances are held to the practical minimum and clearance paths are made as long and narrow as possible to keep leakage to a minimum. Because of the peculiar problems with pneumatic systems, special servo valves have been designed for handling gases. Poppets which achieve better leakage control and reduce friction have been used in pneumatic servo valves in place of close-tolerance spools. For good system design, the response of a pneumatic servo valve should be much greater than that required of the closed loop system which is being used. This is because of the relatively low bulk modulus of the gaseous working fluid, even at high working pressures. This factor of compressibility makes the design of a pneumatic servo appreciably more difficult than that of the hydraulic servo.

5.6.5.5 **FLOW CONTROL.** The most common type of output control for servo valve application is flow control or area control. For given pressure conditions, flow control is achieved as a direct function of the flow area variation. The flow area of a servo valve is generally controlled by axial movement of a spool which uncovers porting areas proportional to the spool movement.

5.6.5.6 **PRESSURE CONTROL.** In a pressure control servo valve, the output pressure differential or load pressure is maintained proportional to the input signal level. One advantage of pressure control is that the acceleration of the load can be limited, since the control of acceleration loads can be an important factor in some systems in the prevention of structural damage. Some distinct advantages have been found for pressure control, as opposed to flow control, for pneumatic control systems. Simple direct actuators can be used and response limitations due to load mass resonances and gas spring effects can be avoided by the use of pressure control servo valves. Pressure control servo valves can provide increased dynamic performance up to and beyond the natural frequency of the load when used with proper stabilization networks. Pressure control servo valves are similar in outward appearances to the flow control types, but the internal design of the servo valve allows it to maintain a differential pressure across the load which is proportional to the directional input. A cross-sectional schematic of a typical pressure control servo valve is shown in Figure 5.6.5.6. Pressure control hydraulic servo valves have features that make the valve less susceptible to the influences of temperature and contamination. The first stage of a pressure control servo valve is similar to the flow control valve including the torque motor, flapper, and nozzles. However, the main spool is designed such that actuator cylinder pressures are sensed respectively on either end of the spool, resulting in an axial spool force proportional to the Δp across the actuator. As nulling is achieved when this force balances the output of the pilot stage, Δp is proportional to the electrical input signal.

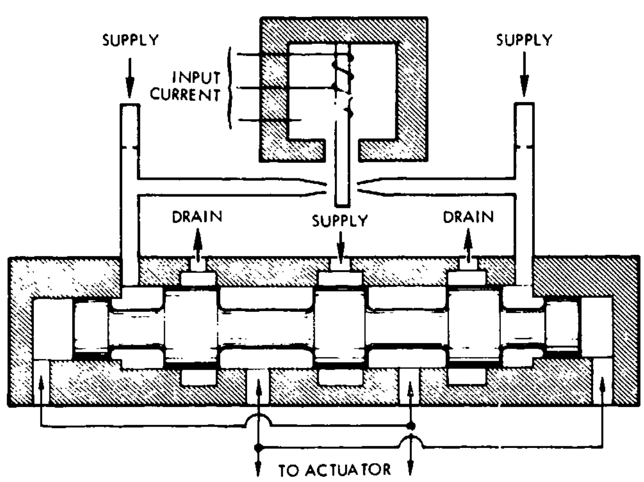


Figure 5.6.5.6. Pressure Control Servo Valve
(Reprinted with permission from "Product Engineering," November 9, 1959, vol. 30, no. 46, W. Ernst, Copyright 1959, McGraw-Hill Publishing Company, Inc., New York, New York)

5.6.6 Servo Valve Inputs

The error signal used to actuate a servo valve usually takes one of three forms: electrical, mechanical, or fluid pressure.

5.6.6.1 ELECTRICAL. An electrical signal drives either a torque motor or a set of two electromagnets. A servo valve with an electrical input is referred to as either an electrohydraulic or an electropneumatic servo valve. The coils of the torque motor or the electromagnet are usually driven in a push-pull manner (Figure 5.6.6.1a). Under null or quiescent conditions a current in both coils is of the same magnitude and direction. It is the relative difference in magnitude between i_1 and i_2 that determines the polarity and signal magnitude. When an error signal is received by the servo amplifier, it produces a Δi proportional to the magnitude of the error signal with the proper polarity. The polarity is determined by which of the two currents has the higher value, where Δi equals i_1 minus i_2. Another wiring scheme for servo valves is the single ended or series system shown in Figure 5.6.6.1b. In this system the current direction changes, thereby changing the polarity. The current flow in this system has twice the effect on the servo valve as does Δi of the push-pull arrangement. The choice of arrangements, whether push-pull or series, depends on the electronics which drive the servo valve rather than on the servo valve itself. Usually a servo valve has the leads from each coil wired so that the user can select the means for driving it.

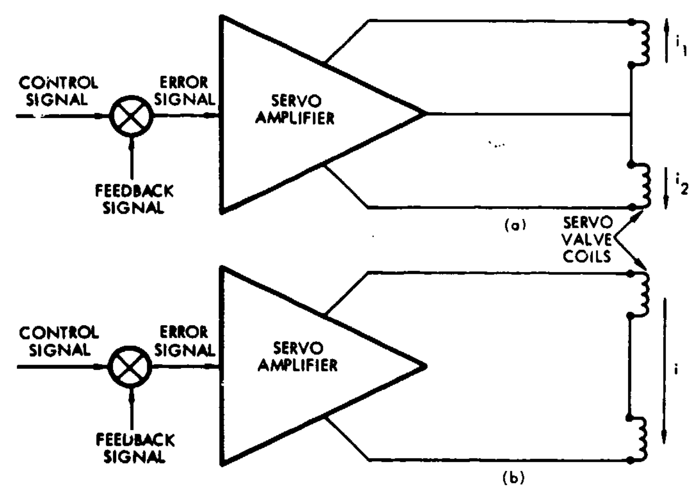


Figure 5.6.6.1a,b. Electrical Inputs for Servo Valve Control: (a) Differentially Connected Servo Valve Coils (b) Series Connected Servo Valve Coils

Torque Motor. Basically, the torque motor is a transducer in that it transforms electrical current into mechanical force. A typical torque motor used in a servo valve is illustrated in Figure 5.6.6.1c. Some type of permanent magnet is employed to set a reference field of flux. The coils that are driven by the input signal will set up fields of flux that oppose each other under quiescent current, but will act with respect to the permanent magnet reference flux when a differential signal is applied to cause the armature to deflect. In order that the amount of armature deflection will be a function of input signal, the armature must move against a spring. This is commonly accomplished by having one end of the armature or flapper rigidly

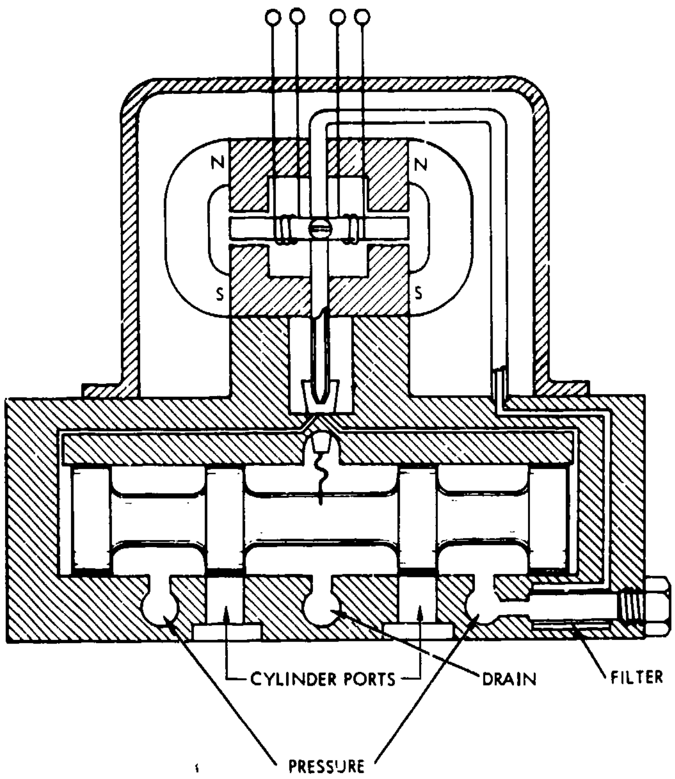


Figure 5.6.6.1c. Typical Jet Pipe Servo Valve with Mechanical Spring Feedback

(Courtesy of Aerospace Division of American Brakeshoe Company, Oxnard, California)

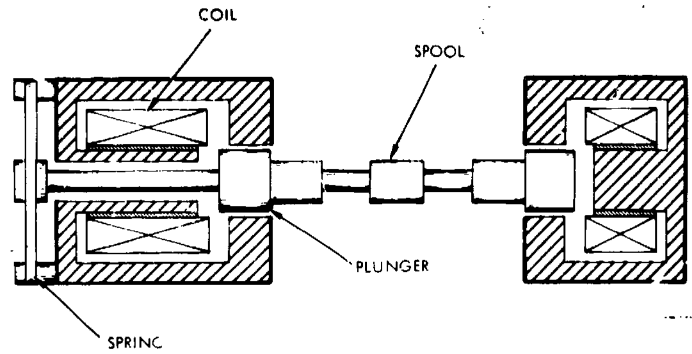


Figure 5.6.6.1d. Plunger-Type Electromagnet

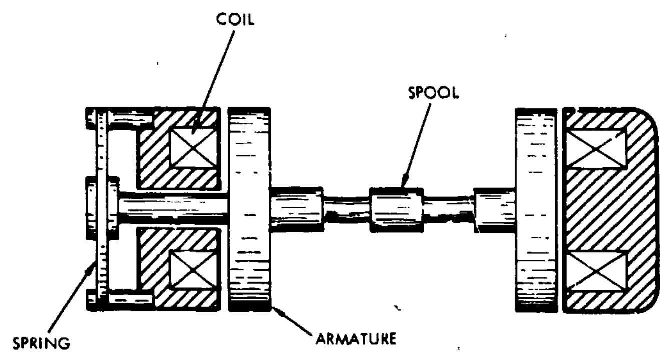


Figure 5.6.6.1e. Armature-Type Electromagnet

mounted to the supporting structure similar to a cantilevered beam. Torque motors can be classified as to their installation — wet, dry, or stale. If the torque motor is sealed from the working fluid, it is known as a dry torque motor; if fluid flows through the torque motor, it is referred to as a wet torque motor. Dry torque motors are favored for contamination resistance. Both elastomer and metallic separating diaphragms are used. If the torque motor is filled with fluid but there is no circulation with the system flow, the design is referred to as a stale torque motor. The stale torque motor has the advantage of cooling the motor without carrying contamination into it, as might be the case in the wet torque motor design. Torque motors are essentially an electromagnetic transducer in which an electrical input is converted into an electromechanical output. Temperature considerations play an important role in torque motor selection, since high temperatures can detrimentally affect the magnetic properties of the torque motor, thereby influencing the operation of the servo valve. (Reference 45-1, Chapter 11 gives a detailed discussion of the design and analysis of servo torque motors.)

Electromagnet. Electromagnets are of two basic types, plunger and armature. In a plunger type (Figure 5.6.6.1d) the output of the electromagnet is a linear movement proportional to the input.

In the illustration, the electromagnet is used in a single-stage servo valve application. The plunger has the advantage of maintaining better linearity over the entire spool travel than the armature-type; however the latter, shown in Figure 5.6.6.1e, has the advantage of a much higher force than the plunger type. Armature-type electromagnets which have been designed for single-stage servo valve applications can generate forces as high as 45 to 55 pounds under normal operating current conditions.

5.6.6.2 FLUID PRESSURE. Fluid pressure, either hydraulic or pneumatic, can be used to actuate a servo valve through a piston or bellows either connected to the pilot stage flapper or connected directly to the valve spool.

5.6.6.3 MECHANICAL. Mechanical input is a type of control used in aircraft hydraulics as a mechanical override for pilot-operated vehicles. Application of a system using a mechanical input is shown in Figure 5.6.6.3. Input to the lever mechanism is provided by an operator or by some device which causes point E of the lever to move about point A, thus causing point S to move a proportional distance. The spool movement allows pressure on one side of the actuator to increase simultaneously, venting pressure on the other side of the actuator. Movement of the actuator causes point A to move about point E as a fulcrum, causing point S to move towards its original position. The spool of the servo valve continues to move towards its null posi-

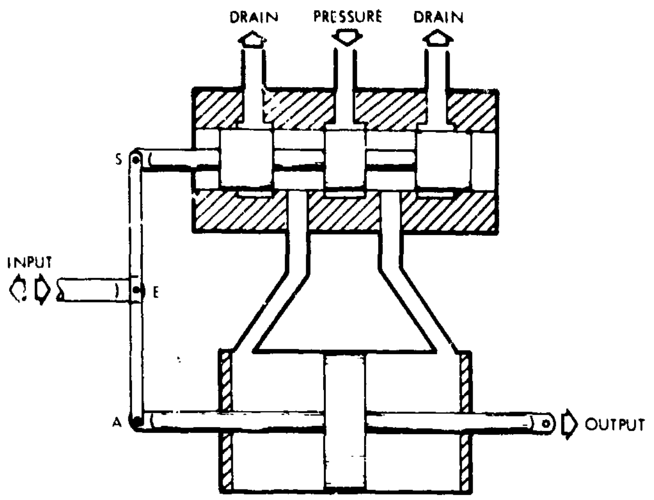


Figure 5.6.6.3. Single-Stage Servo Valve with a Mechanical Input

tion, which will be reached when the load reaches its new commanded position.

5.6.7 Servo Valve Stages

5.6.7.1 PILOT STAGE. A pilot stage, referred to as a hydraulic amplifier or hydraulic relay, is used to amplify the input signal when the force output of the electromagnetic transducer is too small to drive the valve spool directly. Most servo valve applications require the use of a pilot stage to achieve the desired response and flow characteristics. A chart showing comparative characteristics of several types of pilot stages is shown in Figures 5.6.7.1a, b,c,d.

Single Orifice With Flapper. A single orifice with flapper-type pilot stage is shown in Figure 5.6.7.1e. The one-nozzle type has the advantage of ease of manufacture, because only one nozzle must be built instead of two, and there is not the problem of centering the flapper between two nozzles, as is the case with two-nozzle design. Also, should contamination enter the nozzle, the flapper is free to move away to allow passage for the contaminants. However, the flapper, single-orifice type servo valve has the disadvantage of being sensitive to supply pressure level changes, resulting in a corresponding shift in null.

Double Orifice With Flapper. The double-orifice flapper pilot stage (Figure 5.6.2) is the most commonly used flapper-

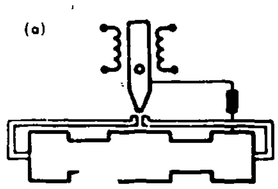
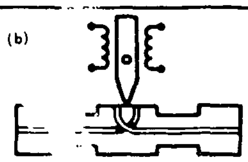
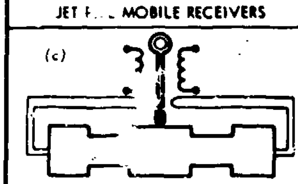
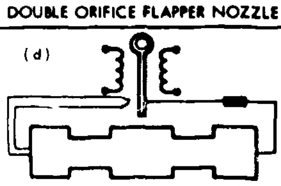
TYPE	MECHANICAL COMPLEXITY			PERFORMANCE		
	TORQUE MOTOR	FEEDBACK	POWER STAGE	CONTAMINATION RESISTANCE	LINEARITY NULL SHIFT	DYNAMIC GAIN
(a)  JET [illegible] FIXED RECEIVERS	LONG STROKE CRITICAL ALIGNMENT	MECHANICAL ATTACHMENT CRITICAL	CONVENTIONAL	GOOD	POOR	AFFECTED BY SYSTEM PRESSURE AND CONTAMINATION
(b)  JET [illegible] MOBILE RECEIVERS	LONG STROKE CRITICAL ALIGNMENT AND NOZZLE FABRICATION	FLUID COUPLING	SPOOL FABRICATION CRITICAL	GOOD	POOR	AFFECTED BY SYSTEM PRESSURE AND CONTAMINATION
(c)  DOUBLE ORIFICE FLAPPER NOZZLE	CONVENTIONAL CRITICAL ALIGNMENT	MECHANICAL CRITICAL ATTACHMENT	CONVENTIONAL	AVERAGE NOZZLE CLEARANCE CRITICAL	GOOD	AFFECTED BY SYSTEM PRESSURE
(d)  SINGLE ORIFICE FLAPPER NOZZLE	CONVENTIONAL CRITICAL ATTACHMENT	MECHANICAL CRITICAL ATTACHMENT	CONVENTIONAL	GOOD	POOR	AFFECTED BY SYSTEM PRESSURE

Figure 5.6.7.1a,b,c,d. Comparison Chart Showing Various Servo Valve Pilot Stage Characteristics

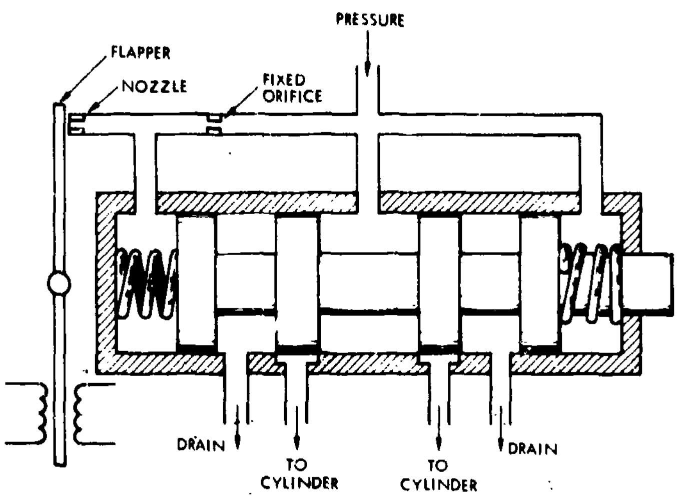


Figure 5.6.7.1e. Typical Single-Flapper Servo Valve

nozzle arrangement. It has the advantage of being relatively insensitive to high pressure variations. However, it has the disadvantage of increased manufacturing costs and flapper centering problems.

Shear Orifices. The shear orifice design is analogous to the double-nozzle type design. However, instead of a moving flapper between two nozzles, it has a slider fork which is controlled by the armature of the torque motor (Figure 5.6.7.1f). The slide fork, which is controlled by the armature of the torque motor, covers two orifices connected respectively to passages drilled from either end of the spool. When a signal is impressed on the torque motor, the slide moves from the null position a distance proportional to the input signal, uncovering one spool port while the other remains closed. Flow of fluid from the end of the spool through the opened port-to-drain results in a pressure unbalance, causing the spool to move toward the vented end (right). The spool moves until a position is reached where the open port is again closed, thus establishing pressure balance across the spool and no further movement of the spool. This design incorporates a unique way of feeding spool position back to the armature.

Jet Pipe. A jet pipe servo valve is illustrated in Figure 5.6.6.1c. The jet pipe (also known as an Askanian nozzle) is attached directly to the armature of the torque motor. An electrical input to the torque motor results in a proportional movement of the jet pipe over two receiver ports. In the null position, the flow of the jet pipe is equally divided between the two ports, resulting in equal pressure on either side of the spool. Movement of the jet from the null position results in a pressure unbalance, shifting the spool in a direction tending to restore balance. The jet pipe requires a larger displacement than the flapper nozzle, but this does not impose any design limitations. In some designs, the jet pipe operates in a manner similar to the flapper valve as far as the power stage spool is concerned. The jet nozzle converts static pressure into fluid velocity, which in turn is reconverted into static pressure at each end of the spool. Some pressure head is lost in the conversion from dynamic to static head, depending upon pressures, flow capacities, and design type. The pressure losses are generally of the order of 5 to 10 percent of the supply pressure.

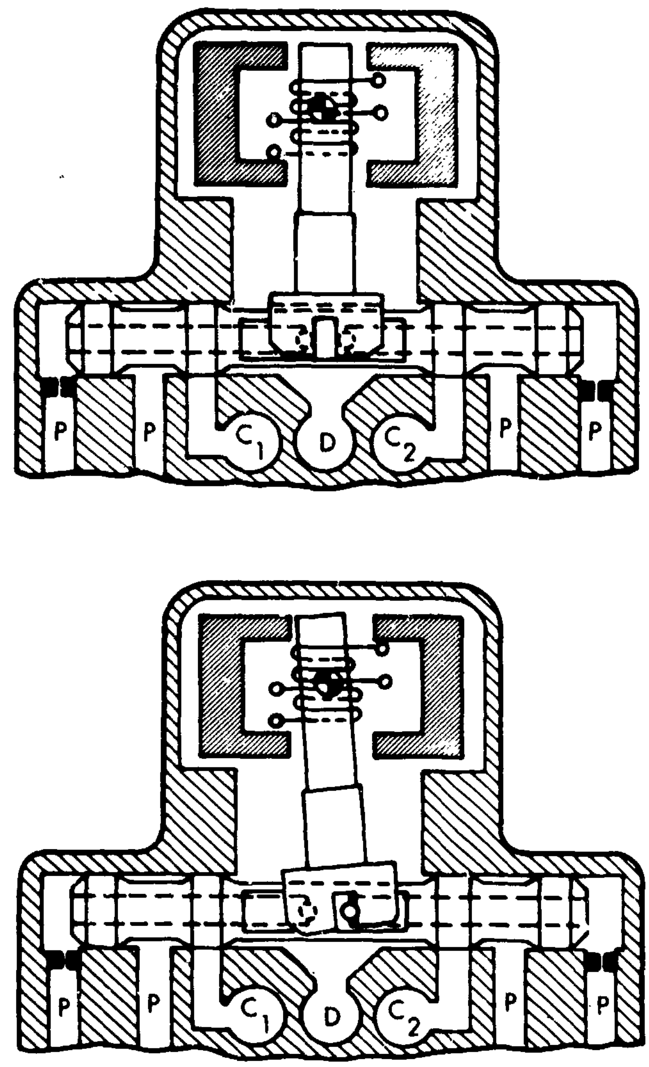


Figure 5.6.7.1f. Shear Orifice Servo Valve Design Shown in the ON Position Immediately After Receiving An Input Signal

(Courtesy of Kearfott Division of General Precision, Inc., Little Falls, New Jersey)

A jet pipe design is relatively less susceptible to clogging than the nozzle design. Spool position feedback in a jet pipe servo valve can be accomplished by either a conventional mechanical spring (Figure 5.6.6.1c) or by a nulling technique utilizing mobile receivers (Detailed Topic 5.6.8.4).

5.6.7.2 POWER STAGE. Although rotary, slide, poppet, and other types of flow control elements are used as the power or second stage of servo valves, by far the most commonly used element is the spool and sleeve or piston valve. A major advantage of the spool valve is the ease of accomplishing pressure balance, extremely important for precise control over a range of flow. Sealing of the spool valve is usuallya ccomplished by closeness of fit between the spool and sleeve.

Four-Way Spool. The four-way spool is the most commonly used type of spool for the power stage of the servo valve. It is used in systems where the effective areas on each side of the actuator piston are equal. The four-way valve is so named because of the required number of fluid lines to the valve. These four lines consist of the supply line, two control lines to the actuator, and at least one return line. The spool itself may have two, three, or four lands, but for any configuration the spool has four metering edges which are lined up with the ports of the valve (sleeve) (Figure 5.6.7.2a,b,c). Although not illustrated, the control orifices are not made an integral part of the valve, but are machined in a sleeve which is inserted in the bore of the valve. It is common practice for the sleeve to be built up from separate short sections, the ports being milled into the end faces of the appropriate sections. Sealing between adjacent ports is accomplished by static O-ring seals between the sleeve and the valve body. As the spool is moved in relation to the fixed sleeve, the orifice areas which are formed by the edges of the spool lands and sleeve ports will vary. These variable area orifices work in pairs, so that one opens to supply more fluid to one side of the piston while the corresponding variable area orifices closes the passageway to the return line. The other two variable area orifices controlling the fluids of the other side of the actuator piston perform the opposite function by closing off the supply pressure and opening the return ports, thus allowing the actuator piston to move in one direction. The combined operation of these four variable orifices divides the flow of fluid to and from each side of the actuator piston in proportion to the movement of the valve spool. The operation of a four-way spool is analogous to the operation of a variable resistor Wheatstone bridge (Figure 5.6.7.2d). Typical valve pressure curves showing pressure versus stroke for each cylinder port are illustrated in Figure 5.6.7.2e. When the spool is in null position, the supply pressure is dropped equally through both sides of the bridge (approximately one-half the supply pressure for hydraulic systems) resulting in a zero differential pressure across the actuator piston. As the spool is moved from its null position, the differential pressure across the actuator piston will increase from zero.

Three-Way Spool. A three-way spool differs from the more common four-way spool in that there is only the one cylinder port, since the three-way valve provides force in only one direction. Some means is required for returning the actuator piston, such as the spring or an unequal area piston supplied with the pressure. With an unequal area piston, a constant supply pressure is maintained on the small area side of the piston, while the servo valve controls the pressure on the large area side of the piston.

Spool Lands. The performance characteristics of a servo valve are largely dependent on the design of control for

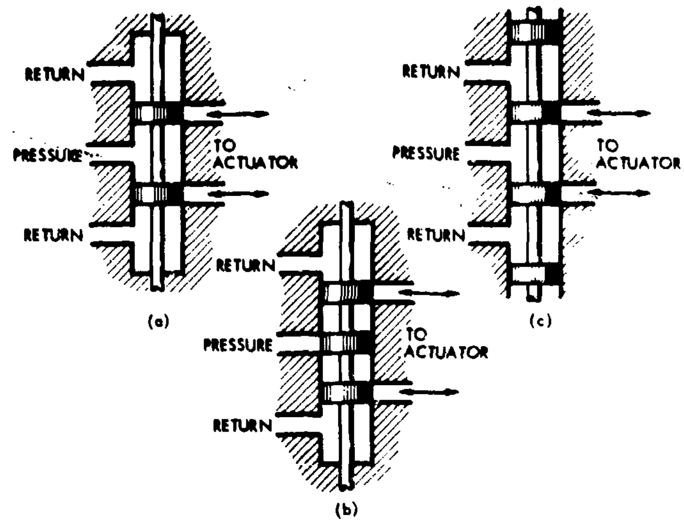


Figure 5.6.7.2a,b,c. Four-Way Spools Having (a) Two, (b) Three, and (c) Four Lands

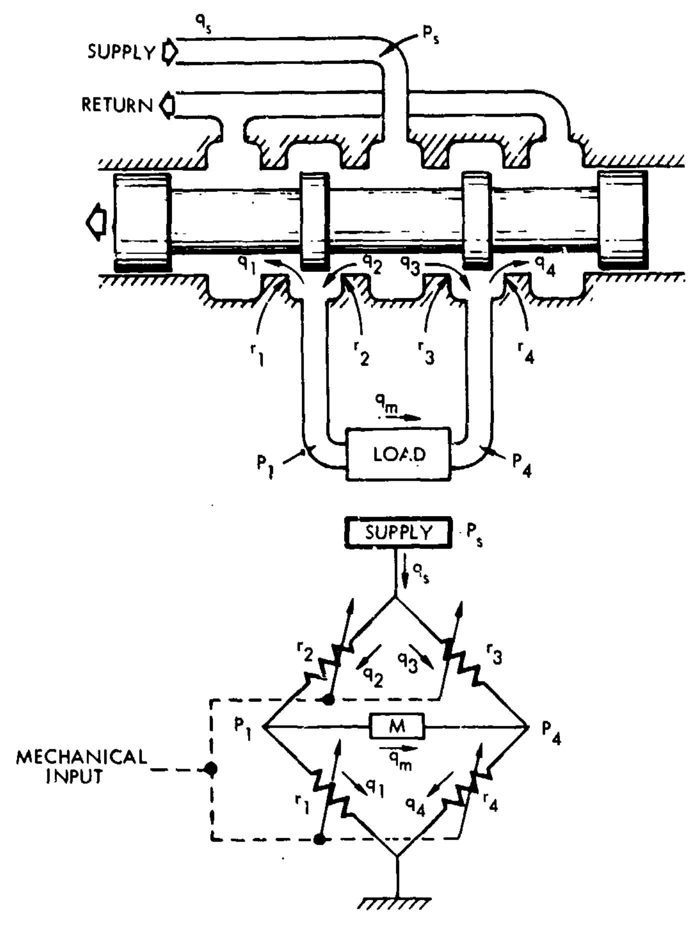


Figure 5.6.7.2d. A Four-Way Spool Valve and Its Equivalent Circuit

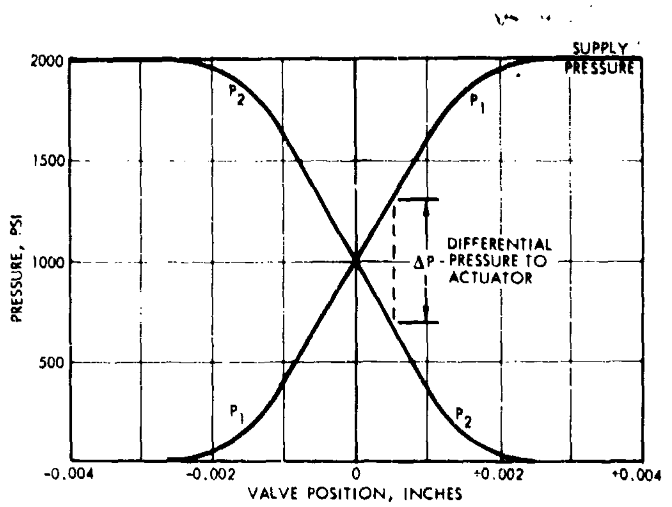


Figure 5.6.7.2e. Typical Pressure Versus Stroke Curves for a Four-Way Servo Valve

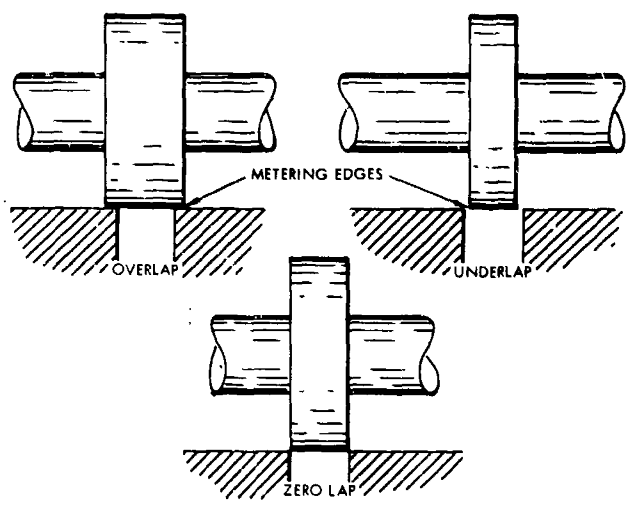


Figure 5.6.7.2f. Illustrations of Spool Lands Showing Overlap, Underlap, and Zero Lap

sealing lands on the spool. The degree of land overlap or underlap (Figure 5.6.7.2f) determines the tradeoff between valve leakage and deadzone. Figure 5.6.7.2g shows the effect of degree of land lap on controlled flow and leakage flow. Although the maximum sealing is achieved by overlapped lands, the valve has a deadzone or deadband equal to the amount of overlap. This deadzone causes a non-linearity resulting in loss of sensitivity near the null position, which could lead to control system instability. As a valve in which the metering edges of the land exactly meet the edges of the ports (zero land) is virtually impossible to manufacture, most valves are designed with a small amount of underlap. This results in an inherent leakage flow back to the return line at the null position. Due to the under-

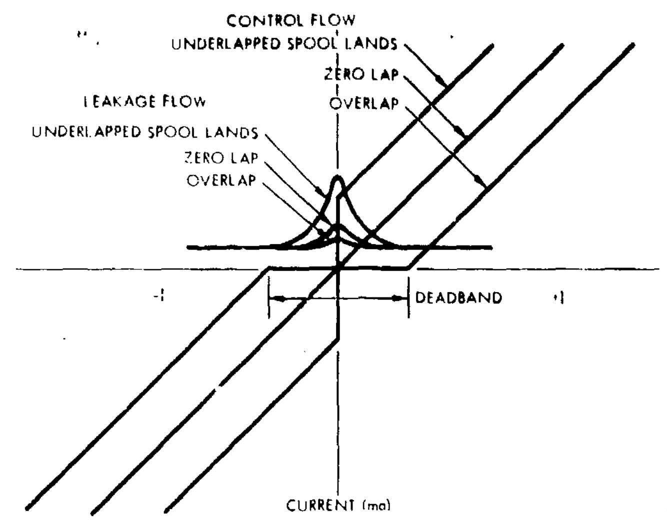


Figure 5.6.7.2g. Leakage and Control Flow Curves Showing the Effect of Spool Land Lap

lapping, the maximum leakage usually occurs at the null position, illustrated in Figure 5.6.5.1. Underlapped valves are commonly referred to as open-center valves, while overlapped valves are called closed-center valves. Overlaps of up to 0.001 inch are provided when it is necessary that leakage be minimized with the valves in null position.

Flow Forces. The flow of fluid through the variable orifices of a servo valve cause hydraulic reaction forces, referred to as Bernoulli forces (pressure variations as a result of non-symmetric fluid velocities). These forces have axial and radial components. The radial force which results primarily from lack of parallelism in the spool lands and sleeves tends to push the spool sideways against the sleeve or wall, causing sticking of the spool. This condition is called "hydraulic lock" and is minimized by locating the valve ports symmetrically around the spool and making the spool and sleeve as truly cylindrical as possible. An axial flow force, resulting from pressure differences across the spool metering edges, usually acts in a direction tending to close the valve. In general, axial forces only become important in single-stage valves where actuation forces are low. The axial force can be reduced by machining cusps between the spool lands. The cusps cause the incoming fluid to change direction, resulting in an axial force that tends to oppose the Bernoulli force. Equations for evaluating these forces are presented in Reference 98-1, pages 397-402. A discussion of spool design techniques for minimizing flow force problems is presented in References 6-36, 6-159, and 26-29.

5.6.8 Load Spring and Spool Position Feedback

Some means is needed to position the servo valve spool proportional to the required electrical input, and to return it to a null position when the error signal reduces to zero because the proper positioning of the actuator has been achieved.

5.6.8.1 SPRING-CENTERED SPOOL. Spring centering of the power stage spool to achieve proportional positioning is illustrated in Figure 5.6.2 and described under Sub-Topic 5.6.3. Coil springs are most commonly used, but some designs have employed leaf type springs. As the travel of servo valve spool is usually less than ±0.015 inch from null position, springs do not show any undue non-linear characteristics. The spring, therefore, provides positioning of the spool proportional to the differential pressure on the spool, which in turn is proportional to the electrical input to the servo valve.

5.6.8.2 MECHANICAL FEEDBACK. A direct mechanical connection between the power spool and the torque motor is a common means of achieving spool position feedback. Either a leaf or coil spring can be utilized by attaching one end of the spring to the spool and the other end to the pilot stage jet pipe or flapper. In the jet pipe design, (Figure 5.6.6.1e) the porting is such that the spool moves in a direction opposite to the jet pipe movement. The spool movement results in a spring feedback force, opposing the jet pipe deflection caused by the torque motor. This feedback force increases until the jet pipe is returned nearly to the neutral position. The jet pipe is offset only enough to hold sufficient differential pressure across the spool to counterbalance the spring force. This offset is very small, since a differential pressure of less than 100 psi will normally hold the spool in the extreme offset condition and the pressure gain of a jet pipe can be approximately 1,000,000 psi per inch of jet pipe movement.

5.6.8.3 PRESSURE FEEDBACK. Fluid pressure can be utilized directly to provide spool position feedback. Proportional position through fluid pressure feedback is utilized in the pressure control servo valve shown in Figure 5.6.5.6.

5.6.8.4 NULLING DEVICES. Another technique of spool positioning is the use of a nulling device wherein the spool seeks a position of pressure balance utilizing ports drilled in the spool itself. Feedback is achieved by positioning of the moving ports with respect to the pilot stage output. This principle is used in the shear orifice design (Figure 5.6.7.1c) and the jet pipe with mobile receivers (Figure 5.6.7.1a). In the jet pipe design, the spool tends to recenter itself under any run position of the jet pipe by virtue of an unbalance axial spool force caused by the pressure distribution across the two receiver holes. It is a nulling device, because the spool tends to null itself on the center of the jet pipe.

5.6.9 Contamination

A servo valve is a precision device. Diametral spool clearances of approximately 0.00005 inch are not uncommon, and surface finishes of 4 to 6 microinches rms are standard requirements for spool and sleeves. Contamination tolerance, or dirt sensitivity, is an important factor in the design and use of servo valves. Servo valve performance and reliability can be detrimentally affected by unavoidable dirt particles. Force balances, flow rate gain, and general electromechanical operation can be influenced by the presence of contaminants within the servo valves. Specific contamination problems consist primarily of the plugging of control orifices and nozzles and the sticking of the spool. Table 5.6.9 show design tradeoffs in servo valves which are influenced by contamination. Extreme care must be taken in servo valve assembly to avoid entrance of contaminants. The use of integral filters to protect orifices and nozzles is common practice. A stuck spool in a servo valve usually requires overhauling and cleaning, or replacement of the valve. However, a stuck spool may sometimes be dislodged by a high amplitude sixty cycle input into the valve. Additional information on the subject of servo valve contamination is given in Detailed Topic 10.6.2.7.

Table 5.6.9. Trade-offs in Servo Valve Designs
(Reference 6-56)

BALANCE THIS:	AGAINST THIS:
Flow rate should be:	
SMALL, to reduce the number of particles passing through the circuit	LARGE, to obtain the best valve time constant
Orifice size should be:	
SMALL, to reduce leakage and consequent power loss	LARGE, to allow passage of small particles which are difficult to filter
Filter pore opening should be:	
SMALL, to catch contaminants and avoid clogging of orifices	LARGE, to avoid clogging the filter
Filter surface area should be:	
SMALL, to permit filter to be small and conserve weight	LARGE, to prevent clogging

5.6.10 Acceleration Switching Servo Valve

A relatively new approach to servo valve control is the use of a time modulated ON-OFF valve action instead of the more common proportional flow, or pressure, servo valve. The basic differences between the switching valve and the standard flow control valve are the absence of spool-centering springs and the fact that the electrical signal to the switching valve has the form of a time modulated squarewave. In response to this switching signal, the valve is driven alternately right and left, opening and closing the two cylinder ports. The rate of change of load flow, hence load acceleration, is proportional to the pilot dwell time. The natural frequency of the load must be lower than the frequency of the square wave input to prevent undesirable jitter. Electrical hysteresis and changes in pilot stage gain, which can vary significantly with temperature, have very little influence on the acceleration switching valve. Primary applications for the valve, therefore, are in high tempera-ture servo systems. For additional information on accel-eration switching servo valves, see References 22-5, 26-205, 27-2, and 207-7.

REFERENCES

6-36	26-205	374-1
6-42	27-2	V-54
6-56	42-1	V-167
6-113	45-1	V-203
6-159	78-1	V-273
19-56	98-1	V-289
22-5	207-7	
26-29	280-2	

5.7 EXPLOSIVE VALVES

5.7.1 Introduction

Explosive valves, sometimes called squib valves, are shutoff valves which use an extremely compact energy source, and are actuated only once in a given mission. In most explosive valves the charge is simply a source of high pressure used to position an actuator. As a great deal of explosive energy is available from a small charge, explosive valves can use brute force actuation to achieve design simplicity. The small size and weight of the exposive energy source permits explosive valves to be lighter than pneumatic or electrically operated valves, with equivalent function. Although it is possible to achieve repeated actuation using multiple squibs, most explosive valves are designed for single operation only, requiring either replacement or refurbishment after use. As the explosive cartridge is expendable, the rest of the valve is usually designed to be expendable also. This provides a minimum weight design, and also permits extremely good sealing before actuation, because parent metal can be ruptured to open the valve.

Since explosive valves cannot be tested non-destructively, their reliability for a given mission must be demonstrated by a statistically designed qualification program. This type of test program requires purchasing for qualification a larger number of explosive valves than conventional valves. References 547-8 and 560-1 treat explosive valve testing.

The purpose of this Sub-Section is to describe explosive valves in terms of explosive actuator characteristics, valve design types, and general characteristics as compared with conventional valves.

5.7.2 Explosive Actuation

5.7.2.1 GENERAL. Explosive valves are actuated by the sudden conversion of potential chemical energy into me-chanical energy. The chemical energy can be released either by deflagration (fast burning), or detonation (the propaga-tion of a shock wave through a high explosive).

In the case of deflagration, work is done by the high pressure of the hot explosive gases acting on a piston. A typical chamber pressure versus time record is shown in Figure 5.7.2.1. Pressure is contained in the cylinder by one or two

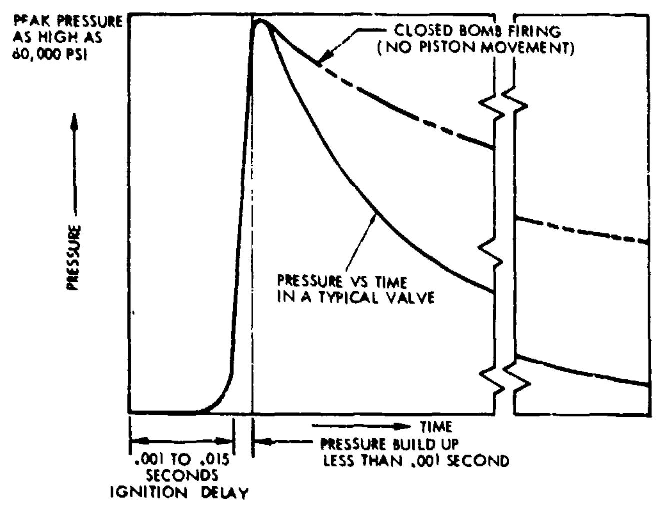


Figure 5.7.2.1. Typical Pressure Versus Time Record for an Explosive Actuator

piston O-rings and backup rings. Some combustion gas generally leaks past the rings, since the seal reaction time is not sufficient to resist the initial pressure buildup. The gas leakage is usually insignificant to the performance of the valve. It is also possible to use a deflagrating charge to drive an interference fit, but extremely high pressures must be obtained. In the case of detonation, the momentum of the shock wave in the explosive is transferred to a piston, equivalent to hitting the end of the piston a sharp blow. The use of a detonating charge permits the piston to be an interference fit in the cylinder. If extremely tight contamination control is required, shock waves from a detonating charge can pass through a permanent metal barrier to actuate the piston.

In either case a small amount of explosive is used. For example, a typical 0.75 inch diameter, 4000 psi working pressure, normally-closed valve uses 140 milligrams of propellant or about 1/20th ounce. The explosive pressure in this valve is approximately 25,000 psi.

A deflagrating charge used in an explosive valve generally consists of small flake-like particles of powder similar to shotgun powder. A typical design would be discs approximately 0.010 inch thick and 0.050 inch in diameter. This shape is used to present a large surface area to the advancing flame front, permitting rapid pressure buildup. A detonating charge consists of a high explosive such as PETN, DDNP, or RDX, either compressed at high pressure to a solid cylindrical pellet or loose loaded.

5.7.2.2 EXPLOSIVE INITIATION. Explosive charges are generally initiated by the transfer of either electrical or mechanical energy to a sensitive primer such as lead styphnate. Ignition may be started by striking the primer a sharp blow through a metal barrier, like the firing pin of a gun, by abruptly piercing the primer container with a sharp probe, or by electrically heating the primer through a bridgewire.

A deflagrating charge is often initiated directly by a primer, while a detonating charge often utilizes a booster to transform the primer output into a detonation wave. A typical booster would be a lead azide pellet pressed against the output charge.

Both detonating and deflagrating charges can be directly initiated without the primer by using an exploding bridgewire. This method requires a firing pulse of several thousand volts, as compared with a conventional firing pulse of about 28 volts (or as low as 5 volts). The extremely high potential first vaporizes the bridgewire, then discharges through the ionized particles remaining in the path, creating an explosion which in turn ignites the main charge. The exploding bridgewire is extremely insensitive to accidental firing, but requires a capacitor discharge firing circuit which is more complex than conventional electrical ignition systems.

Exploding bridgewire (EBW) ignition differs from low-voltage ignition in that the EBW input-energy requirement is primarily a function of voltage, whereas that of the low voltage igniter is primarily a function of current.

Reference 560-1 discusses EBW ignition in detail and suggests the following as typical input-energy relationships:

1) Typical low voltage ignition:

0.2 amp	No samples will fire
0.3 amp	Threshold (minimum current at which explosive system will fire)
0.5 amp	All samples will fire

2) Typical exploding bridgewire (EBW) ignition:

500 volts	No sample will fire
1300 volts	Threshold
2600 volts	All samples will fire

5.7.2.3 TYPICAL EXPLOSIVE CARTRIDGE. A typical explosive cartridge is shown in Figure 5.7.2.3. The charge is electrically initiated, an electrical connector being integral with the cartridge. The unit is hermetically sealed to protect the explosive charge from moisture and damage.

The cartridge shown contains a nominal 188 milligrams of deflagrating charge, sufficient to produce pressures up to 60,000 psi, depending inversely on the dead volume in the firing chamber.

Cartridges such as the one shown in Figure 5.7.2.3 can have a firing current from 1.0 to 5.0 amps, and a firing time from 0.001 to 0.020 second. Cartridges with 5.0 amps firing current can be made to survive 1.0 watts bridgewire power dissipation without igniting. Many cartridges of this type have 2 bridgewires for reliability. Typical units are capable of withstanding 300°F soak for 24 hours without deterioration.

The cartridge shown in Figure 5.7.2.3 weighs 0.03 pound, most of the weight being in the stainless steel case. The electrical conductors, two stainless steel connector pins, are locked into the case by fused ceramic seals. Fused glass seals can be used for low pressure cartridges, while fused ceramic seals are capable of withstanding pressures of 60,000 psi. The cartridge in Figure 5.7.2.3 incorporates a plastic shock plug to attenuate the shock waves generated by the explosive before they reach the ceramic seals. The explosive charge is contained in a thin metal shell which is induction solder sealed to the cartridge body. The thin

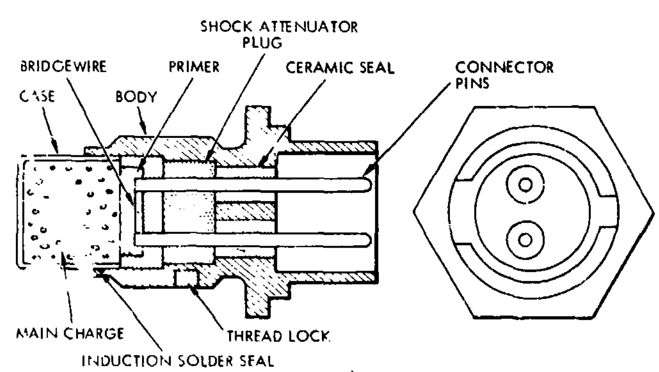


Figure 5.7.2.3. Typical Explosive Cartridge
(Courtesy of Holex, Inc., Hollister, California)

shell bursts when the explosive fires. The cartridge incorporates two nylon pellets to lock the threads against vibration, and is designed to accept a standard O-ring for sealing.

The cartridge in Figure 5.7.2.3 is a specific example of a type of cartridge common to explosive valves. Many variations are possible. Some cartridges have wires leading from the bridgewire to the outside power source instead of using an integral electrical connector; others use a single ceramic or glass seal for the connector pins instead of individual seals for each pin. Some use four pins, two for each bridgewire, while others have but a single bridgewire. Some have a ground pin in addition to the power pins, while some explosive valves use no cartridge at all, instead incorporating the charge directly into the body of the valve.

5.7.3 Normally-Open Valves

Two normally-open explosive valves are shown in Figures 5.7.3a and 5.7.3b. Both valves use a wedging action to seal the flow passage.

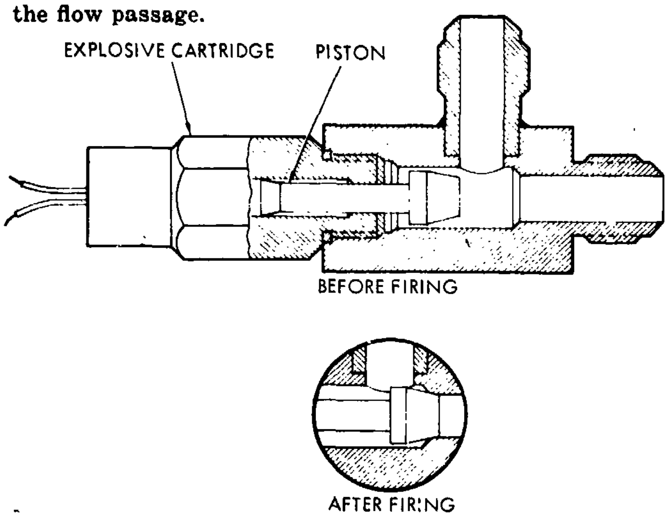


Figure 5.7.3a. Normally-Open Explosive Valve
(Courtesy of Conax Corporation, Buffalo, New York)

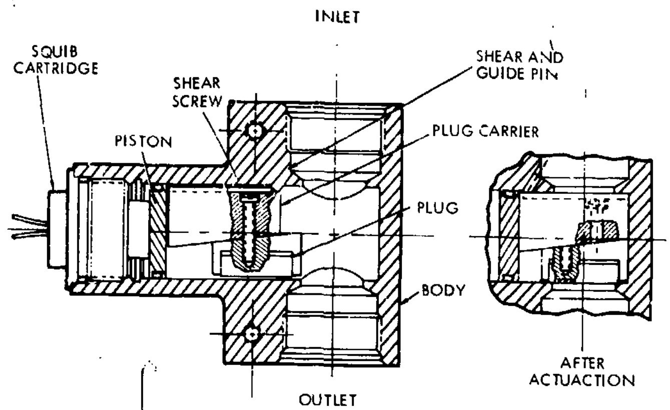


Figure 5.7.3b. Normally-Open Explosive Valve
(Courtesy of Hydro-Space Technology, Inc., West Caldwell, New Jersey)

The valve shown in Figure 5.7.3a drives a conical plunger into the outlet port to seal the valve. This valve has the inlet ports at right angles, although the two ports could be parallel by having a different internal arrangement. This particular valve uses an interference fit between the piston and the valve body to seal the explosive combustion products from the valve passages. One recent design approach utilizes a rolling diaphragm to effect this seal. Care is required in either case to insure against liquid entrapment between the piston and valve body which could alter valve response by creating a dashpot effect. In the valve shown in Figure 5.7.3a the explosive charge is integral with the valve instead of being contained in a separable cartridge.

The normally-open valve in Figure 5.7.3b seals by explosively driving a split wedge into the flow passage. A resilient plug is wedged between the plug carrier and the outlet port after the valve is actuated to seal the passage. This particular valve has a removable explosive charge. The inlet and outlet ports are in line for minimum pressure drop, but there is some flow passage area variation within the valve.

Normally-open valves are available which have a completely unobstructed bore prior to actuation. Figure 5.7.3c illustrates a normally-open explosive valve utilizing another principle of operation. The explosive charge blows the ball into the high pressure fluid stream, causing the ball to wedge into the conical seat

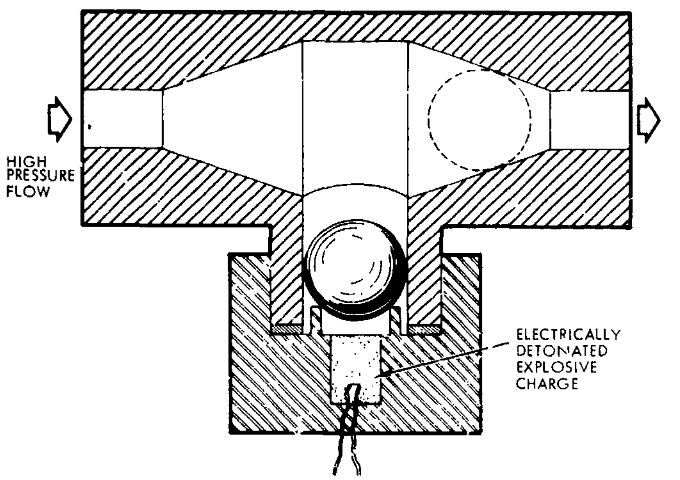


Figure 5.7.3c. Normally-Open Explosive Valve
(Reprinted with permission from "Product Engineering," 24 December 1962, vol. 33, no. 26, M. D. Nelson, Copyright 1962, McGraw-Hill Publishing Company, Inc., New York, New York)

5.7.4 Normally-Closed Valves

Three different normally-closed valves are shown in Figures 5.7.4a, 5.7.4b, and 5.7.4c. In each of these valves, parent metal is sheared by the explosive actuator, permitting fluid to travel from inlet to outlet.

In Figure 5.7.4a, a disc shaped section of metal is punched out by the actuator, clearing the passage between inlet and outlet. The actuator for this valve is identical to the actuator of the valve in Figure 5.7.3a. Because the inlet and outlet ports are not in line, and since the actuator ram creates an obstruction, this valve has somewhat larger pressure drop for an equivalent flow than other explosive valves with straight through connections and no remaining obstacle after firing. However, pressure drop requirements are not usually stringent for the great majority of explosive valve applications.

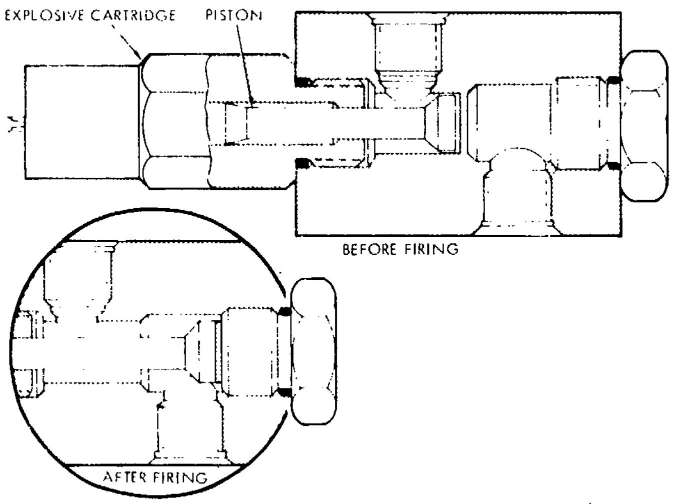


Figure 5.7.4a. Normally-Closed Explosive Valve
(Courtesy of Conax Corporation, Buffalo, New York)

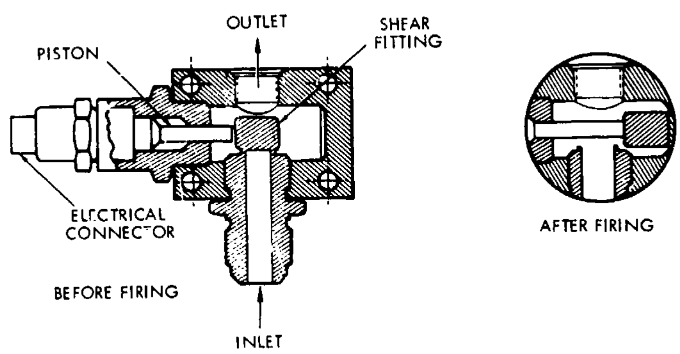


Figure 5.7.4b. Normally-Closed Explosive Valve
(Courtesy of Conax Corporation, Buffalo, New York)

The valve shown in Figure 5.7.4b has a similar actuator to that in Figure 5.7.4a, but has straight-through flow. In this valve, a plug is sheared from the end of the inlet fitting. As in the previous valve, the operating ram remains in the flow path causing some restriction.

The valve shown in Figure 5.7.4c is similar in principle to the previous valve, but has less pressure drop, since the flow is unobstructed after actuation. In this valve, the sheared end of the inlet fitting is carried away by a hole in the operating piston. A new hole in the piston lines up exactly with the centerline of the ports. Two disadvantages of such mating-port valves are the larger mass of the piston assembly (which may in turn require a more powerful explosive charge), and the necessity for careful alignment of the piston with the inlet fitting during assembly

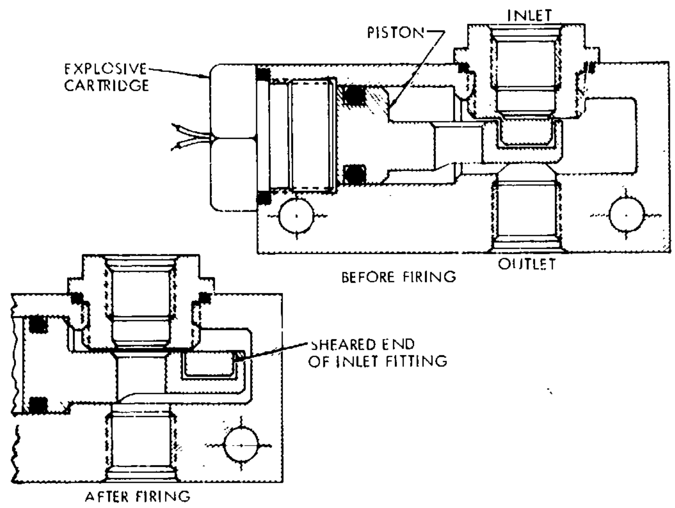


Figure 5.7.4c. Normally-Closed Explosive Valve
(Courtesy of Hydro-Space Technology, Inc., West Caldwell, New Jersey)

5.7.5 Multiple-Passage Explosive Valves

A typical three-way explosive valve is shown in Figure 5.7.5. The outlet port is ducted to one inlet port before firing of the valve, and to the other inlet port after firing. Squib-actuated explosive valves can be designed to provide a variety of porting arrangements.

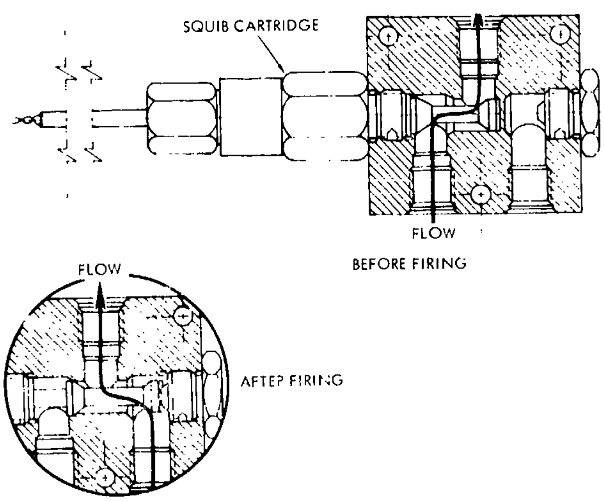


Figure 5.7.5. Explosive Actuated Three-Way Valve
(Courtesy of Conax Corporation, Buffalo, New York)

5.7.6 Pilot-Operated Explosive Valves

In small port sizes (less than one inch), squib valves are usually actuated directly by cartridge pressure. However, as the valve size increases, piloting techniques become more practical, due to the limitations imposed by size of explosive charge, explosive characteristics, heat generation, and safety considerations. In a piloted design, the explosive energy does not directly actuate the valve closure, but serves to release another source of energy which in turn provides the primary actuation power. Two common piloting techniques are:

1) actuation of a small, normally closed explosive valve which in turn releases high energy stored gas, thus providing pneumatic actuation for a larger piston operated valve.

2) release of a latch or pin which allows fluid pressure and/or a compressed spring to actuate the valve.

An in-line poppet-type valve pilot operated by a squib actuated latch is illustrated in Figure 5.7.6. In this design, gas generated by the squib pressurizes the piston, releasing the latch. When unlatched, the poppet is actuated closed by a spring and system pressure. Squib-actuated latch piloting has been used in hermetically sealed valves in sizes upwards of 6 inches with closures of the butterfly and flapper types. The flapper or butterfly is designed such that fluid pressure causes a shearing torque about a fixed pivot point sufficient to rupture a hermetically sealed member. Hermetic sealing is accomplished either by machining the closure as an integral part of the body, or by providing a clamp or welded closure which is opened by rupturing the seal.

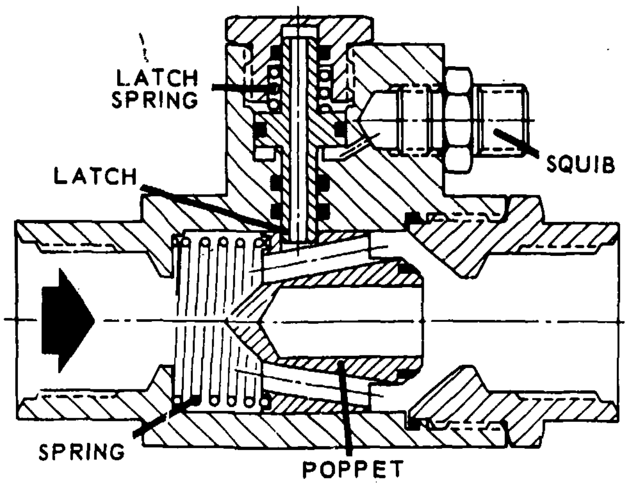


Figure 5.7.6. Pilot Actuated Explosive Valve
(Courtesy of Futurecraft Corporation, City of Industry, California)

5.7.7 Explosive Valves for Repeated ON-OFF Service

Although not normally employed for cyclic operation, explosive actuation can be utilized where ON-OFF service is required. Figure 5.7.7 illustrates an arrangement whereby four parallel flow paths, each composed of one normally-open and one normally-closed explosive valve, provides four ON-OFF operations, one complete ON-OFF cycle for each path. It is also possible, through the use of several explosive cartridges located on either side of the double acting cylinder actuator, to cycle a valve by firing cartridges sequentially. In such a design, check valves are used to protect unfired cartridges during the firing of an adjacent cartridge.

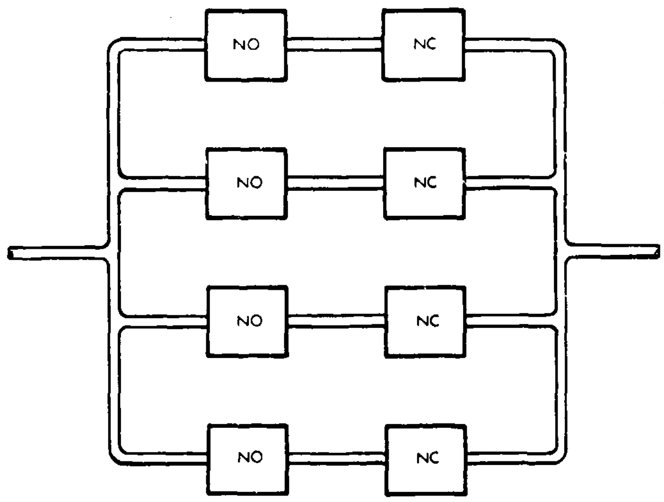


Figure 5.7.7. Four Normally-Open and Four Normally-Closed Explosive Valves Arranged to Provide Four Cycles of ON-OFF Operation

5.7.8 Comparison with Conventional Valves

In the paragraphs which follow, explosive valve parameters are discussed and a comparison is made between explosive valves and conventional valves.

5.7.8.1 COST. The unit cost of an explosive valve is usually less than the unit cost of a conventional valve with the same function. However, the larger number required for qualification testing usually balances the over-all cost. A typical qualification program for a newly designed valve might consist of approximately 50 firings under uniform environmental conditions, plus approximately 20 firings performed at environmental extremes beyond anticipated conditions. Five firings might be made at extreme cold, five at extreme heat, five after overly severe vibration, and perhaps another five after accelerated aging. This entire series of tests would be performed on a sample randomly selected from the production lot. Some previous testing would already have been performed during development of the valve, including under- and over-charge firings to demonstrate margin of safety of the selected charge level.

5.7.8.2 SPEED OF RESPONSE. Response time for most explosive valves is very short, usually falling between 0.001 and 0.025 seconds from receipt of actuating signal to completion of valving element stroke. This total response time actually consists of three distinct phases:

$$t_t = t_a + t_b + t_c$$

where

- t_t total valve response time from switch closed to mechanical stop of moving parts, sec
- t_a elapsed time from switch closed to first indication of explosive burning (pressure rise, etc.), sec
- t_b elapsed time from first indication of explosive burning to start of valving element motion, sec
- t_c valving element response time from start of valving element motion to completion of stroke, sec.

The first phase t_a, usually constitutes the largest portion of the total valve response time, t_t, while the sum of the latter phases, t_b and t_c, is relatively small, usually in the order of 0.0002 second. For this reason, t_a is comparatively easy to measure with conventional instrumentation, whereas t_b and t_c may be determined accurately only with sophisticated instrumentation.

Because t_a represents the total time required to heat the bridgewire to a temperature sufficient to initiate the explosive train, the magnitude of this period is dependent upon:

a) Type of ignition system. A primer design having a high current firing threshold usually incorporates an ignition mix having an inherently high temperature resistance and a correspondingly high ignition temperature. Such high temperature designs therefore require a longer time for a given current to heat the bridgewire to the necessary ignition temperature than do more sensitive systems. In addition, high temperature compounds usually possess relatively slow burning rates (as compared with "brisant" detonating compounds) such that the period from ignition of the primer spot to full burning of the base charge may be several milliseconds.

b) Type and resistance of the bridgewire. A high current firing threshold can be obtained with an explosive ignition mix having a low ignition temperature by reducing the bridgewire resistance.

c) Magnitude of the applied firing current. The greater the firing current, the shorter will be the time required to heat a given bridgewire to the ignition temperature. Figure 5.7.8 is a representative example of this relationship.

d) Ambient temperature. For any explosive, the firing time tends to increase as temperature decreases, because the bridgewire is initially at a lower energy level. This effect is relatively minor, as illustrated by a series of tests wherein firing time of a given design increased from 0.0012 to 0.0015 second as temperature was reduced from 20° to 70°F (Reference V-25).

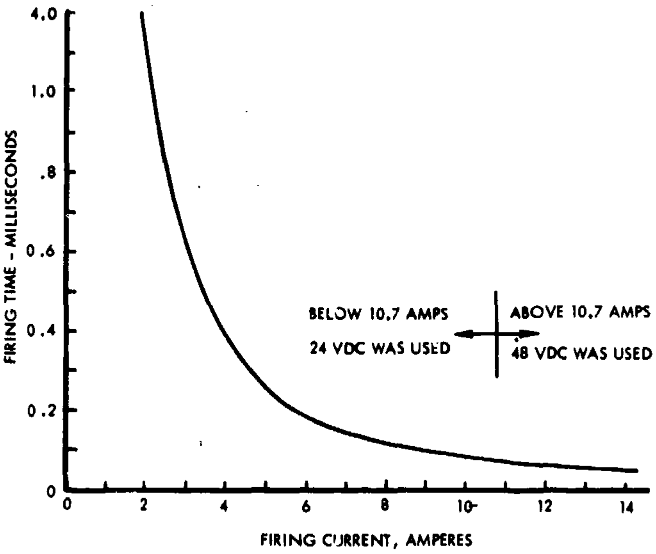


Figure 5.7.8.2. Effect of Bridgewire Current on Firing Time (Typical)

(Courtesy of Conax Corporation, Buffalo, New York)

It may be seen from this discussion that t_a is mainly controlled by the sensitivity of the explosive ignition system, highly sensitive systems being faster than the less sensitive systems. In general, the more sensitive systems are less resistant to accidental firing by static electricity or stray RF current than the less sensitive ignition systems.

The second phase of the response time, t_b, is invariably short relative to t_a. As shown in Figure 5.7.2.1, pressure buildup with a deflagration charge occurs in something less than 0.001 second, and valving element motion can be expected to start at some time well before peak pressure is achieved. The actual elapsed time required for the t_b phase will be a function of:

a) Properties of the explosive material. Detonation charges burn somewhat faster than certain slow-burning deflagration charges.

b) Size of the charge and case. A small charge will require less time to completely react than will a large charge of the same explosive.

c) Mechanical design of the actuating piston and valving element. A deflagration charge design which requires shearing large quantities of parent metal or a large shear pin before stroke commences will require that pressure build up further than a relatively free moving design which permits the stroke to start shortly after pressure begins to rise. Similarly, it is important to note that the valving element is that part of the valve which affects fluid flow when it moves (see Sub-Section 6.2), and does not necessarily start its stroke at the same time as the actuator piston. Phase t_b is not completed until the valving element starts to move.

The final phase of explosive valve response time, t_v, is that required for the valving element to complete its stroke. The duration of this phase is also characteristically very short, but may be significantly increased if the valving element must move through a liquid or further compress a high-pressure gas. Fluid entrapment, wherein the fluid must be displaced through a small clearance or orifice, can slow the valving element down and should be avoided.

The short total response time of explosive valves is advantageous in systems requiring precise synchronization, such as small liquid propellant feed systems or ballistic missile separation devices. The water hammer shock associated with extremely short valving element response times can cause problems in liquid systems. Sub-Topics 3.10.3 and 3.10.4 describe the basic procedures for water hammer analysis.

5.7.8.3 ACTUATING POWER REQUIREMENTS. Because the main source of energy is contained within the explosive valve, little power is required to operate the valve. A typical electrically-initiated cartridge can be safely fired by a 5 amp current for 0.020 second flowing through the 1.0 ohm resistance bridgewire. The actual power requirement would be somewhat greater than 25 watts, since additional resistance would generally be placed in the firing circuit whenever two or more cartridges were fired from the same power source. This would be done to prevent a single cartridge from drawing all available power if it should short out in firing. In general, a firing circuit is designed to supply minimum required power to any single cartridge on the line, even if all other cartridges are shorted.

Mechanically initiated cartridges require correspondingly small amounts of initiation energy.

5.7.8.4 SIZE AND WEIGHT. The size and weight of an explosive valve is lower than that of a corresponding conventional valve, due primarily to the compact energy source and the expendability of the valve body and cartridge. The weight saving reflects further into the over-all missile or aircraft system since the actuating energy requirement is small. Weight as a function of flow area is given approximately by the relations

$$W = 0.10 + 4.52A \quad (A \leq 0.11 \text{ in}^2)$$

$$W = 0.40 + 1.75A \quad (A \geq 0.11 \text{ in}^2)$$

where W = total valve weight, lb_f
A = valving unit flow area, in^2

5.7.8.5 CONTAMINATION. It is possible for products of combustion to leak from the actuation chamber of an explosive valve into the fluid lines unless specific care is taken to prevent this. In general, even a double O-ring on the actuating piston is insufficient to prevent some blow-by of combustion gases, because the pressure build up is too rapid for the pressure assisted seals to follow. The amount of leakage is usually very small, the greatest majority of combustion products being contained. The actuating piston can have an interference fit with the cylinder if greater control of combustion product by-pass is needed.

Another form of potential contamination, the flaking of metal particles from the fracture portion of the valve seal, is not a problem in correctly designed explosive valves. The section to be fractured is necked down to form a localized notch, controlling the failure precisely.

5.7.8.6 PRESSURE DROP. Both normally-open and normally-closed explosive valves can be designed to have straight-through flow with no interruptions or changes of cross-sectional area. The pressure drop is as low as that obtainable with a ball valve.

5.7.8.7 SAFETY. Explosive valves are as safe or safer than conventional valves, since they are usually designed with large factors of safety. The mechanical simplicity of explosive valves permits this to be done easily.

As all significant explosive products are contained, there is no danger to personnel in the area, even if a valve should be accidentally fired. Most explosive cartridges are relatively insensitive to accidental firing, some being able to withstand a continuous power dissipation through the bridgewire of 1.0 watt without firing.

Reference 457-5 presents detailed design criteria for minimizing accidental firing hazards associated with electromagnetic radiation, radio frequency (RF) initiation, static electric discharge, and spurious signal pickup.

5.7.8.8 RELIABILITY. Reliability of even single cartridge explosive valves is extremely high, due partly to their mechanical simplicity and partly to the high reliability of all explosive cartridges. Redundant cartridges can be used to actuate a valve where either extremely high reliability is required or the expense of demonstrating required reliability with a single cartridge would be excessive. This makes the valve somewhat more difficult to design, since it must operate successfully when either or both cartridges fire.

Normally-open explosive valves are often arranged in series, and normally-closed valves in parallel to achieve high system reliability.

5.7.8.9 RESISTANCE TO EXTREME ENVIRONMENT. Explosive valves are generally more resistant to environmental extremes of temperature, shock, and vibration than other types of valves, since they have fewer moving parts and larger factors of safety in their actuating force. Well-designed explosive cartridges have no difficulty surviving extreme missile environments, and can function successfully at temperatures ranging from −320 to +250°F. Elevated temperatures tend to degrade performance of the explosive, which may result in slow response. The explosive has a "memory" with respect to exposure to high temperatures and there is usually no means of determining if a particular valve has been subjected to an extreme environment prior to use.

Biological sterilization, generally recognized as a requirement for a planetary landing vehicle, can be achieved through dry heat, the only practical and certain method. The general criterion for this procedure is heating to 293°F for 36 hours three successive times, with unspecified periods of normal room temperature in between. With respect to such heating cycles, the various explosive devices aboard a vehicle are among the more critical parts. Reference 564-1 describes a broad survey of explosives to see which ones possess sufficient thermal stability to withstand such sterilization.

Extreme radiation environments can also affect the performance of explosive valves. Whereas some explosives such as lead styphnate are relatively insensitive to radiation exposure, most explosives demonstrate some degradation in the form of gas evolution, increased impact sensitivity, and/or decreased brisance (Reference V-25). The level of radiation required to significantly affect explosive performance is indicated in the series of tests conducted by Oak Ridge National Laboratory wherein explosives were subjected to an average of 100,000 .. ntgens per hour for periods of 45 to 90 days. This investigation is reported in "Effects of Gamma Radiation on Explosives," report ORNL-1720, dated November 28, 1962.

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2-7
19-185
262-4
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V-288
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547-8*
560-1*
564-1*

ISSUED: MARCH 1967
SUPERSEDES: MAY 1964

5.8 MULTIPLE-PASSAGE VALVES

5.8.1 Introduction

A multiple-passage valve may be defined as a directional control valve which stops, starts, and diverts flow between three or more alternate flow paths. Multiple passage valves are used to control fluid to and from actuating cylinders and to control the direction of flow in numerous applications where switching or directing flow between various paths is required. Common designations for multiple-passage valves are three-way valves, four-way valves, diverter valves, sequence valves, and shuttle valves. Actuation may be manual, mechanical, hydraulic, pneumatic, or electrical. Multiple-passage valves are identified by: the method of actuation; number of ports; number of positions to which the valve can be actuated; type of valving element (spool, slide, poppet, ball, etc.); and type of sealing. The purpose of this Sub-Section is to describe multiple-passage valves in terms of function terminology and basic design types.

5.8.2 Description and Function of a Multiple-Passage Valve

Multiple-passage valves are used to extend and retract piston-cylinders, rotate fluid motors and actuators, and sequence other hydraulic or pneumatic circuit operations. Multiple-passage valves operate in two or more discrete positions and translate between these positions only during valve shifting. Unlike control valves and servo valves, they are not designed to operate in a proportional or throttling mode. In this respect they may be thought of an an ON-OFF device, as contrasted to a proportional control device. Figure 5.8.2 illustrates the use of multiple-passage valves to control the position of a pneumatic piston-cylinder. The pilot stage consists of a three-way, solenoid-operated, ball-type poppet valve. The main stage is a four-way, two-position, ball-type poppet valve operated by means of a single-acting cylinder, actuated by the first stage. In the de-energized position the main stage cylinder actuator is connected, and the work cylinder is in the down position. When the solenoid is energized, the main stage actuation cylinder moves down, and the upper work cylinder is vented; while the shaft end of the work cylinder is pressurized and the work cylinder moves up. By using a pilot stage and a main stage, a relatively small solenoid with small electrical requirements can operate a large work cylinder.

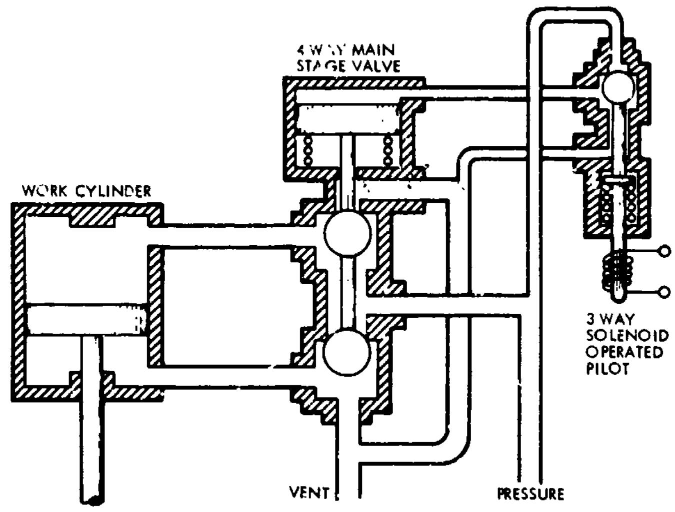

Figure 5.8.2. Four-Way Piloted Solenoid Valve Controlling a Double Acting Cylinder

5.8.3 Multiple-Passage Valve Terminology

5.8.3.1 **DIRECTIONAL CONTROL VALVE.** Directional control valve is a term used to describe all multiple-passage valves, because their primary function is to control the direction of flow from one fluid line to another. Check valves are also considered directional control valves, in that a check valve is included in a system so that flow proceeds in one direction only; that is, it checks reverse flow. Check valves are described in Sub-Section 5.9.

5.8.3.2 **THREE-WAY VALVES.** A three-way valve is a valve with three external port connections. Three-way valves are either two- or three-position valves. The usual three-way valve has one common port, which can be connected to either one of two alternate ports while closing the non-connected port. Normally, these ports are identified as pressure, cylinder, and reservoir (vent) ports. When used to control a single-acting cylinder, the cylinder port is the common port. It is connected alternately to the pressure port and to the vent port.

5.8.3.3 **FOUR-WAY VALVE.** A four-way valve is a valve which has four external port connections usually arranged such that there are two simultaneous flow paths through the valve. The ports for a four-way valve are commonly identified as pressure, reservoir (vent), and two-cylinder ports. Four-way valves are commonly used to actuate double-acting cylinders. In such applications, the valve is connected such that when pressure is applied to one cylinder port the other cylinder port is vented, and vice versa. Four-way valves are normally two- or three-position valves. In a three-position, four-way valve, there is a center position in which all ports are closed.

5.8.3.4 DIVERTER VALVE OR DIVERSION VALVE. A diverter valve is basically a three-way valve, with the common port being the pressure port. Flow can be diverted from the pressure port to either one of two alternate flow paths. Diverter valves are also commonly called diversion valves.

5.8.3.5 SELECTOR VALVE. A selector valve functions similarly to a diverter valve, except that the number of alternate flow paths to which the common pressure port can be connected is unlimited.

5.8.3.6 SEQUENCE VALVE. A sequence valve is a valve whose primary function is to direct flow in a pre-determined sequence between two or more ports. A *shuttle valve* is a type of sequence valve which is pressure actuated such that when a preset system pressure has been reached, the valve automatically actuates, connecting two or more flow paths.

5.8.3.7 VALVE POSITION. The valve position is the point at which the valving elements provide a specific flow condition in the valve. The following terms relate to valve position:

Closed center: all ports are closed in the center position.

Open center: all ports are interconnected in the center position.

Detent position: a pre-determined position maintained by a holding device acting on the valving element of a multiple passage valve.

Hold position: a selective position in a multiple passage valve where the working ports are blocked to hold a power device in a fixed position.

Normal position: the valve position when signal or operating force is not being applied.

Normally-closed: specific flow paths are closed in the normal position.

Normally-open: specific flow paths are open in the normal position.

5.8.4 Multiple-Passage Valves Using Linear Motion

Multiple-passage valves that use linear motion of the valving element are discussed in the following Detailed Topics.

5.8.4.1 SPOOL VALVES. Spool valves control fluid flow by covering and uncovering annular ports with lands on a sliding spool. The number of lands and ports on the spool and valve body determine the porting arrangements that can be achieved, and the geometrical relationship between the lands and ports determine the timing of the valve function. With sharp edged land and port, operation of a spool valve is abrupt. In applications where this would cause undesirable pressure surges, the land edges can be notched, tapered, or chamfered to modify the flow characteristics.

Spool valves are classified as packed or unpacked, depending upon the sealing characteristics. Packed spool valves utilize O-rings or some other type of seal between the spool and valve body to achieve tight shutoff. Unpacked spool valves possess internal leakage, depending upon the clearance between the spool lands and valve body. Annular grooves are sometimes machined on the spool lands to improve lubrication of the valve and to equalize pressure all around the spool to prevent binding on one side of the bore. In addition to eliminating binding, the annular groove centers the valve and in this position the leakage clearance is minimized.

An example of a three-way, two-position spool valve is presented in Figure 5.8.4.1a. In the right hand position, pressure is connected to the cylinder or load. When the valve is shifted to the left, the pressure is sealed off and the cylinder is vented to the reservoir. Figure 5.8.4.1b presents a schematic of a typical four-way spool valve. If this valve is used in a two-position configuration, the spool is placed in either extreme end of the body to cycle the work cylinder back and forth. If the spool is placed in a center position of the body (three-position valve), the work cylinder will be in a locked position and the pressure will be deadheaded in a closed-center condition. Figure 5.8.4.1c illustrates a three-position, four-way hollow spool valve. In the center position, pressure is valved to the reservoir return line.

This valve configuration is suitable for application in a tandem valve system. Tandem valves are valves which are arranged so that pressure is vented to a reservoir line in the center position. This reservoir line is in turn used as the supply pressure for a subsequent valve in the system, etc. Thus, the pressure is used in tandem in a number of valves in the system before fluid is returned to the reservoir.

One unique feature of the spool-type, multiple-passage valve is that the end of the spool can be used as the actuator piston to position the valve. Spools as valving elements are discussed in further detail under "Servo Valves" in Detailed Topic 5.6.7.2.

5.8.4.2 POPPET VALVES. Poppet-type, multiple-passage valves utilize two or more flat, conical, or spherical seats on a translating poppet. Poppet-type, multiple-passage valves lend themselves to three- or four-way operation with a variety of seating and sealing arrangements. Figure 5.8.4.2a illustrates a solenoid-actuated, three-way poppet-type valve. In the de-energized position, the cylinder port is connected to the reservoir, and in the energized position pressure is applied to the cylinder. Figure 5.8.4.2b is a schematic of a four-way, poppet-type, multiple-passage valve with pneumatic piston-cylinder actuation. The valve is pressure-

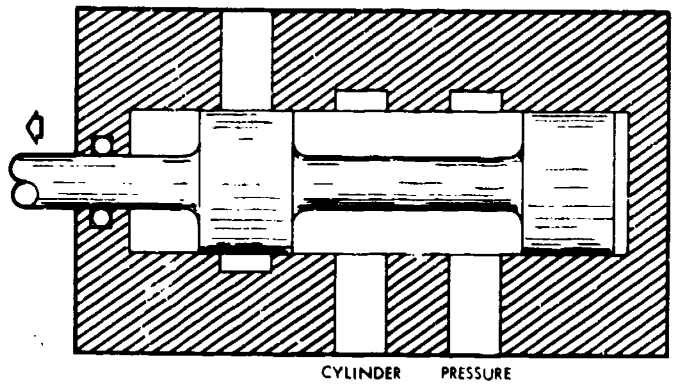

Figure 5.8.4.1a. Three-Way Spool Valve

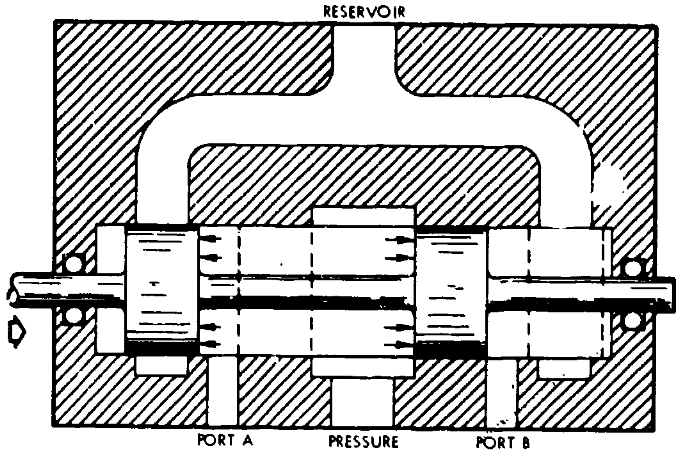

Figure 5.8.4.1b. Four-Way Spool Valve

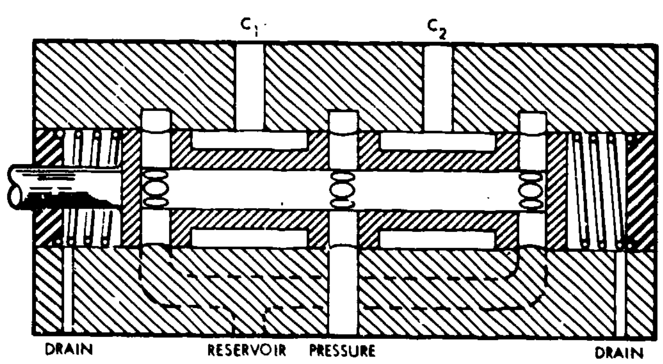

Figure 5.8.4.1c. Three-Position, Four-Way, Hollow Spool Valve With External Drain and Spring Centering Mechanism

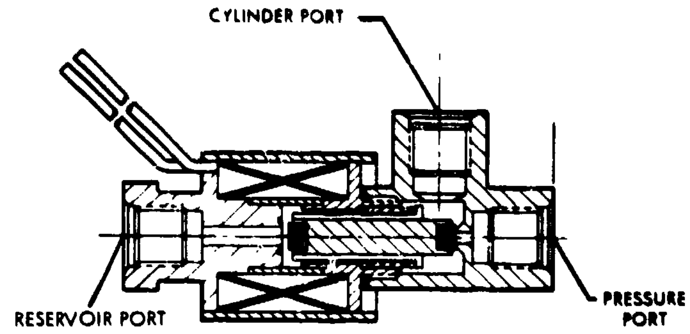

Figure 5.8.4.2a. Solenoid-Actuated Poppet-Type Three-Way Valve
(Courtesy of Valcor Engineering Corporation, Kenilworth, New Jersey)

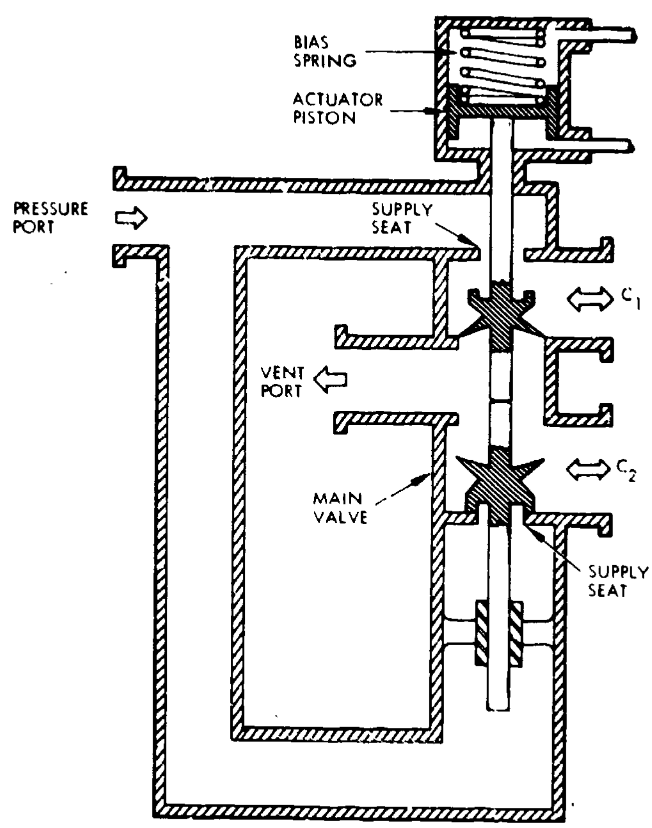

Figure 5.8.4.2b. Schematic of a Four-Way Flexible-Lip Poppet Valve
(Reference 71-1)

actuated. In the position shown, port C_1 is pressurized and port C_2 is vented. Application of pressure to the under side of the actuator piston while venting the top side actuates the valve such that C_2 is pressurized and C_1 is vented. Simultaneous seating of two poppets, as required for a four-way valve, is difficult to achieve with hard seats. In the design illustrated, a Belleville-shaped poppet provides the flexibility required for simultaneous sealing.

5.8.4.3 **SLIDING PLATE VALVES.** The sliding plate valve consists of three main elements, a slide and two plate enclosures. The slide element is sandwiched between the two plates, and contains cored holes and passages which mate with ports in the plates. Many porting arrangements are available, and three- and four-way multiple-passage valves can be easily achieved. One advantage of a sliding plate valve is that it can be easily reworked and lapped to compensate for wear.

5.8.5 Multiple-Passage Valves Using Rotary Motion

Multiple-passage valves utilizing rotary motion between the valving element and the valve body are discussed in the following Detailed Topics.

5.8.5.1 **BALL VALVES.** Ball valves, discussed under "Shutoff Valves" in Sub-Topic 5.2.3, can be readily adapted to operation as multiple-passage valves by the addition of outlets on the body and additional porting in the ball valving element. With three outlet connections on the body, the ball valve can be made into a variety of three-way valves, depending upon the porting utilized in the ball. Figure 5.8.5.1 illustrates two configurations of a three-way ball valve utilizing an L-porting configuration in one ball and a T-porting configuration in the other. Two- and three-position, three-way valve configurations are possible with two or three ports in the ball. A valve body with four outlet ports can be adapted to a number of types of four-way valves depending upon the port configuration used in the ball. Ball valves have low pressure drop and good sealing characteristics, as discussed in Sub-Topic 5.2.3.

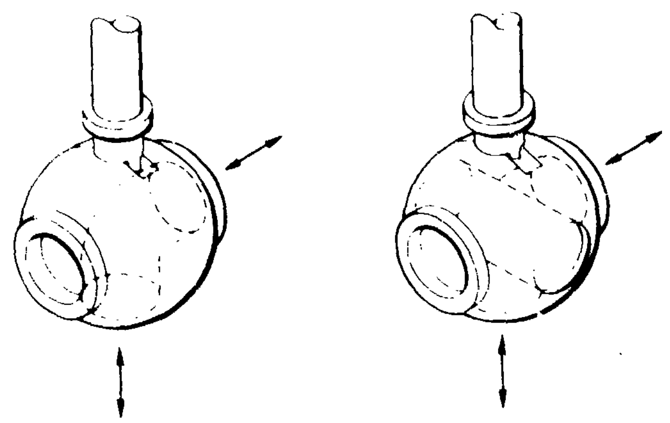

Figure 5.8.5.1. Multiported Ball Valves
(Courtesy of Pacific Valves, Inc., Long Beach, California)

5.8.5.2 **PLUG VALVES.** Plug valves, discussed under "Shutoff Valves" in Sub-Topic 5.2.11, can be adapted to three- or four-way, multiple-passage valve operation by the addition of outlets on the housing and additional porting on the plug. Figure 5.8.5.2a illustrates a top view of a four-way rotary plug valve. The valve illustrated is a closed-center, four-way valve. By variations in porting arrangements in the rotary plug, a number of different types of three- and four-way valves can be achieved, as illustrated in Figure 5.8.5.2b.

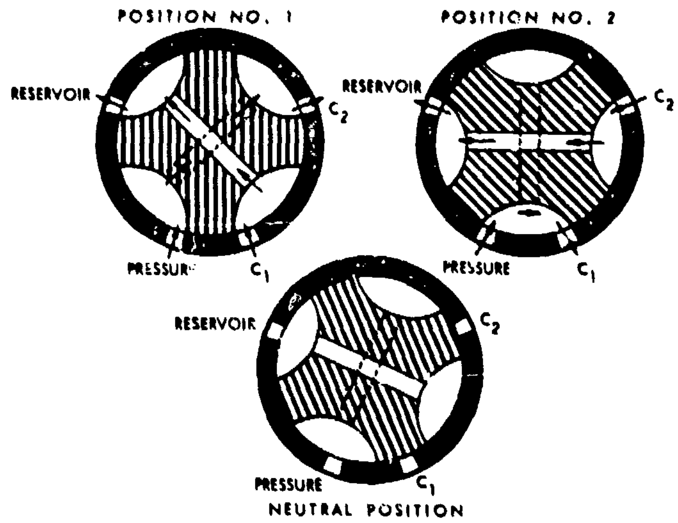

Figure 5.8.5.2a. Four-Way Rotary Plug Valve Closure
(Reprinted from "Product Engineering," June 1955, vol. 26, no. 6, W. Brown, Copyright 1955, McGraw-Hill Publishing Company, New York, New York)

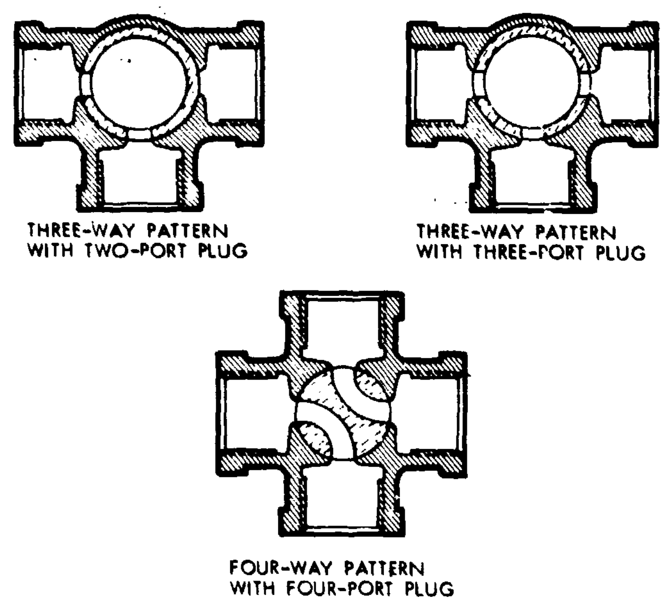

Figure 5.8.5.2b. Porting Arrangements for Three- and Four-Way Rotary Plug Valves
(Courtesy of The Lunkenheimer Company, Cincinnati, Ohio)

5.8.5.3 **ROTARY SLIDE VALVES.** Rotary slide valves, discussed under "Shutoff Valves" in Sub-Topic 5.2.12, are more commonly used as multiple-passage valves than as two-way shutoff valves. Rotary slide, multiple-passage valves consist of two essential parts, the body and a rotating plate. The body contains three or four outlets for either a three- or four-way valve configuration, and the rotary plate contains various porting arrangements to achieve a multiplicity of three- or four-way valve types. Several porting arrangements for three- and four-way rotary slide valves are illustrated in Figure 5.8.5.3.

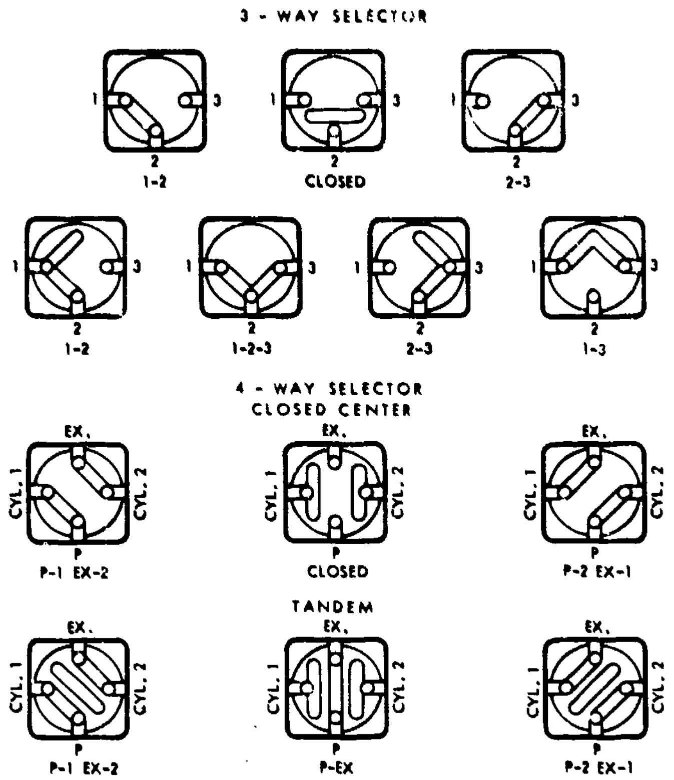

Figure 5.8.5.3. Porting Arrangements for Three- and Four-Way Rotary Slide Valves
(Courtesy of Republic Manufacturing Company, Cleveland, Ohio)

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6-11	19-86	V-61
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6-109	112-4	V-172
19-16	462-1	V-271

5.9 CHECK VALVES

5.9.1 Introduction

The primary function of a check valve is to prevent flow reversal. Check valves pass fluid freely in one direction and, if pressure reverses, close quickly to stop flow in the other direction. Flow reversal in fluid systems may be programmed as a normal occurrence or may be caused by accidental occurrences or failures. Accidental flow reversal must be promptly and effectively halted or reservoirs may overflow, tanks may be over-pressurized, reactive fluids may combine, rotating equipment may overspeed, or other types of equipment damage may occur.

Check valves are entirely automatic in their operation, their valving elements being activated by the forces of the flowing media. Valving elements of check valves are either spring-loaded closed or gravity closed. The spring-loaded type is the only one of significant interest in aerospace applications, due to the uncertainty of vehicle attitude or of the existence of gravitational forces.

Check valves are automatic devices, requiring no external actuation signals or sources of power. They are the simplest of valve types. However, in spite of their simplicity, they are often one of the most troublesome components in a fluid system. Having no actuators, check valves are often lacking in sufficient seating force to accomplish a good seal. Most check valve designs are a compromise between good sealing and low pressure drops.

5.9.2 Design and Selection Parameters

Parameters that must be considered in the selection of a particular check valve configuration are discussed in the following Detailed Topics.

5.9.2.1 **PRESSURE DROP.** In aerospace fluid systems, it is usually desirable to minimize pressure drop across the valve as a function of flow rate. The comparative pressure drop characteristics for several types of check valves are shown in Figure 5.9.2.1. When specifying the pressure drop of the check valve, it is important that the actual operating conditions of flow rate, pressure, temperature, and flow medium are accurately stated.

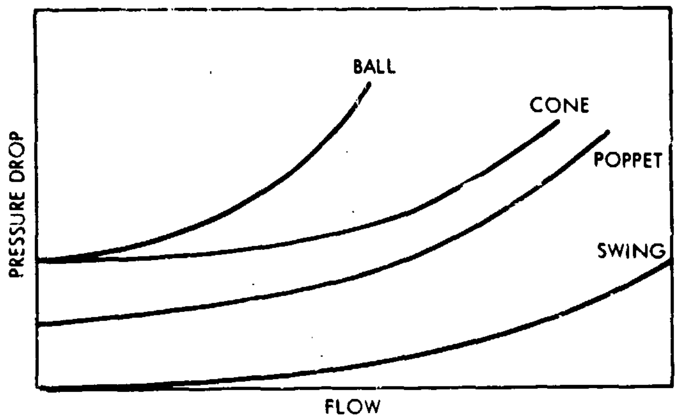

Figure 5.9.2.1. Typical Check Valve Pressure Drop Curves
(Reprinted from "Applied Hydraulics," April 4, 1957, vol. 10, no. 4, D. Van Deerlin, Copyright 1957, Industrial Publishing Corporation, Cleveland, Ohio)

5.9.2.2 **SEALING.** The sealing characteristics of the check valve are dependent upon the seal material, initial spring-loading, and differential pressure across the closure element. When pressure reversal occurs, the check valve will leak until the back pressure becomes high enough to provide an adequate seating force. The requirement for zero leakage at zero or very low reversal pressure necessitates the use of higher initial spring load, which will result in higher pressure drops through the valve in the forward flow direction. The use of resilient seat materials wherever possible will aid in achieving minimum leakage at low seat loadings.

5.9.2.3 **CRACKING AND RESEAT PRESSURE.** Cracking pressure is defined as the forward differential pressure which is required to pass some specified minimum flow rate. Reseat pressure is that reverse differential pressure required to reduce leakage to some minimum specified value. With sufficient spring-loading and proper seat characteristics, the reseating pressure can be zero. Both the cracking and reseating pressure characteristics of the valve are related to the pressure drop and leakage characteristics.

5.9.2.4 **FLOW MEDIUM.** The selection of a check valve for a particular application will depend upon the nature of the fluid medium. The primary consideration will be the compatibility of seal materials, metals, and coatings used in the valve construction. The choice of seals in the valves may be dependent upon whether the fluid medium is a gas or liquid. Special sealing problems are encountred with light gases such as helium and hydrogen as the flow medium.

5.9.2.5 **OPERATING PRESSURE.** The most significant consideration for operating pressure is that the valve housing must be designed for the maximum pressure to which it may be subjected. The possibility of rapid valve closure causing water hammer must be taken into consideration, in that it may cause a pressure rise several times the normal operating pressure. A maximum differential pressure in the reverse direction must be considered in the design of the valve closure and seals to prevent fracture of closure and extrusion of seals. If the valve may be subjected to a large number of pressure cycles, consideration must be given to the possibility of fatigue failure.

5.9.2.6 **OPERATING TEMPERATURE.** The design and selection of the valve must take into account the operating temperature of both the fluid medium and the valve environment. Of particular concern will be the effect of operating temperature on any non-metallic seal materials which may be used in the valve. All materials utilized in the valve must be selected to withstand the worst temperature conditions. Short term temperature excursions of either the ambient or the fluid medium may be tolerable beyond the recommended service temperature of the materials, due to the inherent heat sink of the valve mass. Particular attention should be given to temperature gradients that may exist in the valve because of the possible effect of binding due to differential thermal expansion. The effect of high temperatures must be considered, since they may cause permanent seal damage and resultant leakage; however, low temperatures must also be considered in causing loss of seal resilience with resultant leakage.

5.9.2.7 **CONTAMINATION SENSITIVITY.** Contamination in a fluid system or in the fluid medium may cause a valve malfunction or internal leakage. The degree of contamination to be expected in a fluid system should be considered in the selection of a valve, particularly with respect to the valving element and the type of seals. For contaminated fluid service, close fitting parts and mechanisms in the valving element should be avoided, and seals with the maximum degree of resilience should be used.

5.9.2.8 MAINTENANCE. Maintenance requirements for the valve should be kept minimum, and the need for special tools should be avoided if possible. If seals need to be replaced on a periodic basis, they should be standard available parts and readily accessible. Disassembly or service should require minimum teardown and minimize the possibility of contamination.

5.9.2.9 WEIGHT AND SIZE. Weight and size are of prime consideration for valves for any aerospace application, and these considerations will often dictate the type of valve to be utilized. For example in a ball check valve, the weight of the ball increases with the cube of the line size; whereas in a flapper-type check valve, the weight of the closure would be more nearly proportional to the square of the line size. Therefore, in aerospace applications where weight is of prime consideration, a ball-type check valve would very rarely be used for line sizes greater than 1 inch. For larger diameter lines, a flapper or swing type check valve would be used.

5.9.2.10 COST. Some of the many factors influencing cost of a check valve include operating pressure, temperature, flow rate, leakage requirements, flow medium, weight, reliability, and life cycle requirements. While cost may not be as important as performance and reliability for aerospace applications, none of the above cost factors should be over specified so that they adversely affect the final cost of the valve.

5.9.2.11 OPERATING LIFE. A great many factors can affect the operating life of a valve. Among these are flow medium, contamination, operating temperature, stresses, number of cycles, type of seals, materials of construction, and maintenance procedures. In general, a tradeoff exists between weight of the valve and operating life. To achieve the maximum operating life, the design is made so as to minimize loads and stresses on valve components. However, in order to conserve weight, a valve should not be overdesigned for its life cycle requirement in aerospace flight applications.

5.9.3 Ball Check Valves

5.9.3.1 DESCRIPTION AND OPERATION. A typical ball check valve is illustrated in Figure 5.9.3.1. A closure element of the valve is a hard ball which is spring-loaded against a circular, conical, or spherical seat. To minimize leakage, a soft seat such a Teflon, or other plastic or elastomer may be used. In some designs, a combination of soft and hard seat are used. Flow forces lift the ball off the seat against the loading spring and flow passes around the ball.

5.9.3.2 PERFORMANCE CHARACTERISTICS. The flow must proceed around the ball, resulting in a tendency toward turbulence and higher pressure drop than in other types of check valves. During the course of normal operation, the ball may rotate slightly on the retaining spring,

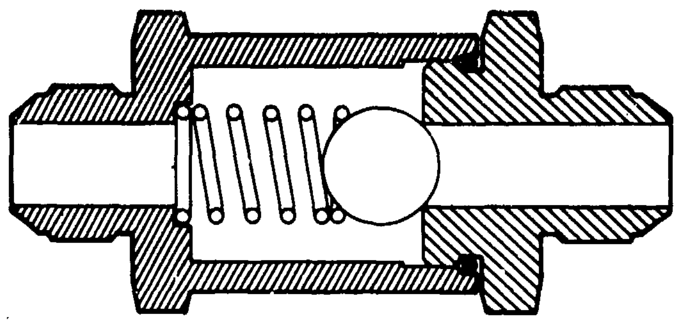

Figure 5.9.3.1. Typical Ball Check Valve

thus causing even wear on the ball and valve seat and minimizing the effects of contamination. Ball check valves have a tendency to chatter, particularly on rapid closure.

5.9.3.3 APPLICATIONS AND LIMITATIONS. Because of their inherent simplicity and low cost, ball check valves the frequently used in applications of small line diameters where pressure drop is not of a particular concern. Practically no damping can be incorporated in the mechanism of the ball check valve and the chattering tendency cannot be eliminated. Therefore, ball check valves are not recommended for applications where chattering is unacceptable.

5.9.4 Cone Check Valve

5.9.4.1 DESCRIPTION AND OPERATION. The cone-type check valve is essentially an outgrowth of the ball type. In cone check valves, the ball is replaced by a sliding element with a conical seating surface at one end, as illustrated in Figure 5.9.4.1a. This conical surface seats against either a circular sharp edge or a conical surface which may be provided with either a soft or hard seat. A variation of the cone check valve which incorporates a spherical seat is illustrated in Figure 5.9.4.1b. The spherical seat geometry minimizes seat and poppet alignment requirements. The particular valve illustrated incorporates a combination hard and soft seat for minimum leakage at low back pressures.

A variation of the cone check valve, called a restriction check valve, is illustrated in Figure 5.9.4.1c. In the restriction check valve, full flow is allowed in the forward direction, while a restricted flow is obtained in the reverse direction through the small orifice in the conical seating element.

5.9.4.2 PERFORMANCE CHARACTERISTICS. Cone check valves generally have less pressure drop for a given size than ball check valves, and have less tendency to chatter because of the guided movement and resultant damping of the valving element. Cone check valves are susceptible to dirt in the seating area and between the piston and body. If contamination lodges between the piston and body it can cause sticking or cocking, with a resultant leakage between the piston and body seating area. The spherical seat variety is less susceptable to leakage caused by cocking of the valv-

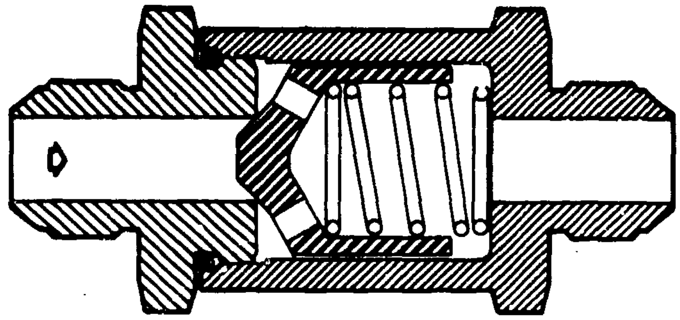

Figure 5.9.4.1a. Typical Conical Poppet Check Valve

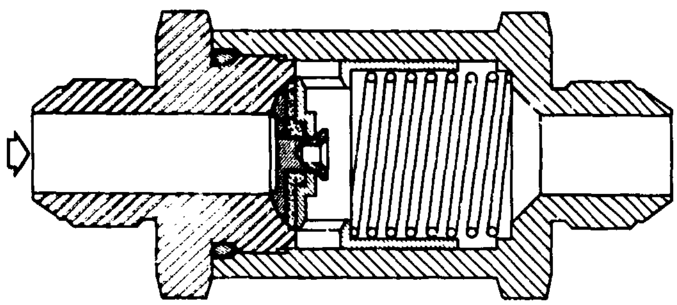

Figure 5.9.4.1b. Poppet-Type Check Valve With a Soft Seal and a Spherical Seat
(Courtesy of Republic Manufacturing Company, Cleveland, Ohio)

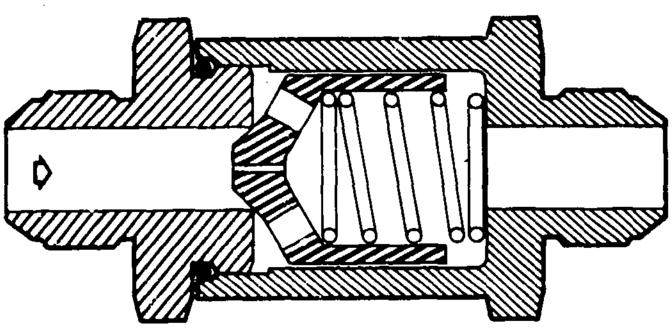

Figure 5.9.4.1c. Restriction Cone Check Valve

ing element with respect to the body due to contamination.

5.9.4.3 APPLICATIONS AND LIMITATIONS. Cone check valves are generally used in the same type of applications as are ball checks. However, cone checks can be used for reduced pressure drop in a given size valve, and can also be used in application where a tendency to chatter cannot be tolerated. Thus, cone checks can be used for somewhat higher flow and higher pressure applications than ball checks. Provision must be made in cone check valve applications to keep contamination to acceptable levels in order to avoid sticking and excessive leakage. A restriction cone check valve, shown in Figure 5.9.4.1c, is useful in applications where full flow is required in one direction and restricted flow is required in the reverse direction. This type of action cannot be achieved in a ball check valve because of the rotation that occurs in the ball element.

5.9.5 Poppet Check Valve

5.9.5.1 DESCRIPTION AND OPERATION. Poppet check valves consist of a mushroom shaped poppet, the stem of which is closely guided in the valve body and the head of which seals against a flat or tapered circular seat. A poppet-type check which utilizes a combination of O-ring and hard conical seat is illustrated in Figure 5.9.5.1. Also shown is an improved O-ring groove design to prevent high velocity fluids from blowing the O-ring demonstration. In this valve, flow forces in the forward direction force the head of the poppet off the seat and flow proceeds through the stem of the poppet, around the head of the poppet, and through the body of the valve.

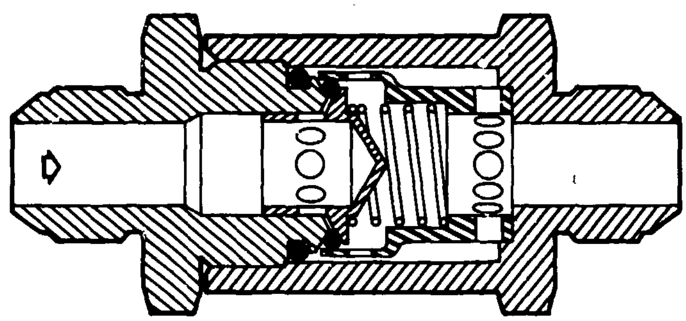

Figure 5.9.5.1. Poppet-Type Check Valve With an O-ring Seal
(Courtesy of Circle Seal Products Company, Inc., Pasadena, California)

5.9.5.2 PERFORMANCE CHARACTERISTICS. Poppet-type check valves, in general, have less pressure drop for a given flow rate than either cone or ball-type check valves. Poppet check valves can be designed to eliminate any tendency toward chatter or hammering by the incorporation of damping chambers in the valve. Because of the close clearances between the poppet stem and valve body, contamination or dirt can cause sticking and leakage.

5.9.5.3 APPLICATIONS AND LIMITATIONS. Poppet check valves must be used only where adequate filtration is provided, to avoid sticking of the poppet. Because poppet check valves can be designed to eliminate chatter, they are useful in applications where chatter cannot be tolerated. Poppet check valves require more parts than ball or cone types and therefore, in general, are more costly. In most cases, poppet check valves can be used in the same applications where ball and cone checks are used, but are more commonly used where it is desirable to improve flow characteristics.

5.9.6 Swing Check Valves

5.9.6.1 DESCRIPTION AND OPERATION. A swing check valve consists of a hinged disc which seats against a flat or tapered surface, or a circular sharp edge. A swing check valve incorporating a soft seat against a flat surface is illustrated in Figure 5.9.6.1. Flow in the forward direction swings the hinged disc from the seat and out of the flow path, allowing a practically unrestricted passage to flow with little pressure drop. When flow through the valve stops or tends to reverse, the disc quickly swings to its seat, sealing the valve against reverse flow.

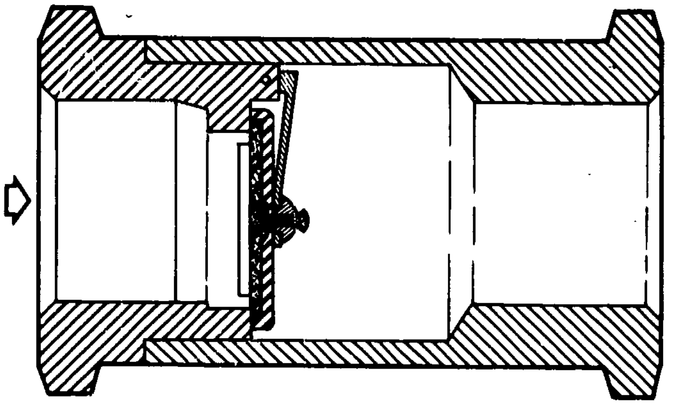

Figure 5.9.6.1. Swing Check Valve Incorporating a Soft Seal
(Courtesy of Republic Manufacturing Company, Cleveland, Ohio)

5.9.6.2 PERFORMANCE CHARACTERISTICS. Because the swing check valve can be designed for small opening forces and the disc can swing out of the flow path, very low pressure drops can be achieved with swing check valves. Swing check valves are particularly subject to water hammer by sudden flow reversal. This is caused by the relatively large disc travel from open to closed position, which can result in a significant reverse flow velocity before the disc is closed. The swing check valve is, therefore, generally inferior to other check valve designs with respect to ability to eliminate reverse flow. Because the disc mechanism has very little damping, the final closing velocity is relatively high. It is this rapid stopping of the reverse flow which produces the water hammer. Fluid contamination has very little effect on the action of the valve disc.

5.9.6.3 APPLICATIONS AND LIMITATIONS. Swing check valves find their widest use in fluid circuits where minimum pressure drop is of paramount importance. Because of their tendency to produce water hammer, swing check valves should not be used in high pressure systems where there is a tendency toward sudden flow reversal. Swing check valves are useful in fluid systems where close control over fluid contamination is not possible.

5.9.7 Flapper Check Valves

5.9.7.1 DESCRIPTION AND OPERATION. Flapper check valves are similar to swing check valves except that instead of a single hinged disc, the flapper check valve has several hinged elements. This makes it possible to design a maximum open area within a given diameter and reduces the moment of inertia of the individual moving parts. A split, flapper check valve, one which utilizes two semi-circular discs or flappers, is illustrated in Figure 5.9.7.1. This valve is designed for flange mounting in an air duct and utilizes flat hard sealing surfaces. Flow through the valve in the forward direction moves the flapper parallel to the flow stream and into the center of the valve.

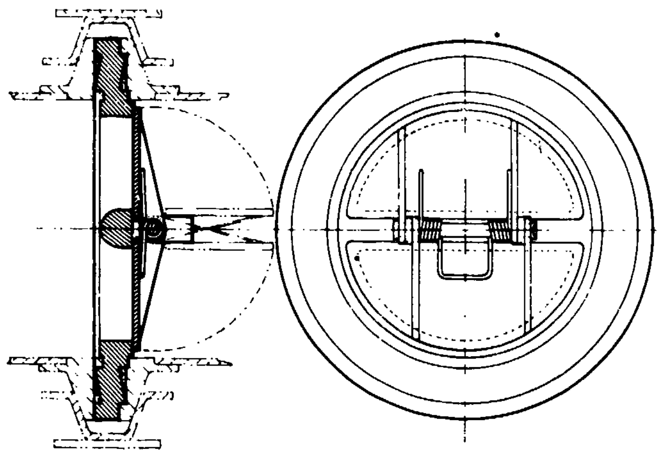

Figure 5.9.7.1. Split Flapper Check Valve for Clamp Mounting Between Flanged Ducts
(Courtesy of Tapco, Thompson Ramo Wooldridge, Inc., Cleveland, Ohio)

5.9.7.2 PERFORMANCE CHARACTERISTICS. A pressure drop across flapper check valves can be kept to a minimum similar to that of swing checks. For a given size valve, flappers are comparatively lighter than swing checks and, because the center of mass is located near the hinge line, they can stand longer and more severe cycle life than a single heavier swinging disc, which would produce greater impact forces when slammed shut. The complicated geometry of a split flapper valve makes good sealing characteristics relatively more difficult to achieve. For a given size valve, flapper check valves can be designed to produce less water hammer than swing check valves, because the flappers do not travel as far to close. The reverse flow velocity and flapper velocity are relatively less at closing.

5.9.7.3 APPLICATIONS AND LIMITATIONS. Flapper check valves have been used primarily in pneumatic fluid systems. Their particular advantage is the minimum pressure drop within a given envelope diameter. Flapper check valves are relatively insensitive to contamination, but would be difficult to design for minimum reverse leakage. In general, flapper check valves are more suitable to applications where sudden flow reversal may occur than swing check valves.

5.9.8 Rubber Check Valves

5.9.8.1 DESCRIPTION AND OPERATION. A rubber check valve is one in which the valve closure elements and sealing surfaces are made of an integral piece of rubber. The valve is designed to take advantage of the elastic properties of rubber. Flow forces in the forward direction deflect the valve closure elements to an open position, while reverse flow closes the valve and produces compression forces in the sealing surfaces. One type of rubber check valve is illustrated in Figure 5.9.8.1. This valve is designed to open to full line size for minimum pressure drop. The conical shape with circular stiffening rings maximizes the reverse pressure that the valve can withstand.

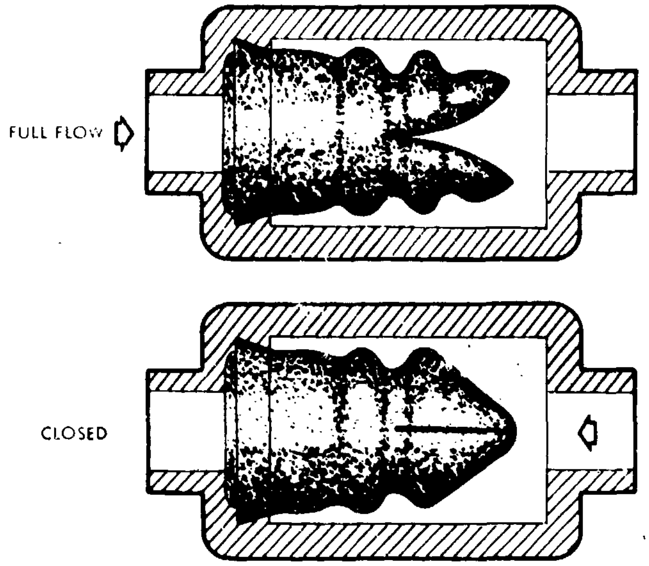

Figure 5.9.8.1. Rubber Check Valve
(Courtesy of Heart Valves, Inc., Mansfield, Ohio)

5.9.8.2 PERFORMANCE CHARACTERISTICS. Rubber check valves can provide mimimum leakage characteristics and are relatively insensitive to forward contamination. They exhibit essentially no tendency to chatter or produce water hammer. A tradeoff exists in the design of rubber check valves of the pressure drop in the forward direction, versus the maximum pressure the valve can stand in the reverse direction.

5.9.8.3 APPLICATIONS AND LIMITATIONS. Obviously, the rubber check valve is limited to fluids which are compatible with the rubber used in the valving element. Primary use of these check valves is on relatively small line sizes, where the reverse pressure that can be exerted is relatively low. The simplicity of this type of check valve makes them relatively low in cost and high in reliability.

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19-187	195-5	V-178
47-2	195-6	V-285
		V-287

5.10 FILTERS AND SEPARATORS

5.10.1 Introduction

Particulate contamination constitutes one of the major causes of fluid component malfunctions. A filter is a device located in a fluid system for the purpose of controlling contamination by trapping particles entrained in the fluid. This sub-section defines terminology commonly associated with filtration and presents a comparison and evaluation of various filter types and performance criteria.

The system performance requirements for a filter can be expressed in terms of the size of the largest particle which can be tolerated in the fluid downstream of the filter under all applicable system conditions. This performance characteristic is a measure of the *degree of protection* offered by the filter and is a direct function of (a) the pore size distribution of the filter medium, and (b) the initial cleanliness level of the filter.

In order to provide this protection throughout the mission/duty cycle of a system, a penalty in total system pressure loss must be tolerated. This may be insignificant when the filter is clean but increases very rapidly as it becomes clogged and approaches its limit of contaminant capacity. This performance characteristic is a measure of the *duration of protection* or useful service life offered by the filter, and is a direct function of the number and geometry of the pores or capillaries of the filter medium provided within a given envelope.

Filtration rating (degree of protection) and contaminant capacity (duration of protection) of a filter are mutually-opposing characteristics in that the finer the filter rating the shorter the service life it can provide within a given cavity or envelope size.

The design criteria of a filter must therefore be based on a tradeoff between (a) filtration rating, (b) contaminant capacity, and (c) envelope size. Once one of these parameters is specified the other two follow a known relationship which is fixed for any given filter medium. If two of these parameters are specified, the third one is automatically established. Similarly, comparisons between different filter media must always be based on total filtration performance, i.e., the ratio between service life and filtration rating per cubic inch of envelope required.

5.10.2 Filter and Separator Terminology

Particle size: defined and measured as the largest dimension of a solid particulate contaminant and quoted in microns as 1 micron, μ, = 10^{-6} meters (Table 5.10.2).

Fiber size: a solid contaminant having a length-to-diameter ratio of 10:1 or greater. Fiber size is measured by diameter and length.

Filter rating, "absolute": the micron size corresponding to removal of 100 percent of all hard, spherical particles (i.e., glass beads) larger than a given size under static blow-down conditions.

Filter rating, "nominal" (obsolescent; seldom used): removal of 98 percent of all incident particles larger than a given size, based on a standard contaminant. The nominal rating is determined by adding a standard contaminant such as glass beads or graduated dust particles of a known size and quantity distribution to the filter influent. The ratio of particles larger than a given size in the effluent to the quantity in the influent is used to determine the nominal rating. The two percent of particles larger than nominal rating which pass through may be of any size up to the maximum size of the test contaminant.

Filter efficiency: the ratio expressed in percent of the weight of particles trapped, divided by the weight of particles introduced into the filter. Filter efficiency is determined using a standardized glass bead contaminant. Efficiency is usually determined on the basis of particles related to the average absolute rating of the filter; however, the efficiency can be determined for any particle size.

Table 5.10.2. Micron Conversion Factors and Comparative Sizes

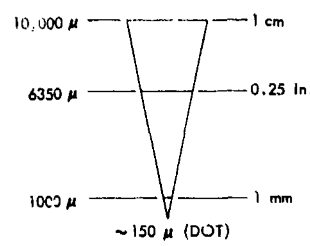
1 MICRON (μ) = 10^{-6} METERS

MICRONS (μ)	INCHES (IN)	MILLIMETERS (mm)
1	$\frac{1}{25.4} \times 10^{-3} = 3.94 \times 10^{-5}$	0.001
25.4	0.001	0.0254
1000	0.0394	1.0
25,400	1.0	25.4

DIA OF HUMAN HAIR ~ 70μ

LOWER LIMIT OF VISIBILITY (NAKED EYE) ~ 40μ

SIZE OF BACTERIA (COCCI) ~ 2μ



SQUARE MESH WIRE SCREEN SIZES

U.S. SIEVE NO. (MESH)	50	100	140	200	270	325	400
WIRES PER INCH	52.36	101.01	142.86	200.0	270.26	323.00	400
WIRE DIAMETER (In.)	0.0085	0.0043	0.0030	0.0021	0.0015	0.0012	0.0010
SIEVE OPENING, (In.)	0.0117	0.0059	0.0041	0.0029	0.0021	0.0017	0.0015
SIEVE OPENING, μ	297	149	105	74	63	44	37

Pressure drop (clean): the pressure differential across a clean filter unit including inlet and outlet ports under specified conditions of flow rate, temperature, pressure, and flow medium.

Contaminant capacity (replaces obsolescent term "dirt holding capacity"): the maximum weight of a contaminant with specified particulate distribution which can be added on the inlet side of a filter without exceeding a specified pressure drop.

Pressure drop at rated contaminant capacity: the pressure differential across a filter unit, including inlet and outlet ports after a specified weight of a contaminant having a specified distribution of particles has been added on the inlet side of a filter under specified conditions of flow rate, temperature, pressure, and flow medium.

Initial element contamination: The contaminant level in a new filter element prior to installation as measured by a specific test method such as ARP 599 and analyzed as described in ARP 598. ARP 849 describes inspection of filter elements for cleanliness.

Media migration: migration of any form of particulate contaminant indentifiable as filter material or the supporting structure.

Collapse pressure: the differential pressure across a filter which will collapse or distort the element in any manner in which the performance of the filter is degraded. This is a test to determine structural integrity of the element.

Initial bubble point: the air pressure in inches of water at which the first bubble appears in a liquid of known surface tension and temperature in which the element is wetted and pressurized with air. The bubble point is an indication of the maximum pore size of a filter medium. The filter element is immersed in the liquid to a known depth, and air pressure is gradually increased until the first bubble appears. The bubble point test procedure is presented in detail in Reference 315-1.

Filter media: the material or combination of materials which are used to remove solids from a fluid stream.

Filtration depth: the property of a filter medium to retain contaminants in more than one plane or layer. Filter media generally do not have uniform pore size, and for this reason are capable of filtering a portion of the particles smaller than the maximum pore size or absolute rating. As the depth of the medium is increased, the efficiency of the medium to retain smaller particles is improved. A detailed presentation of depth filtration performance is described in Reference 17-10.

5.10.3 Filtration and Separation Techniques

5.10.3.1 FILTRATION TECHNIQUES. Two types of filtration are described.

Surface Filtration. Surface filtration is accomplished by impingement and retention of solid contaminants on a matrix of pores or openings in a single plane or surface. Filtration occurs only at the one surface, and contaminants which are not stopped at this surface pass through the media with no further change in direction. Surface filtration is effective in collection of particles larger than the pore size, but ineffective in collection of fibers and particles smaller than the pore size. Contaminant capacity of surface filters is limited by the amount of surface area which can be provided within a given envelope. Examples of surface filtration media are single layer mesh screens, stacked washers, wound metal ribbon, and perforated sheet metal.

Depth Filtration. Depth filtration is accomplished by impingement and retention of solid contaminants in a matrix of pores in series or depth. Sand is a classic example of a depth filter, its filtering action being random absorption and entrapment. Filtration occurs not only at the surface but throughout the thickness of the media. Particles which pass one matrix are subjected to direction changes in a circuitous path. Depth filters are effective in collection of particles larger than the maximum opening and fibers, and will also collect a portion of contaminants smaller than the largest pore size, depending on the media type and thickness. Contaminant capacity of depth type media per square inch of surface area is large because the contaminants may be retained throughout the depth of the material as well as on the surface layer. Examples of depth filtration media are: multiple layers of mesh, wound wire cylinders, stacked paper discs, sintered granulated materials, multiple layers of cloth, compressed or matted organic or inorganic fibers, stacked etched sheet metal discs, open pore plastic foam materials, and stacked membranes.

5.10.3.2 SEPARATION TECHNIQUES. Six separation techniques are described. Water separation refers only to undissolved water.

Adsorptive Separation. Adsorptive separation is removal of contaminants from a fluid by the contaminant becoming attached to the surface of a media. Adsorbent separators differ from filters in that they are capable of separating gum, varnishes, and other contaminants in addition to particles and fibers. Some of the common adsorbent materials are silica gel, Fuller's earth, alumina, and bauxite. These materials have several hundred thousand square feet of surface area per pound of adsorbent. Adsorbent separators are designed to provide a deep bed of adsorbent material and to permit a low velocity through the bed to prevent channeling of fluid. The contaminant is attached to the exposed surface area of the adsorbent material. Some examples of adsorbent separation applications are removal of gums and varnish from petroleum products, removal of oil from compressed gas streams, and removal of carbon dioxide crystals from liquid oxygen or nitrogen.

Magnetic Separation. Magnetic separation is the removal of ferromagnetic contaminants by magnetic attraction. One form of magnetic separator consists of a magnetized plug which is screwed into the storage reservoir of hydraulic or lubrication systems. Another form of magnetic separator consists of magnetized grids in the fluid stream which attract magnetic contaminants. Bar magnets have also been cast into sintered metal filters to enhance their ability to collect magnetic contaminants. Magnetics separators are used in oil reservoirs of reciprocating machinery, hydraulic systems, crankcases, and other applications where magnetic contaminants are anticipated.

Centrifugal Separation. Centrifugal separation is accomplished by rotation of a fluid to separate material of different densities from the fluid. The centrifuge will separate insoluble contaminants which are heavier than the fluid; it will not separate two fluids of the same specific gravity. The centrifuge is useful in separation of water from petroleum base fuels such as the RP series or JP series propellants, and in separating water and high density contaminants from lubricating oils. The centrifuge is widely used in the process and petrochemical field, but has little application as an airborne device due to its large size, heavy weight, and power requirements.

Electrostatic Separation. Electrostatic separation is accomplished by passing fluid over electrostatically-charged plates which separate contaminants by means of electrostatic attraction. This type of separator consists of electrostatically-charged plates separated by a nonconductive collecting mass such as polyurethane foam, glass fibers, or other suitable material. A direct current potential is applied to alternate plates and the contaminants are collected on the collecting mass. The electrostatic separator is efficient in removing extremely small particles from clean fluids, to obtain a super-clean fluid. The separator will also separate water from petroleum products; however, if too much water is present, the separator will not operate after the fluid has become sufficiently conductive to cause electrical breakdown between plates. The relatively bulky and heavy electrical equipment required by this device limits the application of the separator to non-airborne systems.

Freeze Out Separation: Freeze out separation is accomplished by freezing contaminants in a process stream, then collecting the frozen contaminants on a cold coil surface or filter. This technique is used to remove water from compressed air, nitrogen, helium, and other gases. The gas is passed over refrigerated coils or surfaces, and the water freezes out on the cold surface. If the coils or the surfaces are cooled to a sufficiently low temperature, carbon dioxide, lubricating oil, and similar contaminants can be removed from a gaseous process stream. Freeze out separators are often used in pairs, so that one unit may be defrosted or heated to remove frozen contaminants from cryogenic fluids. For example, ice and crystalline carbon dioxide are frozen out in liquid air, liquid nitrogen, or liquid oxygen streams. The crystalline carbon dioxide and ice are removed from the cryogenic liquid by mechanical filtration.

Gravity Separation. Gravity separation of contaminants is accomplished by allowing the material of a different density to settle in a container. The technique is useful in separation of water from petrol[illegible]ducts and for separation of insoluble contami[illegible]ds. The techniques will remove large high [illegible]s rapidly; however, fine particles and sil[illegible] suspension for long periods of time. Gr[illegible]ost effective in flat horizontal t[illegible]h. For continuous gravity separ[illegible]d trays are placed in series [illegible] enters the first tray and [illegible] trays. Settling occurs in [illegible]y. The degree of gravity se[illegible]process is limited by agitation [illegible] the fluid as it [illegible]gh the series of trays.

5.10.4 Mechanical Filter Media

A number of factors must be considered when selecting the proper filter media for a given service. Table 5.10.4 is a mechanical filter media comparison chart based on the more important filter selection parameters. In the following Detailed Topics, mechanical filter media are described with respect to important characteristics, applications and limitations.

5.10.4.1 SQUARE MESH WIRE CLOTH. Square mesh wire cloth (Figure 5.10.4.1) is produced on a loom in the same manner as conventional cloth materials. The warp is the wire that runs the length of the cloth; the shute (or woof) runs perpendicular to the cloth, and is woven into the warp by means of a shuttle. The wire mesh cloth develops a square opening pattern and presents a straight-through opening. Square mesh wire cloth materials are not available in "absolute" ratings much finer than 40 microns. Twilled weaves are also available, which present a straight-through opening to fluid flow. Since there is no mechanical bond between points of intersection of the wire, the cloth will distort under high mechanical stress and vibration, and pore size will vary, allowing particles to pass. Initial cleanliness is difficult to achieve because contaminants are inadvertently woven into the cloth unless special techniques and clean room environments are employed. Contamination will be introduced during fabrication of the filter element when the cloth is welded or attached to the supporting structure.

Although the element may receive a vigorous ultrasonic cleaning, entrapped contaminants will continue to work loose to some extent throughout the life of the filter. This

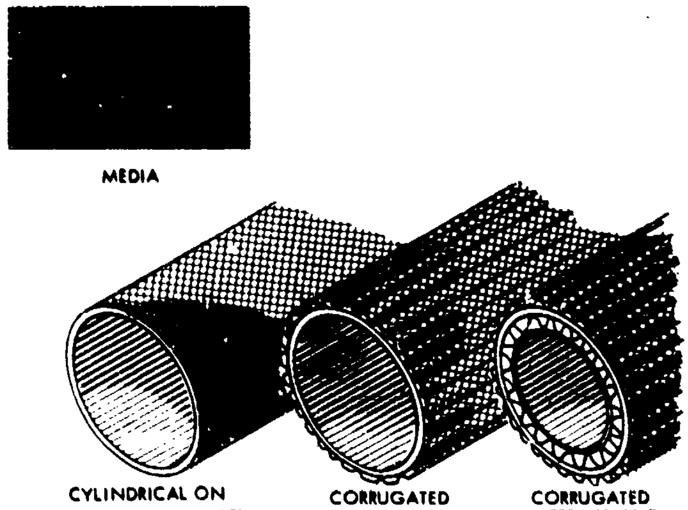


Figure 5.10.4.1. Woven Square Mesh Wire Cloth

type of filter is suitable for noncritical hydraulic, pneumatic, and propellant feed applications. The filter will pass fibers and random large particles.

5.10.4.2 WOUND WIRE MEDIA. Wound wire materials (Figure 5.10.4.2) are a form of thick cloth made by helically cross-winding wire on a mandrel. Wires are then sintered where they cross, and the cloth may be removed from the mandrel and slit into sheet, or may be retained and used in cylindrical form. Small cylindrical elements may be extremely rugged. Cleanliness is difficult to attain due to contaminants being wound into the cloth. Migration of media is comparable to other sintered cloth elements and will continue under dynamic flow, mechanical vibration, and stresses. The elements have depth, and are more effective in stopping fibers than single layer cloth filters. Dirt capacity per unit surface area is good. The elements are usually welded to a header or supporting frame, which introduces additional sources of built-in contamination. This type of filter is not suitable for critical hydraulic, pneumatic, and propellant feed applications.

5.10.4.3 SINTERED WIRE MESH. Both the square mesh wire cloth and wound wire media can be sintered. The square mesh cloth has the warp and woof wires sintered together where they cross. The sintering process fixes the pore size and reduces the possibility of changes in pore size under mechanical stress and vibration. The sintered wire mesh has the same general filtration capability as unsintered mesh and presents a straight-through path for contaminants. The sintering process, however, introduces another source of contamination in the form of diffusion (weld) slag. Initial cleanliness is more difficult to achieve than with unsintered cloth, and migration tends to be more severe. This type of filter is suitable for noncritical hydraulic, pneumatic, and propellant feed applications. The filter will pass fibers.

ISSUED: MARCH 1967
SUPERSEDES: MAY 1964

Table 5.10.4.

Filter Type	Filtration Type	Filtration Rating Range Available (Microns) Nominal (1)	Absolute	Ability for Absolute Cut-off Rating	Temperature Range, °F (2)	Par
Square mesh wire cloth or twilled Dutch single weave wire cloth	Surface		40 and up	Fair	Cryogenic to 1000	Go
Sintered square mesh wire cloth or sintered twilled Dutch single weave cloth	Surface		40 and up	Fair	Cryogenic to 1000	Go
Wound wire cloth	Depth	2 and up		Fair	Cryogenic to 1000	Go
Twilled Dutch double weave wire cloth	Surface		8 to 100	Good	Cryogenic to 1000	Go
Sintered porous metal	Depth	2 to 60	5 to 135	Fair	Cryogenic to 1000	Go
Composite sintered porous metal and mesh	Depth	2 to 60		Fair	Cryogenic to 1000	Go
Etched disc	Derth	0.5 to 75	1 to 150	Good	Cryogenic to 1000	Go
Edge (ribbon or stacked washer)	Surface	40 to 125 25 to 500		Fair	Cryogenic to 1000	Go
Pressed paper	Depth	5 to 100		Poor	Normal Ambient to 275	Po
Matted fibers	Depth	5 to 100 (felted) 10 and over (non-felted)		Poor	Normal Ambient to 275	Fa
Membranes	Surface	0.1 to 12		Good	Normal Ambient to 260	Go
Glass fibers	Depth	5 (matted) to 100		Poor	-100 to 1000	Go
Fretted glass	Depth	0.1 to 125		Fair	-100 to 1000	Go
Sintered plastic	Depth	3 to 15		Fair	Normal Ambient to 300	Go
Fired porcelain	Depth	0.2 to 25		Fair	2000°F maximum	Go
Bonded carbon	Depth	10 to 70		Fair	650°F maximum	Go
Bonded stone	Depth	5 to 100		Fair	2500°F maximum	Go

(1) Nominal rating is shown only for reference when absolute ratings are not commonly employed for the medium.

(2) Temperature ranges for metal elements are based on 300 series stainless steel, monel and Hastelloy C. (Silver and bronze elements have a maximum temperature of 400°F, and Inconel elements have a maximum temperature of 1200°F. Carbon steel elements have a minimum temperature of -100°F.)

(3) Based on comparable filtration ratings and identical external envelope sizes.

(4) Based on comparable fil

(5) Mechanical strength of supported.

(6) Initial cleanliness of the and clean manufacturing during the manufacturin of wear particles during

ISSUED: MARCH 1967
SUPERSEDES: OCTOBER 1965

Table 5.10.4. Mechanical Filter Comparison Chart

Temperature Range, °F (2)	Filtering Ability: Particles	Filtering Ability: Fibers	Clean-Ability After Service	Mechanical Strength	Initial Pressure Drop Per Unit Envelope (3)	Contaminant (Dirt Holding) Capacity Per Unit Envelope (3)	Absence of Media Migration
Cryogenic to 1000	Good	Poor	Fair	Good (5)	Low	Good	Fair-Good
Cryogenic to 1000	Good	Poor	Fair	Good (5)	Low	Good	Fair
Cryogenic to 1000	Good	Good	Fair	Good	Low	Fair	Fair
Cryogenic to 1000	Good	Good	Fair	Good (5)	Low	Good	Fair-Good
Cryogenic to 1000	Good	Good	Fair	Fair	High	Fair	Poor
Cryogenic to 1000	Good	Good	Fair	Fair	High	Fair	Poor
Cryogenic to 1000	Good	Good	Good	Good	Medium	Fair	Good
Cryogenic to 1000	Good	Poor	Good	Good	Medium	Poor	Good
Normal Ambient to 275	Poor	Fair	Nil	Poor	Medium	Fair	Poor
Normal Ambient to 275	Fair	Good	Nil	Poor	Medium	Good	Poor
Normal Ambient to 260	Good	Good	Nil	Poor	High	Poor	Fair-Good
-100 to 1000	Good	Good	Poor	Poor	Medium	Good	Poor
-100 to 1000	Good	Good	Fair	Fair	High	Fair	Poor
Normal Ambient to 300	Good	Good	Fair	Fair	High	Fair	Poor
2000°F maximum	Good	Good	Fair	Fair	High	Fair	Poor
650°F maximum	Good	Good	Fair	Fair	High	Fair	Poor
2500°F maximum	Good	Good	Fair	Fair	High	Fair	Poor

(4) Based on comparable filtration rating and service life performance requirements.

(5) Mechanical strength of wire cloth filters can be good if the wire cloth is adequately supported.

(6) Initial cleanliness of these filters can be very good, but only if in-process cleaning and clean manufacturing techniques in clean room environments are employed during the manufacturing cycle. This includes careful attention to the generation of wear particles during fabrication and assembly operations.

B

Mechanical Strength	Initial Pressure Drop Per Unit Envelope (3)	Contaminant (Dirt Holding) Capacity Per Unit Envelope (3)	Absence of Media Migration	Initial Cleanliness	Shock and Vibration Resistance	Weight (4)
Good (5)	Low	Good	Fair-Good	Fair-Good (6)	Fair	Low
Good (5)	Low	Good	Fair	Fair	Fair	Low
Good	Low	Fair	Fair	Fair	Good	Medium
Good (5)	Low	Good	Fair-Good	Fair-Good (6)	Fair	Low
Fair	High	Fair	Poor	Poor	Fair	Medium
Fair	High	Fair	Poor	Poor	Poor	Medium
Good	Medium	Fair	Good	Fair-Good (6)	Good	High
Good	Medium	Poor	Good	Fair-Good (6)	Good	High
Poor	Medium	Fair	Poor	Poor	Poor	Low
Poor	Medium	Good	Poor	Poor	Poor	Low
Poor	High	Poor	Fair-Good	Good	Poor	Low
Poor	Medium	Good	Poor	Poor	Fair	Low
Fair	High	Fair	Poor	Poor	Poor	Medium
Fair	High	Fair	Poor	Poor	Fair	Low
Fair	High	Fair	Poor	Poor	Poor	Medium
Fair	High	Fair	Poor	Poor	Poor	Medium
Fair	High	Fair	Poor	Poor	Poor	Medium

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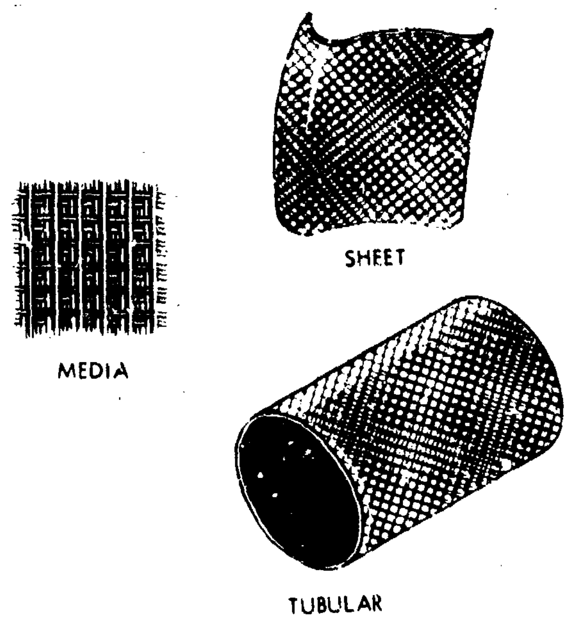

Figure 5.10.4.2. Woundwire Media

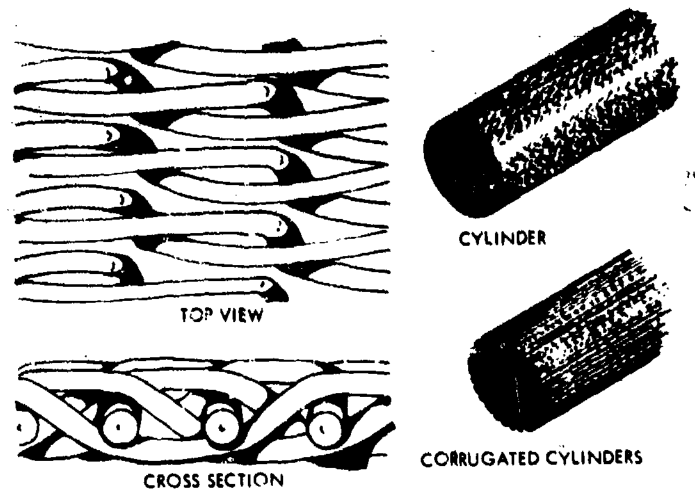

Figure 5.10.4.4. Dutch Twill Weave Wire Cloth

5.10.4.4 TWILLED DUTCH DOUBLE WEAVE WIRE CLOTH. Twilled Dutch double weave wire cloth (Figure 5.10.4.4) is a woven cloth whose weave presents a more intricate flow path than square mesh patterns. For this reason the material has somewhat better ability to stop fibers. This medium cannot be classed as a depth filter, however, even though its fiber removal performance will be considerably better than square weave cloth. Initial cleanliness will be comparable to square mesh elements, and migration characteristics will also be comparable to the square mesh elements. Sintering twilled Dutch double weave cloth will increase migration of the element. Twilled Dutch double weave cloth is generally heavier than square mesh, and will generally require less structural reinforcement than square mesh cloth. This type of filter is suitable for critical pneumatic, hydraulic, and propellant feed applications if manufactured with controlled in-process cleaning and non-dirt generating production techniques within a class 100,000 clean room (see Table 5.10.4.4).

5.10.4.5 STACKED WASHER OR RIBBON EDGE FILTER. Edge filters (Figure 5.10.4.5) consist of stacked washers or ribbon wound to form a hollow cylinder. The filtration surface is the outside diameter of the cylinder. Each stack of discs, or ribbon, is supported mechanically by a cage which compresses the stack. The pore size of such elements is fixed and will not vary with flow, mechanical stress, vibration, or temperature, if properly supported. Initial cleanliness of the element can be attained to a level determined by the ability to clean all individual washers or ribbons and to assemble them without particle generation within a controlled clean room. Migration of media will depend upon the method used to stamp, cut, or fabricate the discs or ribbon. These filters are surface filters and will pass fibers of larger dimensions than the absolute pore size. Dirt capacity is poor since the element has no depth. Depending upon the construction, this type of filter may be recleaned by back flushing, if the stack can be loosened.

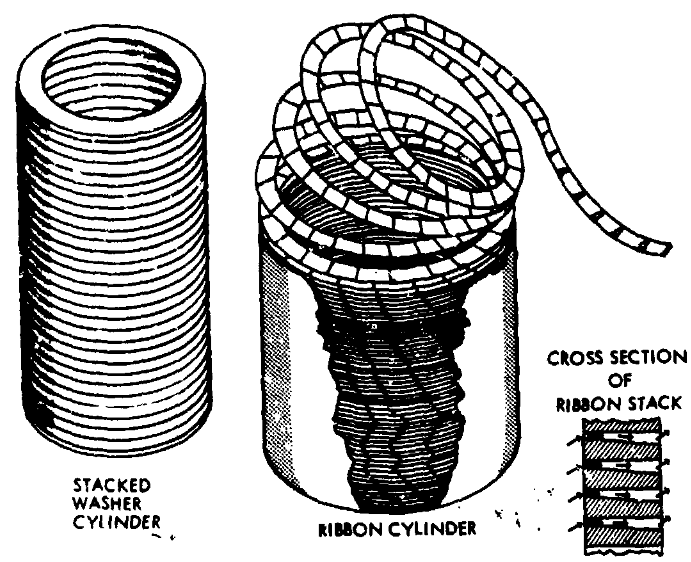

Figure 5.10.4.5. Edge Filters

It is suitable for non-critical hydraulic, pneumatic, and propellant feed applications. The element will pass fibers. The size and weight may be excessive for airborne propellant feed applications.

5.10.4.6 SINTERED METAL. There are a great many types of filter elements fabricated by sintering beads, metal powder, and other metallic granular material to produce a filter element with depth characteristics (Figure 5.10.4.6).

Table 5.10.4.4. Standard Grades of Twilled Dutch Double Weave Wire Cloth

(Courtesy of Wintec Corporation, a division of the Cemarc Corporation, Inglewood, California)

Grade	Nominal Mesh Count Warp x Shute	Absolute Micron Rating	Wire Diameter (microns) Warp	Wire Diameter (microns) Shute	Nominal Thickness (Inches)	Nominal Weight (lb/ft^2)
01	24 x 230	100.0	380	220	0.0350	-
02	30 x 250	75.0	250	200	0.0280	0.719
03	50 x 250	50.0	250	200	0.0280	0.633
04	80 x 745	35.0	100	76	0.0110	0.241
28	165 x 1350	20.0	70	40	0.0057	0.159
41	200 x 1350	15.0	70	40	0.0057	0.161
51	250 x 1350	12.5	55	40	0.0051	0.160
67	325 x 2100	10.0	38	25	0.0033	0.100
75	375 x 2150	7.5	34	24.5	0.0031	0.095

ALLOWABLE TOLERANCES

Warp Wire Mesh Count		Shute Wire Mesh Count		Wire Diameter	
Warp Wire Mesh Count	Number of Tolerance Wires per Inch	Shute Wire Mesh Count	Number of Tolerance Wires per Inch	Wire Diameter (microns)	Diameter Tolerance (microns)
Up to 49	± 0.3	Up to 250	± 20	Up to 25	± 1/2
50 to 74	± 0.5	251 to 1000	± 25	26 to 99	± 1
75 to 99	± 1.1	1001 to 2000	± 30	100 to 199	± 2
100 to 400	± 2.0	2001 to 3000	± 50	200 to 400	± 3

AVAILABLE MATERIALS

Finest Available Grade	Finest Wire Size (microns)	Type of Material
Grade 75 - 375 x 2150	18	AISI 304, 304L SS
Grade 41 - 200 x 1350	25	AISI 316, Monel, Nickel
Grade 04 - 80 x 745	50	AISI 321, 347 SS

The elements are produced in a wide variety of configurations including discs, cylinders, washers, corrugated tubes, etc. The degree of bonding of this material may vary from poor to good, depending upon the quality control during manufacture, and structural strength may vary from fragile to rugged. Initial cleanliness of this type of element is difficult to achieve and migration of element material including flakes, beads, or large granules is possible. The ability to withstand thermal shock is very critical with this material, and mechanical shock and vibration resistance of these materials is poor. Elements of this type may be plated by various methods to reduce the migration properties of these elements; however, these characteristics cannot be eliminated. This type of element may be suitable for removal of gross contamination of fluids in non-critical airborne hydraulic, pneumatic, or propellant feed applications.

5.10.4.7 COMPOSITE SINTERED METAL AND WIRE CLOTH. In order to reduce the migration characteristics of sintered metal type elements, filter elements are fabricated from a combination of sintered metal granules bonded to wire mesh, (Figure 5.10.4.7). This combination was developed to obtain the desirable depth characteristics of the sintered granule, and to reduce the possibility of migration and element collapse by providing an integral wire cloth backup. Laboratory tests have demonstrated that this construction is very poor from the standpoint of initial cleanliness, and migration of element material is severe. The interaction of the cloth with the sintered granular material will release large numbers of particles under mechanical vibration and operating stresses. This medium should not be used for critical airborne pneumatic, hydraulic, or propellant feed applications; however, it may be a good mate-

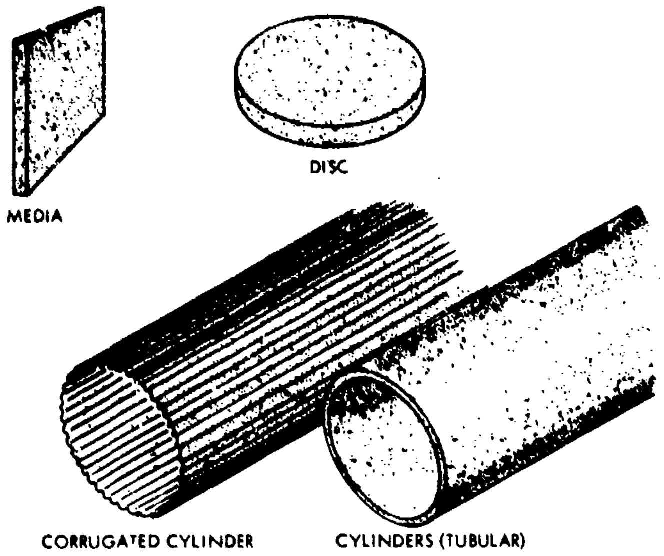

Figure 5.10.4.6. Sintered Porous Metal Filter

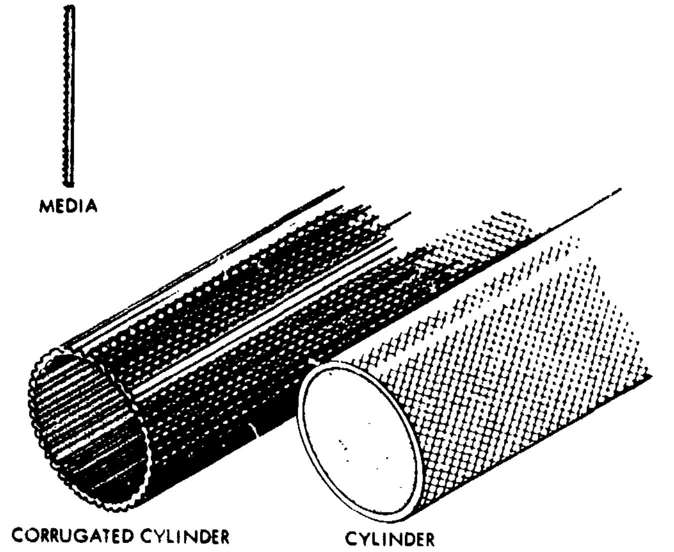

Figure 5.10.4.7. Composite Sintered Porous Metal with Wire Mesh

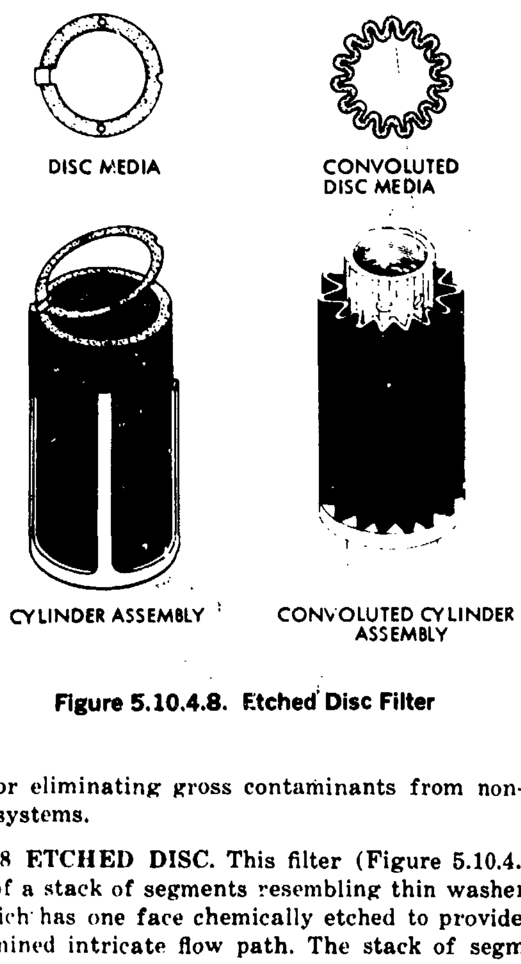

Figure 5.10.4.8. Etched Disc Filter

rial for eliminating gross contaminants from non-critical fixed systems.

5.10.4.8 ETCHED DISC. This filter (Figure 5.10.4.8) consists of a stack of segments resembling thin washers, each of which has one face chemically etched to provide a predetermined intricate flow path. The stack of segments is held rigidly by a supporting cage, and is tightly compressed. The initial cleanliness is attained by cleaning after assembly but before compression of the stack. Since there are no welds, wires, chips, or machined surfaces on the element, migration of media is low. The depth property of the element is achieved by requiring the contaminants to cross the surface of each segment through a tortuous path. Contaminant capacity is fair to good, depending primarily upon the specific disc configuration. The element may be cleaned by releasing the compression on the segments and backflushing. This type of element is suitable for critical hydraulic, pneumatic, and propellant feed applications. Although size, weight, and initial pressure drop are relatively high for airborne propellant feed applications, etched disc filters have been used when other types were unable to meet stringent initial cleanliness or low micron rating requirements. Recent advances in wire cloth filter technology have broadened the choice of filter types for these critical applications, but the etched disc filter remains the only all-metal filter for filtration below 8 microns. In addition, the relatively low area-to-mass ratio of some disc configurations makes the etched disc filter desirable for use with strong oxidizers such as liquid fluorine.

5.10.4.9 PRESSED PAPER. Pressed paper elements (Figure 5.10.4.9) are available in many different configurations including cylinders, corrugated cylinders, discs, tubes, and others. Since paper is not a uniform material, wide variation may exist in the pore size, bonding, and physical properties of the material. The pore size may also vary appreciably under mechanical stress and vibration. Filters of this type will shred and migrate fibers and particles. Initial cleanliness is difficult to attain. This type of element may be suitable for removal of gross contamination from non-

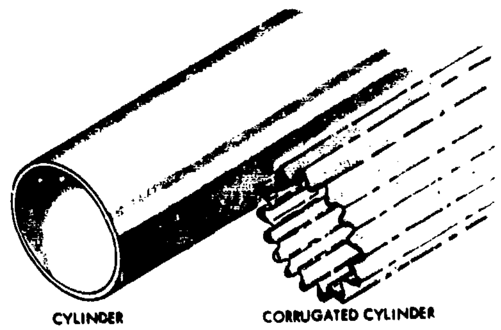

Figure 5.10.4.9. Pressed Paper or Fiber Filter

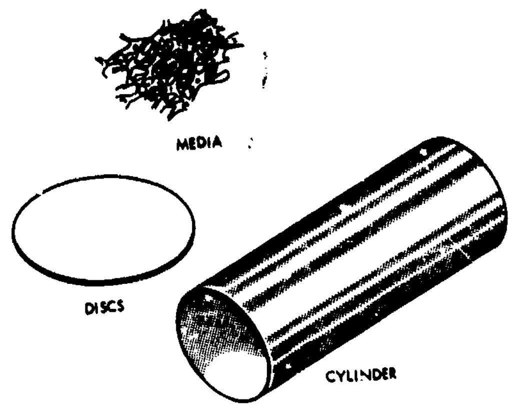

Figure 5.10.4.11. Membrane Filter

critical fixed systems, but it should not be used in critical airborne hydraulic, pneumatic, or propellant feed systems.

5.10.4.10 **MATTED FIBERS.** The properties of matted fiber filter elements (Figure 5.10.4.9) are similar to those discussed under pressed paper filters. Matted fiber elements may have better contaminant capacity than the pressed paper elements due to the greater depth. This type of element may be suitable for removal of gross contamination from non-critical fixed systems, but it should not be used in critical airborne hydraulic, pneumatic, or propellant feed systems.

5.10.4.11 **MEMBRANE.** The membrane filter is a thin porous element containing precision capillary pores evenly distributed over the surface area to form a monolithic structure which is not subject to media migration into the effluent. Filter elements are available in both wafer and cylinder configurations. The elements are limited to low pressure differentials and cannot be cleaned, but must be discarded after use.

Minute porosity grades and ease of contaminant examination make the membrane filter an indispensible tool for laboratory analysis, fluid sampling, and contamination monitoring. Porosity grades are precisely controlled and range from 10 millimicrons to 10 microns. All particles larger than the pore size are screened from the fluid and retained on the filter surface where they can be examined microscopically.

Although other materials are used for membrane filters, cellulose ester is by far the most common. This material is compatible with water, dilute acids and alkalies, and many hydraulic fluids.

5.10.5 Filter Cases

5.10.5.1 **CONFIGURATION.** A series of standard filter elements and filter cases have been developed to cover hydraulic systems for aircraft. These standards cover the range of approximately one-half gpm to 30 gpm, and are covered by MIL-F-5504 and MIL-F-8815. For special applications, even within these flow ranges, there are almost unlimited designs and configurations of filter cases and elements. Filter cavities are also machined in the body of a component such as a servocylinder, pump, or regulator. There is very little standardization within the filter industry except for these MIL specification units. For this reason, procurement specifications for filters require considerable detail concerning application. A series of standard designs is shown in Figure 5.10.5.1.

5.10.5.2 **ACCESSIBILITY.** The degree of accessibility required often determines the general type of filter case. The in-line filter case is normally the most inaccessible type of filter, and usually requires that the case be removed from the piping system before a filter element may be removed.

The T-type or pot-type are configured such that the element can be removed by removing the filter pot and the internal element. The Y-type filter is usually designed in such a manner that the element may be removed through one leg of the Y by removal of a blank flange or cover plate. This configuration reduces the clearance required for removal of the element, and requires somewhat less space initially. A wide variety of combinations and variations of these basic types are available, and may be specified with flanged connections, threaded connections, weld fittings and others.

5.10.5.3 **INTERNAL SEALS.** The entire function of a filter can be voided by a poorly sealed element, and for this reason it is important that a proper seal be specified and checked by test. Seal operation is particularly important under dynamic vibration environment and flow conditions. An improperly designed seal will not only leak and by-pass contaminants around the element, but may actually wear the seal, producing shredded particles and fibers. In critical propellant feed applications, internal seals should be avoided whenever possible for reasons of material compatibility and particle generation during assembly. (See Sub-Section 6.3 for static seal information.)

5.10.5.4 **INTERNAL PARTS.** There is usually a requirement to seal the element at the interface with the case, and often supporting cages, bolts, nuts, O-rings and other de-

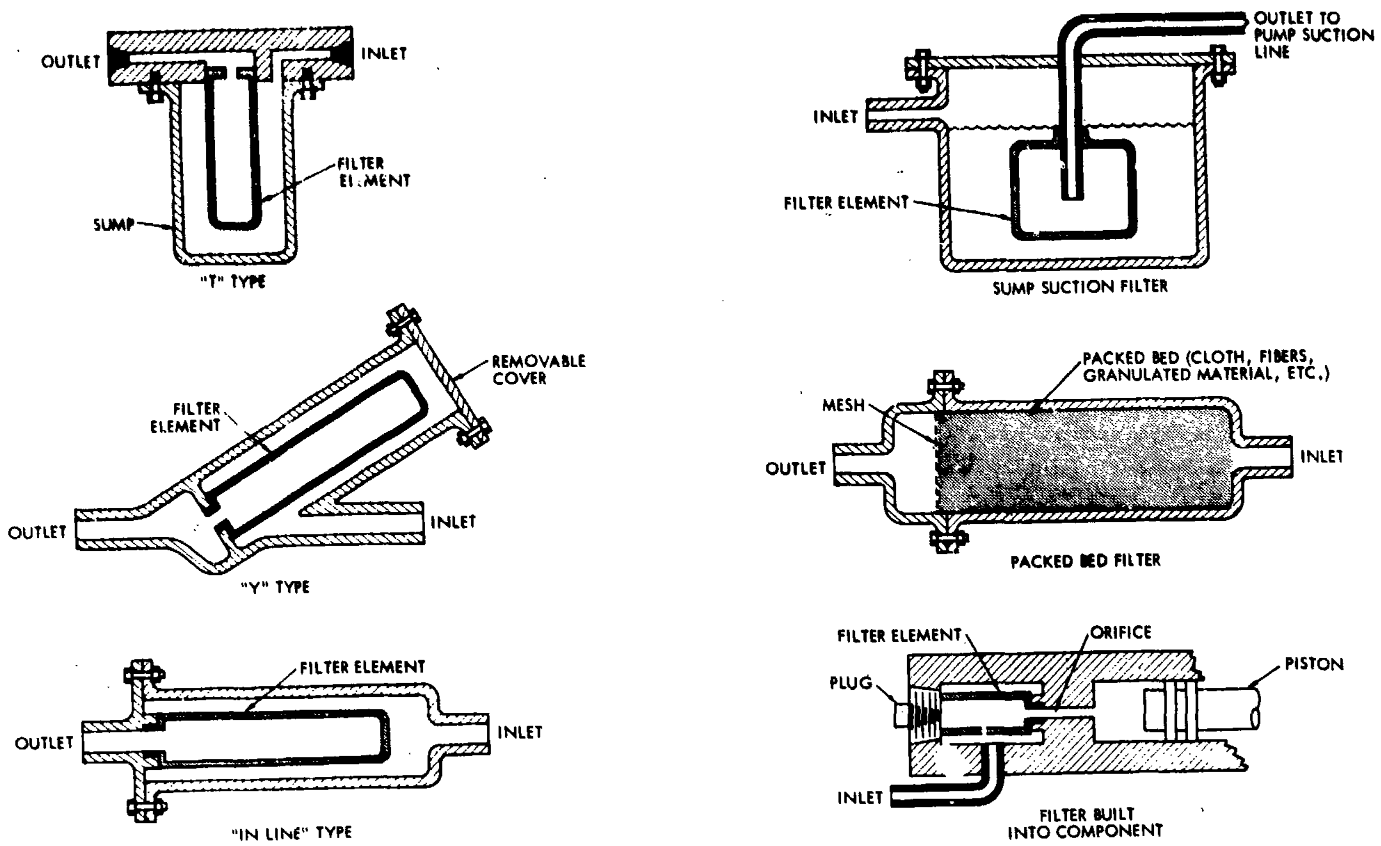

Figure 5.10.5.1. Filter Case Configurations

vices satisfy this requirement. Many systems have been damaged by parts of the filter case or structure migrating downstream. This problem can readily be eliminated by proper design of seals or locking devices, and by arranging removable parts so that they cannot accidentally be left inside the case.

5.10.5.5 DIFFERENTIAL PRESSURE INDICATORS. Many filters are designed with differential pressure indication devices to indicate abnormally high differential across the element. In many systems the device is ideal; however, in systems in which gas and liquid flow intermittently, the indicator will often show a high differential pressure when the filter is clean. (Two-phase flow often occurs in cryogenic systems.) Surges in systems will also give false indications under certain circumstances. Dial indicating differential pressure gages are used in some systems. They have the advantage of a continuous reading device, and the disadvantage of requiring that the indicator be monitored. The integral pressure differential indicator is more often applied to manned aircraft systems which run several thousand hours and gradually build up contaminant levels. The device has limited application on rocket and missile systems which operate for a very short duration, and do not build up contamination appreciably during the operating time.

5.10.5.6 BY-PASS VALVES. Many filters in lubrication and hydraulic systems have a built-in by-pass valve which will open and by-pass the filter when the differential pressure builds up to a pre-determined value. This device is incorporated because it is often considered better to continue to keep a system running with contaminated fluid and no filtration, than to have the system fail completely by lack of fluid when a filter becomes plugged. The device has limited application on rocket and missile systems which operate for a very brief duration and do not build up contamination appreciably during the operating time. The by-pass valve also complicates the filter case, is difficult to clean, and serves as a trap for contaminants.

5.10.6 Filter Cleaning

5.10.6.1 BACKFLUSHING. A common method of cleaning filters is to reverse the flow through the element, which flushes the contaminant from the inlet side of the filter. This method will remove the loose contaminants; however, it will not remove contaminants which are tightly wedged within the filter media. Backflushing alone will seldom restore a filter element to its initial cleanliness status.

5.10.6.2 ULTRASONIC CLEANING. Ultrasonic cleaning is accomplished by immersing a fluid element in a solvent solution and applying sonic energy to the system through transducers mounted on or within the tank. The sonic energy will produce cavitation on the surface of elements immersed in the solution which will loosen contaminants on the filter element. The effectiveness of the cleaning process depends on thorough flushing or rinsing through the element between ultrasonic cycles. Flow while in the ultrasonic environment is relatively ineffective due to decreased cavitation. The successive application of ultrasonic cleaning treatments to new or contaminated elements will produce a decreasing downstream particle count until a minimum level is reached. Beyond this point subsequent application of ultrasonics will not usually produce significant reduction in particle count. Procedures for determining the cleanliness level of filter elements are outlined in SAE Aeronautical Recommended Practice (ARP) 599.

5.10.6.3 FLUSHING. Flushing a filter element in the normal flow direction after ultrasonic cleaning or backflushing will usually dislodge contaminants which have been loosened during cleaning and will improve the cleanliness level of the filter element. Contaminant level tests are usually performed during the final flush in the normal flow direction.

5.10.6.4 PURGING. Purging an element with pre-filtered dry gas (usually air or nitrogen) will dry the element and remove residual solvent from previous cleaning steps. Purging an element, even in combination with mechanical vibration, is usually less effective as a cleaning method than ultrasonic cleaning and backflushing.

5.10.6.5 INITIAL CLEANING. Initial cleaning of new filter elements is necessary because contaminants are introduced during manufacture and assembly of the media into the filter element and case.

A combination of ultrasonic cleaning, flushing, and backflushing will produce a decreasing downstream particle count until a minimum level is reached. Initial cleanliness level of stainless steel cloth elements may be improved by acid passivation after assembly or by annealing the assembly in a hydrogen atmosphere. These two processes reduce the oxides on the element and may produce a significant improvement in cleanliness level attainable during subsequent ultrasonic cleaning and flushing.

5.10.7 Filter Requirements by System Type

5.10.7.1 HYDRAULIC. Airborne hydraulic systems usually contain orifices, servo valves, metering devices, and other parts which have extremely small clearances and openings. As a result, these systems are often extremely sensitive to contamination. For these reasons, the proper selection and application of filters for these systems is extremely critical. A typical control servo valve may have a nozzle of 800 microns diameter (0.032 inch), clearance between nozzle and flapper of 15 microns (0.00059 inch), and orifices of 125 microns (0.005 inch) (Reference 40-1). To prevent plugging or interference between nozzle and flapper, filtration to the 15 micron level may be required. To prevent interference between spool and bushing, which may cause sluggish spool operation, filtration to the 2.5 micron level may be required. A system containing components with clearances of this type may also require filtration to remove accumulation of large numbers of particles smaller than the maximum permissible particle. Removal of these smaller particles, or silt, may be required to reduce wear on the moving parts, reduce hysteresis, and prevent degradation of the response characteristics.

The selection of filters for a hydraulic system may be accomplished by the following steps:

a) Determine the maximum particle or fiber size the critical system components can tolerate. This may be determined by an analysis of system component clearances, orifice sizes, and similar design factors.

b) Review the system components to determine which items will generate contamination. (Tests have demonstrated that valve stem packing, pump seals and packing, valve seats, rubber and organic hoses, gaskets, and similar components will produce contamination within an initially clean system.)

c) Provide filtration at the location or locations in each system which will protect the critical components from contamination from outside sources as well as self-generated contaminants. Keep the number of joints between filter and critical component to a minimum.

d) Specify a filter or system of filters which will satisfy the system requirements. Particular emphasis should be placed on initial cleanliness of the filter, because tests have indicated that a so-called clean element may be a primary source of contamination.

e) Specify a filter or system of filters which will not allow media migration into the system. (Many systems have failed because filter media migrated or disintegrated.) Specify and require tests to assure that the filter purchased will perform under the actual shock, pulsation, vibration, and other environments to which it will be subjected.

f) Make certain that the filter has adequate service life, i.e., an envelope large enough to handle the required dirt capacity at the desired micron rating.

5.10.7.2 PNEUMATIC. Pneumatic systems usually contain valves which must reseat during and after functioning, as well as orifices and close tolerance clearances. As a result, many pneumatic systems are sensitive to contamination, and must be provided with filters.

The selection of filters for a pneumatic system may be accomplished by the following steps:

a) Determine the maximum particle or fiber size the critical system components can tolerate. This may be determined by an analysis of system component clearances and orifice sizes, or by tests on similar components.

b) Provide filtration at the location or locations in each system which will protect the critical components. Typical filter applications are inlet of pressure regulators, inlet of pressure reducing valve, inlet to pilot valve, and outlet of gas storage tank. (See (c) above under "Hydraulic.")

c) Specify the surge conditions anticipated for the system, and require that the filter be tested to withstand the specified surges. If system surges cannot be predicted or simulated in test, structural strength may be attained by requiring the differential collapse pressure to equal normal operating pressure.

d) Specify the initial cleanliness level required for the filter element. The high velocities encountered in most pneumatic systems will shake loose these contaminants trapped in the filter element.

e) Specify media migration requirements for the filter elements. The extremely high velocities encountered in most pneumatic systems will cause migration of a poorly bonded or weak filter element. Require that the elements be tested for media migrations.

f) Specify low temperature requirements for the filter if it is in a system which may experience low temperature due to expansion of gas or handling gas at cryogenic temperatures. A filter which performs adequately at room temperature may not be satisfactory at lower temperatures, due to seal embrittlement or differential expansion between the element and the case, which may permit leakage past the element seal and by-passing of contaminants.

g) See (f) above under "Hydraulic"

5.10.7.3 PROPELLANT FEED. Airborne propellant feed systems usually contain pumps, valves, orifices, nozzles, and other components which may be sensitive to particles and fibers. The selection of filters for this type of system should be based on the maximum size particle which the engine propellant system can digest. This value is often difficult to determine by theoretical means; however, actual measurements of contaminant levels on test stands for similar engines are often useful in making this determination.

Propellant feed filters are normally installed close to the engine, and are subject to extreme vibration, noise, heat, and often accommodate very high flow rates. The predicted environment for these filters should be thoroughly specified, and tests should be performed to verify that the filter will meet the requirements.

Another function of the propellant feed filter is to catch parts of other equipment which may have worked loose or failed before they can do damage to critical engine components. In this function, the ruggedness of the element plays an important part.

FILTER SPECIFICATION REQUIREMENTS*

1. General Configuration, Size, Shape and Weight Limitations
2. Operating Fluid
3. Materials of Construction
   a) Metals
   b) Seals
   c) Filter media
4. Port Size Inlet and Connection Type
5. Port Size Outlet and Connection Type
6. Identification
   a) Name of unit
   b) Pressure rating
   c) Absolute filtration rating
   d) Maximum allowable differential pressure
   e) Procurement specification number
   f) Manufacturer's part number, case, and element
   g) Manufacturer's serial number
   h) Contractor's order number
   i) Manufacturer's name or trademark
   j) Flow arrow
7. Packaging Requirements
8. Interchangeability
9. Acceleration (vibration)
10. Temperature Range
11. Internal Parts, Locking Requirements
12. Pressure
    a) Operating
    b) Proof
    c) Burst
13. Rated Flow of Operating Fluid
14. Pressure Drop at Rated Flow, Clean
15. Pressure Drop at Rated Flow, and Rated Dirt Holding
16. Reverse Flow Requirement and Pressure Drop (if applicable)
17. Filter Rating, absolute
18. Filter rating, nominal (preferably deleted)
19. Dirt Holding Capacity
    a) Weight of contaminant to be added and add rate

*For additional details, refer to Air Force Ballistic System BSD-TDR-63-15

b) Maximum pressure drop increase allowed during contaminant addition

c) Type of contaminant, particles or beads

20. Element Collapse Pressure Differential

21. External Leakage

22. Internal Leakage Past Element Seal and Relief Valve (if applicable)

23. Duty Cycle, Including Surge Requirements

24. Initial Cleanliness Requirement

25. Media Migration

26. Relief or By-Pass Requirements

27. Environmental Requirements, Including Corrosive Conditions Due To Propellants, Sand, Salt Atmosphere

28. Quality Assurance Provisions and Test Requirements

a) Initial evaluation
   1) Examination of product
   2) Examination and test for initial contamination level
   3) Media and foreign particle migration test
   4) Proof pressure and external leakage test
   5) Bubble test
   6) Pressure drop test
   7) Dirt holding capacity test
   8) Element collapse test
   9) Re-cleanability test

b) Production test, each filter
   1) Examination of product
   2) Bubble test
   3) Proof pressure and external leakage test
   4) Examination and test for initial contamination level

REFERENCES

*References added March 1967

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1-26	6-162	49-2
1-105	6-164	62-47
1-107	6-195	68-32
6-2	6-206	70-6
6-3	6-207	155-2
6-23	17-10	279-1
6-26	19-48	315-1
6-27	19-140	445-1
6-28	23-25	447-1
6-38	23-26	447-2
6-50	23-51	447-3
6-56	40-1	450-3
6-211*		

5.11 QUICK-DISCONNECT COUPLINGS

5.11.1 Introduction

A quick-disconnect coupling consists of a male plug and a female socket witn a locking device for rapid connecting and disconnecting of fluid lines. The coupling or de-coupling operation normally requires less than one second time and is executed by a simple slide and/or rotary motion of the coupling ring.

In this Sub-Section the basic types of quick-disconnect couplings are described and compared, and cross-sectional drawings are presented to illustrate their construction and operation. In addition, factors to consider in choosing materials, seals, and locking devices are discussed.

5.11.2 When to Use Quick-Disconnect Couplings

Quick-disconnect couplings are used in hydraulic, pneumatic, storable propellant, and cryogenic systems, when one or more of the following conditions prevail:

- Ease of installation is required
- Rapid connecting and disconnecting is required
- Filling or draining of lines is to be eliminated
- The entry of contamination is to be prevented
- Fluid components are frequently removed for repair or replacement
- Breakaway connections are required
- Remote disconnecting of fluid lines is required

5.11.3 Quick-Disconnect Selection Factors

5.11.3.1 GENERAL. The selection of a quick-disconnect coupling involves determining the valve configuration best suited, the body materials and seals compatible with the system characteristics, and the locking device capable of withstanding the required number of operations, adaptable to remote disconnection, and operable in the required space envelope.

5.11.3.2 VALVE CONFIGURATIONS. The "Quick-Disconnect Coupling Selection Diagram" in Figure 5.11.3.2a is a flow diagram intended to serve as a guide in selecting valve configurations for fluid quick-disconnect coupling applications. Three parameters are considered in construction of the diagram in the order of their importance: self-sealing features, air inclusion and fluid loss, and pressure drop. Each parameter is discussed below:

Self-Sealing Requirements. Based on system requirements, self-sealing of both ends of the disconnected line may be mandatory, required for one end only, or may be unnecessary for either disconnected end. When the requirement is established, the field of coupling configurations is narrowed to a few types, differing principally in the valve arrangement (Figure 5.11.3.2a).

Air Inclusion and Fluid Loss During Connecting and Disconnecting. The air inclusion and fluid loss characteristics apply only to self-sealing, double valved couplings where a discrete volume of air is trapped within the coupling during connecting, and an equivalent volume of fluid is lost during disconnecting. This results in air being forced into the system during coupling and fluid being lost during coupling. Typical values for these characteristics for one-half inch tube size quick-disconnect couplings are presented in the "Fluid Quick-Disconnect Characteristic Chart" in Table 5.11.3.2. Additional fluid is lost in disconnecting due to leakage past the valves during closing. This quantity will depend upon such factors as fluid properties, fluid temperature, fluid pressure, valve closure rate, wear, and valve seal design. This additional leakage can be determined accurately by test, and must be added to the trapped volume to determine the total fluid loss.

Quick-Disconnect Coupling Pressure Drop. The pressure drop across a quick-disconnect coupling is determined by the shape of the flow path through the coupling, the fluid flow rate, and the fluid properties. The flow-path differs markedly between the various types of valve designs (Fig-

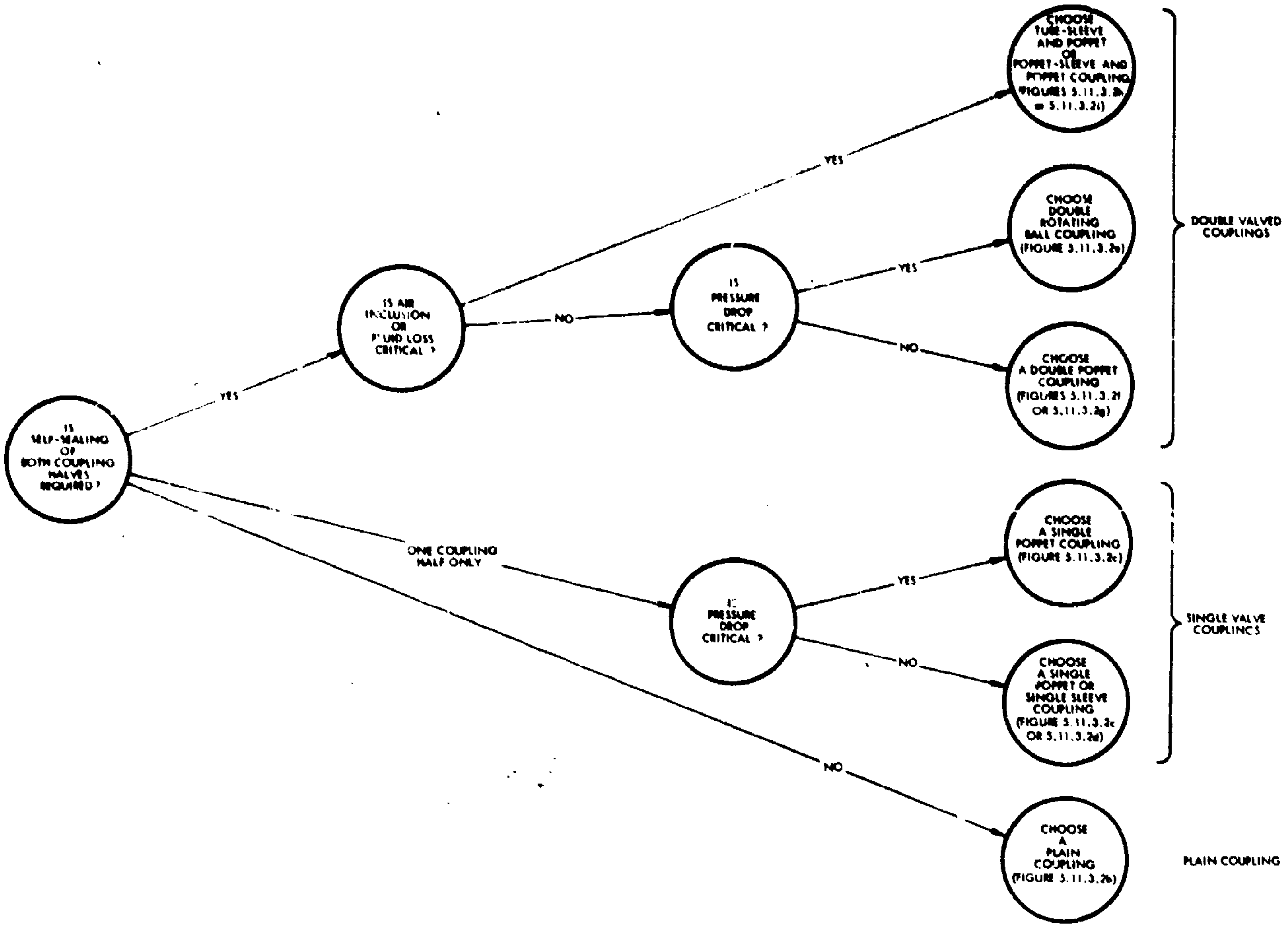


Figure 5.11.3.2a. Quick-Disconnect Couplings Selection Diagram

Table 5.11.3.2. Fluid Quick-Disconnect Comparison Chart

QUICK-DISCONNECT CONFIGURATION	FIGURE	SELF-SEALING: BOTH COUPLING HALVES	SELF-SEALING: ONE COUPLING HALF ONLY	TYPICAL SIZE RANGES AVAILABLE TUBING O.D. (in.): MINIMUM	MAXIMUM
Plain	5.11.3.2b			1/4	6
Single poppet	5.11.3.2c		X	1/4	6
Single sleeve	5.11.3.2d		X	1/4	3/4
Double rotating ball	5.11.3.2e	X		1/4	2
Double poppets (drilled)	5.11.3.2f	X		1/4	2
Double poppets (solid)	5.11.3.2g	X		1/4	6
Tube-sleeve and poppet	5.11.3.2h	X		1/4	1-1/4
Stem-sleeve and poppet	5.11.3.2i	X		3/16	2-1/2

QUICK-DISCONNECT CONFIGURATION	LOCKING ACTION: ROTARY	ROTARY AND PUSH-PULL	PUSH-PULL	TYPICAL CHARACTERISTICS FOR ONE-HALF INCH TUBE SIZE: PRESSURE DROP, psi AT 6 GPM MIL-H-5606 AT 60F	VOLUME TRAPPED DURING CONNECTING OR DISCONNECTING (cc)	WEIGHT (STEEL, 2000 PSI RATING) (lb)
Plain		X	X	0.6	NA*	0.5
Single poppet		X	X	3.4	NA*	0.6
Single sleeve			X	3.6	NA*	0.6
Double rotating ball	X			3.0	3.5	1.1
Double poppets (drilled)			X	3.3	5.8	1.7
Double poppets (solid)	X	X	X	4.8	3.3	0.8
Tube-sleeve and poppet			X	5.5	0.8	1.3
Stem-sleeve and poppet			X	6.0	0.03	1.2

*N.A: Not applicable

ures 5.11.3.2b through 5.11.3.2i), and also varies to some degree between different designs of the same valve type. The latter may be due to streamlining of poppets for reduced pressure drop, or reduction in flow area (for strengthening internal parts) causing increased pressure drop. For comparative purposes, typical pressure characteristics for one-half inch tube size couplings are presented in Table 5.11.3.2.

5.11.3.3 BODY MATERIALS AND SEALS. The selection of body materials and seals is dependent upon the following factors:

System Fluid and Temperatures. Materials commonly used for coupling bodies are steel, stainless steel, aluminum, and brass. Other body materials used for special application include plastics, bronze alloys, titanium, Monel, and copper. Seal materials commonly used in quick-disconnect couplings are the natural and synthetic rubbers, Teflon and Teflon products, and metals. Which of the above or other materials is chosen is based on fluid-material compatibility, a subject to be included in Section 12.0, "Materials."

System Pressure. The system pressure dictates the coupling's strength requirement, which is a function of the yield strength and thickness of the stressed materials. The material thicknesses in turn affect the coupling weight, pressure drop, and size. The material chosen is of such strength that the thicknesses of the stressed parts do not result in excessive weight, restricted flow causing excessive pressure drop, or excessive space. This is significant in that an unrealistic restriction on weight, size, or pressure drop will result in unwarranted compromising of the other factors and/or the body material.

5.11.3.4 LOCKING DEVICES. Locking devices have two functions to perform: to lock the plug and socket together, and to hold the coupling valves open. The strength of the locking device is determined by the type of locking materials and the size and number of locking balls, threads, fingers, pins, or other devices incorporated within the socket which mate with a locking groove or slot in the plug. In selecting a locking device, other factors which are to be considered are number of required operating cycles while pressurized and depressurized; the mode of locking and unlocking; forces required to connect and disconnect; and space allowed for actuation of the locking device. Consideration should also be given to remote disconnection using pneumatics, squibs, or lanyards.

5.11.4 Common Types of Quick-Disconnect Couplings

5.11.4.1 PLAIN QUICK-DISCONNECT COUPLINGS. The plain, quick-disconnect coupling is an open, straight-through flow coupling without shutoff valves in the plug or socket. A typical plain coupling is shown in Figure 5.11.3.2b, which utilizes a cam ring, ball and groove device for locking, and an O-ring for sealing. The coupling is connected by mating the plug and socket and rotating the cam ring, forcing the balls radially into the plug groove. This action draws the coupling halves together, compressing the O-ring to achieve an external seal. The coupling is unlocked and disconnected by rotating the cam ring in the opposite direction (allowing the balls to move from the plug groove into the ball cage) and pulling the coupling halves apart.

5.11.4.2 SINGLE VALVE QUICK-DISCONNECT COUPLINGS. Two types are described:

Single Poppet Quick-Disconnect Coupling. The single poppet, quick-disconnect coupling consists of an open plug and a valved socket (Figure 5.11.3.2c). The mating parts are pushed together, first forming an external seal between the plug and socket. Further travel of the plug forces the poppet valve open, and retraction of the actuating ring confines the balls in the groove, locking the coupling halves. Disconnecting is accomplished by retracting the actuating ring and pulling the coupling halves apart.

Single Sleeve Quick-Disconnect Coupling. The single sleeve, quick-disconnect coupling consists of a valved socket and plain plug, (Figure 5.11.3.2d). Connecting is accomplished by mating the plug and socket, retracting the coupling ring, and pushing the coupling halves together. The outer diameter of the plug forms a seal with an O-ring on the inner diameter of the socket as the plug slides into the socket and opens the sleeve valve. When the valve is fully opened, the plug ball groove is aligned with the locking balls which drop into the groove. The coupling ring is then released, locking the coupling. The coupling is disconnected by retracting the coupling ring and pulling the plug and socket apart.

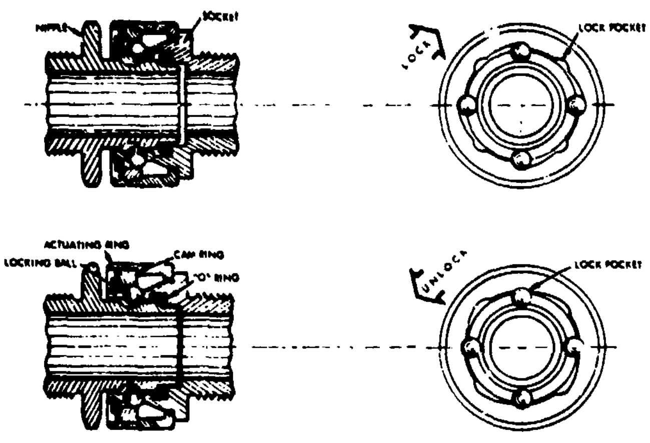


Figure 5.11.3.2b. Plain Quick-Disconnect Coupling
(Courtesy of Roylyn, Inc., Glendale, California)

5.11.4.3 DOUBLE VALVE QUICK-DISCONNECT COUPLINGS. Five types are discussed:

Double Rotating Ball Quick-Disconnect Coupling. The double rotating ball, quick-disconnect coupling consists of two coupling halves, each containing a ball valve, (Figure 5.11.3.2e). It is connected by mating the coupling halves and twisting the coupling ring. The coupling ring actuates a pinion gear, which rotates the ball valves to the open position. Disconnecting is accomplished by rotating the coupling ring, shutting the ball valves and unlocking the coupling.

Double Poppet (Drilled) Quick-Disconnect Coupling. The double poppet (drilled) quick-disconnect coupling consists of a plug and socket, each containing a drilled poppet valve, (Figure 5.11.3.2f). The mating parts are pushed together forming an external seal between the plug and socket. Further movement of the plug causes the poppet valves to butt against each other, opening the valves. When the poppet valves are in the full open position, the locking fingers snap into position forming a lock between the plug and socket. The coupling is disconnected by retracting the actuating ring and pulling the coupling halves apart.

Double Poppet (Solid) Quick-Disconnect Coupling. The double poppet (solid) quick-disconnect coupling (Figure 5.11.3.2g) consists of a plug and socket, each containing a solid poppet valve. The coupling halves are pushed together, first forming an external seal and next opening both poppet valves. Retraction of the coupling ring allows the balls to drop in the plug locking race. Release of the coupling ring confines the balls, locking the coupling halves together. Disconnecting is accomplished by retracting the coupling ring and pulling the coupling apart.

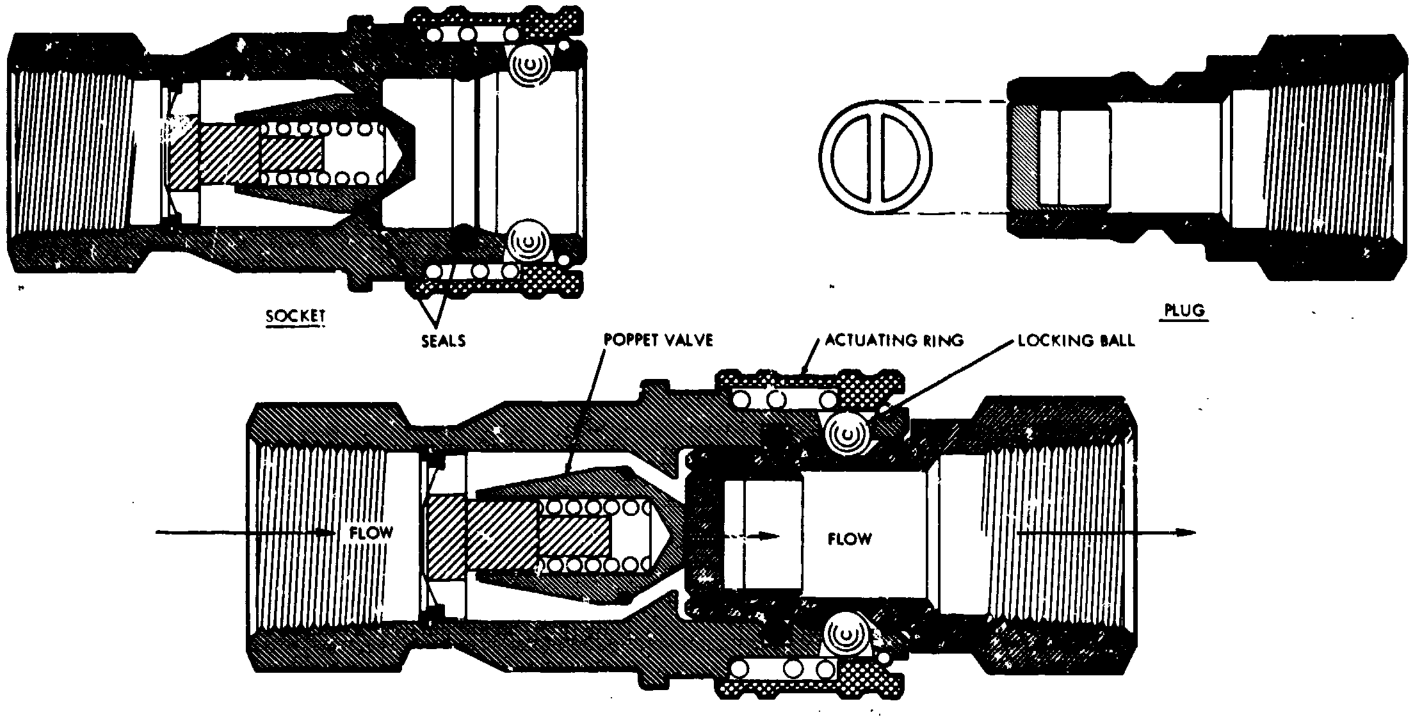


Figure 5.11.3.2c. Single Poppet Quick-Disconnect Coupling
(Courtesy of the Hansen Manufacturing Company, Cleveland, Ohio)

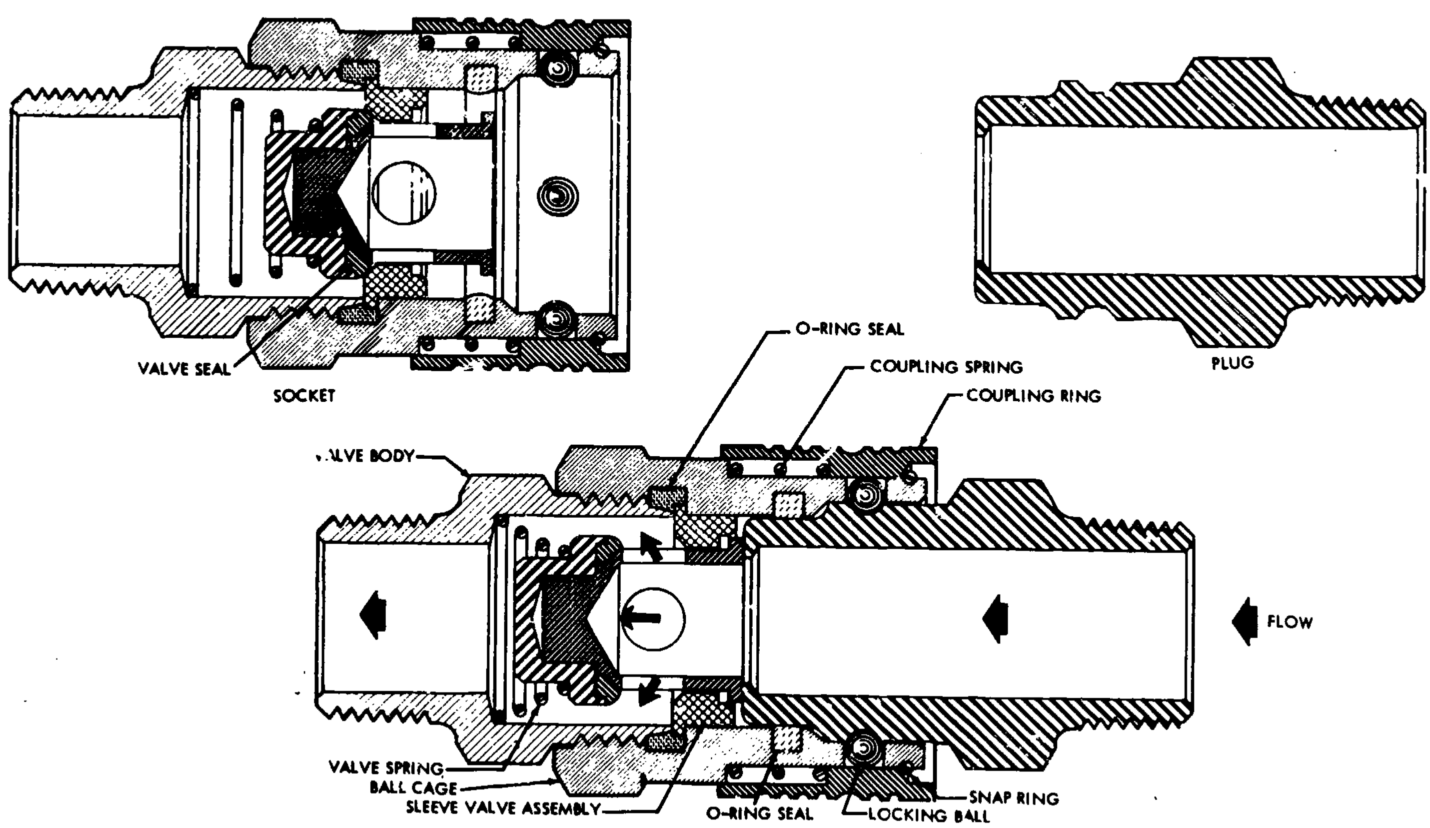


Figure 5.11.3.2d. Single Sleeve Quick-Disconnect Coupling
(Courtesy of Tru-Flate, Inc., Oakland, California)

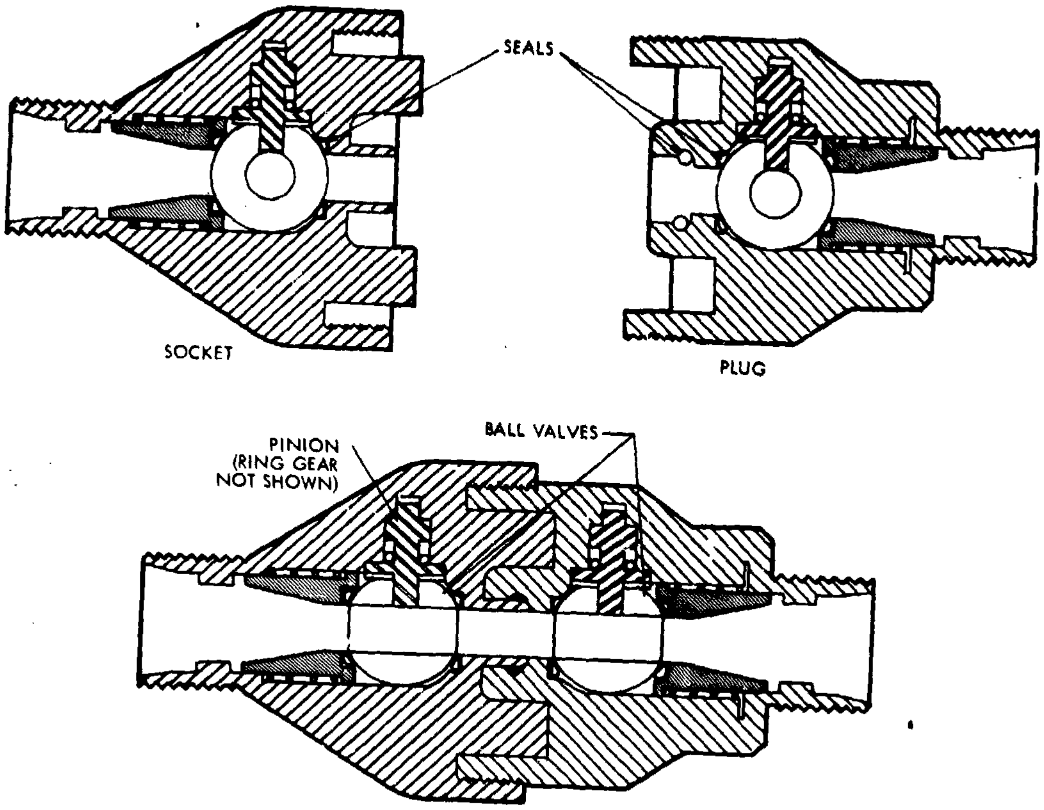
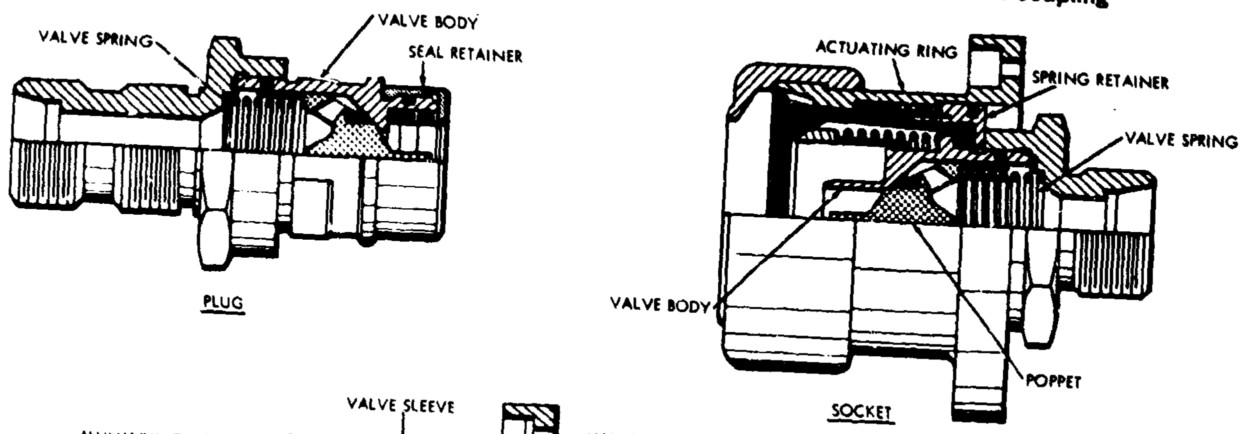


Figure 5.11.3.2e. Double Rotating Ball Quick-Disconnect Coupling



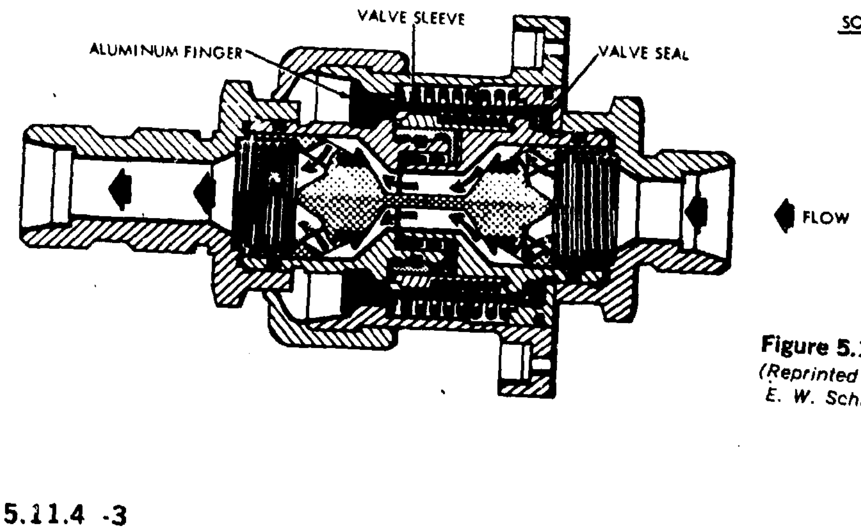


Figure 5.11.3.2f. Double Poppet Quick-Disconnect Coupling
(Reprinted from "Design News," April 24, 1961, vol. 16, no. 9. E. W. Schrader, Copyright 1961, Design News, Englewood, Colorado)

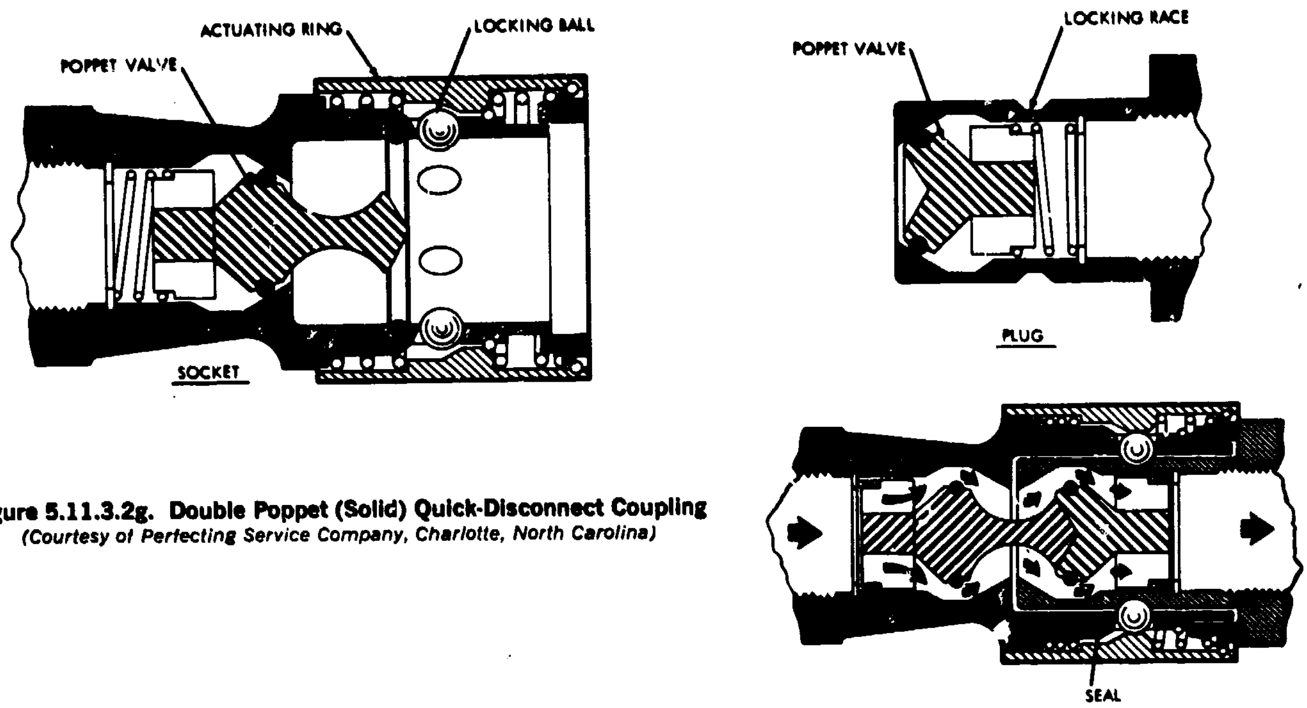


Figure 5.11.3.2g. Double Poppet (Solid) Quick-Disconnect Coupling
(Courtesy of Perfecting Service Company, Charlotte, North Carolina)

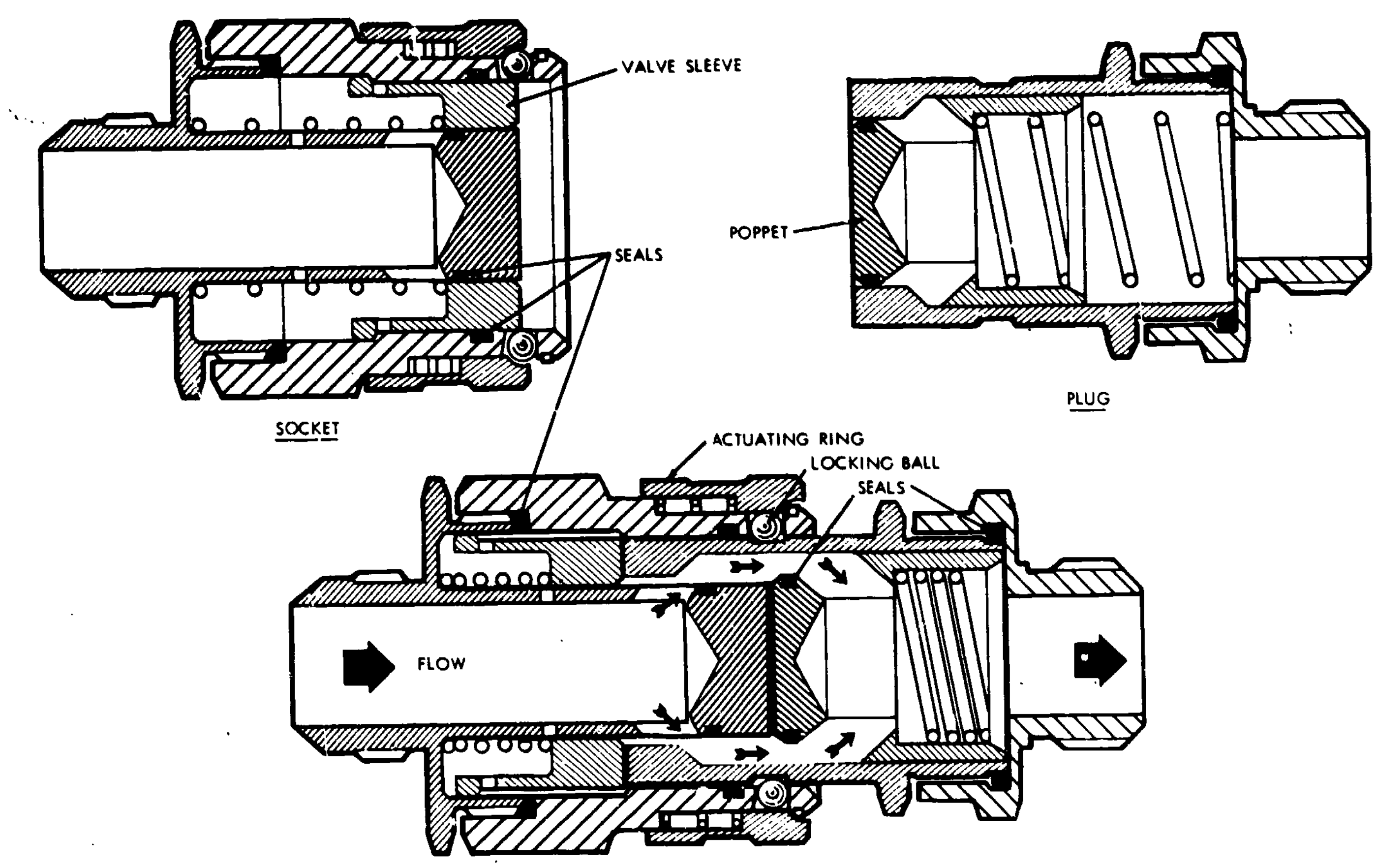


Figure 5.11.3.2h. Tube Sleeve and Poppet Quick-Disconnect Coupling
(Courtesy of Snap-Tite, Inc., Union City, Pennsylvania)

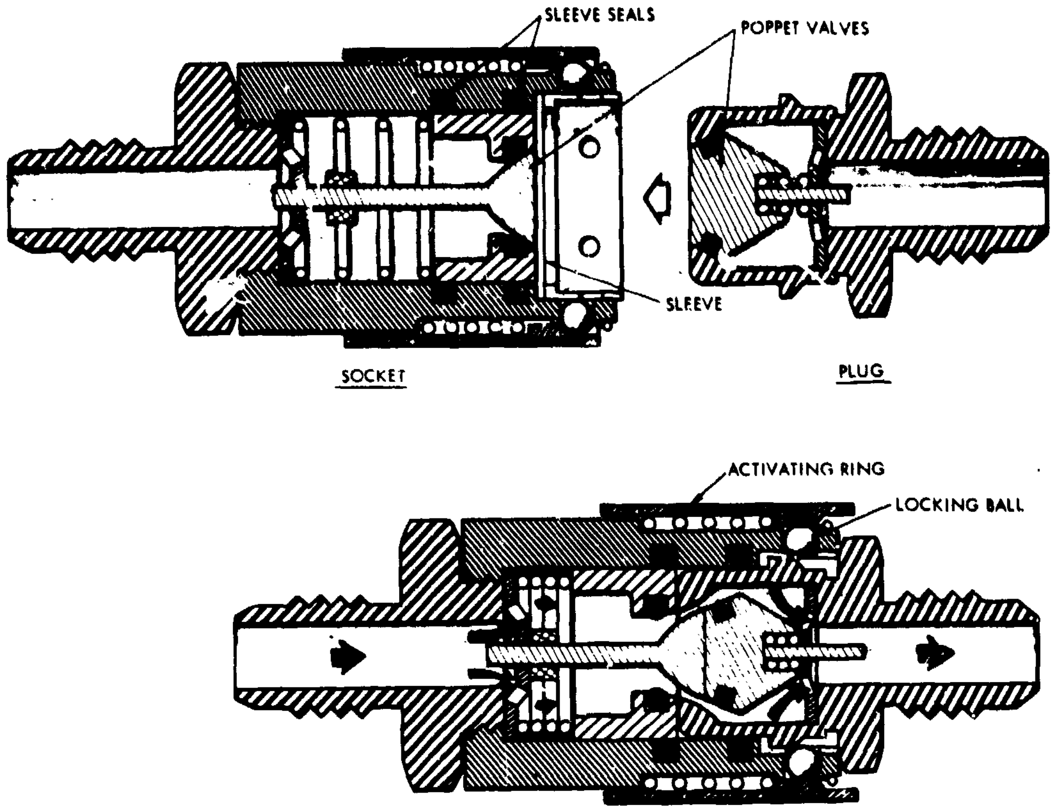


Figure 5.11.3.2i. Poppet Sleeve and Poppet Quick-Disconnect Coupling
(Courtesy of Seaton-Wilson Manufacturing Company, Inc., Burbank, California)

Sleeve and Poppet Quick-Disconnect Coupling. The sleeve and poppet quick-disconnect coupling (Figure 5.11.3.2h) consists of a plug containing a poppet valve and a socket containing a tubular stem and sleeve valve. The plug and socket are pushed together, first forming an external seal between the plug outside diameter and the O-ring seal on the inside diameter of the ball cage. The valve sleeve is pushed into the socket by the plug, which open the tubular stem and sleeve valve. Simultaneously, the tubular stem butts against and opens the poppet valve in the plug. When the valves are fully opened, the balls drop into the locking groove of the plug and the actuating ring slides over the balls, locking the coupling halves together. The coupling is disconnected by retracting the actuating ring and pulling the coupling halves apart.

Poppet-Sleeve and Poppet Quick-Disconnect Coupling. The poppet-sleeve and poppet quick-disconnect coupling (Figure 5.11.3.2i) consists of a socket containing a combination poppet and sleeve shutoff valve, and a plug containing a poppet shutoff valve. The coupling halves are pushed together forming an external seal between the outer diameter of the plug and the inner diameter of the socket. On initial contact, the poppet and sleeve in the socket retract until the poppet has reached its full open position as determined by the stop located on the poppet stem. Further travel of the plug causes the socket poppet to open the poppet in the plug. The plug, in turn, forces the socket sleeve to the full open position. Retraction of the actuating ring allows the locking balls to pass over the locking ridge on the plug. Release of the actuating ring confines the balls in the locking race, forming a positive lock between the coupling haves. Disconnecting is accomplished by retracting the actuating ring pulling the coupling halves apart.

REFERENCES

Plain
6-59, 19-89, V-103, V-106, V-250

Single Poppet
6-59, 19-52, 19-89, V-72, V-103, V-214, V-235

Single Sleeve
6-59, V-252

Double Rotating Ball
1-31, 6-59, V-97

Double Poppets (Drilled)
6-59, V-69, V-212

Double Poppets (Solid)
1-31, 6-59, 19-52, V-72, V-98, V-103, V-116, V-153, V-106, V-180

Tube-Sleeve and Poppet
1-31, 6-59, 19-52, 19-89, V-103, V-116, V-160, V-180

Stem-Sleeve and Poppet
1-31, 73-95, V-30, V-130

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5.12.1 Introduction

Connectors, commonly referred to as fittings, are devices used in fluid systems for the purpose of joining lines or components and sealing the joint thus made. Connectors may be used for line-to-line, line-to-component, or component-to-component applications. All connectors, regardless of type, employ three functional elements: (1) a seal, (2) the connector-to-line joint, and (3) the load carrying structure. The distinction between connectors involves the manner by which these three functions are accomplished. In some separable connectors there is a direct relation between the functional elements and the mechanical parts of the connector, while in permanent connectors all three functions may be combined, as in a single welded joint.

Connectors can be broadly categorized as separable, permanent, and semi-permanent. *Separable connectors* have at least one mechanical joint which can be easily separated and reconnected a number of times. In addition to the mechanically separable joint, separable connectors may employ a permanent (swaged, welded, or brazed) connection for joining the connector mating halves to the tube and/or component. The separable connector category includes threaded connectors (Sub-Topic 5.12.3) and bolted-flange connectors (Sub-Topic 5.12.4). Quick disconnects, which are a special type of separable connector, are covered in Sub-Section 5.11. *Permanent connectors* employ welded, brazed, soldered, swaged, or adhesive-bonded connections rather than mechanical joints, and are used where reduced weight and reliability are more important than ease of separation. Two most common types of permanent connectors are brazed and welded connectors, covered respectively in Sub-Topics 5.12.5 and 5.12.6. Swaged and adhesive-bonded connectors are treated in Sub-Topics 5.12.7 and 5.12.8.

Semi-permanent connectors are a special class of permanent connectors consisting of permanent joints (welded, brazed, or swaged) which can be used for a limited number of separation and reconnection cycles without requiring removal or replacement of lines, or special adapters or fittings in making reconnections. Discussion of semi-permanent connectors is included under the welded, brazed, and swaged connector types.

The purpose of this sub-section is to present design and selection factors for separable, permanent, and semi-permanent connectors, and to present data on specific connector designs within each category.

5.12.2 Connector Selection Considerations

Numerous factors, often conflicting, enter into the selection of a connector for a given fluid system application. The following paragraphs discuss factors to be considered in selection of connectors.

As a general rule, the number of connectors in any system, regardless of type, should be kept to a minimum. Any connector is a potential source of problems. In addition it adds to system weight and costs.

5.12.2.1 PERMANENT OR SEPARABLE CONNECTORS. The first decision which must be made in selecting a connector is whether the joint or joints under consideration should be made separable or permanent. The fundamental distinction between permanent and separable connectors involves the manner in which the connector seal is effected. The seal of a permanent connector is effected by intermolecular cohesion at the sealing interface, whereas the separable connector seal involves mechanical mating, usually involving local deformation of one of the sealing surfaces. The intermolecular bond effected in the permanent connector has a far higher probability of sealing than the mechanical mating of surfaces with the separable connector; however, the permanent connector seal cannot be easily separated, whereas the separable connector can be readily disconnected with a minimum of tooling and, in many instances, reused.

In addition to higher probability of sealing due to lack of mechanical parts, permanent connectors also offer a weight advantage over most separable connectors, often second in importance only to reliability for flight systems. Other attributes of permanent connectors are small envelope and resistance to shock and vibration.

On the other hand, the distinct advantage of separable over permanent connectors is the ease of making and breaking joints, often a very important consideration during system development, as well as for component replacement, repair, and system maintenance.

The heat required for making most permanent joints must be carefully considered regarding any possible detrimental effects on the system. Heat damage to polymeric seals in valves and thermal distortion are common problems in welded installations. Loss of strength in heat-affected areas must also be considered. Accessibility of the connectors to the necessary welding or brazing tools can be another limitation to the application of permanent connectors, depending upon the system configuration. Permanent connectors which join dissimilar materials such as titanium-stainless steel, or aluminum-stainless steel, present problems which can more readily be overcome with mechanical-type joints. Susceptibility of brazed or welded joints to corrosion is another potential problem which must be considered in selecting permanent connectors.

5.12.2.2 PERMANENT CONNECTOR SELECTION. The choice of permanent connectors offers several possible joining techniques including welding, brazing, bonding, soldering, and swaging with welding and brazing the most common. Factors which influence the selection between types of permanent joints include permanence of the joint, temperature range, fluid compatibility, and suitability of the connector and/or tube materials and geometry for either welding or brazing. Although providing a permanent connection, brazing lends itself more readily than welding to making a joint which can be separated and rejoined if necessary, since a brazed connection can sometimes be reheated, separated, and later rebrazed. At the expense of added weight, however, it is possible to provide welded joints with limited reusability. Such joints are commonly referred to as semi-permanent welded joints. Elevated temperature and joint strength are two of the major limitations with brazed joints, resulting from the relatively low melting points of brazing alloys. A practical upper temperature limit for most brazed joints is approximately 1000 F, the actual limit for specific applications being dependent upon the materials used.

5.12.2.3 SEPARABLE CONNECTOR SELECTION. Most separable connectors fall into one of two categories: (1) threaded connectors and (2) bolted flange connectors (in which category clamped flange connectors are considered for the purpose of this discussion).

A basic difference between threaded connectors and bolted flange connectors is the number of tension (or tensile) members in the separable joint. Threaded connectors utilize a single nut, while flanged connectors use several bolts. The choice between threaded and flanged connectors is determined by line size and the amount of preloading required to establish a satisfactory seal over the entire range of connector loads. In smaller line sizes, threaded connectors have a weight advantage. As the line size approaches 1 inch, applied torque limitations reduce the preloading ability of the threaded connector structural members, and only by increased size (and, therefore, increased weight) of these structural members can satisfactory preloading be maintained.

In general, a 2000 inch-pound torque limitation is used for a threaded connector because of wrench size and nut strength limitations. Thus, a practical limitation exists for the amount of torque applied to a single threaded connector and, consequently, the use of threaded connectors is limited by system fluid pressure and line size (Sub-Topic 5.12.3 further discusses preload limitations). Size and weight of the connector are other factors which place a line size limitation on the use of threaded connectors. For smaller line sizes (less than 1 inch) threaded connectors have a weight advantage over flanged connectors. A practical limitation on the use of flanged connectors in small line sizes is the fact that in small flange sizes the bolt becomes too small for reliable torquing. Threaded connectors are favored for use with smaller line sizes because of their relative ease of installation and the simplicity with which the tube may be joined to the connector, especially with flared connectors. One disadvantage of threaded connectors is the danger of damage to the tube or line if the threaded flange is not restrained while the nut is torqued down.

In addition to nuts and bolts, clamps offer a method for joining flanges. Ring clamps, such as the V-band clamp, and snap-on clamps provide advantages with respect to simplicity of installation and ease of removal. Snap clamps, however, have no capability for providing variable preload, while ring clamps are limited to relatively low preloads, thus they are used primarily for low pressure ducting or in small line sizes.

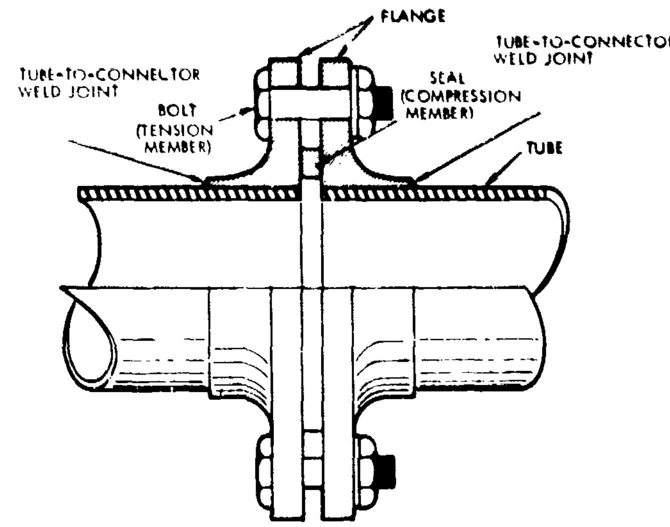

Figure 5.12.3a. Typical Bolted Flange Connector Components

5.12.3 Threaded Connectors

A threaded connector is a line fitting which provides a separable mechanical joint secured by a single threaded nut, whereas a bolted flange connector uses several bolts, a clamp, or combinations of these to secure the joint.

Figures 5.12.3a and b illustrate the functional similarities of typical connectors with separate seals. Both connectors contain three functional elements: (1) the seal, (2) the tube-to-connector joint(s). The functional elements are not necessarily restricted to specific mechanical components of the connectors (Reference 44-14). In the bolted flange connection shown in Figure 5.12.3a, the seal as a functional element consists of the two flanges and the seal; in the threaded connector shown in Figure 5.12.3b, the stub end flange, seal, and threaded flange constitute the seal. The functional similarity between the bolts in Figure 5.12.3a and the nut in Figure 5.12.3b, the tension member of the load-carrying structure, is apparent. The tube-to-connector joints are effected in both instances by weld joints.

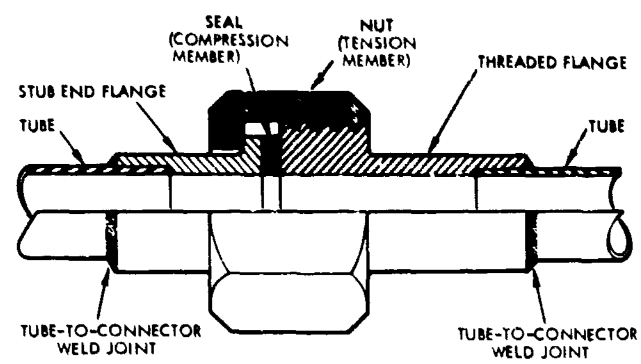

Figure 5.12.3b. Typical Threaded Connector Components

Detailed Topics 5.12.3.1 through 5.12.3.11 discuss considerations which are equally applicable to threaded and bolted flange connectors. Detailed Topics 5.12.3.12 through 5.12.3.15 treat threaded connectors only.

5.12.3.1 DESIGN AND SELECTION CRITERIA. Table 5.12.3.1 summarizes important factors relevant to the design and selection of threaded and bolted flange connectors. Some of the factors noted are discussed in greater detail in the following detailed topics. It should be noted that the design and selection criteria listed under each of the five major headings do not necessarily appear in their order of importance.

5.12.3.2 CONNECTOR PRELOAD. Connector preload is the axial structural load which is required to maintain sealing in any connector; it is provided by tightening the tension member. Preload requirements commonly control the basic design of the connector. The preload has three functions: (1) to seat the seal, (2) to provide force to resist applied structural loads tending to separate the connector joint, and (3) to provide stored elastic energy to maintain sealing load, in spite of dimensional changes resulting from axial and bending loads, stress relaxation, and thermal gradients.

In order to maintain adequate seal load it is necessary to preload the connector to an amount greater than the combined expected axial pressure end load and other externally applied loads. In addition, sufficient load must be provided for dimensional changes caused by thermal gradients. A

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Table 5.12.3.1. Threaded and Bolted Flange Connector Design and Selection Criteria

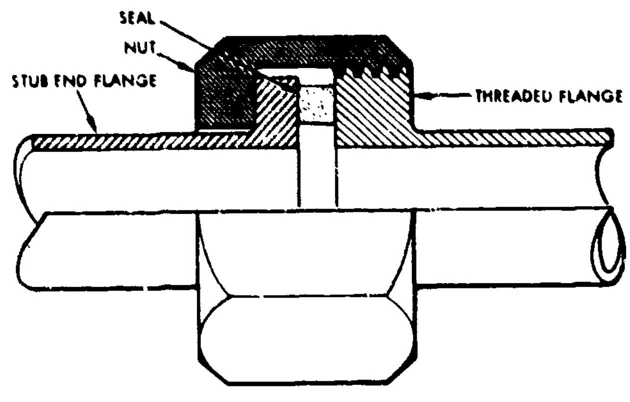
Performance and Environmental Criteria
Leakage
Temperature Range
Temperature Cycling
Pressure Range
Reverse Bending, or Flexure
Impulse-Vibration Cycling
Torque Relaxation
Vacuum-Radiation Tolerance
Material Degradation
Configuration Criteria
Weight
Envelope
Cost
Tube Entry Clearance
Suitable Materials
Preload Relationship
Angularity Tolerance
Manufacturing Criteria
Tolerances Required
Machines Required
Tooling Required
Inspection Required
Tubing Requirements
Reliability and Quality Assurance Criteria
Torque Latitude for Sealing
Torque Latitude for Preload
Effect of Axial Load on Seal
Failure Mode, Effect on System
Inspectability
Assembly Criteria
Torque Required
Cleanliness Required
Care Required
Special Tools Required
Inspectability
Reconnectability

balance must be achieved, however, between a preload which is sufficient to prevent connector separation or seal leakage, and a preload so large that it may cause eventual failure of the tensile members of the connector.

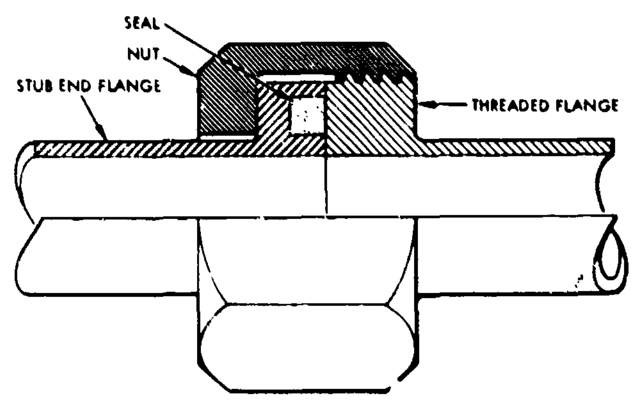
Connector preload can be classified as either series or parallel, as illustrated by the arrangements shown in Figure 5.12.3.2a. In the series arrangement, all of the compressive force of the preload is transmitted through the sealing element, and any change in applied load is felt by the seal. In the series arrangement, therefore, the tension member must have sufficient spring follow-up to account for seal flow or relaxation without the load being reduced below the minimum sealing load. In the parallel arrangement, the seal carries all of the load only until the seal is compressed to the point where the two connector flange halves are drawn together; further application of axial force preloads the connector directly through the flanges and very little additional load is carried by the seal.

Factors the designer should consider in determining the connector preloads are discussed below and illustrated with simplified preload diagrams in Figures 5.12.3.2b, c, d, and e.

Minimum Compressive Seal Load Needed to Prevent Leakage. After the seal has been subjected to an initial seal load, a minimum residual compressive seal load must be maintained during service conditions to prevent leakage. The residual load should give an apparent seal stress greater than the counteracting tensile stresses caused by system operating conditions, including pressure end load and bending, which tend to separate the sealing surfaces. In theory, ideal sealing surfaces would remain in contact until the apparent seal stress decreased to zero. In practice, however, a certain prescribed minimum residual compressive load must be maintained at all times in order to maintain a satis-



TYPICAL SERIES-PRELOADED JOINT



TYPICAL PARALLEL-PRELOADED JOINT

Figure 5.12.3.2a. Series and Parallel Preloaded Joints

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factory seal. The required seal load is the major factor affecting connector design for small line sizes and low system pressures. For large sizes and high pressures, pressure end loads have a more important influence on connector design than seal seating load.

Maximum Allowable Stress in the Tensile Members. The maximum allowable stress in a connector is based on the connector tensile member material yield stress at maximum operating temperature, derated by some safety factor. In an optimized connector design, the maximum operational load considered as an upper limit for connector operation would coincide with the maximum allowable tensile member load.

Deflection Rates of Compression and Tension Members. The deflection rate, which is the reciprocal of the spring rate, is a measure of change in deflection for a given change in load. The deflection rate for the compression members of a connector is invariably much less than the deflection rate of the tensile member in any connector.

Effects of Thermal Gradients. A thermal gradient, caused by the passage of a hot fluid through a connector, generally causes the connector compression members to become hotter than the tension members. This results in an increase in preload, since the compression members expand, transferring load to the tension members. If the tension members are overstressed, material yielding may occur. Such a condition could result in immediate connector structural failure or eventual structural failure as a result of thermal cycling. Material yielding, or creep, might also result in leakage when thermal equilibrium is restored, because the seal load is decreased. In the case where a cold fluid passes through the connector, a contraction of the compression members occurs, resulting in a relaxation of load on the tension members. The load relaxation must not be so great that the sealing load is reduced below the minimum necessary to prevent leakage. Since the connector structure is rather complex from a thermal analysis standpoint, an experimental determination of the thermal gradients expected in a particular connector service application may be necessary.

Magnitude of Structural Loads. Connector structural loads are a function of axial, torsional, and bending loads, each of which is discussed in greater detail in subsequent sub-topics. For an optimum connector design, the maximum tensile load anticipated on the tension members would coincide with the maximum allowable connector load determined from material properties. In addition, the compressive load on the seal interface must never be reduced below the minimum required seal load. The maximum allowable load in the tension member occurs when the maximum hot thermal gradient and maximum tensile end load are applied simultaneously. The minimum compressive load on the seal interface exists when the maximum cold thermal gradient and the maximum tensile end load occur simultaneously.

Variations in Initial Preload. The initial preload on a connector is dependent on many factors, e.g., thread form, thread alignment, thread fit, amount of lubrication, surface finishes of parts, type of lubrication, environment, etc. It is important for the designer to prescribe a preload that will meet the requirements of a satisfactory connector, one which is not so large or small as to cause connector failure (structural or leakage) at temperature extremes under full load. An acceptable preload range can be built into a connector by proper selection of the tension member(s) deflection rate, i.e., selection of tension member(s) with a greater than optimum deflection rate. (The optimum deflection rate is defined as that which produces coincidence of (a) the maximum allowable tensile load, with the tensile load created by the maximum hot thermal gradient, combined with the maximum connector end load, and/or (b) the minimum compressive seal load, with the load created by the maximum cold thermal gradient, combined with the maximum connector end load.

A preload diagram provides a convenient design tool for correlating the factors described above. It is basically a plot of load versus deflection for the tension and compression members of a connector. In the preload diagram shown in Figure 5.12.3.2b, the line O-T is a stress-strain type curve for the tension member showing the expected deflection for applied load. Note that the load-deflection curve for tension member(s), O-T, goes through the origin, i.e., zero deflection at zero load. The load deflection curve for connector compression member(s), c-c', has a negative slope, indicating that increasing load deflection of the compression member(s) is opposite in direction to the deflection of the tension member(s). Note that the intersection of the load-deflection curve of the compressive member(s) with the deflection axes at zero load is quite arbitrary. That is, when a connector nut is tightened or torqued so as to produce a tension member load corresponding to O-a, the compression member(s) of the connector deflect an amount δ_a. When the

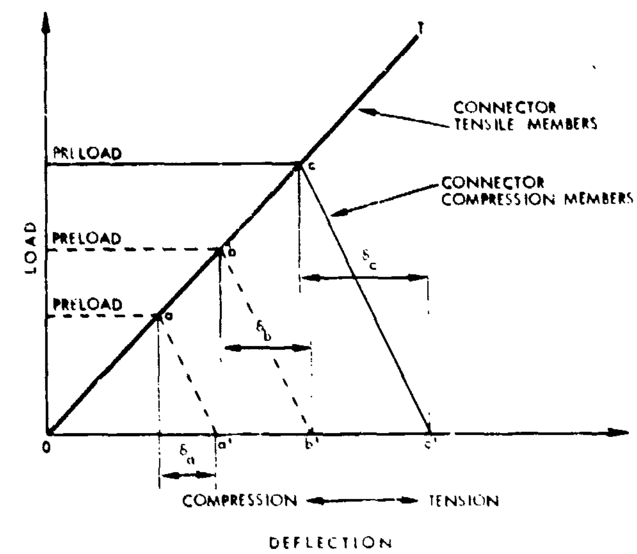


Figure 5.12.3.2b. Load-Deflection Curve

connector nut is further tightened to O-b, the compression members deflect an amount δ_b, and so on until the nut is tightened to the desired preload value, producing δ_c deflection. In all instances the slope or deflection rate of the compression members is constant, but as increasing preload is applied the deflection increases as indicated.

To elaborate further on the load deflection diagram, consider Figure 5.12.3.2c. Three values of preload, designated 1, 2, and 3, are indicated. These preloads establish the deflection of tension and compression members of the connector. When additional load is imposed on the connector in the form of fluid pressure, bending, axial loads, etc., the connector tends to separate. The tension member deflection increases, and the compression member deflection decreases. At preload 1 in Figure 5.12.3.2c, a load L_1 is required to reduce the compression member deflection to zero (by stretching the tension member sufficiently to relieve the preload). If preload 2 or 3 is initially applied, it is necessary to apply a load L_2 or L_3, respectively, to cause connector separation, i.e., zero deflection of compression members.

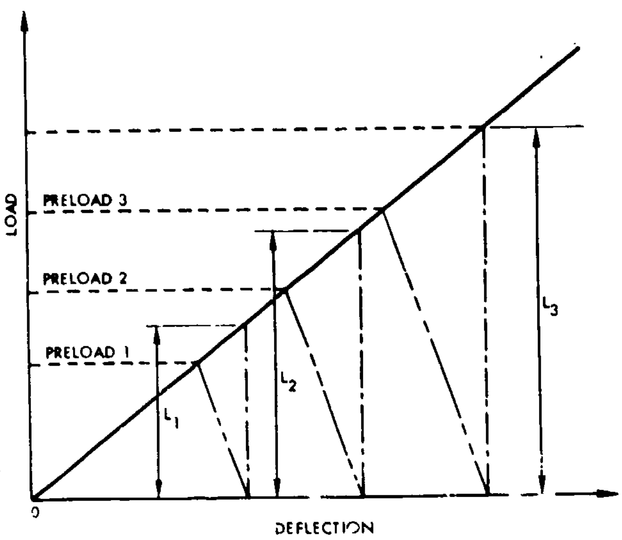


Figure 5.12.3.2c. Preload and External Loading

The foregoing example considers only constant temperature cases. If an initially preloaded connector is subjected to thermal shock, a temperature gradient will exist between the tension and compression members. Most often the thermal shock is imposed from within the line; as a result, the case where tension members are suddenly cooled or heated is generally of less consequence than when the compression members of the connector are cooled or heated. When the compression members are suddenly heated they will expand and transfer load to the tension members. When the compression members are suddenly cooled they will contract, and load will be removed from the tension members. These cases are illustrated in Figure 5.12.3.2d. In addition, this figure shows how an externally applied load, L, affects the connector when thermal shock is imposed.

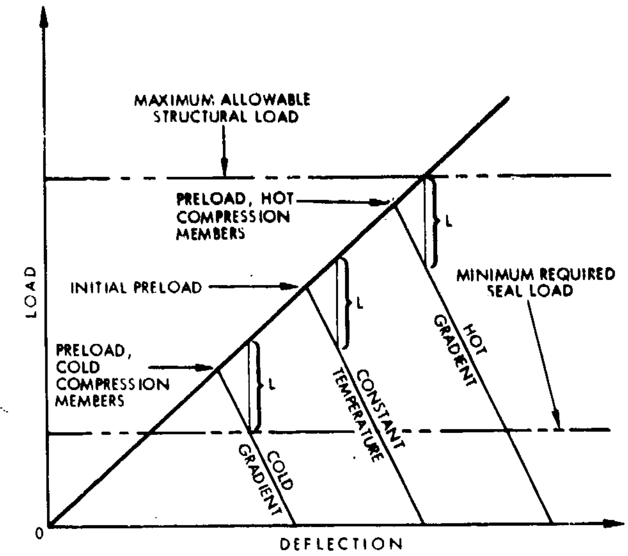


Figure 5.12.3.2d. Effect of Thermal Gradients

The system-imposed tensile end load, L, in conjunction with the maximum hot thermal gradient (gradient from inside to outside of connector) establishes the maximum load on the connector, controlled by structural limitations. The load, L, in conjunction with the maximum cold thermal gradient, establishes the minimum load on the connector, controlled by sealing requirements.

Another point worth noting in considering the preload in a separable connector is the proper selection of the tensile member deflection rate. As previously noted, the tensile member deflection rate is the measure of change in length (deflection) with change in load. Several different deflection rates for separable connector tensile members are illustrated in Figure 5.12.3.2e.

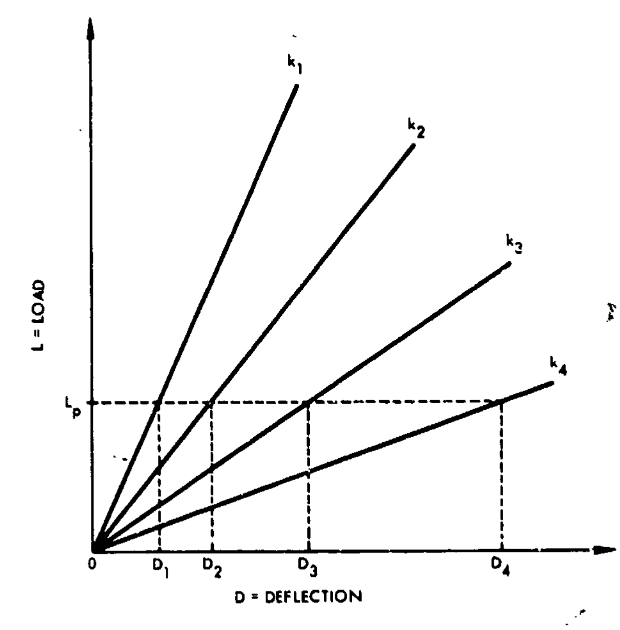


Figure 5.12.3.2e. Tensile Member Deflection Rates

As seen from Figure 5.12.3.2e, a given initial load, L_p, creates differing levels of deflection D_1, D_2, D_3, and D_4 in the tensile members dependent upon the deflection rates k_1, k_2, k_3, and k_4 of each tensile member. When selecting a tensile member deflection rate for a particular application, it is always conservative to use an actual deflection rate greater than the desired deflection rate, except for the case when the anticipated deflection causes yield of the tensile member.

5.12.3.3 SEAL SEATING LOAD. The seal seating load is the initial axial compression load on the seal required to produce intimate contact between sealing surfaces. The seal seating load should not be confused with the minimum residual compressive load necessary to maintain a leak-tight seal under operating conditions. The minimum residual compressive load need only provide an apparent seal stress slightly greater than the total seal unloading due to connector operating loads. The seal seating load can be estimated as

$$F_S = \pi D_{eff} G \qquad \text{(Eq 5.12.3.3a)}$$

where

D_{eff} = effective seal diameter, in.

G = initial required seal seating load, pounds-per-inch of seal length

The value of G in pounds-per-inch for the gasket seating load is a function of the seal material and geometry. Tables 6.3.3.2b,c, and d in Sub-Section 6.3, "Static Seals," list recommended seal stress for several types of seals. Figure 5.12.3.3 is a plot of the required seal seating load as a function of line diameter for representative values of G, assuming D_{eff} equals 1.1 times the line outside diameter.

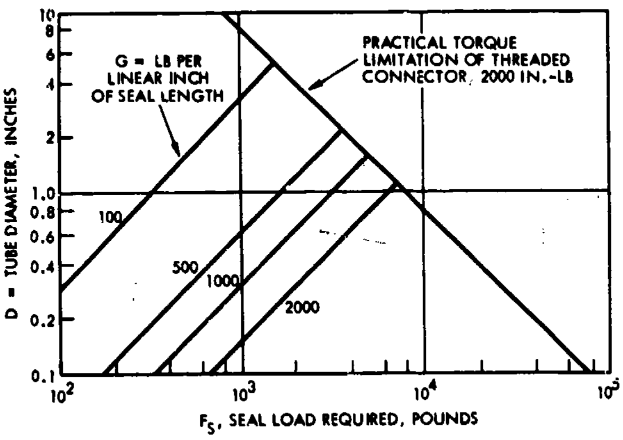


Figure 5.12.3.3. Required Seal Load as a Function of Seal Diameter

To obtain the seal seating load in threaded connectors, it is necessary to apply torque to the connector nut. The amount of torque necessary to obtain some required preload, assuming standard connector threads, may be approximated by the following equation (Reference 44-14):

$$\tau = 0.2 d F \qquad \text{(Eq 5.12.3.3b)}$$

where

τ = required preload torque, in-lb

d = nominal thread diameter, in.

F = required preload, lbs

Unless better information is available, d may be approximated as 1.30 times the tube outside diameter. Using the upper limit of torque applied to a threaded connector as 2000 in-lb (see Detailed Topic 5.12.2.3) an upper limit of connector loading due to preload may be established as

$$F_{max} = \frac{7690}{D} \qquad \text{(Eq 5.12.3.3c)}$$

Figure 5.12.3.3 shows this upper limit as a function of tube diameter. When F_{max} is used in conjunction with the required seal seating loads shown on the same curve, the designer is able to determine whether a given seal, with a specified value of G, can be used in a particular size tubing system since gasket seating load is torque limited.

5.12.3.4 PRESSURE END LOAD. The pressure end load is a result of system pressure, and is equal to

$$F_e = \frac{\pi}{4} D_{eff}^{2} \qquad \text{(Eq 5.12.3.4)}$$

where

F_e = pressure end load, lbs

D_{eff} = effective seal diameter, in.

P = internal pressure, psi

Assuming that D_{eff} = 1.1 times the tubing outside diameter, because the effective diameter of most connector seals is about this much larger than the tube OD, Figure 5.12.3.4 gives values of the pressure end load, F_e, for various combinations of tubing diameter and internal fluid pressure.

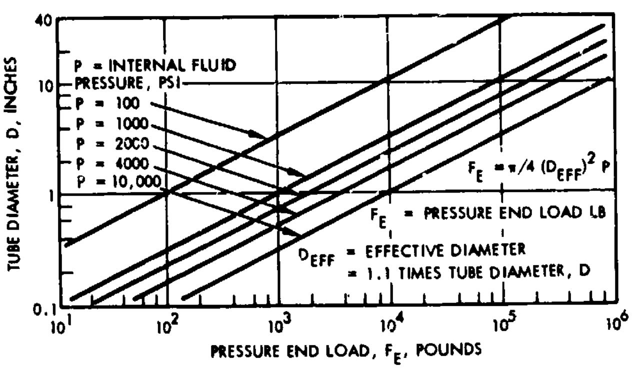


Figure 5.12.3.4. Connector Pressure End Load as a Function of Tube Diameter and Design Pressure

The system operating pressure is a major determining factor in the design or selection of a threaded connector. In a ground-based system, the pressure end load is frequently the design load for the connector. Although anchors, fluid components, and other restraints absorb a part of the pressure end load, design practice necessitates the connector be made to withstand the entire pressure end loading. Pressure impulses must be considered from this same standpoint, i.e., the connector must be designed to withstand the impulse load. The system pressure in conjunction with the tubing diameter will also be a major selection factor in choosing between a threaded connector or a flanged separable connector. Structural limitations of the threaded connector and practical means of applying large torques commonly restrict applications of threaded connectors to tube sizes of one inch or less.

5.12.3.5 BENDING MOMENT LOAD. A bending moment, M, producing bending loads, F_B, may be present because of tubing misalignment, thermal expansion or contraction of the tubing system, vibration, displacement of anchors, or acceleration forces. Bending moments imposed on a fitting in a tubing system cannot usually be determined in advance, since these moments depend upon the specific tubing system, its operating conditions, and the location of the threaded connector in the system. However, some limits, even if arbitrary, must be established so that the designer can gain a feeling for the bending loads imposed on a connector. In general, the bending load can be expressed, in terms of the maximum stress that can be transmitted by the tubes, as

$$F_B = \pi D t \sigma_m \qquad \text{(Eq 5.12.3.5a)}$$

where

D = tubing outside diameter, in.

t = tube wall thickness, in.

σ_m = limiting longitudinal stress of tube material, psi

In the previous equation, the value of t (tube wall thickness) is obtained, if not already known, from the tubing design considerations. For thin-walled tubing, t may be evaluated (Reference 44-14) from

$$t = \frac{PD}{2\sigma_c} \qquad \text{(Eq 5.12.3.5b)}$$

where

P = internal pressure rating of system, psi

D = tube outside diameter, in.

σ_c = limiting circumferential (hoop) stress of tube material, psi

The bending load limit then becomes

$$F_B = \frac{\pi}{2} PD^2 \frac{\sigma_m}{\sigma_c} \qquad \text{(Eq 5.12.3.5c)}$$

The problem of determining the maximum allowable bending load now becomes one of determining the appropriate limiting values for σ_m and σ_c in Equation (5.12.3.5c).

For aerospace applications it is desirable to maintain minimum weight throughout. In this respect, σ_c may be taken as either two-thirds of the material yield strength or one-half of the material ultimate strength, whichever is less. This is based on a proof pressure rating of 1.5 times system operating pressure and a burst pressure rating of 2 times system operating pressure.

The value of σ_m in a moderate application could be taken as the tube material yield strength. However, the severity of an aerospace environment usually imposes more stringent limitations on values of σ_m. For example, cyclic loading resulting from vibration would require σ_m to be selected on the basis of tube material fatigue strength. Additional factors such as impact loads, temperature extremes, etc., would further modify the value chosen for σ_m. A typical value for σ_m might therefore be based on one-half the tube material endurance strength.

5.12.3.6 EXTERNAL AXIAL AND TORSIONAL LOADS. As with bending loads, external axial and torsional loads imposed upon the connector assembly by the line are largely a function of the line diameter. Usually the axial loads which may be transmitted by small diameter tubing to a threaded connector are minor in comparison with preload and pressure forces, and may be discounted. Torsional loads may present a problem if sufficiently large to cause rotation of one part of the joint with respect to the other.

The larger lines normally associated with bolted-flange applications may transmit forces to the connector appreciably in excess of those associated with internal pressure and seal preload requirements. As with bending loads, this is particularly true of short, large diameter lines whose terminal points are subjected to significant displacements, as shown in Section VIII-B-3 of Reference 34-10.

5.12.3.7 VIBRATION. The vibration spectrum for a threaded connector is wholly dependent on the intended system application. Some representative vibration environments are presented in Sub-Topic 13.3.5 and Sub-Section 7.3. In any threaded connector application involving thin walled tubing, experience has shown that a simple vibration loading may be expected to induce structural failure in the tubing or piping before connector structural failure occurs. However, leakage, the most common type of threaded connector failure, may develop prior to structural failure of the tube. Reference 147-1 describes a number of vibration tests of threaded connectors and shows tube rupture to be more prevalent with flared-tube connectors, whereas sleeve seal connectors tended to leak under vibration. At low system pressure in the presence of vibration, inadequate seal seating stresses may lead to separation of sealing surfaces and subsequent leakage. At elevated system pressures, the vibration-induced bending load, in addition to the pressure end load, may cause fatigue failure of a connector or may cause material yield and eventual leakage due to insufficient preload.

The precise location of tubing failure due to vibration is largely a function of the connector type. In the connectors that incorporate tube flaring, the root of the flare becomes the most likely point of failure. This is due to the cold working of the tube material in the flaring process. For connectors that utilize a swaging process to hold the tube in place, the point where the tube is held is most likely to fail because of stress concentration induced in swaging. For connectors utilizing a weld joint to the tubing, the weld area is susceptible to failure under vibration as the result of exposure to extreme heat in the welding process. For connectors using brazed joints, the brazing process must insure adequate wetting of surfaces or tube pull-out may result from insufficient shear strength. Although other failure modes exist and must be investigated, those mentioned above represent the most commonly encountered failure modes under vibration.

5.12.3.8 CONNECTOR ASSEMBLY. The need for special assembly tools may be a determining factor in the design or selection of a connector. The tube-to-connector joint may necessitate special welding, brazing, swaging, holding, or other tools. The connector-to-connector joint may also require some form or combination of the above-mentioned tools. Therefore, before the type of connector is selected, consideration must be given to the method of assembly. See SAE ARP 683, "Installation Procedures and Torques for Fluid Connections."

In addition to the special tools requisite in a connector assembly, the methods in which these tools are used bear consideration. The critical steps in connector-to-tube assembly, connector-to-connector assembly, and proper handling and installation of seals must be evaluated. Details of torquing and the effect of over- or under-torquing a connector must also be investigated. Tube material properties and the ability of the tube to be cold-worked for flaring or swaging as well as weldability, brazability, and bondability may also be a consideration in a connector selection or design.

5.12.3.9 CONTAMINATION. At any point in a tubing or piping system where a connector is installed, there exists a point of potential system contamination. When a connector is disconnected, adequate protection of the tubing or piping immediately surrounding the fitting must be provided in order to prevent foreign particulate matter from entering the system. In addition to environmentally introduced contamination, potential contamination emanating from the connector must be considered. Some connectors generate and collect more contamination than others, due to notches, grooves, recesses, or other cavities where contamination is likely to collect. The threads of a threaded connector are prime candidates for contamination generation, as are certain weld and braze joints. In this regard, the method of assembly may become a critical factor. The performance of the threading operation in joining the threaded connector, as well as the use of thread lubricants and sealants, may necessitate limitations, controls, special methods, etc. Aside from any special precautions, it is good practice to wash the connector and the immediately surrounding tubing or piping with an acceptable solvent prior to making a disconnect (opening the system).

5.12.3.10 TEMPERATURE AND THERMAL SHOCK. The temperature environment is a determining factor in the selection of materials of construction for a separable connector. At elevated temperatures, the connector must be designed to withstand the effects of material creep and stress relaxation. At low temperatures, material ductility must be sufficient to prevent brittle failure. As discussed in Detailed Topic 5.12.3.2, "Connector Preload," thermal shock and the associated thermal gradients must be taken into account when determining an appropriate preload range for a separable connector.

5.12.3.11 ADDITIONAL FACTORS. Some additional factors in a threaded or bolted flange connector design worth consideration are:

a) The total number of parts involved in a particular connector and the associated storage, handling, availability, and logistics problems

b) The critical dimensions and/or critical surfaces, with related machining, handling, and protection

c) The necessity of special containers and/or individual handling of separate seals, special connector devices, etc.

d) The overall size and weight of the connector

e) The quality control necessary in the production, procurement, and installation of a given connector

f) The ease with which a given connector can be inspected when installed, and the associated equipment necessary to accomplish inspection.

g) The necessity for insuring that assembly and inspection personnel are thoroughly familiar with both the design and functional characteristics of the connector. This factor is essential if reliable, leak-tight systems are to be consistently achieved.

5.12.3.12 THREADED CONNECTOR TYPES. Major distinguishing features between various types of threaded connector designs are the seals, both tube-to-connector seals and connector-to-connector separable joint seals. Probably the most common classification of threaded connectors is the flared tube seal connector, a design which combines features of the tube-to-connector seal and separable joint seal by using the flared tube as a gasket clamped between the connector halves.

A second threaded connector type is the sleeve seal connector, employing a sleeve which is attached to the tube; instead of the tube forming the separable joint seal, the sleeve and connector together form the seal. Sleeve seal connectors include those commonly referred to as flareless connectors.

A third and distinctive type of connector is the separate seal connector, employing a separate removable sealing element between the connector halves (flanges) to make the separable joint seal. In this connector design, a permanent type joint (i.e., welded or brazed) is commonly employed to provide the tube-to-connector seal. Swaging is also employed in some separate seal connector designs to accomplish the tube-to-connector seal.

In the following sub-topics, samples of specific connector design types which can be classified under each of the above-mentioned threaded connector seals are described. In these descriptions, reference to a "tube-to-tube joint" has been adopted for simplicity, and use of the term "union" is adhered to in this case, although other types of joints such as tube-to-component, etc., exist.

5.12.3.13 FLARED TUBE SEAL CONNECTORS. The flared tube seal type connector is probably the most common threaded connector. This connector utilizes the tubing flare to accomplish both the connector-to-connector seal and the tube-to-connector seal. This design has numerous variations, most of which employ the same basic geometry but differ in the manufacturing and quality control requirements. The AN (Air Force Navy Aeronautical), MS (Military Standard), and MC (Marshall Center) connectors are common flared tube connectors which can be procured to government or industry specifications. These connectors and others are discussed in the following paragraphs.

AN Flared Connectors. The AN flared connector, illustrated in Figure 5.12.3.13a, consists of a sleeve, a connecting nut, and a union. This connector is manufactured in accordance with military specification MIL-F-5509B, titled "Fitting, Flared Tube, Fluid Connection." In making a tube connection with this type of connector, the following sequence of operations is generally followed: (1) The nut and the sleeve are placed on the tubing, (2) the tubing is flared by a hand tool or by a machine, and (3) the connection is secured by placing the flared tubing in contact with the nose of the union, sliding the sleeve and nut against the flared tubing end, and threading the nut into place on the union. The sleeve is forced against the back of the tube flare by the nut as it is threaded onto the union. As the nut is tightened, the sleeve compresses the back of the tube

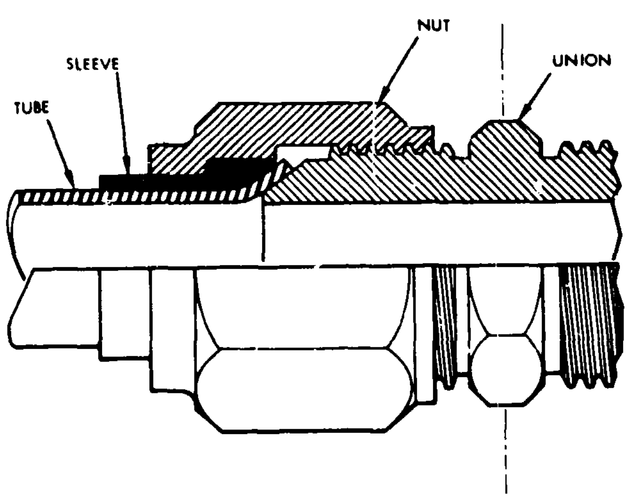


Figure 5.12.3.13a. AN Flared Connector

flare against the nose of the union, thus forming the seal. Actual sealing is accomplished by deformation of the flared tube against the mating surface of the union.

A major advantage of the AN flared connector is its simplicity; however, several disadvantages have given rise to modifications and even completely new connector design concepts. Some of the outstanding disadvantages include:

a) The torque relaxation with time characteristics. This is a tendency of the flared fitting to lose the initial assembly torque after a period of time, resulting in a loose connection.

b) The inability to obtain sufficient sealing stresses in ½ inch and larger connector sizes. To obtain satisfactory metal-to-metal sealing against helium leakage, recent work indicates that a contact stress of 1 to 3 times the yield strength of the softer material is required. Using the minimum torque requirements given in AND 10064 for steel tubing and the relation $\tau = 0.2$ dF (see Detailed Topic 5.12.3.3), values of available seating stress can be obtained and compared with those values of seating stress required for specific applications.

c) Inadequate quality control requirements. The inspection sampling frequency is generally considered insufficient, and surface finish requirements on sealing surfaces are inadequate for satisfactory sealing in the presence of helium, low leakage requirement, high pressure, or cryogenic temperature.

d) The tendency of hard steel tubing to crack when subjected to flaring operations. Aerospace applications frequently demand the use of high strength tubing. This type of material is frequently not capable of being cold worked without having extremely high residual stress levels result, therefore requiring stringent material and manufacturing specifications.

To solve the problem of the AN connector a modification in the form of a separate seal has been devised. This seal, shown in Figure 5.12.3.13b, is a truncated cone, and fits over the end of the 37-degree union nose. The seal is made of a soft material, generally copper or aluminum, offering lower required seating stresses. While presenting a solution to the required seating stress problem, the separate seal has the disadvantage of adding another piece to the connector (which could be omitted if proper care is not taken during assembly), and it does not eliminate the disadvantages previously identified in (a), (c), or (d).

MS Flared Connector. The MS flared connector, often referred to as a precision flared fitting, utilizes the same nut and sleeve as the AN connector, and is procured to the same military specification. The primary difference between the MS and AN connector exists in the union: the MS union (and other parts to which the connector nut attaches) is procured to MS (Military Standards), whereas the AN union is procured by Air Force/Navy Aeronautical Standards. Some important characteristics of the MS unions are:

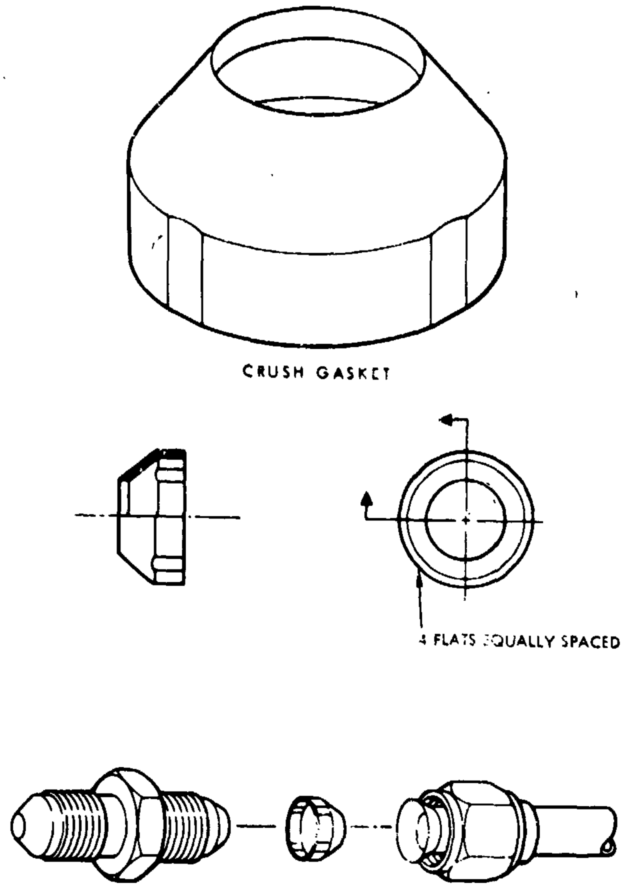


Figure 5.12.3.13b. Conical Seal for AN Connector
(Courtesy of Voi Shan Industries, Culver City, California)

a) Finer finish on the 37-degree cone of the sealing surface
b) 100 percent inspection of unions for thread concentricity to each other, thread relief lengths and diameters, surface smoothness of thread reliefs and hex faces, and surface finish of cone ends
c) Tighter dimensional tolerances restricting the overall length of the connector (union or other) hex nut-to-tip dimension.

These changes were incorporated as an attempt to obtain better sealing quality by imposing tighter quality control, but they do not resolve the sealing stress requirement problem mentioned above under the discussion of the AN flared connector.

MC Fittings. MC fittings, which were developed by NASA, are basically similar in appearance and operation to the AN and MS flared fittings shown in Figure 5.12.3.13a. The MC connector is procured to NASA Specification MC 146. Some of the refinements incorporated into the MC fittings in order to increase reliability are:

a) The finish of the conical sealing surface is limited to 10 to 14 microinches rms profile for steel, and 22 to 28 microinches rms for aluminum.
b) The sealing surfaces are round, within 0.0003 inch, and limited to a maximum deviation of 0.0001 inch in any 60-degree arc.
c) Concentricity of the sealing surface with the thread pitch diameter is within 0.003 inch TIR (total indicator reading).

Globe Aerospace Connector. This connector, shown in Figure 5.12.3.13c, uses the standard AN, MS, or MC sleeve and union, but employs a different concept in the locking nut. The nut is composed of two parts: a hex nut similar to an AN818 with a groove in the shoulder length, and a locking nut with the same size hex and six barbed fingers. The fingers of the lock nut fit over the shoulder of the nut proper in such a way that all six fingers of the lock nut engage the groove in the nut proper. Both nuts are threaded onto the union when forming a connection. The nut proper is torqued first and accomplishes the seal between tube flare and union. The lock nut is then torqued to an equal or greater value, and the barbed fingers grip the lip of the groove on the nut proper and lock it in position. The lock nut innovation precludes the necessity for safety wiring in the presence of a vibration environment.

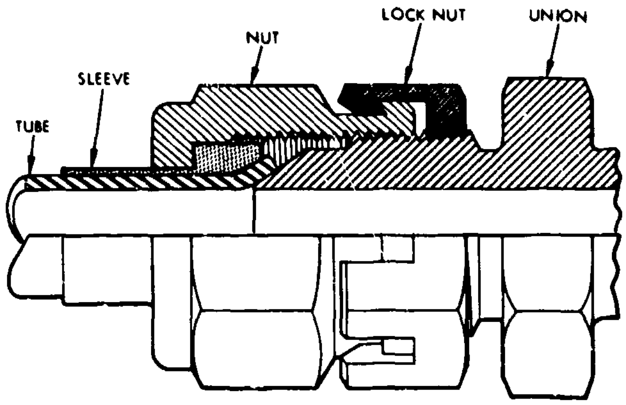


Figure 5.12.3.13c. Globe Aerospace Connector
(Courtesy of Globe Aerospace, North Hollywood, California)

DL Connector. The DL connector, as shown in Figure 5.12.3.13d, is in essence an AN connector, but with two modifications:

a) The sleeve of the DL connector is the same thickness throughout its entire length.
b) The nut of the DL connector has a groove in the outer periphery at about the midway point of its length, and has greater material volume at the point where it contacts the sleeve.

These modifications remove material from the sleeve, place it in the nut, and the nut is thus able to carry heavier loading.

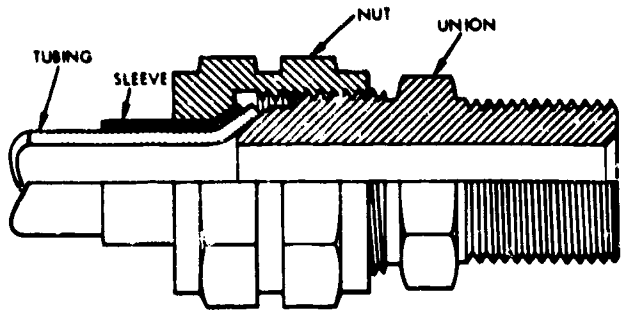


Figure 5.12.3.13d. DL Connector
(Courtesy of E. B. Wiggins, Inc., Los Angeles, California)

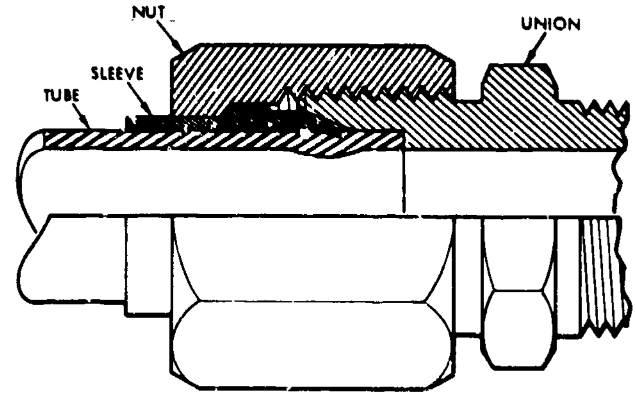


Figure 5.12.3.14a. MS Flareless Connector

5.12.3.14 SLEEVE SEAL CONNECTORS. The sleeve seal connector is characterized by effecting a seal between the union and a sleeve. In some cases two parallel seals are required, the first as previously mentioned and the second between the tube and the sleeve. The sleeve is usually attached permanently to the tubing by swaging, brazing, or welding. The following paragraphs describe various sleeve seal connectors.

MS Flareless. The MS flareless connector shown in Figure 5.12.3.14a is made up of a union, a sleeve (often referred to as a ferrule), and a nut. As the tubing is inserted into the union, it is seated on an internal shoulder; the ferrule is then slid along the tube and seated against the internal taper of the union. Following this, the nut is tightened and the sharp edge of the ferrule is driven into the tubing, forming a seal and forcing the tube forward against the shoulder. At this same time, the seal of ferrule-to-internal taper of the union is formed, as the ferrule bows up to make contact with the internal taper. After the initial connection, the ferrule is locked on the tubing. Upon disassembly, a certain amount of springback takes place in the ferrule, and as long as the connector is not overtorqued, reuse is acceptable. Overtorquing can cause yielding of the ferrule, loss of the ferrule, and/or springback, and may result in poor sealing.

It has been found that machine presetting the ferrule on the tubing results in a better quality connector. However, even this operation requires a certain degree of skill, although machines such as the Weatherhead fixed stop or Parker hydraulic pressure-balanced preset machines are well suited for repeatability and rapid, uniform presets. Another limitation of this connector is the fact that the exterior surfaces of the tubing and ferrule must be protected from damage if a leak-free connector is to be insured.

Flodar Fitting. The Flodar connector, manufactured by the Flodar Corporation and shown in Figure 5.12.3.14b, is similar to the MS flareless connector in external appearance. The Flodar, like the MS connector, uses a union, sleeve, and nut. In the Flodar fitting, as the nut is tightened onto the union the tube is flared slightly at the end, a result of the internal part of the union having a V-shaped seating surface. The sleeve bears against the back of the tube flare to retain the tubing, preventing tube "pull out" and forming a secondary seal. The rear portion of the sleeve is slotted every 90 degrees to give a "collet grip," which aids in dampening vibration.

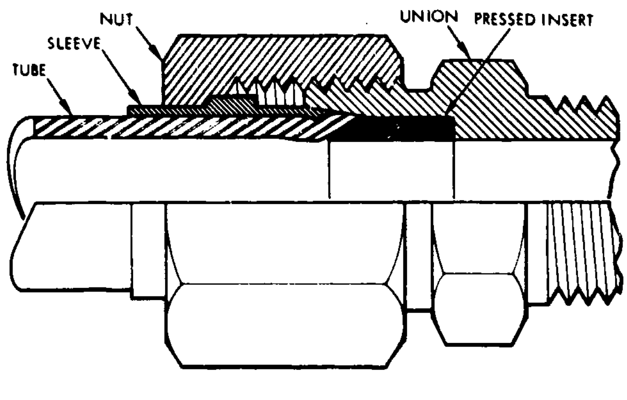


Figure 5.12.3.14b. Flodar Connector
(Courtesy of Flodar Corporation, Cleveland, Ohio)

Swagelok. The Swagelok connector, manufactured by the Crawford Fittings Corporation and shown in Figure 5.12.3.14c, is also similar to the MS flareless connector. As in the MS connector, the tube is seated against an internal shoulder within the union. However, instead of a single piece sleeve, the Swagelok employs two pieces, the front ferrule and the rear ferrule, both of which are contained within the nut.

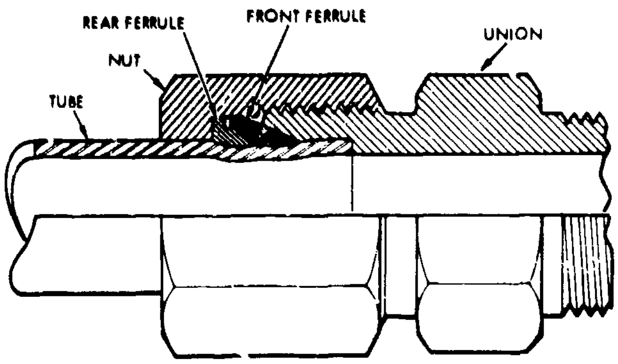


Figure 5.12.3.14c. Swagelok Connector
(Courtesy of Crawford Fitting Corporation, Cleveland, Ohio)

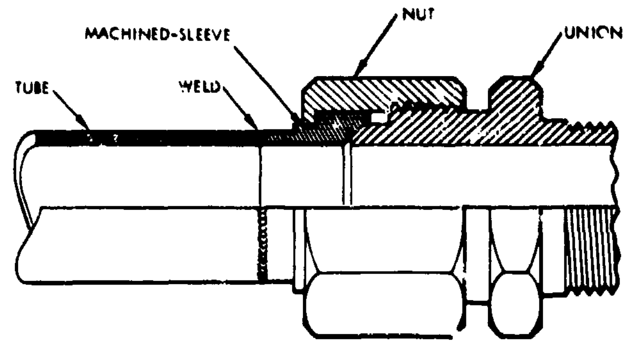


Figure 5.12.3.14e. Machined-Sleeve Connector
(Reference 36-11)

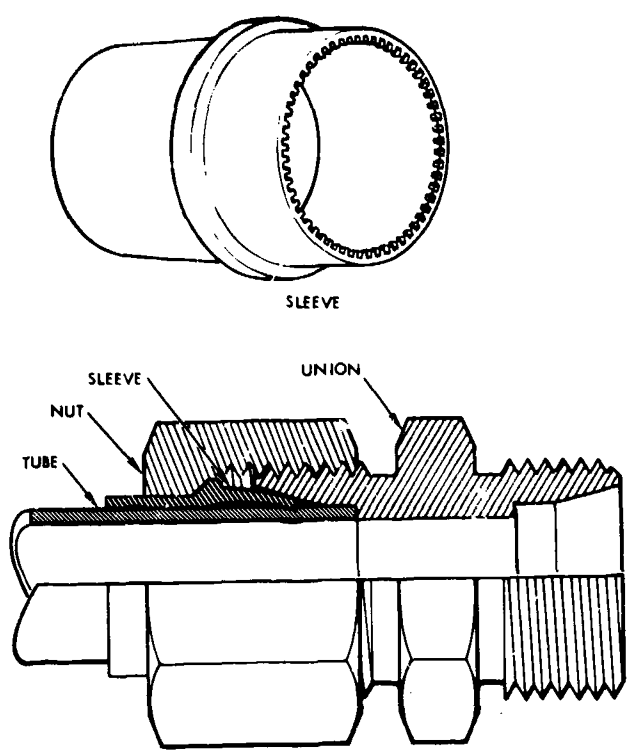


Figure 5.12.3.14d. Sermeto Serrated-Sleeve Connector
(Courtesy of The Weatherhead Company, Cleveland, Ohio)

In seating the tube against the union, the nut forces the rear ferrule against the front ferrule and both ferrules bite into the tube, forcing the tube against the inner shoulder of the union. A secondary seal of front ferrule-to-union is also formed upon tightening the connector nut. The double ferrule offers greater tube holding than the MS flareless connector.

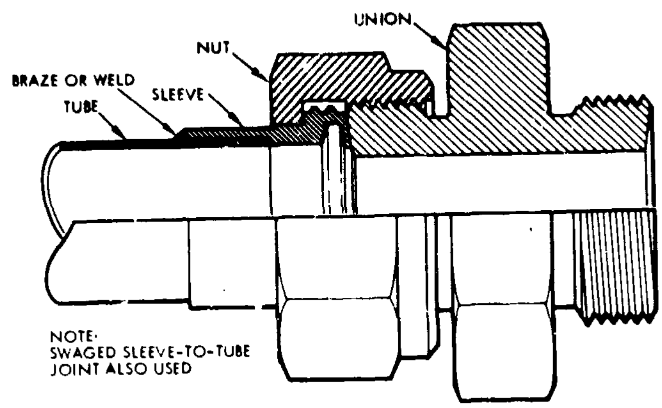


Figure 5.12.3.14f. Dynatube Connector
(Courtesy of Resistoflex Corporation, Roseland, New Jersey)

Sermeto Serrated Sleeve. The Sermeto Serrated Sleeve, manufactured by the Weatherhead Company and shown in Figure 5.12.3.14d, seats the tubing against the inner shoulder of the union in exactly the same fashion as the MS connector. The sleeve of this connector is a single piece, but is similar to the combination of front and rear ferrule in the Swagelok connector in that it is contained completely within the nut. In addition, the leading edge on the sleeve inner diameter is serrated, giving a greater cutting edge area than a smooth sleeve, and providing additional retention of the tube.

Machined Sleeve. This type of connector, as shown in Figure 5.12.3.14e, utilizes standard AN nuts and mating parts. The sleeve, however, is a specially machined piece that is permanently attached to the tubing. In this connector, the sleeve mates with the tapered nose of the union to form the seal. The machined sleeve connector is generally considered to be a stronger connector than standard AN or MS flared connectors, and is less sensitive to vibration environments.

Dynatube. The Dynatube connector, manufactured by the Resistoflex Corporation and shown in Figure 5.12.3.14f, utilizes as the sealing member a specially designed sleeve which is directly and permanently connected to the tubing. The face of the sleeve seals against the union (mating part) as the nut is drawn tight. Internal pressure assists in making the seal during operation. A feature of this connector is the small threaded portion on the exterior of the sleeve which engages the threads of the nut when the fitting is disconnected, preventing the sealing surface from being exposed, as well as preventing the nut from slipping back on the tubing.

Hi-Seal. The Hi-Seal connector, manufactured by Imperial Eastman Corporation and shown in Figure 5.12.3.14g, is composed of a sleeve, a nut, and a union. The sleeve has a precision 12-degree cone which slides into the union and provides the seal as the nut is drawn onto the union. The nut also drives the serrated edges on the inside of the sleeve into the tubing, thus swaging the sleeve permanently to the tubing.

Lo-Torque. The Lo-Torque connector, manufactured by Parker Aircraft Company and shown in Figure 5.12.3.14h, is similar to the machined sleeve connector previously described. Generally, the sleeve and threaded flange are brazed to the tubing instead of being welded.

5.12.3.15 SEPARATE SEAL CONNECTORS. The separate seal connector employs an individual piece, functionally identical to a flange gasket, to accomplish connector sealing. This type of connector generally has all components except the seal attached to the tubing system, thus the seal is the only "loose" piece. The major disadvantage of separate seal connectors is the possibility of omitting the seal at assembly. Several types of separate seal connectors are described in the following paragraphs.

AFRPL Connector. The AFRPL connector, shown in Figure 5.12.3.15a, was developed for the Air Force by Battelle Memorial Institute (Reference 44-14) to temperature and environment specifications beyond the capability of most commercially available connectors. In this connector, a separate seal, referred to as the "bobbin seal," is used as the sealing member, so named because of its resemblance to the bobbin of a sewing machine.

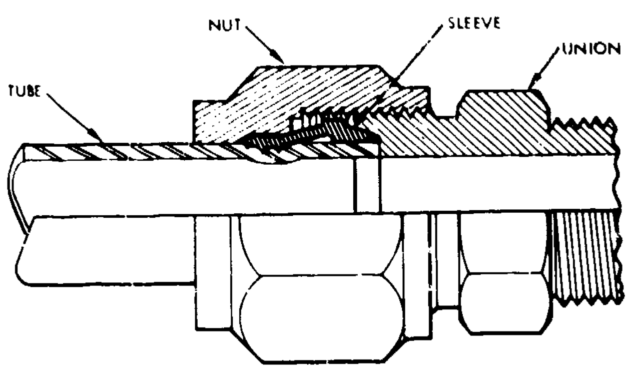


Figure 5.12.3.14g. Hi-Seal Connector
(Courtesy of Imperial Eastman Corporation, Chicago, Illinois)

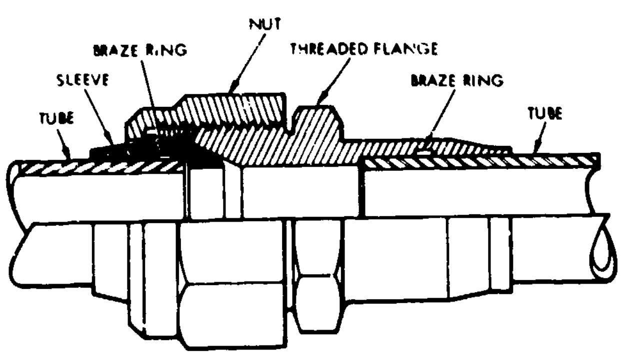


Figure 5.12.3.14h. Parker Lo-Torque Connector
(Courtesy of Parker Aircraft Co. Los Angeles, California)

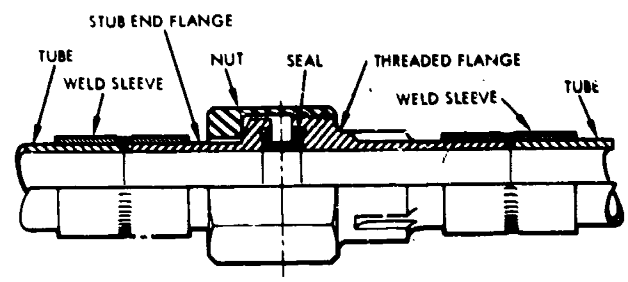


Figure 5.12.3.15a. AFRPL Connector, Showing Tube-to-Tube Connection

The conical discs of the bobbin seal mate with the plain stub end on one side, and with the threaded stub end on the other. As the connector is tightened, plastic deformation of the seal legs occurs on the radial surfaces and the necessary connector seal is created. Reuse of the seal is not recommended, since the sealing surfaces have been deformed and an intimate mating of surfaces cannot be guaranteed. Depending upon the materials involved and the intended usage, the bobbin seal may or may not be plated. The AFRPL connector is normally welded to the tube, as illustrated.

Advantages of this connector are: (1) positive sealing can be achieved by the development of sufficient sealing stresses without excessive torque, and (2) this design affords a wide range of preloading values and can be made to give parallel loading. The major disadvantage of this connector is that the seal must be replaced after a single usage. The connector can be inspected for possible seal omission: if the seal is left out, the nut will be threaded well beyond the thread form on the threaded flange, thus providing obvious detection.

Gamah Connector. This connector, manufactured by the the Gamah Corporation and shown in Figure 5.12.3.15b, employs a seal member which is essentially a belleville spring. Sealing is accomplished by plastic deformation of the seal member. In this connector, the seal can be reversed and used a second time, but further use is not recommended. To achieve a minimum envelope size, the connector uses a circular nut, which requires a spanner wrench for tightening. The connector parts that are joined to the tubing are usually swaged on to the tube, but may be welded or brazed.

Astro-Weight Connector. This connector, manufactured by Harrison Manufacturing Company and shown in Figure 5.12.3.15c, utilizes a K-seal which attains sealing by metallic elasticity as well as pressure energization. Upon assembly, the seal makes intimate contact with the sleeve and the connector. When system pressure is applied the seal is forced against the sleeve and the connector to assist sealing still further. The sleeve and male nut of the connector can be swaged, welded, or brazed to the tubing.

Roylyn Connector. This connector, manufactured by Roylyn Incorporated and shown in Figure 5.12.3.15d, is a four-piece unit that generally has the male fitting and the sleeve welded to the tubing, with the nut loose on the tubing but prevented from falling off by the position of the sleeve. The separate seal is of the belleville spring type.

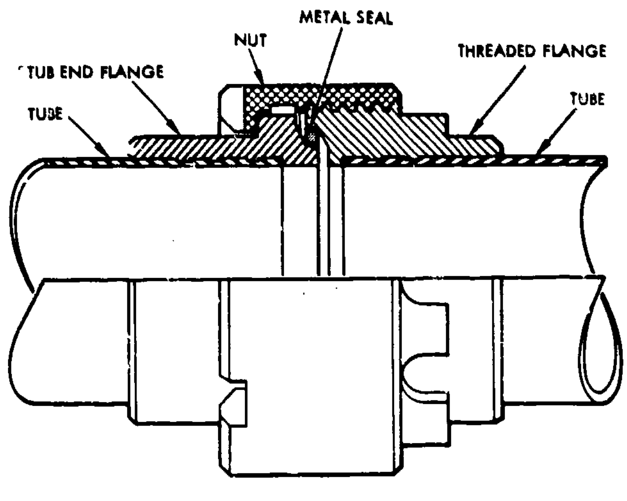


Figure 5.12.3.15b. Gamah Connector, Showing Tube-to-Tube Connection
(Courtesy of Gamah Corporation, Denver, Colorado)

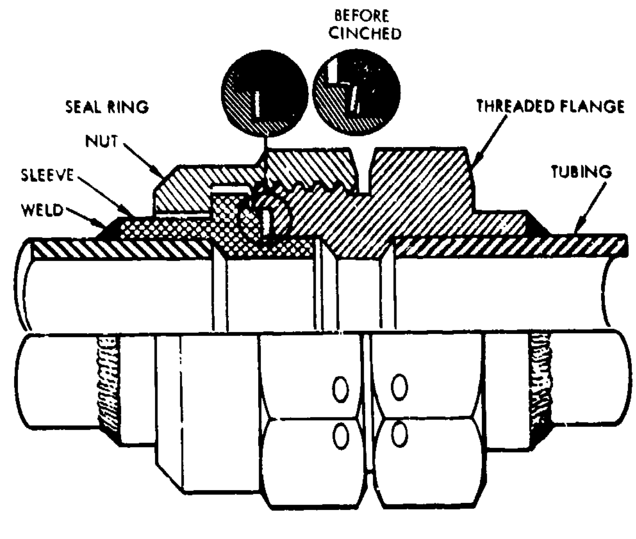


Figure 5.12.3.15d. Roylyn Connector, Showing Tube-to-Tube Connection
(Courtesy of Roylyn Incorporated, Glendale, California)

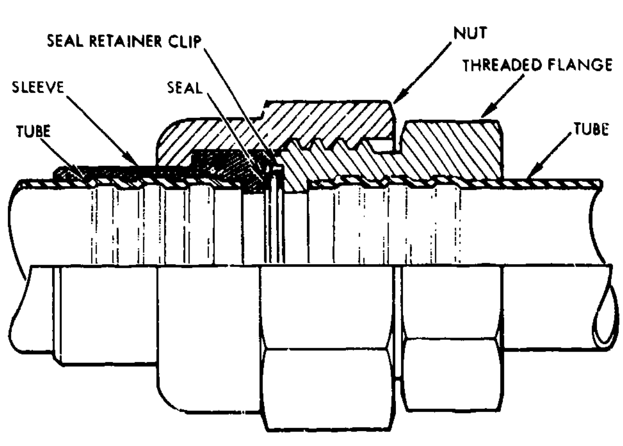


Figure 5.12.3.15c. Astro-Weight Connector Showing Tube-to-Tube Connection
(Courtesy of Harrison Manufacturing Co., Burbank, California)

Rubbernek Connector. This connector, manufactured by the Chicago Fittings Corporation and shown in Figure 5.12.3.15e, is another of the flareless tube type that employs a special five-piece sealing and tube-retaining element in addition to the standard union and connecting nut. The seal is molded from synthetic rubber compounds. In connecting the nut to the union, the retainer and guide force the anchor to clamp the tubing while the seal is forced against the union. The anchor is essentially a belleville spring with a sharp edge that provides the means for tube retention.

Aeroquip Marman Conoseal Connector. This connector, manufactured by Aeroquip Corporation/Marman Division and shown in Figure 5.12.3.15f, is a four-piece unit that has the male flange and the sleeve flange welded to the tubing. The connector nut is loose on the tubing, but is prevented from falling off by the position of the sleeve. The separate seal is of the belleville spring type.

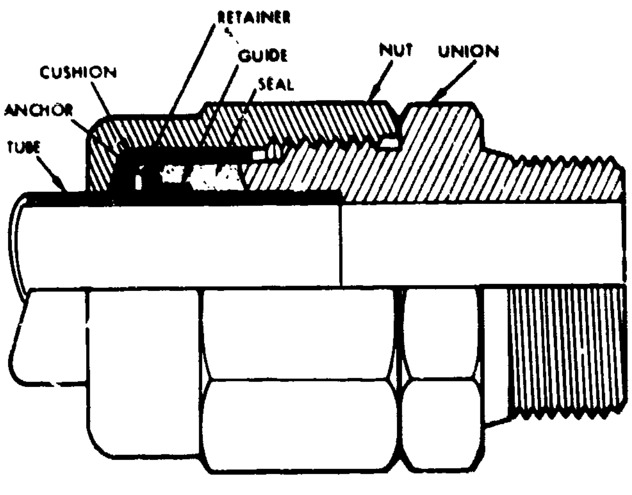


Figure 5.12.3.15e. Rubbernek Connector
(Courtesy of Chicago Fittings Corporation, Broadview, Illinois)

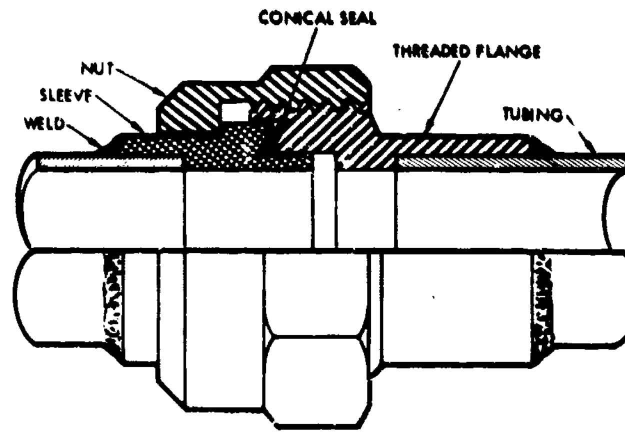


Figure 5.12.3.15f. Aeroquip Marman Conoseal Connector, Showing Tube-to-Tube Connection
(Courtesy of Marman Division, Aeroquip Corporation, Jackson, Michigan)

5.12.4 Bolted Flange Connectors

5.12.4.1 GENERAL DESCRIPTION. Bolted flange connectors incorporate the same three basic elements found in threaded connectors: the seal, the connector-to-line joint, and the load-carrying structure (see Sub-Topic 5.12.3). The outstanding difference between bolted flange connectors and threaded connectors is the number of tensile members in the load-carrying structure. Whereas threaded connectors have a single threaded element, in flanged connectors the tensile load is normally divided between four or more threaded elements. Static seals are usually used between the flanges, while the flange is usually welded or brazed to the line.

Bolted flange connectors can be generally classed as having either integral or separate flanges (see Figure 5.12.4.1). For the same flange material, the integral type will usually be lighter than a corresponding separate flange connector; however, alignment of bolt holes is easier with the separate type flange. Separate flanges are also referred to as loose or floating flanges. In some cases it may be possible to select a higher strength material for the separate flange, since it need not be compatible with the fluid in the system, nor weldable or brazable to the line material. With both integral and separate flange connectors there are numerous possible specific designs utilizing various seal and facing configurations, some of which are illustrated in Figure 6.3.2.4b.

In the following detailed topics, some of the more important factors related to the design and selection of a bolted flange connector for aerospace applications are considered.

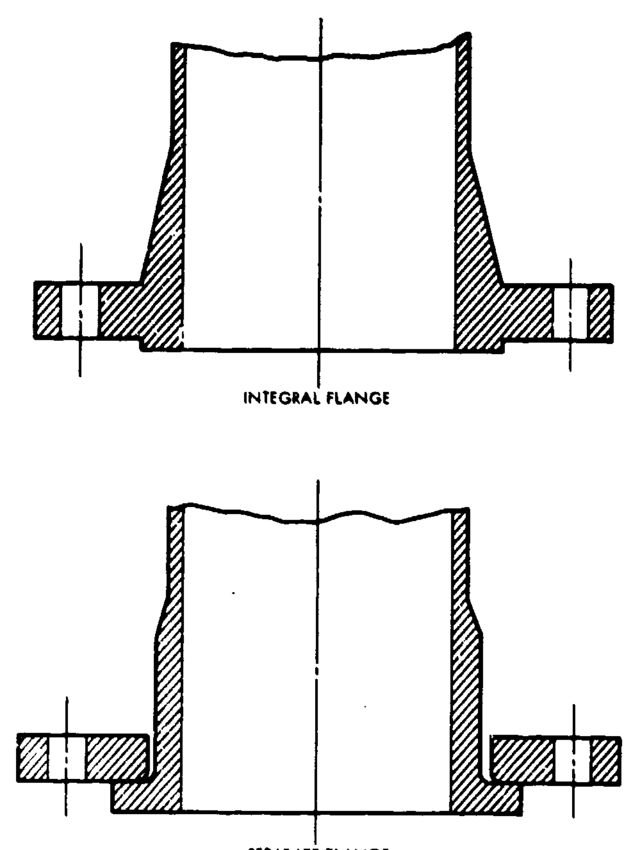


Figure 5.12.4.1. Integral Flange and Separate (Loose) Flange

Those basic separable connector considerations equally applicable to threaded and bolted flange connectors are treated under "Threaded Connectors" as follows:

Factor	Detailed Topic
Threaded Connector Design and Selection Criteria	5.12.3.1
Connector Preload	5.12.3.2
Seal Seating Load	5.12.3.3
Pressure End Load	5.12.3.4
Bending Moment Load	5.12.3.5
External Axial and Torsional Load	5.12.3.6
Vibration	5.12.3.7
Connector Assembly	5.12.3.8
Contamination	5.12.3.9
Temperature and Thermal Shock	5.12.3.10
Additional Factors	5.12.3.11

5.12.4.2 ENVIRONMENTAL AND SERVICE CONSIDERATIONS. Prior to designing or selecting any bolted flanged connector, the anticipated environment and service must be known. Important parameters include:

a) Maximum and minimum operating pressure

b) Maximum and minimum operating temperature

c) Chemical compatibility of flange materials with contained fluid

d) Vibration environment

e) Compatibility of flange and tubing from the standpoint of welding or brazing

f) Required service life, anticipated material creep

g) Maximum thermal gradients anticipated

h) Types and magnitudes of external loads

i) Maximum permissible leakage rate(s)

j) Size and weight limitations.

5.12.4.3 BOLT TORQUING. Bolt torque is determined by flange preload requirements. The preload must be high enough to properly seat the seal and provide greater compressive strain than the expected strain relaxation caused by tensile loads, yet must not be too high so that overstressing of the gasket and/or flange tensile members results. Parallel gasket loading, described in Detailed Topic 5.12.3.2, protects the gasket from over-torquing, but also places extreme importance on seal gland design. Establishing the proper preload value is essential in obtaining seal seating and maintaining a seal during operation. The factors that must be considered in determining the required preload are:

a) Spring constants of the compression and tension members

b) Minimum compressive load on the flange members needed to prevent leakage

c) Maximum allowable stress in the tensile member

d) Magnitude of the structural loads effects on thermal gradients.

A detailed discussion of preload is given in Detailed Topic 5.12.3.2.

Table 5.12.4.3 shows the relationship between bolt stress and torque. It should be noted that this table makes no allowance for friction, therefore the values given represent upper limits, actual values being lower depending on friction. Because of the loss of torque caused by friction, and the variable it introduces, every attempt should be made to minimize friction by lubricating bolts and nuts, and to avoid the use of high-friction self-locking nuts.

The sequence of bolt tightening is discussed in Detailed Topic 5.12.4.8.

5.12.4.4 FLANGE DEFLECTIONS AND DISTORTIONS. Three problems common to bolted flange joints which can be avoided by proper flange design and selection procedures are (1) flange rolling, (2) flange scrubbing, and (3) flange bowing (see Figure 5.12.4.4). Flange rolling describes the tendency of a bolted flanged connector to separate at the sealing surface while simultaneously making more intimate contact outside the bolt circle. Flange rolling may

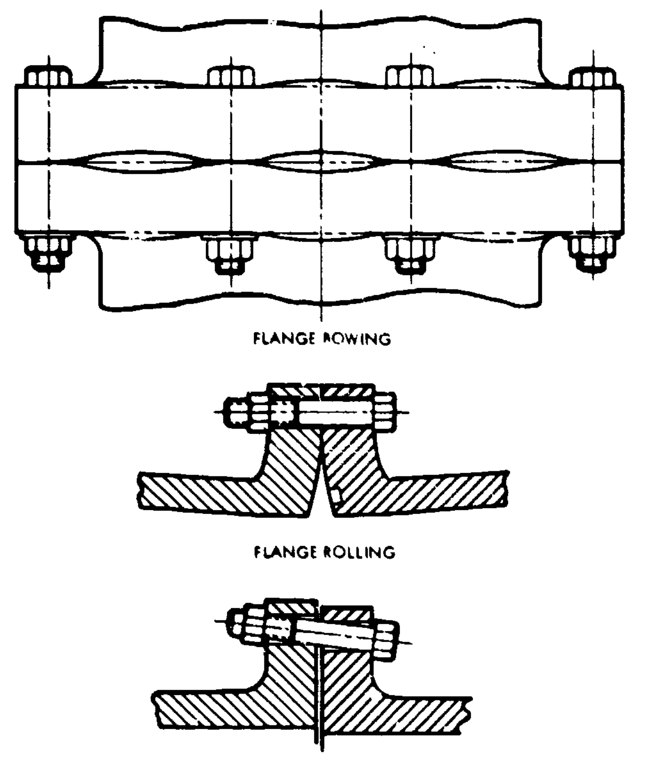


Figure 5.12.4.4. Deflections and Distortions in Flanges

Table 5.12.4.3. Relationship of Bolt Stress to Torque for Various Bolt Sizes

(This table makes no allowance for friction, therefore the values of torque and stress given here represent upper limits)

Bolt Size (in.)	No. Threads per inch: Coarse	Fine	Extra Fine	Root Area (in^2)	Tensile Stress for Torque of 1 in-lb (psi)	Axial Load with Torque of 1 in-lb on Nut (lbs)
1/4	20			0.0269	1,115	20.6
		28		0.0326	879	21.5
			32	0.0352	807	21.7
5/16	18			0.0454	512	16.6
		24		0.0524	432	17.1
			32	0.0590	377	17.6
3/8	16			0.0678	292	14.3
		24		0.0809	261	14.9
			32	0.0890	213	15.3
7/16	14			0.0933	179	12.2
		20		0.1090	150	12.6
			28	0.1217	132	13.0
1/2	13			0.1257	118	10.9
		20		0.1486	97	11.4
			28	0.1634	86	11.7
9/16	12			0.1620	82.3	9.9
		18		0.1888	68.8	10.3
			24	0.2054	63.3	10.5
5/8	11			0.2018	59.3	8.72
		18		0.2400	48.2	9.24
			24	0.2586	44.3	9.33
3/4	10			0.3020	32.1	7.34
		16		0.3513	27.0	7.64
			20	0.3725	25.3	7.68
7/8	9			0.4183	20.4	6.52
		14		0.4805	17.6	6.78
			20	0.5200	16.1	6.92
1	8			0.5510	13.6	5.73
		14		0.6464	11.6	5.99
			20	0.6921	10.6	6.09
1-1/8	7			0.6931	9.61	5.08
		12		0.8118	8.06	5.32
			18	0.8772	7.43	5.43
1-1/4	7			0.8898	6.71	4.62
		12		1.0238	5.73	4.80
			18	1.0969	5.35	4.90
1-3/8	6			1.0541	5.17	4.18
		12		1.2602	4.24	4.40
1-1/2	6			1.2938	3.84	3.86
		12		1.5212	3.31	4.16
			18	1.6101	3.17	4.32

result where tube walls are thin and afford little resistance to rolling moments; where flanges lack sufficient rigidity and consequently deflect under high end loads; or where there is no flange contact outside of the bolt circle, in which case the outermost point of flange contact serves as a fulcrum. Flange rolling can be minimized by making flanges rigid, by making the bolt circle as close to the pipe or tube as possible, and through the use of full face gaskets. The adverse effects of flange rolling can be minimized through the use of raised face flanges.

Flange scrubbing is a condition where there is radial displacement of one flange with respect to the other, i.e., motion of flanges in a direction perpendicular to the tubing axis. This condition can result from differential contraction or expansion of dissimilar flange materials, or from flange misalignment. Scrubbing causes motion at the sealing interface, which can materially degrade the seal.

Bowing is a problem that results from spacing bolts too far apart and/or insufficiently rigid flanges.

5.12.4.5 ALLOWABLE BOLTED FLANGE CONNECTOR STRESSES. In flange design where creep is not a factor, allowable stresses may be related to the short-time yield strength of the material. Yield strength selected as a basis for design should be that at the maximum anticipated service temperature.

An allowable stress for flange material, equal to two-thirds of the material yield strength, conforms to general pressure vessel design practice. (The ASME Code, Reference 68-70, uses 62.5 percent for ferrous materials, two-thirds for non-ferrous materials.) Since in flanged joints the criterion of failure involves excessive deformation rather than rupture, ultimate tensile strength of the material is not directly considered. An allowable stress of two-thirds of the yield strength implies that at a proof test pressure of 1.5 times the operating pressure, the stress may reach the yield strength of the material.

Allowable bolt stresses no higher than 50 percent of the material yield strength are suggested. The allowable stress for bolting is set lower than for the flanges for the following reasons:

a) Bolts are loaded in tension with no ability to transfer load to some other part of the structure in case of overload

b) The only bolt stress calculated is that due to the axial forces whereas, in service, the bolts will have some bending and shear stresses due to rotation of the flanges

c) The threads on the bolts form a notch with accompanying stress intensification

d) In service, the bolts are loaded in initial bolt make-up by tightening with a wrench, which introduces shear stresses and a reduction in axial yield strength.

For a proof test pressure of 1.5 times the operating pressure, it will be necessary to tighten the bolts beyond the recommended 50 percent up to 75 to 90 percent of their yield strength. Typical allowable stress values for several materials based on the above criteria are given in Table 5.12.4.5. It should be noted that materials which obtain their strength by heat treatment may be altered in welding or brazing flanges to the tube. However, if the joining process is such as to limit the heat-affected zone to a small volume of material, design of the bolted flange connection may be based on the heat treatment properties of the material.

5.12.4.6 FACING CONSIDERATIONS. The subject of flange facing types is discussed in Detailed Topic 6.3.2.4, "Gland Design," where seal glands are classified as unconfined, semi-confined, and fully-confined and are illustrated in Figure 6.3.2.4b. Some additional considerations influencing the selection flange facing are discussed in the following paragraphs.

Fully confined (tongue-and-groove) or semi-confined facings aid in locating and holding the gasket during assembly and reduce the possibility of gasket blowout. Also, at least one of the faces is partially protected against damage during handling and assembly of the joints. With metal gaskets, and where low-leakage rates with gases are desired, protecting the flange seating surface during shipment, storage, fabrication, assembly, and possible reassemblies, becomes very important. The ring joint or double groove facing provide particularly good protection of the flange sealing surfaces.

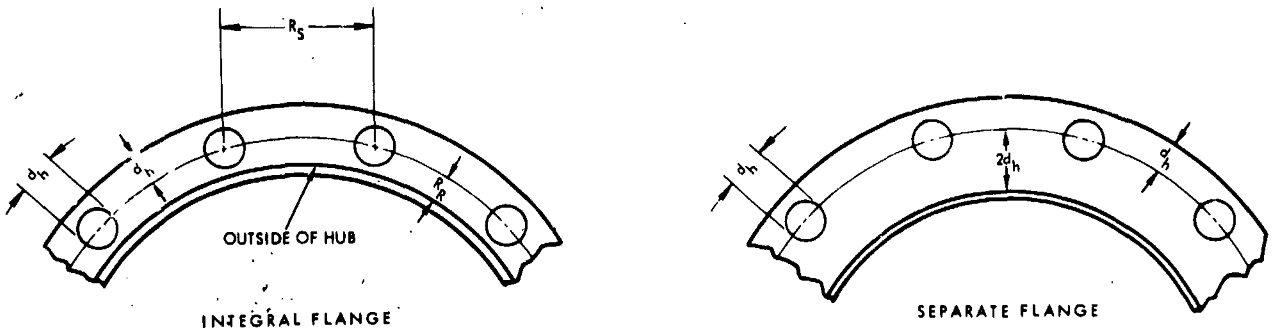
Where the joints will be subjected to high temperatures in service, there is a possibility of partial welding of metal gaskets to the flange faces. Use of dissimilar metals will help to prevent this problem, and in cases where fluid compatibility permits, a graphite lubricant film on the seating surfaces may aid in disassembly. Nonmetallic gaskets will generally harden after long periods of high-temperature service. The raised face configuration has some advantage in this respect, since a much-used gasket of this design can be removed more easily than one with the tongue-and-groove or semiconfined facings.

5.12.4.7 BOLTING DIMENSIONAL DETAILS. Tables 5.12.4.7a and b give pertinent details on bolting suitable for bolted-flanged connections designed for minimum weight. It is desirable to locate the bolts as close to the tube wall as possible. The bolting listed requires a minimum of radial clearance between bolts and flange hub or stub-end wall. In some instances it may be desirable to spot-face the hub or stub end wall to minimize bolt-to-tube wall distance. Socket-head cap screws are suggested in small sizes (No. 5 through No. 12 Am. Std. socket-head cap screws), and NAS 624/636 external wrenching bolts in larger sizes (¼-inch through one inch). NAS-type bolts are suggested for larger sizes, since standard socket-head cap screws may have inadequate bearing areas. The radial and between-bolts minimum clearances shown in the table are based on allowances for fillets, use of standard socket wrenches, and for minimum space between bolts. The "rule-of-thumb" is that the distance between bolt holes should be at least equal to the bolt hole diameter.

Table 5.12.4.5. Typical Allowable Stress Values Using the Following Criteria:
Allowable Stress for Flanges = Two-Thirds Yield Strength
Allowable Stress for Bolts = One-Half Yield Strength

Material (Yield strengths are specified minimums)	Allowable Stress for Indicated Temperature, psi			
	Up to 100 °F	150 °F	200 °F	250 °F
Aluminum alloy, 6061-T6 35,000-psi yield strength 38,000-psi ultimate tensile strength				
Flanges	23,300	23,000	22,700	21,900
Bolts	17,500	17,200	17,000	16,400
Aluminum alloy, 2014-T6 55,000-psi yield strength 65,000-psi ultimate tensile strength				
Flanges	36,700	35,800	34,500	31,200
Bolts	27,500	26,800	25,700	23,400
	Up to 100 °F	300 °F	500 °F	700 °F
Type 347, annealed stainless steel 35,000-psi yield strength 75,000-psi ultimate tensile strength				
Flanges	23,300	20,000	18,900	18,900
Bolts	17,500	15,100	14,200	14,200
Type 301, cold-rolled stainless steel 90,000-psi yield strength 115,000-psi ultimate tensile strength				
Bolts	45,000	40,000	38,000	36,000
	Up to 100 °F	300 °F	500 °F	600 °F
AM 355, precipitation hardened 150,000-psi yield strength 290,000-psi ultimate tensile strength				
Flanges	100,000	85,000	79,000	73,000
Bolts	75,000	63,000	59,000	54,000
	Up to 100 °F	400 °F	600 °F	800 °F
Rene' 41, nickel-base alloy 130,000-psi yield strength 170,000-psi ultimate tensile strength				
Flanges	86,000	84,000	82,000	80,000
Bolts	65,000	63,000	61,000	60,000

Table 5.12.4.7a. Flange Bolting Dimensions for Socket Head Capscrews



Nominal Size* (in.)	Threads per inch	Root Area, A_{BJ} (sq. in.)	Radial Clearance, R_R	Minimum Space, R_S	Hole Size, d_h	Maximum Head Diameter (in.)
#5	44	0.00716	0.132	0.396	0.141	0.200
#6	40	0.00874	0.143	0.429	0.152	0.221
#8	36	0.01285	0.165	0.495	0.180	0.265
#10	32	0.0175	0.186	0.558	0.209	0.306
#12	28	0.0226	0.202	0.606	0.240	0.337
1/4	28	0.0326	0.259	0.657	0.281	0.438
5/16	24	0.0524	0.306	0.796	0.344	0.531
3/8	24	0.0809	0.365	0.974	0.406	0.649
7/16	20	0.109	0.415	1.125	0.469	0.750
1/2	20	0.149	0.464	1.242	0.532	0.828
9/16	18	0.189	0.519	1.407	0.594	0.938
5/8	18	0.240	0.575	1.575	0.687	1.050
3/4	16	0.351	0.665	1.845	0.812	1.230
7/8	14	0.480	0.779	2.157	0.937	1.438
1	12	0.625	0.870	2.437	1.062	1.620

* Sizes #5 through #12, American Standard Socket Head Cap Screws, ASA B18.3-1954

R_R = (Max Head Dia./2) + 0.030 inches

R_S = 3 R_R

Sizes 1/4" through 1" Dimensionally to National Aircraft Standards, NAS 624 through 636

R_R = 1/2 (Maximum Head Diameter) + 0.040 inches for 1/4" through 7/16" nominal size

1/2 (Maximum Head Diameter) + 0.050 inches for 1/2" through 3/4" nominal size

1/2 (Maximum Head Diameter) + 0.060 inches for 7/8" and 1" nominal size

R_S = 1.5 + Maximum Head Diameter

Table 5.12.4.7b. Estimation of Bolt Size and Spacing for External-Wrenching Bolts
(Reference 46-29)

(1) Total Bolt Area (Bolt Circle Radius)			(2)	(3) Root Area			(4)	(5)
Fine (in.)	Coarse (in.)	8-Thread (in.)	Bolt Size (in.)	Fine (in^2)	Coarse (in^2)	8-Thread (in^2)	Bolt Spacing* (in.)	Edge Distance (in.)
0.273	0.225	-	1/4	0.0326	0.0269	-	3/4	3/8
0.406	0.352	-	5/16	0.0524	0.0454	-	13/16	7/16
0.508	0.426	-	3/8	0.0809	0.0678	-	1	1/2
0.609	0.521	-	7/16	0.1090	0.0953	-	1-1/8	9/16
0.748	0.633	-	1/2	0.1486	0.126	-	1.25	0.62
1.007	0.845	-	5/8	0.2400	0.202	-	1.50	0.75
1.262	1.083	-	3/4	0.3513	0.302	-	1.75	0.81
1.464	1.278	-	7/8	0.4805	0.419	-	2.06	0.94
1.743	1.537	1.537	1	0.6245	0.551	0.551	2.25	1.06
2.040	1.740	1.828	1-1/8	0.8118	0.693	0.728	2.50	1.12
2.29	1.988	2.08	1-1/4	1.024	0.890	0.929	2.81	1.25
2.69	2.16	2.37	1-3/8	1.260	1.054	1.155	3.06	1.37
2.94	2.50	2.71	1-1/2	1.521	1.294	1.405	3.25	1.50

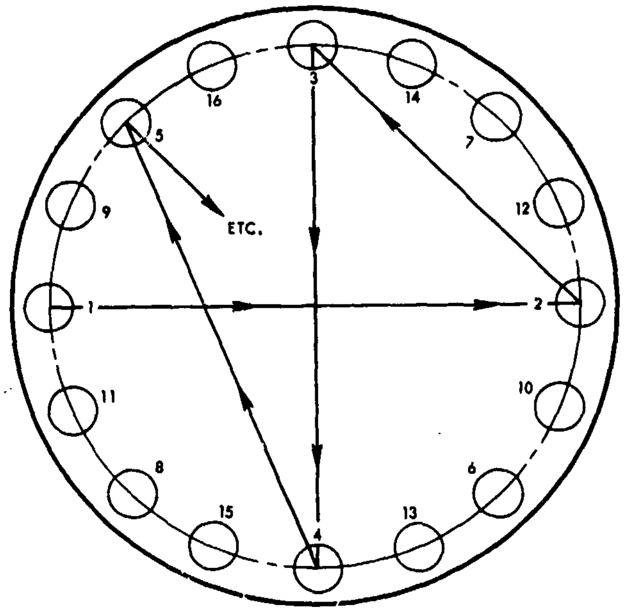
*This spacing can be reduced with internal-wrenching bolts, or by use of special thin-wall wrenches, causing an appreciable reduction in flange weight, since the bolt size can be reduced one or two sizes.

A maximum spacing between bolts equal to $2(d_h + t)$ is suggested, where d_h is the bolt hole diameter and t is the flange thickness. If bolts are too far apart, flange bowing may result in gasket leakage.

Bolting shown in Table 5.12.4.7a may be used with nuts, or the bolts may be studded into tapped holes in a mating flange. The studded design is advantageous from the standpoint of assembly, and provides a slight weight reduction. Use of bolts and nuts also permits easier replacement in case of damaged or seized threads. This is a particularly important consideration if disassembly and re-assembly are required after service at high temperature or under a hard vacuum, since thread seizing under these conditions is a strong possibility.

5.12.4.8 **BOLTED FLANGE CONNECTOR HANDLING AND ASSEMBLY.** Bolt load should be controlled by using a torque wrench when tightening bolts. Since torque-bolt load relationships vary with such factors as material, thread class, surface finishes, and lubricant, the relationship for the particular combination of bolting, surfaces, and lubricant should be checked. The magnitude of bolt torque to be used is covered in Detailed Topic 5.12.4.3.

To facilitate proper assembly, whether using a nut and bolt or a stud-end nut arrangement, it is necessary to clean the threads and to be sure that the nuts do not bind when assembled. Flange seating surfaces and gasket should be checked for cleanliness. The gasket should be carefully placed in position and all bolts inserted and tightened finger tight. Bolts should then be tightened in sequence, as indicated in Figure 5.12.4.8. The number of tightening rounds (the number of times each bolt is tightened in sequence) will vary depending upon the specific application. With some aerospace applications involving o-rings, it is standard practice to torque all bolts to 100 percent of final torque on the first round. In other instances, such as some critical joints using laminated asbestos gaskets, bolts are torqued down in increments of 10 percent of final torque per round. The majority of aerospace applications call for bolt tightening in increments of one-third or one-fourth of final torque per round. It is desirable to recheck the bolt torque after a period of 24 to 48 hours and, if possible, after a short period of operation under service conditions.



TIGHTENING SEQUENCE FOR 16-BOLT FLANGE

Figure 5.12.4.8. Example of Bolt-Tightening Sequence

5.12.4.9 GASKET CONSIDERATIONS. The types of gaskets employed with flanges consist generally of (1) annular ring gaskets, (2) pressure energized gaskets, and (3) full face gaskets. The most widely used gasket for bolted-flange connections is the annular ring type. The pressure energized gasket (gaskets with sealing action aided by internal pressures) are shown in Table 6.3.3.5b, (Sub-Section 6.3). Full face gaskets generally require higher bolt loads than comparable connections using "inside gaskets" (i.e., annular ring gaskets). Disadvantages of full faced gaskets include less tolerance against thermal gradients and a tendency for the load on the gasket to be concentrated at the bolt holes and on the portion of the gasket outside the bolt circle. Full faced gaskets are widely used for mild service conditions, but seldom for severe service conditions involving either high (above 450°F) or low (cryogenic) temperatures, or pressures above 300 psig.

Gasket thickness must be sufficient enough to provide adequate "conformity," since thicker gaskets generally are able to conform to surface scratches and to compensate for sealing waviness. However, gaskets must be thin enough to provide stability and prevent "blowout" when residual stresses are low and internal pressures high.

Two important design criteria related to gaskets and their proper selection are (1) seating stress, and (2) residual stress. Suggested design values for seating stresses are given in Detailed Topic 6.3.3.2.

5.12.4.10 FLANGE DESIGN PROCEDURES. Step-by-step design procedures for bolted flanges, specifically oriented toward aerospace applications, are contained in the following documents:

> "Tentative Separable Connector Design Handbook," Contract NAS 8-4012 Advanced Technology Laboratories, General Electric Company (46-29)
>
> "Development of Mechanical Fittings, Phases I and II," Technical Documentary Report RTD-TDR-63-1115, Battelle Memorial Institute (44-14)

Other references dealing with bolted flanged connector design for general applications are:

> Taylor Forge Bulletin 502, 5th Edition (V-357)
>
> ASME Boiler and Pressure Vessel Code, Section VIII (68-70)

5.12.4.11 RING CLAMP FLANGES. Ring clamps (see Figure 5.12.4.11) were devised in an attempt to duplicate a flanged connection without using a great number of bolts. Generally, the ring clamp is made of two mating flanges with tapered outer surfaces, over which is fitted a split hoop whose inside surfaces are tapered at the same angle as the flange. Small projections with bolt holes are provided where the two halves of the hoop are mated; usually two bolts are used, but if one side of the ring is pivoted, a single bolt may be used. When the bolt(s) are tightened, a clamping force normal to the tapered faces is imposed. The axial component of the clamping force is the only force available for sealing and preloading. The ring clamp is used extensively for many commercial applications and has been used in missile systems, usually for large-diameter, low-pressure, lightweight joints. Its major limitations are:

a) It is difficult to attain high preloads and control initial preload within narrow limits. This is largely due to the friction in the V-band clamp, which results in uneven loading around the circumference.

b) The weight of a fitting of this type would far exceed that of standard-type flanges for high-pressure applications.

Advantages of ring clamp flanges include speed of assembly or disassembly, and the lack of requirements for wrench accessability around the entire perimeter of the connector.

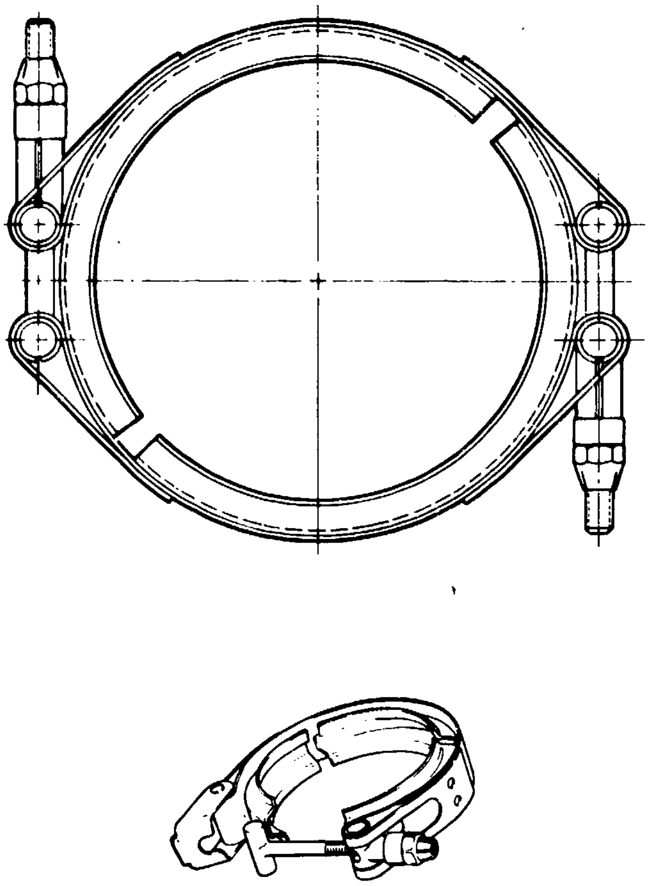


Figure 5.12.4.11. Typical Ring Clamp
(Courtesy of Marman Division, Aeroquip Corporation, Los Angeles, California)

Ring clamp flanges may be expected to perform well in those non-critical low-pressure applications requiring frequent disassembly, especially in large diameters which would require either large numbers of bolts or heavy flanges (for stiffness) if conventional bolted-flange designs were used.

5.12.5 Brazed Joints and Connectors

5.12.5.1 GENERAL. Brazed joints are being used increasingly in aerospace fluid system applications because of their advantages of low leakage under extremes of temperature, pressure, and vibration while providing a capability for disconnecting and rejoining. Brazing is used in making tube-to-connector joints in both threaded and flanged-type connectors, as well as to perform both the joining and sealing functions in brazed connectors. Brazed connectors include a variety of brazed sleeves used for joining tubing. This sub-topic describes various brazed connector configurations, and considers brazed-joint design and installation parameters.

5.12.5.2 THE BRAZING PROCESS. Brazing is a metal-joining operation performed at temperatures ranging between those of welding and soft soldering, with soft soldering temperatures considered to be below 800°F. Brazing differs from welding in that (1) bonding results from wetting rather than melting the base alloy, (2) the brazing filler metal (brazing alloy) is made to flow into the joint capillary, and (3) the brazing filler metal is an alloy having a composition different from that of the metals being joined. Almost all similar metals and alloys, or metals and alloys with similar coefficients of expansion above 1100°F, may be joined by brazing.

The American Welding Society and the American Society for Testing Materials list brazing filler metals under the following seven classifications:

1) Aluminum-silicon (used only on grades of aluminum with relatively high melting temperatures)
2) Copper-phosphorous (used primarily for joining copper and copper alloys)
3) Silver (used on virtually all ferrous and non-ferrous metals except aluminum, magnesium, and several other low melting-point metals)
4) Copper-gold (used primarily in electron tube assembly)
5) Copper and copper-zinc (used for joining both ferrous and non-ferrous metals)
6) Magnesium (used for magnesium alloy base material)
7) Heat-resistant brazing filler metals (used primarily where extreme heat or corrosion resistance is required).

There are other special purpose filler metals in addition to these seven classes, some of which are listed in the Welding Handbook, 4th Edition, Section 3, Chapter 46, (Reference 504-2).

Some of the commonly used filler metals are listed in Table 5.12.5.2, along with their composition and working temperature range.

5.12.5.3 BRAZED CONNECTOR DESIGN FACTORS. Proper design (or selection) of a brazed connector demands consideration of several factors, some of the more important of which are discussed in the following paragraphs.

Braze Alloy Compatibility with Tube, Connector, and Fluids. Compatibility of a braze alloy with tubing and connector materials should be considered in terms of wettability, flow, and corrosion. The wettability and flow characteristics of candidate brazing alloys may be determined by heating small specimens of the tube and fitting materials on which have been placed samples of the brazing alloy. The specimens are heated to progressively higher

Table 5.12.5.2. Composition, Melting Temperatures and Brazing Temperatures of Common Brazing Alloys
(Reference 320-7)

Brazing Alloy	Chemical Composition (Percent)										Melting Temperature Range (°F)	Minumum Recommended Brazing Temperature (°F)
	Au	Ag	Ni	Pd	Cr	Cu	Si	Li	B	Fe		
72Ag-28Cu-Li	-	71.8	-	-	-	28	-	0.2	-	-	1410	1450
72Au-22Ni-6Cr	72	-	22	-	6	-	-	-	-	-	1785 to 1835	1950
35Au-3Ni-62Cu	35	-	3	-	-	62	-	-	-	-	1787 to 1886	1890
82Au-18Ni	82	-	18	-	-	-	-	-	-	-	1742	1850
Ni-Cr-B	-	-	81.5	-	15	-	-	-	3.5	-	1930	2150
Ni-Cr-B-Si-Fe	-	-	83.5	-	6	-	5	-	3	2.5	1830	1900
60Pd-40Ni-0.3Li	-	-	40	59.7	-	-	-	0.3	-	-	2100	2150
82Au-18Ni-Li	81.7	-	18	-	-	-	-	0.3	-	-	1710	1825
70Au-22Ni-8Pd	70	-	22	8	-	-	-	-	-	-	1825 to 1910	1950

Table 5.12.5.3. Wettability and Flow of Braze Alloys
(Reference 320-7)

Base (Tube) Material	Brazing Alloy	Wettability	Flow
AISI Type 321 or 347 Stainless Steel	72Ag-28Cu-Li 72Au-22Ni-6Cr 82Au-18Ni	Good Good Good	Good Fair Fair
AM 350 Stainless Steel	72 Ag-28Cu-Li 82Au-18Ni-Li	Good Excellent	Good Excellent
Rene' 41	72Au-22Ni-6Cr 35Au-3Ni-62Cu 82Au-18Ni 60Pd-40Ni-0.3Li Ni-Cr-B Ni-Cr-B-Si-Fe 82Au-18Ni-Li 70Au-22Ni-8Pd	Good Good Good Good Fair Fair Excellent Excellent	Fair Fair Fair Good Poor Poor Excellent Excellent

temperatures in a controlled atmosphere furnace and, after the desired temperature is reached, are withdrawn from the furnace in order to evaluate the melted braze alloys. Unaided visual examination will reveal how well the braze alloy flows, and to what degree the specimen tube metal surface has been wetted. Micrographic examination will show how well the braze metal and the base metal have joined. The wettability and flow characteristics of several brazing alloys and companion tubing materials are presented in Table 5.12.5.3. Although the results of this type of test are qualitative in nature, they establish both the wetting compatibility and an approximate brazing temperature of the braze alloy and tube materials.

The effect of braze alloy on the base metal is of particular concern when brazing 300-series stainless steels, for the use of certain braze alloys with stainless steel can lead to intergrannular attack of the steel by the alloy. Another problem in selecting a brazing alloy is the possibility of carbide precipitation, caused by the alloy, which results in reduced corrosion resistance. Unless special protection is to be provided, the braze alloy selected must be compatible with the fluids with which it is to be used, just as the tube and connector materials must be compatible.

Strength of Brazed Joint. For a particular application, the selection of a specific alloy s ould be based upon experience of previous users, or upon results of a preliminary block-shear test, conducted to determine the joint strength and using the specific metals involved. Details on how these tests may be conducted are included in Reference 320-7. Figure 5.12.5.3a is a plot of shear strength versus temperature for palladium-nickel-lithium, gold-nickel, and silver-copper based braze alloys. These alloys are used in design of tube joints for rocket propulsion fluid systems. The figure shows that the palladium-nickel based alloy exhibits greater strength than the silver and gold based alloys.

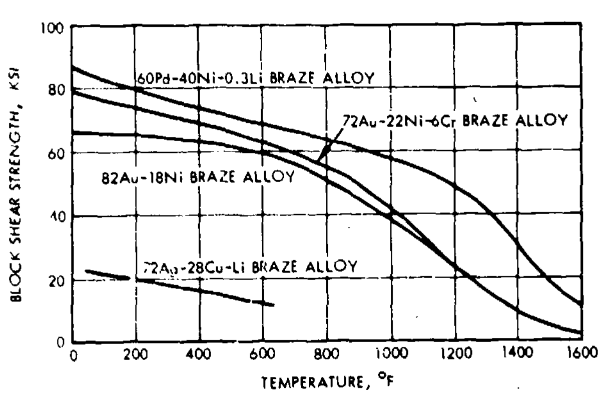


Figure 5.12.5.3a. Block/Shear Strength Versus Temperature Properties of Braze Alloys
(Reference 320-7)

Connector Wall Thickness. The wall thickness of brazed joint sleeves will vary depending upon the design configuration. Sleeves which have no braze alloy retention grooves, or grooves which are outboard of the prepared surfaces, need not have wall thicknesses any greater than the wall thickness of the tubing being joined. Sleeves which have braze alloy retention grooves cut in the middle of, or inboard from, the prepared surfaces should generally have a wall thickness equal to or greater than the tubing, to compensate for the stress concentrations caused by the braze alloy grooves.

Connector Length. The lengths of the brazed connector sleeve and the prepared surfaces are critical to the strength of the brazed joint. Primary concern is that the joint be able to withstand bending loads, together with axial loading, which tends to pull the tube ends out of the sleeve. This combined loading condition is the result of stresses produced by a vibration and/or impulse environment while the tubing is under full internal pressure. Development of the maximum percentage of tube strength under flexure and impulse loading at elevated temperatures is probably the most difficult test which brazed fittings must pass. Under such conditions the tubing tends to fail just at the point where it enters the fittings. To overcome this tendency, (1) the ends of the fittings are tapered or stepped to relieve stress concentrations, and (2) the flow of brazed alloy is usually stopped short of the end of the connector by concentrating the brazing heat towards the center of the sleeve, or by using sleeve locating lands for capillary control. Concentrating the heat toward the center helps to prevent thermal degradation of the strength properties of the tubing at the location where it enters the connector sleeve.

The shear strength of the brazing alloy and accessability give a first approximation for sizing the connector sleeve length.

Tube Sizing. Machine sizing of tubing and connectors is generally required to control the variation in diametrial spacing between the braze connector and the tube OD. Sizing is also necessary because the tolerances on diameter and roundness of commercial tubing are not within those required to form a satisfactory capillary gap between the brazed connector and tubing wall.

Tube sizing can be accomplished satisfactorily in a number of ways, such as by the use of hydraulic presses and sizing dies, swaging operations, machining, and hand fitting. The first three methods require the use of fairly heavy forming equipment, used primarily for in-shop work. In field maintenance and repair operations, however, hand sizing methods using portable sizing tools are generally employed. One configuration of hand sizing tool using an explosive cartridge is shown in Figure 5.12.5.3b. The energy required to size the tubing is obtained by the expansion of gases from a .22, .32, or .38 caliber cartridge. High energy tube sizing tools such as this can be used in the field, providing normal safety precautions are observed. The tools may be fitted with a variety of split dies

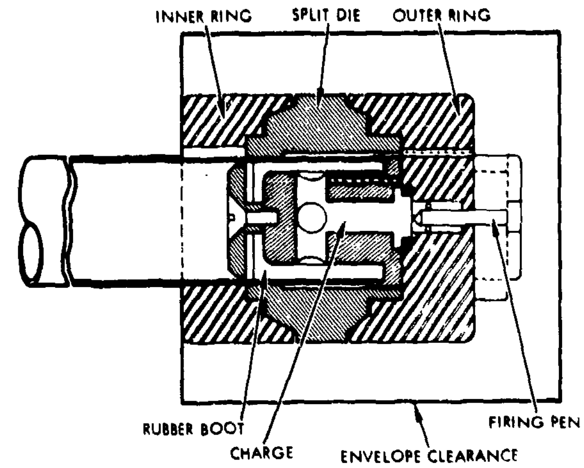


Figure 5.12.5.3b. Portable High-Energy Tube-Sizing Tool
(Reference 320-7)

to hold the tubing to correct diameter and wall thickness. These tools are designed to size tubing to 0.010 (+0.003, −0.000) inches above the nominal tubing diameter. Sizing tools have been made for tubing up to approximately 2 inches in diameter with wall thicknesses up to approximately 0.065 inches. For heavier wall thicknesses sizing is usually accomplished by machining.

Clearance Between Members of the Joint. The clearance between members of the joint determines the thickness of the alloy film that will be formed between the parts during brazing, and thus has an important influence on the joint strength. The clearance must be large enough to allow entrance of the molten braze alloy, and escape of the molten flux and gases developed during heating. On the other hand, if the joint clearance is too large, the braze alloy will not flow in the joint by capillary attraction. Clearance should normally be in the range of 0.001 to 0.005 inches, and should be maintained at the brazing temperature to get proper flow.

When determining what clearances to use for brazing dissimilar metals, the thermal expansion of the parts must be considered, allowing sufficient clearance for entrance of the braze alloy at the brazing temperature.

Line Size. The line size limitation for brazed connectors depends on design parameters which include:

a) Total expected end loading
b) Braze alloy material available
c) Shear strength of braze alloy
d) Power requirements to accomplish a satisfactory braze joint.

While brazed tube connections have been made in line sizes ranging up to six inches in diameter, non-aerospace applications have utilized brazing of much larger diameter joints, such as in the copper ducting of steam condenser cooling water systems. In considering the application of brazed connectors to any particular line size, the expected design end loads will determine the lower limits of strength required by the brazed joint, making it possible to provide a suitable braze alloy for these strength limitations. Once this information has been obtained, it is then necessary to determine whether or not sufficient heat and/or power can be provided to accomplish the brazing operation. Heat rate and power requirements are discussed further in Detailed Topic 5.12.5.4.

Theoretically, there is no upper limit of line diameter for brazed joints, nor for the length of the braze sleeve. Normal practice, as recommended in Reference 320-7, is for braze sleeve length not to exceed 1.5 times the line outside diameter. For any given braze sleeve length-to-diameter ratio, it is apparent that the maximum system pressure is not a function of line diameter, since the total shear area of the connector will increase as the line diameter increases, at the same rate as will the cross-sectional area of the line. At the larger diameters, however, it may be found that the problems associated with applying the necessary heat to accomplish the brazing operation will limit line diameters more often than will the length of braze sleeve and the larger number of braze rings required with longer sleeves.

5.12.5.4 BRAZING PARAMETERS. Important brazing parameters which affect considerably the degree of success in making a brazed tube joint are:

a) Parameters Independent of Material Properties
   - Surface cleanliness
   - Atmosphere control
   - Heating rate and power requirements

b) Parameters Dependent on Material Properties
   - Connector sleeve design
   - Induction heating coil design (or other heat source)
   - Tube sizing

Those parameters independent of material properties are discussed in the following paragraphs.

Surface Cleanliness. An effective tube braze cannot be made unless conditions of absolute cleanliness are maintained. Impurities or foreign deposits on surfaces to be brazed will decompose upon heating and give off contaminating and bubble forming gases, thus leaving residues which prevent adhesion of the brazing alloy. Procedures for prebraze cleaning of high strength stainless steel alloys are outlined in Reference 320-7.

Atmosphere Control. An inert gas envelope or fluxing is necessary for tube brazing, in order to minimize oxidation of the metal surfaces and provide maximum possible adhesion of the brazing alloy. Any one of several different methods may be used for inert gas shielding, all of which are generally based upon providing a plenum chamber of some form around the brazed area, which is then purged and kept under a positive pressure with inert gas such

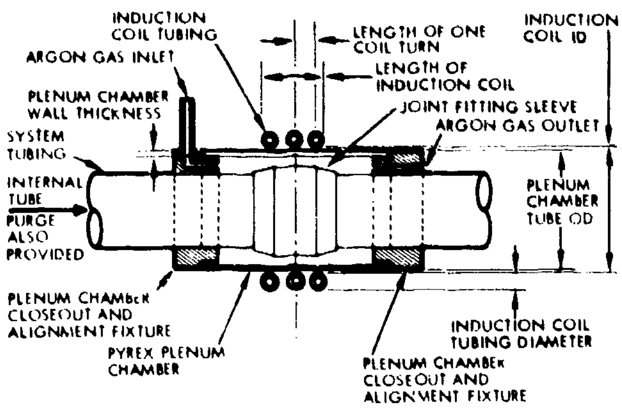


Figure 5.12.5.4. Schematic of Braze Tube Joining Induction Coil and Plenum Chamber
(Reference 320-7)

as nitrogen, helium, or argon (see Figure 5.12.5.4). In addition, it is usually necessary to purge the inside of the tubing with the same inert atmosphere. Inert gas pressure inside and outside of the brazed joint should be the same, so that no gas can pass through the molten brazing alloy, causing voids or the expelling of molten brazing alloy from the joint capillary.

The inert gas used for purging during brazing operations must be free of moisture. If even small amounts of water are present in the purge gas, the continuous flow introduces enough moisture in the braze area to create a serious contamination problem. To prevent this, a drying train should be used in the inert gas feed systems to insure that not more than 10 parts per million of moisture remain.

During the braze operation, a continuous low flow of inert gas should be maintained to sweep off any adsorbed gases which may be released from the metal surfaces upon heating, and to prevent possible leakage of air into the brazed area. The gas flow rate should be continued even after the brazing operation has taken place, and during cooling of the joint, to prevent air from entering the area upon contraction of the cooling inert gas.

Another method for controlling brazing atmosphere is by use of a brazing flux. Upon the application of heat, the flux decomposes, forming a gaseous envelope around the braze area. A disadvantage of using flux for areospace fluid system brazing is the difficulty of removing the flux residue from the brazed joint, particularly from the inside of the tubing. If the flux is not removed, it serves as a nucleus for corrosion and can generate particulate contamination in the fluid system. An important advantage of using flux is that it greatly simplifies the brazing operation by eliminating the cumbersome inert gas system and stringent prebraze joint cleaning requirements. Self-fluxing braze alloys, such as those containing lithium, are available which eliminate the necessity of using a separate brazing flux.

Heat Rate and Power Requirements. Induction heating is currently the most widely used method of generating heat for in-place fluid systems brazing. Methods such as resistance heating, gas torches, furnaces, and exothermic reactions, are also used, but to a lesser extent. Induction heating is a process by which heat is produced in a metal which is in proximity to a rapidly varying magnetic field produced by an alternating current. The heat is generated by the resistance of the metal to the flow of eddy currents induced by the varying magnetic field, and also because of hysteresis effects in the metal. The required magnetic field is produced by conducting high frequency current through a work coil, or inductor, which acts as the primary winding of a simple transformer, while the workpiece acts as a secondary. By shaping the coil properly, the heat can be localized, or spread throughout the work, as required. The work coil may consist of a single turn, or of many turns, depending on the desired heat pattern, the work material, the current frequency, and the distance from the coil to the work. Heat distribution patterns can be obtained which are not possible with conventional methods of heating.

When performing initial production brazing runs on a specific material and tubing system, power settings of the induction heating machine should be kept sufficiently low, to insure uniform heating of the tube and connector assemblies. Slow heating rates will allow time to observe visually the wetting and flow action of the brazing alloy. Once the characteristics of the materials have been established, heating rates can be increased and times reduced to the point where effective brazes can be made in seconds. Table 5.12.5.4 gives some typical brazing times for several aerospace tubing and connector materials.

The choice of frequency in induction heating depends upon the particular application. The degree to which the induced currents and, in turn, the heating effect, penetrate the work is generally an inverse function of the frequency of the applied alternating current. Frequencies from 60 cps into the megacycle range have been used for induction heating. The higher frequencies from radio frequency (RF) generators, in the approximate range from 200,000 to 450,000 cps, usually produce a very intense, fast, and localized heat pattern, desirable for brazing thin-walled tubing. A more diffuse and slower heating effect with a deeper heating penetration is produced by the lower motor-generator frequencies in the range from 1000 to 10,000 cps. The choice of a particular frequency to be used for a given application may be determined by the characteristics of the equipment available, rather than by more strictly technical consideration (see Reference 320-7). In many cases it is not critical, and almost any frequency may be used.

5.12.5.5 JOINT REBRAZING. Experience with brazed joints to date indicates that debrazing and rebrazing of joints can be accomplished easily and result in a satisfactory joint, providing the necessary steps are taken to insure that oxidation of the joining surfaces does not occur. The number of times which a joint may be rebrazed depends considerably upon the brazing alloys involved and their

Table 5.12.5.4. Brazing Parameters for Induction Braze Joining of Tubing
(Reference 320-7)

Tubing Material	AISI 347 Stainless Steel	AISI 347 Stainless Steel	AM 350 CRT Stainless Steel	AM 350 SCT Stainless Steel	Rene' 41 Alloy
Tube Size:					
Outside diameter	1.000	3.000	0.250	1.000	0.125
Wall thickness	0.083	0.250	0.042	0.134	0.010
Fitting Sleeve Material	AISI 347 Stainless Steel	AISI 347 Stainless Steel	AM 355 SCT Stainless Steel	AM 355 SCT Stainless Steel	Rene' 41 Alloy
Composition of Brazing Alloy	71.8 Ag 28.0 Cu 0.2 Li	71.8 Ag 28.0 Cu 0.2 Li	81.7 Au 18.0 Ni 0.3 Li	81.7 Au 18.0 Ni 0.3 Li	81.7 Au 18.0 Ni 0.3 Li
Brazing Control Temperature*	1450 F	1500 F	1500 F	1450 F	1900 F
Brazing Heating Time	47 seconds	360 seconds	15 seconds	45 seconds	45 seconds
Induction Heating Machine Information:					
Frequency	250 Kc	250 Kc	450 Kc	250 Kc	450 Kc
Rated capacity	30 Kw	30 Kw	2.5 Kw	30 Kw	2.5 Kw
Power setting	20 percent	45 percent	75 percent	35 percent	65 percent
Plate voltage	3.5 Kv	6 Kv	2.7 Kv	4 Kv	2.3 Kv
Plate amperage	1.8 amperes	2.3 amperes	0.7 amperes	2.0 amperes	0.7 amperes

*Control temperature measured by thermocouple tack welded to tube OD 1/32 inch from edge of fitting sleeve

susceptibility to oxidation. With gold-nickel alloys, rebrazing can be accomplished as many as 5 to 8 times with a satisfactory joint resulting each time. With lead-nickel-lithium brazing alloys, however, rebrazing using the original alloy is difficult because the lithium in the brazing alloy dissipates with heat. In this case, the lithium (which serves as a volatile flux and improves wettability and flow characteristics of the alloy) is dissipated by evaporation. Rebrazing results in poor wetting and flow characteristics and, thus, an unsound joint. For any brazing alloy containing small amounts of an essential ingredient which can be dissipated or otherwise degraded by reheating, it is not recommended that rebrazing be attempted.

Repairs of brazed joints can be accomplished successfully in ways other than rebrazing. For example, a repair can be made by completely removing the defective joint by cutting the tubing at both sides of the joint, then replacing the cut-out section by brazing a new section in place using new connectors, as required. Although successful, this method does add weight to the fluid system. Where weight considerations are overriding it may be desirable to rebraze the original joint.

Rebrazing should be undertaken only when absolutely necessary. New braze alloy should be used when possible, even when the original connector sleeves are reused because of mechanical loss and degradation of the original brazing alloy in the reheat cycle and during the cleaning operation. Reasons why rebrazing is not desirable are:

a) Repeated heating exposes the joint components to more opportunities for contamination, oxidation, and adverse heat treating.

b) Fluxing agents in the original braze alloy become depleted either by evaporation or by a reaction with the surface oxides on the joint components, and effect on the brazing operation is lost.

c) At elevated temperatures there is a gradual diffusion of the braze alloy into the base metal, which changes the composition and properties of both the remaining alloy and the joint component materials. These material changes can be detrimental both from a strength and corrosion standpoint.

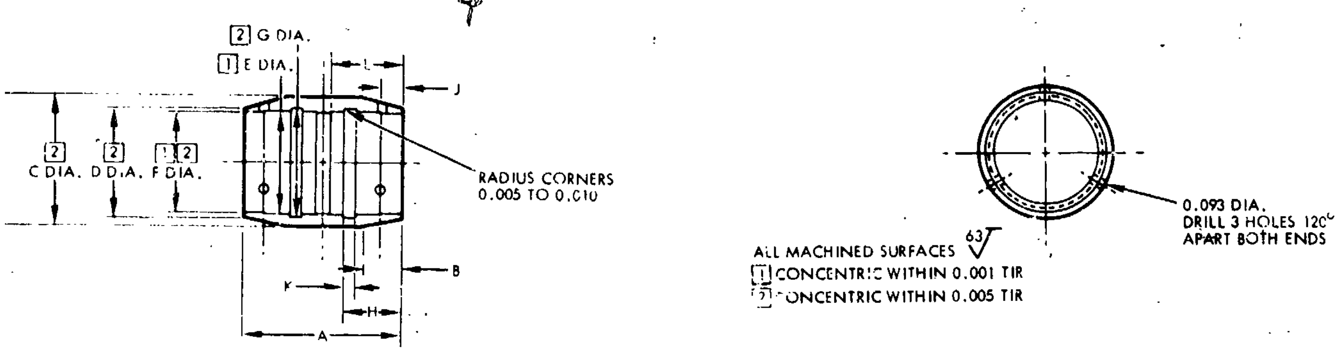
d) Extended times at brazing temperatures can produce damaging metallurgical changes such as carbide precipitation in certain stainless steels, or overaging and loss of strength in precipitation hardening materials such as AM 350 and René 41.

Further information on rebrazing may be obtained from Reference 320-7.

5.12.5.6 BRAZED JOINT INSPECTION. Probably the single most important objection to the use of brazed connectors is the difficult problem of inspecting joints for integrity, following the brazing process. The use of visual inspection holes in a connector only assures proper flow of the braze alloy in the joint (see Figures 5.12.5.7c and d, and Table 5.12.5.7). The most reliable methods developed to date for inspecting the integrity of joints involve the use of X-ray and sonic inspection equipment. Complete inspection using X-rays requires two photographs taken at positions 90 degrees apart. Because both the near and far sides of the tubing joint will appear on the X-ray plate, considerable skill is required of the inspector to determine in which quadrant a defect may be located. In general, the main item which shows up in the X-ray photograph is the extent of alloy distribution and existence of voids on the prepared surfaces of the joint. The X-ray photograph, in effect, indicates the extent of braze alloy wetting, but it does not indicate whether a satisfactory bond exists. This can be determined by use of an ultrasonic inspection technique which involves the transmission of high frequency sound to the brazed joint and monitoring of the time interval and frequency of the sound echo. If a solid joint has been

Table 5.12.5.7. Dimensions of Brazed Connector Sleeves

(Reference 320-7)



Sleeve Material	Tube OD	Sleeve Dimensions										
		A	B	C	D	E	F	G	H	J	K	L
AISI Type 347 Stainless Steel	1	1.500 ±0.005	0.375 ±0.010	1.255 ±0.005	1.082 +0.005 -0.000	1.002 +0.001 -0.000	1.0005 +0.001 -0.000	1.055 +0.005 -0.000	0.550 ±0.005	0.195 ±0.005	0.100 +0.005 -0.000	0.6875 +0.005 -0.000
	3	3.335 ±0.005	0.875 ±0.010	3.720 +0.010 -0.000	3.130 ±0.010	3.003 +0.001 -0.000	3.001 +0.001 -0.000	3.200 +0.005 -0.000	1.200 ±0.005	0.245 ±0.005	0.205 +0.005 -0.000	1.550 ±0.005
AM 355 SCT Stainless Steel	1/4	0.559 ±0.005	0.156 ±0.010	0.400 +0.005 -0.000	0.321 +0.005 -0.000	0.254 +0.001 -0.000	0.251 +0.001 -0.000	0.276 +0.005 -0.000	0.223 ±0.005	0.100 ±0.005	0.040 +0.005 -0.000	0.245 +0.005 -0.000
	1	1.500 ±0.005	0.375 ±0.010	1.304 +0.005 -0.000	1.082 +0.005 -0.000	1.007 +0.001 -0.000	1.004 +0.001 0.000	1.055 +0.005 -0.000	0.550 ±0.005	0.195 ±0.005	0.100 +0.005 -0.000	0.6875 +0.005 -0.000
René 41	1/8	0.500 ±0.010	0.125 ±0.010	0.187 ±0.005	0.158 +0.005 -0.000	0.128 +0.001 -0.000	0.128 +0.001 -0.000	-	-	-	-	-

made, the echo will originate from the inside wall of the tubing, but if a poor joint or lack of bonding exists, the echo will emanate from the unbonded surface between the connector and the outside wall of the tubing. By knowing the precise distance from the sound emitter to the brazed joint, and accurately determining the time interval between signal transmission and receipt of the echo, it can be determined quite reliably if a satisfactory bond exists at the joint.

Although usable information regarding the structure of the brazed joint is obtained from X-ray and ultrasonics, only a final leak test and proof pressure test can determine the brazed joint integrity.

5.12.5.7 BRAZED JOINT AND CONNECTOR CONFIGURATIONS. A brazed connector is basically a sleeve which fits over the ends of the tubes to be joined. The sleeve may contain internal grooves for the braze alloy rings. Figure 5.12.5.7a illustrates the use of braze alloy rings in various locations in several typical brazed tube joints.

Brazed connectors generally assume one of two shapes. The first, illustrated in Figure 5.12.5.7b, is a simple through bore connector with the braze alloy ring butted between the ends of the tube to be brazed. The second, shown in Figure 5.12.5.7c, contains two braze alloy rings in the sleeve and employs three capillary clearance lands.

The diameter and length of the braze alloy retention grooves must be selected so that the notch effect is minimized and fracture of the connector sleeve under load is avoided. The dimensions of the grooves must also be chosen to provide for the containment of sufficient braze alloy to assure complete filling of the joint capillary under any brazing condition. A braze alloy groove volume of approximately three to seven times the maximum joint capillary volume is recommended.

Because of difficulties which would be encountered in machining the connectors and assembling the joint, it is recommended that one-quarter-inch diameter tube be the smallest size using internal braze alloy retention grooves. The braze alloy retention groove should always be located slightly closer to the interior end of the joint capillary than to the outer end. Thus, if the braze alloy is observed to have flowed the longer distance to the edge of the sleeve, some assurance is provided for a full braze alloy flow to the sleeve center. Inspection holes are usually placed in the connector sleeve to permit a more complete evaluation of the extent of alloy flow.

The capillary clearance lands at the center and ends of the connector shown in Figure 5.12.5.7c serve to align the tube ends as well as provide a positive capillary dimension in the assembled joint. The locating lands should have sufficient area to withstand the socket loads applied by misalignment of the installed tubing. When a sleeve is designed with locating lands, the overall length of the sleeve will be the length of the braze capillary plus the total length of the locating lands. Dimensions of grooved bore connector sleeves are shown in Table 5.12.5.7.

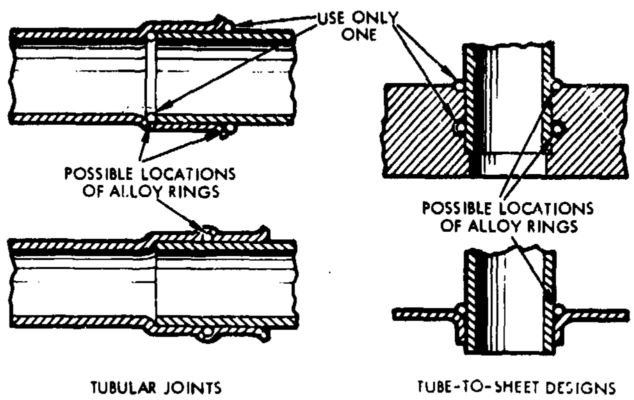


Figure 5.12.5.7a. Typical Designs for Brazed Joints
(Reference 46-16)

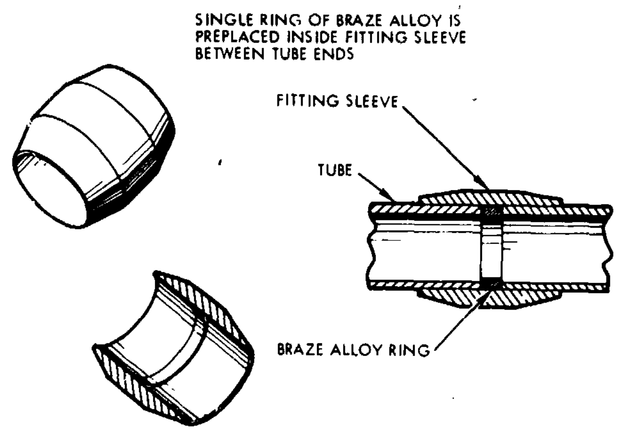


Figure 5.12.5.7b. Straight-Through Bore Connector Sleeve
(Reference 320-7)

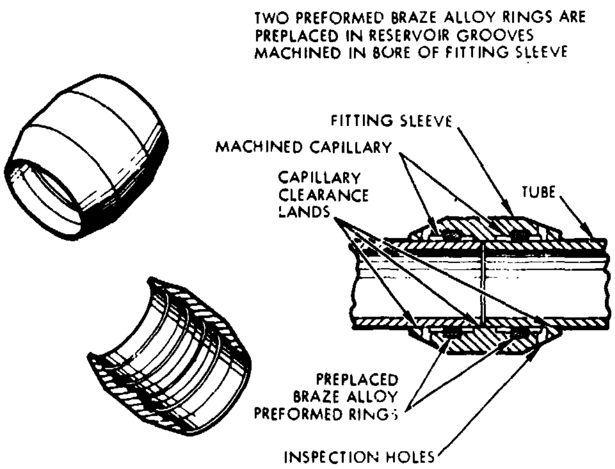


Figure 5.12.5.7c. Groove Bore Connector Sleeve
(Reference 320-7)

Two other variations of connector sleeves are shown in Figures 5.12.5.7d and 5.12.5.7e, both employing the use of preplaced brazing alloy and alloy retention grooves. The configuration in Figure 5.12.5.7d is generally restricted to lightweight tubing which can be upset beaded to form the brazing alloy groove. The configuration in Figure 5.12.5.7e is a machined connector which employs an exothermic reaction pack (shown) to provide brazing heat. The primary advantage of this connector is that in-place joints can be made in remote or limited space locations with no equipment other than the reaction pack and a 6-volt battery. Also, because of the fast reaction time and rapid temperature rise in exotherm brazing, the deleterious effects of time-at-temperature on heat treatment and grain structure of the tubing are reduced. This connector is presently commercially available for use with one-half-inch OD tubing.

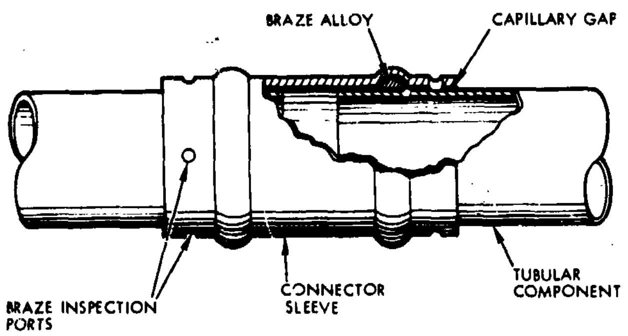


Figure 5.12.5.7d. Brazed Connector Featuring Upset Bead Braze Alloy Retention Grooves
(Courtesy of Marman Division, Aeroquip Corporation, Jackson, Michigan)

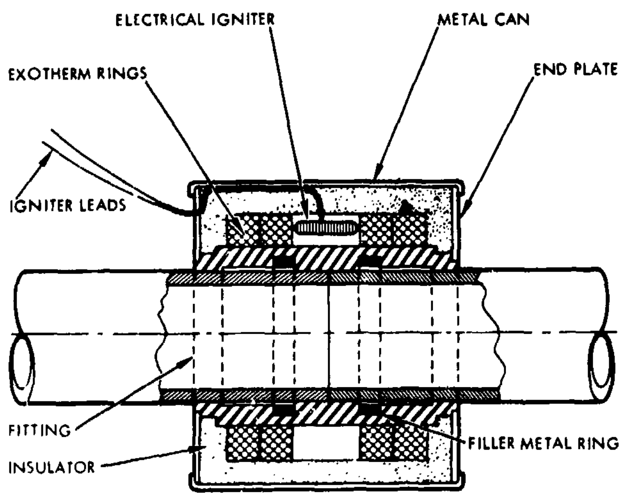


Figure 5.12.5.7e. Exotherm Brazed Connector
(Courtesy of The Deutsch Co., Los Angeles, California)

5.12.6 Welded Joints and Connectors

5.12.6.1 GENERAL. Welded connectors provide minimum weight, leak-tight joints with maximum joint strength, and avoid the dissimilar metals compatibility problems associated with brazed joints. The primary drawback of a welded connector or joint is that due to its permanent nature, physical cutting or machining is required for separation. Basic information related to the design of welded connectors and joints, as well as applicable tube weld methods, is discussed in this sub-section.

5.12.6.2 THE WELDING PROCESS. Welding is the process of joining two pieces of metal by establishing a metallurgical bond between each piece. Welding is accomplished by the application of heat and/or pressure, leading to the classification of (1) pressure welding, and (2) fusion welding. Pressure welding may be accomplished with or without a heat source; however, the time required to accomplish the weld is lessened with a heat source. The temperature requirements for pressure welding normally fall in the plastic forging range, as opposed to the melting temperature range for fusion welding.

5.12.6.3 WELDED CONNECTOR DESIGN FACTORS. The designer must be aware of a number of factors when considering the use of welded connectors and/or joints in a tubing or piping system. The factors which follow are considered to be of major significance.

Connector Material. The type of materials to be welded is the most important factor in any weldment, dictating the type of welding process(es) available. For example, when welding lines and connectors made of ferrous materials, consideration must be given to:

a) The tendency of the weld metal and/or base metal to crack during the welding operation or while cooling after welding
b) Changes in mechanical properties of weld or base metals due to welding process
c) Changes in metallurgical structure of base metal due to welding
d) Changes in chemical composition of base metal due to volatilization of alloying components, air oxidizing, or reactions with shielding gases.

One of the most commonly used materials for aerospace tubing applications is austenitic stainless steel. For this class of metals, Table 5.12.6.3 gives recommendations for electrode or welding rod types to be used with various grades of stainless steel.

Tube Thickness. Tube thickness influences the selection of weld method. Thick sections that require multiple passes to form a weld seam are usually best welded with an inert-gas shielding-arc using a consumable electrode. Sections of lesser thickness in the one-sixteenth to three-sixteenth-inch range may be welded by a number of processes, depending on the overall size and maneuverability of the workpiece. Detailed Topic 5.12.6.4 further discusses this aspect of welding.

Table 5.12.6.3. Welding Rod or Electrode Recommendations for Wrought Austentitic Stainless Steel Pipe Grades

(From "Welding Handbook," 4th Ed., Section II, American Welding Society, New York, 1962)

AISI Designation	Popular Designation	Recommended Electrode or Welding Rod*
304	19-9	E, ER 308
304L	19-9 (extra low carbon)	E, ER 308L
309	25-12	E, ER 309
309 Cb	25-12 Cb	E 309 Cb
310	25-20	E, ER 310
310 Cb	25-20 Cb	E, ER 310 Cb
310 Mo	25-20 Mo	E, ER 310 Mo
316	18-12 Mo	E, ER 316
316L	18-12 Mo (extra low carbon)	E, ER 316L, or ER 318
317	19-13 Mo	E, ER 317
318	18-12 Mo Cb	ER 318
321	18-8 Ti	E, ER 347, ER 321
347	18-8 Cb	E, ER 347
348	18-8 Cb	E, ER 347, ER 348

*AWS and ASTM Specifications: E means grade recognized by AWS-ASTM as covered electrode; ER as bare electrode and welding rod

Joint Strength. From a strength standpoint, a welded joint can be analyzed by the use of standard strength of materials formulas. The allowable working stresses to use for the weld joint in the case of many standard materials can be obtained from several sources, including the American Welding Society "Welding Handbook."

The designer should use caution, however, in the design of welded joints utilizing the super-strength alloys such as austenitic and semiaustenitic stainless steels, nickel base alloys, and high-strength aluminum, because of the detrimental effects of weld heat on the strength of these metals. When welded joints and connectors are to be used extensively in a fluid system, it is recommended that test specimens utilizing the actual materials and joining techniques be prepared and tested to verify design calculations.

Fluid Compatibility. The fluid to be contained and its corrosive nature must be considered in selecting a weld method. Although the microstructure of the parent metal may be compatible with the operating fluid, changes in microstructure resulting from the welding process may make this area of the tubing more susceptible to corrosion. Stress corrosion may take place in weld areas if stress concentrations are allowed to occur. Therefore, the designer should take into consideration the changes in material composition resulting from the welding process, and ascertain the compatibility of any such material changes in the presence of the system fluid.

Welding Equipment Portability. The location and position of welds affects the choice of welding method, primarily from the standpoint of size and portability of the welding equipment. Gas welding equipment is highly portable, whereas resistance welding equipment for flash butt welding is usually permanently installed. Also, metal arc welding equipment is quite portable, requiring only a standard motor-driven DC generator. The inert gas shielded methods require more elaborate equipment, including a gas supply; however, this type of welding can be carried out in all positions, including overhead which is advantageous when in-place welding must be conducted. Other fusion welding methods which depend on floating oxides out of the molten weld puddle are usually limited to downhand welding in the horizontal position.

Production Requirements. The number of like welds to be made considerably influences the selection of the welding method. Large production quantities justify making an investment in elaborate tooling, and automatic welding machines, whereas low production usually requires the use of standard equipment for hand-welding methods. The tubing size also influences the choice of weld method, since practicality will limit the use of automatic equipment in large diameter tubing. The welding methods that lend themselves to automation are the inert gas, atomic hydrogen, and all resistance welding processes. For random in-place tube welding or one-of-a-kind welds, gas welding is often employed because of the simplicity of the equipment involved. This is also true for hand-operated inert shielded-arc welding, with this method offering the advantage of not contaminating the work with corrosive flux that has to be removed. Inert gas metal-arc welding, which uses an automatically fed consumable electrode, is a method often used for production welding because it sets a pace for the welding operator.

5.12.6.4 WELDING TECHNIQUES. Any one of several tried and proven welding techniques can be used for the joining of fluid system lines and components. Equipment for welding tubing includes hand-held tools, semi-automatic welding equipment, and fully automatic equipment.

The success of any welded joint in a fluid system depends to a large extent upon selection of the proper welding method and execution of the weld in a satisfactory manner. Following are discussions of those methods that are the most likely to be used for fluid system welding.

Gas Tungsten-Arc Welding. This process, often referred to as TIG welding, is an inert gas shielded, metal-arc process using a nonconsumable tungsten electrode. The inert gas shield is usually provided by helium or argon. The weld is accomplished by fusing workpieces with a tungsten arc without adding filler material. The primary advantage of this method is that it does not require a flux to form a satisfactory weld, obviating post cleaning of the weld area. This is important for closed fluid systems where flux removal would be difficult. If flux is used it must be removed, otherwise it will accelerate corrosion of the weld metal and contaminate the system.

Because there is no flux to absorb impurities in the TIG welding process, cleanliness is an important consideration. Combustible materials in the path of the arc will burn and generate gases that contaminate the inert shroud, causing scum to form over the molten metal and making it difficult for the welding operator to obtain smooth flow. Also, it is a potential cause of cracks and porosity in the weld. Tubing which is to be joined by the TIG process should be cleaned mechanically, when necessary, and with solvent.

Another advantage of TIG welding is that it can be accomplished in all positions, because there is no slag to be worked out of the weld puddle by gravity. Consequently, overhead welding is entirely feasible, solving practical work problems when it is difficult or impossible to position the work. In addition, visual control of the weld area is excellent, as the gas envelope around the arc is transparent and the weld puddle clean. An operator making a weld by hand can do an excellent job because he does not have to contend with smoke and fumes and can see more clearly what he is doing. In addition, this process provides minimum distortion of the metal near the weld because the heat is concentrated at the weld area. The intensity of the average tungsten arc is approximately 10,000 to 15,000 amps per square inch of electrode.

Table 5.12.6.4a lists a number of weld parameters for joining, by the TIG process, tubing materials which are most commonly found in aircraft and rocket propulsion fluid systems. Materials included in this table are AISI Type 347, AM 350 CRT, and AM 350 SCT stainless steels, René 41 Alloy, and 6061-T6 aluminum. Further details for joining these materials by TIG welding can be obtained from Reference 320-7.

Table 5.12.6.4a. Weld Parameters for Joining Tubing by TIG Process for Aerospace Fluid Systems

(Reference 320-7)

Tubing System Material	Tube Size (in.)		Sleeve Thickness (in.)	Weld Type and No. of Passes	Diameter of Electrode (a) (in.)	Weld Current (amperes)	Arc Voltage (volts)	Travel Speed (sec per revolution)	Welding Start Position	Shielding Gas and Flow Rate		Purge Time (min)
	OD	Wall								Torch	Backup	
AISI Type 347 Stainless Steel	1	0.083	0.030	External 1 pass	3/32	84	10.0	26.5	6 (b)	75% Argon 25% Helium 20 cfh	Helium 40 cfh	5
	3	0.250	None	Internal 1 pass plus External 1 pass	3/32	100	9.5	180	6	75% Argon 25% Helium 50 cfh	Helium 50 cfh	1
					3/32	110	9.5	210	12	75% Argon 25% Helium 60 cfh	Helium 40 cfh	3
AM 350 CRT Stainless Steel	1/4	0.042	0.015	External 1 pass	1/16	15 (c)	12.0	17	Traveling	75% Argon 25% Helium 20 cfh	Helium 50 cfh	10
AM 350 SCT Stainless Steel	1	0.134	None	External 1st pass 2nd pass	3/32	95 55	14.0 14.0	45 45	7:30	75% Argon 25% Helium 40 cfh	Helium 50 cfh	15
René 41 Alloy	1/8	0.010	0.010	External 1 pass	1/16	5 (c)	18.0	8.5	Traveling	75% Argon 25% Helium 15 cfh	Helium 30 cfh	10
	1	0.065	0.031 (d)	External 1 pass	1/16	52	15.0	27	9	75% Argon 25% Helium 35 cfh	Helium 10 cfh	15
6061-T6 Aluminum	1	0.058	0.045 Diam. 4043 Alum. Alloy Wire	External Wire Feed (e)	1/16	26 (f)	13.0	48	11 (e)(f)	Helium 50 cfh	Argon 30 cfh (g)	1

Notes: (a) All welds made with two percent thoriated electrodes.
(b) Number denotes electrode position in terms of hour positions of clock; electrode travel is clockwise direction unless otherwise noted.
(c) Electrode travel started before weld current initiated.
(d) Wire feed was at rate of 12 inches per minute for electrode traveling speed of 4 inches per minute, counterclockwise travel.
(e) Current of 20 amperes was used to preheat start area for 20 seconds; current was then increased to 26 amperes after which electrode travel and wire feed were started within five seconds. Approximately two-thirds around, circumference weld current was reduced to compensate for effect of preheating ahead of electrode. After overlapping weld start, wire feed was sloped off, then weld current was sloped off to zero by the time approximately one-half a revolution was made.
(f) Solar 202 flux was also used for inside surfaces of tube and joint at joint area.

Gas Metal-Arc Welding. This welding process is an offspring of the TIG welding process. Gas metal-arc welding (more appropriately termed gas-shielded consumable metal-electrode welding) utilizes a consumable metal electrode and a gas shield, usually CO_2, helium, or argon. In this process, a special welding tool feeds the electrode, usually of the same composition as the material being welded, as it is consumed. The use of electrode in the form of coiled wire increases the speed of this welding process, and a further time savings is effected since no weld slag forms which must later be removed.

When using helium or argon for a gas shield, this process can be used for welding a great variety of materials. However, cost usually limits the use of these gases to the welding of aluminum, magnesium, or stainless steels where a completely inert atmosphere is required.

For welding heavy-walled steel tubing, a granular flux is often combined with a CO_2 gas shield. Hollow electrodes with a self-contained flux in the center are also used. Another method uses a magnetic flux added to the inert gas, which adheres to the electrode wire as a result of magnetic force caused by current flow in the wire.

Gas Welding. This is another welding process which can be adapted sufficiently well for in-place fabrication of fluid systems. It is a versatile process in that existing portable torches and equipment can be used. Generally speaking, oxyacetylene and oxyhydrogen flames are used, however, natural gas and propane may also be utilized. When gas welding is employed, it is often necessary to supply a filler material to build up the weld area to the necessary volume and strength. In addition, the use of a flux is required to prevent oxidation of the molten weld metal, as discussed above. However, this imposes the disadvantages of post-weld cleaning. Since the development of shielded arc welding, the gas welding process has been largely limited to applications where portability of welding equipment is a must.

Flash Welding. Flash welding is a resistance welding process. In joining two pieces by this process, the welding voltage is first applied between the pieces, a flashing or arcing follows for a prescribed time, and finally the pieces are forced together upsetting the metal and forming a weld joint. Although this is a rapid method of welding, when tubing is welded in this manner it is often impossible to remove the flash that is formed on the inside of the tubing. Flash welding schedules for welding tubing of thicknesses from 0.010 to 1.0 inch are given in Table 5.12.6.4b.

Electron Beam Welding. A recently developed welding technique which has proven invaluable for the welding of high strength and high refractory metals and alloys is the electron beam welding method (EB welding). An electron beam welding machine consists of a vacuum chamber in which electrons are produced and directed toward the work piece. The heart of the electron beam welder is an electron gun in the form of a triode. Its components are an electron-emitting cathode, an accelerating electrode, and a focusing electrode or lens. Figure 5.12.6.4 illustrates schematically the Pierce electron gun and shows how the beam of electrons can be focused on a small area.

Three primary advantages of the electron beam process are:

1) In addition to forming welds under ideal conditions of purity, a large amount of heat can be concentrated in a very small area. This means that the fuse zone which forms the weld bead can be deep and narrow with a depth-to-width ratio of from 4 to as high as 25, compared to about 1 for other welding processes such as TIG and gas.

2) Because the weld zone is so narrow, there is less molten metal to give off heat during cooling, and consequently, there is less distortion in the weld zone, and the microstructure of the parent metal adjacent to the weld zone

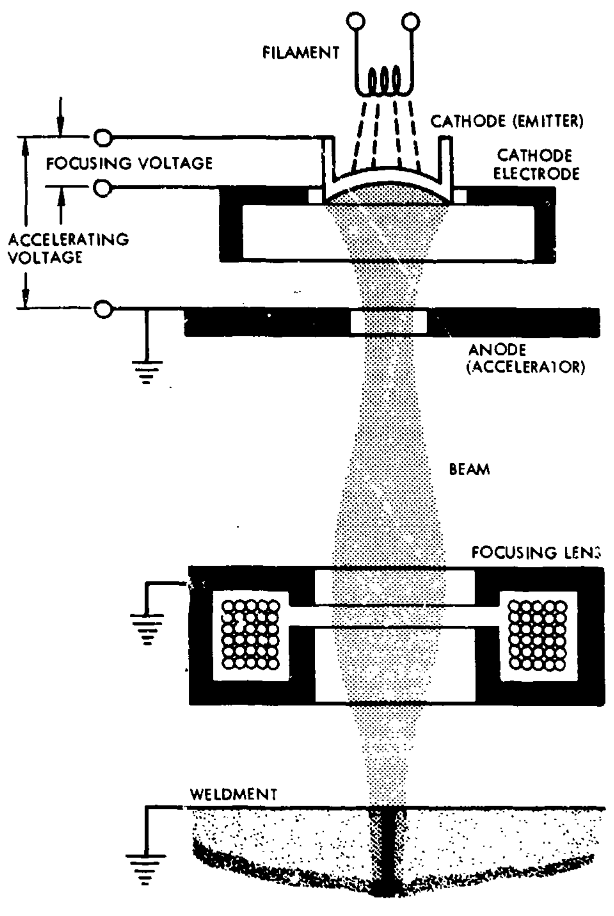


Figure 5.12.6.4. Sketch of Pierce Electron Gun
(Reprinted with permission from "International Science and Technology," No. 4, April 1962, R. F. Bunshah, Conover-Mast Publications, New York, New York)

is less affected. This is important because the heat-affected zone adjacent to the fusion zone is very often the location where welds fail because of the inability to control the micro-structure of this area.

3) The amount of heating can be varied smoothly and precisely by varying the control voltages. This means that preheating and post-heating can be accomplished easily and weld spatter is seldom a problem.

The major drawback or disadvantage of using the electron beam process for in-place welding is that all welding must be accomplished in a vacuum of approximately 10^{-3} torr. This requires remote manipulation of the work piece, or electron gun, and greatly limits the work which can be welded by this process. To date, this welding method has been confined primarily to special welding tasks requiring design of special equipment and set-up, or to general welding of small components. Additional information on this process can be obtained from References 411-2 and 65-42.

Atomic Hydrogen Welding. This welding process is a type of arc welding. Although the inert-gas welding processes have displaced atomic-hydrogen welding to a large extent, the process is worthy of note in that it is particularly suited to welding high alloy steels and relatively thin materials, while maintaining the same composition in the weld as in the base metal. Atomic-hydrogen welding is usually accomplished with a special hand-held tool that contains two tungsten electrodes and a hydrogen gas feed through one or both of the electrode holders. No filler metal is used in this process. As the workpieces are positioned, the two electrodes are energized and an AC arc is maintained with a hydrogen gas envelope. The hydrogen dissociates in the arc and then combines with the cooler base metal. In combining with the base metal, substantial heat is released. The heat of combination plus the heat of the arc produces a higher temperature than is available from ordinary arc welding processes. The hydrogen gas also provides a reducing envelope that shields the weld area from oxygen and nitrogen in the free atmosphere.

5.12.6.5 WELDED CONNECTOR AND JOINT TYPES. Basically, there are two types of welded joints used in aerospace fluid systems. One is the plain joint in which the tube ends or components are fitted together, either butted or lapped, and then welded by the application of heat with or without the use of a flux and filler material. The other type of joint is termed a sleeve joint, employing the use of a separate connector, or sleeve, which is slipped over the ends of the two parts to be joined and welded to the parts, usually at the joint interface. Some specific joint types are discussed below.

Butt Joints. When using the arc welding method to form the straight butt joint illustrated in Figure 5.12.6.5a, no surface preparation of tube ends is necessary if relatively thin-walled tubing is used and joining is accomplished by a single pass of the electrode. Preparation of the tube ends is required, however, when wall thickness makes more than one weld pass necessary (see Figure 5.12.6.5b).

A straight butt joint may also be made by any of several other welding methods, as discussed in Detailed Topic 5.12.6.4.

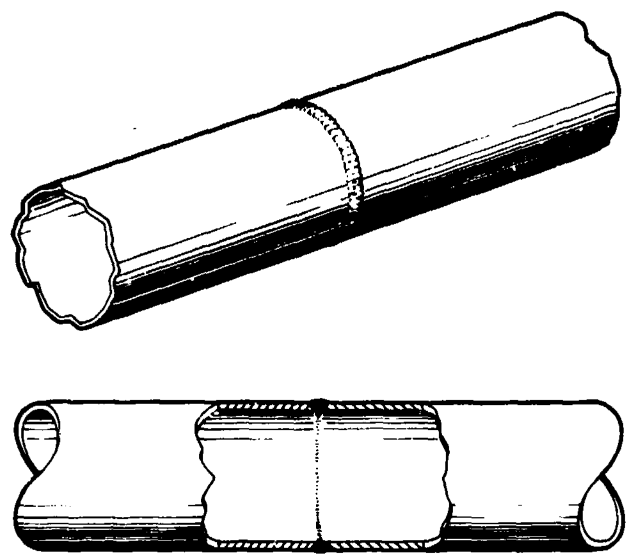


Figure 5.12.6.5a. Straight Butt Tube Weld Joint

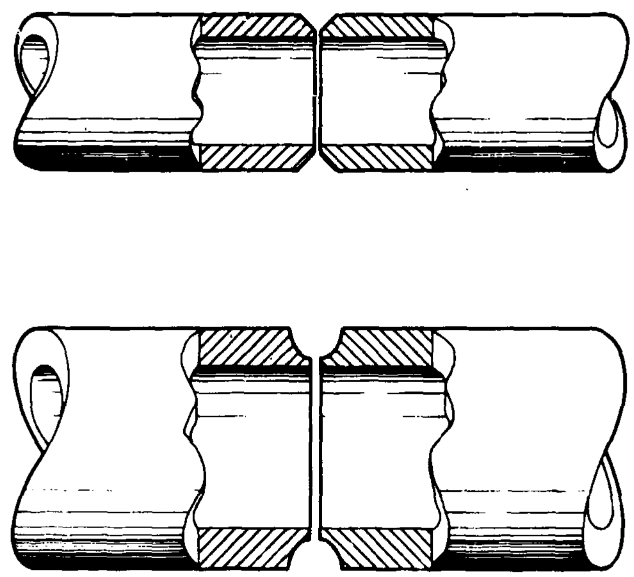
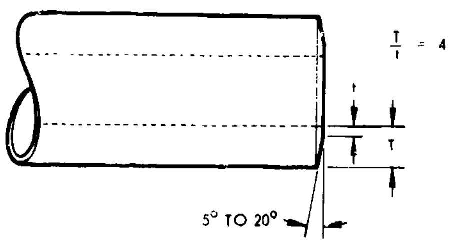


Figure 5.12.6.5b. Edge Preparation Required for Welding Thick-Walled Tubing

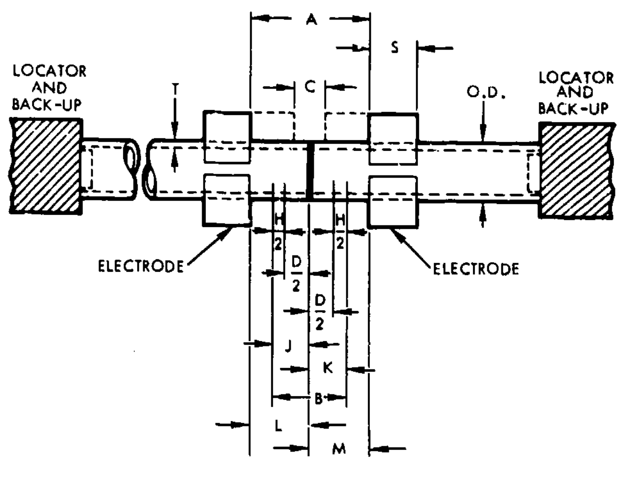
Table 5.12.6.4b. Flash Welding of Tub

(From "Welding Handbook," 4th Ed., Section II, American



NOTE: WHEN "T" IS 3/16 IN. OR THICKER, BEVEL ONE WORK PIECE

RECOMMENDED END PREPARATION
TUBING



T = TUBE WALL OR SHEET THICKNESS
A = INITIAL DIE OPENING
B = MATERIAL LOST
C = FINAL DIE OPENING
D = TOTAL FLASH-OFF
H = TOTAL UPSET
J = K = MATERIAL LOST PER PIECE
L = M = INITIAL EXTENSION PER PIECE
O.D. = OUTSIDE DIA. OF TUBING
S = MINIMUM NECESSARY LENGTH OF ELECTRODE CONTACT

T, In.	A, In.	B, In.	C, In.	D, In.	H, In.	J = K, In.
0.010	0.110	0.060	0.050	0.040	0.020	0.030
0.020	0.215	0.115	0.100	0.080	0.035	0.058
0.030	0.325	0.175	0.150	0.125	0.050	0.088
0.040	0.430	[illegible]	0.200	0.165	0.065	0.115
0.050	0.530	[illegible]	0.250	0.205	0.075	0.140
0.060	0.620	0.330	0.290	0.240	0.090	0.165
0.070	0.715	0.385	0.330	0.280	0.105	0.193
0.080	0.805	0.435	0.370	0.315	0.120	0.218
0.090	0.885	0.475	0.410	0.345	0.130	0.238
0.100	0.970	0.520	0.450	0.375	0.145	0.260
0.110	1.060	0.570	0.490	0.410	0.160	0.285
0.120	1.140	0.610	0.530	0.440	0.170	0.305
0.130	1.225	0.650	0.575	0.470	0.180	0.325
0.140	1.320	0.700	0.620	0.510	0.190	0.350
0.150	1.390	0.730	0.660	0.530	0.200	0.365
0.160	1.470	0.770	0.700	0.560	0.210	0.385
0.170	1.540	0.800	0.740	0.580	0.220	0.400
0.180	1.620	0.840	0.780	0.610	0.230	0.420
0.190	1.690	0.870	0.820	0.630	0.240	0.435
0.200	1.760	0.900	0.860	[illegible]	0.250	0.450
0.250	2.010	1.010	1.000	0.730	0.280	0.505
0.300	2.245	1.120	1.125	0.810	0.310	0.560
0.350	2.460	1.210	1.250	0.880	0.330	0.605
0.400	2.640	1.290	1.350	0.930	0.360	0.645
0.450	2.780	1.350	1.430	0.970	0.380	0.675
0.500	2.910	1.410	1.500	1.020	0.390	0.705
0.550	3.040	1.465	1.575	1.055	0.410	0.733
0.600	3.135	1.505	1.630	1.085	0.420	0.753
0.650	3.245	1.555	1.690	1.125	0.430	0.778
0.700	3.360	1.610	1.750	1.160	0.450	0.805
0.800	3.525	1.675	1.850	1.210	0.465	0.838
0.900	3.660	1.730	1.930	1.250	0.480	0.865
1.000	3.800	1.800	2.000	1.300	0.500	0.900

* Data based on welding, without preheat, two pieces of
Fig. 31.12 for assembly of parts.

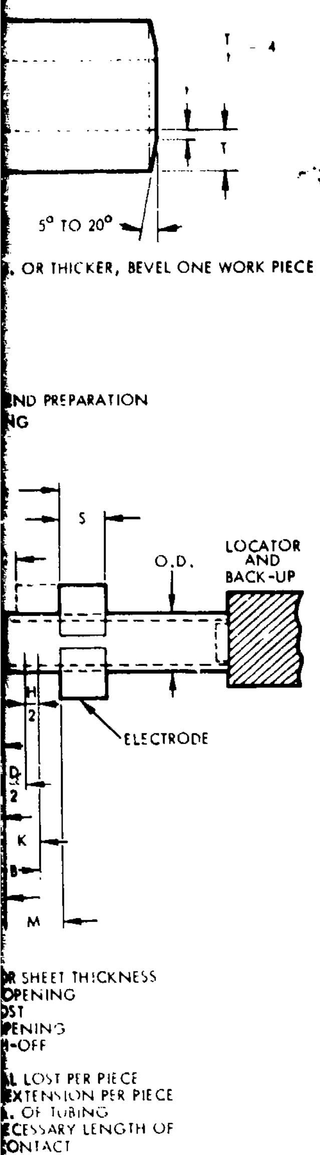


Table 5.12.6.4b. Flash Welding of Tubing and Flat Sheets

(From "Welding Handbook," 4th Ed., Section II, American Welding Society, New York, 1962)

T, In.	A, In.	B, In.	C, In.	D, In.	H, In.	J = K, In.	L = M, In.	Flashing Time, Sec.	O.D., In.	S With Locator	S Without Locator
0.010	0.110	0.060	0.050	0.040	0.020	0.030	0.055	0.40	0.250	0.375	1.00
0.020	0.215	0.115	0.100	0.080	0.035	0.058	0.108	0.80	0.312	0.375	1.00
0.030	0.325	0.175	0.150	0.125	0.050	0.088	0.163	1.25	0.375	0.375	1.50
0.040	0.430	0.230	0.200	0.165	0.065	0.115	0.215	1.75	0.500	0.375	1.75
0.050	0.530	0.280	0.250	0.205	0.075	0.140	0.265	2.25	0.750	0.500	2.00
0.060	0.620	0.330	0.290	0.240	0.090	0.165	0.310	2.75	1.000	0.750	2.50
0.070	0.715	0.385	0.330	0.280	0.105	0.193	0.358	3.50	1.50	1.000	3.00
0.080	0.805	0.435	0.370	0.315	0.120	0.218	0.403	4.00	2.00	1.250	
0.090	0.885	0.475	0.410	0.345	0.130	0.238	0.443	4.50	2.50	1.750	
0.100	0.970	0.520	0.450	0.375	0.145	0.260	0.485	5.00	3.00	2.000	
0.110	1.060	0.570	0.490	0.410	0.160	0.285	0.530	5.75	3.50	2.25	
0.120	1.140	0.610	0.530	0.440	0.170	0.305	0.570	6.25	4.00	2.50	
0.130	1.225	0.650	0.575	0.470	0.180	0.325	0.613	7.00	4.50	2.75	
0.140	1.320	0.700	0.620	0.510	0.190	0.350	0.660	7.75	5.00	2.75	
0.150	1.390	0.730	0.660	0.530	0.200	0.365	0.695	8.50	5.50	3.00	Not Recommended Without Use of Locator
0.160	1.470	0.770	0.700	0.560	0.210	0.385	0.735	9.00	6.00	3.25	
0.170	1.540	0.800	0.740	0.580	0.220	0.400	0.770	9.75	6.50	3.50	
0.180	1.620	0.840	0.780	0.610	0.230	0.420	0.810	10.50	7.00	3.75	
0.190	1.690	0.870	0.820	0.630	0.240	0.435	0.845	11.25	7.50	4.00	
0.200	1.760	0.900	0.860	0.650	0.250	0.450	0.880	12.00	8.00	4.25	
0.250	2.010	1.010	1.000	0.730	0.280	0.505	1.005	16.00	8.50	4.50	
0.300	2.245	1.120	1.125	0.810	0.310	0.560	1.123	21.00	9.00	4.75	
0.350	2.460	1.210	1.250	0.880	0.330	0.605	1.230	27.00	9.50	5.00	
0.400	2.640	1.290	1.350	0.930	0.360	0.645	1.320	33.00	10.00		
0.450	2.780	1.350	1.430	0.970	0.380	0.675	1.390	38.00			
0.500	2.910	1.410	1.500	1.020	0.390	0.705	1.455	45.00			
0.550	3.040	1.465	1.575	1.055	0.410	0.733	1.520	50.00			
0.600	3.135	1.505	1.630	1.085	0.420	0.753	1.568	56.00			
0.650	3.245	1.555	1.690	1.125	0.430	0.778	1.623	63.00			
0.700	3.360	1.610	1.750	1.160	0.450	0.805	1.680	70.00			
0.800	3.525	1.675	1.850	1.210	0.465	0.838	1.763	83.00			
0.900	3.660	1.730	1.930	1.250	0.480	0.865	1.830	97.00			
1.000	3.800	1.800	2.000	1.300	0.500	0.900	1.900	110.00			

* Data based on welding, without preheat, two pieces of the same welding characteristics. See Fig. 31.12 for assembly of parts.

Lap Joints. The plain lap weld joint, as shown in Figure 5.12.6.5c, is not normally used in high-pressure/high-temperature aerospace fluid systems because of the increased weight and increased pressure drop as a result of turbulence across the male tube end. Another disadvantage is that system fluid can become trapped between the mating tube surfaces because both ends of the joint are generally not welded. Trapped fluids can cause corrosion and cleaning problems. As shown in Figure 5.12.6.5c, the joint can be made by either expanding the female member or reducing the male member. Expansion and reduction of tubing can be accomplished by such methods as swaging, or explosive forming. One advantage of the lap joint is that it is generally stronger in bending than the butt joint because the moment of inertia is increased by the double wall. The change in diameter of the tube wall must be gradual, however, to avoid stress concentrations. This type of joint is normally restricted to easily worked tubing which can withstand the forming operations without cracking or excessive thinning.

When the longitudinal forces in a fluid system are the determining strength consideration, the female member of the lap joint may be serrated, notched, or slotted (see Figure 5.12.6.5c) to afford greater weld bead length and hence a greater allowable shear force. If a lap joint has been made properly, it will seldom fail radially under burst pressure; the main mode of failure will be in tension as the weld is placed in shear.

Burn-Down Flange Joints. A third type of plain weld joint used in the aerospace industry is the burn-down flange joint, depicted in Figure 5.12.6.5d. This joint is used for relatively thin wall tubing which is difficult to butt weld because of burn-through tendencies. Fabrication of this joint requires forming a flange on each of the two parts to be joined. Welding is accomplished by butting the flanges together and applying concentrated heat at the peripheral intersection, thereby melting or burning down the two flanges to form a weld bead. The advantage of this joint is that burn-through and loss of weld puddle control are eliminated because material is specifically provided to form the weld

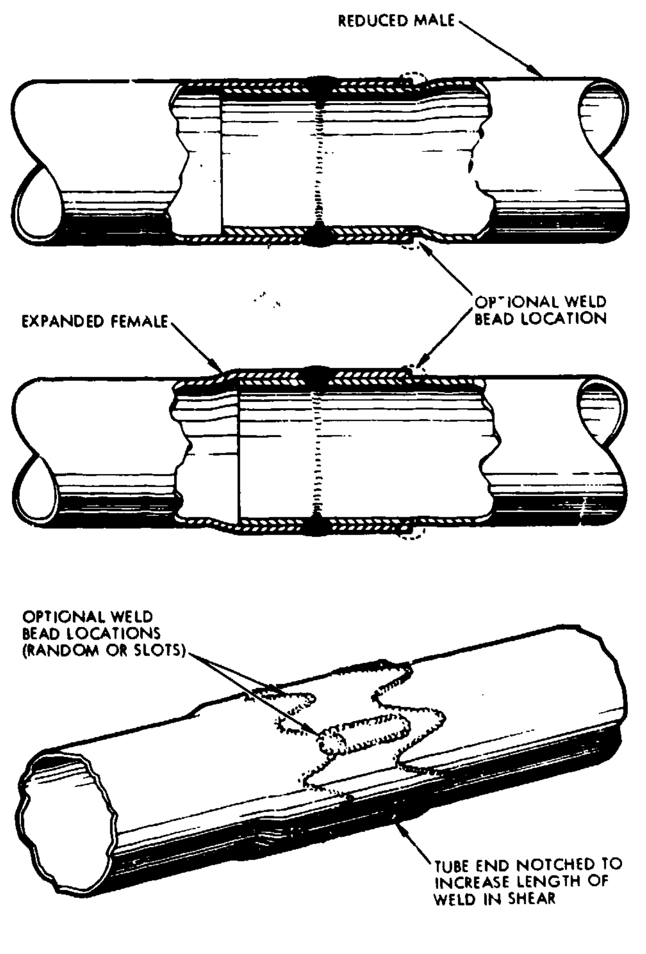


Figure 5.12.6.5c. Plain Lap Weld Joints for Tubing

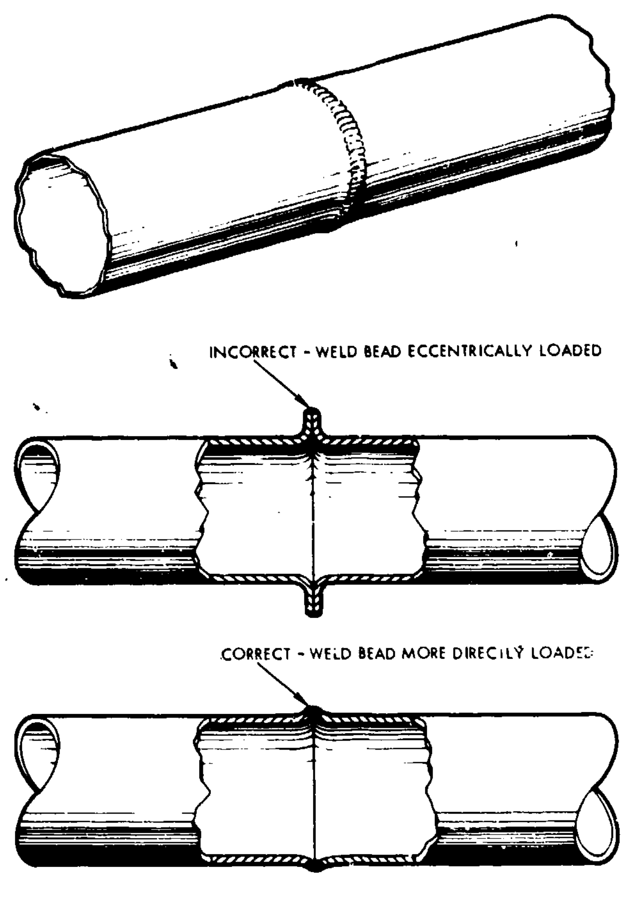


Figure 5.12.6.5d. Burn-Down Flange Weld Joint

bead at a greater diameter than the tube wall. The strength of such a joint will not be as great as that of a straight butt joint, unless the weld bead extends down to the tubing wall OD to prevent eccentric loading (see Figure 5.12.6.5d). The burned-down flange joint is more expensive than a butt joint because joint preparation requires an extra forming operation.

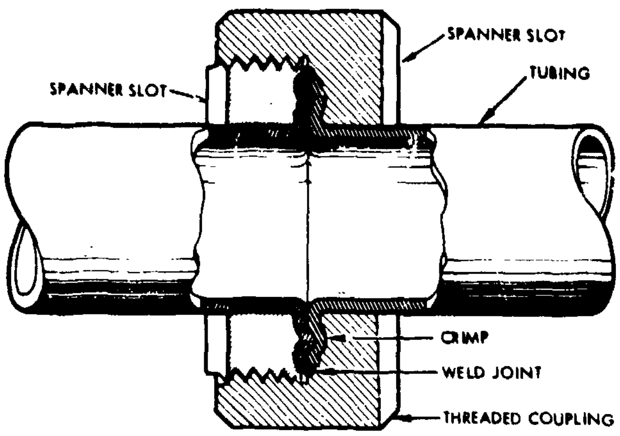


Figure 5.12.6.5e. Modified Burn-Down Flange Joint

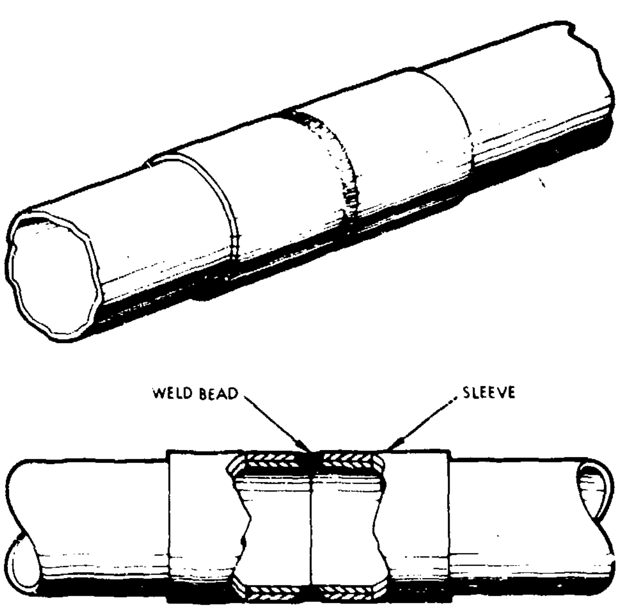


Figure 5.12.6.5f. Welded Sleeve Joint

Modified Burn-Down Flange Joints. A modification of the burn-down flange joint is shown in Figure 5.12.6.5e. Since in this connection the weld is used only to accomplish sealing and is not relied on for strength of the connection, a very small weld is all that is required. In making this type of tube connection the tube ends are first flared to make a butt joint; next a set of aluminum split ring chill blocks are installed onto the tube joint for the purpose of clamping the tube ends together and providing a heat sink during the welding of the flared tube ends. Note that a slight crimp in the flared tube ends is created by the chill blocks, preventing welding contamination from entering the inside of the tubing. After the weld has been made, a threaded coupling is slipped over the weld joint for the purpose of carrying pressure loading. This threaded coupling can be very light, since it is not required to maintain any gasket sealing load, but merely adds the requisite material to resist pressure loading. This type of connection may be classified as semi-permanent, since the seal weld can be broken and rewelded if necessary.

Sleeve Joints. When high strength alloys are used, it is often difficult to form or machine tubing and connectors in preparation for welding. In addition, the butt welding of thin wall high strength tubing is difficult without burn-through or sagging of the weld bead into the tube. To circumvent these problems, a weld connector or sleeve may be provided over the joint area, as shown in Figure 5.12.6.5f. The weld is made by heating the sleeve just over the joint between the two tubes, fusing the sleeve and tubes together in one operation. The connector sleeve is generally made from a piece of tubing the next size larger than the tubing being welded, or is an expanded piece of the same tubing. The wall thickness of the weld sleeve may be the same as or thinner than the tubes being joined. The length of the sleeve is not critical but should be long enough to give support to the heat-affected zone adjacent to the weld bead. Reference 320-7 contains additional details on weld sleeve design.

To assure that the weld is made in the proper location when using a sleeve, it is necessary to position and hold it so that when the weld bead is laid it will fall just above the gap between the two parts being joined. The difference between the outside diameter of the tubing and the inside diameter of the sleeve is not critical, providing that the gap between the two does not exceed 0.005 to 0.010 inch. Because of variations in tolerances on seamless steel tubing diameter, mismatch between two joined tubes can be as great as 0.003 inch on one- and two-inch diameter tubing without causing difficulty in welding, providing that weld penetration is approximately equivalent on both sides of the joint. When extremely heavy sections of tubing are welded, it often is necessary to provide a backup torch to make a weld pass on the inside diameter. This, of course, can only be accomplished when there is access to the inside of the tubing. When using a weld sleeve, the tubes should be butted together rather than allowing a gap, although satisfactory welds can be made with gaps as high as 0.030 inch. Gaps larger than this may cause weld bead concavity, which can result in stress concentration in the weld area.

5.12.7 Swaged Connectors

5.12.7.1 GENERAL. Swaging, as applied to connectors, is the process of joining two pieces by the cold forming of metal to both make a seal and form a connection to carry the structural load. This type of connection is generally considered to be permanent in nature, although recent developments have exploited the technique for obtaining a semi-permanent connector which does not require the addition of heat (Reference 36-14). The swaging process as such is utilized in some existing connectors as a means of locating a ferrule or sleeve on a piece of tubing, and is receiving increasing attention in new connector concepts. As an example of present usage the MS flareless connector ferrule is swaged to the tubing during the initial connector assembly process.

The swaging process offers some distinctive advantages over other permanent connectors, such as:

a) Heat is not required to make the connection, and therefore tubing is not weakened as a result of heat concentration or annealing.

b) Inspection to assure proper mechanical locations of swaged connector parts can be readily accomplished by visual operation.

5.12.7.2 SWAGED CONNECTOR TYPES. One type of swaged connector is the Parker "H" fitting shown in Figure 5.12.7.2a. In this connector, the tubing is inserted into the ends of the connector body. The sleeves, or slides in this case, are then pulled up onto the tapered lands of the connector body resulting in the tubing being locked in place by the sharp edges of the connector body inner diameter. Either a special hand-operated or hydraulically operated tool is required to set the sleeves onto the connector body. A more recent development, also from Parker Aircraft, is the "X" connector illustrated in Figure 5.12.7.2b. The development of this connector for NASA's Marshall Space Flight Center (as a potential improvement over MC connectors) is described in Reference 36-14. This concept is unique in that it is under development for not only the one-half-inch tubing size illustrated, but for a 4-inch duct size. The concept is being considered for all sizes of lines from one-quarter-inch tubing to the largest diameter ducts currently visualized in launch vehicle design (about 20-inch diameter, according to Reference 34-10). The sleeves of high strength stainless steel are stretched beyond their yield stress when the tubing ends are swaged into place. During assembly, the gold-plated primary seals of the stainless steel collar are seated against the sleeve seating surfaces by the axial force resulting from radial deformation of the collar by an assembly tool. This toggle action results in negligible relaxation of the axial sealing force when the collar lips spring radially outward slightly after the assembly tool is removed. Upon disassembly the collar is either "unswaged" or cut off and discarded; a new collar (and hence new gold-plated seals) is used for each subsequent reassembly.

5.12.8 Adhesive Bonding Joints and Connectors

5.12.8.1 GENERAL. Adhesive bonding is a process suitable for joining metal-to-metal, metal-to-nonmetal, or nonmetal-to-nonmetal. The types of adhesives used include thermosetting resins, thermoplastic resins, and various artificial elastomers. Joints that can be brazed can generally be adhesive bonded also, with the exception that high-temperature applications and high-tensile strength requirements are limiting factors for adhesive bonds.

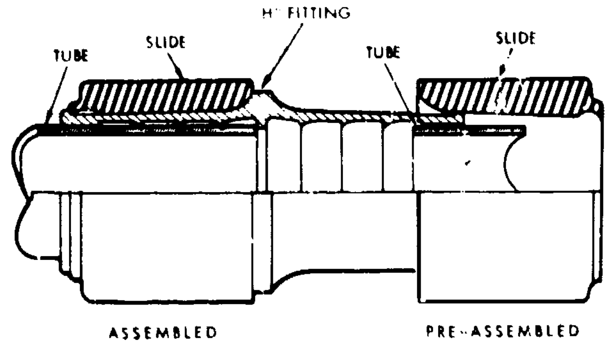


Figure 5.12.7.2a. Parker "H" Connector
(Courtesy of Parker Aircraft Co., Los Angeles, California)

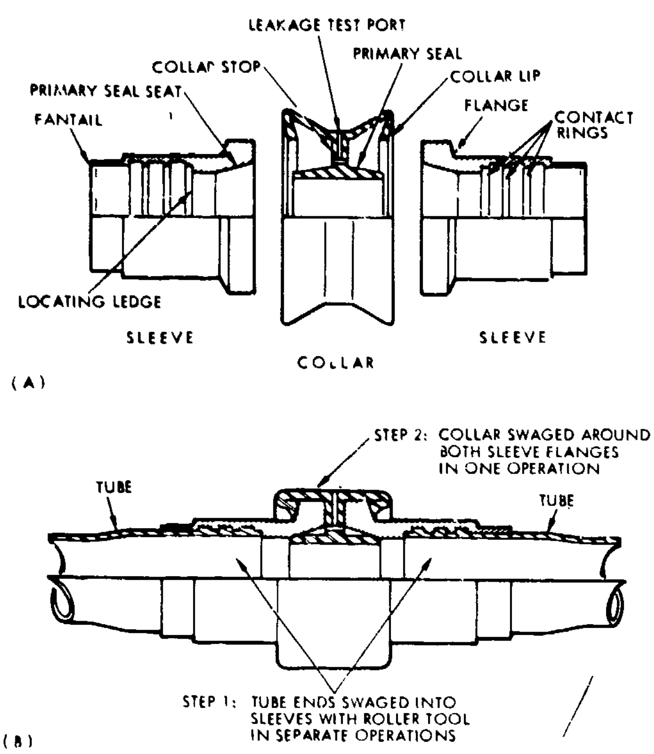


Figure 5.12.7.2b. Parker "X" Connector
(Reference 36-14)

5.12.8.2 APPLICATION. Adhesive bonding has seen limited usage in aerospace fluid connector applications to date, although it has been used in related components, such as liners for filament-wound reinforced-plastic helium pressure vessels. Adhesive bonding of relatively low-pressure plastic pipe using sleeve or socket-type connectors has been widely used in numerous military, industrial, and commercial applications for reasons of chemical compatibility, weight, or cost.

Increased interest in the use of filament-wound reinforced plastic for liquid-propellant tankage and even high-pressure rocket engine fluid-transfer lines (Reference 34-10) may be expected to focus more attention on the use of adhesive bonding in future aerospace applications.

5.12.8.3 DESIGN FACTORS. The design of an adhesive bonded joint is similar to that of a brazed joint, in that full advantage must be taken of the shear strength of the adhesive. Butt joints are therefore avoided in preference to lap type joints.

As with brazing, a satisfactory bonded design must also give consideration to surface preparation. Generally acceptable methods in preparing surfaces to be adhesively bonded include:

a) Cleaning all contaminants and grease from surfaces
b) Etching surfaces to make them chemically receptive to adhesives and to provide good wetting characteristics
c) Rinsing
d) Drying
e) Applying a primer.

Thin materials may be successfully joined by adhesive bonding. In addition, adhesive bonding allows a smooth contour at the joint, can provide thermal and electrical insulation, can protect against galvanic action between dissimilar metals, and may be used as a dampener against vibration or sound.

Three factors that severely limit the use of adhesively bonded connectors and joints are:

a) Instability of many adhesives at temperatures above about 400°F
b) Poor tensile strength
c) Limited compatibility of the adhesive with fluid media.

REFERENCES

*References added March 1967

Threaded Connectors

1-216	36-14
6-30	44-14
6-100	44-15
6-111	46-6
6-130	46-7
6-154	46-29
6-161	46-31
6-187	147-1
6-209	151-2
18-1	151-3
19-76	151-4
23-38	233-1
23-55	447-4
36-11	V-97
36-12	44-24*

Connector Preload Theory

36-11	44-15
36-12	46-29
36-14	46-31
44-14	44-24*

Flanged Connector Design

19-76	46-6
23-38	46-7
26-213	46-29
26-214	46-31
36-11	68-62
36-12	132-1
36-14	196-1
44-14	400-1
44-15	

Swaged Connectors

36-14	46-31
36-20*	

Welded and Brazed Connectors

65-42	132-1
68-62	320-7
77-5	504-1
77-6	504-2
36-20*	513-2*
320-9*	

Adhesive Bonded Connectors

26-209	132-1
34-10	400-1
93-8	

5.13 FLEXIBLE FLUID COUPLINGS

5.13.1 Introduction

Flexible fluid couplings consist of many types of flexible hoses, bellows, and coiled tubing configurations for fluid systems.

5.13.2 When to Use Flexible Couplings

Flexible couplings (hoses, bellows, and coiled tubing) are considered for use in lieu of rigid tubing under any of the following conditions:

Angular or lateral misalignment of points to be connected exists.

Fluid components are frequently removed for repair or replacement.

Relative motion occurs caused by thermal expansion, component movement, or structural bending.

Severe vibration is present.

5.13.3 Which Flexible Couplings to Use

The choice of which flexible coupling to use depends upon many factors, some of which are listed in Table 5.13.3. Relative values of the factors have been assigned for each coupling to serve as a guide in the selection of a connector for a specific application.

5.13.4 Flexible Hoses

A flexible hose assembly consists of a flexible inner liner which conducts the flowing fluid, a reinforcement for bracing the inner liner, and end fittings for connecting the hose ends to the fluid system (Figure 5.13.4). The selection of a flexible hose assembly involves determining first the inner liner material and design, second the reinforcement materials and design, and last the type and method of attaching the end fittings.

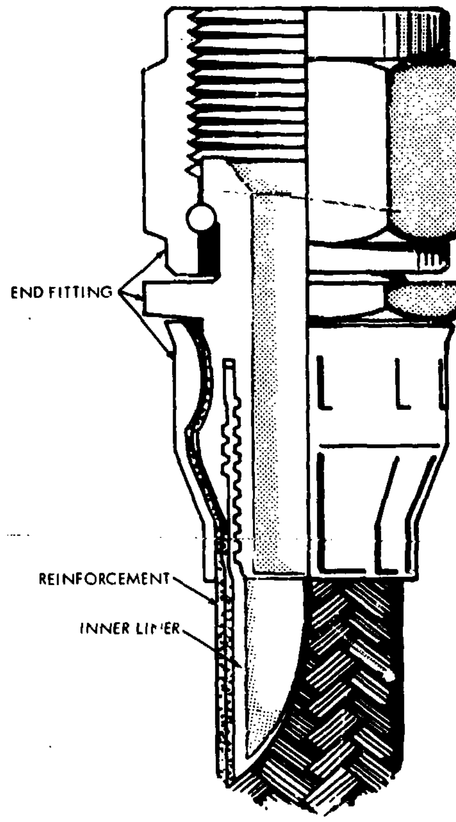


Figure 5.13.4. Flexible Hose Assembly
(Courtesy of the Weatherhead Company, Cleveland, Ohio)

5.13.4.1 FLEXIBLE HOSE INNER LINER DESIGN FACTORS. Several factors affect flexible hose inner liner design.

Materials. Hose inner liner materials are grouped in three classifications—elastomers, plastics, and convoluted metals. Which type of liner material is best suited for a given application depends upon the following factors:

Fluid-material compatibility
High and low temperature tolerance
Flexibility-minimum bend radii
Surface roughness
Leakage susceptibility
Available sizes

Material compatibility and physical property data are included in Section 12.0, "Materials," for use as a guide in selecting hose liner materials.

Construction. Inner liners of natural rubber, synthetic rubber, or extruded plastic are smooth flexible tubes as shown

Table 5.13.3. Flexible Fluid Coupling Selection Chart

CONNECTOR	RATINGS*							
	FLEXIBILITY	WEIGHT	PRESSURE DROP	SPACE	PRESSURE	TEMPERATURE	RELIABILITY	IMPULSE LIFE
Flexible Hoses	1	3	2	2	2	2	2	2
Bellows Joint	1	2	2	1	3	2	2	3
Coiled Tubing	2	1	1	1	1	1	1	1

1) First choice
2) Second choice
3) Third choice

in Figure 5.13.4.1a. The strength of the liner is dependent upon the material yield stress, bore, and wall thickness which are related to the internal pressure as follows:

$$p_i = \frac{2 S_y t}{D} \quad \text{(Eq 5.13.4.1)}$$

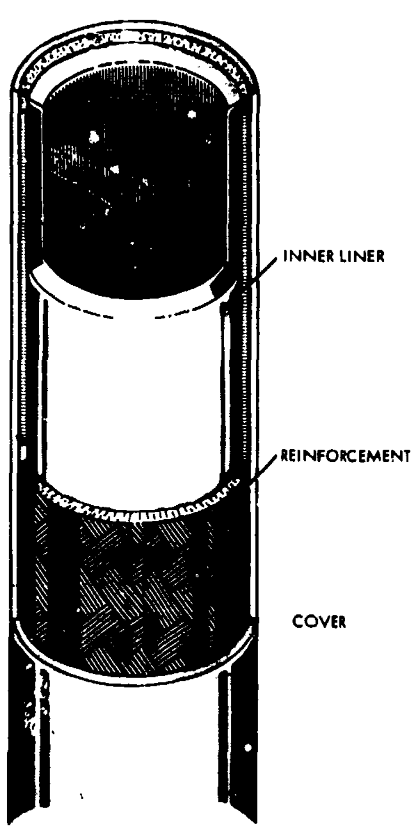


Figure 5.13.4.1a. Non-metallic Interliners
(Courtesy of Imperial-Eastman Company, Chicago, Illinois)

where p_i = internal fluid pressure, psi
S_y = material yield stress, psi
t = material wall thickness, in.
D = tubing bore diameter, in.

Metallic inner liners may have helical or annular convolutions which can be constructed by corrugating a plain tube or by spiraling a continuous strip of metal while bonding the edges. The strength and flexibility of the inner liner varies with the type of convolution, the depth, pitch, and wall thickness of the liner, and the mechanical properties of the material. Typical metal liner configurations are shown in Figure 5.13.4.1b. Generally, stripwound liners have better strength and flexibility characteristics than corrugated tube liners, but are more susceptible to leakage.

5.13.4.2 **FLEXIBLE HOSE REINFORCEMENT DESIGN FACTORS.** Several factors affect flexible hose reinforcement design.

Materials. Hose reinforcement materials commonly used in flexible hose assemblies are natural fibers (cotton), synthetic fibers (rayon, dacron, nylon), and metallic wire (carbon steel, stainless steel, Monel, bronze). Often the inner liner and reinforcement material are the same. The choice of which material or combinations of materials to use depends upon the specific application; such factors as temperature, pressure, flexibility, corrosion resistance, and weight should be considered. Generally, stainless steel reinforcement provides a strong, corrosion-resistant support, usable for cryogenic and high temperature (1000°F) applications, but is heavier and stiffer than a synthetic fiber (such as nylon) which is usable in the temperature range of −100°F to +225°F.

Construction. The two common reinforcement configurations employed in aerospace flexible hoses are wire or fiber braid (basket weave), and spiral wire (Figure 5.13.4.2a). The strength of the reinforcement depends upon the fiber or wire material, diameter, and coverage. The braid type also depends upon braid angle, which is normally 35 degrees to the hose centerline. One method of applying the

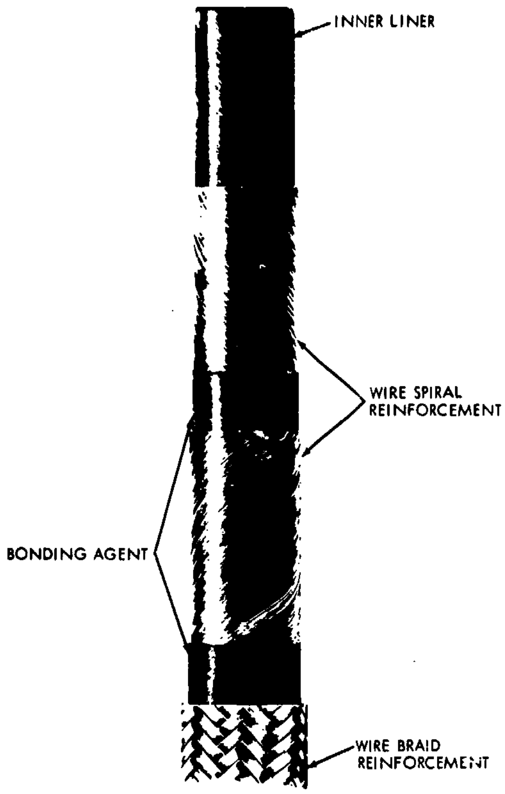


Figure 5.13.4.1b. Metal Interliners
(Reprinted from "Chemical Engineering," February 1961, vol. 68, no. 4, E. M. Ramberg. Copyright 1961, McGraw-Hill Publishing Company, New York, New York)

reinforcement is to cover the inner liner with a coating of cement, and then wind the braid or spiral ply tightly over the liner, bonding the reinforcement to the liner. Additional layers of reinforcement are also bonded to the surface underneath using the same technique. The entire assembly is then aged at the curing temperature of the bonding cement. Spiral ply reinforcement is not subject to stress concentrations such as exist in braid reinforcement where wires cross, and is therefore less apt to fracture. In the hose assembly shown in Figure 5.13.4.2a, the spiral ply is used in the inner layers for strength while braid is used on the outer layer for protective covering and additional strength. Many other variations of combinations of spiral ply and braid reinforcement are available, but are not illustrated. One special configuration worth mentioning is jacketed flexible hose where cryogenic or high temperature fluids are transported and heat gain or loss is critical. Figure 5.13.4.2b illustrates a jacketed flexible hose of all metal construction suitable for such applications.

5.13.4.3 METHODS OF ATTACHMENT. The end fitting is required to form a seal with the inner liner and distribute all working forces to the reinforcement. The ability to do this depends upon the method of attachment which may be accomplished by swaging, crimping, or threading non-metallic hoses; or threading, welding, brazing, or soldering of corrugated metal hoses. Figure 5.13.4.3 illustrates these seven methods of fitting attachment. In selecting the method of attachment, consideration should be given to reusability, leakproof reliability, temperature and corrosion resistance to brazing, soldering, and welding alloys, as well as gripping strength.

5.13.4.4 HOW TO INSTALL FLEXIBLE HOSES. An important consideration in obtaining maximum useful life of

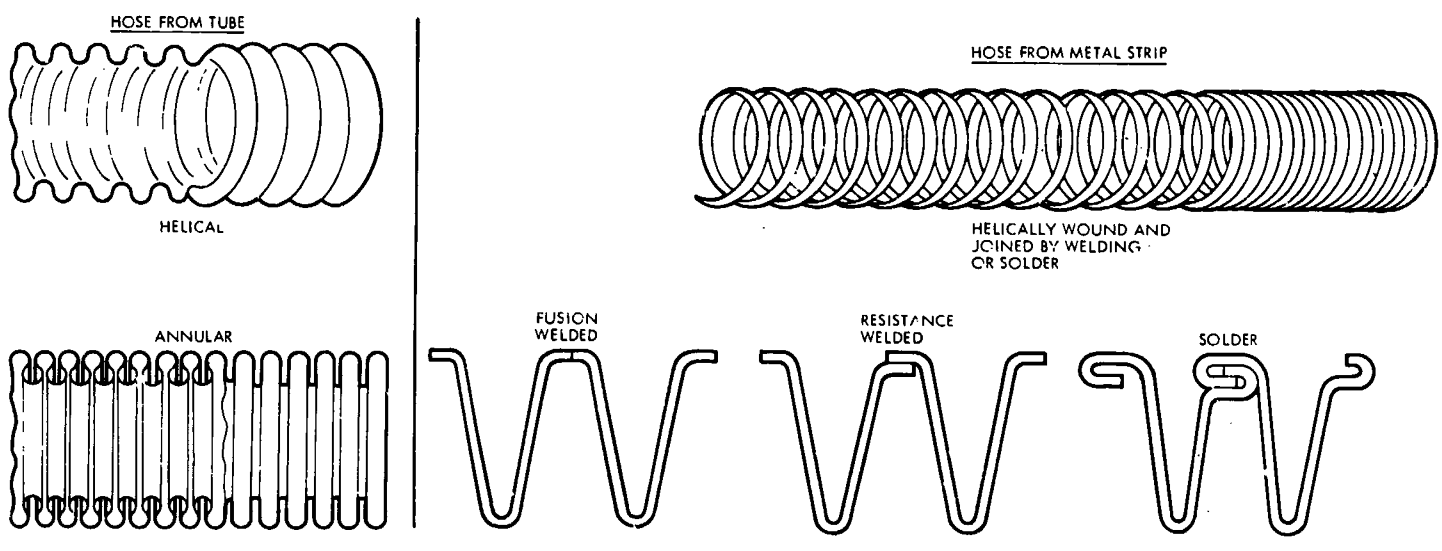


Figure 5.13.4.2a. Braid and Spiral Wire Reinforcements
(Courtesy of Resistoflex Corporation, Roseland, New Jersey)

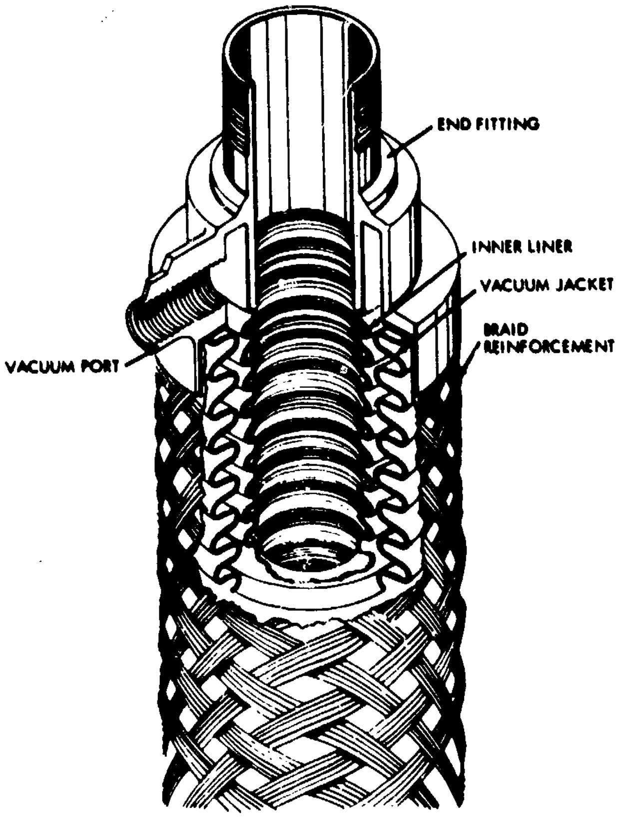


Figure 5.13.4.2b. Vacuum Jacketed Flexible Hose
(Courtesy of Allied Metal Hose Company, Long 'sland City, New York)

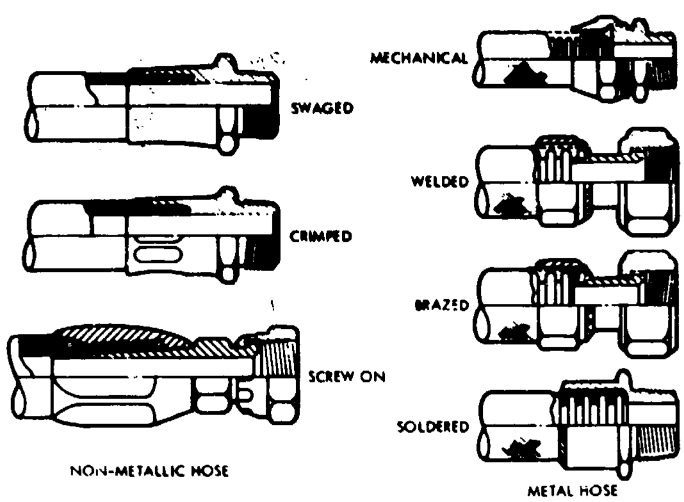


Figure 5.13.4.3. Methods of Fitting Attachment
(Reprinted from "Applied Hydraulics and Pneumatics," December 1959, vol. 12, no. 12, D. W. Fentress, Copyright 1959, Industrial Publishing Corporation, Cleveland Ohio)

a hose assembly is the method of installation. Good rules to practice in installing hose assemblies are:

1) Avoid Torsion. Install the hose such that bending occurs within a plane through the hose centerline. This can be controlled by locating the end fittings (Figure 5.13.4.4a) within the plane of bending.

2) Provide Slack. Install hose between coaxial fittings such that sufficient slack is available to compensate for length changes caused by pressurization, mechanical movement, or thermal expansion.

3) Avoid Short Bend Radii. Short bend radii impose higher stresses on the hose inner liner and reinforcement, resulting in reduced operating life and reduced operating pressure. Select elbow fittings and piping to prevent excessive bending (Figure 5.13.4.4b).

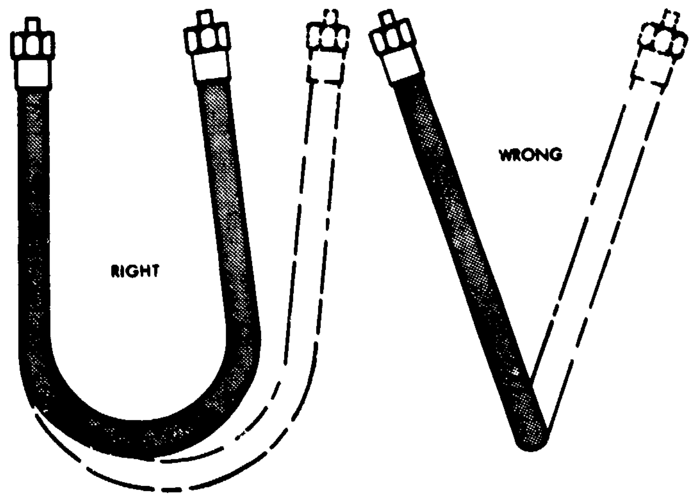


Figure 5.13.4.4a. Avoid Torsion in Connector Installation
(Reprinted from "Machine Design," December 1959, vol. 31, no. 26, H. W. Larose, Copyright 1959, Industrial Publishing Corporation, Cleveland, Ohio)

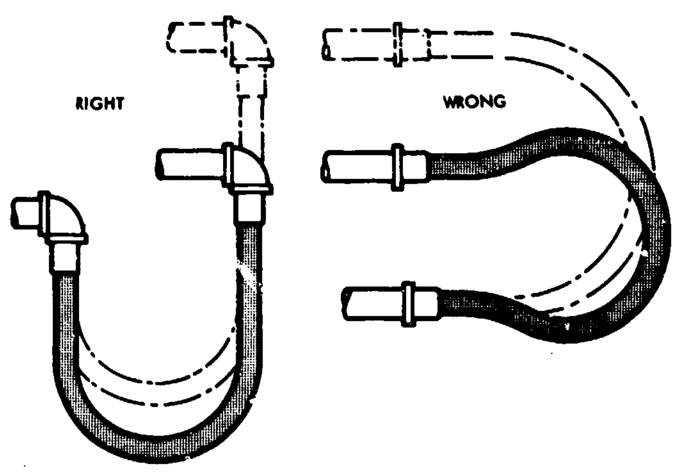


Figure 5.13.4.4b. Prevent Excessive Bending in Connector Installation
(Reprinted from "Machine Design," December 1959, vol. 31, no. 26, H. W. Larose, Copyright 1959, Penton Publishing Company, Cleveland, Ohio)

4) **Provide Heat and Abrasion Protection.** Select proper hose coverings to insulate the hose assembly from contiguous hot surfaces. Also, in installations where hose rubbing cannot be prevented, select a hose covering which is abrasion resistant and relatively frictionless.

5.13.5 Bellows Joints

A bellows joint is an elastic, corrugated, tubular connector used for conducting fluid between points of relative angular, transverse, lateral, or combined motion. The shape of the bellows connector is described by its free length, span, mean radius, inside diameter, outside diameter, convolution shape, and pitch (Figure 5.13.5a).

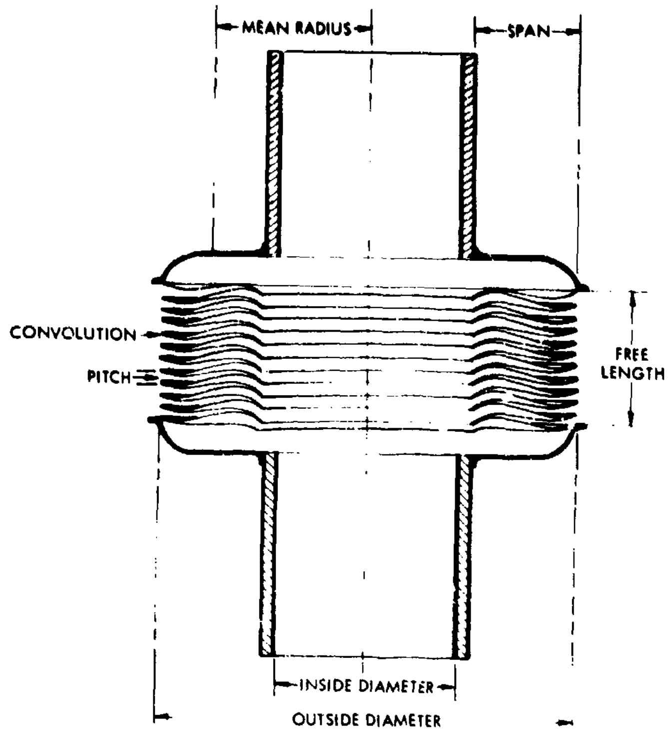


Figure 5.13.5a. Bellows Joint
(Courtesy of the Belfab Corporation, Daytona Beach, Florida)

Bellows joints are considered for use in applications when axial, lateral, angular, or combined deflections between contiguous interconnected fluid components exist. Examples of these applications are illustrated in Figure 5.13.5b.

The selection of a bellows joint involves choosing a bellows material and contour which will meet the compatibility and performance requirements established for the given application. Additionally, if axial expansion due to internal pressure is excessive, a means of restraint will be required. This Sub-Topic is limited to descriptions of bellows contours and restraints. The reader is referred to Sub-Section 6.6, "Bellows," for additional information. SAE ARP 735, "Aerospace Vehicle Cryogenic Ducting" includes a discussion of bellows joint design.

5.13.5.1 CHARACTERISTICS OF THE BASIC BELLOWS CONTOURS. The four basic bellows contours are the flat plate, nesting ripple, single sweep, and torus, as illustrated in Table 5.13.5.1. Numerous variations of each contour are possible by changing the bellows, materials, inside diameter, outside diameter, pitch, and/or material thickness. The chart in Table 5.13.5.1 can serve as a guide for selecting the bellows contour which best suits the intended application. Generally, the flat plate and nesting ripple contours are limited to low pressure applications (below 300 psig), and the single sweep and torus contours are used for high pressure applications (above 300 psig).

Table 5.13.5.1. Characteristics of Bellows Contours

	FLAT PLATE	NESTING RIPPLE	SINGLE SWEEP	TORUS
RESISTANCE TO PRESSURE	FAIR	POOR	GOOD	EXCELLENT
LONG STROKE CAPABILITY	POOR	EXCELLENT	FAIR	FAIR
LINEARITY OF FORCE OUTPUT WITH PRESSURE	EXCELLENT	POOR	FAIR	EXCELLENT
LINEARITY OF STROKE WITH PRESSURE	EXCELLENT FOR SHORT STROKE	GOOD	FAIR	GOOD
SPRING RATE	LOW	LOW	HIGH	HIGH

5.13.5.2 CHARACTERISTICS OF FLEXIBLE BELLOWS RESTRAINTS. Bellows restraints are incorporated in bellows assemblies to prevent axial expansion caused by internal pressure. When selecting a restraint, such factors as flexing capability (modes and moments), pressure drop, weight, and envelope size are considered. A list of such factors is presented in Table 5.13.5.2, where ratings are assigned for each of twelve restraint mechanisms. The table is intended to serve as a guide in selecting, designing, or modifying a restraint mechanism.

5.13.6 Flexible Coiled Tubing

The coiled tubing assembly consists of a length of tubing coiled to proper dimensions to permit flexing of the ends

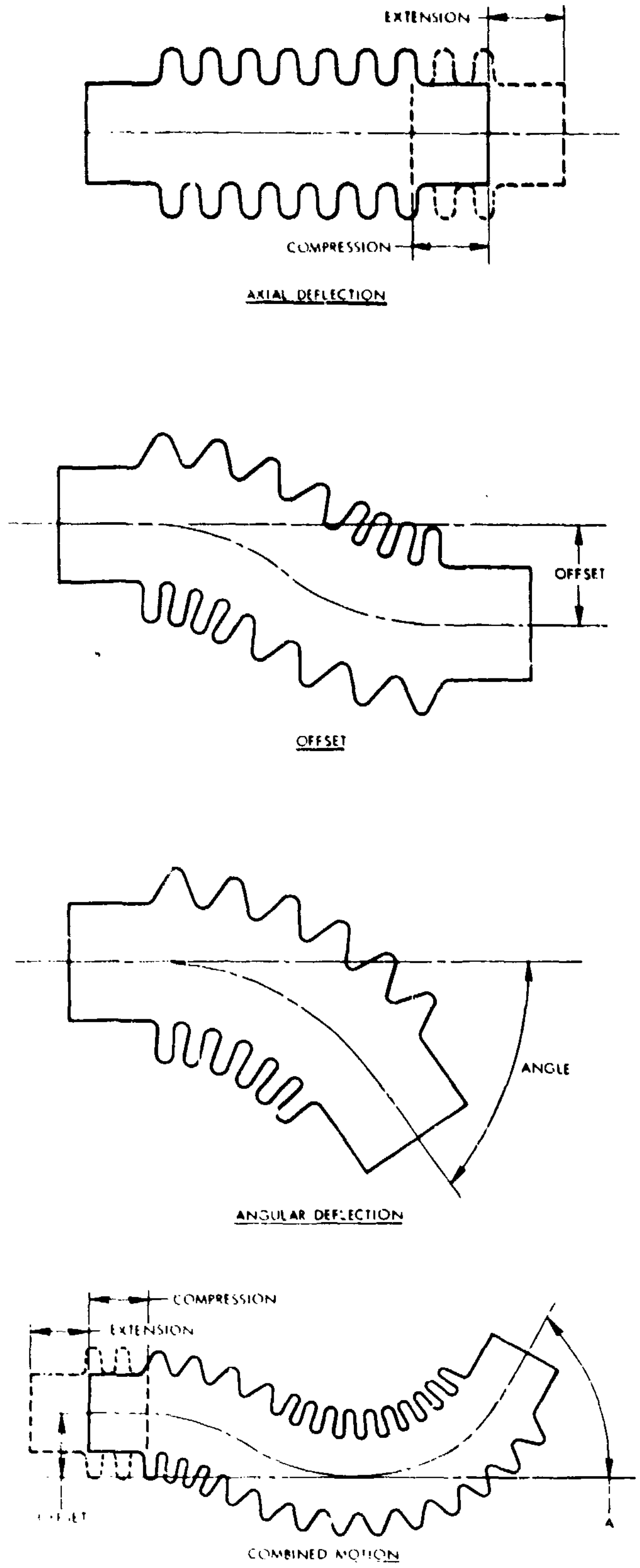


Figure 5.13.5b. Applications of Bellows Joints

without causing excessive reaction forces and stresses. A typical coiled tubing assembly is illustrated in Figure 5.13.6.

Flexible coiled tubing assembly design entails selecting the tubing material, determining tubing coil geometry, and selecting end fittings. Consideration must be given to compatibility with the fluid and environment, space limitations, and flexibility requirements. Factors affecting the selection of the material, geometry, and end fittings are discussed separately in the following Detailed Topics.

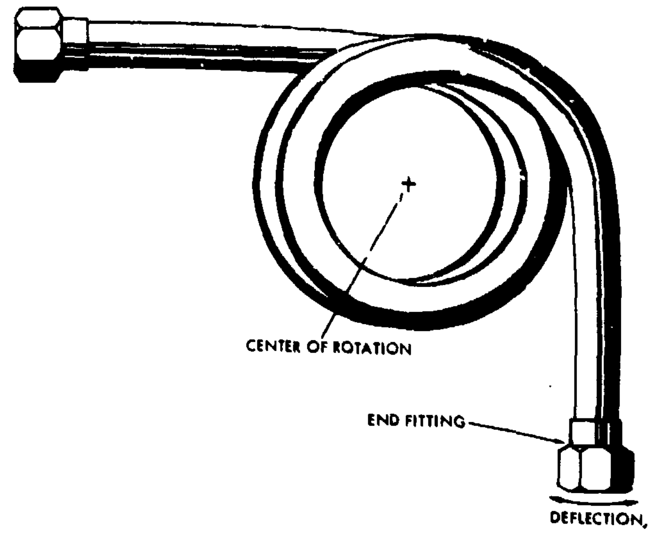


Figure 5.13.6. Coil Tubing

5.13.6.1 TUBING MATERIAL. The coiled tubing material is dictated by the fluid and environment compatibility, and strength requirements. Generally, the same material used in fluid lines throughout the system is suitable for the coiled tubing assembly. However, if greater flexibility is required, another material with lower modulus of elasticity may be used, providing it still meets the compatibility and strength requirements.

5.13.6.2 GEOMETRY. The following steps may be used to design a coiled tubing assembly:

a) Select tubing of the same diameter, wall thickness, and material as used in adjacent fluid lines.

b) Using a 10:1 coil diameter-to-tubing diameter ratio, lay out a coil with at least one turn, with connecting legs which depart from the coil tangentially and are coaxial with the connecting fittings.

c) Determine the relative motion between the points to be connected.

d) Verify that the coiled tubing assembly has ample clearance in the available space envelope.

e) Compute the stresses throughout the tubing assembly under the conditions of maximum coil unwind and wind-

Table 5.13.5.2. Factors in the Selection of Flexible Bellows Restraints

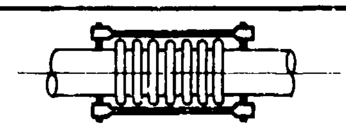
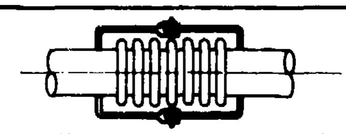
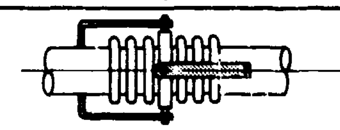
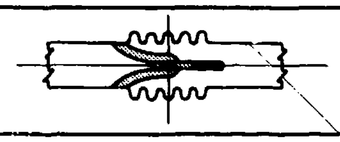
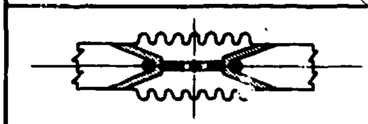
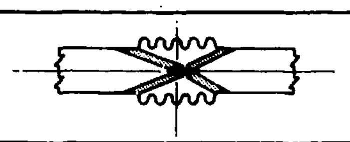
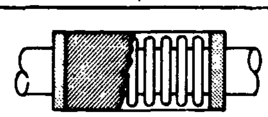
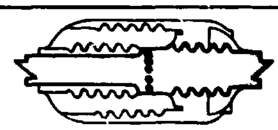
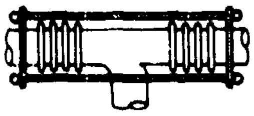
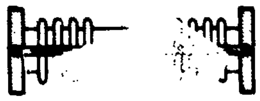
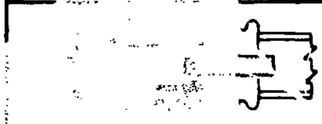
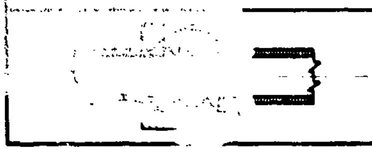
CONFIGURATION	FLEXIBILITY	LOAD TO DEFLECT	WEIGHT	PRESSURE DROP	ENVELOPE SIZE
	DOUBLE HINGE SINGLE PLANE ANGULATION, LATERAL OFFSET	LOW	MEDIUM	THAT OF BELLOWS ONLY	LARGE IN ONE PLANE
	SINGLE HINGE SINGLE PLANE ANGULATION	LOW	MEDIUM	THAT OF BELLOWS ONLY	LARGE IN ONE PLANE
	EXTERNAL GIMBAL ANGULATION IN ANY PLANE	LOW	HEAVY	THAT OF BELLOWS ONLY	LARGE
	INTERNAL HINGE ANGULATION IN ANY PLANE	LOW	LESS THAN EXTERNAL GIMBAL	1/3 VELOCITY HEAD OR MORE	SMALL (NOTHING EXTERNAL OF BELLOWS)
	INTERNAL CABLE ANGULATION IN ANY PLANE, LATERAL OFFSET	LOW	LESS THAN EXTERNAL GIMBAL	1/3 VELOCITY HEAD OR MORE	SMALL (NOTHING EXTERNAL OF BELLOWS)
	INTERNAL BALL AND SOCKET ANGULATION IN ANY PLANE	LOW	LESS THAN EXTERNAL GIMBAL	1/3 VELOCITY HEAD OR MORE	SMALL (NOTHING EXTERNAL OF BELLOWS)
	BRAIDED WIRE RESTRAINT ANGULATION IN ANY PLANE	HIGH	MEDIUM	THAT OF BELLOWS ONLY	SMALL (NOT MUCH LARGER THAN BELLOWS)
	PRESSURE BALANCED OR COMPENSATED AXIAL, ANGULATION IN ANY PLANE, LATERAL OFFSET	BELLOWS SPRING RATE	HEAVY	TWO LENGTHS OF BELLOWS	VERY CUMBERSOME

Table 5.13.5.2. Factors in the Selection of Flexible Bellows Restraints (Continued)

CONFIGURATION	FLEXIBILITY	LOAD TO DEFLECT	WEIGHT	PRESSURE DROP	ENVELOPE SIZE
	PRESSURE BALANCED AXIAL, ANGULATION IN ANY PLANE, LATERAL OFFSET	BELLOWS SPRING RATE	MEDIUM	THAT OF BELLOWS ONLY	LARGE AND VERY LONG
	SWING JOINT LATERAL OFFSET, ANGULATION IN ONE PLANE	BELLOWS SPRING RATE PLUS BEARING FRICTION	LOW	TWO LENGTHS OF BELLOWS	LARGE AND VERY LONG
	INTERNAL GIMBAL ANGULATION IN ANY PLANE	LOW	HEAVY	DEPENDS ON STRUCTURE IN FLOW STREAM	SMALL
	BALL-BELLOWS JOINT ANGULATION IN ANY PLANE	HIGH	HEAVY	THAT OF BELLOWS ONLY	LARGE

up. Reference 68-62, pages 336 through 358, discusses techniques used in analyzing the stress distribution throughout a coiled tubing assembly. This technique uses a combination of Castigliano's theorem of least work, and von Karman's correction for cross sectional distortion.

f) If in the stress analysis it is shown that the endurance stress is exceeded, it is apparent that greater flexibility is required. Flexibility can be increased by changing the material to one of lower modulus of elasticity, decreasing the wall thickness, increasing the number of coils, or locating the coil center farther from the points of relative motion.

REFERENCES

Flexible Hoses

1-35, 1-36, 1-49, 1-111, 1-229, 6-35, 6-66, 6-67, 19-170, 19-183, 29-1, 49-22, 51-7, 73-21, 115-1, 149-1, 193-4, 264-2, V-46, V-70, V-76, V-90, V-97, V-116, V-218, V-253, V-254, V-255, V-256

Bellows Connectors

1-58, 1-72, 1-158, 19-116, 19-139, 19-154, 19-172, 62-6, 73-1, 73-6, 73-8, 73-9, 73-92, 192-1, V-117

5.14 ROTARY FLUID JOINTS

5.14.1 Introduction

A rotary fluid joint is a coupling used for conducting fluid between points of relative motion. It consists essentially of a body, a tubular shaft, a rotary shaft seal, a bearing, and threaded ports (Figures 5.14.1a through 5.14.1c). Descriptions and uses of rotary joints for aerospace fluid system applications are discussed in this Sub-Section.

5.14.2 When to Use Rotary Fluid Joints

Rotary joints in conjunction with metal tubing are considered for use in lieu of flexible hose when:

- Greater stiffness is required to improve system response
- Continuous rotation is required
- Small radius bends are required
- Connector movement is limited to a small predetermined space
- High reaction moments and forces cannot be tolerated

5.14.3 Selection of a Rotary Joint

Rotary joints are presently available for use in cryogenic and high temperature (above 1000°F) fluid systems, moderate vacuum to high pressure (above 10,000 psi) applications, and corrosive fluid applications. The success of a given application depends upon judicious selection of seals, bearings, materials, and types of rotary joints. This chapter will dwell on descriptions of the types of rotary joints commonly used in aerospace applications. Sub-Section 6.4 "Dynamic Seals," 6.8 "Bearings," and 12.0 "Materials" contain information relative to these subjects.

5.14.4 Types of Rotary Fluid Joints

5.14.4.1 UNBALANCED ROTARY JOINTS. Unbalanced rotary joints may be of the ball and socket (self-aligning) or single plane (continuous rotation) designs (Figure 5.14.1a and 5.14.1b, respectively). In both types, internal pressurization causes thrust loading on the seals and bearings, resulting in increased operating torques. This disadvantage is offset somewhat by the low pressure drop characteristic of the straight-through design. In applications where pressure drop is critical and operating torque is unimportant, either of the two unbalanced designs is satisfactory. If self-alignment is required, the ball and socket joint (Figure 5.14.1a) is best suited, and if continuous rotation is required, the single plane rotary joint (Figure 5.14.1b) is preferred.

5.14.4.2 BALANCED ROTARY JOINTS. Balanced rotary joints are designed such that the swiveling or rotary tubular shaft is pressure balanced, eliminating high sealing and bearing friction forces caused by axial pressure thrust (Figure 5.14.1c). This design requires drilling the rotary shaft to form a fluid passage through the joint. The drilled passageway offers a significant flow restriction, which must be considered in selecting this type of joint. This joint is well suited in high speed, constant rotation applications, where friction and heat generation are critical.

REFERENCES

Unbalanced

19-160, 19-161, 19-162, 19-184, 49-37, 339-1, V-21, V-72, V-81, V-126, V-259.

Balanced

6-116, 19-160, 19-184, 339-1, V-97, V-126.

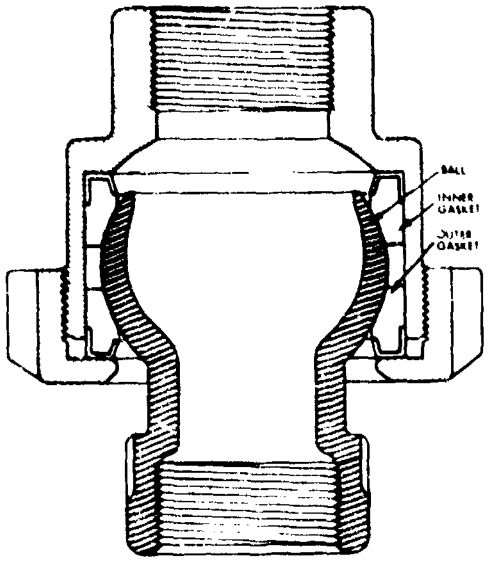


Figure 5.14.1a. Ball and Socket Rotary Fluid Joint
(Courtesy of Aeroquip Corporation, Jackson, Michigan)

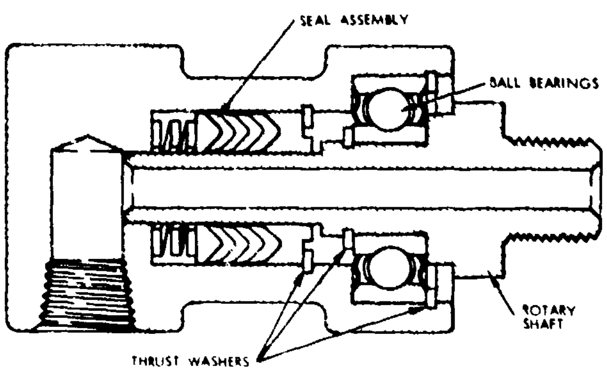


Figure 5.14.1b. Single Plane Rotary Joint
(Courtesy of Aeroquip Corporation, Jackson, Michigan)

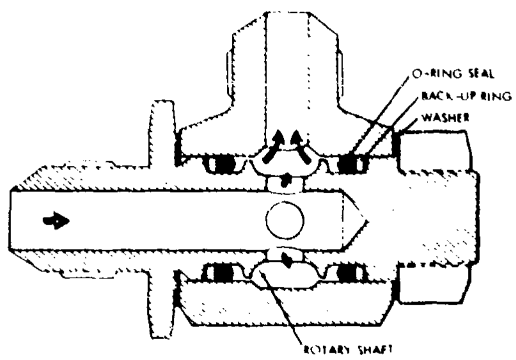


Figure 5.14.1c. Pressure Balanced Rotary Joint
(Courtesy of Aeroquip Corporation, Jackson, Michigan)

5.15 INSTRUMENTATION

5.15.1 Introduction

This Sub-Section describes the basic operational and performance characteristics of instruments which sense pressure, temperature, liquid level, and position.

For each instrument category (temperature, pressure, etc.) typical data characterizing various instrument types are presented in comparison charts. The generalizations made in the charts can only be considered typical, since they do not necessarily reflect all the many design variations between instruments using essentially the same principle of operation. Wherever possible, manufacturers' data should be consulted to ascertain the performance standards of specific, available instruments. An extensive manufacturers' survey of instruments and their characteristics has been published as the *ISA Transducer Compendium* (Reference 460-1) by the Instrument Society of America.

References which provide greater topic depth are included in the text.

5.15.2 Pressure Instrumentation

5.15.2.1 COMPARISON CHART FOR PRESSURE INSTRUMENTS. Table 5.15.2.1 characterizes typical selection data for pressure instruments, ranging from vacuum to high pressure applications.

5.15.2.2 LIQUID DISPLACEMENT GAGES. Four types of liquid displacement gages will be discussed.

U-Tube Manometer. A U-tube manometer measures the differential pressure impressed across the two legs of the U-tube. The magnitude of the pressure is determined by measuring the liquid height differential and multiplying it by the specific weight of the liquid. Thus

$$\Delta P = w\,(h_2 - h_1) \qquad \text{(Eq 5.15.2.2a)}$$

where ΔP = pressure difference, psf

w = specific weight, lb_f/ft^3

$h_{1,2}$ = respective liquid heights in the manometer legs, measured from a common datum plane, ft

The additional pressure exerted by the difference in height of liquid or gas in the manometer legs is small, and is neglected in Equation (5.15.2.2a). A variation of the U-tube manometer is the well manometer, where the differential pressure is determined by measuring the height of one leg only. In this manometer, there is a large difference between the cross-sectional areas of the two manometer legs; thus there is a negligible change of liquid height in the large leg (well) compared to the measured change in the small leg.

The most commonly used liquids for manometers are water and mercury. Pressure differentials are often recorded in terms of the liquid used. With water, ΔP is recorded in inches of water; with mercury, inches Hg or millimeters Hg are the most common units.

The height of the liquid is subject to several correction factors which include variation of density with temperature, variation of gravitational force with latitude and elevation, meniscus, and thermal expansion. Correction factors are tabulated in Reference 160-38, pages 62 and 63.

Absolute Pressure Gages. This instrument is a mercury filled U-tube with one leg completely evacuated and sealed, and the other leg open to the measured pressure. Since there is zero pressure above the sealed leg, absolute pressure is the difference in height of the mercury column multiplied by the specific weight of mercury. Thus

$$P = w\,(h_2 - h_1) \qquad \text{(Eq 5.15.2.2b)}$$

where P = absolute pressure, psf

w = specific weight, lb_f/ft^3

$h_{1,2}$ = respective heights of mercury in the manometer legs, measured from a common datum plane

The absolute pressure may also be recorded in terms of the height of the mercury column (e.g., in Hg, or mm Hg).

Inclined Tube. The inclined tube (Figure 5.15.2.2a) is a well-type manometer with the small leg inclined, thus producing a longer scale which increases the accuracy of visual measurement. The differential pressure, determined by the height of liquid rise, h, is related to the length of the liquid column measured along the inclined tube by the expression

$$\Delta P = wL \sin \alpha \qquad \text{(Eq 5.15.2.2c)}$$

where w = specific weight

L = length of liquid column in inclined tube angle of inclination

α = measured above a fixed datum (zero line)

The liquid used most often is mercury. By reducing the angle α, the scale length can be greatly increased; for best

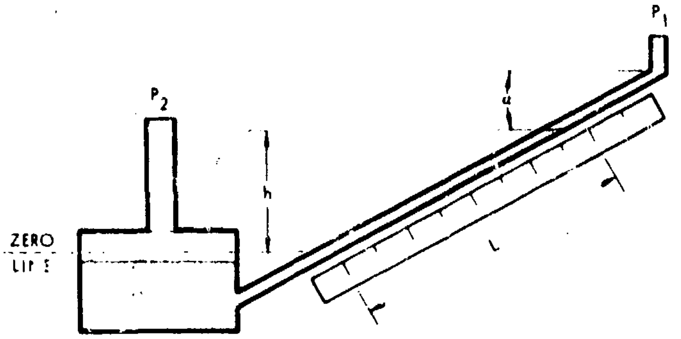


Figure 5.15.2.2a. Inclined Tube Manometer

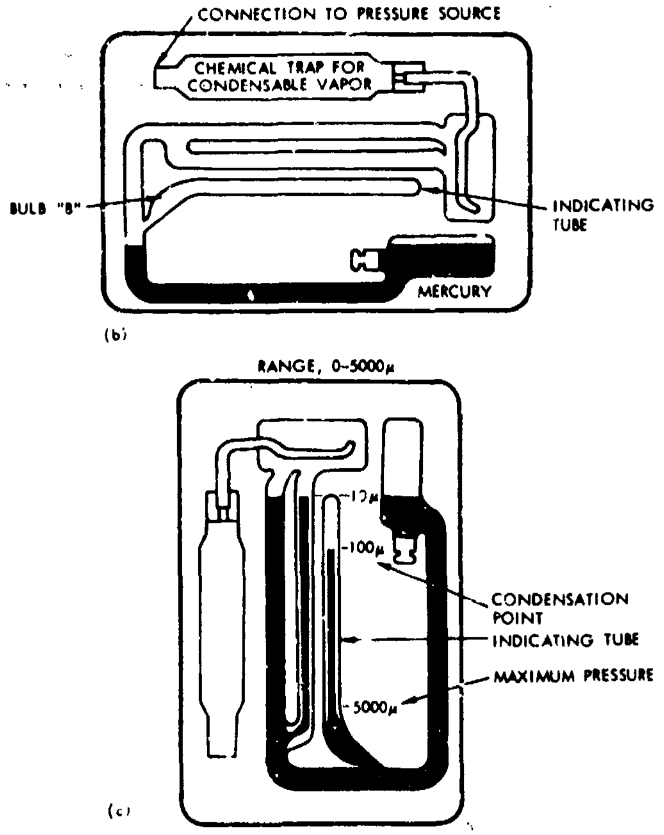


Figure 5.15.2.2b,c. McLeod Gage in (b) Charging Position; in (c) Reading Position

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operation, however, the ratio of L to h should not exceed 10:1. Only small pressure differentials can be measured with this device (Reference 410-1).

Pressure Multiplying Manometer (McLeod). The McLeod gage is a direct reading device which measures low absolute pressures. A fixed and known volume of gas in a vacuum system is trapped by mercury in the gage when it is tilted from a horizontal charging position (Figure 5.15.2.2b) to the measurement position (Figure 5.15.2.2c). The gas entrapped is compressed into a much smaller known volume in the indicating tube. The pressure exerted by the compressed gas is then sufficient to support a measurable column of mercury. The original pressure may be found by Boyle's law. Since Boyle's law applies only to non-condensable gases, higher vacuums than actually exist will be measured if condensation takes place. There is no correction or multiplication factor to adjust or compensate for the error.

The simplest method of protection against condensation errors is to provide a built-in chemical trap comprised of a glass tube, large enough to cause no flow restriction and filled with a chemical having a vapor pressure well below the minimum pressure to be encountered. The chemical trap allows true total pressure including condensables and non-condensables to be measured. Although only dry air is in the indicating tube, this air is present at a pressure balanced by the total pressure of condensable vapor and air on the other side of the trap (Reference 160-25).

This device is not continuous reading, is awkward to manipulate, and requires at least one minute for pressures to equalize (Reference 166-3). Unless a chemical trap is provided, vapors of the mercury will diffuse into the vacuum system. Despite these shortcomings, this device is used extensively as the high vacuum standard. More details of the McLeod gage can be found in Reference 160-25.

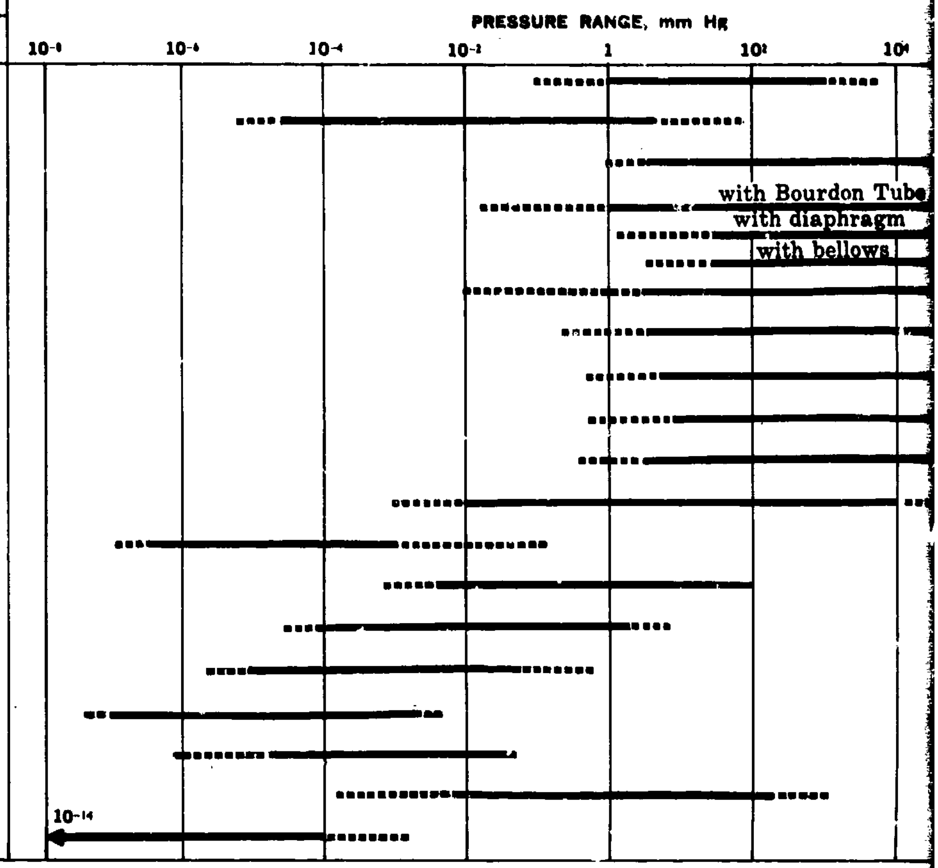
5.15.2.3 BOURDON TUBE GAGES. As defined in Reference V-37, a bourdon tube refers "to any of a number of pressure-responsive elements which are fabricated from tubing flattened into some non-circular cross-section, either curved or twisted along their length, which respond to a change in pressure by movement or rotation of one or both unrestrained ends."

Principle of Operation. The movement or rotation of the unrestrained end of the bourdon tube is caused by a change in internal pressure. The tube deforms elastically until the stresses created in the tube balance the forces arising from a pressure change. Increasing pressure tends to return the flattened cross-section to its original circular shape. Opposing compressive and tensile stresses create a bending moment that tends to change the curvature of the tube (Reference V-37).

Tube Types. Four types of Bourdon tubes commonly used are plain, spiral, helical, and twisted. Each is shown in Figure 5.15.2.3a with various cross-sections used in their design.

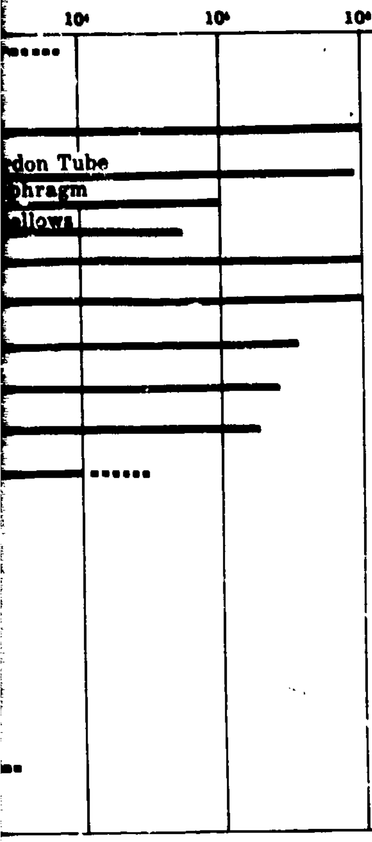
Table 5.15.2.1. Comparis

GAGE TYPE	PRESSURE: ABSOLUTE	PRESSURE: GAGE	PRESSURE: ΔP	PRESSURE: VACUUM
U-tube manometer	X	X	X	X
McLeod Gage	X			X
Bourdon Gage	X	X	X	X
Potentiometer	X	X	X	X
Strain gages (wire)	X	X	X	X
Piezoelectric	X	X		X
Capacitance	X	X		X
Magnetic (inductance)	X	X		X
Magnetic (reluctance)	X	X		X
Vibrating element	X	X		X
Knudsen	X			X
Thermocouple	X			X
Pirani	X			X
Thermistor	X			X
Triode	X			X
Cold cathode (philips)	X			X
Alphatron	X			X
Ultra high vacuum	X			X



Normal range
Extended range
H = High
I = Indefinite
L = Low
M = Medium
N = No
S = Sometimes
T = Temporary
Y = Yes

Comparison Chart for Pressure Instruments



ACCURACY (%)	CONTINUOUS READING	MOVING PARTS	SENSITIVE TO GAS COMPOSITION	ELECTRICAL PARTS	OPTICAL PARTS	AFFECTED BY CONTAMINATION	RESISTANCE TO SHOCK	LIFE EXPECTANCY	OUTPUT POWER	FREQUENCY RESPONSE	RESOLUTION
0.05-4	S	Y	Y	N	S	Y	N	I		H	M
2	N	Y	Y	N	S	Y	N	I		—	M
0.5-2	Y	Y	S	N	N	Y	S	T		H	M
0.5-2	S	Y	S	Y	N	N	N	T	H	M	LM
0.25-1	S	Y	N	Y	N	N	Y	T	L	H	VH
0.5-2	S	Y	N	Y	N	N	Y	I	M	H	H
1-2	S	Y	N	Y	N	N	N	I	L	H	H
2	S	Y	N	Y	N	Y	N	I	MH	H	H
2	S	Y	N	Y	N	N	Y	I	M	H	H
0.1	Y	Y	N	Y	N	Y	Y	I	H	H	H
	S	Y	Y	N	Y	S	N	I	L	L	L
	N	N	N	Y	N	S	Y	T	L	L	L
	Y	N	N	Y	N	S	Y	I	L	L	L
	S	N	N	Y	N	S	Y	I	L	L	L
	Y	N	N	Y	N	Y	S	T	L	L	L
	Y	N	N	Y	N	S	Y	I	L	L	L
	Y	N	N	Y	N	S	S	T	L	L	L
	Y	N	N	Y	N	Y	N	I	L	L	L

B

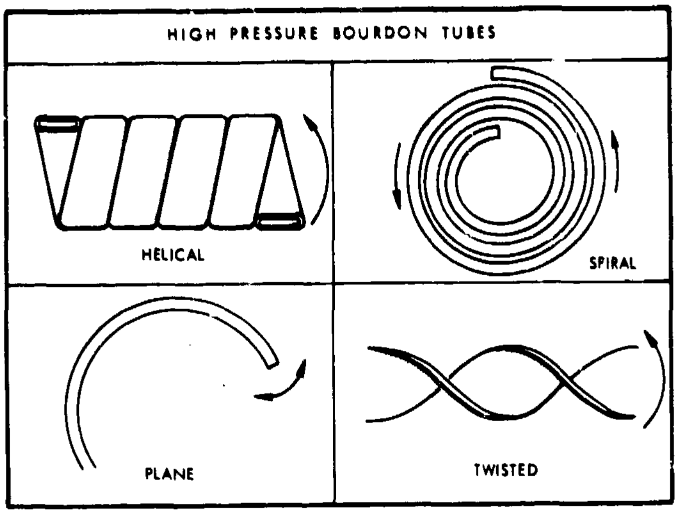


Figure 5.15.2.3a. Bourdon Tube Designs

The plain tubes are either U-tubes or C-tubes. In the U-tube the pressure inlet is located at the center of the tube length, and the two ends are free to move. In the C-tube, the pressure inlet is at one end and fixed with the other end free to move. The C-tube configuration is the most widely used for gages; a more complete description is presented later.

The spiral tube is similar to the plain tube except that its radius of curvature changes along the length of the tube, creating a greater length of tube for a given circumference. The number of turns in the spiral vary from three to six. Due to the larger active length of tube, the arc length of stroke is greater. A straight-line pickoff from this larger angle of sweep causes a larger non-linearity than with a plain Bourdon tube.

The helical tube permits a long tube length for a given radius of curvature. The average tube may have from two to six turns. The movement of the tip will be greater than that for the plain tube and will be more nearly circular. The increase in length of the helical tube is negligible, and the travel of the tip lies in a plane perpendicular to the centerline of the helix. The tip travel will increase as the active tube length increases and will have the same order of magnitude per unit length of tube as an equivalent plain Bourdon tube.

The twisted tube maintains a straight centerline along the length of the tube, but is twisted about the centerline at a uniform rate. The opening of the tube due to internal pressure causes the tube to unwind some finite angle about the tube centerline. The change in tube length is negligible. Different angles of twist alter the angular rotation for any given length of tube.

The design, manufacturing, and performance considerations of these tubes are described in detail in Reference 156-1.

The Bourdon Pressure Gage. The most common type of Bourdon gage uses a plain C-tube as its pressure-sensing element, connected to an indicator or pointer through a mechanical linkage. The various gage components are depicted in Figure 5.15.2.3b, and their individual characteristics are described in Reference 1-14, pages 120 to 128.

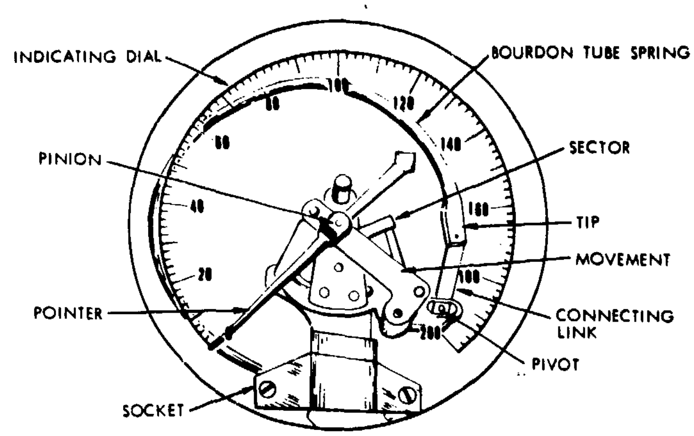


Figure 5.15.2.3b. Bourdon Pressure Gage
(Reprinted with permission from "Machine Design," 23 July 1959, vol. 31, no. 15, W. L. Buffenmyer, Copyright 1959, Penton Publishing Company, Cleveland, Ohio)

Pressure gages for system use are commonly classified according to accuracy (by ASA standards), style of case and end use (Table 5.15.2.3). Pressure gages with accuracies of 0.1 percent are used for calibrating instruments. A notable example is the Heise gage. Dial sizes are approximately 12 to 18 inches for accurate visual reading.

Table 5.15.2.3. Gage Classification Methods
(Reference 1-14)

ASA STANDARDS

Grade AA — Test Gages: Error in pressure indication at any point on scale does not exceed one-half percent of maximum pressure for which scale is graduated. Widely used in laboratories, production, testing, field inspection, test stands, and recalibration work. Also used for process and general industrial applications. Available with better accuracies, such as one-fourth percent, for more exacting measurements.

Grade A — High-Grade Commercial Gages: Error in pressure indication does not exceed 1 percent of scale range at any point within middle half of scale and 1½ percent for rest of scale.

Grade B — Commercial Gages: Error in pressure indication does not exceed 2 percent of scale range at any point within middle half of scale, and 3 percent for rest of scale.

CASE STYLES

Drawn Case: Usually ASA Grade B type in dial sizes from 1½ to 4½ inches. Can be used economically on original equipment where volume production requires large gage quantities.

Table 5.15.2.3 (Continued)

Cast Case: Meet ASA Grade A standards or better in dial sizes from 3½ to 16 inches, or larger.

END USES

Commercial: Usually of drawn-case type with ASA Grade B accuracy. Used primarily on items for domestic use, household heating units, fire extinguishers, small pumps, and compressors. Have low unit cost and are devoid of refinements to simplify maintenance and repairs. Often called "throw-away" gages.

Industrial: Usually of cast-case type with ASA Grade A accuracy. Used primarily in industrial application for measuring steam, oil, and water-line pressures. Also used on equipment destined for such applications. Have limited adjustments for repair and maintenance.

Process: Usually of cast-case type with ASA Grade AA accuracy. Used primarily for measuring process pressures in oil, power, and chemical plants. Have long life and reliability, and provide for maximum ease of repair and maintenance.

Test: Includes all gages of ASA Grade AA or better, regardless of application.

The above instruments measure gage pressure. By installing another tube, wound in a direction counter to the first tube, differential pressure can be measured. As shown in Figure 5.15.2.3C, one tube is held rigid at one end, while the other is linked to the pivot end of a second tube. The free end of the second tube measures the relative movement of both elements (Reference 410-1).

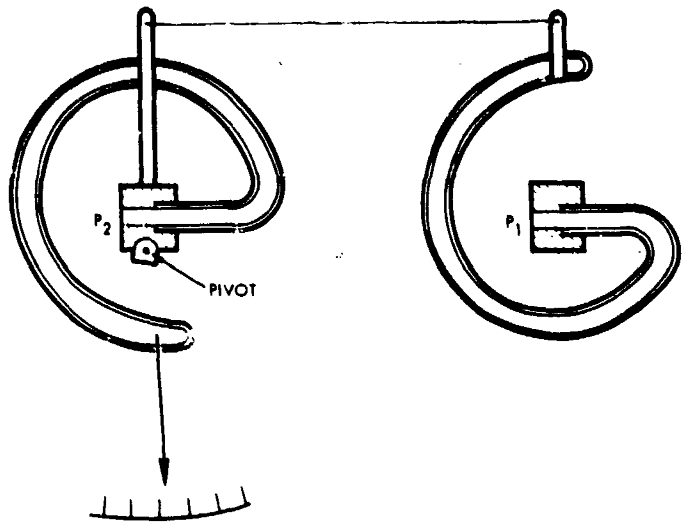


Figure 5.15.2.3c. Differential Pressure Bourdon Tube Gage
(Reprinted with permission from "Process Instruments and Controls Handbook," Considine, (ed), Copyright 1957, McGraw-Hill Book Company, Inc., New York, New York)

5.15.2.4 POTENTIOMETER GAGES. A typical potentiometer gage has a flexible sensing element (e.g., diaphragm, bellows, or bourdon tube) that responds to changes in pressure by moving an attached wiper arm across the resistance coil of a potentiometer. As shown in Figure 5.15.2.4, a balancing potentiometer in a bridge circuit is positioned until the ammeter reads zero. Voltage, e/E, is proportional to the pressure. The elements comprising a potentiometer gage are discussed in the following paragraphs.

The Diaphragm. A diaphragm is a thin circular plate supported around its periphery. When pressure is impressed against one of its surfaces, the diaghragm deflects. Diaphragms may be classed into two groups, flat and convoluted. The term *corrugated* is used interchangeably with *convoluted*.

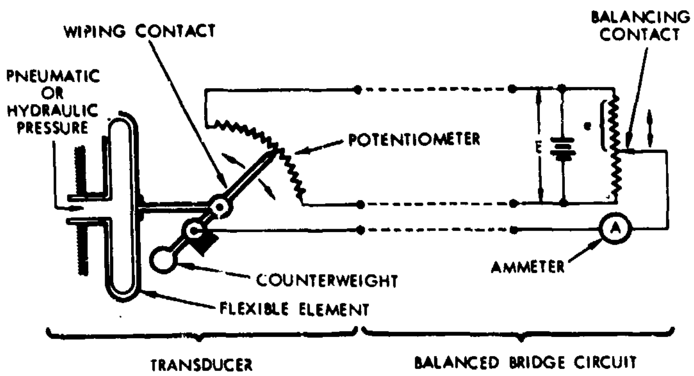


Figure 5.15.2.4. Potentiometer Gage
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The flat diaphragm group encompasses the true flat, stretched, spherical and catenary designs. The true flat and stretched diaphragms can be used to measure plus and minus pressures, since they can deflect either way. Spherical and catenary diaphragms can be used only when external pressure is greater than internal pressure.

Many designs contain a centrally located reinforcing "boss" to transmit the deflection. The boss will not affect linearity as long as its diameter is less than 40 percent of diaphragm's effective area, and is constructed of the same material as the diaphragm. The deflection of a catenary diaphragm is a function of the support tube, and not the diaphragm. The tube material's proportional limit will determine the maximum linear displacement and its yield stress will limit the maximum operating pressure. Typical materials for flat diaphragms are stainless steels, AISI 347, 416, and 410 (Reference 156-1).

Convoluted diaphragms are normally used in pairs, bonded together by soldering, brazing, and welding, forming a capsule. The total deflection is equal to the sum of the displacement of all capsules stacked axially. Although exceptions exist, capsulated diaphragms will be linear to within 1 percent if the displacement is below 2 percent of diaphragm diameter. Ni-Span C* is a commonly used material for convoluted diaphragms, although Inconel X, K Monel, phosphor bronze, beryllium copper, and stainless steel are also widely used.

The diaphragm can be located at the tip of the pressure port or end of the case (flush diaphragm), or it can be installed deep within the case (recessed diaphragm). In the event of material incompatibility, a thin membrane of a compatible material is used to isolate the diaphragm from the fluid, with silicone oil between the membrane and diaphragm to transmit the pressure (Reference 74-35).

The design and manufacture of diaphragms is presented in detail in Sub-Section 6.7 and References 156-1 and 410-1.

The Bellows. A bellows is a one-piece expandable and collapsible pressure-sensing device which is axially flexible. The convoluted seamless bellows is the most commonly used type. Other types such as the welded, machined, and electroless metal-deposited bellows, are primarily custom made. Pressures may be applied internally or externally to the bellows. If the bellows' length exceeds its outside diameter, pressure is usually applied externally, reducing the stresses which would tend to distort or buckle the bellows if internal pressure were applied.

Typical bellows materials include stainless steels, Monel, beryllium copper, and phosphor bronze. Hastelloy and Inconel X have been used for corrosive fluid and high temperature service, respectively.

Details of design and use are described in Sub-Section 6.6.

The Bourdon Tube. See Detailed Topic 5.15.2.3 for a discussion of Bourdon tubes.

The Potentiometer. A potentiometer is an instrument which measures an unknown potential difference by balancing the difference against a known potential difference. The resistance ranges of a potentiometer lie usually between 100 and 30,000 ohms.

For the most part, the potentiometer limits the response time and resolution of the complete gage. The large mass of the potentiometer and its wiper keep the response time of the gage relatively large. Inertia and friction must be overcome, as well as its large displacement for pressure change. The resolution of a wirewound potentiometer is limited by the number of turns of the resistance element. For example, a 100 turn potentiometer detects a pressure change when the wiper moves from one turn to an adjacent turn, or changes no smaller than 0.01 or 1 percent of full scale. Some gages use film potentiometers where resolution is stepless and is limited only by the film granules.

Gage Types. Potentiometer gages, in various configurations, may be used to measure absolute, gage, and differential pressure. A common design uses two sensing elements positioned side by side and linked together across a fixed pivot. If one sensing element is evacuated, pressure applied externally to the sensing element will be measured in absolute, gage, or differential terms, depending on whether the second element is respectively under vacuum, barometric pressure, or at some reference pressure.

5.15.2.5 STRAIN GAGE PRESSURE TRANSDUCERS.

Two types will be discussed, metal wire and piezoelectric.

Metal Wire. A metal wire strain gage pressure transducer consists of an elastic element that converts pressure into displacement, and a wire resistance strain gage element that measures the displacement (Reference 19-122). When the sensing element is displaced, the wire length and diameter are altered and its electrical resistance is changed. If only two wire strain gage elements are used, a half-bridge is formed by connecting them in series. Most transducers use four active elements connected as a full bridge, and mounted to the deforming member such that two elements are in tension and two in compression when the elastic element deflects. Connecting the two resistance-increasing elements and the two resistance-decreasing elements in diagonally opposite arms of the bridge results in maximum output.

Strain gage elements may be either bonded or unbonded. Some designs bond the strain gage element directly to pressure-sensing element, such as a Bourdon tube or diaphragm. More commonly, a secondary or auxiliary member is used as the deforming member. Such members include a cantilever beam, a force ring (Flader sensing ring), or a thin-walled cylinder (strain tube). The bellows is commonly used as the sensing element for the cantilever beam. For pressures above atmosphere, a single bellows is used; for differential pressures, a second bellows is installed axially opposing the movement of the first bellows. When measuring absolute pressure, the second bellows is evacuated.

A diaphragm and piston are used as the sensing element for the force ring. The catenary diaphragm is used as the sensing element for the strain tube. A complete description of these auxiliary members and sensing elements is presented in Reference 19-122.

In the unbonded gage, the wire elements are stretched and unsupported between a frame and a movable armature. The sensing element for low pressure is usually a metal

*Registered trademark of International Nickel Company.

bellows; for high pressure, a diaphragm. The diaphragm is actually a rigid disc of great strength. The sensing element isolates the measuring elements from fluid yet transmits the force, or deformation, to the gage armature.

All strain gage transducers contain compensating and adjusting resistors which permit zero and balance adjustment, thermal zero-shift compensation, and sensitivity adjustment. With the added voltage drop across the compensating resistors, full scale output voltage may be reduced as much as 50 percent. A typical resistance network is shown in Figure 5.15.2.5a.

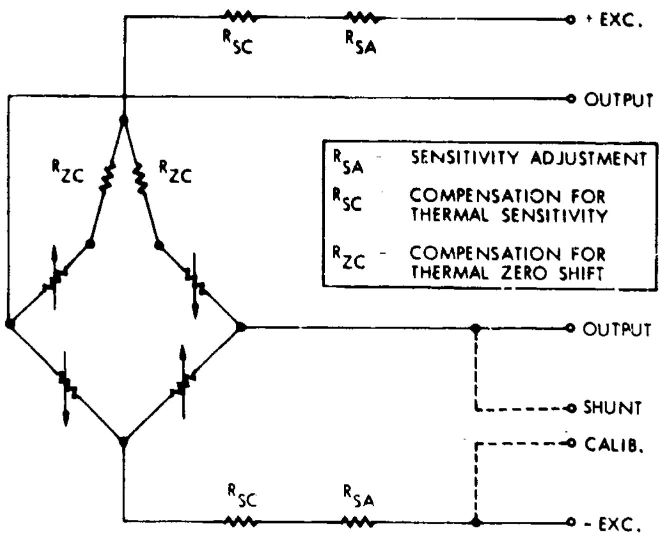

Figure 5.15.2.5a. Resistance Network for a Full-Bridge Strain Gage Pressure Transducer
(Reprinted with permission of "Instruments and Control Systems," March 1963, vol. 36, no. 3, H. N. Norton, The Instruments Publishing Company, Inc., Pittsburgh, Pennsylvania)

Piezoelectric. Elastic deformation of certain crystals along defined stress planes produces an electric potential in the crystal. Active crystals include quartz, tourmaline, barium titanate, and Rochelle salts. The sensitivity of most crystals is approximately 10^{-11} coulombs/psi. They are generally used in the measurement of transient pressures from fractions of a psi up to approximately 10^{4} psi, with durations from 10^{-3} sec to several seconds (Reference 111-24). The signal response to pressure variations is linear, since it follows the elastic stress-strain curve of the crystal.

The advantages of strain gage transducers are fast response (3-5 milliseconds), high resolution, minimum motion of mechanical parts, ease of temperature compensation, and good shock and vibration resistance.

The disadvantages of strain gage transducers are the difficulty of obtaining zero output at zero pressure, and the need for signal amplification since output voltage levels are below standard telemetry levels.

A schematic of the transducer system is shown in Figure 5.15.2.5b, and a summary of strain gage pressure transducer characteristics is shown in Table 5.15.2.5. The natural frequency of the gages is indicative of frequency levels for resonance.

Some gages use a single crystal but, more commonly, crystals are stacked in parallel to retain the high frequency response of the small crystal and provide the necessary sensitivity common to larger crystals. A typical transducer is shown in Figure 5.15.2.5c. Electric and magnetic shielding and short cable lengths are required to protect the low voltage generated from external noise (References 111-24 and 410-1).

The primary advantages of piezoelectric gages are linearity, high frequency response, high sensitivity, and good reproducibility under controlled temperature conditions (Reference 19-113).

The primary disadvantages are sensitivity to temperature change, vibration, and electrical or magnetic noise; weak signal generation, requiring high-gain amplifiers (Reference 19-113).

Table 5.15.2.5. Characteristics of Strain Gage Pressure Transducers
(Reference 19-122)

CLASS OF SENSING ELEMENT	TYPE STRAIN GAGE	AVAILABLE RANGES, psia	ACCURACY PERCENT, FULL RANGE	INPUT CURRENT	NATURAL FREQUENCY, cps
I — Bourdon Tube and Bellows	Bonded	0-10 to 0-50,000	± ¼	AC or DC	300 to 1,000
I — Bellows and Diaphragm	Unbonded	0-0.05 to 0-10,000	± ½	AC or DC	270 to 10,000
II — Bellows and Cantilever Beam	Bonded	0-10 to 0-2500	± ½	AC or DC	—
III — Ring Type	Bonded	0-25 to 0-10,000	± 1	AC or DC	Above 1,500
IV — Catenary Diaphragm	Bonded	0-100 to 0-10,000	± 1	AC or DC	20,000 to 45,000

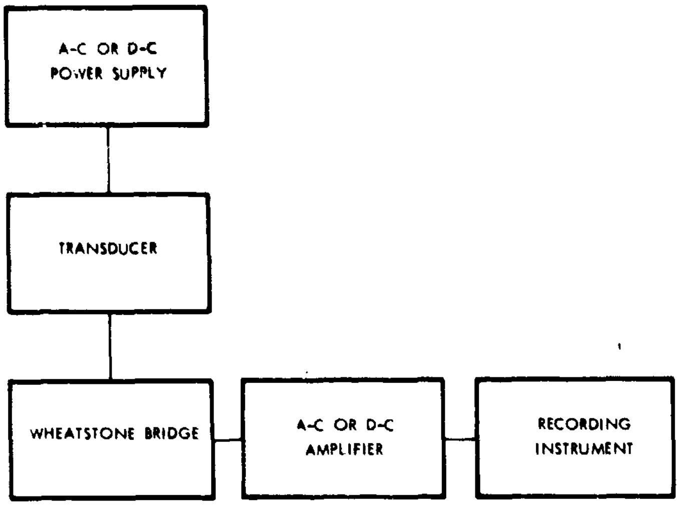

Figure 5.15.2.5b. Schematic Diagram for a Strain Gage Transducer System

(Reprinted from "Product Engineering," January 1954, vol. 25, no. 1, J. Grey, Copyright 1954, McGraw-Hill Publishing Company, Inc., New York, New York)

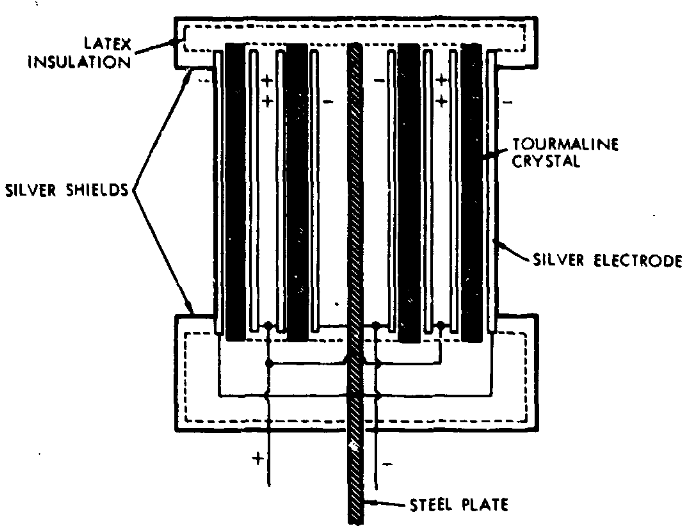

Figure 5.15.2.5c. A Piezoelectric Transducer

A schematic diagram of a crystal transducer system is shown in Figure 5.15.2.5d. For more information consult References 19-113 and 410-1.

5.15.2.6 **VARIABLE CAPACITANCE.** Electrical capacitance transducers may be either distance-sensitive or pressure-sensitive. The distance-sensitive transducers use two parallel plates — one stationary, the other subject to pressure. As the pressure varies, the latter plate deflects, changing the distance between the plates and, therefore, the

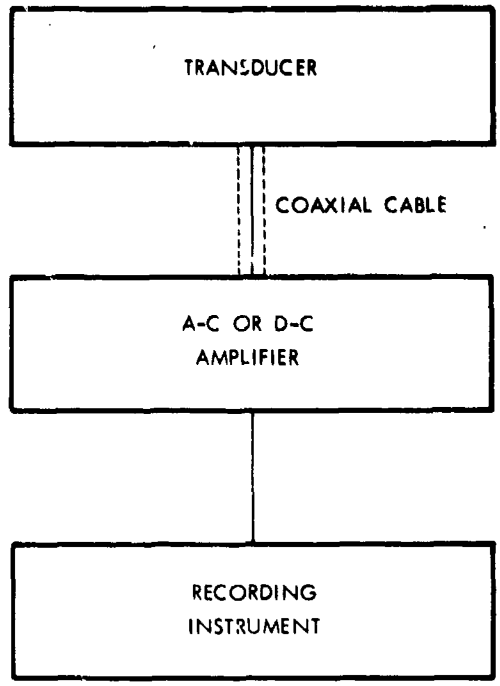

Figure 5.15.2.5d. Schematic Diagram for a Piezoelectric Transducer System

(Reprinted from "Product Engineering," January 1954, vol. 25, no. 1, J. Grey, Copyright 1954, McGraw-Hill Publishing Company, Inc., New York, New York)

capacitance. The pressure-sensitive transducer uses a crystal, similar to those used in the piezoelectric transducer, which changes dielectric constant with pressure. The capacitance is proportional to the dielectric constant between the plates.

The advantages of the variable capacitance transducer include small size, high frequency response, good linearity, and resolution. The disadvantages are temperature drift, low shock and vibration resistance, and more complex electronic equipment (Reference 19-113). A schematic diagram of a capacitance transducer system is shown in Figure 5.15.2.6.

5.15.2.7 **MAGNETIC TRANSDUCERS.** Magnetic transducers are of two types, inductance and variable reluctance.

Inductance. The inductance transducer is a linear variable differential transformer (LVDT) in which the inductance of a coil is changed by the introduction of a movable core of magnetic material attached to a pressure sensor, e.g., a bellows or diaphragm. The center or primary coil is supplied with alternating current. The inductance of the remaining two coils is changed by equal but opposite amounts as the core moves. The differential transformer doubles the sensitivity of the single winding inductance coil. This transducer combines high accuracy and linearity with low mechanical friction and high power output. However, it has a slow response time and, generally, has little resistance to vibration or shock. An example is shown in Figure 5.15.2.7a.

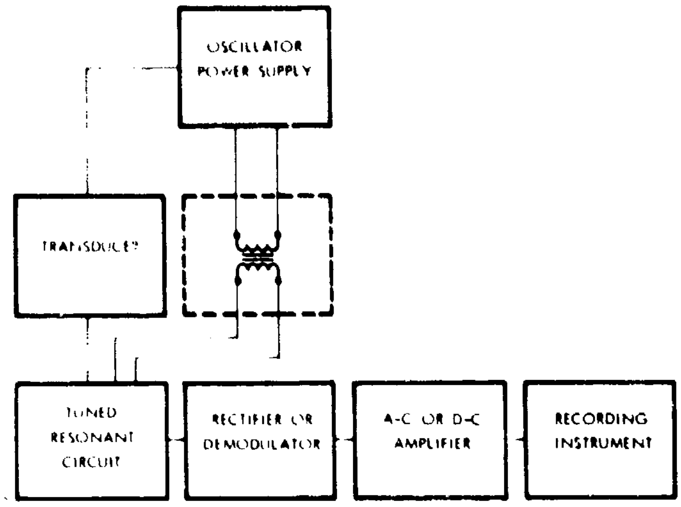

Figure 5.15.2.6. Schematic Diagram for a Capacitance Transducer System

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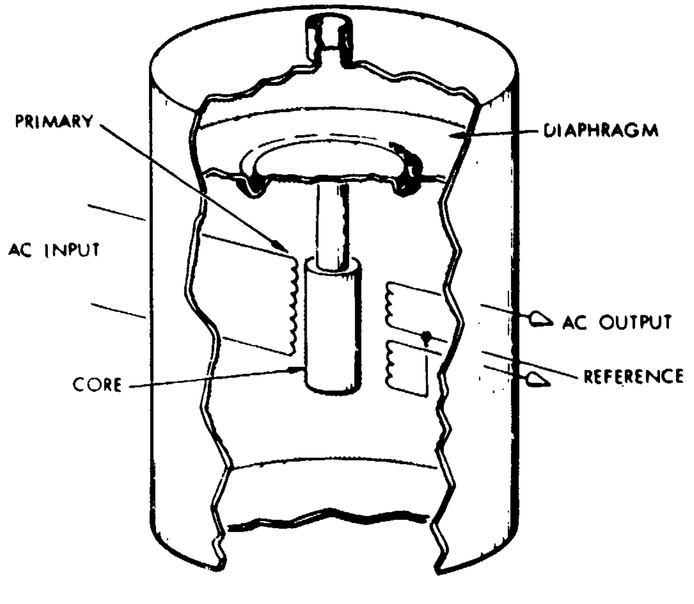


Figure 5.15.2.7a. LVDT Transducer

(Reprinted from "Product Engineering," 12 November 1962, vol. 33, no. 23, E. Rubinstein, Copyright 1962, McGraw-Hill Publishing Company, Inc., New York, New York)

Variable Reluctance. A reluctance-type transducer operates on the principle of electromagnetic induction, which produces an electric current (or voltage) by the movement of a conductor through a magnetic field. It consists of a sensing element such as a Bourdon tube or diaphragm, a magnetic core, an armature to complete the magnetic circuit, and two or four inductance coils. A typical two coil unit is shown in Figure 5.15.2.7b. The armature is fastened to the twisted Bourdon tube and rotates in proportion to the pressure. The rotation changes the air gap, causing a change in the circuit inductance. The inductance coils are represented in the bridge circuit (Figure 5.15.2.7c). Generally, the output is sufficiently high that amplification is unnecessary, but the output of the bridge may be doubled by adding an E-core opposite the one shown, creating a four-arm bridge circuit. Any inductance change can be used to modulate the amplitude of a carrier voltage, or change the oscillator frequency. A schematic diagram of the instrumentation system for a variable reluctance transducer is shown in Figure 5.15.2.7d.

5.15.2.8 VIBRATING ELEMENT (REFERENCE 156-2). The operation of a vibrating element transducer involves the use of a vibrating mechanical element whose resonant frequency is made to change with applied pressure.

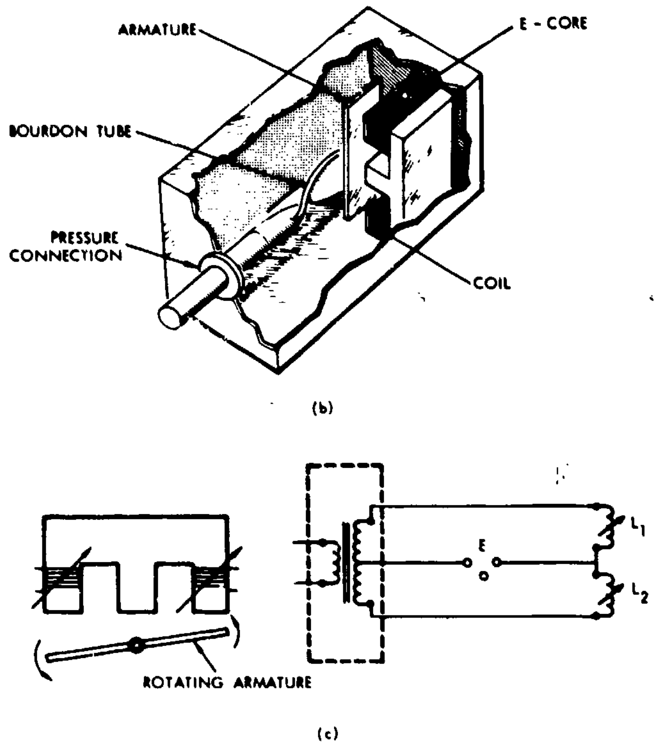


Figure 5.15.2.7b,c. Variable Reluctance Transducer and Electrical Circuit

(Reprinted with permission of "Product Engineering," (b) 12 November 1962, vol. 33, no. 23, E. Rubinstein; (c) January 1964, vol. 25, no. 1, J. Grey, Copyright 1962, 1954, McGraw-Hill Publishing Company, Inc., New York, New York)

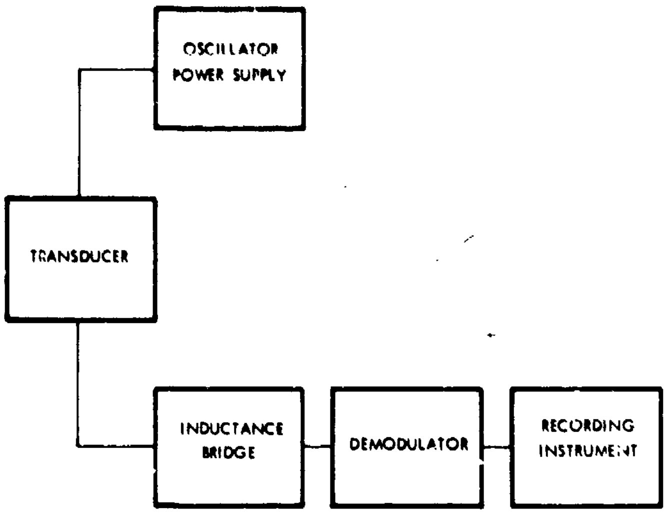


Figure 5.15.2.7d. Schematic Diagram for a Reluctance Transducer System

(Reprinted from "Product Engineering," January 1954, vol. 25, no. 1, J. Grey, Copyright 1954, McGraw-Hill Publishing Company, Inc., New York, New York)

One type of instrument employs a fine wire stretched between a fixed point and a pressure-sensitive metal diaphragm. The wire, which is in a permanent magnetic field, is set into vibration by an alternating current passing through it. The current is supplied by an amplifier connected in an oscillator circuit in such a manner that the frequency of oscillation is determined by the resonant frequency of the wire. A change in the applied pressure results in slight displacement of the diaphragm and consequently, a change in the tension of the wire. The resonant frequency of the wire is also changed, since it is a function of the tension, resulting in a change in the frequency of the amplifier output signal. The change in the applied pressure is determined by measuring the change in frequency of this output signal.

A second type of instrument utilizes a thin-walled metal cylinder maintained in forced vibration in elliptical mode by electromagnetic driving. Pressure to be measured is applied externally to the cylinder. Changes in the applied pressure produce changes in the hoop stresses which, in turn, result in changes in the resonant frequency.

5.15.2.9 RADIOMETER GAGE (KNUDSEN). The Knudsen gage is a vacuum instrument which consists of a light-weight vane and mirror supported on a torsion suspension mounted between two electrically heated surfaces. When gas molecules strike the higher temperature surface they rebound with an increase in momentum, which, when striking the vane, deflect the vane against its torsion suspension. The resulting deflection of the light beam from the mirror is read on a translucent scale, and the pressure is obtained from a calibration curve.

For proper operation, the mean free path of the gas molecules should be much greater than the gap between the vane and high temperature surface. Its major disadvantages are mechanical awkwardness and lack of remote measurement. Details can be found in Reference 166-3, page 157.

5.15.2.10 THERMAL CONDUCTIVITY GAGES. Thermal conductivity gages are vacuum instruments which respond to changes in the thermal conductivity of the residual gases in a vacuum system. The gages depend on the loss of heat from a surface by means of heat conduction through the gas at pressures where the mean free path of the gas molecules is comparable with the dimensions of the gage tube and sensing elements. Pressure is related to the measured temperature of a heated wire, the higher the pressure the greater the heat loss. Chief advantages are continuous reading, electrical output for recording, and total and absolute pressure measurements.

These gages consist of three types (Reference 74-37):

Thermocouple. In a thermocouple gage, the temperature of a heated wire is measured by a thermocouple, forming a thermoelectric junction with the wire. Several thermocouples grouped together are called a thermopile. The main distinction between the thermocouple and thermopile is the method of heating. In the thermocouple gage (Voege type) the thermocouple is in contact with the heated wire, while in the thermopile (Hastings type), the thermopile is heated directly, and includes a rate of change of temperature-compensating thermocouple in the pile.

Pirani. In the Pirani gage, a heated wire is installed in one leg of a Wheatstone bridge. Variation in temperature causes a change in wire resistance, affecting the balance of the bridge.

Thermistor. The thermistor gage uses a bead of semiconductor material in thermal contact with the heated wire. The temperature of the wire is determined by the resistance of the thermistor.

5.15.2.11 IONIZATION GAGES. Ionization gages are primarily high vacuum instruments which, as defined in Reference 74-36, "depend on the collection and measurement of ions formed by collisions between gas molecules and high vibrating electrons or other energetic particles." A few of the gages are described below.

Triode Gages. Triode gages are also called hot filament, hot cathode, or thermionic gages. In these gages, electrons from a thermionic cathode are accelerated through a more positive grid structure. If the accelerating voltage is greater than the potential of gas ionization, the electrons from the cathode will ionize gas molecules on collision. If each electron makes only one ionizing collision while traversing from cathode to plate, the number of positive ions formed will be proportional to the gas pressure. As electrode negative with respect to the cathode collects a constant fraction of the ions produced, resulting in an ion current proportional to the gas pressure.

The upper pressure limit for these gages is about 10^{-4} mm Hg. With higher pressures, the ion current becomes saturated, reaching a constant value independent of pressure. Furthermore, chemically active gases react with the hot filament.

Cold Cathode (Philips). As shown in Figure 5.15.2.12, the cold cathode gage consists of a ring-shaped anode located between two cathode plates. A magnetic field is impressed by a permanent magnet. A micrometer and 1-megohm resistance are connected in series with the tube across a 2000 volt DC source.

The cold cathode gage uses positive ions, accelerated by the cathode potential, to bombard a cold cathode of zirconium or thorium, producing secondary electrons. The electrons are deflected by a magnetic field so that they travel to the anode in helical paths. Since the total path length traveled by the electrons is long, the ionization produced by each electron is large. The total current is equal to the sum of the positive ion current to the cathode and is a measure of pressure. Unstable oscillations will commonly occur in the discharge, and will cause discontinuities as high as 10 percent from the calibration curve.

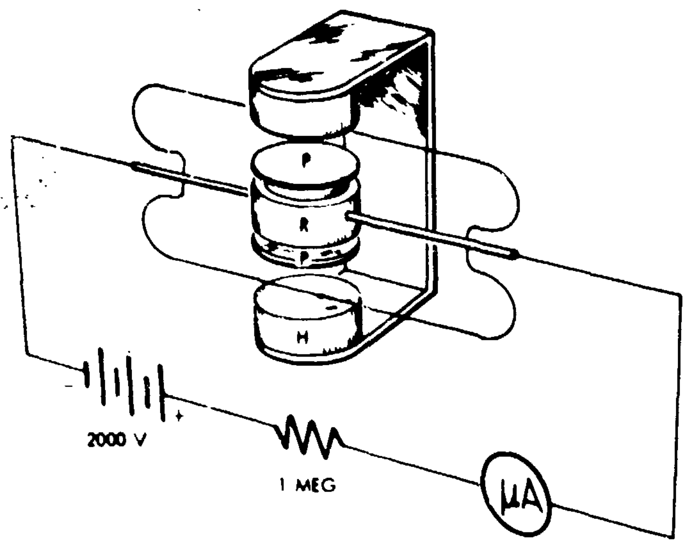


Figure 5.15.2.12. Philips Cold-Cathode Ionization Gage
(Reprinted with permission of "Instruments and Control Systems," March 1963, vol. 36, no. 3, J. M. Lafferty and T. A. Vanderslice, The Instruments Publishing Company, Pittsburgh, Pennsylvania)

Alphatron. In this gage, ionization is produced by alpha particles from radium sources. A DC potential is applied between an ionization chamber and the ion collectors. The ionization current produced is amplified and then read on a standard microammeter. The primary advantages of such a gage are no filament to burnout, and no possibility of chemical activity between the gas and cathode. Furthermore, it will measure higher pressures than either the triode gage or Philips gage.

Ultra-High Vacuum Gages. For pressure measurements below 10^{-8} mm Hg, very sensitive gages are required. All are ionization gages, many of which are alterations or improvements to the triode and cold cathode gages described previously. These gages are:

1) Bayard-Alpert gage
2) Cold cathode, inverted magnetron gage
3) Hot cathode, magnetron ionization gage
4) Partial pressure gage
5) Omegatron
6) Magnetic-deflection mass spectrometer

Reference 74-36 gives an explanation of operation.

5.15.3 Temperature Instrumentation

5.15.3.1 COMPARISON CHART FOR TEMPERATURE INSTRUMENTS. Table 5.15.3.1 presents typical selection data for several types of temperature sensors.

5.15.3.2 THERMOMETERS. Thermometers include a variety of devices discussed in the following paragraphs under liquid in glass, filled system, bimetallic, resistance, and thermistor.

Much of this Detailed Topic was reprinted from *Automation*, J.J. Combes, vol 7, no. 5, May 1960, Copyright 1960, Penton Publishing Company, Cleveland Ohio.

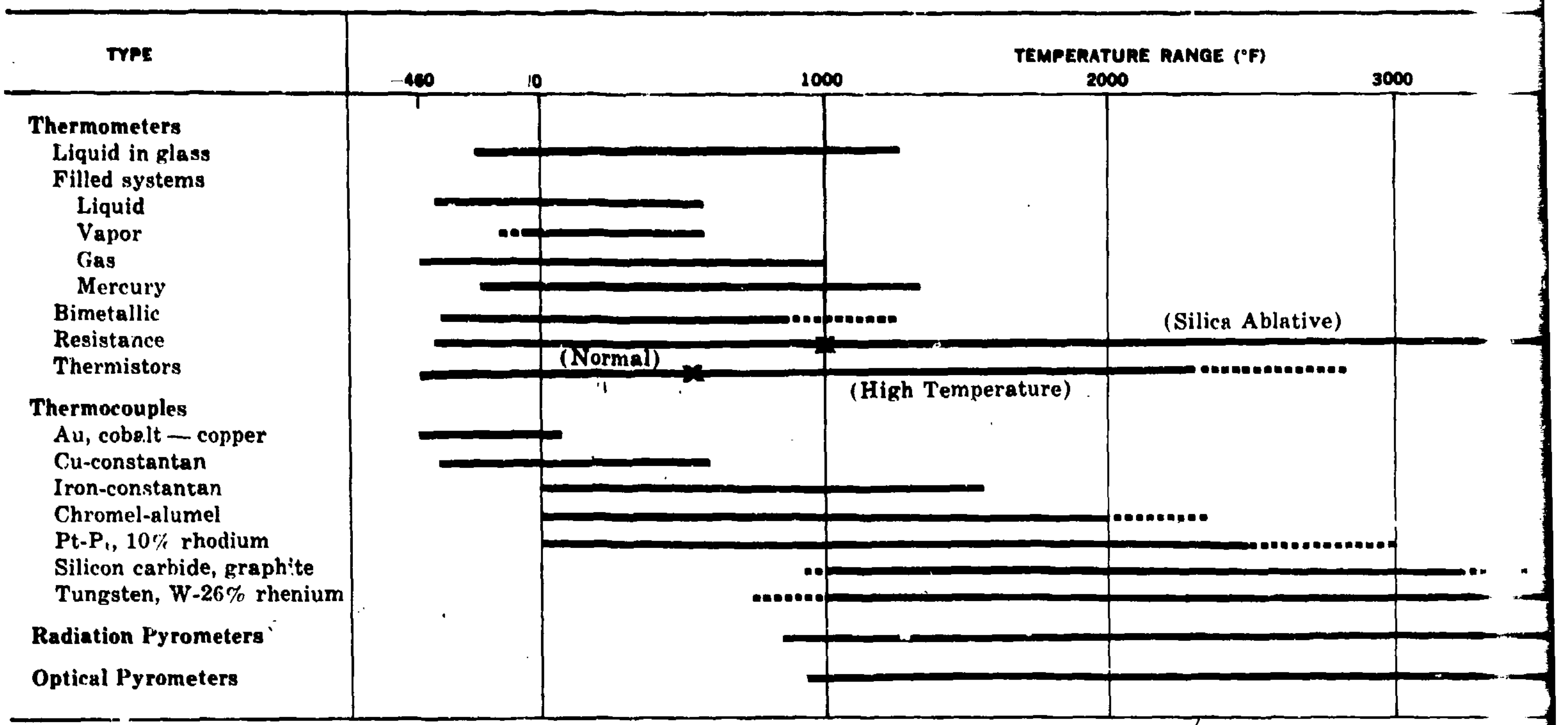
Liquid in Glass (Reference 10-9). This type of thermometer measures temperature as a function of the change in volume of a liquid as it is subjected to a change in temperature. It is made of a relatively thick-walled, long glass tube, with a hollow tube on one end. The bulb communicates with a blind capillary bore in the center of the tube. The bulb, the volume of which is large compared with that of the capillary, is usually filled with mercury, mercury-thallium, gallium, alcohol, toluol, pentane, or silicones.

As the bulb is subjected to a change in temperature, its liquid also changes temperature. This changes the volume of the liquid, causing it to rise or fall in the capillary bore of the tube. The height of the liquid in the bore, compared to a calibrated scale on or next to the tube, is then an indicator of the measured temperature.

Approximate limits of operation are from −300 to +1200 F. This type of temperature gage is the most widely used in industry, in laboratories, and by the general public.

Filled System (Reference 10-9). A filled system (Figure 5.15.3.2a) consists of a bulb subjected to the measured temperature, a flexible connecting capillary, and an elastic element; all are filled with a material responsive to changes in temperature. There are two types of these temperature elements:

Table 5.15.3.1. Compar... on



━━ = Normal range
▪▪▪▪ = Extended range
F = Fair
G = Good
H = High
L = Low
M = Medium
N = None, No
S = Sometimes
SL = Slow
V.G. = Very Good
Y = Yes

rison Chart For Temperature Instruments

4000	ACCURACY (°F)	CALIBRATION SENSITIVITY (STABILITY)	OUTPUT POWER	REQUIRED OVERRANGE PROTECTION	RESPONSE	REMOTE RECORDING	AUXILIARY EQUIPMENT REQUIRED	AFFECTED BY CONTAMINATION
	0.04 to 0.5	V.G.	N	N	H	N	N	N
	1.5 to 9	G	N	Y	SL	Y	S	N
	1.5 to 6	G	N	Y	SL	Y	S	N
	1.5 to 6	G	N	Y	M	Y	S	N
	1.5 to 6	G	N	Y	SL	Y	S	N
	1 to 10	F	N	N	SL	N	N	Y
5500F	0.05 to 0.1	V.G.	H	Y	H	Y	Y	N
	0.2 to 2.0	F	M	Y	H	Y	S	N
		G	L	S	H	Y	Y	S
	2 to 15	G	L	S	H	Y	Y	S
	2 to 15	G	L	S	H	Y	Y	S
	2 to 15	G	L	S	H	Y	Y	S
	2 to 15	G	L	S	H	Y	Y	S
	25 to 150	G	L	S	H	Y	Y	S
6000F	25 to 150	G	M	S	H	Y	Y	S
7000F	10 to 25	F	N	N	H	N	N	N
7600F	7 to 25	F	N	N	H	N	N	N

B

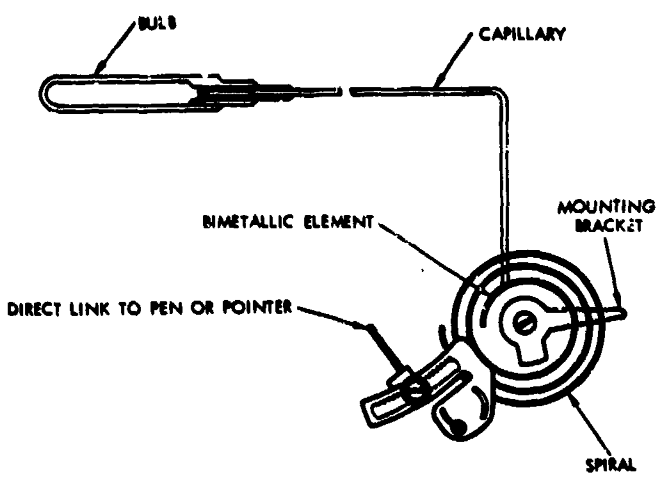


Figure 5.15.3.2a. A Filled System Temperature Transducer
(Reprinted with permission of "Automation," May 1960, vol. 7, no. 5, J. J. Combes, Copyright 1960, Penton Publishing Company, Cleveland, Ohio. Figure Courtesy of Honeywell, Inc., Philadelphia, Pennsylvania)

1) *Volume Responsive.* This type is completely liquid-filled. The liquid expands with temperature rise, and the net volume change is taken up by the elastic element. The accompanying pressure change is secondary.

2) *Pressure Responsive.* This type of filled system is either partially filled with a volatile liquid or is completely gas filled. Pressure in the system increases with temperature rise. Pressure change expands the elastic element; the volume change is secondary.

Each type of filled system has been separated into two classes by the Scientific Apparatus Makers Association. Volume response devices include liquid-filled (Class I) and mercury-filled (Class V) types. Pressure-responsive devices are either vapor-filled (Class IIA, IIB, and IIC) or gas-filled (Class III).

Liquid Filled — Class I. Range of operation for Class I bulb thermometers is —300 to +500°F. A minimum span of 25 to 50°F for a given instrument is limited by bulb size. A maximum span of 250°F is limited by the non-linear expansion characteristics of the filling liquid. The speed of response is relatively slow because of low thermal conductivity and system heat capacity.

Mercury Filled — Class V. Range of operation is from —38 to +1200°F. The minimum span is 40°F. The maximum span is limited only by the range; the speed of response is relatively slow.

Vapor Filled — Class II. Range of operation is —300 to +600°F. Instruments in this class are divided into three subclasses as follows:

1) IIA is used when the bulb temperature will always be higher than the ambient temperature. A large bulb contains vapor and volatile liquid; the capillary and elastic element are filled with volatile liquid.

2) IIB is used when the bulb temperature will always be lower than the ambient temperature. A small bulb contains volatile liquid and vapor, and the capillary and elastic element contain vapor.

3) IIC is used when the bulb temperature might be either higher or lower than the ambient temperature. The filling material can be located as in either IIA or IIB. Change-over can require from several minutes to an hour.

Gas Filled — Class III. Range of operation is from —450 to +1000°F. Span is limited only to that required to insure reasonable linearity. This type requires a relatively large size bulb, but its speed of response is very fast.

Bimetallic (Reference 10-9). These devices measure temperature as a function of the differential change in size of two metals as they are subjected to a change in temperature. They are made of strips of two or more metals laminated into a multi-layer piece. As the laminated metal strip is subjected to a change in temperature, its individual laminations expand or contract in different amounts, causing the strip to bend or change curvature. The bimetallic element of an industrial thermometer is usually fabricated in the form of a helical coil, with one end fixed and the other end free to turn as a result of differential expansion. The turntable end operates a pointer which rotates over a calibrated dial. This thermometer has a similar, but not quite as broad, field of use as the liquid in glass thermometer. Its approximate limits of operation are from —300 to +1200°F (but not over 800°F for sustained operation). Its accuracy is about 1 percent of scale range and its response is reasonably good.

Resistance. A resistance thermometer (Figure 5.15.3.2b) measures temperature as a function of the change in electrical resistance of a wire as it is subjected to a change in temperature. This type is the most accurate of the three remote reading temperature measuring systems. The resistance thermometer is the standard for the International Temperature Scale in the range of —297 to +1167°F.

The resistance thermometer is made by winding a length of insulated wire (usually nickel, copper, or platinum) around a high-conductivity solid metal core. In use, the tip of the core is kept in contact with the bottom of its well so that heat is transferred rapidly to the resistance winding for high speed of response. As the temperature of the winding increases, its resistance increases. The resistance is continuously and automatically measured by a modified Wheatstone Bridge-type instrument. Output signals from the instrument operate a pen or pointer on a calibrated chart or scale to indicate the measured temperature. Resistance thermometers are normally used in the measurement range of —325 to +1000°F. (Reference 10-9).

Extremely high temperatures — up to the melting point of the material in the probe — can also be measured with resistance elements, but the probe is destroyed. An example

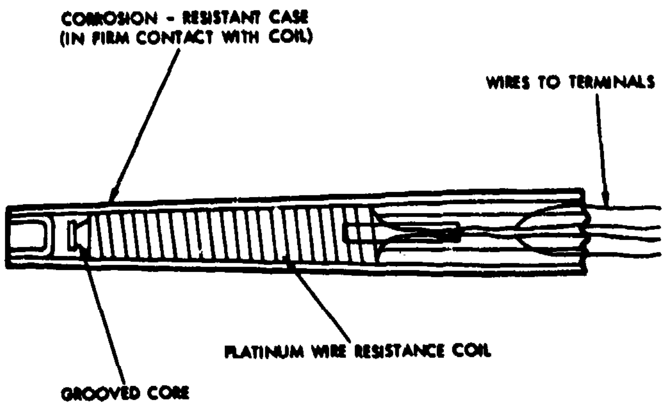


Figure 5.15.3.2b. A Resistance Thermometer
(Reprinted with permission of "Automation," May 1960, vol. 7, no. 5, J. J. Combes, Copyright 1960, Penton Publishing Company, Cleveland, Ohio. Figure Courtesy of Leeds and Northrop Company, Philadelphia, Pennsylvania)

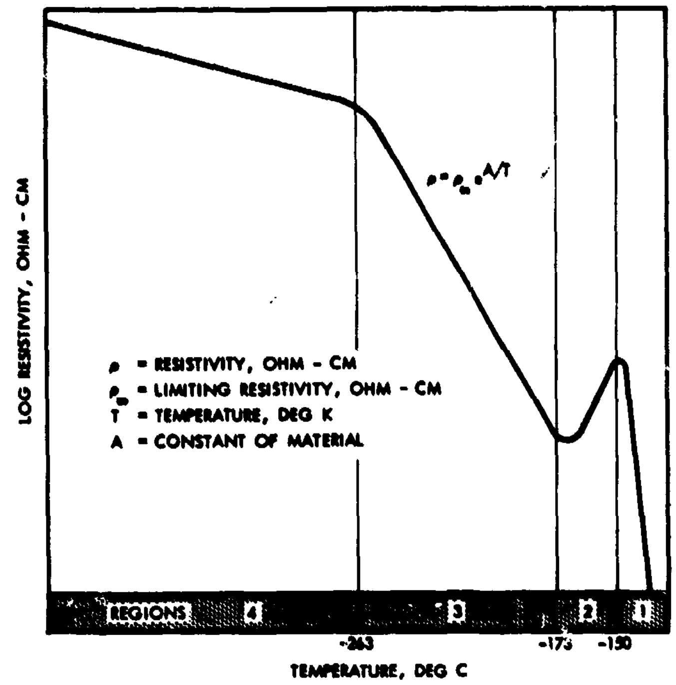


Figure 5.15.3.2c. Variation of Resistivity with Temperature for a Germanium Sample
(Reprinted from "Control Engineering," September 1961, vol. 8, no. 9, A. Nussbaum, Copyright 1961, McGraw-Hill Publishing Company, New York, New York)

is measuring the surface temperature of ablating silica in a missile nose cone undergoing test. A piece of tungsten wire imbedded in the silica increases in resistance gradually with temperature until the instant when the surface ablates back as far as the wire. The wire now burns through instantly and the resistance jumps. The last reading before the jump indicates temperature. Values over 5500°F have been recorded (Reference 19-71).

Thermistor. A thermistor is a semiconductor that changes electrical resistance with changes in temperature. Most thermistors have negative temperature coefficients (NTC), designated as N-type thermistors. Typical materials are germanium and metal oxides of cobalt, nickel, manganese, iron, and zinc. Thermistors are made from powder and are available in bead, rod, disc, and washer forms.

Germanium thermistors with appropriate choices of impurities (doping with phosphorus, arsenic, antimony) have good performance throughout the cryogenic region. With an N-type, resistance increases greatly as the temperature decreases, reducing the lead wire resistance errors. The response and sensitivity is non-linear throughout its temperature range. The reader is referred to References 27-30 and 74-24 for a discussion of the qualitative physical characteristics of germanium. A resistivity-temperature plot for a typical germanium sample is depicted in Figure 5.15.3.2c. Output voltage is over one hundred times more sensitive than the thermocouple and thus is extremely sensitive to small temperature changes.

5.15.3.3 THERMOCOUPLES. A thermocouple measures temperature as a function of the voltage generated at the junction of wires of two dissimilar metals in close contact (Reference 10-9). Two junctions made are the sensing junction and the reference junction. The higher the temperature at the sensing junction, the larger the generated voltage (emf). A cold reference junction is normally kept at constant temperature, usually at 32°F (0°C). If the cold junction temperature can change, a correction must be made by adding to the observed emf that emf which the thermocouple would develop if the reference junction were at 32°F and if the sensing junction were at the actual temperature of the reference junction. The simplest two-metal thermocouple is shown in Figure 5.15.3.3a.

Thermocouples may be either sheathed or bare wire (several types are shown in Figure 5.15.3.3b). The higher the temperature environment the more critical the sheathing requirements become.

The five most commonly used thermocouples are:

Gold, cobalt — copper	(−450 to 80°F)
Copper — constantan	(−300 to 600°F)
Iron — constantan	(0 to 1550°F)
Chromel — alumel	(0 to 2000°F)
Platinum — platinum, 10 percent rhodium	(0 to 2500°F)

Higher temperature thermocouples have become available using refractory, metal couples and non-metal couples such as silicon carbide and graphite. Some have been calibrated to 4000° Fahrenheit. Measurements to 9000°F have been

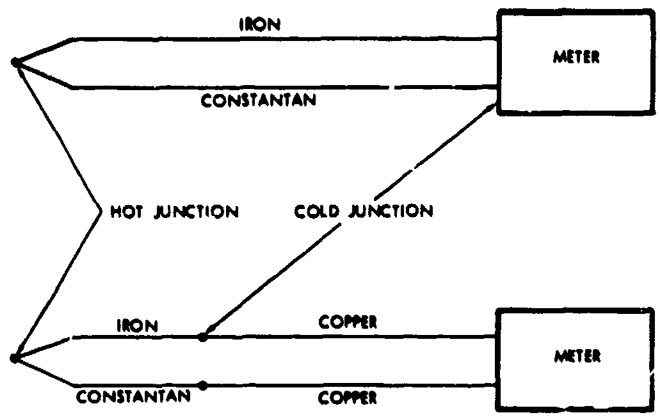


Figure 5.15.3.3a. Elementary Thermocouple With and Without Extension Leads

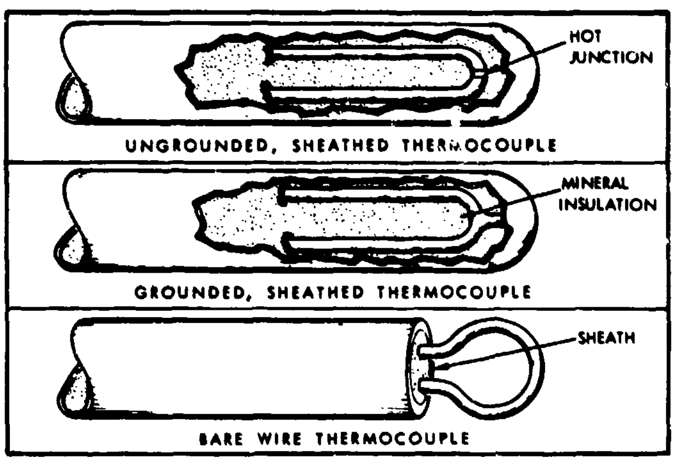


Figure 5.15.3.3b. Types of Thermocouples
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accomplished by alternately exposing high temperature thermocouples to the hot environment and a water cooled jacket. At elevated temperature three problems exist which affect thermocouple selection. These problems are (1) corrosive nature of environment, (2) high temperature electrical insulation for wires and (3) sheathing material with high endurance. A survey of thermocouples for temperatures over 1000°C is presented in Reference 72-2. Emf outputs for various thermocouple types are shown in Figure 5.15.3.3c. Response time is denoted by the thermocouple characteristic time, usually designated as τ. The characteristic time is the time required for a thermocouple to complete 63.2 percent of its response to a step change in gas temperature and may range from a few milliseconds to several seconds. The characteristic time is an empirical parameter and is affected by non-uniform conditions. These conditions are listed in order of severity: conduction, weld-bead size, junction shape, radiation, and thermocouple orientation. A discussion of these conditions and their effects is presented in Reference 52-13.

5.15.3.4 PYROMETERS. Two types of pyrometers are discussed, radiation and optical.

Radiation Pyrometers.* Radiation pyrometers measure temperature as a function of the intensity of the radiation given off by a hot body. A radiation pyrometer system (Figure 5.15.3.4a) includes an optical system which focuses radiant energy from the source upon a blackened receiving disc. Attached to the disc are the measuring junctions of a thermopile (multiple thermocouples connected in series). The lead wires of the thermopile are connected to a temperature indicator or recorder similar to those used with thermocouples. The temperature of the disc rises in proportion to the intensity of the radiation focused upon it, until the rate of its heat loss equals its rate of heat absorption from the source. Since the reference junctions of the thermopile are at the ambient temperature of the instrument housing, the difference in temperature between the hot and cold (measuring and reference) junctions produces a millivoltage output which is read on an indicator or recorder in terms of temperature of the source. Radiation pyrometers provide greater temperature sensitivity at the high ends of their ranges, mainly due to the fact that the intensity of hot body radiation is proportional to the fourth power of the absolute temperature body.

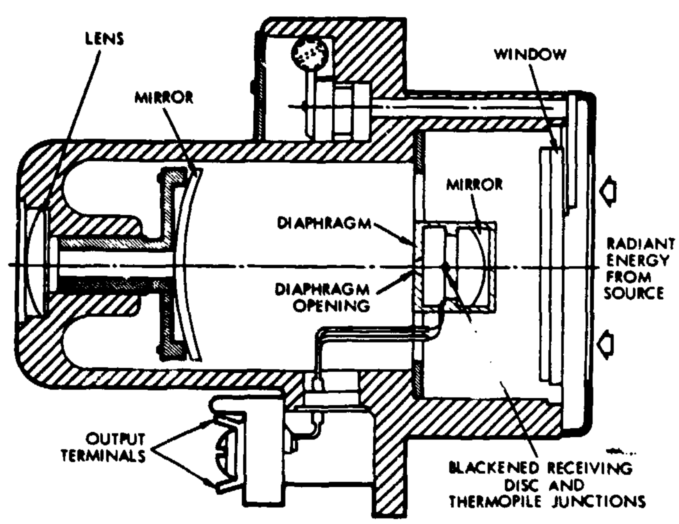

Figure 5.15.3.4a. A Radiation Pyrometer
(Reprinted with permission of "Automation," May 1960, vol. 7, no. 5, J. J. Combes, Copyright 1960, Penton Publishing Company, Cleveland 13, Ohio. Figure Courtesy of Leeds and Northrop Company, Philadelphia, Pennsylvania)

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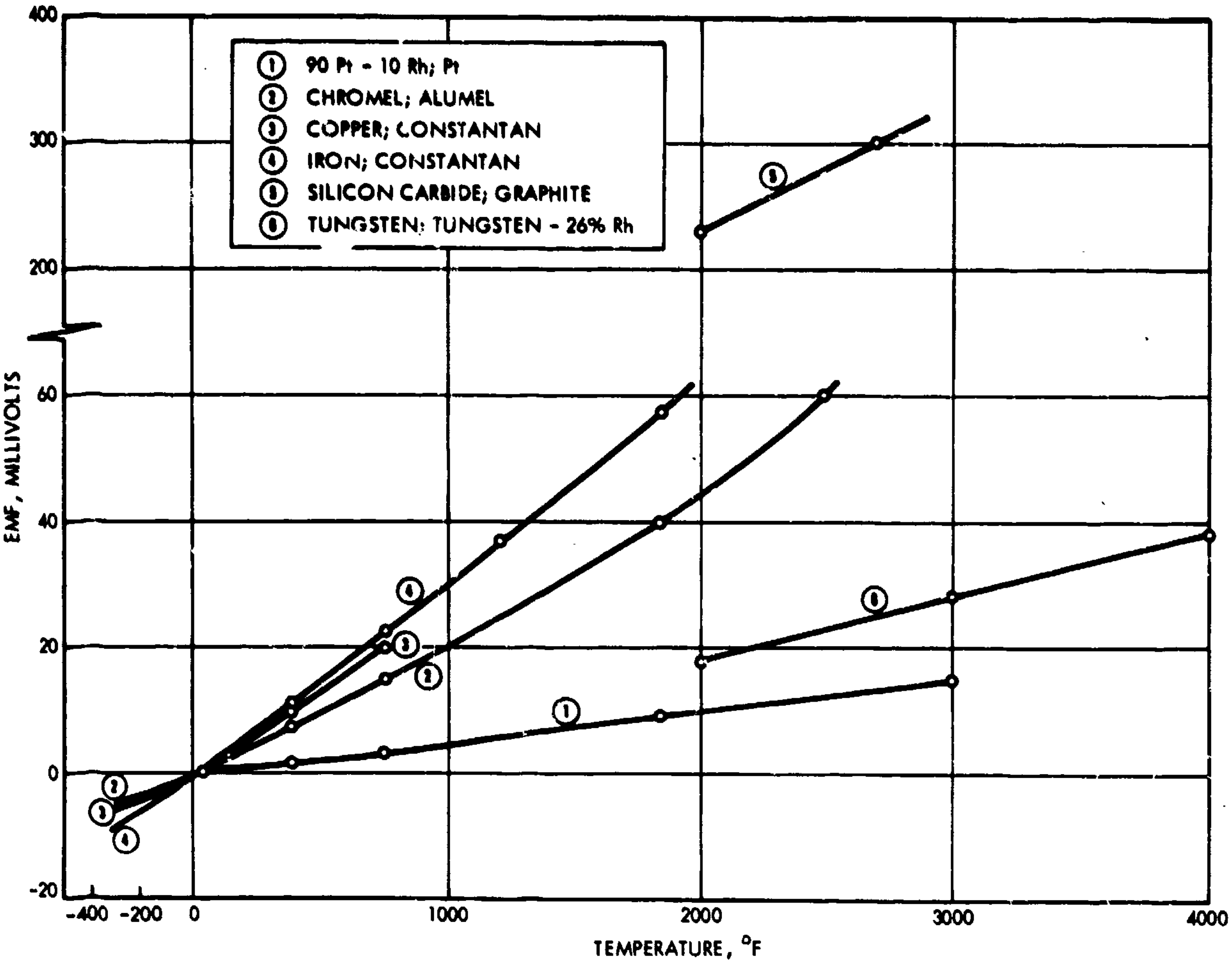

Figure 5.15.3.3c. Variation of Emf with Temperature for Several Thermocouples
(References 410-1 and 19-71)

Since the amount of radiation received is dependent upon the distance of the instrument from the hot body and the emissivity of the hot body, the instrument is calibrated in the field by setting it to the temperature indicated by a thermocouple or optical pyrometer at one point on its scale. When set for one point, the measurements will be correct over the entire instrument range.

The instrument reading is lowered by absorption of radiation by smoke and furnace gases in its line of sight. These gases may be invisible but possess absorption bands within the wave length range of the instrument. Therefore, when being used to measure temperature within a combustion or reaction chamber, the device is often focused upon the inside of a protecting tube which extends into the combustion chamber. The tube reaches the gas temperature but excludes the interfering smoke and gases. This expedient, however, imposes an additional time lag when fluctuating temperatures are being measured.

To improve accuracy and reproducibility for the measurement, and because ambient temperatures can vary considerably in the area of application of these instruments, automatic compensation is usually provided. This is accomplished by connecting a thermal conducting member, of the correct thermal coefficient and size, from the measuring junction of the reference junction. The thermal conductor, of course, is insulated from the electrical circuit. By this means, the heat flow pattern from measuring to reference junctions can be set to maintain constant output and, therefore, constant temperature indication despite varying housing temperature. Because no external reference junction compensation is required, no special extension lead wires are required and ordinary insulated copper wires may be used.

The range of operation of radiation pyrometers is from 800 to 7000°F; they may have spans from 700 to 2000°F. Changes in the sighting lens opening, radiation focus of the instrument, location of the instrument (distance to the target), and changes in the electrical circuit are the means by which the span for a given instrument may be shifted over the range and widened or narrowed.

Improvements to the basic radiation pyrometer have been made to compensate for varying solid surface emissivities and gas emission characteristics. One method is to filter out optically all but the desired wave lengths, leaving the instrument to read only one or two wave lengths at a time.

A one-color radiation pyrometer reads the varying intensity of one wave length with temperature, but is affected by changes in emissivity. A two-color pyrometer reads two wave lengths and compares the ratio of intensities, which cancels out changes in emissivity. If the emissivity for each wave length has a unique pattern of change, then a two-color pyrometer is little better than the one-color type.

Additional information can be found in References 387-1 and 72-2.

Optical Pyrometers.* Optical pyrometers measure temperature as a function of the brightness of the body whose temperature is being measured. Instruments of this type include a telescope and a potentiometer (Figure 5.15.3.4b). The telescope contains a filament which is made to glow by regulating the flow of current through it when it is focused upon the hot, glowing object whose temperature is being measured. The filament in the telescope view field appears to be superimposed upon the hot object. In measuring, the filament current is adjusted until its brightness exactly matches that of the hot object. Then a fine measurement is made of the filament current by balancing a galvanometer. This same adjustment positions a temperature scale to the temperature of the filament which, by being at the same brightness, is also at the same temperature as the hot object. This is true, of course, only if the hot object is emitting black body radiation. This would be the case if the object were completely surrounded by other objects all at the same temperature, as on the inside of a furnace which has reached equilibrium. Certain objects that are nonreflecting such as carbon, oxidized iron, steel, and nickel surfaces, are considered to be emitting black body radiation even when viewed in the open. When an optical pyrometer is being used to measure the temperature of reflecting, shiny objcts, it must be equipped with compensating screens to make it direct-reading for the emissivity of the objects.

The disappearing-filament method of temperature measurement, used in the optical pyrometer, is the method which is used for defining the International Temperature Scale between fixed calibrated points above the melting point of gold (1945°F).

The range of operation of optical pyrometers is from 1400 to 7600 F. For temperatures from 1400 to 2250 F the filament and object are compared directly. For higher temperatures, an absorption screen is placed between the filament and the object. This reduces the apparent brightness of the object so that it can be observed by the eye, and is within the normal range of the filament brightness. With a screen in position, the instrument is direct-reading on a higher scale.

Optical pyrometers are calibrated in the manufacturer's factory. High grade instruments are calibrated in accordance with National Bureau of Standards requirements as follows: low range $\pm 7°F$, high range $\pm 14°F$, extra high range $\pm 25°F$.

The limit of error for a particular measurement cannot be stated because measurements are subject to sources of errors such as object emissivity, operator experience, smoke and fumes in the sighting path, and dirt in the optical system. Experienced, competent operators and good maintenance can minimize these causes of error.

More information on pyrometry can be found in Reference 72-2.

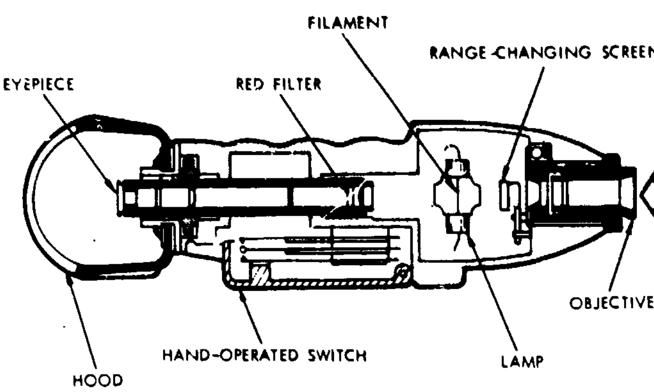

Figure 5.15.3.4b. An Optical Pyrometer
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5.15.3.5 DISCRETE DEVICES. Discrete temperature sensors are devices which indicate a specific temperature or temperatures within a narrow band. Two discrete temperature devices are those operating on the principle of color change or state change.

Change of Color. Because the color of a hot body is a function of its temperature, color charts can be used for rough approximation of temperature. Special paints may also be used which change color on heating, are irreversible, and provide a permanent record after cooling.

Change of State. The temperature of a body may be determined from the melting points of crayons, paint, and pellets, applied to the surface before heat is applied. Crayons and paints can be obtained to record temperature in 50°F increments in the range from approximately 100 to 2000 F. Pellets range from approximately 100 to 2500°F.

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Table 5.15.4.1 Comparison Chart for Level Instruments

TYPE	ACCURACY, in.* (POINT SENSORS)	RESPONSE TIME, MILLISECONDS (TYPICAL)	ACTIVATED BY SURFACE DISTURBANCE	AFFECTED BY CONTAMINATION	SYSTEM WEIGHT	REMOTE READING	FLUID DENSITY CORRECTION	LOW TEMPERATURE FLUIDS	RUGGED	PRESSURIZED TANK	POINT SENSING	CONTINUOUS READING
Floats	0.06	125	Y	Y	H	S	N	N	Y	S	Y	Y
Hydrostatic	—	50	N	N	L	S	Y	S	S	S	N	Y
Thermoelectric	0.06	40	Y	Y	L	Y	N	Y	N	Y	Y	N
Capacitance gage	0.25	40	S	Y	H	Y	Y	Y	N	Y	Y	Y
Ultrasonic	0.003	25	Y	N	L	Y	N	Y	S	Y	Y	Y
Photoelectric	0.09	20	Y	Y	M	Y	N	Y	S	Y	Y	Y
Nuclear radiation	—	—	N	Y	H	Y	N	Y	Y	Y	N	Y

*Calibration conditions and quiescent liquid surface

H = high
L = low
M = medium
N = no
S = sometimes
Y = yes

5.15.4 Liquid Level

5.15.4.1 COMPARISON CHART FOR LEVEL INSTRUMENTS. Table 5.15.4.1 presents a summary of liquid level sensor data which can serve as a guide in the selection of instruments to satisfy a particular application.

5.15.4.2 FLOAT GAGES. One of the most elementary liquid level gages is the float gage, consisting of a buoyant member, mechanically or magnetically linked to a calibrated indicating system. As the liquid level varies in height, the buoyant float changes position. Float gages are generally used for non-cryogenic liquids. At near cryogenic temperatures, probable ice buildup on the float will reduce accuracy. Furthermore, low density cryogenics such as hydrogen and helium offer little buoyancy.

Float gages should not be used in airborne systems because of the errors that develop due to the deviations in the liquid surface attitude. For ground use, float gages are reliable, easy to maintain, and accurate.

Float gages may be either continuous-reading or point-sensing. Several types of float gages are discussed below. The first four are continuous reading floats, differing primarily in the linkage between float and indicator.

1) Rod and bearing linkage. A rod connected to the float is fastened to rotary shaft, oriented perpendicularly through a bearing to the pointer. These gages are used in tanks with pressure above the liquid at atmospheric pressure.

2) Gages with stuffing boxes. For tanks under pressure the rotary shaft of the gage described in (1) operates through a packing gland, called a stuffing box.

3) Chain or tape. The float is connected to a flexible chain or tape which is held taut over a rotating pulley-pointer combination by a counterweight.

4) Magnetic float. The float is a doughnut-shaped magnet, guided vertically by a non-magnetic dip pipe. Inside the dip pipe is a similar magnet suspended from a level mechanism connected to an indicator. As the liquid level changes, the inner magnet by magnetic attraction, changes height with the outer magnetic float.

5) Flotation system. The system consists of a series of hermetically sealed and magnetically actuated contacts which are spaced vertically along the tank. A magnetic float actuates the contact as the liquid level rises or falls.

More information on float gages can be found in Reference 410-1.

5.15.4.3 HYDROSTATIC LEVEL MEASUREMENT. Hydrostatic level can be measured either by measuring a single pressure or differential pressure.

Static Pressure. Liquid height may be measured by a conventional pressure gage, recording pressure at the minimum level of the tank. The height of liquid above the minimum tank level is

$$h_l = \frac{P}{w} \qquad \text{(Eq 5.15.4.3a)}$$

where h_l = liquid height, ft

P = static pressure, lb_f/ft^2

w = liquid specific weight, lb_f/ft^3

Differential Pressure. A simple U-tube mercury manometer can also measure liquid level in an open tank by connecting one leg to a minimum tank level tap. The liquid height above the minimum tank level is

$$h_l = \frac{h_m S_m}{S_l} \qquad \text{(Eq 5.15.4.3b)}$$

where h_l = liquid height, ft

h_m = mercury height, ft

$S_{l,m}$ = specific gravities of liquid and mercury, respectively

5.15.4.4 THERMOELECTRIC LEVEL GAGES. Thermoelectric gages are used as point sensors for common liquids and cryogenics. In these gages, current heats the gage and the difference in heat transfer rates between the liquid and gas environments indicates the point of liquid contact. For liquid hydrogen special safeguards must be provided, since the currrent must be increased over that used with other cryogenics because of the high rate of heat conductivity and heat transfer. In some cases, circuit pulsing is used to limit the heat input (Reference 20-6). In a tank, several thermoelectric transducers are clamped at pre-selected stations to a Stillwell tube. The exact liquid level cannot be determined except at the instant when a given station is uncovered.

Thermistor. The thermistor or semiconductor gage with a negative temperature coefficient (resistance increase with temperature decrease), is well suited as a level gage for cryogenics. An electrical input heats the thermistor to a temperature of approximately 10° above its environment. The change in environment from the cold gas or vapor to the cryogenic liquid produces a large electrical signal. The heat input to the cryogenic liquid from the thermistor, and the formation of gas bubbles around the thermistor decrease the performance accuracy. A detailed discussion of thermistors as level gages is presented in Reference 108-1.

Hot Wire Probe. This gage uses a thin (1 mil) platinum wire, commonly spotwelded to suitable suspensions and leads, heated by AC or DC. When the wire is surrounded by vapor, the heat produced by the current is carried away slowly. When surrounded by a liquid, the heat is carried away so rapidly that the wire temperature approaches the liquid temperature (Reference 29-6). The resistance change in the platinum wire, caused by the temperature change, is detected by a bridge circuit.

5.15.4.5 CAPACITANCE GAGES. Capacitance gages can be either continuous-reading or point-sensing. Point-sensing requires continuous reading but a relay is installed between the gage and display system which is not actuated until the capacitance changes beyond a predetermined setpoint. The capacitance transducer consists of two conductors separated by a dielectric. The capacitance between two flat conductors or plates depends on three primary factors: (1) the dielectric constant of the medium between the plates, (2) the facing areas of the plates, and (3) distance between the plates. A change in any of these three factors will result in a change in capacitance (Reference 19-204).

Capacitance gages for measurement of liquid level are of two types—for liquids that are dielectric, and for liquids that are conductors. The two systems are shown in Figure 5.15.4.5.

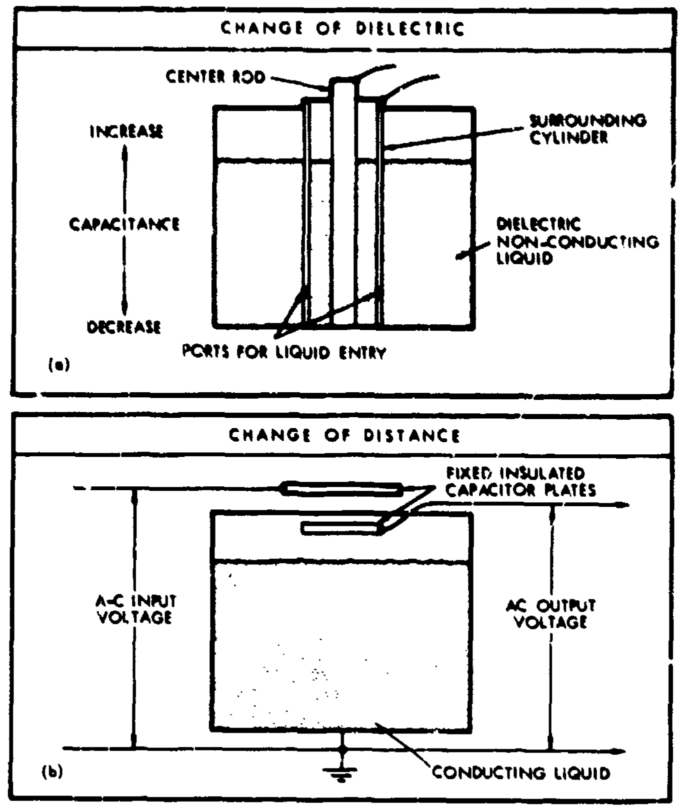

Figure 5.15.4.5. Capacitance Gages for Liquid Level Measurement: (a) Non-Conducting Liquids (b) Conducting Liquids

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The level of non-conductive liquids can be measured by a change in dielectric constant of a gage transducer. The dielectric liquid rises or falls between two concentric cylinders which act as plates. Since the dielectric constant of such liquid is substantially greater than air, a rise in the liquid level increases effective capacitance. In such applications the capacitance change with level variation is essentially linear. Principal errors encountered are produced by change of dielectric constant with temperature (Reference 19-204).

In cases where non-linear operation between capacitance and level is desired, tube profiling is used. One method is to distort the capacitance tubular plates to the desired pattern. This involves severe mechanical difficulties in manufacture. Another method is to deposit a thin, metallic film on the capacitance plate in such a manner as to conform to the non-linearity desired. More information concerning the implementation of this technique can be found in Reference 418-1, pages 216 to 220.

For conductive liquids such as mercury, electrolytes, or water, the change in capacitance between the liquid and a fixed plate above the liquid provides a variation in an AC output signal. The prime effect used here is a change in distance between two capacitor plates, one plate being the upper surface of the liquid. The output is non-linear with respect to liquid variation (Reference 19-204).

The AC excitation voltage, which may be low frequency, is applied between the upper plate and the liquid. The potential thus induces a charge and, consequently, a voltage on the lower fixed plate (Reference 19-204).

5.15.4.6 ULTRASONIC. One of the most familiar types of liquid height sensor is sonar, measuring the transit time for a pulse emitted from a sound wave transmitter to travel to gas-liquid interface and return to a receiver. Time is correlated to liquid height. Sonar requires large amounts of auxiliary equipment and is highly sensitive to the attitude of the surface level and tank.

Another ultrasonic method is composed of a series of miniature ultrasonic probes spaced along a Stillwell tube. The probe is a piezoelectric sensing element connected to a remote transistorized oscillator and subminiature relay. As a unit, it functions as a crystal oscillator. When liquid surrounds the probe tip, the high acoustic impedance of the liquid prevents circuit oscillation. When exposed to a vapor or gas, the acoustic impedance drops abruptly and the circuit starts to oscillate. The relay is actuated by the oscillator and energizes the display or control system. (Reference V-281).

5.15.4.7 PHOTOELECTRIC. In the photoelectric gage (Figure 5.15.4.7) a light beam is transmitted to a crystal. If the crystal is in a gas, the light is reflected to a photocell. If the crystal is submerged in a liquid, the light is deflected and is not observed by the photocell. The difference in signal from the photocell may be taken as the discrete point of liquid level (Reference V-18).

5.15.4.8 NUCLEAR RADIATION. A radiation gage consists of a radioactive source, such as a commercially available radium salt, and a detector. Two commonly used detectors are the geiger counter and the ionization cell. The intensity of radiation received by the detector varies in proportion to the thickness of material interposed and is in inverse proportion to the square of the distance between source and detector.

The source may be located in the liquid at the bottom of the tank. Varying liquid levels vary the intensity of radiation. The source may also be located on a guided float. Varying liquid levels change the distance between source and detector. Installations must be made such that there is no radiation hazard. The use of radium salt makes such gages costly.

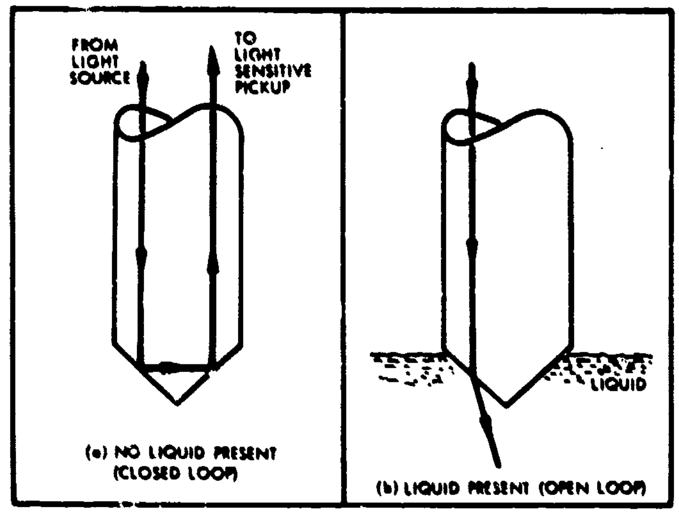

Figure 5.15.4.7. Photoelectric Liquid Level Gage
(Courtesy of Pioneer-Central Division, The Bendix Aviation Corporation, Davenport, Iowa)

5.15.5 Position Measurement Instrumentation

5.15.5.1 GENERAL. The transducers discussed in Sub-Topic 5.15.2 which measure pressure through member displacement, can be used to measure position. These include capacitors, potentiometers, variable inductance and reluctance, and strain gages. The first four can be used as either rotating or linear displacement measuring devices, while the strain gage is only a linear device. In pressure transducers, the relationship between displacement and resistance is usually linear; while for displacement devices, potentiometers can provide sine, hyperbolic, and logarithmic functions. The variable reluctance and strain gage are limited to small displacements. All have electrical outputs and only the potentiometer and variable inductance devices possess high outputs such that they do not require amplification.

A few transducers applicable to displacement measurement are discussed in the following Detailed Topics. A great many others are presented in Reference 19-205.

5.15.5.2 DISCRETE POINT SENSING. A typical discrete point sensor is a limit switch. The part being moved trips the switch, initiating a control system or display panel. Usually cams or tabs are attached to the moving part to contact the switch. In some cases, dimpled metal tape is used.

5.15.5.3 VARIABLE PITCH SPRING. The helical spring with a variable pitch (Figure 5.15.5.3), will change its resistance as it extends, and will act as a variable resistance element in an electrical circuit. Either AC or DC current can be used, but the resistance is small and does require amplification.

5.15.5.4 FIELD GRADIENT TRANSDUCER. This transducer, sometimes called the Hall device, uses a crystal in which a voltage is generated in proportion to the intensity of a magnetic field as the crystal moves in that field. The crystal is excited with DC current. The output voltage is approximately a few microvolts and requires large amplification. Displacement of 25×10^{-6} inches can be measured.

5.15.5.5 INTERFEROMETER. The interferometer compares an outgoing light beam with a beam of light reflected from a mirror attached to the object, and distance of which is to be measured. The phase shift is zero if the total distance is equal to an integral number of wave lengths (plus a constant). For intermediate distances, the instrument can interpolate very accurately. Output is optical and no amplification is required.

5.15.5.6 SYNCHRO. The synchro, an angular measurement device, is essentially a transformer with a rotating secondary winding (Figure 5.15.5.6). The primary winding is stationary and is energized with an AC current. As the secondary winding is rotated relative to the primary winding, a voltage is induced in the secondary winding as a sine function of the angle of rotation. Resolution is as high as 0.005° of arc and power output is high, eliminating the need for amplification.

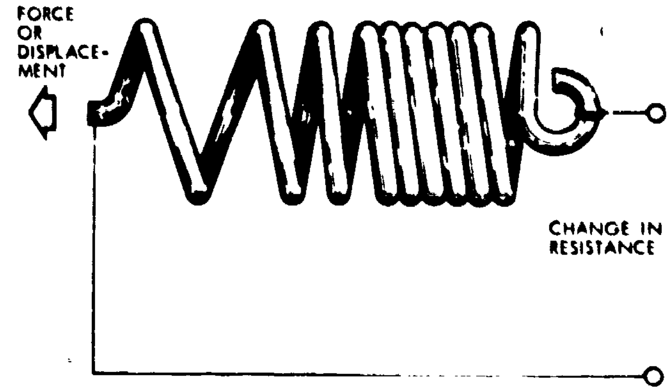

Figure 5.15.5.3. Variable Pitch Spring Linear Displacement Transducer

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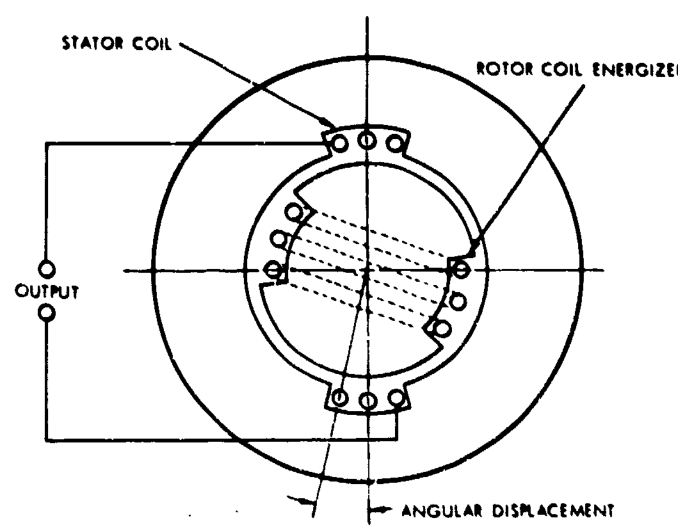

Figure 5.15.5.6. Synchro Rotary Transducer

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Pressure
1-14, 19-113, 19-122, 19-175, 74-16, 74-17, 74-25, 74-35, 74-36, 74-37, 111-24, 156-1, 156-2, 160-25, 160-31, 160-38, 166-3, 410-1, V-37

Temperature
10-9, 19-71, 19-202, 19-206, 19-208, 27-30, 47-12, 52-13, 72-2, 74-13, 74-24, 387-1, 410-1

Level
19-204, 20-6, 29-6, 47-6, 52-4, 108-1, 410-1, V-281, V-290

Position
19-204, 19-205

General
19-91, 19-207, 47-17, 349-1, 410-1, 418-1, 460-1

5.16 PRESSURE SWITCHES

5.16.1 Introduction

This Sub-Section discusses considerations important to the design and selection of pressure switches. Pressure switches are ON-OFF transducers which convert fluid pressure changes above or below a given set point into an electrical signal. Pressure switches are used in numerous aerospace fluid systems for pressure indication and pressure control applications. Switches used by ballistic missiles and space vehicles must exhibit high reliability under extreme temperatures and high vibration levels. Pressure switches have been generally regarded as poor in resistance to vibration, suffering from chatter, contact point arcing and fusing, and contact resistance changes at the 25 to 50g vibration levels produced by rocket engines. However, recent design improvements in pressure switches have alleviated many of these problems. Rocket system control applications utilizing pressure switches have been made using pressure switches in conjunction with shutoff valves to control pressure by functioning as either ON-OFF regulators or relief valves.

5.16.2 Pressure Switch Operation

A pressure switch is composed of three basic modules—a pressure sensing element, a reference load spring, and electrical contacts. The various modules in a pressure switch are illustrated in Figure 5.16.2, and are described in Sub-Topic 5.16.3. The reference load spring is connected to the pressure-sensing element and opposes any movement of the pressure-sensing element until tank or system pressure increases and overcomes the spring pre-load. The load spring is usually adjustable over a pressure range, providing flexibility of switch operation. Several sensing element designs possess sufficient inherent spring rate such that the requirements for a separate load are eliminated. The electrical switch is mechanically joined through the sensing element. As pressure increases to a value to overcome the spring pre-load, the sensing element deflects and actuates the switch contact assembly. When pressure decreases below that value required to overcome the spring force, the switch is deactivated and the contacts are reset. The pressure at which reset occurs is always less than the actuation pressure because of friction, hysteresis, and contact increment differential inherent in the pressure switch. The difference in pressure between actuation and reset is defined as the deadband.

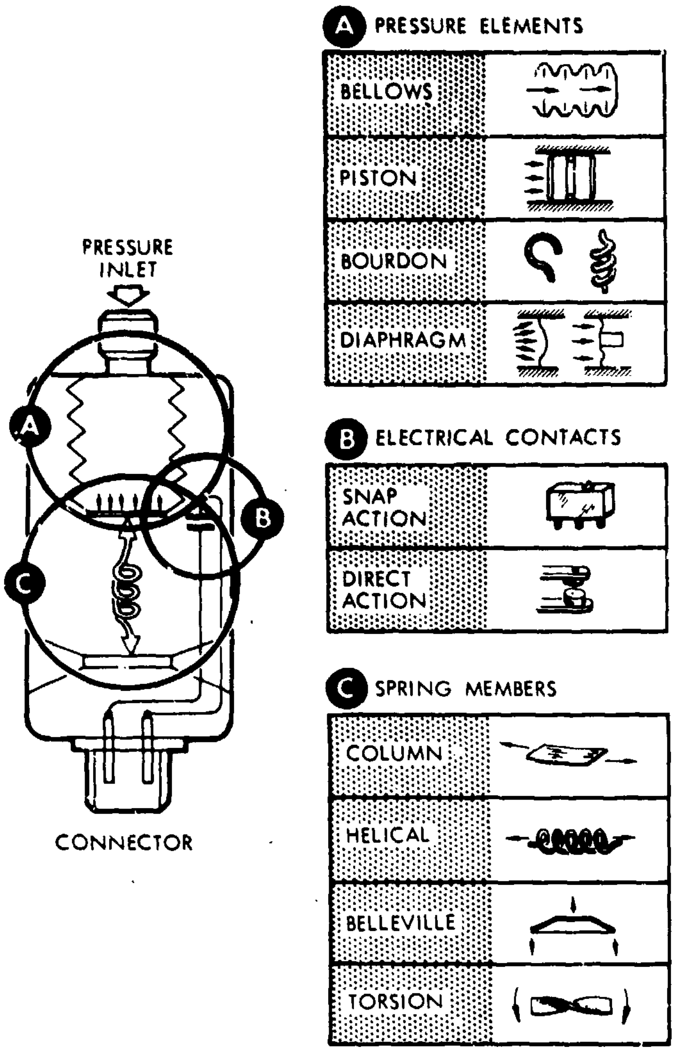

Figure 5.16.2. Pressure Switch Modules
(Reference 34-1)

The pressure switch is usually designed for snap (detent) action in the load spring and/or in the electrical switch.

Snap action gives stability under vibration, increased life cycle performance motion, amplification, and more energy available for switching (Reference 6-197).

5.16.3 Pressure Switch Modules

Pressure switch modules include sensing elements, loading springs, and electrical contacts.

5.16.3.1 **SENSING ELEMENTS.** Sensing elements for pressure switches may be categorized as Bourdon tubes, pistons, diaphragms, and bellows.

Bourdon Tube. The Bourdon tube is the most widely used sensing element for pressure switches where vibration, pressure oscillations, and temperature changes are moderate. A Bourdon tube pressure switch is illustrated in Figure 5.16.3.1a. The deflection of the Bourdon tube due to internal pressure is explained in Detailed Topic 5.15.2.3, and design considerations are presented in Reference 156-1. The Bourdon tube is a simple design possessing an inherent high positive spring rate quality and requiring no reference loading spring. It responds to pressure impulses of shorter duration than any other sensing element. Compensation, however, is ordinarily required to dampen system pressure oscillation such as pump ripple. The tube is sensitive to vibration and is subject to calibration drift with temperature change (Reference 1-5).

Any material that is sufficiently malleable to be formed into the desired shape can be used. Common materials are phosphors bronze and Ni Span C.

Piston. For some applications, a piston using a sliding seal such as an O-ring, provides a satisfactory pressure switch sensing element. A separate reference loading spring is provided in conjunction with the piston. Friction created by the seal depends upon piston alignment, lubrication, and force to seal. By increasing friction between the sliding surfaces, the seal can add a self-damping effect which will reduce pressure switch response to system oscillations, but will make the piston element impractical for low pressure use. The piston type pressure switch is used primarily for high pressure or high temperature system applications. A piston type pressure switch is illustrated in Figure 5.16.3.1b.

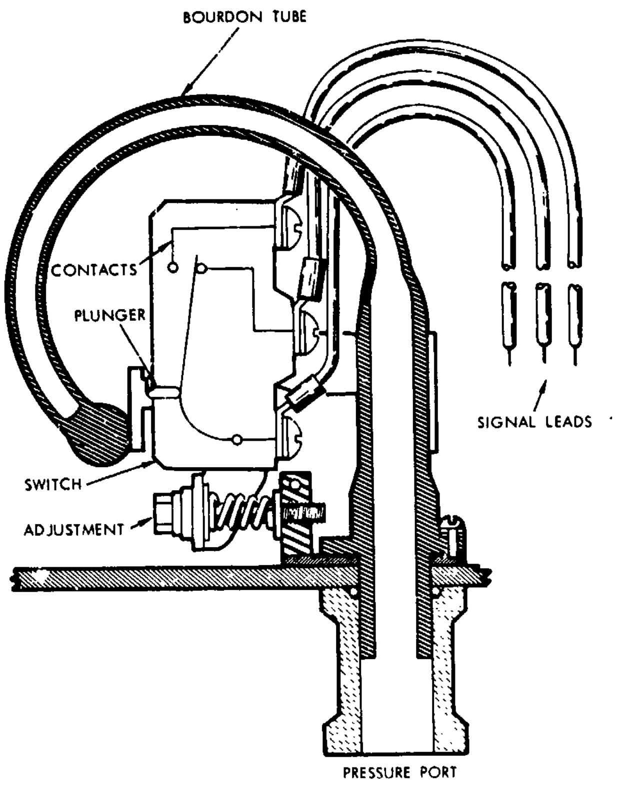

Figure 5.16.3.1a. Bourdon Tube Pressure Switch
(Reprinted from "Machine Design," August 1957, vol. 29, no. 17, W. B. Wallace, Copyright 1957, Penton Publishing Company, Cleveland, Ohio)

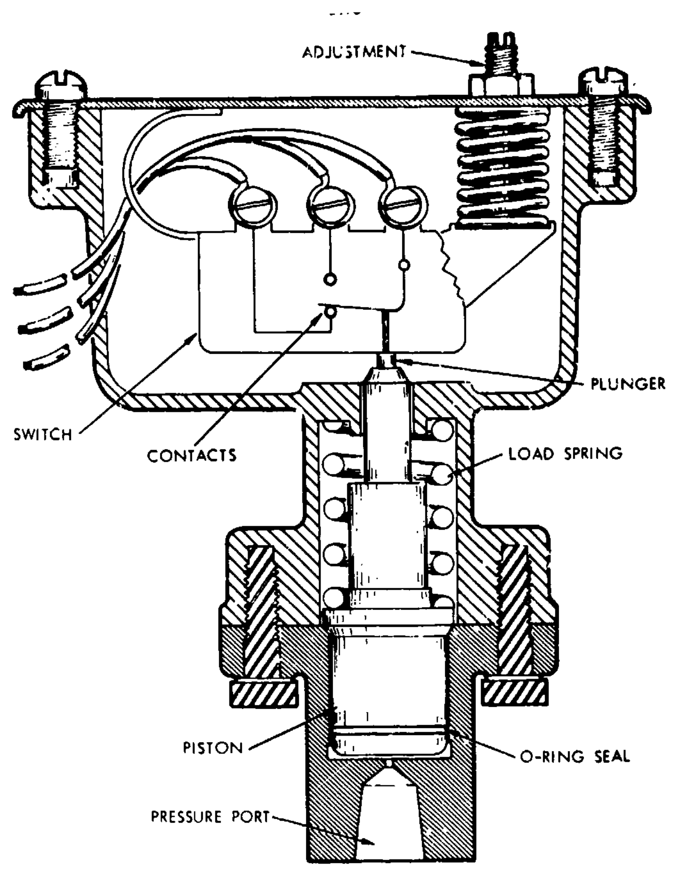

Figure 5.16.3.1b. Pressure Switch With a Piston-Sensing Element
(Reprinted from "Machine Design," August 1957, vol. 29, no. 17, W. B. Wallace, Copyright 1957, Penton Publishing Company, Cleveland, Ohio)

Diaphragm. Diaphragms are generally used with tank pressures ranging from 15 to 3000 psia. Thin diaphragms have low hysteresis, but as diaphragm thickness requirements increase with increasing pressure there is a corresponding increase in hysteresis lag.

Diaphragms can be either flat or convoluted and are made of both metals and non-metals. Metal diaphragms, in some cases, do not require a separate reference loading spring. They are, however, susceptible to cracking and spring rate change if the number of flexing cycles is excessive. Typical metals include stainless steel, aluminum alloys, and beryllium copper.

Non-metal diaphragms usually require a separate load spring. Typical non-metals include rubber, neoprene, and Teflon-filled glass fibers. Non-metallic diaphragms are quite variable in stiffness over wide temperature ranges, and are susceptible to deterioration due to the corrosivity of the operating fluid or to aging. Diaphragms are discussed in Sub-Section 6.7. Use of a diaphragm as a pressure switch sensing element is illustrated in Figure 5.16.3.1c.

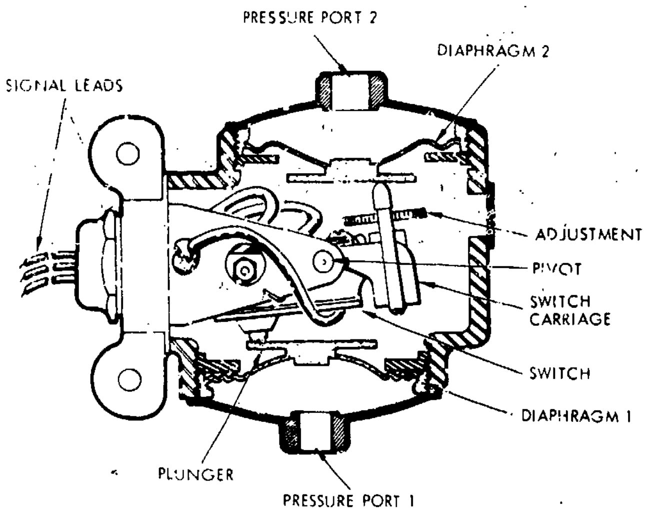

Figure 5.16.3.1c. Differential Pressure Switch With Two Diaphragm-Sensing Elements
(Reprinted from "Machine Design," August 1957, vol. 29, no. 17, W. B. Wallace. Copyright 1957, Penton Publishing Company, Cleveland, Ohio)

Bellows. Bellows are used as low pressure sensing elements in much the same sense as diaphragms. The bellows permits a longer stroke and larger sensing area than the diaphragm, but is more susceptible to vibration. It possesses a spring rate but it is usually augmented by a separate loading spring. Figure 5.16.3.1d shows a pressure switch with a bellows sensing element.

5.16.3.2 **LOADING SPRINGS.** Springs for pressure switches can be classified as linear springs (helical, torsion, leaf, and beams) and non-linear springs (Belleville, buckling column, and mechanical toggles). These designs can be used individually or in combination. Differing load-

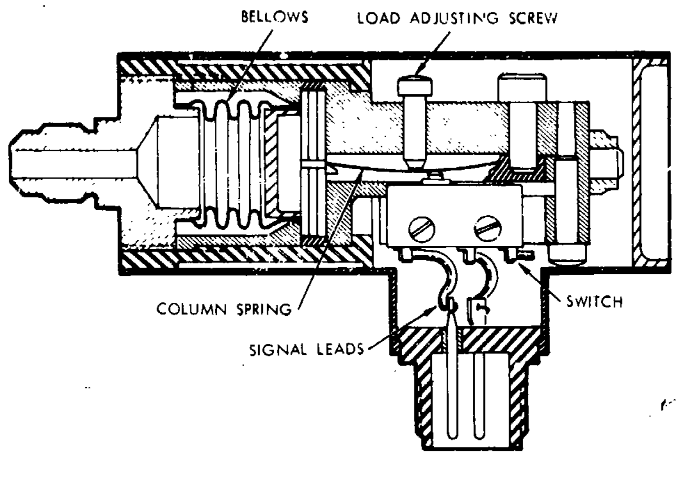

Figure 5.16.3.1d. Bellows Deflecting A Column Spring
(Reference 34-1)

deflection characteristics of the two spring classifications are illustrated in Figure 5.16.3.2a. Snap action is achieved by the change from positive to negative spring rates in the Belleville and buckling column springs. Figure 5.16.3.1d illustrates the use of a column spring-loading element. Actuation of the electrical switch is achieved when the spring is loaded to the buckling point.

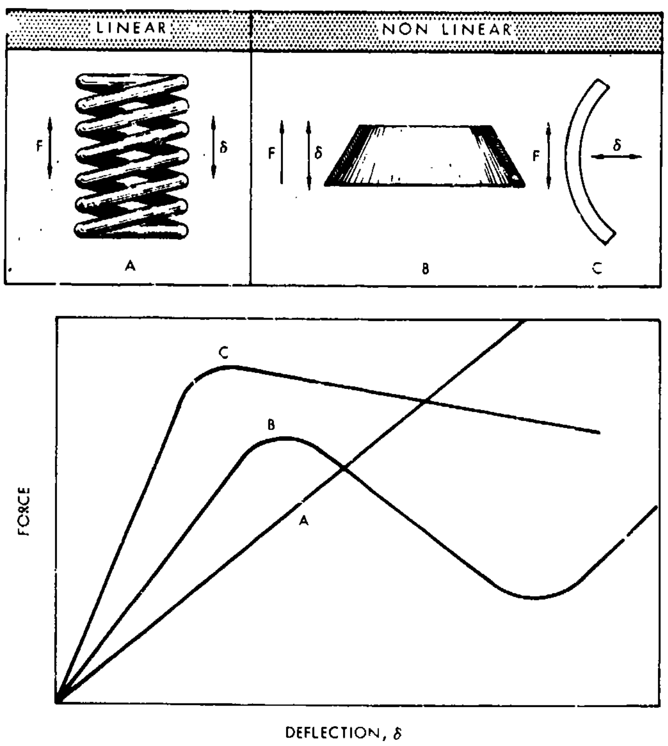

Figure 5.16.3.2a. Spring Load-Deflection Characteristics

A pressure switch utilizing Belleville springs is illustrated in Figure 5.16.3.2b. A series of Belleville springs in parallel are used as the loading element for the pressure switch. The Belleville springs are initially preloaded to the upper portion of the negative slope (See Figure 5.16.3.2c). When the diaphragm applied pressure force reaches the pre-load point, the spring is unstable and snaps to a stop near the lower part of the negative slope, actuating the switch. Deadband is established by the relative position of the stops. The Belleville spring has the advantages over a helical spring of requiring no additional pressure for further spring displacement at the actuation pressure, lower dimensional height (giving smaller calibration change with temperature variations), smaller size, lower weight, and reduced vibration sensitivity.

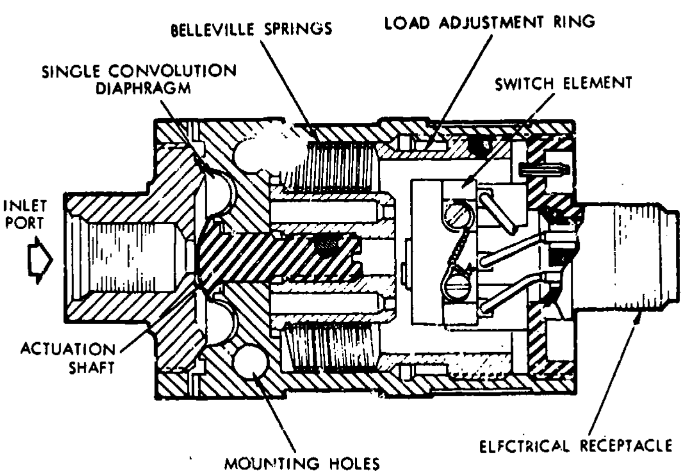

Figure 5.16.3.2b. Pressure Switch Using Belleville Spring Loading
(Reference 21-4)

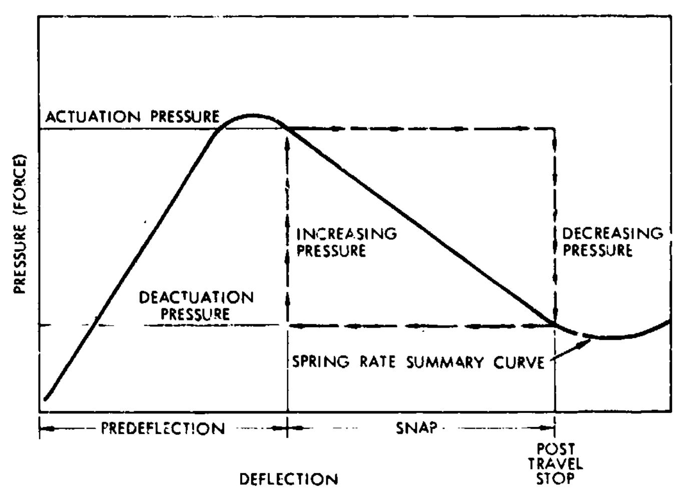

Figure 5.16.3.2c. Snap Action of a Belleville Spring

5.16.3.3 ELECTRICAL CONTACTS. Electrical contacts can be either positive action or snap action.

Positive Action Contacts. Positive action contacts move as a linear function of pressure-sensing element movement. These contacts are simpler and less expensive than snap action contacts, but are highly susceptible to contact arcing, particularly when the sensing element travel is slow. For rapid pressure increases, such as with surges, arcing is minimized. Positive action switches are highly sensitive to vibration unless the spring loading system is in the negative rate range. Under these conditions, the switches do not rely on light spring forces to return to the normally closed position. Typically contact elements of this type are leaf spring and mercury switches.

Snap Action Contacts. Snap action contacts are held together by a spring that reverses its force direction at an overcenter position. A plunger in contact with the sensing element moves the spring to the overcenter position and, when the spring force reverses, contacts snap into a new position. Arcing is reduced between contacts because they move quickly from full open to full closed. Furthermore, the contacts cannot be alternately opened and closed, as with direct action, under fluid system pulsation. A snap action switch maintains contact pressure up to the point where switch transfer occurs. Switch chattering under vibration begins as soon as contact pressure becomes less than the g load exerted under vibration. With a positive spring rate system, snap action switches are inherently less sensitive to vibration than positive action switches. Subminiature switches are snap action switches commonly used as electrical contacts for pressure switch assemblies (Figure 5.16.3.3).

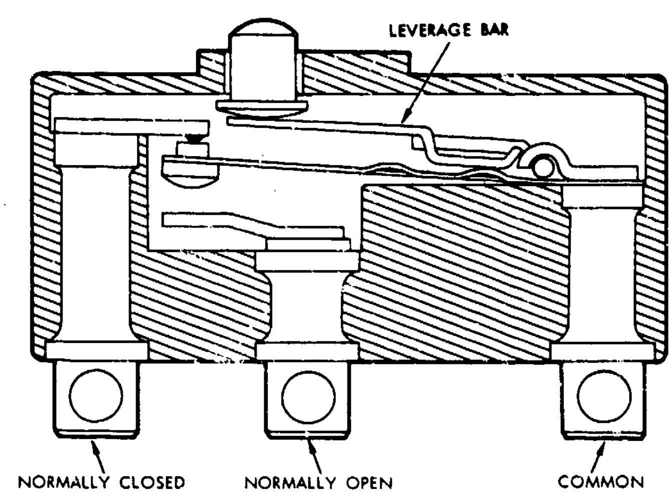

Figure 5.16.3.3. Snap Action Contact Assembly
(Reference 21-4)

5.16.4 Pressure Switch Types

Pressure switch types discussed include absolute, gage, differential, and calibration.

5.16.4.1 ABSOLUTE TYPE. The absolute pressure switch has the pressure port connected to one side of the pressure sensing element with the other side of the element exposed to a constant pressure which is either hermetically sealed in the switch or ported to it. Hermetically sealed switches are usually purged with a dry nitrogen-helium mixture to protect the switch from atmospheric moisture. Absolute pressure switches are unaffected by changes in ambient barometric pressure or by attitude variations, but compensation must be made for temperature variations. Typical absolute pressure switch designs are shown in Figure 5.16.3.1c and 5.16.3.1d.

5.16.4.2 GAGE TYPE. The most common pressure switch is the gage design in which system pressure is ported to one side of a pressure-sensing element and the other side of the element is ported to the atmosphere. The gage switch actuates when the difference between system pressure and atmospheric pressure reaches the set point of the switch. Gage-type pressure switches are used in flight systems where pressure is to be controlled near system or tank burst pressures, since pressure loading on a system is a function of the differential between system pressure and atmospheric pressure. A variation of the gage pressure switch which utilizes two mechanically connected diaphragms is shown in Figure 5.16.4.2, one diaphragm exposed to system pressure counteracted by the second diaphragm exposed to atmosphere. Dual pressure sensing elements are preferred in order to keep the switch and loading spring in an uncontaminated and inert atmosphere.

5.16.4.3 DIFFERENTIAL TYPE. Differential-type pressure switches may have either a single or double sensing element. In the single element differential pressure switch shown in Figure 5.16.3.1c, pressure from two locations is applied respectively on either side of the sensing diaphragm. In the design shown, port 1 is the high pressure port. The high pressure port must be connected to the higher of the two system pressure, and is used only when high pressure will always occur at the high pressure port.

If either pressure may become the higher system pressure, or if system fluid is detrimental to the switch assembly, a double sensing element pressure switch must be used. Figure 5.16.3.1c illustrates a double diaphragm pressure switch. Pressure applied to the pressure port 1 deflects diaphragm 1 against the switch plunger. An equal pressure applied to port 2 swings the contact assembly about the pivot point and moves the plunger away from the sensing element 2.

5.16.4.4 CALIBRATION TYPE. When a pressure switch has been installed in a system, its condition of operation cannot be determined unless the system is pressurized or unless the pressure switch is disconnected from the system and tested independently. A recent requirement for check-

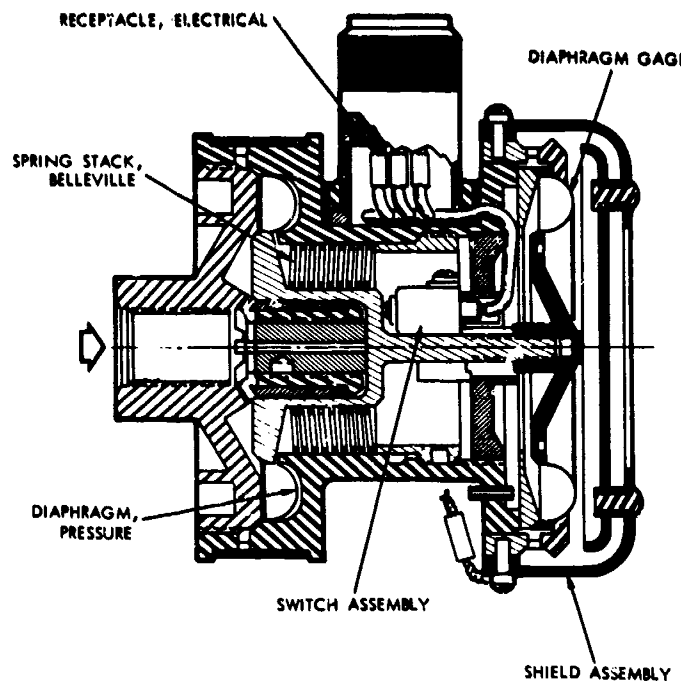

Figure 5.16.4.2. Low Pressure Switch
(Reference 21-1)

out of the pressure switch while it is installed in the system has resulted in a design incorporating a separate calibration stage. The additional calibration stage can be used with absolute, gage, or differential types. A pressure switch design with a calibration stage is shown in Figure 5.16.4.4. The switch can be actuated by pressure applied to either the calibration or system port. Operation of the switch from system pressure is independent of the calibration portion, since the system and calibration diaphragm are kept separate by a disjointed plunger Checkout can be achieved by pressurizing the calibration diaphragm which with adjoining pre-load moves the main plunger.

5.16.5 Selection Parameters

The following parameters should be considered when selecting a pressure switch.

5.16.5.1 OPERATING PRESSURE. The operating pressure is limited by the pressure limitations of the sensing element. Pressure switches are available for pressures from 30 inches of Hg vacuum to over 10,000 psi. Diaphragm and bellows elements are most satisfactory for low pressure with Bourdon tube and piston elements best suited for high pressure.

5.16.5.2 LIFE CYCLES EXPECTED. The cycle life of a pressure switch can be limited by either of its three major elements. The sensing element and loading springs are limited by the fatigue life of the material. Particularly

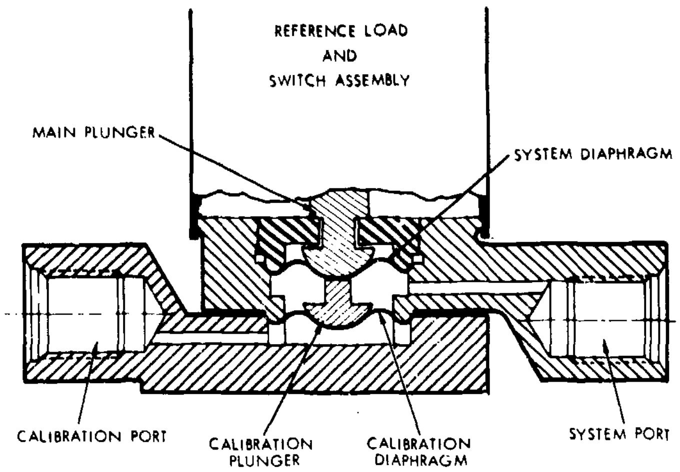

Figure 5.16.4.4. Pressure Switch With Calibration Stage
(Courtesy of Southwestern Industries, Inc., Los Angeles, California)

susceptible to fatigue failures are diaphragms, Bourdon tubes, and Belleville springs. Corrosive environments and large thermal gradients will lessen estimated life. Electrical contact assembly life is limited by wear and arcing. If the contacts are not properly sealed within the housing, contamination and corrosion will reduce the effectiveness of positive contact. Cycle lives for pressure switches commonly range from 10,000 to over one million cycles. Most missiles and space vehicles require cycle life of approximately 2500 to 5000 cycles.

5.16.5.3 RESPONSE. Response times for pressure switches are measured as the time lapse between reaching the actuation pressure and the triggering of an electrical response. Response times can range from less than 5 milliseconds for diaphragm and bellows type switches to over 1 second for piston type switches.

5.16.5.4 REPEATABILITY. Repeatability is defined as the ability of the pressure switch to actuate repeatedly at the desired set point. Diaphragm, bellows, and Bourdon tube pressure switches used at pressures of 30 psia and above are accurate to within $\pm \frac{1}{2}$ percent while sealed piston switches are accurate to nearly ± 2 percent for moderate environmental conditions. Under temperature and vibration extremes the accuracy tolerances for diaphragm, bellows, and Bourdon tube switches can increase to ± 1 or ± 2 percent. At pressures below 30 psia, repeatability is more nearly ± 1 percent with ± 2 percent for operation under temperature and vibration extremes.

5.16.5.5 VIBRATION AND SHOCK RESISTANCE. Vibration is a critical environmental factor and has been one of the primary disadvantages of pressure switches for incorporation on rocket systems. Vibration resistance has been greatly improved with the availability of pressure switches using snap action spring, or snap action contact assemblies. Properly designed negative rate spring system driving positive action switches are available with vibration resistance to 100g's and shock resistance to 120g's.

5.16.5.6 TEMPERATURE EFFECTS. Temperature changes will alter dimensions, increase friction, change spring elastic moduli, increase difficulty to seal, and may increase the deadband. Deadband variations are more likely with pistons or elastomeric diaphragms than with metal diaphragms. Although temperature compensation techniques can be used to maintain accuracy over wide temperature ranges, the accuracy for pressure switches which must operate from cyrogenic to normal ambient temperature is often reduced by a factor of two or more. However, with proper material selection and design, accuracies of ± 1 percent over a temperature range from -300 to $300°F$ are possible. The effect of extreme hot or cold temperatures will be reduced if a short length of small diameter tubing, sometimes called a "pigtail," is used to separate the switch from the environment (Reference 10-7). Pressure switches have been used successfully for monitoring rocket thrust chamber pressures with a combustion gas temperature over 5000°F, by having the pressure switch isolated by a length of tube.

5.16.5.7 LINE LENGTH. A long sensing line acts like a surge damper, decreasing switch sensitivity to short duration pressure ranges. If response to short peaks is desired, the sense line length should be kept to an absolute minimum (Reference 10-7).

5.16.5.8 ELECTRICAL RATING. For optimum accuracy and life, electrical loads should be kept to a minimum. When high loads are required, a relay may be used in conjunction with the switch. Some positive action switches are capable of carrying 10 amps resistive and 5 amps inductive at 30 v. D.C. and, in some cases, can eliminate the use of relays.

5.16.5.9 MEDIA. Media refers to either the working fluid or the surrounding atmosphere environment. If the working media or atmosphere is corrosive, the material of the sensing element and switch body must be compatible or protected by coatings or plating. In some cases it might be desirable to use an inert liquid between the corrosive fluid and switch. The liquid may be located in a chamber divided by a corrosion-proof diaphragm, or installed in a U-tube configuration filled with a buffer liquid such as mercury (Reference 10-7).

5.16.5.10 DEADBAND. Pressure switch deadband is defined as the difference between the actuator pressure and the deactuation pressure, usually expressed as a percentage of the actuation pressure. Pressure switches can be designed with deadbands ranging from 2 percent to 50 percent.

REFERENCES

1-5	19-75	34-1
6-197	21-1	34-2
10-7	21-4	156-1
10-13		V-269

5.17 FLOWMETERS

5.17.1 Introduction

The principles of operation, applications, and limitations of fluid flowmeters are discussed in this Sub-Section. Symbols used are tabulated, defined, and dimensionalized on a fold-out sheet at the end of the Sub-Section.

5.17.2 Classification of Flowmeters

Flowmeters are classified in five major groupings called mass, velocity, volumetric, gaseous leakage, and calibration flowmeters. Table 5.17.2 summarizes the flowmeter types discussed under each grouping.

5.17.3 Comparison Chart for Flowmeters

Table 5.17.3 is a flowmeter comparison chart which summarizes flowmeter data and is intended to serve as a guide for the selection of flowmeters for fluid system applications. The factors considered in preparation of the chart are listed below.

5.17.3.1 FLUID MEASURING CAPABILITY. The choice of fluids considered suitable for each flowmeter was based on flowmeter materials compatibility, contamination sensitivity, and the flowmeter principle of operation.

5.17.3.2 PERFORMANCE CHARACTERISTICS. The range of maximum flow rates (maximum flow rating of smallest and largest flowmeters), rangeability (ratio of maximum to minimum usable flow rate), and accuracy (percent of full scale) for each flowmeter type were obtained from supplier's literature and/or analysis of the flowmeter principle of operation. The response time (time required for the flowmeter to respond to a step change in fluid velocity and indicate the new velocity within 2 percent) for each flowmeter depends upon the secondary element (pickup, amplifier, and indicator) as well as the primary element (sensor), hence estimates of the response times were based on the fastest response times available for secondary elements. Supplier's literature and analysis of the flowmeter principle of operation were considered in assigning the typical values of response times for each flowmeter type.

5.17.3.3 ENVIRONMENTAL LIMITATIONS. The values assigned to the usable temperatures and maximum pressure were based on materials and design considerations for normal usage of the flowmeters, and may be exceeded at the expense of reduced accuracy and reduced reliability.

5.17.3.4 ADVANTAGES AND DISADVANTAGES. Factors considered were determined from analysis of the construction and principle of operation of each flowmeter and are summarized below.

Advantages
- Bi-directional flow
- Obstructionless
- Linear output reading
- Continuous reading
- No moving parts
- No dynamic seals
- No bearings
- Auxiliary power not required for primary element.

FLUID COMPONENTS

TABLE 5.17.3. FLOWM

FLOWMETER	FLUID MEASURING CAPABILITY: CORROSIVE	CYROGENIC	HYDROCARBON	HYDRAULIC	LIQUID METALS	GASES	MIXED PHASE	CONTAMINATED FLUIDS	SLURRIES	PERFORMANCE CHARACTERISTICS: RANGES OF MAXIMUM FLOW RATES: MASS-LIQUID POUNDS/MIN: MIN	MAX	VOLUMETRIC-LIQUID GAL/MIN: MIN	MAX	GASES (1) POUNDS/MIN (2) STD CC/MIN: MIN	MAX	FLOW RANGEABILITY MAX/MIN	ACCURACY RATING %F.S.	TYPICAL RESPONSE TIME MILLISEC	ENVIRONMEN: FLUID TEMP EXTREMES: LOW
TWIN TURBINE	X	X	X	X		X	X			370	100,000	45	12,000	(1) 10	15,000	10:1	0.5	10	CRYOGENIC
TURBINE-CONSTANT SPEED IMPELLER	X	X	X	X		X	X		X	40	100,000	4	12,000	(1) 10	15,000	10:1	0.5	1,000	CRYOGENIC
CONSTANT TORQUE IMPELLER		X	X	X		X					>1,000		>120		>150	10:1	0.5	1,000	CRYOGENIC
GYROSCOPIC		X	X	X		X	X	X	X	100	4,000			(1) 20	600	10:1	0.5	5,000	CRYOGENIC
CORIOLIS			X	X				X	X	<14	> 20					20:1	1	10	-65
MAGNUS			X	X				X	X	<10	>1,000					4:1	8	1,000	-65
THERMOPILE						X								(1) 0.004	80	20:1	3	5,000	-20
BOUNDARY LAYER	X	X	X	X	X	X		X	X	90	30,000			(1) 10	15,000	> 100:1	3	1,000	CRYOGENIC
VARIABLE HEAD FLOWMETERS WITH DENSITOMETERS	X		X	X						1.6	120,000	0.2	15,000			4:1	2	100	-65
VELOCITY METERS WITH DENSITOMETERS	X		X	X						1.6	120,000	0.2	15,000			10:1	1	100	-65
VARIABLE HEAD FLOWMETERS WITH VELOCITY COMPENSATION	X	X	X	X						1.6	120,000	0.2	15,000			4:1	2	100	CRYOGENIC
THIN PLATE ORIFICE	X	X	X	X	X	X	X	X	X			0.2	3,500	(2) 1.4×10^{6}	2.8×10^{9}	4:1	2	100	CRYOGENIC
FLOW NOZZLE	X	X	X	X	X	X	X	X	X			0.5	15,000	(2) 2.8×10^{6}	1.4×10^{10}	4:1	2	100	CRYOGENIC
VENTURI	X	X	X	X	X	X	X	X	X			0.5	15,000	(2) 2.8×10^{6}	1.4×10^{10}	4:1	2	100	CRYOGENIC
DALL FLOW TUBE		X	X	X		X	X					0.5	15,000	(2) 2.8×10^{6}	1.4×10^{10}	4:1	2	100	CRYOGENIC
TWIN THROAT VENTURI		X	X	X		X	X					0.5	15,000	(2) 2.8×10^{6}	1.4×10^{10}	4:1	2	100	CRYOGENIC
CAPILLARY	X	X	X	X	X	X						2×10^{-4}	>3	(2) 4×10^{-5}	> 0.15	20:1	2	100	CRYOGENIC
POROUS		X	X	X		X						2×10^{-4}	>3	(2) 5,000	10,000	20:1	2	100	CRYOGENIC
PITOT TUBE	X	X	X	X	X	X		X	X			0.5	15,000	(2) 2.8×10^{6}	1.4×10^{10}	4:1	2	100	CRYOGENIC
TAPERED TUBE AND FLOAT		X	X	X		X		X	X			2.6×10^{-5}	10,000	(2) 75	68,000	10:1	2	5,000	CRYOGENIC
SLOTTED CYLINDER AND PISTON		X	X	X								0.2	15,000			10:1	2	5,000	CRYOGENIC
TURBINE	X	X	X	X	X	X						0.1	15,000	(2) 2.8×10^{4}	4.5×10^{10}	15:1	0.5	10	CRYOGENIC
DEFLECTION OF SOUND BEAM	X		X	X								1500	3,000			15:1	2	10	-50
ACOUSTIC VELOCITY	X		X	X	X							0.1	800			20:1	2	10	-50
ELECTROMAGNETIC	X	X		X	X			X	X			0.002	$> 10^{5}$			20:1	1.5	3,000	CRYOGENIC
RADIOISOTOPE	X	X	X	X	X	X	X	X	X			<0.1	$> 10^{5}$	(2) > 0	4.5×10^{10}	> 20:1	1	D.N.A.	CRYOGENIC
DRAG FORCE	X		X	X	X			X	X			0.1	15,000			10:1	0.5	10	-65
NUTATING DISC		X	X	X								5	1,000			> 20:1	1	100	CRYOGENIC
ROTARY VANE			X	X								5	1,000			> 20:1	1	100	CRYOGENIC
LOBED IMPELLER						X								(2) 1.4×10^{6}	4.8×10^{7}	> 20:1	[illegible]	100	0
RECIPROCATING PISTON			X	X								<0.1	> 100			> 100:1	1	10	-65
AXIAL PISTON METERING PUMPS			X	X								<0.1	> 100			> 100:1	1	100	-65
MASS SPECTROMETER (GASES)						X								(2) 10^{-11}	> 1000	> 100:1	10	1,000	40
ULTRASONIC TRANSLATOR (GASES)						X								(2) 1	> 1000	> 100:1	> 50	100	40
WATER DISPLACEMENT (GASES)						X								(2) 4×10^{-7}	> 1000	> 100:1	1	D.N.A.	40
VARIABLE AREA (TRI-FLAT)			X	X		X						2.64×10^{-5}	5.55×10^{-2}	(2) 75	68,000	> 1000:1	2	5,000	0
STATIC WEIGHING CALIBRATOR	X	X	X	X	X			X	X	< 1	$> 10^{6}$	<0.1	$> 10^{5}$			> 1000:1	0.15	D.N.A	CRYOGENIC
DYNAMIC WEIGHING CALIBRATOR	X	X	X	X	X			X	X	< 1	$> 10^{6}$	<0.1	$> 10^{5}$			> 1000:1	0.25	D.N.A	CRYOGENIC
TIME VOLUME CALIBRATOR	X	X	X	X	X			X	X			< 0.1	$> 10^{5}$			> 1000:1	0.20	D.N.A.	CRYOGENIC
METER PROVER CALIBRATOR	X	X	X	X				X	X			< 100	$> 10^{5}$			> 100:1	0.10	D.N.A.	CRYOGENIC

5.17.3 FLOWMETER COMPARISON CHART

ENVIRONMENT LIMITATIONS			ADVANTAGES								DISADVANTAGES							
FLUID TEMP EXTREMES, °F		MAXIMUM PRESSURE PSIG	BI-DIRECTIONAL	OBSTRUCTIONLESS	LINEAR OUTPUT	ADAPTABLE FOR CONTINUOUS READING	NO MOVING PARTS	NO DYNAMIC SEALS	NO BEARINGS	NO AUXILIARY POWER REQ FOR PRIMARY ELEMENT	ACCURACY AFFECTED BY					ACCURATE CONSTANT SPEED MOTOR REQUIRED	CAREFUL CALIB REQUIRED	FLOWMETER
LOW	HIGH										MECHANICAL FRICTION	DENSITY CHANGE	VISCOSITY CHANGE	TEMP CHANGE	PRESSURE CHANGE			
CRYOGENIC	650	> 5,000			X	X		X		X	X						X	TWIN TURBINE
CRYOGENIC	300	> 5,000	X		X	X					X					X	X	TURBINE-CONSTANT SPEED IMPELLER
CRYOGENIC	300	> 5,000			X	X					X						X	CONSTANT TORQUE IMPELLER
CRYOGENIC	650	5,000	X	X	X	X					X					X	X	GYROSCOPIC
-65	650	5,000	X	X	X	X					X					X	X	CORIOLIS
-65	650	300	X		X	X										X	X	MAGNUS
-20	220	4,500				X	X	X	X								X	THERMOPILE
CRYOGENIC	1200	> 10,000	X	X		X	X	X	X				X				X	BOUNDARY LAYER
-65	[illegible]	> 5,000			X	X	X	X	X								X	VARIABLE HEAD FLOWMETERS WITH DENSITOMETERS
-65	300	> 5,000			X	X		X									X	VELOCITY METERS WITH DENSITOMETERS
CRYOGENIC	1000	> 5,000			X	X	X	X	X								X	VARIABLE HEAD FLOWMETERS WITH VELOCITY COMPENSATION
CRYOGENIC	2000	> 10,000				X	X	X	X	X		X	X	X	X			THIN PLATE ORIFICE
CRYOGENIC	2000	> 10,000				X	X	X	X	X		X	X	X	X			FLOW NOZZLE
CRYOGENIC	2000	> 10,000				X	X	X	X	X		X	X	X	X			VENTURI
CRYOGENIC	2000	> 10,000				X	X	X	X	X		X	X	X	X		X	DALL FLOW TUBE
CRYOGENIC	2000	> 10,000				X	X	X	X	X		X	X	X	X		X	TWIN THROAT VENTURI
CRYOGENIC	2000	5,000	X	X	X	X	X	X	X	X		X	X	X	X		X	CAPILLARY
CRYOGENIC	160	5,000	X		X	X	X	X	X	X		X	X	X	X		X	POROUS
CRYOGENIC	2000	100							X	X		X	X	X	X		X	PITOT TUBE
CRYOGENIC	1600	5,000			X	X		X	X	X		X	X	X	X		X	TAPERED TUBE AND FLOAT
CRYOGENIC	1600	100			X	X		X	X	X		X	X	X	X		X	SLOTTED CYLINDER AND PISTON
CRYOGENIC	1000	> 5,000			X	X		X		X							X	TURBINE
-50	750	300	X	X	X	X	X	X	X			X		X	X		X	DEFLECTION OF SOUND BEAM
-50	750	300	X	X	X	X	X	X	X			X		X	X		X	ACOUSTIC VELOCITY
CRYOGENIC	400	5,000	X	X	X	X	X	X	X								X	ELECTROMAGNETIC
CRYOGENIC	2000	> 5,000		X	X		X	X	X	X							X	RADIOISOTOPE
-65	300	> 5,000	X			X	X	X	X			X	X	X	X		X	DRAG FORCE
CRYOGENIC	400	5,000			X	X				X								NUTATING DISC
CRYOGENIC	400	5,000			X	X				X								ROTARY VANE
[illegible]	275	1,200			X	X				X								LOBED IMPELLER
-65	650	5,000	X		X	X			X	X								RECIPROCATING PISTON
-65	400	5,000			X	X				X								AXIAL PISTON METERING PUMPS
40	120	< 10		X	X	X	X	X	X								X	MASS SPECTROMETER (GASES)
40	120	< 10		X	X	X	X	X	X								X	ULTRASONIC TRANSLATOR (GASES)
40	120	< 10		X	X		X	X	X	X								WATER DISPLACEMENT (GASES)
[illegible]	160	10			X	X		X	X	X		X	X	X			X	VARIABLE AREA (TRI-FLAT)
CRYOGENIC	2000	< 400		X	X	X		X	X	X	X							STATIC WEIGHING CALIBRATOR
CRYOGENIC	2000	< 400		X	X	X		X	X	X	X							DYNAMIC WEIGHING CALIBRATOR
CRYOGENIC	2000	< 400		X	X	X		X	X	X				X				TIME VOLUME CALIBRATOR
CRYOGENIC	400	< 300	X	X	X	X			X	X				X				METER PROVER CALIBRATOR

B

Table 5.17.2. Fluid Flowmeter Classification

Mass Flowmeters

Momentum
- Twin turbine
- Turbine-constant speed impeller
- Constant torque impeller
- Gyroscopic
- Coriolis
- Magnus

Thermal
- Thermopile
- Boundary layer

Inferential
- Variable head flowmeters with densitometers
- Velocity meters with densitometers
- Variable head flowmeters with velocity compensation

Velocity Flowmeters

Variable head
- Thin-plate orifice
- Flow nozzle
- Venturi
- Dall flow tube
- Twin throat venturi
- Capillary
- Porous
- Pitot tube

Variable area
- Tapered tube and float
- Slotted cylinder and piston

Turbine

Ultrasonic
- Deflection of sound beam
- Acoustic velocity

Electromagnetic

Radioisotope

Drag force

Positive Displacement Volumetric Flowmeters

Nutating disc

Rotary vane

Lobed impeller

Reciprocating piston

Axial piston metering pump

Gaseous Leakage Flowmeters and Detectors

Mass spectrometer

Ultrasonic translator

Water displacement

Variable area

Flowmeter Calibration Systems

Static weighing calibrator

Dynamic weighing calibrator

Time volume calibrator

Meter prover calibrator

Disadvantages

Accuracy reduced by changes in mechanical friction, fluid density, fluid viscosity, fluid temperature, and fluid pressure.

Accurate constant speed motor required.

Careful calibration required; proportionality constants must be determined experimentally.

5.17.4 Mass Flowmeters

5.17.4.1 GENERAL. Mass flowmeters indicate flow rate in gravimetric units (lb/sec). This classification includes momentum, thermal, and inferential fluid flowmeters. *Momentum* types sense reaction forces caused by changing the direction of fluid flow. *Thermal* types sense heat flow rates which vary with fluid mass flow rates, and *inferential* types sense velocity and density separately and combine the two signals to result in a mass flow rate meter readout. Descriptions of various configurations of these three types are presented in the following paragraphs.

5.17.4.2 MOMENTUM. Six flowmeters will be discussed.

Twin Turbine Flowmeter. The sensing element in the twin turbine meter consists of two turbines having different fixed blade angles interconnected by a torsional member. Fluid flowing through the meter (Figure 5.17.4.2a) causes rotation of the element, and the two turbines tend to seek different speeds as a result of the different blade angles. The torsional member, which interconnects the two turbines, deflects angularly until the torsional forces caused by the turbines are balanced by the internal strain energy of the interconnecting torsional member. The angular displacement between the two turbines is directly proportional to the flow momentum, mV. Small magnets in the turbine blades generate signals as they pass by the coils on the meter body, and the phase difference of the signals generated at the turbine coils determines the angular displacement of the torsional member. The period of time, t, required for the phase angle to traverse a reference point (pickup coil) is directly proportional to the phase angle and inversely proportional to the angular velocity of the turbine. This time period, t, is directly proportional to the mass flow rate through the meter as explained below:

Mass flow rate

$$\dot{m} = K_0 V \rho \qquad \text{(Eq 5.17.4.2a)}$$

Momentum flow

$$p = K_1 V^2 \rho \qquad \text{(Eq 5.17.4.2b)}$$

Turbine phase angle

$$\phi = K_2 V^2 \rho \qquad \text{(Eq 5.17.4.2c)}$$

Turbine angular velocity

$$\omega = K_3 V \qquad \text{(Eq 5.17.4.2d)}$$

Time period for ϕ to pass a reference point

(Eq. 5.17.4.2e)

$$t = \frac{\phi}{\omega} = \frac{K_2 V^2 \rho}{K_3 V} = K_4 V \rho$$

Time period is proportional to mass flow rate

$$t = K_4 \dot{m} \quad \text{(Eq 5.17.4.2f)}$$

It should be noted that although Figure 5.17.4.2a illustrates an analog readout, this type of flowmeter is commonly instrumented to provide digital flow rate indication or mass flow totalization.

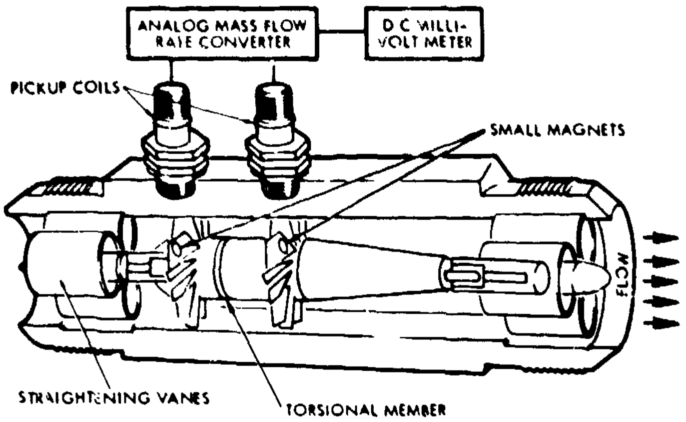

Figure 5.17.4.2a. Twin Turbine Mass Flowmeter
(Courtesy of Potter Aeronautical Corporation, Union, New Jersey)

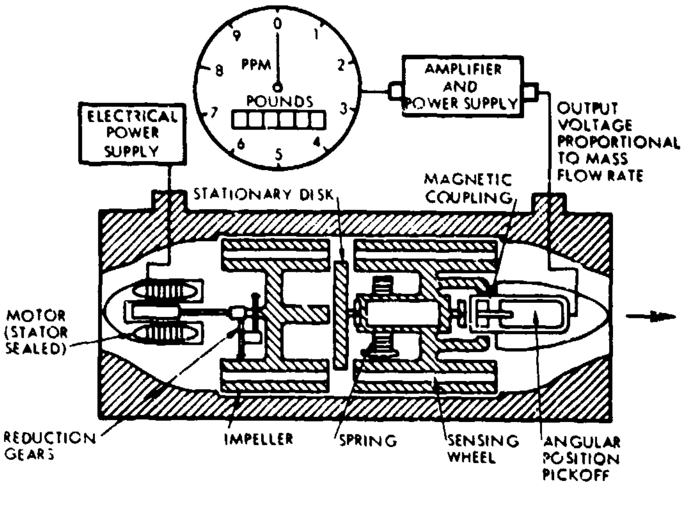

Figure 5.17.4.2b. Constant Speed Impeller Flowmeter
(Reprinted from "Product Engineering," 8 May 1961, vol. 32, no. 19, C. C. Miesse, Copyright 1961, McGraw-Hill Publishing Company, New York, New York)

Turbine-Constant Speed Impeller Flowmeter. The sensing element in the turbine-impeller meter consists of a turbine (sensing wheel) restrained by a torsional member (spring) as shown in Figure 5.17.4.2b. Upstream of the turbine is an impeller driven at a constant speed to impart a constant angular momentum to the flowing fluid. Recovery of this angular momentum by the turbine results in angular deflection of the torsional member which is directly proportional to the fluid mass flow rate. The angular deflection is detected by the angular position pickoff which produces a signal for direct flow measurement and integration. The theory applying to this principle of operation is presented below:

Restraining torque

$$L = \frac{I\omega}{t} \quad \text{(Eq 5.17.4.2g)}$$

Moment of inertia of element

$$dI = k^2 dM \quad \text{(Eq 5.17.4.2h)}$$

Rate of change of angular momentum of element

(Eq 5.17.4.2i)

$$\frac{dM_a}{dt} = \omega k^2 \frac{dM}{dt} = L = \omega k^2 \dot{m}$$

Restraining torque is proportional to mass flow rate

$$L = K\dot{m} \quad \text{(Eq 5.17.4.2j)}$$

Constant Torque Impeller Flowmeter. The constant torque impeller flowmeter consists of an electric motor, a magnetic clutch, an impeller, and a pick-up coil (Figure 5.17.4.2c). The electric motor rotates the magnetic clutch assembly (permanent magnet) which drives the impeller at a constant torque. The impeller angular velocity is detected by the pick-up coil, which generates a signal at a frequency inversely proportional to the fluid mass flow rate. The interrelationship of the impeller speed ω, mass flow rate m, and impeller torque L, is developed in the discussion of "Turbine-Constant Speed Impeller" and is given by

$$\dot{m} = K_0 \frac{L}{\omega} \quad \text{(Eq 5.17.4.2k)}$$

Since the impeller driving torque is constant, the mass flow rate is inversely proportional to the impeller angular velocity:

$$\dot{m} = \frac{K_1}{\omega} \quad \text{(Eq 5.17.4.2l)}$$

The signal output from the coil may be directed to either an analog or digital readout device.

Gyroscopic Flowmeter. The gyroscopic flowmeter operates on gyroscope principles, where fluid motion in a circular path is equivalent to the spinning wheel. As the spin axis (circular path) is displaced, the flowmeter will tend to move in a third orthogonal axis, unless restrained. The force developed to move the flowmeter in a third axis is

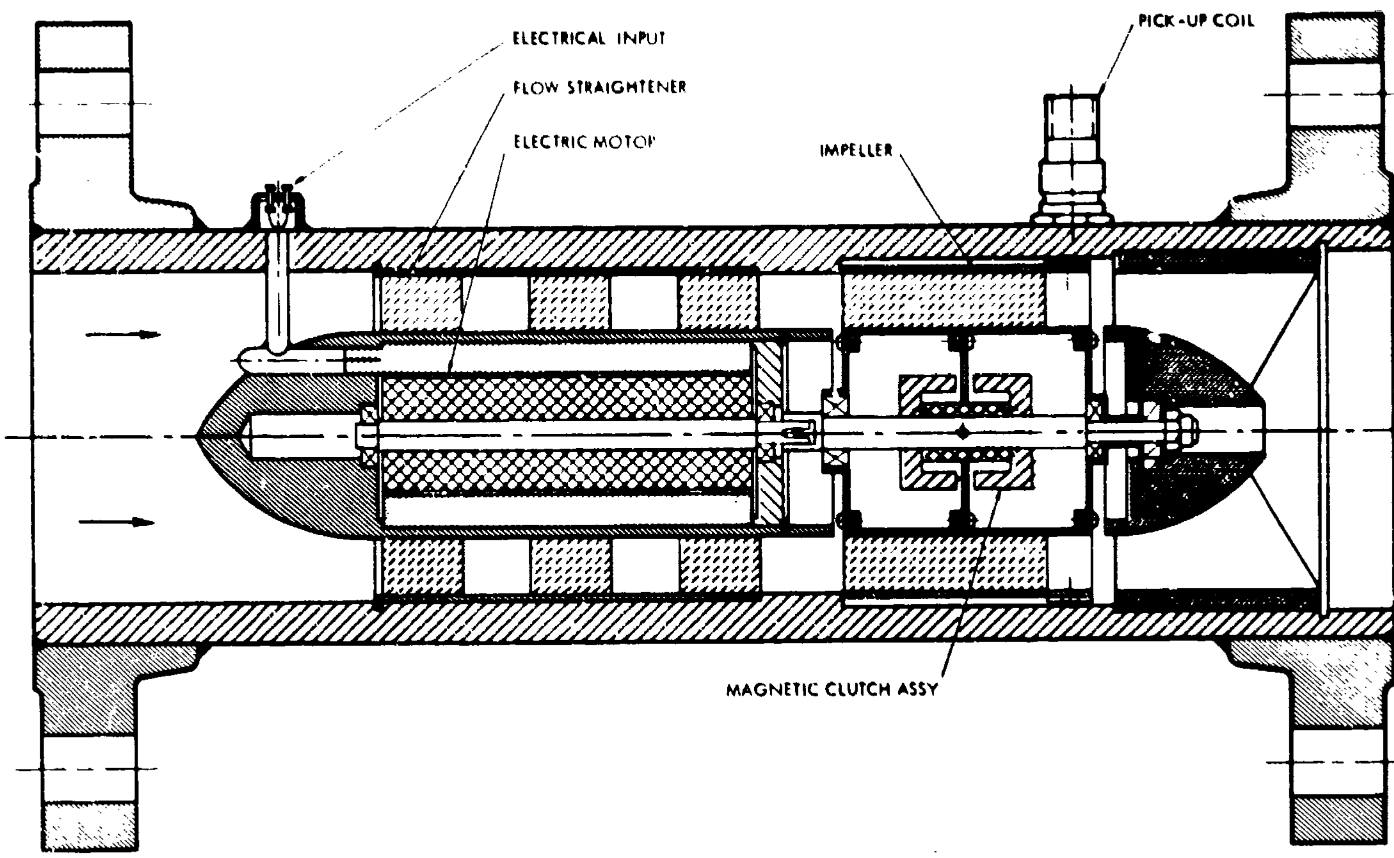

Figure 5.17.4.2c. Constant Torque Impeller Mass Flowmeter
(Courtesy of The Foxboro Company, Waugh Engineering Div., Van Nuys, California)

proportional to the fluid mass flow rate and can be measured by a force or torque sensor. The schematics in Figure 5.17.4.2d illustrate the flow direction, the direction of constant velocity, and the direction of gyroscope action. The torque produced by restraining precession is directly proportional to the mass flow rate as shown below:

Gyroscope reaction torque

$$L = I_s \omega_s \Omega_h \qquad \text{(Eq 5.17.4.2m)}$$

where I_s = polar moment of inertia of fluid about spin (flow) axis $= \rho(2\pi r A) r^2$

ω_s = angular velocity of fluid mass about spin axis $= \dfrac{V}{r}$

Ω_h = precessional angular velocity about horizontal axis $= \dfrac{K}{2\pi r^2}$

V = velocity of fluid in conduit

r = radius

Gyroscope reaction torque is proportional to mass flow rate:

$$L_v = K\dot{m} \qquad \text{(Eq 5.17.4.2n)}$$

Coriolis Flowmeter. The coriolis flowmeter in Figure 5.17.4.2e resembles a centrifugal pump, where fluid enters at the center of an impeller which is rotated at a constant angular velocity. The torque imparted on the impeller and detected by the torque tube in accelerating the fluid mass (by coriolis acceleration) is directly proportional to the mass flow rate as explained below:

Coriolis acceleration (Reference 19-178)

$$\alpha = 2V\omega \qquad \text{(Eq 5.17.4.2o)}$$

From Newton's Second Law, the total force acting on the impeller

$$F = \int_{r_1}^{r_2} \alpha \, dm \qquad \text{(Eq 5.17.4.2p)}$$

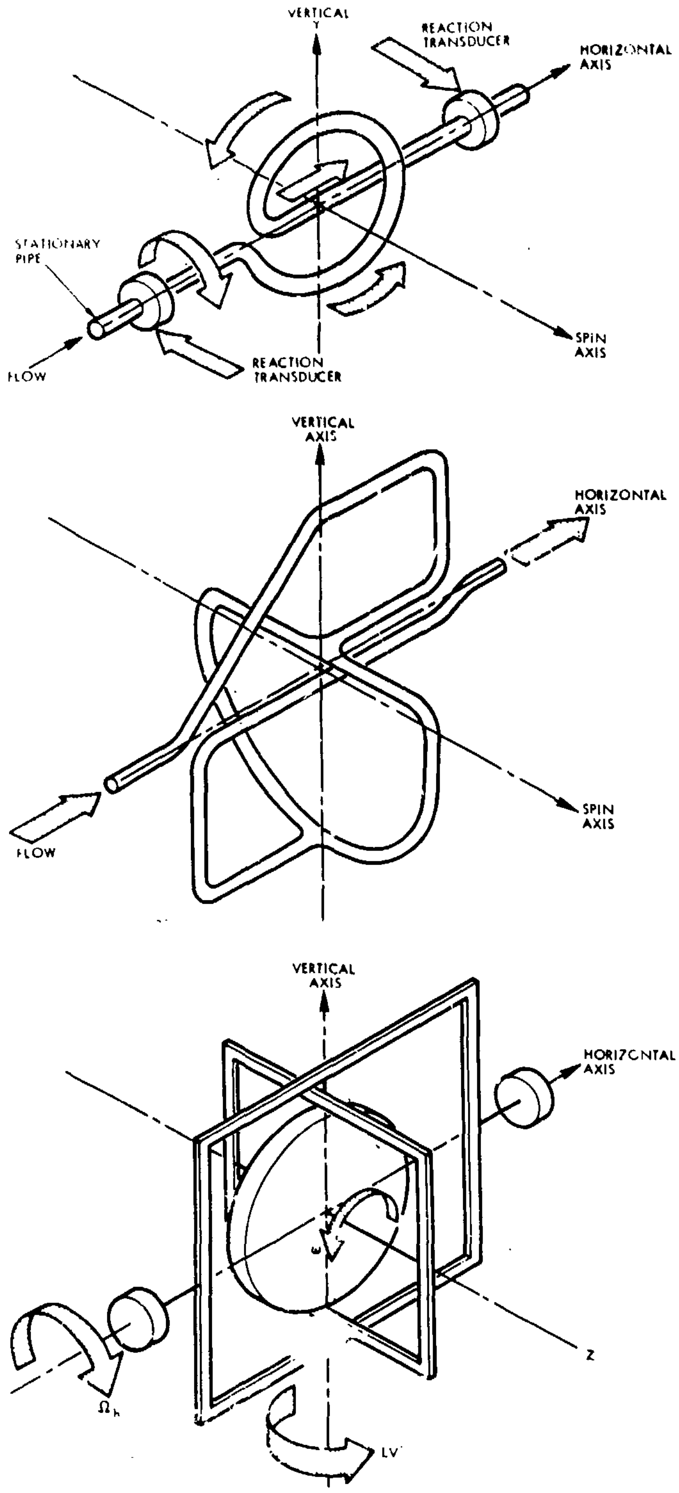

Figure 5.17.4.2d. Gyroscopic Mass Flowmeter
(Reprinted from "ISA Journal," June 1960, vol. 7, no. 6, C. M. Halsell, Copyright 1960, Instrument Society of America, Pittsburgh, Pennsylvania)

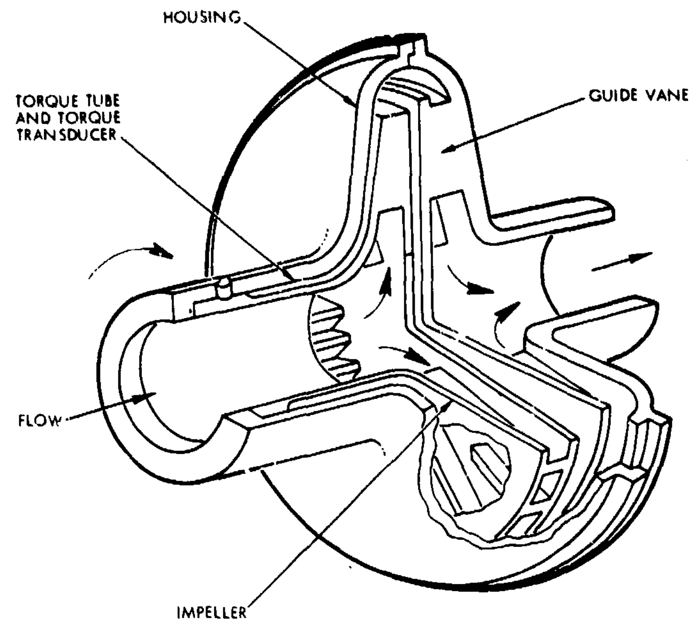

Figure 5.17.4.2e. Coriolis Mass Flowmeter
(Reprinted from "Product Engineering," 8 May 1961, vol. 32, no. 19, C. C. Miesse, Copyright 1961, McGraw-Hill Publishing Company, New York, New York)

The element mass

$$dm = \rho A dr \qquad \text{(Eq 5.17.4.2q)}$$

The total torque about the impeller axis

$$L = \int_{r_1}^{r_2} 2V\omega\rho A r dr \qquad \text{(Eq 5.17.4.2r)}$$

Torque

$$L = \omega (r_2^2 - r_1^2) \dot{m} \qquad \text{(Eq 5.17.4.2s)}$$

Torque due to coriolis force is proportional to the mass flow rate

$$L = K\dot{m} \qquad \text{(Eq 5.17.4.2t)}$$

Magnus Flowmeter. The principle of operation of the magnus flowmeter is represented in the schematic shown in Figure 5.17.4.2f. Fluid enters the meter and is subjected to induced circulation caused by the rotation of the cylinder located in the center of the flow stream. This results in increasing the velocity of the flow of fluid in the part of the flowmeter where the flow velocity and the cylinder tangential velocity are in the same direction, and results also in a decrease in velocity where the flow velocity and the cylinder tangential velocity are opposing. The pressure differential measured at the points of maximum and minimum flow velocity is proportional to the mass flow rate as explained below:

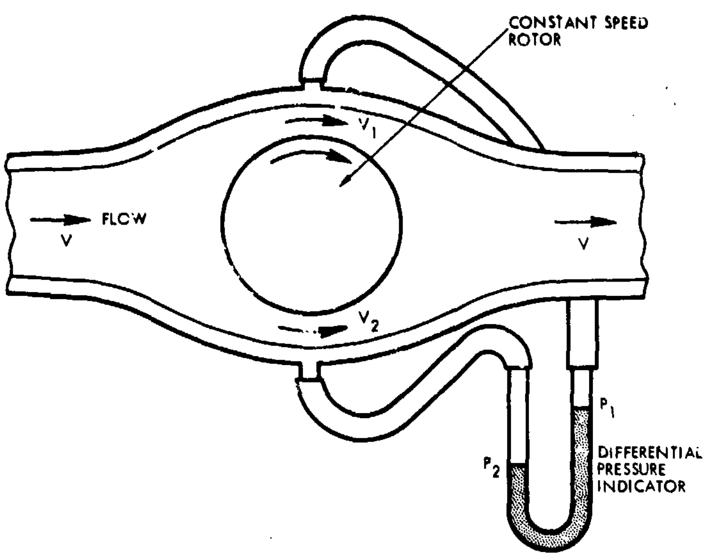

Figure 5.17.4.2f. Magnus Mass Flowmeter
(Reprinted from "Instruments," March 1951, vol. 24, no. 3, D. Brand and L. A. Ginsel, Copyright 1951, Instruments Publishing Company, Pittsburgh, Pennsylvania)

Assuming Bernoulli's equation applies,

$$\frac{p_1}{\rho} + \frac{V_1^2}{2g_c} = \frac{p_2}{\rho} + \frac{V_2^2}{2g_c} \qquad \text{(Eq 5.17.4.2u)}$$

where $V_1 = V + V_r$; $V_2 = V - V_r$

and $p_1 - p_2 = \dfrac{2V\,V_r}{g_c}$

if $\dot{m} = \rho V A$ (Eq 5.17.4.2v)

$$p_1 - p_2 = \frac{2V_r\,\dot{m}}{\rho A g_c} \qquad \text{(Eq 5.17.4.2w)}$$

The pressure differential is proportional to the mass flow rate

$$p_1 - p_2 = K\dot{m} \qquad \text{(Eq 5.17.4.2x)}$$

5.17.4.3 THERMAL. Two thermal flowmeters will be discussed.

Thermopile Flowmeter. The sensing element in the thermopile flowmeter shown in Figure 5.17.4.3a consists of two noble metal thermocouples which are heated by alternating current, and a third cold junction thermocouple which senses heat in the flow stream. As the mass flow rate increases, the heated thermocouples and the cold junction thermocouples tend to seek the same temperature, which reduces the output in the DC meter circuit and the output asymptotically approaches zero. A careful calibration of the meter is necessary to obtain the mass flow rate—DC potential characteristic.

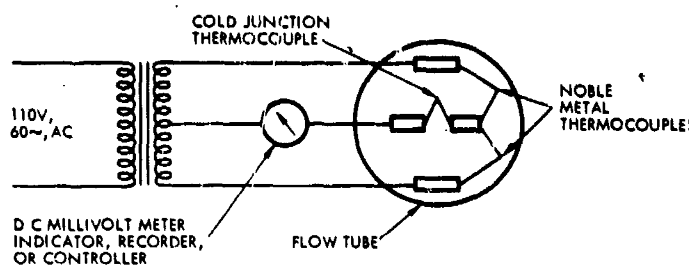

Figure 5.17.4.3a. Thermopile Mass Flowmeter
(Courtesy of Hastings-Raydist, Inc., Hampton, Virginia)

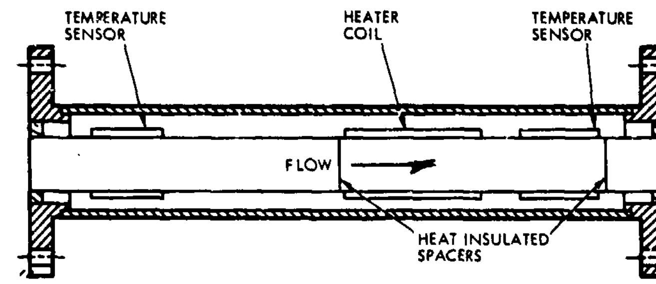

Figure 5.17.4.3b. Boundary Layer Mass Flowmeter
(Reprinted from "Instruments and Control Systems," April 1961, vol. 34, no. 4, J. H. Laub, Copyright 1961, Instruments Publishing Company, Pittsburgh, Pennsylvania)

Boundary Layer Flowmeter. The boundary layer mass flowmeter consists of a long straight tube with a heater coil attached to the external surface, and temperature sensors attached upstream and downstream of the coil (Figure 5.17.4.3b). The heater coil may produce heat by inductance or resistance heating, which is absorbed by the fluid from heat flow through the tubing and boundary layer to the fluid stream. The temperature sensors measure the temperature of the outside surface of the boundary layer on both sides of the heater coil, the differential of which is related to the heat flow rate as shown below:

For turbulent flow

$$q = K_0\,\Delta T\,(c_p)^{0.4}\,(k)^{0.6}\,(\dot{m})^{0.8}/(\mu)^{0.4} \qquad \text{(Eq 5.17.4.3a)}$$

For laminar flow

$$q = K_1\,\Delta T\,(c_p)^{1/3}\,(k)^{2/3}\,(\dot{m})^{1/3} \qquad \text{(Eq 5.17.4.3b)}$$

where q = heat flow rate
$\dot{m}$ = mass flow rate
ΔT = temperature differential

Electrical power to the heater coil is controlled by either variable temperature, constant power or variable power, constant temperature. If the temperature differential is held constant and assuming c_p, k and μ remain constant, the heater power demanded will be proportional to the heat flow rate, which is related to the mass flow rate as follows:

For turbulent flow

$$q = K_2(\dot{m})^{0.8} \qquad \text{(Eq 5.17.4.3c)}$$

For laminar flow

$$q = K_3(\dot{m})^{1/3} \qquad \text{(Eq 5.17.4.3d)}$$

If the electrical power to the heater coil is held constant and c_p, k, and μ remain constant, the heat flow rate will remain constant, and the mass flow rate will be related to the temperature differential as follows:

For turbulent flow

$$\Delta T = K_4(\dot{m})^{-0.8} \qquad \text{(Eq 5.17.4.3e)}$$

For laminar flow

$$\Delta T = K_5(\dot{m})^{-1/3} \qquad \text{(Eq 5.17.4.3f)}$$

5.17.4.4 INFERENTIAL. Three flowmeters will be discussed.

Variable Head Flowmeters with Densitometers. The inferential mass flowmeter consists of any one of the variable head flowmeters listed in Detailed Topic 5.17.3.2 coupled with one of the densitometers listed below:

a) Float type — senses density by measuring buoyant force on float submerged in a by-pass chamber located near the head meter.

b) Dynamic densitometer — senses density by measuring hydraulic coupling between a rotating impeller and a static turbine.

c) Piezoelectric crystal densitometer — senses density by measuring the voltage output of the crystal, which is directly proportional to the density and acoustic velocity of the fluid. Changes in temperature or pressure will effect a change in the acoustic velocity of the fluid, which must be compensated for.

A schematic of a typical variable head meter-densitometer installation is presented in Figure 5.17.4.4a. The densitometer provides an electrical signal proportional to the fluid density, which is fed into a computer. The differential pressure cell output signal is proportional to the fluid density times the square of the volumetric flow rate, and this signal is fed to the computer. The computer calculates the product of the two signals and extracts the square root, resulting in the fluid mass flow rate. The computer output signal can be integrated to provide a total mass flow over a given time period.

Velocity Meters with Densitometers. An inferential mass flowmeter can be constructed by combining a turbine, ultrasonic, or electromagnetic flowmeter with a float, dynamic, or piezoelectric densitometer as sketched in Figure 5.17.4.4b. The function of the computers is merely to multiply the density and velocity signals to provide a readout which is

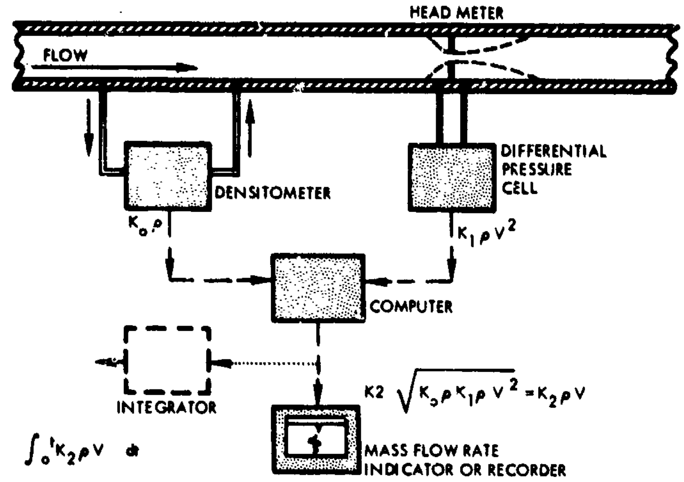

Figure 5.17.4.4a. Variable Head Flowmeter with Densitometer
(Reprinted from "ISA Journal," June 1960, vol. 7, no. 6, C. M. Halsell, Copyright 1960, Instrument Society of America, Pittsburgh, Pennsylvania)

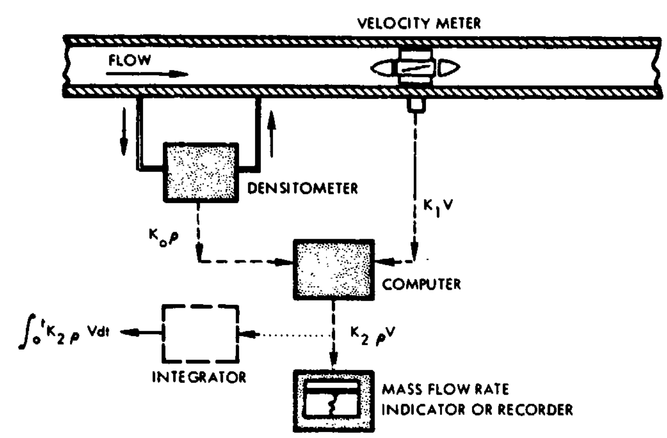

Figure 5.17.4.4b. Velocity Flowmeter With Densitometer
(Reprinted from "ISA Journal," June 1960, vol. 7, no. 6, C. M. Halsell, Copyright 1960, Instrument Society of America, Pittsburgh, Pennsylvania)

proportional to the mass flow rate. The computer output signal can be integrated to provide a total mass flow readout over a given time period.

Variable Head Flowmeters with Velocity Compensation. An inferential mass flowmeter can be constructed by combining the outputs of a velocity flowmeter and a variable head flowmeter (Figure 5.17.4.4c). The output from the velocity flowmeter is proportional to the fluid flow velocity, and the output from the variable head flowmeter is proportional to the fluid density times the flow velocity squared.

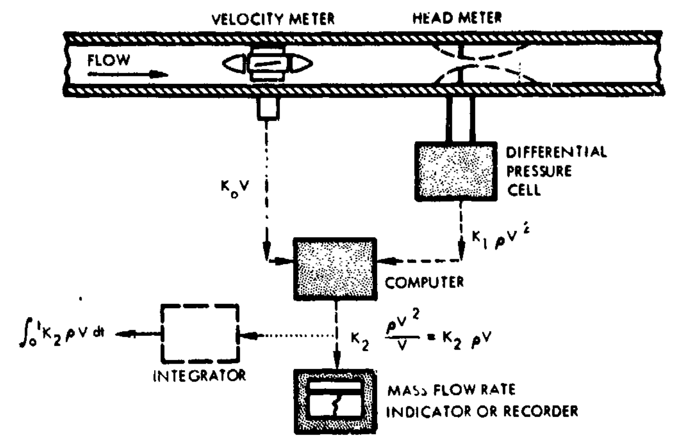

Figure 5.17.4.4c. Variable Head Flowmeter With Velocity Compensation

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By dividing the output from the variable head flowmeter by the output from the velocity flowmeter (using a suitable computer) a signal will be produced which is directly proportional to the fluid mass flow rate. This output signal can be integrated to provide a total mass flow over a given time period.

5.17.5 Velocity Flowmeters

5.17.5.1 GENERAL. Velocity flowmeters are defined as fluid flow measuring instruments which sense fluid flow velocity or react to changes in velocity. Meters included in this broad definition are variable head, variable area, turbine, ultrasonic, electromagnetic, radioisotope, and drag force. The variable head and variable area flowmeters are considered to be inferential, since the former senses the pressure drop due to a change in velocity, and the latter senses the change in the flow area (annular orifice) required to maintain a constant pressure drop across the float.

5.17.5.2 VARIABLE HEAD. Eight types will be discussed.

Thin-Plate Orifice Flowmeter. The orifice plate flowmeter consists of a concentric, square-edged circular hole ranging from 10 to 80 percent of the line diameter, a pressure drop sensor, and pressure ports, one upstream (point 1) and one downstream (point 2) of the orifice plate (Figure 5.17.5.2a). As fluid passes through the orifice meter, a drop in pressure and an increase in velocity results. The relationships between the pressure drop, fluid velocity, fluid properties, and orifice geometry for gases and liquids are presented respectively in Detailed Topics 3.8.2.2 and 3.8.2.3 of the handbook. Values of the discharge coefficient for orifices with flange, vena-contracta, and pipe taps are tabulated in Reference 68-1, pages 142 through 168.

Flow Nozzle Flowmeter. The flow nozzle flowmeter consists of an inlet section which is a smooth convergent nozzle, followed by a cylindrical throat (Figure 5.17.5.2b). The throat diameter generally ranges from 25 to 70 percent of the line diameter. The relationships between pressure drop, fluid velocity, fluid properties, and nozzle geometry for gases and liquids are the same as those for the orifice plate (Detailed Topics 3.8.2.2 and 3.8.2.3) differing only in the values of the discharge coefficient. Values of the discharge coefficient for flow nozzles with corner taps, plus one ID and one-half ID pressure taps are presented in Reference 68-1, pages 134 to 137.

Venturi Flowmeter. The venturi flowmeter is constructed as shown in Figure 5.17.5.2c, with a pressure tap followed by a convergent section, a throat section with a pressure tap, and a long divergent section. As flow enters the divergent section, the static pressure gradually decreases and, correspondingly, the fluid velocity gradually increases to a maximum at the throat. Flow through the divergent section causes a decrease in the fluid velocity and an increase in the static pressure to within about 10 percent of the inlet flow pressure. The relationships between the pressure drop, fluid velocity, fluid properties, and venturi geometry are the same as for the flat plate orifice (Sub-Topic 3.8.3) and differ only in the value of the discharge coefficient, C. The value of the discharge coefficient of the venturi type shown in Figure 5.17.5.2c, is presented in Reference 68-1, page 125.

Dall Flow Tube Flowmeter. The Dall flow tube flowmeter was developed to obtain a flow measuring instrument which would have a low over-all head loss, a short laying length, and a stable discharge coefficient over a wide range of Reynolds numbers. As flow enters the Dall flow tube (Figure 5.17.5.2d) it collides with a dam at A, passes through the inlet cone, striking the sharp edge at B and another at C. The flow then passes over the open throat, striking two more sharp edges, D and E; it then expands rapidly to the line diameter in a short divergent cone. The total length of the Dall flow tube is approximately two times the inlet diameter. The differential pressure produced by a Dall flow tube is approximately 1.9 to 2.4 times as great as that for a corresponding flow nozzle or venturi tube under identical flow conditions. The relationships between the pressure drop, fluid velocity, fluid properties, and meter geometry for gases and liquids are the same as those for the orifice plate (Detailed Topics 3.8.2.2 and 3.8.2.3), differing only in the value of meter coefficient, C. Values for the discharge coefficient with a fixed diameter ratio at various Reynolds numbers are presented in Reference 26-92, Figure 5; values at a fixed Reynolds number of 350,000 with varying diameter ratios are presented in Reference 26-92, Figure 6.

Twin Throat Venturi Flowmeter. The twin throat venturi flowmeter resembles the ordinary venturi meter. The major differences are the short laying length of the twin throat venturi and the shape of the throat section (Figure 5.17.5.2e). The low pressure tap is located at the point of minimum throat diameter, where pressure is further reduced by increased velocity over the streamlined throat

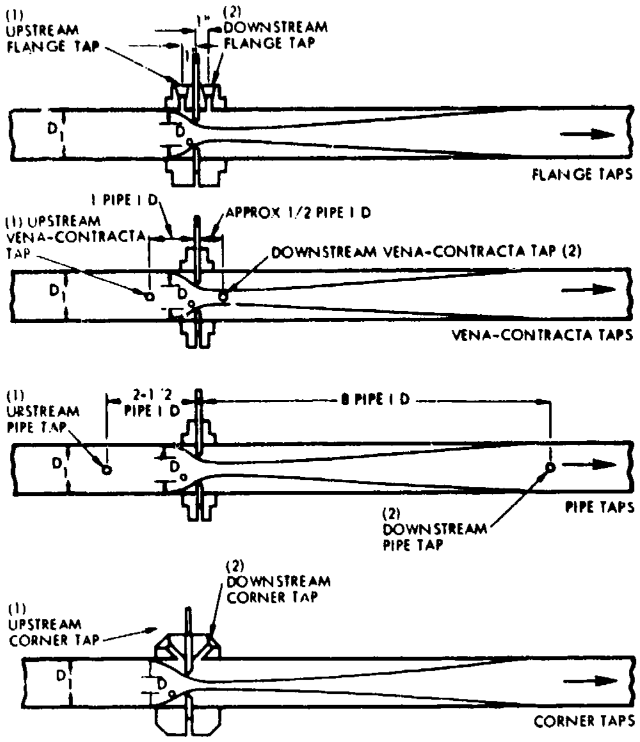


Figure 5.17.5.2a. Thin Plate Orifice Flowmeter With Various Tap Locations

(Reprinted from "Power," February 1956, vol. 100, no. 2, M. R. Skrokov, Copyright 1956, McGraw-Hill Publishing Company, New York, New York)

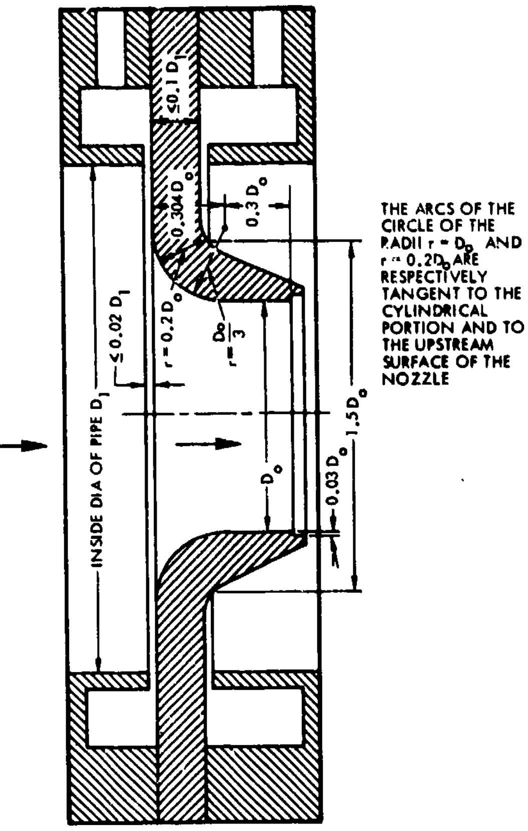
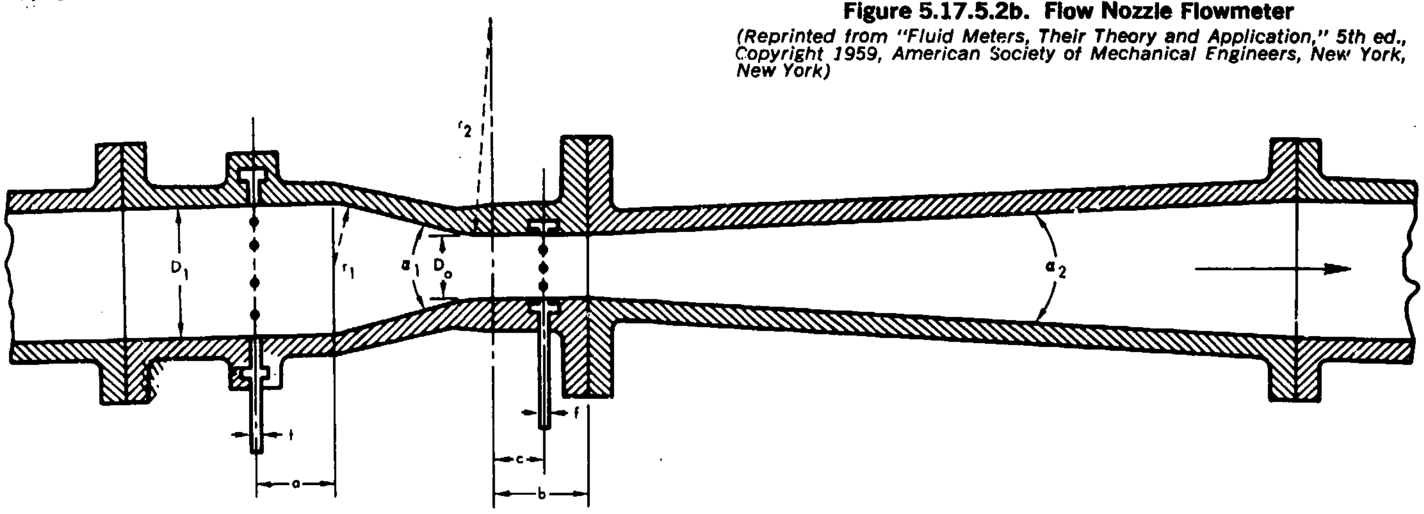


Figure 5.17.5.2b. Flow Nozzle Flowmeter

(Reprinted from "Fluid Meters, Their Theory and Application," 5th ed., Copyright 1959, American Society of Mechanical Engineers, New York, New York)



D_1 = PIPE DIAMETER INLET AND OUTLET

D_o = THROAT DIAMETER AS REQUIRED

a = $0.25D_1$ TO $0.75D_1$ FOR $4" \leq D_1 \leq 6"$, $0.25D_1$ FOR $6" < D_1 \leq 32"$

b = D_o

c = D_o

f = 3/16 IN. TO 1/2 IN. ACCORDING TO D_1. ANNULAR PRESSURE CHAMBER WITH AT LEAST 4 PIEZOMETER VENTS

r_2 = $3.5D_o$ TO $3.75D_o$

r_1 = 0 TO $1375D_1$

α_1 = 21° ± 2°

α_2 = 5° TO 15°

Figure 5.17.5.2c. Venturi Flowmeter

(Reprinted from "Fluid Meters, Their Theory and Application," 5th ed., Copyright 1959, American Society of Mechanical Engineers, New York, New York)

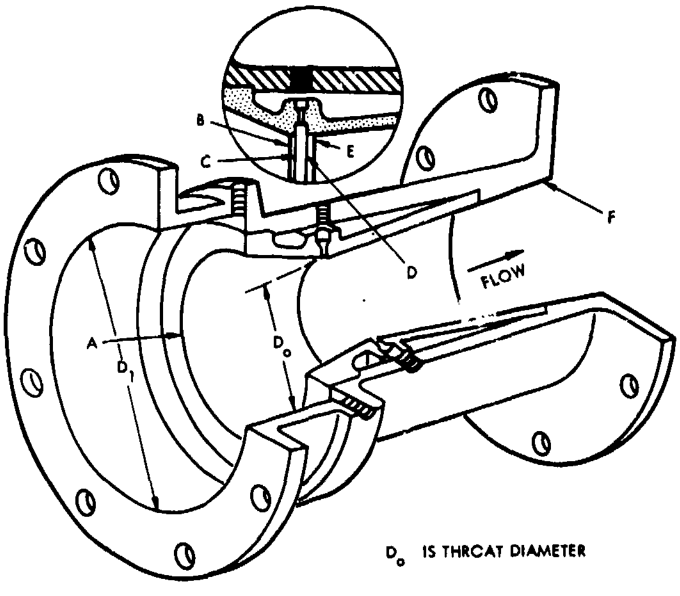


Figure 5.17.5.2d. Dall Flow Tube Flowmeter
(Reprinted from "ASME Trans.," April 1956, vol. 78, no. 3, I. O. Miner, Copyright 1956, American Society of Mechanical Engineers, New York, New York)

section. The twin throat venturi meter has been standardized in sizes from 4 to 48 inches line diameter, with throat to pipe diameter ratios from 0.225 to 0.785, and a two-inch minimum throat diameter restriction. The relationship between pressure drop, fluid velocity, fluid properties and meter geometry for gases and liquids are the same as those for the orifice plate (Detailed Topics 3.8.2.2 and 3.8.2.3), differing only in the value of the discharge coefficient, C. Values for the discharge coefficient for Reynolds numbers ranging from 40,000 to 1,000,000, and throat-to-tube diameter ratios from 0.316 to 0.789 are presented in Reference 26-55, Figure 5.

Capillary Flowmeter. The capillary flowmeter is used for measuring the flow rate of liquids or gases at Reynolds numbers less than 2,000 where the flow rate varies linearly with the pressure differential of the capillary tube. The sensor of this meter consists of a small diameter, straight length of tubing with pressure taps located circumferentially at two stations along the tube (Figure 5.17.5.2f). The relationship between pressure drop, fluid velocity, fluid properties, and meter geometry is presented below:

$$V^2 = \frac{2g_c d_c (p_1 - p_2)}{fL\rho} \qquad \textbf{(Eq 5.17.5.2a)}$$

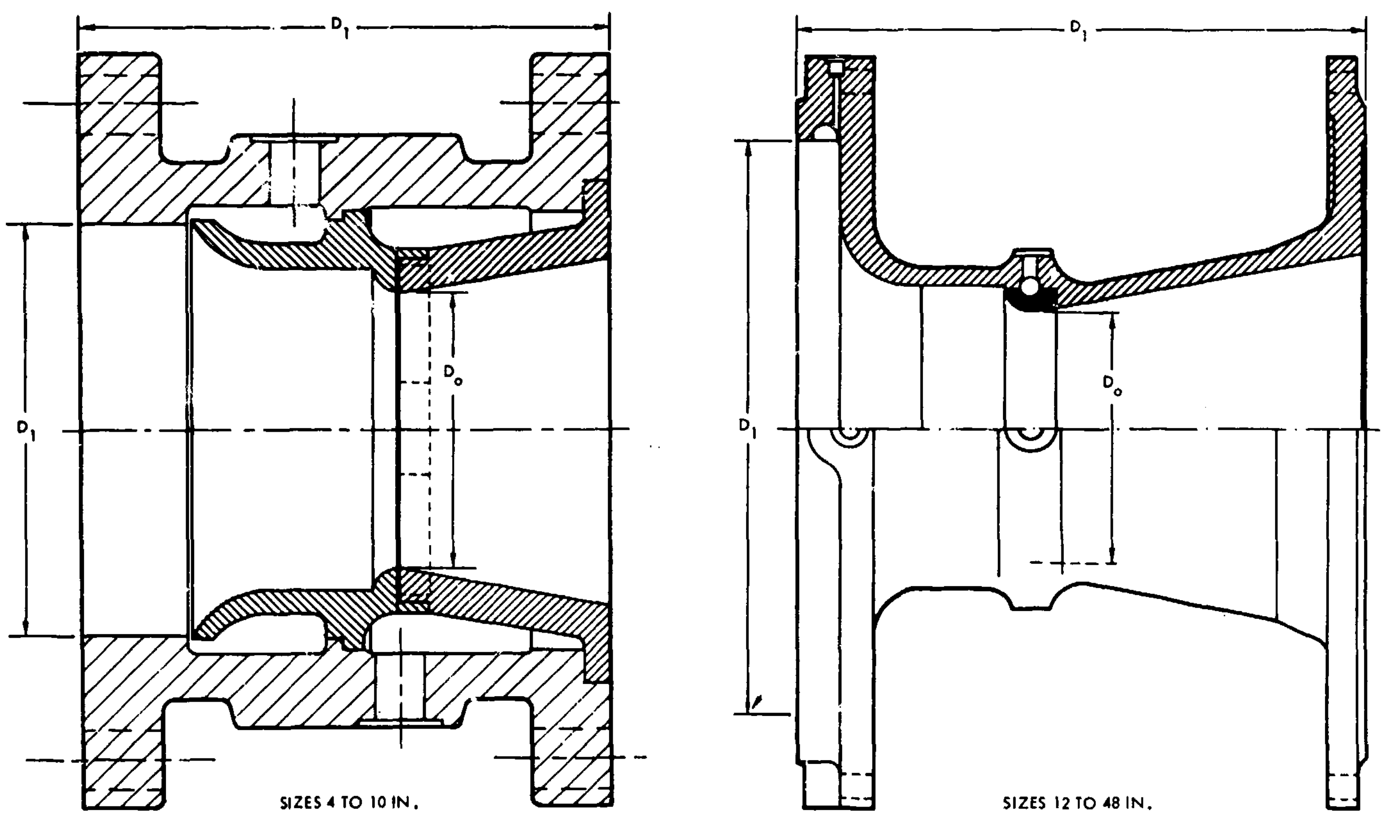


Figure 5.17.5.2e. Twin Throat Venturi Flowmeter
(Reprinted from "ASME Trans., Series D, Journal of Basic Engineering," September 1960, vol. 82, no. 3, A. A. Kalinske, Copyright 1960, American Society of Mechanical Engineers, New York, New York)

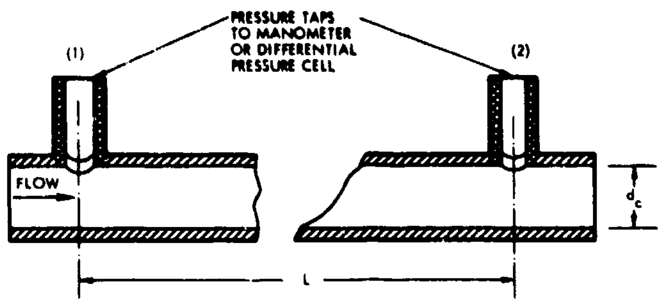


Figure 5.17.5.2f. Capillary Flowmeter
(Reprinted from "Instruments and Control Systems," April 1961, vol. 34, no. 4, L. M. Polentz, Copyright 1961, Instruments Publishing Company, Pittsburgh, Pennsylvania)

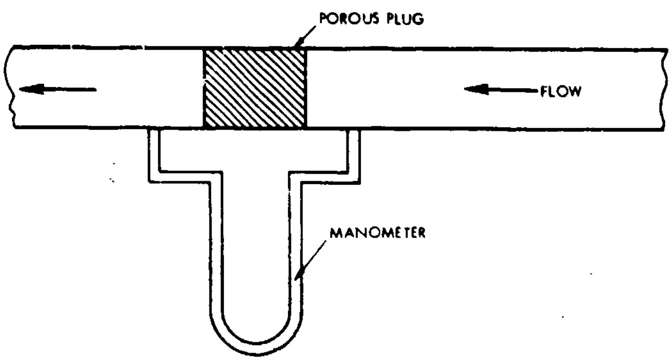


Figure 5.17.5.2h. Pitot Tube Flowmeter Installation
(Reprinted from the "ARS Journal," May-June 1953, vol. 23, no. 3, J. Grey and F. Liu, Copyright 1953, American Rocket Society, New York, N[illegible] York)

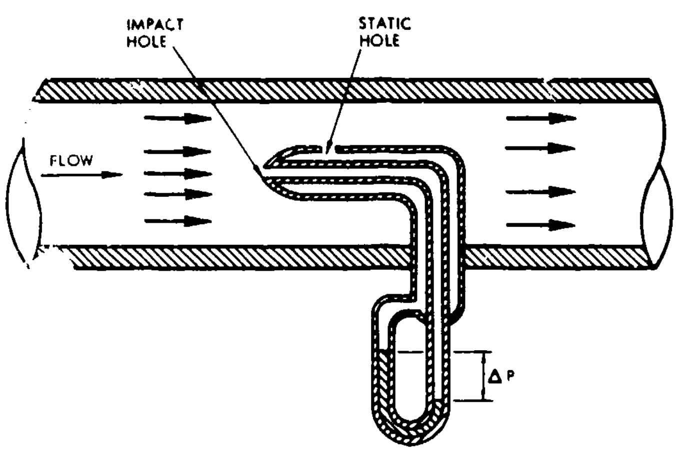


Figure 5.17.5.2g. Porous Flowmeter
(Reprinted from the "ASME Trans.," July 1951, vol. 73, no. 5, S. W. Fleming and R. C. Binder, Copyright 1951, American Society of Mechanical Engineers, New York, New York)

where $f = \frac{64}{Re}$

and $Re = \frac{V d_c \rho}{\mu}$

$$V = \frac{(p_1 - p_2)\, d_c^2 g_c}{32 \mu L} \qquad \textbf{(Eq 5.17.5.2b)}$$

Porous Plug Flowmeter. The porous plug flowmeter consists of a porous plug fastened inside of a straight section of tubing, with circumferential pressure taps located at stations upstream and downstream of the porous plug (Figure 5.17.5.2g). The porous material may consist of sintered metals, steel wool, cotton fibers, glass wool, or layers of fine screening, which are selected for suitability with the fluid to be measured. Tests conducted by the authors of Reference 26-88 have shown that the pressure drop varies linearly with the fluid flow velocity, and further, the dimensionless ratios presented below are useful in correlation of data from various types of porous media:

(Eq 5.17.5.2c)

$$\frac{V L \mu S^2}{p_1 - p_2} = f\,(w_p/w_f)$$

Pitot Tube Flowmeter. The sensing element in the pitot tube flowmeter (Figure 5.17.5.2h) consists of two pressure taps, one facing into the flowing stream to measure the total static and velocity head, and the other located perpendicular to the flow path for measuring the static pressure head. The differential pressure is then the velocity head. Since the fluid velocity varies from a minimum at the wall to a maximum at the center line, it is necessary to take readings over the diameter of the tube to establish the velocity profile. A simple method of averaging the flow velocity is to break up the tube into five equal areas as shown in Figure 5.17.5.2i, and sample the velocity at the ten points shown. The velocity is computed using the following relationships:

Liquids:

(Eq 5.17.5.2d)

$$V = C \sqrt{\frac{2(p_2 - p_1) g_c}{\rho}}$$

Gases:

(Eq 5.17.5.2e)

$$V = C \sqrt{\frac{2 g_c p_1 \gamma}{w_1 (\gamma - 1)} \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}\right]}$$

The C value for pitot tubes varies between 0.965 and 1.00, and should be established experimentally when a high degree of accuracy is required.

5.17.5.3 VARIABLE AREA. Two variable area flowmeters are described in the following paragraphs.

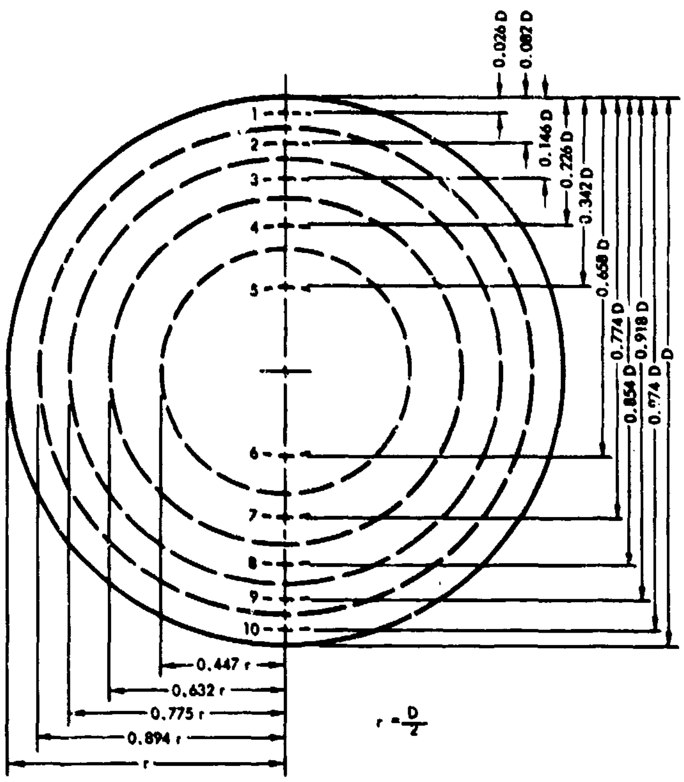


Figure 5.17.5.2l. Pitot Tube Flowmeter Probe Locations
(Reprinted from "Fluid Meters, Their Theory and Application," 5th ed., Copyright 1959, American Society of Mechanical Engineers, New York, New York)

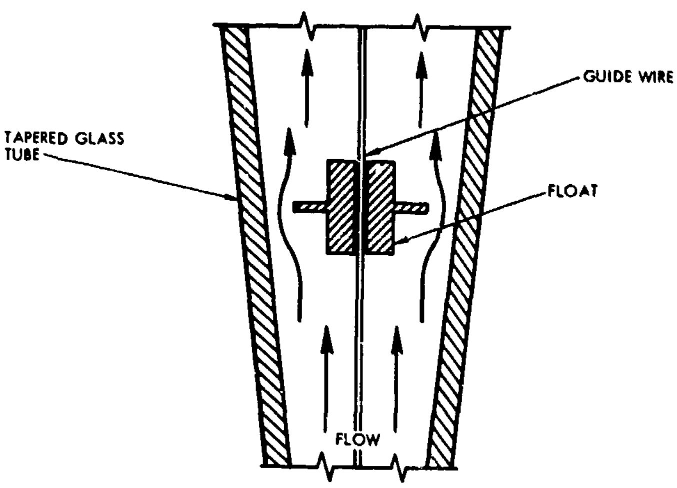


Figure 5.17.5.3a. Tapered Tube and Float Flowmeter
(Reprinted from the "ARS Journal," May-June 1953, vol. 23, no. 3, J. Grey and F. F. Liu, Copyright 1953, American Rocket Society, New York, New York)

Tapered Tube and Float Flowmeter. The tapered tube and float flowmeter shown in Figure 5.17.5.3a consists of a vertical tapered tube, plus a tapered float which is free to move vertically on a guide which centers the float within the tube. The tube is installed vertically with the smaller tube area at the bottom, which is the fluid entrance point. As flow is increased the float is forced higher in the tube, resulting in an increase in the annular area between the float and tapered tube. When equilibrium is established between the float weight (less the fluid buoyancy) and the upward force produced by flow, the float comes to rest at a position which is proportional to the fluid velocity. The pressure drop across the float remains constant, since the float weight is constant. The relationship between pressure drop, fluid velocity, fluid properties, and meter geometry is presented below:

Equilibrium requirement

(Eq 5.17.5.3a)

$$p_1 - p_2 = \frac{F_v\,(\rho_f - \rho_L)\,g}{F_a g_c}$$

Annular orifice equation

(Eq 5.17.5.3b)

$$V = \frac{A_T C}{A}\sqrt{\frac{2\,(p_1 - p_2)\,g_c}{\rho_L}}$$

Combined equations

(Eq 5.17.5.3c)

$$V = \frac{A_T C}{A}\sqrt{\frac{2F_v\,(\rho_f - \rho_L)\,g}{F_a \rho_L}}$$

Slotted Cylinder and Piston Flowmeter. The slotted cylinder and piston flowmeter consists of a cylinder which has equal sized orifices arranged in a uniform helical pattern, plus a free floating piston, as shown in Figure 5.17.5.3b. As the flow velocity increases, the number of exposed orifices increases in direct proportion to the travel of the piston and the fluid flow velocity. An indicator attached to the piston travels inside a calibrated sight glass installed on the meter housing, indicating the fluid velocity or volumetric flow rate.

5.17.5.4 TURBINE FLOWMETER. The turbine flowmeter consists of a turbine in the flow stream which rotates at an angular velocity proportional to the fluid velocity, and a magnetic detector for determining the angular velocity of the turbine. A permanent magnet, imbedded in the rotor, generates an alternating current in a coil which is located in the magnetic detector (Figure 5.17.5.4). The frequency of the alternating current is a direct measurement of the angular velocity of the turbine, which is in turn directly proportional to the fluid flow velocity. The speed of the turbine is related to the flow velocity, the turbine blade angle, and the average radius of the blade center of pressure as shown below:

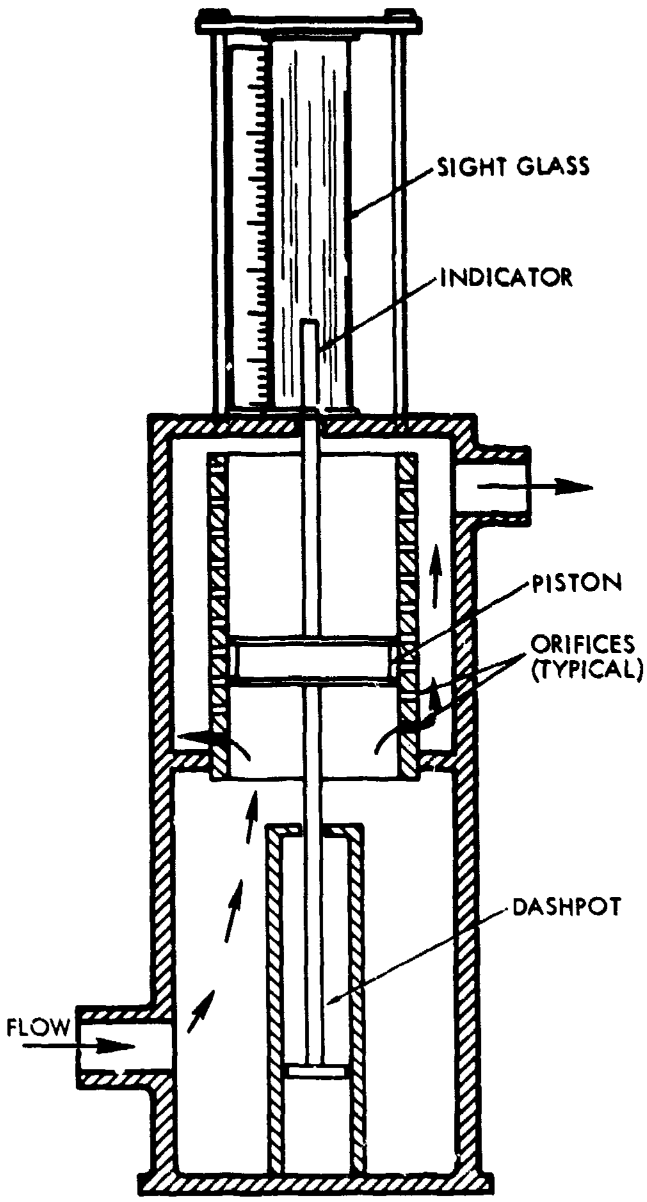


Figure 5.17.5.3b. Slotted Cylinder and Piston Flowmeter
(Reprinted from "Fluid Meters, Their Theory and Application," 5th ed., Copyright 1959, American Society of Mechanical Engineers, New York, New York)

Turbine angular velocity

$$\omega = \frac{V \tan\theta}{R} \qquad \textbf{(Eq 5.17.5.4a)}$$

Flow velocity

$$V = \frac{R\omega}{\tan\theta} \qquad \textbf{(Eq 5.17.5.4b)}$$

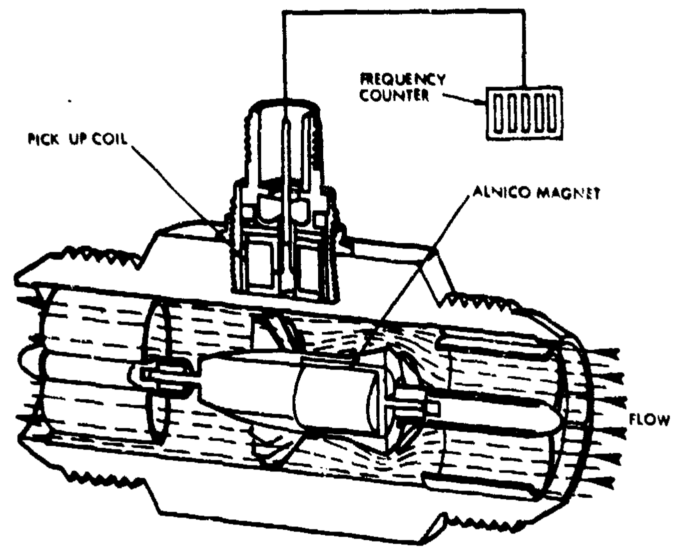


Figure 5.17.5.4. Turbine Flowmeter
(Reprinted from "Jet Propulsion," February 1956, vol. 26, no. 2, J. Grey, Copyright 1956, American Institute of Aeronautics and Astronautics, New York, New York)

Flow rate

$$Q_v = \frac{(A_h - A_r) R\omega}{\tan\theta} \qquad \textbf{(Eq 5.17.5.4c)}$$

Velocity or volumetric flow rate is proportional to turbine angular velocity

$$Q_v = K\omega \qquad \textbf{(Eq 5.17.5.4d)}$$

The electrical output signal from the magnetic detector may be directed to electronic frequency meters for flow rate measurements, or can be connected to an electronic frequency counter for flow totalization.

5.17.5.5 ULTRASONIC FLOWMETER. Two ultrasonic flowmeters are described in the following paragraphs:

Deflection of Sound Beam. This type of ultrasonic flowmeter operates on the principle that the flowing fluid will deflect an ultrasonic beam propagated orthogonally to the flow direction. The meter consists of a straight flow tube with an ultrasonic transmitter mounted on one side, and upstream and downstream receivers mounted opposite to the transmitter (Figure 5.17.5.5a). An ultrasonic pulse is transmitted to the normal direction of the flowing fluid, and is deflected at an angle whose tangent is the ratio of the fluid velocity to the acoustic velocity. The ratio of the fluid velocity to the acoustic velocity is determined by measuring the electrical potential output of the receiver transducers and by using the following relationship (Reference 52-7):

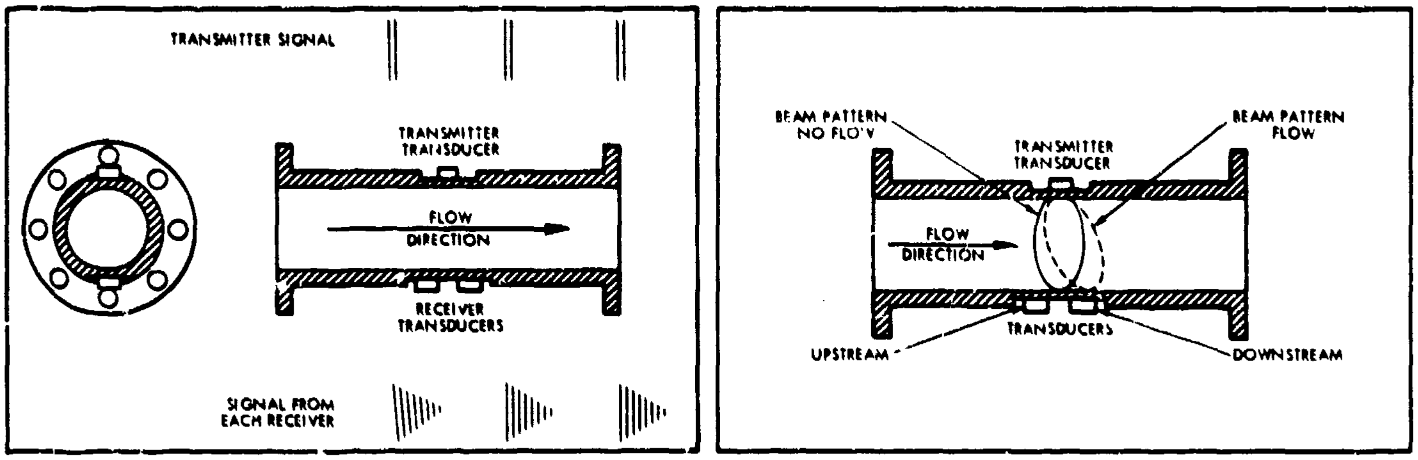


Figure 5.17.5.5a. Deflection of Sound Beam Flowmeter
(Reprinted from the "ISA Journal," October 1960, vol. 7, no. 10, H. E. Dalke and W. Welkowitz, Copyright 1960, Instruments Society of America, Pittsburgh, Pennsylvania)

Average fluid velocity

$$V = \frac{KV_sE_u}{E_D} \qquad \text{(Eq 5.17.5.5a)}$$

Flow rate

$$Q_V = \frac{KV_sAE_u}{E_D} \qquad \text{(Eq 5.17.5.5b)}$$

Flow velocity and volumetric flow rate are proportional to the output ratio of the receivers

$$Q_V = \frac{K_1E_u}{E_D} \qquad \text{(Eq 5.17.5.5c)}$$

Acoustic Velocity Flowmeter. The acoustic velocity flowmeter consists of a straight tube with two sets of ultrasonic transmitters and receivers, one set angled upstream and the other downstream, as shown in Figure 5.17.5.5b. A short train of 10 megacycle oscillations is generated at the transmitter, and is propagated through the liquid to the mating receiver. The signal is then amplified, and retriggers the transmitter to initiate another train of oscillations. The frequency of this retriggering operation for each set of transmitters and receivers depends on the acoustic velocity of the liquid, the fluid flow velocity, and the direction at which the ultrasonic energy is beamed (upstream or downstream). The repetition frequencies for the upstream and downstream directions of ultrasonic energy and their relation to the volumetric flow rate are presented below:

Upstream repetition frequency

(Eq 5.17.5.5d)

$$f_u = \frac{V_s + V\cos\theta_s}{2d_s}$$

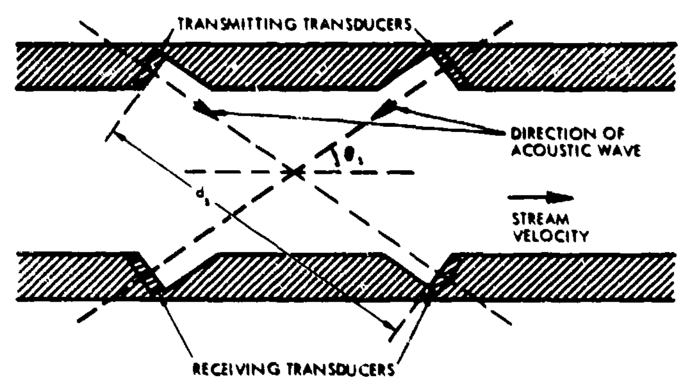


Figure 5.17.5.5b. Acoustic Velocity Flowmeter
(Reprinted from "Product Engineering," May 1961, vol. 32, no. 19, C. C. Miesse and O. E. Curth, Copyright 1961, McGraw-Hill Publishing Company, New York, New York)

Downstream repetition frequency

(Eq 5.17.5.5e)

$$f_d = \frac{V_s - V\cos\theta_s}{2d_s}$$

Beat frequency

(Eq 5.17.5.5f)

$$f_u - f_d = \frac{V\cos\theta_s}{d_s}$$

Average flow velocity

(Eq 5.17.5.5g)

$$V = \frac{(f_u - f_d)\,d_s}{\cos\theta_s} = K\,(f_u - f_d)$$

Flow velocity and volumetric flow rate are proportional to the beat frequency

$$Q_V = K_2 (f_u - f_d) \qquad \textbf{(Eq 5.17.5.5h)}$$

5.17.5.6 **ELECTROMAGNETIC FLOWMETER.** The electromagnetic flowmeter operates on the principle that an electric conducting fluid passing through a magnetic field will generate an electrical potential which is directly proportional to the fluid flow velocity. This flowmeter consists of a flow tube, magnetic poles arranged so that the magnetic field is orthogonal to the direction of fluid flow, and electrodes on the sides of the tubes (Figure 5.17.5.6) for measurement of the electrical potential which is generated by the conducting fluids flowing through the magnetic field. The equations utilizing this flowmeter principle are presented below:

Faraday's law of electromagnetic inductipn

$$E = KBdV \qquad \textbf{(Eq 5.17.5.6a)}$$

Average velocity

$$V = K_1 E \qquad \textbf{(Eq 5.17.5.6b)}$$

Volumetric flow rate is directly proportional to the induced emf

$$Q_V = K_1 AE \qquad \textbf{(Eq 5.17.5.6c)}$$

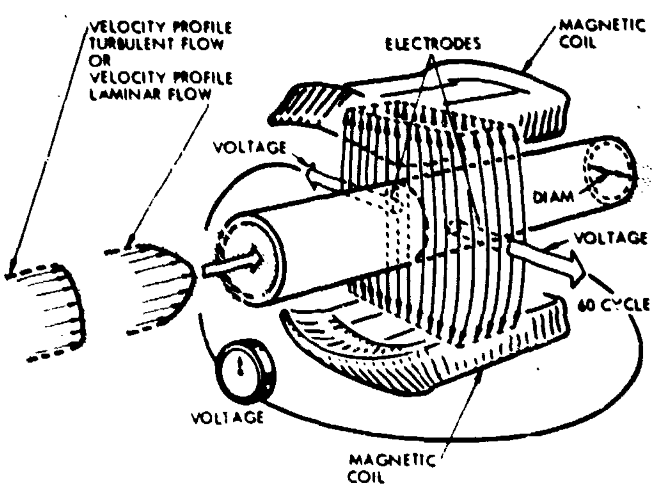


Figure 5.17.5.6. Electro Magnetic Flowmeter
(Reprinted from "Instruments and Control Systems," September 1960, vol. 33, no. 9, E. D. Woodring, Copyright 1960, Instruments Publishing Company, Pittsburgh, Pennsylvania)

5.17.5.7 **RADIOISOTOPE FLOWMETER.** The radioisotope flowmeter consists of a straight tube of known cross-sectional area with a port for injecting radioactive material, two sensors spaced at a known distance, and a timer connected to the two sensors for measuring the time for the radioactive particles to pass between two known points. A schematic of this flowmeter is shown in Figure 5.17.5.7. The average velocity and volumetric flow rate are then determined from the time measured for flow between the two sensors.

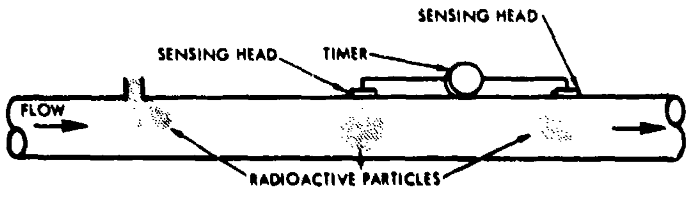


Figure 5.17.5.7. Radioisotope Flowmeter
(Reprinted from "Automation," September 1957, vol. 4, no. 9, J. J. Combes and M. J. DePasquale, Copyright 1957, Penton Publishing Company, Cleveland, Ohio)

5.17.5.8 **DRAG FORCE FLOWMETER.** The drag force flowmeter consists of a contoured drag body, lever rod, and strain tube, as illustrated in Figure 5.17.5.8. It operates on the principle that volumetric flow rate is proportional to the square root of the drag forces acting on a body placed in a flow stream. Flow around the contoured drag body creates drag forces which are transmitted through the lever rod to the strain tube. Strain gages attached to the strain tube sense the bending stress, which is proportional to the drag forces.

The drag forces are related to the volumetric flow rate as follows:

$$F = \frac{C_d \rho}{2Ag_c} Q^2 \qquad \textbf{(Eq 5.17.5.8a)}$$

for $R_e > 400$ (Reference V-258)

The volumetric flow rate is proportional to the square root of the drag forces

$$Q = K\sqrt{\frac{F}{\rho}} \qquad \textbf{(Eq 5.17.5.8b)}$$

5.17.6 Positive Displacement Volumetric Flowmeters

5.17.6.1 **GENERAL.** Positive displacement volumetric flowmeters are defined as fluid flow measuring instruments where each cycle of movement—whether rotary, reciprocating, or otherwise—represents a specific volume of fluid passing through the meter. Fluid flowmeters included in this broad definition are nutating disc, rotary vane, lobed impeller, reciprocating piston, and axial piston. The volumetric flow rate is determined by computing the product of meter frequency multiplied by the meter characteristic of volume per cycle. Volumetric totalization is obtained by multiplying the accumulated meter cycles from a frequency counter by the meter characteristics.

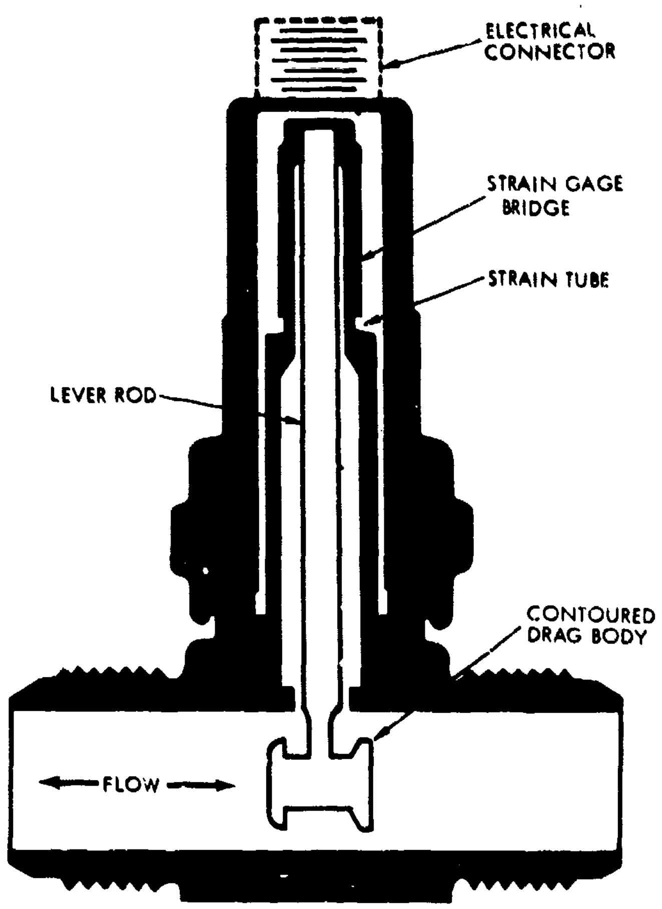


Figure 5.17.5.8. Drag Force Flowmeter
(Courtesy of Ramapo Instrument Company, Inc., Bloomingdale, New Jersey)

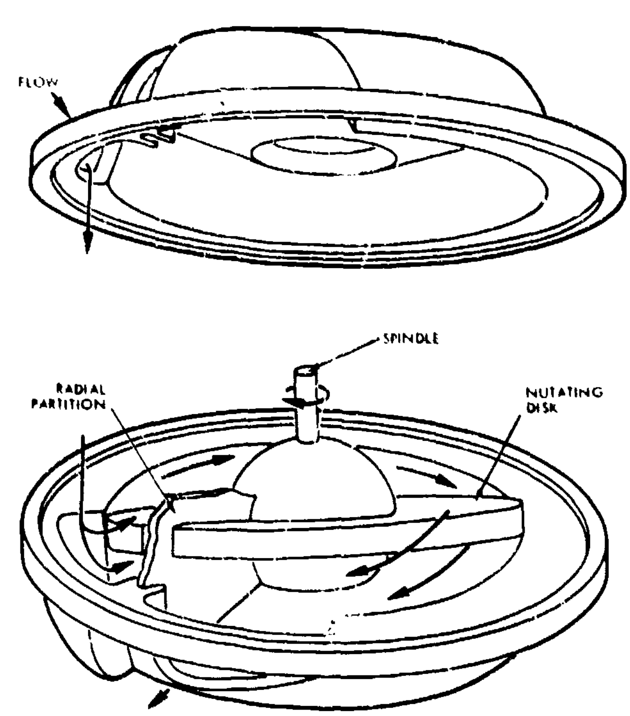


Figure 5.17.6.2. Nutating Disc Flowmeter
(Reprinted from "Automation," September 1957, vol. 4, no. 9, J. J. Combes and M. J. DePasquale, Copyright 1957, Penton Publishing Company, Cleveland, Ohio)

5.17.6.2 NUTATING DISC FLOWMETER. The nutating disc flowmeter consists of a disc with a ball at its center which mates in a socket in the meter case (Figure 5.17.6.2). Fluid enters the chamber on one side of the radial partition impinging on the disc, causing it to nutate up or down, thus allowing the fluid to flow through the chamber to the outlet. The disc spindle makes one revolution each time the nutating disc travels from the chamber bottom to the chamber top and back again. An electronic or mechanical counter attached to the disc spindle is used to display the number of revolutions or cycles of the meter. The instantaneous cycling frequency is a direct measurement of the instantaneous volumetric flow rate, and the number of cycles in a measure of flow totalization.

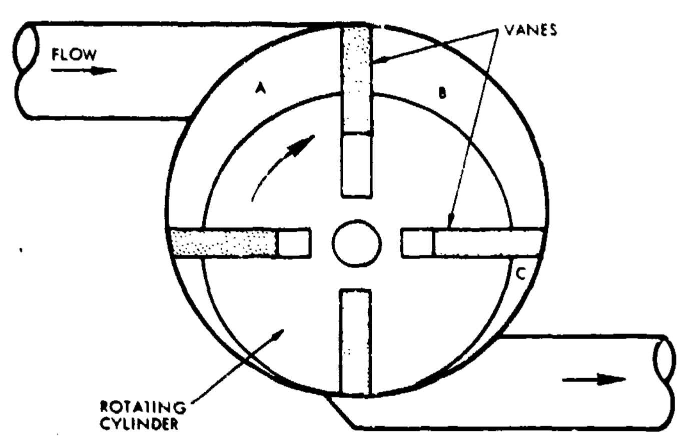


Figure 5.17.6.3. Rotary Vane Flowmeter
(Reprinted from "Automation," September 1957, vol. 4, no. 9, J. J. Combes and M. J. DePasquale, Copyright 1957, Penton Publishing Company, Cleveland, Ohio)

5.17.6.3 ROTARY VANE FLOWMETER. The rotary vane flowmeter consists of a cylindrical body, a rotating cylinder, and spring-loaded sliding vanes attached to the rotating cylinder (Figure 5.17.6.3). Fluid enters the meter at space A, which causes the flowmeter to rotate clockwise due to the larger area on one end. Space B, which can be considered the metering space, is moved to space C, which is the point where fluid exits from the meter. The liquid volumetric flow rate is determined by measuring the angular velocity of the rotor. Each revolution of the rotor displaces four times the volume represented in space B. Instantaneous flow rate can be determined by measuring the instantaneous angular velocity of the rotor, and total flow can be determined by mechanically or electronically counting the number of cycles of revolution of the rotor.

5.17.6.4 LOBED IMPELLER FLOWMETER. The lobed impeller flowmeter (Figure 5.17.6.4) consists of a flow chamber and a pair of lobed impellers which are kept in proper relationship to each other by means of equal tooth gears. The lobes are carefully machined to provide small clearances between the lobes and the flow chamber to reduce the leakage to a minimum without increasing friction. The angular velocity of the rotors is directly proportional to the volumetric flow rate through the meter. Totalization of flows can be obtained merely by counting the number of revolutions of the rotors and multiplying this by the meter displacement per revolution.

5.17.6.5 RECIPROCATING PISTON FLOWMETER. The reciprocating piston flowmeter consists of a cylinder, a double-acting piston, a velocity transducer, and an accurate DC millivolt meter (Figure 5.17.6.5). As fluid enters the cylinder from one end, the piston moves at a velocity proportional to the volumetric flow rate. A velocity transducer

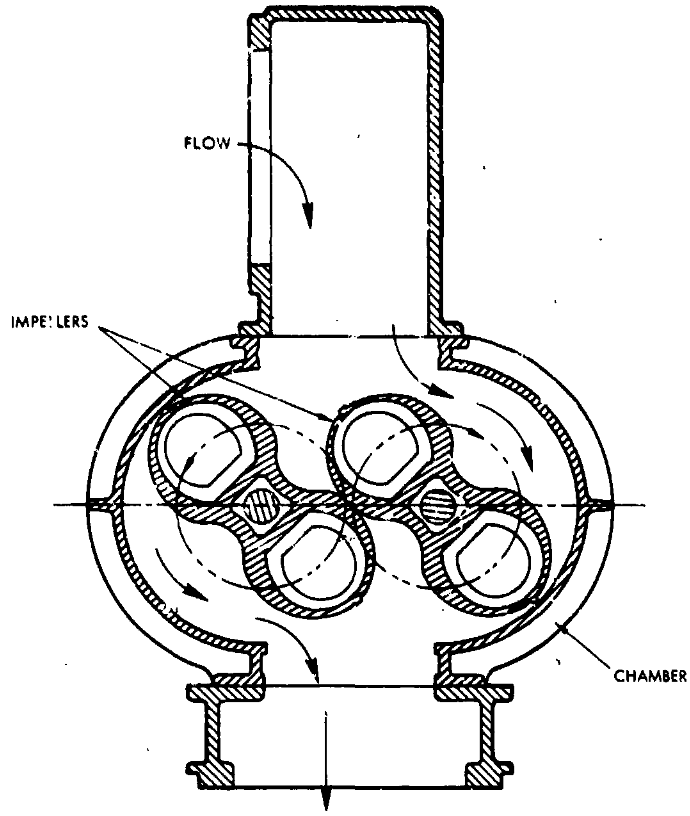


Figure 5.17.6.4. Lobed Impeller Flowmeter
(Reprinted from "Fluid Meters, Their Theory and Application, 5th ed., Copyright 1959, American Society of Mechanical Engineers, New York, New York)

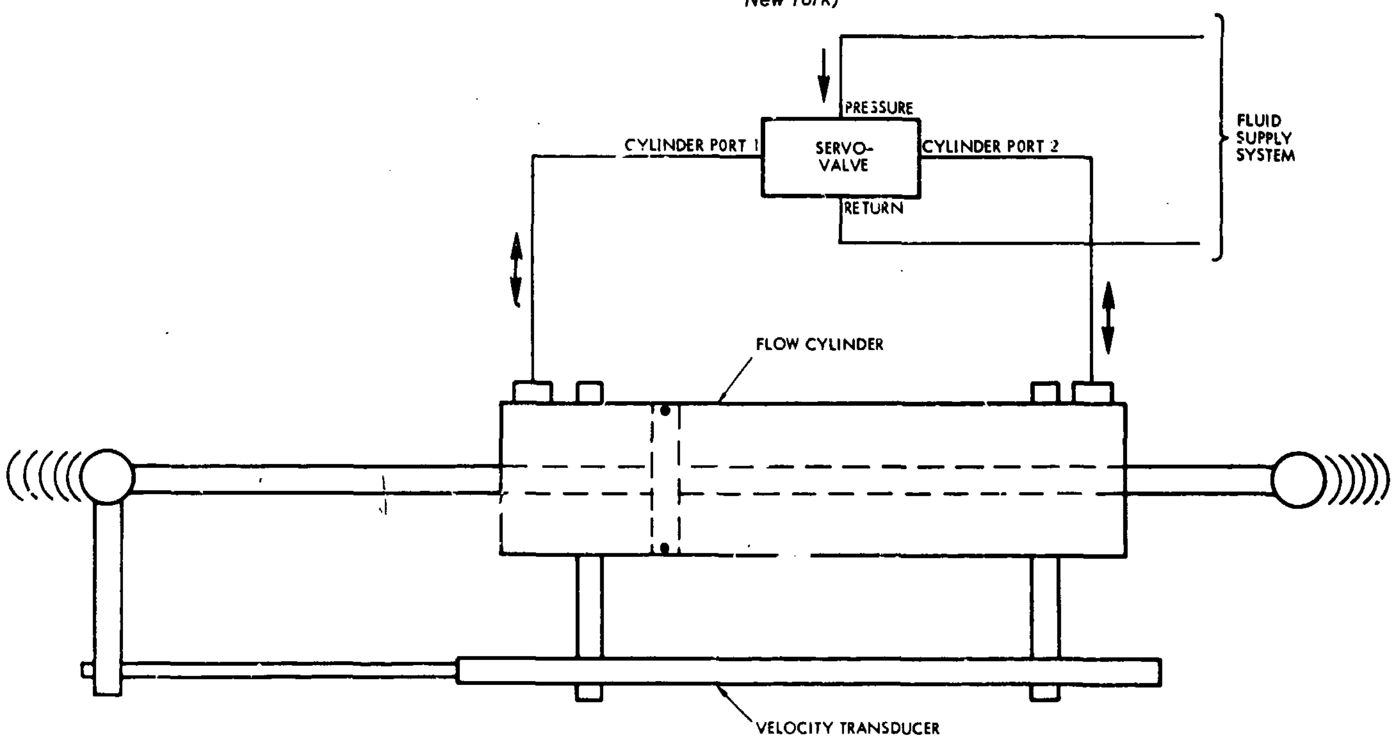


Figure 5.17.6.5. Reciprocating Piston Flowmeter

generates a signal which is indicated on the millivolt meter. For measuring static servo valve characteristics, the millivolt meter is ordinarily a high accuracy XY plotter, and for determining frequency response it may be a high speed oscillograph or a frequency response analyzer. The piston either reverses its direction in responding to a reversed input signal to the servo valve, or may be reversed automatically using auxiliary equipment.

5.17.6.6 AXIAL PISTON METERING PUMPS. Axial piston metering pumps can be used as positive displacement flowmeters. The pump, driven as if it were a hydraulic motor without load, will react to the pressure on the pistons, causing the pump shaft to rotate at an angular velocity proportional to the volumetric flow rate. The flow rate can be determined by measuring the angular velocity of the pump shaft and multiplying this by the volume per revolution characteristic of the pump. A cross-section of an axial piston pump is presented in Figure 5.17.6.6.

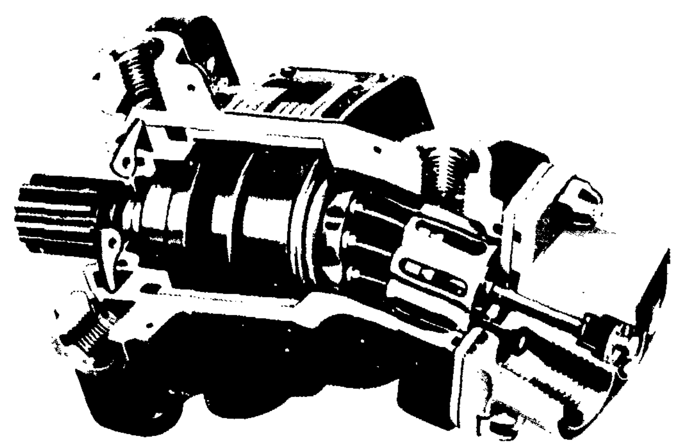


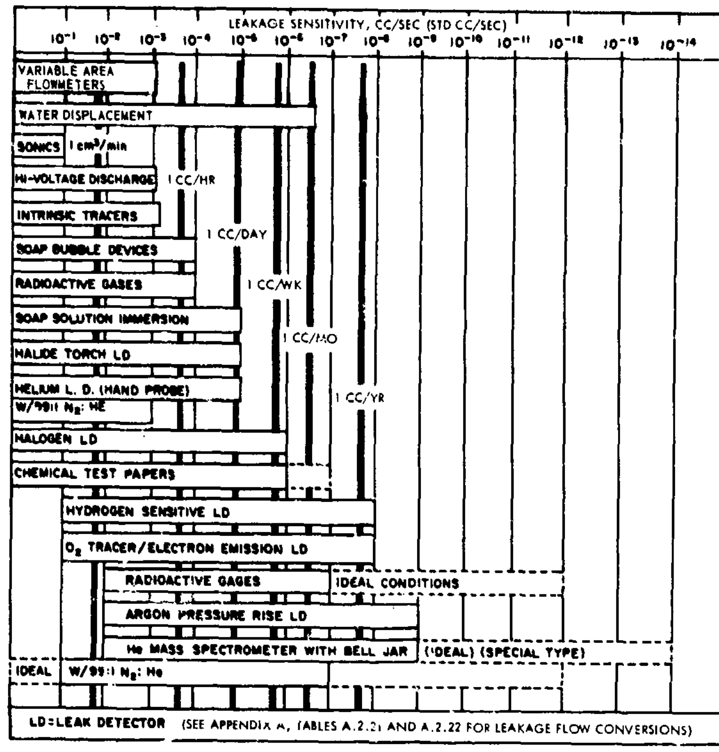
Figure 5.17.6.6. Axial Piston Pump Flowmeter
(Courtesy of Vickers, Inc., Detroit, Michigan)

5.17.7 Gaseous Leakage Flowmeters and Detectors

5.17.7.1 GENERAL. The detection and measurement of gas leakage in the flow range of 10^{-11} to 10^{3} standard cc/sec are discussed in the following Detailed Topics. Included are descriptions of the mass spectrometer, ultrasonic translator, water displacement and variable area flowmeter techniques. A comparison chart (Table 5.17.7.1) of their usable flow ranges is also presented.

5.17.7.2 MASS SPECTROMETER. The mass spectrometer is sensitive to minute quantities of inert tracer gas, which may be helium, neon, or argon. Molecules of the tracer gas are drawn into the instrument, ionized by an electron beam, and separated from all ions of different molecular weight. The tracer gas ions are then focused on a collector, producing an electric current which is amplified to operate the

Table 5.17.7.1. Comparison Chart



panel meter. The panel meter is used as a comparator between a standard leak and the part being tested. The sensitivity of the leak detector in standard cc/sec/meter division is determined as follows:

$$S = \frac{L}{n \times \text{meter multiplier}}$$

(Eq 5.17.7.2)

where S = sensitivity, standard cc/sec/meter division

n = number of divisions on panel meter registered at standard leak rate

L = leakage rate, standard cc/sec

The illustrations in Figure 5.17.7.2 show the various modes of vacuum and pressure testing for which the mass spectrometer can be used. Vacuum testing is preferred to pressure testing because dilution of the helium effluent by atmospheric air does not occur. Further information concerning techniques, calibration, and application of the mass spectrometer are presented in the papers collected in Reference V-222.

5.17.7.3 ULTRASONIC TRANSLATOR. The ultrasonic translator detects ultrasonic sound waves created by the flow of gas molecules as they pass through minute holes and fissures. The sound waves are amplified electronically and directed to a loud-speaker or earphones for detection

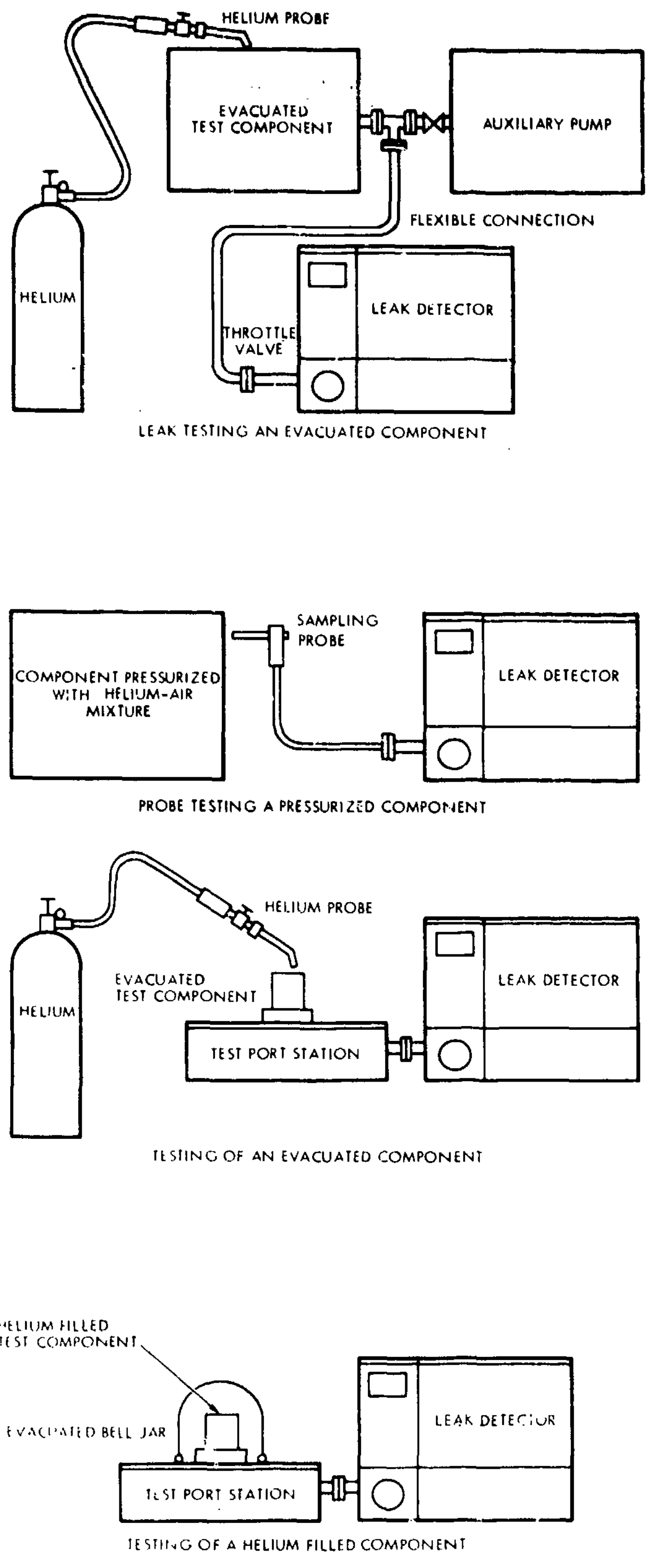


Figure 5.17.7.2. Mass Spectrometer Gas Leakage Detector
(Courtesy of Consolidated Electro Dynamics Corporation, Pasadena, California)

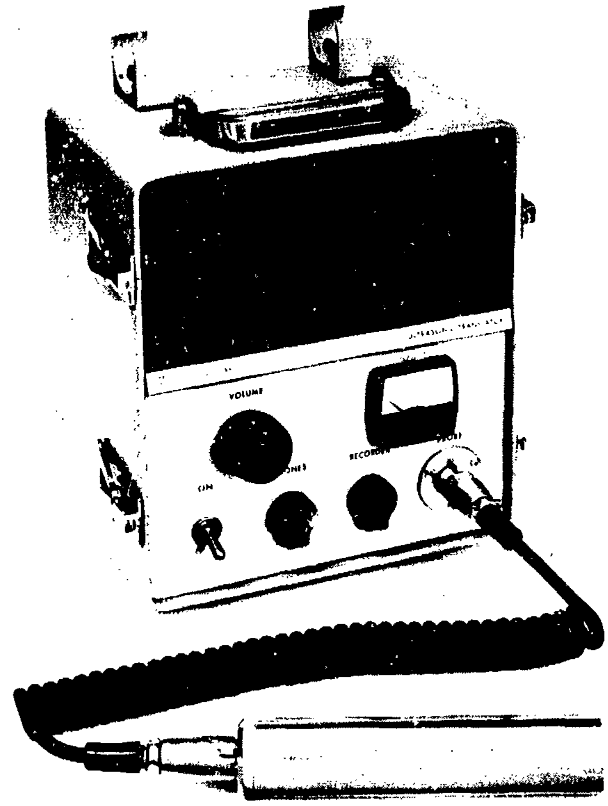


Figure 5.17.7.3. Ultrasonic Translator Gas Leakage Detector.
(Courtesy of Delcon Corporation, Palo Alto, California)

by the operator. Although this instrument serves primarily as a leakage detector, quantitative data can be estimated based on the detection distance. When tests of an ultrasonic translator were conducted (Reference 70-2) it was concluded that the unit is capable of detecting gas leakages of approximately 100 standard cc/min in surroundings of extraneous noise. A photograph of an ultrasonic translator is presented in Figure 5.17.7.3.

5.17.7.4 WATER DISPLACEMENT LEAKAGE FLOWMETER. A simple, reliable method of measuring gaseous leakage, where the leakage point can be sealed within a duct, is illustrated in Figure 5.17.7.4. The gaseous leakage flow is ducted to an inverted graduate, where the gas is trapped and displaces the water. By measuring the time required to displace a given calibrated volume, the leakage flow rate can be established. Where the leakage gas readily dissolves in water, allowances must be made for the amount of gas lost in solution. The graduate should be sized according to the leakage rate to permit sufficient time to make an accurate time reading. Volumetric corrections should be made to account for the increase in gas pressure due to the increase in the height of the water surface above the gas-water interface.

5.17.7.5 VARIABLE AREA FLOWMETER. The tri-flat variable area flowmeter consists of a tapered glass tube (Figure 5.17.7.5) with its axis vertical and the smaller end at the bottom. A spherical float, which is free to move

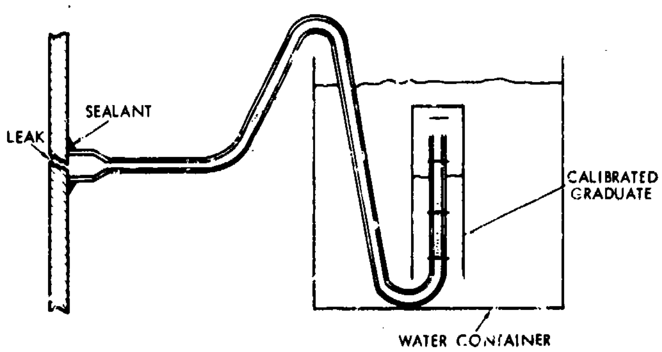


Figure 5.17.7.4. Water Displacement Gas Leakage Flowmeter

vertically, is placed within the tube. The gas entering the tube from the bottom raises the float, increasing the area between the float and tube until the flow forces are balanced by the net weight of the buoyed float. The tube is available in sizes from 1/16 to 3/8 inch nominal diameter, and in four float materials—sapphire, stainless steel, Carboloy, and Tantalum. This flowmeter can be used only if the leakage point can be sealed to a line connected to the flowmeter.

5.17.8 Flowmeter Calibration Systems

Four types of flowmeter calibration systems are discussed.

5.17.8.1 STATIC WEIGHING CALIBRATOR. The static weighing calibrator shown in Figure 5.17.8.1 consists of a flow control valve, a flow diverter valve, an electronic timer, a weigh tank, and a platform scale. Filtered fluid of known physical properties and at a controlled temperature enters the system, first passing through a flow straightener, the flowmeter to be calibrated, a flow control valve, a nozzle, and finally through a diverter valve to the storage tank. The tare weight of the weigh tank and its contents is carefully determined while fluid is being transferred through the calibration system to the storage tank. When the calibration flow rate has been established, the diverter valve is actuated, automatically starting the electronic timer. When sufficient fluid has been collected in the weigh tank, the diverter valve is again actuated, automatically stopping the automatic timer and directing the fluid to the storage tank. The weight of the weigh tank and its contents is carefully measured, and the tare weight subtracted to determine the total quantity of fluid which has been collected over the time interval indicated on the electronic timer. All weighings are performed under static conditions, with the weigh tank free of all connecting lines and hoses. For calibration of cryogenic or other high vapor pressure liquids, it is necessary to use a pressurized receiver tank in place of the weigh tank to reduce loss by evaporation. This requires the use of flexible lines to the receiver tank, tending to reduce the accuracy of the calibration system.

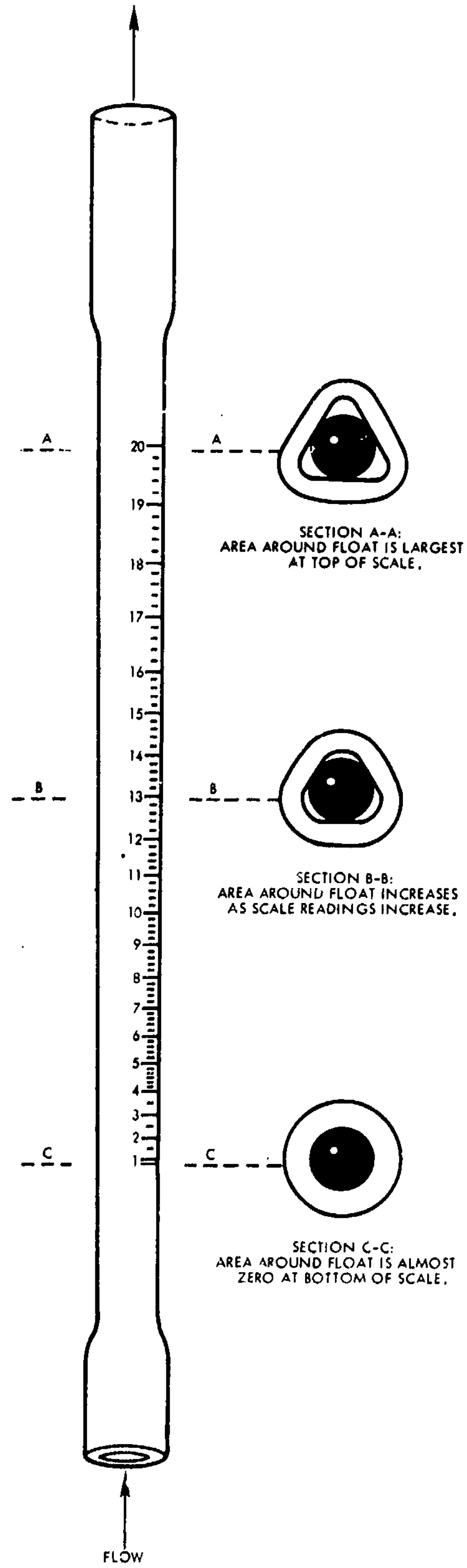


Figure 5.17.7.5. Variable Tri-Flat Flowmeter
(Courtesy of Fischer and Porter Company, Warminster, Pennsylvania)

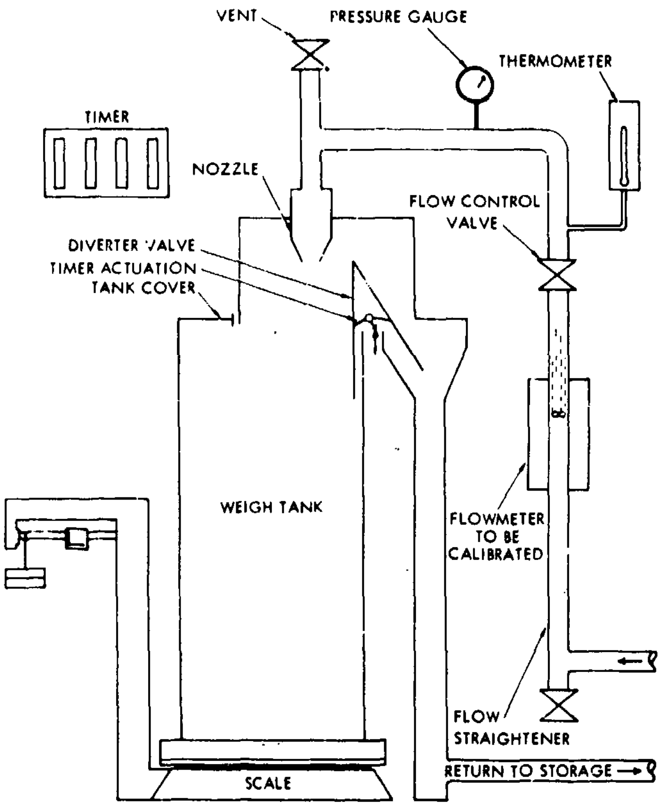


Figure 5.17.8.1. Static Weighing Flowmeter Calibrator
(Reprinted from "ASME Trans.," October 1958, vol. 80, M. R. Shafer and F. W. Ruegg, Copyright 1958, American Society of Mechanical Engineers, New York, New York)

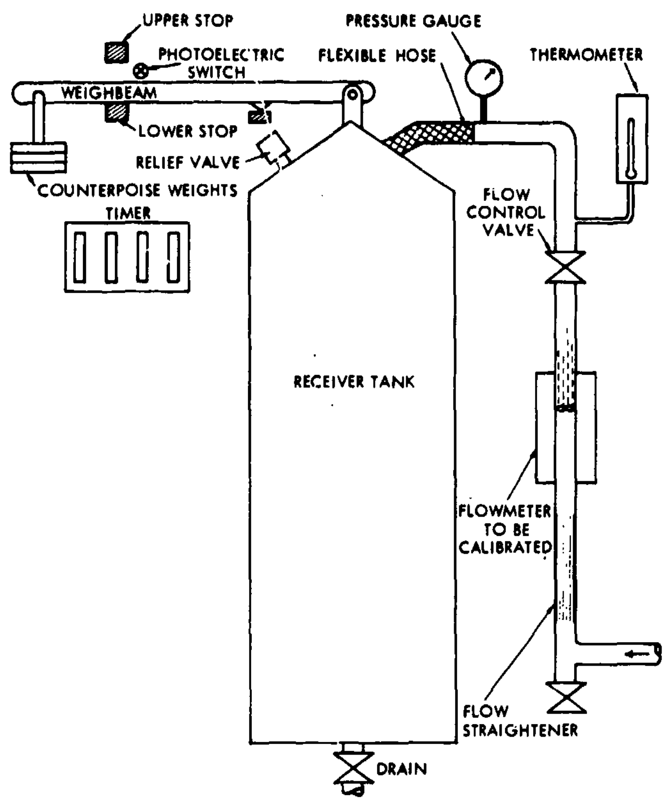


Figure 5.17.8.2. Dynamic Weighing Flowmeter Calibrator
(Reprinted from "ASME Trans.," October 1958, vol. 80, M. R. Shafer and F. W. Ruegg, Copyright 1958, American Society of Mechanical Engineers, New York, New York)

5.17.8.2 DYNAMIC WEIGHING CALIBRATOR. The dynamic weighing calibrator (Figure 5.17.8.2) is used to determine the time interval required to collect a predetermined weight of liquid. Filtered liquid of known physical properties and at a controlled temperature is transferred through the system to the receiver tank. When the weight of liquid in the receiver tank overcomes the force of the counterpoise weights on the weighbeam, the weighbeam rises and interrupts a light beam in a photoelectric timer circuit, automatically starting an electronic timer. Counterpoise weights are added to the weight tray in an amount equivalent to the weight desired to be collected in the receiver tank. The weighbeam then returns to the lower stop, allowing light to pass through to the photoelectric timer switch. The timer switch circuit is designed such that on the initial interruption of the light beam the electronic timer starts, and on the second interruption of the light beam the electronic timer is stopped. When the weight of liquid in the receiver tank overcomes the forces of the counterpoise weights on the weighbeam, the weighbeam rises, interrupting the light beam in the photoelectric timer circuit and automatically stopping the electronic timer. The weight flow rate is then determined by dividing the weight of the counterpoise weights by the time interval indicated on the electronic timer. The time required for the weighbeam to travel from the lower to upper stops increases as weight on the platform scales increases, introducing an error in the time measurement. This error can be reduced, however, by locating the photoelectric timer switch such that the light beam is interrupted on the first movement of the weighbeam. In Reference 26-50, a mathematical method of predicting the time required for the weighbeam to travel the distance between the stops is presented in terms of the mass flow rate, the collected weight, the length of weighbeam balance arm, and the acceleration of gravity. The author of Reference 28-5 has successfully employed the dynamic weighing calibrator technique to calibration of cryogenic flowmeters, which uses auxiliary refrigeration and pressurization equipment to reduce evaporation losses of the calibrating liquid.

5.17.8.3 TIME-VOLUME CALIBRATOR. The time-volume calibrator (Figure 5.17.8.3) consists of a calibration tank, a flow straightener, pressure and temperature sensors, a flow control valve, and a receiver tank. Calibration of a flowmeter is performed by pressurizing the expulsion tank

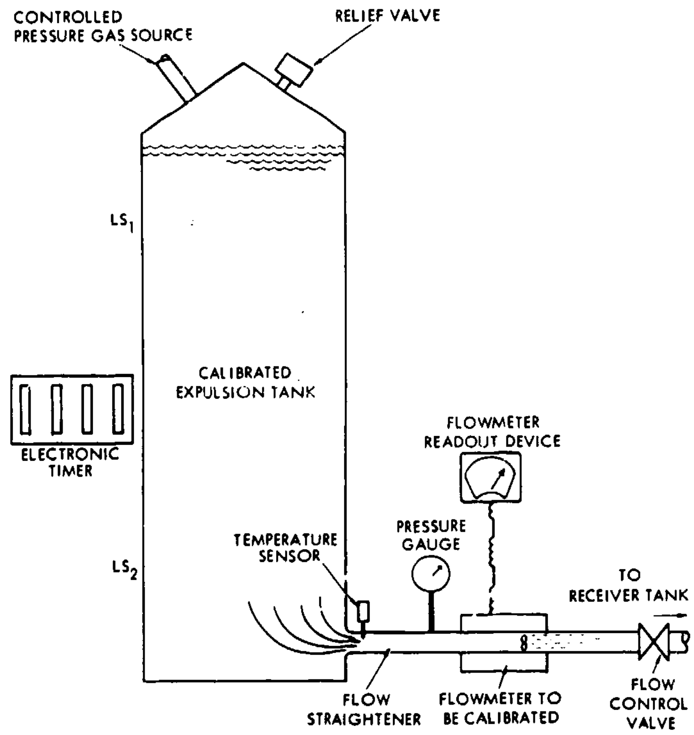


Figure 5.17.8.3. Time-Volume Flowmeter Calibrator

and regulating the flow control valve until the desired flow rate (as indicated by the flowmeter to be calibrated) is established and stabilized. Stabilized flow continues as level switch LS-1 is actuated by the falling liquid surface, starting a timing device. Likewise, level switch LS-2 is actuated, stopping the timing device. The calibrated volume between LS-1 and LS-2 divided by the indicated time yields the average volumetric flow rate, to be compared with the volumetric flowmeter readout. The volume between LS-1 and LS-2 is determined by weighing the quantity of water required to actuate the level switches, and applying temperature-density corrections for converting to volume at the calibrated temperature. Due to thermal contraction or expansion in calibrating cryogenic or high temperature fluids, the volume between LS-1 and LS-2 must be re-determined at the extreme temperature using a method similar to the water calibration.

5.17.8.4 **METER PROVER CALIBRATOR.** The meter prover consists of a length of select stainless steel tubing and a fluid-driven piston which forms a tight seal against the inside tube wall. Liquid flow drives the piston the length of the meter prover, interrupting two switches in transit. The two switches, which are separated by a known calibrated volume, start and stop an electronic timer, thus measuring the time to displace a known volume of liquid.

A typical bidirectional meter prover system is shown in Figure 5.17.8.4. The maximum design velocity for a bi-directional system is 10 ft/sec, and for a unidirectional system is 2.5 ft/sec.

REFERENCES

Twin Turbine
2-5, 52-24, 160-32, V-73

Turbine-Constant Speed Impeller
14-4, 19-74, 26-13, 26-101, 27-15, 52-24, 68-15, 160-32, 223-1, V-261

Constant Torque Impeller
V-96

Gyroscopic
19-74, 27-15, 52-24, V-29

Coriolis
2-13, 10-2, 19-74, 19-178, 26-17, 26-83, 52-24, 160-32

Magnus
19-74, 26-83, 52-24, 165-1

Thermopile
10-2, 19-74, 26-17, 68-1, 225-2, 249-1, V-109

Boundary Layer
10-2, 19-74, 27-14, 68-1, 74-7

Variable Head Flowmeters with Densitometers
27-15, 52-24, 52-37, 74-3

Velocity Meters with Densitometers
52-24

Variable Head Flowmeters with Velocity Compensation
52-24

Thin Plate Orifice
2-13, 10-2, 19-28, 19-74, 19-79, 26-42, 26-52, 26-91, 26-119, 26-134, 26-135, 26-146, 26-148, 26-204, 28-6, 49-12, 52-1, 52-28, 60-11, 68-1, 68-5, 7 -12, 73-17, 73-75, 74-4, 74-28, 110-15, 112-11, 160-16, 160-23, 160-32, 160-33, 165-21, 193-15

Flow Nozzles
10-2, 19-28, 19-74, 26-9, 26-91, 26-159, 49-12, 52-28, 60-11, 68-1, 73-17, 73-75, 74-4, 112-11, 160-32

Venturi
2-13, 10-2, 19-74, 26-19, 26-91, 26-115, 26-123, 28-1, 52-28, 60-2, 68-1, 73-17, 73-75, 74-4, 92-1, 112-11, 160-32

Dall Flow Tube
26-92, 74-1, 167-5, 206-3

Twin Throat Venturi
26-55

Capillary
19-74, 68-1, 74-5, 160-76

Porous Plug
26-88, 26-142, 68-1, 74-9, 163-1, 166-2, V-194

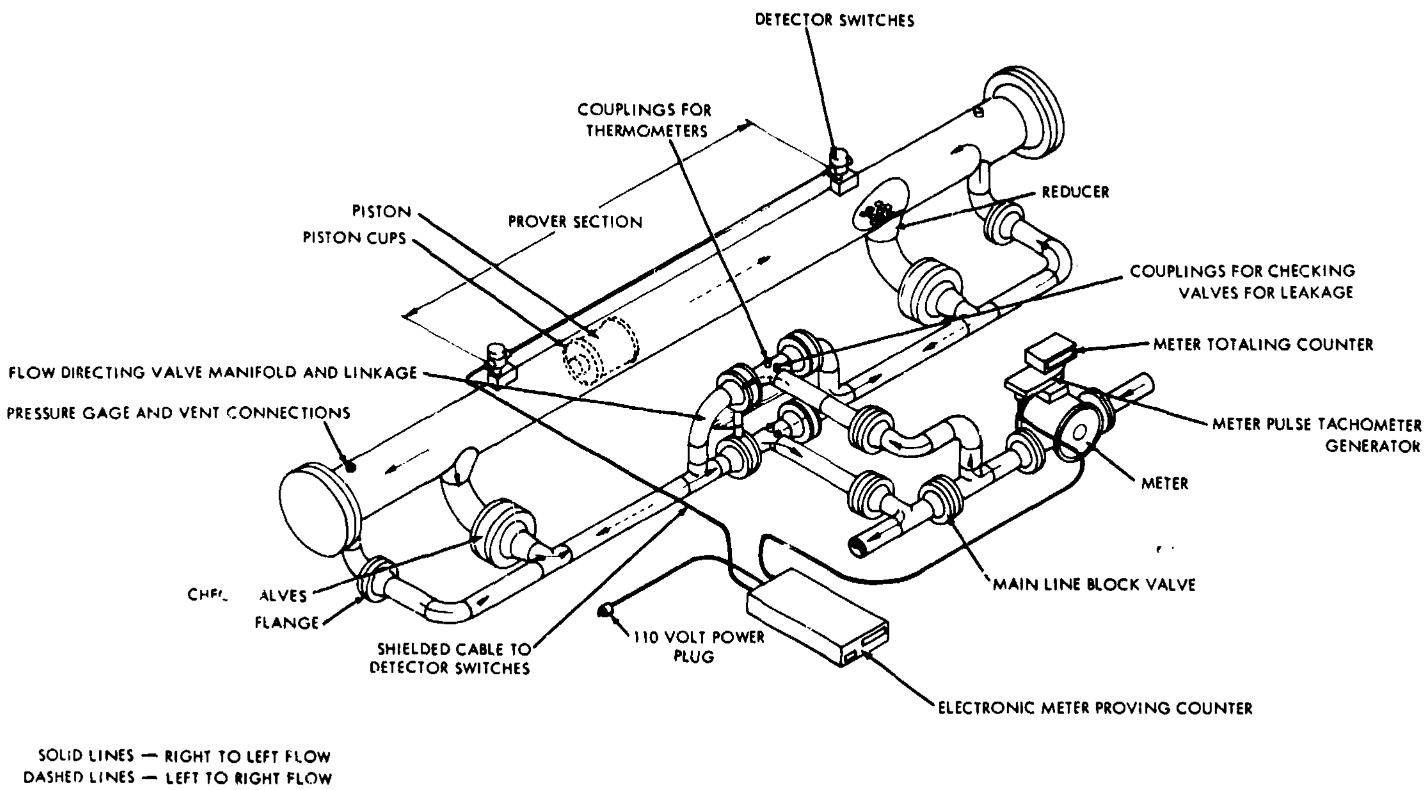


Figure 5.17.8.4. A Meter Prover System
(Courtesy of J. A. Halpine and Son, Inc., Tulsa, Oklahoma)

Pitot Tube
2-13, 10-2, 26-96, 26-99, 26-117, 68-1, 74-6, 160-32

Tapered Tube and Float
2-13, 6-103, 10-2, 19-74, 26 [illegible] 52-28, 61-12, 68-1, 74-1, 74-10, 74-29, 160-32, 160-3 [illegible] 7-1, V-204, V-228

Slotted Cylinder and Piston
19-74, 68-1

Turbine
2-5, 2-13, 6-103, 10-2, 19-74, 27-15, 52-24, 52-28, 52-43, 74-1, 74-3, 92-5, 160-32, V-73, V-96, V-228

Deflection of Sound Beam
52-7, 52-24, 52-28, 64-17, 203-1, 238-1, V-236

Acoustic Velocity
10-2, 19-74, 27-15, 52-24, 52-28, 52-51, 61-1, 68-1, 83-7, 83-8, 160-22, 160-32, 238-1, 290-2

Electromagnetic
2-13, 6-103, 10-2, 19-74, 26-17, 26-70, 26-82, 52-24, 52-28, 64-15, 64-16, 64-17, 68-1, 74-1, 117-11, 160-32, 165-26, 238-4, 300-4, V-40, V-228

Radioisotope
10-2

Drag Force
V-258

Nutating Disc
10-2, 19-74, 58-1, V-148

Rotary Vane
10-2, 19-74

Lobed Impeller
10-2, 68-1, V-220

Reciprocating Piston
10-2, 19-74, 27-4, 68-1

Axial Piston Metering Pump
V-95

Mass Spectrometer
V-222

Ultrasonic Translator
70-2, V-237

Water Displacement
70-2

Variable Area
V-228

Static Weighing Calibrator
26-50

Dynamic Weighing Calibrator
26-50, 28-4, 28-5

FLOWMETERS

SYMBOLS, UNITS, AND DIMENSIONS

SYMBOLS	QUANTITY	UNITS	DIMENSIONS	
			M, L, t, T	F, L, t, T, M, H
A	FLOW AREA ORTHOGONAL TO FLOW PATH	FT^2	L^2	L^2
A_h	INTERNAL AREA OF HOUSING CROSS-SECTION	FT^2	L^2	L^2
A_r	MAXIMUM CROSS-SECTIONAL AREA OF ROTOR NORMAL TO FLOW	FT^2	L^2	L^2
A_T	AREA THROUGH ANNULAR ORIFICE	FT^2	L^2	L^2
B	MAGNETIC FLUX DENSITY	GAUSS		
C	MODIFIED DISCHARGE COEFFICIENT	DIMENSIONLESS		
c_p	SPECIFIC HEAT AT CONSTANT PRESSURE	BTU/LB °F	$L^2 t^{-2} T^{-1}$	$HM^{-1}T^{-1}$
c_v	SPECIFIC HEAT AT CONSTANT VOLUME	BTU/LB °F	$L^2 t^{-2} T^{-1}$	$HM^{-1}T^{-1}$
d	TUBE DIAMETER	FT	L	L
d ( )	DIFFERENTIAL	DIMENSIONLESS		
d_c	CAPILLARY DIAMETER	FT	L	L
d_s	DISTANCE BETWEEN ULTRASONIC TRANSMITTER AND RECEIVER	FT	L	L
D_1	DIAMETER AT POINT 1	FT	L	L
D_0	DIAMETER AT THROAT	FT	L	L
dm	MASS OF ELEMENT $\rho A dr$	LB_m	M	M
dr	LENGTH OF ELEMENT Adr	FT	L	L
E	ELECTRICAL POTENTIAL	MILLIVOLTS		
E_D	ELECTRICAL POTENTIAL OUTPUT FROM DOWNSTREAM RECEIVER	MILLIVOLTS		
E_U	ELECTRICAL POTENTIAL OUTPUT FROM UPSTREAM RECEIVER	MILLIVOLTS		
F	FORCE	LB_f	ML/t^2	F
f	FRICTION FACTOR $64/N_R$ (LAMINAR)	DIMENSIONLESS		
F_a	LARGEST FLOAT CROSS-SECTIONAL AREA NORMAL TO FLOW	FT^2	L^2	L^2
f_d	DOWNSTREAM REPETITION FREQUENCY	CPS	$1/t$	$1/t$
f_u	UPSTREAM REPETITION FREQUENCY	CPS	$1/t$	$1/t$
F_v	FLOAT VOLUME	FT^3	L^3	L^3
g	ACCELERATION OF GRAVITY	FT/SEC^2	L/t^2	L/t^2
g_c	CONVERSION CONSTANT = 32.2	$LB_m FT/LB_f SEC^2$		ML/Ft^2
I	MOMENT OF INERTIA	$LB_m\ FT^2$	ML^2	ML^2
I_s	POLAR MOMENT OF INERTIA ABOUT SPIN AXIS	$LB_m\ FT^2$	ML^2	ML^2
k	THERMAL CONDUCTIVITY	$BTU/HR\ FT^2$	ML/t^3T	H/tLT
k	RADIUS OF GYRATION	FT	L	L
$K, K_o, K_1, K_2\ K_n$	CONSTANTS			
L	LENGTH OR LENGTH OF FLOW PATH	FT	L	L
L	TORQUE	$FT\ LB_f$	ML^2/t^2	FL
L	LEAKAGE RATE	ATM. CC/SEC	L^3/t	L^3/t
L_v	GYROSCOPE REACTION TORQUE ABOUT VERTICAL AXIS	$FT\ LB_f$	ML^2/t^2	FL
M	MASS	LB_m	M	M

SYMBOLS, UNITS, AND DIMENSIONS

SYMBOLS	QUANTITY	UNITS	DIMENSIONS	
			M, L, t, T	F, L, t, T, M, H
M_a	ANGULAR MOMENTUM	$LB_m\ FT^2/SEC$	ML^2/t	ML^2/t
$\dot{m}$	MASS FLOW RATE	LB_m /SEC	M/t	M/t
n	NUMBER OF PANEL METER DIVISIONS	DIMENSIONLESS		
Re	REYNOLDS' NUMBER $VD\rho/\mu_m$	DIMENSIONLESS		
P_1	PRESSURE AT POINT 1 (ABSOLUTE)	LB_f/FT^2	M/Lt^2	F/L^2
P_2	PRESSURE AT POINT 2 (ABSOLUTE)	LB_f/FT^2	M/Lt^2	F/L^2
q	HEAT FLOW RATE	BTU/HR	ML^2/t^3	H/t
Q	VOLUMETRIC FLOW RATE	FT^3/MIN	L^3/t	L^3/t
R	AVERAGE RADIUS OF BLADE CENTER OF PRESSURE	FT	L	L
r	RADIUS FROM SPIN AXIS TO CENTERLINE OF FLOW PATH	FT	L	L
r_1	RADIUS TO POINT 1	FT	L	L
r_2	RADIUS TO POINT 2	FT	L	L
S	SENSITIVITY OF LEAKAGE FLOWMETER	ATM. CC/SEC/DIV	L^3/t	L^3/t
S	SPECIFIC SURFACE, (SURFACE AREA EXPOSED TO FLUID/TOTAL VOLUME OF SOLID MATTER IN POROUS PLUG)	1/FT	1/L	1/L
t	TIME	SEC	t	t
T	TEMPERATURE, FAHRENHEIT	°F	T	T
V	FLUID VELOCITY	FT/SEC	L/t	L/t
V_r	VELOCITY OF THE CIRCULATING FLUID STREAM	FT/SEC	L/t	L/t
V_S	FLUID SONIC VELOCITY	FT/SEC	L/t	L/t
V_1	FLUID VELOCITY AT POINT 1	FT/SEC	L/t	L/t
V_2	FLUID VELOCITY AT POINT 2	FT/SEC	L/t	L/t
w_f	SPECIFIC WEIGHT OF MATERIAL IN POROUS PLUG	LB_f/FT^3	M/t^2L^2	F/L^3
w_p	SPECIFIC WEIGHT OF POROUS PLUG	LB_f/FT^3	M/t^2L^2	F/L^3
w_1	SPECIFIC WEIGHT OF FLUID AT POINT 1	LB_f/FT^3	M/t^2L^2	F/L^3
α	CORIOLIS ACCELERATION	RADIANS/SEC^2	$1/t^2$	$1/t^2$
γ	c_p/c_v	DIMENSIONLESS		
Δ	INCREMENT	DIMENSIONLESS		
θ	BLADE ANGLE, DEGREES	DIMENSIONLESS		
θ_s	ANGLE BETWEEN ULTRASONIC BEAM DIRECTION AND FLOW DIRECTION, DEGREES	DIMENSIONLESS		
μ	ABSOLUTE VISCOSITY	$LB_f SEC/FT^2$	M/Lt	Ft/L^2
ρ	DENSITY	LB_m/FT^3	M/L^3	M/L^3
ρ_f	FLOAT DENSITY	LB_m/FT^3	M/L^3	M/L^3
ρ_L	LIQUID DENSITY	LB_m/FT^3	M/L^3	M/L^3
ϕ	PHASE ANGLE, DEGREES	DIMENSIONLESS		
ω	ANGULAR VELOCITY	RADIANS/SEC	1/t	1/t
ω_s	ANGULAR VELOCITY ABOUT SPIN AXIS	RADIANS/SEC	1/t	1/t
Ω_h	PRECESSIONAL ANGULAR VELOCITY ABOUT HORIZONTAL AXIS	RADIANS/SEC	1/t	1/t

B

Time-Volume Calibrator
26-50

Meter Prover Calibrator
V-260

5.18 GLOSSARY OF FLUID COMPONENT TERMINOLOGY

Actuator: a device which converts mechanical, fluid, or electrical energy into rotary or linear mechanical motion of a valve stem.

Balanced Valve: a valve having no net pressure force acting on the valving element.

Ball Valve: a spherical plug valve having removable seats.

Blade Valve: a valve having a sliding flat plate closure element that moves perpendicular to the flow stream, cutting through the fluid to accomplish shutoff. The blade is usually designed to seal on one side, with fluid pressure providing all or part of the sealing force.

Block Valve: (see Shutoff Valve)

Bonnet: the upper part or cap of a valve through which the stem projects. The bonnet usually contains the stem guide and stem seal or packing, and provides the mean or supporting the valve actuator.

Butterfly: a disc or vane mounted on a rotating shaft. It serves as the closure element for a butterfly valve.

Check Valve: a valve that permits flow to proceed in one direction only, checking reverse flow.

Closure: the valving element and its associated seat or, in the case of seatless valve, the mating port in the valve housing.

Closure Element: (see Valving Element)

Closure Member: (see Valving Element)

Cock: a simple conical plug valve having no removable seals or packing. Common designations for cocks are drain cock, stop cock, and pet cock.

Control Valve: a valve which regulates or otherwise controls volumetric flow rate of a fluid by means of a variable area flow restriction. A control valve has an infinite number of operating positions, in contrast to a shutoff valve which is either fully open or fully closed. The term *control valve* is used synonymously with *throttle valve*. Small manual control valves are also called *metering valves*.

Control Element: (see Valving Element)

Controller: a device which converts an input signal from the controlled variable (temperature, level, pressure or flow rate) to a valve actuator input (pneumatic, hydraulic, electrical, or mechanical) to vary the valve flow rate in response to the required correction of the controlled variable.

Directional Control Valve: any valve which controls the direction of flow between one or more fluid lines.

Explosive Valve: a valve having a small explosive charge which provides a source of high pressure gas used to position an actuator. Explosive valves are also called squib valves, squib referring to the device containing the explosive charge.

Flapper: a closure element which consists of a flat plate or disc having a hinge displaced from, and in the plane of, the sealing surface. A flapper is a common closure in check valves. The terms *flapper* and *swinging gate* are often used synonymously.

Flow Coefficient (C_v): the volume of water in U. S. gallons that will flow per minute through a valve body, with the pressure drop of 1 psi across the valve when it is in the fully open position.

Gate: (see Gate Valve)

Gate Valve: a valve having a closure member, called a gate, which moves perpendicular to the flow stream, cutting through the fluid and wedging between two annular seats to accomplish closure. With few exceptions, gate valves are double seated, both sides of the gate acting as sealing surfaces. The gate may be either solid or split.

Globe Body (pattern): a poppet valve body having coincident or parallel inlet and outlet ports, with the closure traveling perpendicular to the centerline of these ports.

Globe Valve: any valve having a poppet type closure, that is, a closure which travels perpendicular to a plane through the seating surface. A globe valve can have a variety of body shapes such as angle, Y, inline, or globe body pattern.

Globe Pattern Valve: a valve having a globe body (see Globe Body).

Headroom: the vertical space requirements of a valve and its actuator used in a horizontal line.

Inline Poppet Valve: a poppet valve with the centerline of the control port coincident with the centerline of the inlet and outlet ports of the valve.

Lands and Grooves: Lands are the annular areas between circumferential grooves or recesses in a spool piston control element. Flow metering is accomplished by land edges.

Linear Slide Valve: a valve consisting of a ported sliding plate held either closely against one seating surface, or between two seating surfaces having ports mating with the slide ports.

Loss Producing Element: (see Valving Element)

Metering Valve: a term used in conjunction with regulators and relief valves to mean the combined valve seat, valving element, and actuator. This term is also used to describe small control valves.

Metering Element: (see Valving Element)

Multiple Passage Valve: a directional control valve which stops, starts, and diverts flow between three or more alternate flow paths.

Needle Valve: a poppet (globe valve) having a sharply tapered plug and a narrow seating area such that relatively low stem loads provide high seating stresses.

Non-Rising Stem: a stem design in which the valve stem does not rise, but merely turns with the actuator, raising and lowering the valving element. Most non-rising stems consist of an externally threaded lower stem secured so it cannot turn, and an internally threaded hollow upper stem into which the lower section is drawn or telescoped as the upper section is turned. Non-rising stems may be either the internal screw type or the external screw type.

Normal Position: that position of a valve assumed when there is no signal for operating force applied to the valve stem.

Normally Closed Position: a valve closure which is closed in the normal position.

Normally Open Position: a valve closure which is open in the normal position.

Normally Closed Valve: any powered valve which returns to a closed position upon shutoff or failure of the actuating energy signal.

Normally Open Valve: any powered valve which returns to an open position upon shutoff or failure of the actuating energy signal.

Packless Valve: a valve that utilizes a diaphragm or bellows stem seal in place of the customary packing gland to prevent fluid leakage around the stem.

Pilot Valve: an auxiliary or relay valve which permits the use of low energy circuits for the control of high energy systems.

Pintle: a long tapered plug used in a globe-type valve.

Piston Valve: a valve having a piston-type closure which moves in a cylinder, controlling flow by covering or uncovering ports located circumferentially around the cylinder. Cylinder ports may be round, diamond, or wedge-shaped. The piston may be spring-loaded, pressure-actuated, or connected to a conventional rising stem. Although throttling is accomplished by a shearing action, piston valves can be designed with a seat for positive sealing.

Plug: a poppet-type valving element having a contoured control surface which controls flow by a variable restriction between the seat or seat ring. There are many forms and shapes of valve plugs, each of which provides a different flow characteristic through the valve, as described in Sub-Topic 5.3.3. Three of the most common types of plugs are the parabolic plug, the rectangular port, and the V-port plug. The parabolic plug is a paraboloid which has a linear flow characteristic; however, the term parabolic plug is often used to describe any plug having the general appearance of a paraboloid covering a wide range of flow characteristics. The rectangular and V-port plugs are cylinders having either rectangular or V patterns cut from the lower edge. The rectangular plug provides a linear flow characteristic, while a straight sided, V-port plug provides a parabolic flow characteristic.

Plug, Rotary: a ported closure member which rotates on an axis perpendicular to the axis of flow. Flow is controlled by changing the relationship of the closure port with respect to the valve body ports. Rotary plugs can be cylindrical, conical, or spherical.

Plug Valve: a seatless valve having a rotating cylindrical, conical, or spherical closure member called the plug (see Plug, Rotary), providing unobstructed flow between two or more inlet and outlet connections. Flow control is accomplished by a shearing action of the plug past stationary ports in the valve body. In a two-way valve, 90° rotation of the plug moves the valve from the full open to full closed position. A spherical plug valve is more commonly called a ball valve. A cock is a simplified version of a plug valve which has no removable seals or packing.

Pop Valve: (see Relief Valve)

Poppet: a valving element which travels perpendicular to a plane through the sealing surface. A poppet can be a flat disc, a sphere, a cone, or any surface of revolution. The term *poppet* is often used interchangeably with *plug*. The distinction is sometimes made where the plug achieves flow control by being inserted into the control port, in contrast to the poppet which is located externally to the port. This distinction, however, is not clearly defined.

Poppet Valve: any valve in which the closure travels perpendicular to a plane through the seating surface. The term *poppet valve* is used synonymously with *globe valve*, a globe valve in its broadest sense being any valve using a poppet-type closure (see Poppet).

Positioner: a device used on control valves to impart sensitivity to the valve in obtaining the positioning required by the control signal. A positioner consists of an actuator coupled with the feedback loop between the control signal to the valve and the valve stem position.

Pressure-Reducing Valve: (see Pressure Regulator)

Pressure Regulator: a control valve that uses no auxiliary source of power during operation. The term *pressure regulator* is used synonymously with *self-operated controller* and *pressure reducing valve*.

Regulator: (see Pressure Regulator)

Relief Valve: a pressure relieving device which opens automatically when a pre-determined pressure is reached. Relief valves opening rapidly to full flow are generally referred to as safety valves or pop valves.

Rising Stem: a valve stem so guided that the external portion to which the actuator is secured rises and descends as the closure opens and closes the valve to fluid flow.

Rotary Slide Valve: a valve having a closure consisting of a ported rotor or disc which is held against a flat seating surface having mating ports. Closure is accomplished by rotating the disc until the flow port or ports are blocked.

Safety Valve: (see Relief Valve)

Seat: an annular portion of the valve body (integral) or an insert (removable) against which the valving element seats to regulate or prevent flow. Although a valve seat can also serve as a seal, a distinction should be made between the seat and the sealing device which is commonly inserted, either in the seat or in the valving element, for controlling leakage in the closed position.

Seat Ring: an integral or replaceable ring in the valve body which provides an opening that may be partially or completely obstructed by the movable valving element for purposes of flow regulation. The seat ring is sometimes referred to as the valve port.

Seatless Valve: a valve having a grooved or ported closure member which slides or rotates against the valve body to open and close fluid flow passages through a shearing rather than a seating action.

Self-Operated Controller: (see Pressure Regulator)

Selector Valve: a valve whose function is to direct fluid flow from one input port to one of two or more selected output ports.

Sequence Valve: a valve whose primary function is to direct flow in a predetermined sequence between two or more ports. A *shuttle valve* is a type of sequence valve which is pressure actuated such that when a preset system pressure has been reached, the valve automatically actuates, connecting two or more flow paths.

Shutoff Valve: a valve which is normally used in only two positions—either closed or full open—in contrast to control valves which take intermediate positions to modulate flow between full flow and no flow. The term *shutoff valve* is used synonymously with *stop valve* and *block valve*.

Shuttle Valve: (see Sequence Valve)

Sleeve Valve: a valve which controls flow by the shearing action of a sleeve sliding over control ports. A sleeve valve can be provided with a seat for tight sealing.

Slide Valve: any valve which accomplishes closure by sliding a ported member over a seating surface having matching ports (see Gate Valves, Blade Valves, and Rotary Slide Valves).

Solenoid Valve: a poppet, spool, or piston valve actuated by an integrally mounted solenoid actuator.

Spool: a solid, elongated valve closure member with annular rings forming lands and grooves. The spool moves with a sliding, reciprocating action within the valve body.

Spool Valve: a seatless valve which controls flow by covering and uncovering annular ports with lands on a sliding spool. Spool valves can have sliding seals (packed) or depend solely on close tolerances for sealing (packless). A spool is inherently a pressure balanced closure member.

Squib Valve: (see Explosive Valve)

Stem: that part of the valve which is attached to or acts upon the closure to lift, rotate, or otherwise position the closure to control fluid flow.

Stem Guide: a cylindrical bearing area within the valve body or bonnet supporting the portion of the valve stem between the actuator and closure member.

Stop Valve: (see Shutoff Valve)

Stuffing Box: a cylindrical recess provided within the valve bonnet at the point where the stem valve emerges, into which a packing may be compressed by a gland to provide a pressure tight seal.

Swinging Gate: (see Flapper)

Throttle Valve: (see Control Valve)

Valve: a device for controlling flow rate, flow direction, or pressure of a fluid in a pressure system. A valve consists of a valving element, a housing, and an actuator. A distinction is sometimes made between a complete valve and a basic valve, the basic valve being a complete valve less the actuator.

Valve Port: an opening through which fluid passes within a valve body. It includes both internal and external openings.

Valve Positioner: (see Positioner)

Valve Stem: that part of the valve connected to the actuator which acts to lift, rotate, or otherwise position the member which controls fluid flow.

Valve Trim: internal elements in contact with the fluid which accomplish flow control. The trim includes the valving element and seat.

Valving Element: the moving element of a valve which controls flow either by its proximity to a seat or its relationship to a closure port, as in the case of a seatless valve. The following terms are used synonymously with valving element: control element or member, closure element or member, metering element, and loss producing element.

Valving Unit: the combination of valving element plus seat or mating closure port(s).

Vent Valve: a pressure-relieving shutoff valve which is externally operated, as contrasted to a relief valve which opens automatically.

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SUPERSEDES: OCTOBER 1965

6.1 INTRODUCTION

The design of fluid components consists, to a large extent, of the integration of the basic elements or modules which must be selected and/or designed by the fluid component designer. The purpose of this section is to present the designer with fluid component module selection criteria, design formulae, and design data. The fluid component modules included in this section are valving units, static and dynamic seals, springs, bearings, bellows, diaphragms, and actuators.

6.2 VALVING UNITS

6.2.1 Introduction

A *valving unit*, also referred to as a closure or inner valve, is the combination of a valving element and its mating seat or flow port(s). The valving element is movable and controls flow resistance or direction while the seat or flow port is fixed to, or is a part of, the valve housing.

Valving units are commonly characterized by the valving element configuration, with the type of motion added, in some instances, for clarification. Valving units discussed in this sub-section are:

Butterfly	Sleeve
Ball	Spool
Rotary plug	Piston
Gate	Poppet
Blade	Flexible valving element
Linear slide	Unconventional concepts
Rotary slide	

A valving unit may be required to perform three functions:

1) Shutoff — effecting a seal to prevent fluid flow.
2) Flow modulation — controlling fluid flow rate by controlling resistance to fluid flow, usually by opening or closing an aperture.

3) Flow diversion — controlling direction of fluid flow by diverting flow from one passage to another.

The manner in which sealing is accomplished varies greatly with the valving unit configuration. Valves may have hard seals, soft seals, or no seals (packless). In poppet valves, sealing is obtained by moving the poppet against its seat; in spool, rotary plug, and ball valves, sealing is achieved by blocking the flow port through a sliding or shearing motion of the valving element. Valves which utilize sliding valving elements such as spools, with leakage controlled by minimizing clearances between the valving element (spool) and its bearing surfaces (sleeve or housing), are generally referred to as packless valves.

Flow modulation can be achieved with any valving element, but flexibility in varying flow characteristics is restricted to relatively few designs, such as the poppet, gate, blade, piston, spool, and sleeve. Various throttling characteristics are achieved by contouring the poppet, blade, or gate, or by changing the configuration of the port uncovered by the spool, piston, or sleeve.

The function of flow diversion is usually performed through valve porting patterns. A typical example is a multiple-land spool controlling flow between three or four ports in the valve housing. Rather specialized designs for flow diversion use multiple passage valving units, typified in Figures 5.8.5.2a and b.

Table 6.2.1 lists sub-topics and detailed topics located elsewhere in the handbook where specific valving unit designs are discussed with regard to function. Blank spaces in this table reflect only a lack of specific examples and do not necessarily indicate that a particular valving unit is not applicable to a given function.

Due to a lack of available documented design data, much of valving unit design is an empirical process involving past experience, design ingenuity, and a "cut-and-try" development procedure. The design procedure commonly involves a minimum of analysis, limited primarily to static force and flow capacity, combined with designer-company experience in producing a prototype which is then modified through development testing to meet design requirements. In many instances hundreds of designs or modifications of existing designs are tried before a final design is established. This empirical approach will undoubtedly continue until such time as the phenomena of valve seating and flow control are fully understood and reduced to analytical terms and or suitably generalized parametric data. It is the purpose of this sub-section to (1) focus the attention of the designer on factors which influence valving unit performance, (2) direct the reader to those sections of the handbook where these influencing factors are discussed at length, and (3) include either available generalized data for valving unit design or data pertaining to representative examples of valving unit design.

Table 6.2.1. Valving Unit Functions

Valving Unit	Shutoff	Flow Modulation	Flow Diversion
Ball	3.8.4.2 5.2.3.1	-	5.8.4.1
Blade	5.2.7.1	-	5.8.4.3
Butterfly	3.8.4.2 5.2.4.1 5.5.6.4	5.3.5.5	-
Flexible Element	3.8.4.2 5.2.9	5.3.5.4 5.4.15	-
Gate	3.8.4.2 5.2.7.1	-	-
Piston	5.2.10.4 5.5.6.4 5.5.7.2	5.3.5.6 5.4.8.3	-
Poppet	3.8.4.2 5.2.5.1 5.2.6.1 5.5.6 5.5.18 5.7.3 5.7.4 5.7.5 5.7.6 5.8.4.2 5.9.5 5.11.4.2 5.11.4.3	3.8.4.2 5.3.3 5.3.5 5.4.4 5.4.5 5.4.6 5.4.7	5.8.4.2
Rotary Plug	5.2.11	5.3.5.7	5.8.5.2
Rotary Slide	5.2.12	-	5.8.5.3
Sleeve	-	5.3.5.6 5.4.15	-
Spool	-	5.3.5.6 5.6.7.2	5.8.4.1

6.2.2 Design and Selection Factors

Many valve problems can be traced to simple design oversights involving factors of environment, materials, stress, and dynamics, and their various interrelationships. The important factors which must be considered in valving unit design and or selection are presented in the following detailed topics.

6.2.2.1 PRESSURE DROP AND FLOW CAPACITY. Pressure drop and flow capacity are essentially measures of the same parameter — resistance to fluid flow. Pressure drop is a measure of resistance to flow determined with a valve in a fluid system passing a given flow rate, whereas flow capacity is associated with the maximum flow rate which a valve will pass for a given pressure differential across the valve. As discussed in Sub-Sections 3.8 and 3.9, pressure drop and flow capacity are related by any of the following conventions:

Discharge Coefficient. Discharge coefficient, C_D, is an adaptation of the sharp-edged orifice equation for use with valves.

$$Q = C_D A_o \sqrt{2g \frac{\Delta P}{w}} \qquad \text{(Eq 3.8.1f)}$$

where

Q = volumetric flow rate, ft^3/sec
C_D = discharge coefficient — experimentally determined, dimensionless
A_o = throat area of the orifice, ft^2
g = gravitational constant, ft/sec^2
ΔP = static pressure differential across the loss-producing element, lb_f/ft^2
w = specific weight, lb_f/ft^3

Flow Coefficient. Flow coefficient, C_v, is an experimentally determined measure of the volumetric flow rate (in gallons per minute) of water at 60°F that will flow through a valve with a pressure drop (Δp) of 1 psi across the valve. For liquids

$$Q = C_v \sqrt{\frac{\Delta p}{S}} \qquad \text{(Eq 3.8.4.2b)}$$

where

Q = volumetric flow rate, gpm
C_v = flow coefficient, gpm $\sqrt{lb_f/in^2}$
Δp = static pressure differential across the loss-producing element, lb_f/in^2
S = specific gravity of the liquid, dimensionless

Equations (3.8.4.2e) and (3.8.4.2f) present more complex C_v expressions for use with gases and vapors, respectively.

Where weight flow rather than volumetric flow is of interest, as is frequently the case with rocket engine components, a similar coefficient, K_w, can be experimentally determined using water or any other convenient liquid. Once the K_w factor for a given component is determined, the weight flow rate, $\dot{w}$, of any liquid through that component is

$$\dot{w} = K_w \sqrt{\Delta p\, S} \qquad \text{(Eq 6.2.2.1)}$$

where

$\dot{w}$ = weight flow rate, lb_f/sec
K_w = K_w factor, $\sqrt{lb_f\, in^2}/sec$
Δp = static pressure differential across the loss-producing element, lb_f/in^2
S = specific gravity of the liquid, dimensionless

Equivalent Orifice Area. Equivalent orifice area, A_o, is the area of an equivalent sharp-edged orifice (discharge coefficient, C_D = 0.60) which gives the same pressure drop/flow relation as the flow element in question. The orifice equation is then expressed

$$Q = 0.60\, A_o \sqrt{2g \frac{\Delta P}{w}} \qquad \text{(Eq 3.8.4.1a)}$$

where

Q = flow rate, ft^3/sec
A_o = equivalent orifice area, ft^2 (C_D = 0.60)
ΔP = pressure drop across valve, measured between points where upstream and downstream diameters are equal, lb_f/ft^2
w = specific weight, lb_f/ft^3

Resistance Coefficient. Resistance coefficient, K, is an empirical coefficient which relates pressure drop through a flow device to velocity head

$$\Delta P = Kw \frac{V^2}{2g} \qquad \text{(Eq 3.9.5.2a)}$$

where

ΔP = total pressure drop across the loss-producing element, lb/ft^2
K = resistance coefficient, dimensionless
w = specific weight, lb_f/ft^3
V = fluid velocity in pipe of nominal size of valve, ft/sec

Table 3.9.5.2 gives values of K for certain valves having various valving unit configurations. Resistance coefficient is also related to equivalent length of pipe by the following expression

$$L_e = \frac{L}{D} = \frac{K}{f_D} \qquad \text{(Eq 3.9.5.1a)}$$

where

L_e = equivalent length in pipe diameters
L = pipe length
D = internal diameter of pipe
f_D = Darcy friction factor

Table 3.9.5.1 presents equivalent lengths for certain valves having various valving unit configurations.

NBS Flow Factor. NBS flow factor, F, is a measure of gas flow through valves and orifices and is defined as the ratio of air flow in SCFM to the upstream pressure in psia when p_2/p_1 = 0.5, or

$$F = \frac{Q\,(max)}{p_1} \qquad \text{(Eq 3.8.4.3a)}$$

where

F = flow factor, SCFM/psia

$Q(max)$ = air flow in SCFM at $P_2/P_1 = 0.5$

p_1 = upstream pressure, psia

p_2 = downstream pressure, psia

The NBS flow factor is used primarily with air at pressure ratios below the critical. Once F has been determined for a specific component, as described in Detailed Topic 3.8.4.3 and Reference 82-6, subsonic air flow rate may be obtained from the expression

$$Q = p_1 F \sqrt{\frac{4}{3}} \times \sqrt{1 - \left(\frac{p_2}{p_1}\right)^2} \quad \text{(Eq 3.8.4.3b)}$$

where

Q = air flow rate in SCFM at $p_2/p_1 \geq 0.5$

F = flow factor, SCFM/psia

p_1 = upstream pressure, psia

p_2 = downstream pressure, psia

Detailed Topic 3.8.4.3 discusses alternative flow equations using F, and includes conversion factors and equations for adapting the NBS flow factor for use with gases other than air.

Available data concerning pressure drop and flow capacity for valves having particular valving unit types are included in Sub-Topic 6.2.3. The optimization of a design for minimum pressure drop, or maximum flow capacity, is discussed in Detailed Topic 6.2.2.18. Leakage flow through a closed valving unit is discussed in Detailed Topic 6.2.2.3, and flow through partially closed valving units in Detailed Topic 6.2.2.2.

In general, when minimum pressure drop or maximum flow capacity are prime considerations in a particular valve application, valving units should be chosen which

a) Provide for a full-open flow passage devoid of any protrusion of the valving element (as with some ball, gate, and blade designs).

b) Provide a straight-through flow passage with no requirement that the fluid change direction, such as most sleeve and poppet designs demand.

c) Provide smooth walls throughout the flow passage by minimizing roughness at the seat interface, or by sealing or filling the cavity into which a gate or blade retracts. Most ball and rotary plug configurations and some linear and rotary slide configurations have been designed to achieve this.

d) Provide for constant cross-sectional area throughout the flow passage. If variation is necessary, the change in area should not be abrupt, but should be achieved as gradually as length limitations permit.

It is interesting to note that none of the valving unit types which potentially meet these requirements (ball, rotary plug, gate, and blade) are inherently pressure-balanced. Should size, pressure, or sealing considerations dictate some form of pressure-balanced valving unit such as a sleeve or poppet, full-open pressure drop may be minimized by

a) Avoiding reduction (or increase) in cross-section area through the valving unit, to eliminate contraction and expansion losses

b) Where such changes in area are necessary, making the transition as gradual as possible

c) Optimizing changes in flow direction, to minimize work done by the fluid (decrease in total fluid energy).

6.2.2.2 FLOW VERSUS STROKE. Beyond the complex flow regimes in the near-closed position, flow rate varies predictably with changes in flow resistance and changes in pressure differential. The valving element controls flow resistance by varying the area and direction of the flow path in a predictable manner as a function of stroke. The change in flow rate as a function of valving element position under conditions of constant temperature and constant pressure drop is called the inherent flow characteristic. Common flow characteristics are linear, equal percentage, and square root, as depicted in Figure 6.2.2.2a. Flow characteristics can be tailored by varying the contour of the valving element with practically any valving unit configuration; but certain types such as poppet (linear plug) valves are common choices for flow control applications because of the ease with which a wide range of flow characteristics may be obtained.

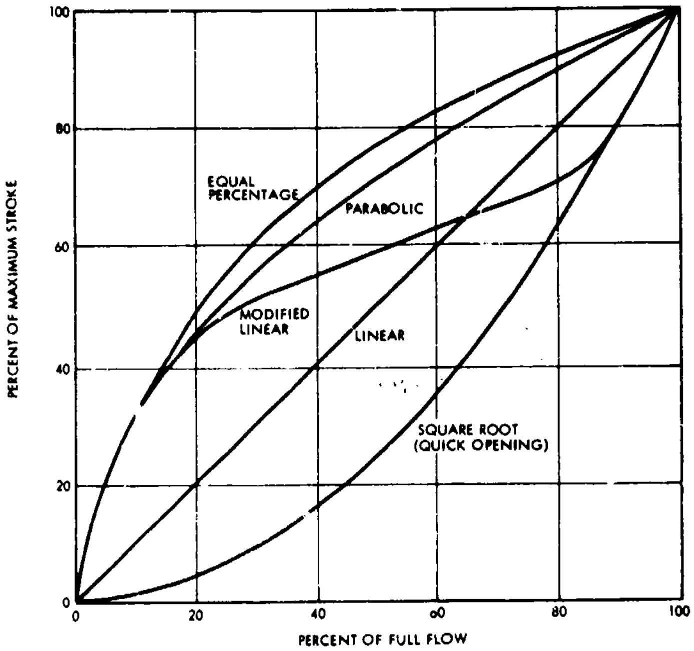


Figure 6.2.2.2a. A Comparison of Control Valve Flow Characteristics

With control valves in particular, the required flow characteristic may be expected to dictate the final valving unit configuration, within the constraints imposed by full-open pressure drop or flow capacity requirements on the one hand, and seating geometry requirements imposed by leakage requirements on the other. The means whereby flow characteristics of particular valving unit types may be tailored are discussed under Sub-Topic 6.2.3.

Reference 35-14 includes a comprehensive theoretical analysis (accompanied by experimental verification) of the flow through a flat seat poppet valving unit in the near-closed condition. Figure 6.2.2.2b shows the various theoretical flow regions and their transitions as calculated for the model poppet valving unit shown in Figure 6.2.2.2c. While this example is for one particular valving unit configuration with gaseous nitrogen, it is indicative of the complexity associated with flow-versus-stroke analysis of any valving unit in the near-closed range. Other leakage flow data may be found in Sub-Section 3.11 and Detailed Topics 6.2.3.11 and 6.3.2.3.

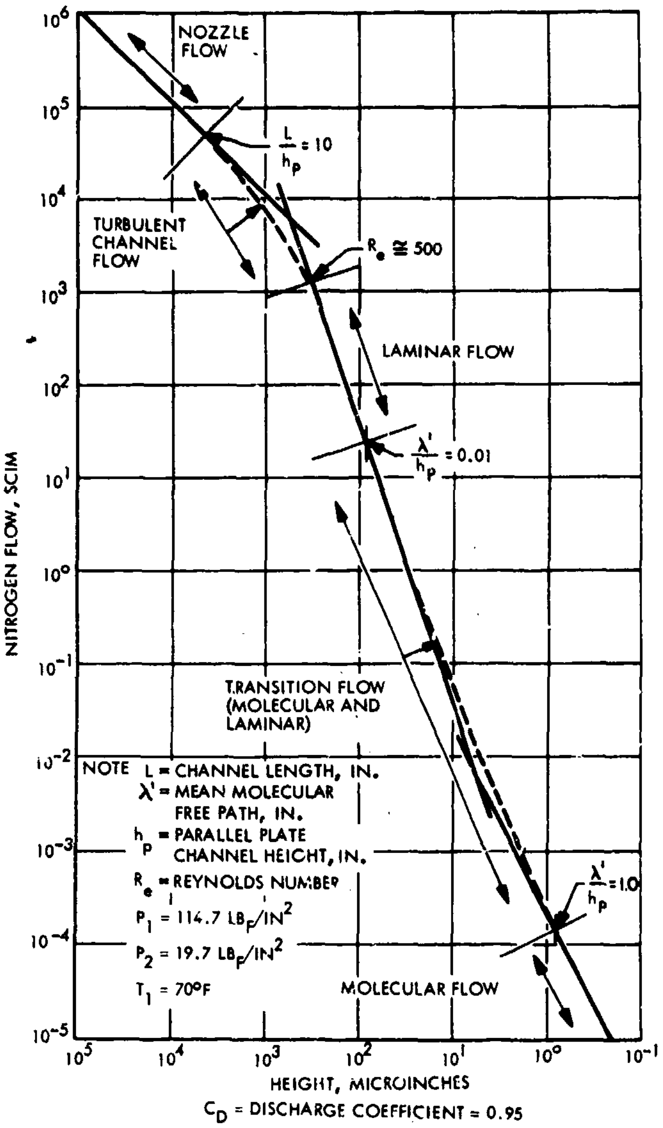


Figure 6.2.2.2b. Theoretical Nitrogen Flow Through Flat-Seat Poppet Valve Model Shown in Fig. 6.2.2.2c

(Reference 35-14)

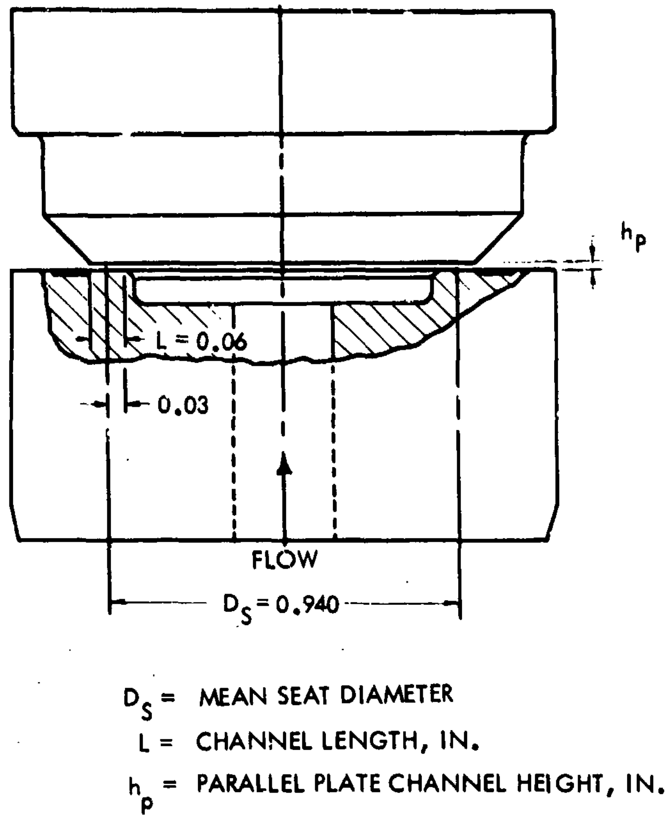


Figure 6.2.2.2c. One-Inch Poppet and Seat Model

(Reference 35-14)

6.2.2.3 LEAKAGE. Leakage requirements for a particular valve application may influence the choice of valve type and will almost without exception be the primary consideration in the design of the seating elements of the valving unit. A recent survey of aerospace valve manufacturers has shown that the technology most needed to advance the state-of-the-art is in the field of valve seating, with primary emphasis placed upon leakage. Leakage requirements, or leakage allowances in some instances, may differentiate between *primary leakage* (leakage past the valving unit, from upstream to downstream through the system) and *secondary leakage* (external leakage past the stem of a ball or butterfly-type valve, or leakage into an integral actuator in a sleeve or poppet-type valve). The following discussion of valving units considers only primary leakage.

With respect to sealing, valving units may be classified as packless (no seals), soft seal (elastomer or plastic), or hard seal (metal, ceramic, ceramet, or graphite). Packless valving units, usually pistons or spools, are used for flow diversion or control valve applications having generous leakage allowances (quiescent flow). Soft seals offer the advantage that low stress is required to effect a seal. Hard seals are used primarily for high temperatures and pressures and for sealing highly corrosive fluids. The cost of a hard seal valving unit increases significantly as leakage requirements become more stringent, since better sealing requires more refined machining techniques to achieve progressively better surface finishes, and close tolerances are required to maintain the necessary alignment Smooth finishes are not as important for soft seal designs as for hard seals. In valving units using soft seals, elasticity and resiliency are the more important material properties, rather than hardness, and permeability becomes a consideration. In elastomeric seals, scratches and pitting are not as common as fretting and feathering or surface scratching. The sealing of soft seals is obtained by elastically deforming the nonmetals into the asperities of the interfacing metal. The force to seal must be sufficient to deform the elastomer or plastic to fill these asperities. On the other hand, if the sealing stress is too high some plastics will cold flow or extrude if they are not properly contained.

While leakage allowances in critical applications (such as spacecraft) may present the most severe requirement in the design or selection of the valving element and seat, the approach taken to satisfy the requirement is often greatly dependent upon otners factors, as discussed under parametric interactions in Detailed Topic 6.2.2.17. References 36-13, 131-6, and 131-25 describe the severity of this problem in detail and present several potential advanced technology solutions, including the following concepts:

- Elastic metal tubular valve seat (Detailed Topic 6.2.3.13)
- Cone labyrinth poppet valve (Detailed Topic 6.2.3.13)
- Curtain flap valve (Detailed Topic 6.2.3.13)
- Diffusion valve
- Thermoelastic freeze valve
- Wet seal
- Floating poppets.

In addition, Reference 35-14 describes a very comprehensive study of leakage past seats and poppets. This subject is discussed further in Detailed Topic 6.2.3.11.

6.2.2.4 FLUID MEDIUM. Fluid medium considerations which are important in valving unit selection are the physical properties of the fluid and the compatibility of the fluid with construction materials. Compatibility factors dictate materials selection when corrosive fluids must be handled. Materials compatibility data are presented in Sub-Section 12.5. The state and physical properties of the fluid become particularly important in determining pressure drop and flow capacity. These factors are also important in the design of seals, gases being more difficult to seal than liquids.

In flow calculations, gases are normally treated as compressible fluids and liquids as incompressible fluids. Certain physical properties, such as viscosity of liquids or gas constant for gases, have a significant bearing on valving unit design, especially in determining whether flow under given conditions is nozzle, turbulent, laminar, or molecular. Two-phase flow is frequently encountered in aerospace valving unit applications, especially with cryogenic fluids. It is discussed in Sub-Section 3.5 and Detailed Topic 3.8.2.5. Other fluid properties data of use to the valving unit designer may be found in Sub-Section 3.3 and Section 12.

While critical design features are usually based upon one primary fluid, consideration must also be given to secondary fluids with which the valving unit will be required to operate, such as cleaning, testing, purging, and pressurizing fluids.

6.2.2.5 FLUID PRESSURE. The importance of fluid pressure to the design of a valving unit is largely dependent upon the following factors:

Size. Structural strength becomes an increasingly important consideration with increasing valve size because pressure loads are a function of the square of the valve size.

Balance. If the valving unit is inherently pressure-balanced, the influence of pressure upon such parameters as size and actuation forces will be far less than in the case of an inherently unbalanced valving unit such as a ball, butterfly, or gate.

Pressure-induced strain. Binding of certain close-tolerance valving units can result if the design fails to take into consideration the strain which takes place in each part due to pressure loads.

Conditions of pressure. Circumstances under which the valving unit is subjected to high pressure must be considered. A drain valve, for example, may be required to seal against high pressure, but never be required to open until after pressure has been relieved. The main propellant control valves of some pump-fed liquid rocket engines are required to seal tightly against relatively low tank pressures (100 psi) for long periods of time, followed by a transient during which fluid pressure increases to several thousand psi as the valve moves to full open. This high pressure to which the valve is exposed decays as the valve is closed during engine shutdown, with the result that the valve is never required to seal against a differential pressure comparable to the high pressure of the fluid which flows through the valve in the open position.

6.2.2.6 FLUID TEMPERATURE. System and environmental conditions combine with fluid temperature to establish thermal gradients and thermal transients in the valving unit. Among the problems which thermal extremes and transients may cause at the valving unit are the following:

Differential thermal expansion. Differential expansion at the valving unit may be either the result of thermal gradients or different coefficients of thermal expansion. In either case leakage or binding between the valving element and the port may result.

Ice formation. Ice resulting from condensation and entrapment of moisture in cryogenic fluid systems may cause sticking of the valving element, or it may interfere with seating. Stems or shafts that are rotary actuated provide easy ice removal compared to linear action where ice must be sheared.

Changes in material properties. The reduction in allowable stress at high temperature requires careful material selection and mechanical design, especially for hot gas applications. In addition, thermal cycling can occasionally result in dimensional changes, as parts which expand at very high temperatures may yield partially and fail to return to original size when cooled.

Thermal stress. Thermal stresses resulting from extreme thermal gradients across elements of the valving unit often occur when cryogenic fluids or hot gases are first admitted to the valve.

Thermal shock. The term *thermal shock* is frequently used in valve design and essentially consists of combinations of the effects listed above when the valving unit is rapidly exposed to very cold or very hot fluids.

6.2.2.7 RESPONSE TIME. The total valve response time is the sum of the response times of the actuator and valving unit. Valving unit response is primarily a function of the following:

a) Mass of moving parts

b) Type and magnitude of motion of each moving part

c) Applied (actuator) force

d) Fluid dynamic forces

e) Friction forces

f) Fluid displacement.

Response times less than 5 milliseconds are possible if sufficient actuator forces are available. Reference 36-13 indicates that few aerospace applications require response times of less than 5 milliseconds, although a lightweight gate valve has been used at Ames Research Center which opens in less than 1 millisecond. In larger valves, seating configuration and materials may limit the velocity at which the valving element may contact the seat, possibly permitting more rapid opening than closing. A serious consideration, particularly with unbalanced valving units such as ball, butterfly, gate, and blade, may be the very high peak pressure differentials across the valving unit resulting from water hammer effects if valves are closed rapidly in systems with high flow velocities (Reference 34-10).

Although high response (short response time) requires relatively high actuation force, this actuation force may be minimized if a balanced valving unit (sleeve, spool, or balanced poppet) is designed for minimum friction (packless or seat and poppet sealing) and minimum valving element mass and stroke.

6.2.2.8 ACTUATION FORCES. The forces which must be overcome to move the valving element relative to the seat or valve ports are frictional forces, inertial forces, pressure forces, and fluid flow forces.

Frictional Forces. Frictional forces in poppet-type valving units are often of negligible magnitude; however, most other valving units require a significant breakaway force as part of the actuation force. Most unbalanced valving units can have a relatively high frictional breakaway force as well as appreciable dynamic frictional forces. With pressure-energized seals, the frictional load increases directly with pressure. In such valving units as the ball, plug, spool, piston, and sleeve, leakage past the valving element and housing may float the valving element, reducing coulomb frictional forces and creating small viscous damping frictional forces.

Inertial Forces. Inertial forces are a function of valving element mass, geometry, and type of motion. The influence of these inertial forces on the actuation force requirements is largely dependent upon required response time as well as the direction and magnitude of accelerations to which the system is subjected. Obviously linear acceleration has little effect upon the torque required to rotate a ball valve, but if such acceleration occurs along the axis of a sleeve, poppet, or other linear-motion valve it may have significant bearing upon actuation force requirements.

Pressure Forces. Pressure forces arise from any net pressure unbalance acting on the valving element. Depending upon the functional design of the valve, the pressure force may increase, decrease, or virtually have no effect on the actuation force. In an unbalanced valve design such as a conventional poppet, upstream pressure normally acts in a direction to seat the valve so that an increasing upstream pressure will tend to force the valving element tighter against its seat. The use of pressure unbalance to aid in sealing requires a higher actuation force to open the valve. When the size of the valve and the magnitude of pressure demand excessively large actuation forces, a balanced design and/or piloting can be used. Complete pressure balancing reduces actuation forces to those required to overcome friction forces, spring forces, and any required sealing forces. Pressure-balanced valving units are particularly advantageous in applications where accurate positioning of the valving element is desired. This is exemplified in servovalve and regulator design where spools and balanced piston or sleeve valving units are used. A pressure force which acts in a direction tending to open the valve against the sealing force is the antithesis of the usual shutoff valve design described above. It is applied primarily in relief valve design where it is desired that an increasing upstream pressure reduce the seating force and actually open the valve.

Fluid Flow Forces. Fluid flow forces arise from two basic effects that exist at the interface between the flowing fluid and the surface wetted by that fluid. These effects are:

a) Normal forces, due to the static pressure distribution set up under flow conditions at the boundary surface

b) Tangential frictional forces, caused by the viscosity of the flowing fluid.

In valve applications, the friction drag forces are substantially smaller than pressure forces and are usually neglected.

There are two basic analytical techniques available for estimating the steady-state flow forces or torques on a valving element:

1) The momentum theorem

2) The integrated surface pressure, or free streamline potential flow.

Momentum Theorem. The momentum theorem is often a more practical approach than the potential flow theory because it can include such real fluid effects as friction and compressibility by accounting for these factors in the velocity and pressure condition. However, the technique is limited in two respects. First, it must be possible to establish the velocity vectors and pressure across suitably selected inlet and outlet flow cross-sections. Second, in the case of calculating flow forces on the valving element, the momentum theorem is limited to those cases where the resultant fluid forces can be properly apportioned between the valving element and flow passage in the valve body. For example, in the venturi valve, the flow passage is close to the valving element throughout the valve stroke, and the momentum forces cannot be accurately apportioned between the two elements. However, in conventional poppet relief valves the fluid is exhausted to a large cavity and is not affected by the exit passage.

Assuming steady-state conditions and balanced pressure forces, the momentum equation may be expressed as

$$\vec{F} = \vec{M}_{out} - \vec{M}_{in} \qquad \textbf{(Eq 6.2.2.8a)}$$

where

$\vec{F}$ = force exerted by the valve on the fluid

$\vec{M}_{out}, \vec{M}_{in}$ = momentum vectors through the control volume surface

In Cartesian coordinates, Equation (6.2.2.8a) may be written with pressure forces

(Eq 6.2.2.8b)

$$\vec{F}_x = \frac{\rho Q}{g_c}(\vec{V}_{2x} - \vec{V}_{1x}) + P_2A_{2x} - P_1A_{1x}$$

(Eq 6.2.2.8c)

$$\vec{F}_y = \frac{\rho Q}{g_c}(\vec{V}_{2y} - \vec{V}_{1y}) + P_2A_{2y} - P_1A_{1y}$$

where

$\frac{\rho Q}{g_c}$ = steady-state mass flow rate, lb_m/sec

ρ = density, lb_m/ft^3

Q = volumetric flow rate, ft^3/sec

g_c = gravitational conversion constant, 32.2 $\frac{lb_m ft}{lb_f\, sec^2}$

$\vec{V}$ = steady-state velocity, ft/sec

P = static pressure, lb_f/ft^2

A = area, ft^2

Subscripts 1, 2 = inlet and exit stations, respectively, on control volume surface

x, y = horizontal and vertical directions, respectively

Depending upon the orientation of the valving element, one of the forces (F_x or F_y) may be zero due to valve symmetry. The application of the momentum theorem to poppet valves is presented in References 26-59 and 26-94, and to spool valves in Reference 45-1.

Integrated Surface Pressure or Free-Streamline Potential Flow Techniques. When the momentum theorem technique cannot be used to determine valving element flow forces, an alternate approach is the integrated surface pressure technique which can be used to calculate the desired flow force on the valving element. In this case it is necessary to establish the static pressure-shear distribution over the surface of the valving elements. This in itself is a formidable task, limiting its widespread use. By integrating over the entire contributing surface, the flow force components can be established.

In general, the corresponding force components can be expressed as follows:

$$R_x = \int_s (\tau_o \sin\theta - p\cos\theta)\, dA \qquad \textbf{(Eq 6.2.2.8d)}$$

$$R_y = \int_s (\tau_o \cos\theta - p\sin\theta)\, dA \qquad \textbf{(Eq 6.2.2.8e)}$$

where p is the static pressure, τ_o is the shear stress at the wetted surface, θ is the angle between the normal-to-the-surface and the positive x-axis, and the integration is carried out over the entire surface on which the fluid reaction force is desired. For low viscosity fluids such as air and water, the pressure term is usually predominant, allowing simplification by eliminating the shear stress term. When the static pressure-shear distribution is established, the equivalent center of pressure can also be determined: when suitably combined with the force components in Equations (6.2.2.8d) and (6.2.2.8e), the corresponding flow torque on a valving element such as a butterfly can be determined.

Potential flow theory is based on an ideal two-dimensional, nonviscous, incompressible, irrotational steady flow that can be described by either Ψ (x, y) or Φ (x, y), the potential and stream functions, respectively, which satisfy

Laplace's equation. The solution of Laplace's equation is given by harmonic functions which must satisfy the prescribed boundary conditions, and is usually very difficult to complete. The analytical approach used is treated in Reference 68-64. Digital computer techniques are outlined in Detailed Topic 8.3.4.4, which may be applied to this analysis. Additional computer techniques that handle viscous flow are covered in the following references:

Pearson, C. E.: "A COMPUTATIONAL METHOD FOR TIME-DEPENDENT, TWO-DIMENSIONAL INCOMPRESSIBLE VISCOUS FLOW PROBLEMS"
Sperry Rand Research Center, Sudbury, Mass. Rept. SRRC-RR-64-17, Program 11-372-10, February 64, 114 pp., figs., tbls., refs.

Fromm, J. E.: "A METHOD FOR COMPUTING NON-STEADY, INCOMPRESSIBLE, VISCOUS FLUID FLOWS"
Los Alamos Scientific Lab, Los Alamos, N. Mex., Contr. W-7405-Eng. 36, Rept. LA-2910, VC-32, 19 September 63, 152 pp., figs., refs.

The real fluid viscosity effects can be treated by Prandtl's boundary layer approach, which assumes all viscosity effects to be confined to a thin boundary layer at the surface of the wetted boundary. The balance of the fluid outside this boundary layer can still be treated as being ideal. If there is no flow separation from the wetted surface, the boundary layer contributes the additional shear stress, τ_0, at the boundary where the fluid velocity goes to zero. Due to the thinness of the boundary layer, it usually has little effect on the pressure distribution determined by potential flow theory. Thus the flow force for no-flow separation can be established within the accuracy of engineering design calculations by the use of potential flow theory. This approach breaks down if separation occurs, since the point of separation must be established to use the added free-streamline feature. In this case the growth of the boundary layer in the laminar and then the turbulent flow region along the wetted surface must be estimated to determine the point of separation, usually a difficult task relying heavily on experimental data correlations. For fluid component configurations with sharp corners, e.g., sharp-edged orifices, spools, etc., the point of separation can be taken at the sharp corner. In other cases, such as in diffusing sections, with no abrupt changes in geometry and turbulent fluid flow, the determination of the separation point can be only established experimentally, since present boundary layer theory is inadequate to predict this. However, once the separation point is established, the free-streamline feature of this approach assumes that a free-streamline forms off the separation point and adjusts itself so that in the separation region the pressure is constant and equal to the downstream pressure. This latter assumption is in agreement with experimental observation for steady, separated flows. The flow pattern compatible with this free-streamline is then determined by potential flow theory to establish the pressure distribution and corresponding flow force for the separated flow condition.

6.2.2.9 SIZE. Valve size (expressed in terms of some common denominator such as valve inlet port diameter) has an effect upon the valving unit configuration selected for any particular valve type, as well as upon the type of valve to be used for a given application. Valves are not usually considered to be "scaleable" for a number of reasons, including:

a) *Clearances.* The small clearance required, or possible, between the valving element and valve housing in certain small designs may be completely impractical in larger sizes due to thermal expansion, structural distortion, concentricity limitations, or other considerations. Increasing the clearance in a larger version of the valving unit might require use of a completely different sealing concept from that employed in the smaller size.

b) *Seating.* Considering the flat seat and poppet approach treated in detail in Reference 35-14, it is obvious that large sizes of flat seats and poppets present totally different requirements in terms of such factors as flatness retention, alignment, contamination sensitivity, and effects of thermal gradients.

c) *Pressure-balancing.* As illustrated in Reference 34-10, unbalanced valving units such as balls and butterflies appear far less practical in larger sizes (greater than 5 inches) for high-pressure systems than do pressure-balanced types such as sleeves.

d) *Fabrication practicability.* Valving units which require machining and assembly of a relatively complex valving element assembly within the valve port diameter (such as a butterfly valving unit) obviously do not lend themselves as well to very small applications, as do simpler types such as poppets, where the valving element itself can be quite simple.

e) *Flow regimes.* The near-closed flow regimes of laminar and molecular flow (illustrated in Figure 6.2.2.5 for a 1-inch diameter flat poppet and seat) would account for a far larger proportion of the total stroke of a very small valving unit than for a large valving unit. Such differences could account for significant variations in flow characteristics for grossly different sizes of a given valving unit configuration, even if seat and valving element proportions were held constant.

6.2.2.10 WEIGHT. Weight considerations alone will seldom dictate the choice of a particular valving unit configuration, but must often be evaluated when comparing unit types or sealing methods to be employed.

Weight savings realized through careful valving unit design may come from associated modules rather than from the valving unit itself. For example, a pressure-balanced poppet configuration might weigh essentially the same as an unbalanced design, but permit the use of a smaller and lighter actuator. Such considerations are an important part of valving unit optimization, as discussed in Detailed Topic 6.2.2.18.

6.2.2.11 ENVELOPE. As with other design and selection factors, envelope limitations are more inclined to influence the selection of basic valve type (ball, poppet, etc.) than details of the specific valving unit. The overall system layout in numerous applications may be significantly improved by ingenuity on the part of the valving unit designer, however. Unlike commercial applications, where a system is usually designed around off-the-shelf valves, the stringent requirements of many aerospace systems necessitate the design of individual valves for each specific application. Among the valving unit design and selection factors which system envelope requirements may influence are:

a) *Flow directions.* Where system layouts require a change in flow direction in the vicinity of the valve, such changes may often be most efficiently designed into the valve. The low pressure drop characteristics of a ball valving unit may be lost if a 90-degree change in flow is required through or near the valve. In such cases an angle poppet or sleeve configuration may be more efficiently incorporated.

b) *Length limitations.* Length limitations must be evaluated on the basis of direction and nature of the limitations. In certain instances, for example, assembly and maintenance requirements may indicate need for a very small flange-to-flange dimension, suggesting the use of a blade or gate valving unit which may be undesirable for a number of reasons. Possibilities may exist, however, which would permit designing ball, butterfly, poppet, or other valving units whose protrusion some distance into the line or mating component could be accepted.

c) *Diameter limitations.* These are also directionally sensitive but are sometimes erroneously specified on the basis of the nearest interferring item from the centerline of the valve. Proper orientation of a ball or butterfly stem, or a gate housing, may satisfy some applications. Others may entail the use of such concepts as in-line venturi poppets with integral actuators, no part of which extends significantly beyond the basic line outside diameter. Sometimes the choice between a ball or butterfly valving unit is based upon the fact that the butterfly (less actuator and shaft) is completely contained within the line ID, whereas the ball valving element must be appreciably larger in diameter than the line, unless a reduced port size is used (see Figure 6.2.3.2c.

6.2.2.12 CONTAMINATION SENSITIVITY. Since no system is completely free of particulate contamination, valves themselves being contamination generators, sensitivity of a valving unit to contamination is an important consideration in valve design. Contamination can lodge between the seal faces, thus (1) creating a leak path (although the particles may be crushed in the interfaces, they may elastically deform the valve seat as it is mated with the valving element); and (2) scratching or deforming the valving element or seat plastically (depending upon the size and hardness of the contaminant), thereby reducing the valve's further usefulness as a sealing device. Contamination and its effect upon fluid systems and components is treated in detail in Section 10, "Contamination and Cleaning."

The damage susceptibility of hard seats to particulate contamination is acknowledged by most valve manufacturers and is cause for concern. Where pressure drop limitations permit, many manufacturers of small valves incorporate wire mesh filters at the inlet port to screen out particulate matter. Others tend to de-emphasize the contamination problem by claiming the probability is low of a particle of sufficient size causing failure by lodging itself on a seat land at the time of closure. Manufacturers of internal combustion engines have for a long time faced the problem of carbon contamination of conical poppet valves, some choosing relatively narrow seats to minimize contamination probability and to increase crushing stresses on those particles which are trapped, others preferring wide seats to minimize the effect of pits in the seat or poppet faces resulting from contamination or corrosion.

The damage susceptibility of soft seats or seals depends greatly on the resiliency of the nonmetal used. With elastomeric seal materials, the contaminant can be pressed locally into the valve seal during closure without plastically deforming the seal. However, a continual subjection to such treatment may cause seal fretting and feathering. If the nonmetal is relatively hard (e.g., Teflon) the size of the contaminant which can be impressed is diminished, and at the same time the nonmetal may become scratched or nicked, creating a leak path.

In unpacked (sealless) spool valves, there is a tradeoff between contamination sensitivity and leakage based on clearances between the valving element and its housing. If the designer becomes over-zealous in minimizing leakage by reducing the clearance between members, a larger number of contaminant particles can become lodged, causing valve failure. The clearance values allotted should be checked at both of the temperature extremes to which the valve will be subjected, in order to ensure adequate design for the maximum contamination particle size anticipated.

An appreciation of the relationship between valving unit dimensions and particle size may be obtained from Table 10.6.2.7a, which lists average critical dimensions for hydraulic servo valves, and Table 6.2.2.12, which shows typical particle sizes from various sources.

The formation of ice on the valving unit surfaces of cryogenic valves may be treated as a form of contamination. Such ice formations have been the source of numerous problems with practically all types of cryogenic propellant valves, including:

a) Interference and adhesive action at the valve seat, preventing valve opening, especially with butterfly valving units

b) Ice formation on the seat or valving element while the valve is open, preventing seating when valve is closed.

Table 6.2.2.12. Typical Sources of Wear Particles
(Reference 36-13)

Source	Size (microns)
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

c) Damage to lip or other soft seals when the seal slides over ice formed on the valve body or valving element, especially in sleeve valve designs.

Section 10 and References 131-6 and 131-25 describe in detail techniques for minimizing contamination sources and increasing system tolerance to contamination.

One of the more frustrating paradoxes facing the valve designer is the fact that those designs least likely to generate contaminant particles, such as flat poppets and seats with no scrubbing action, are most susceptable to contamination; whereas those designs least susceptable to contamination, such as the cone labyrinth or other scrubbing concepts, are themselves prime generators of wear particles.

It is interesting to note that some aerospace applications, such as hydraulic aircraft control systems, where some leakage is tolerable, actually exploit the normal contamination found in certain systems to reduce seat and poppet leakage by permitting the accumulation of contamination particles to "dam up" any leak paths.

6.2.2.13. VALVE LIFE. Valving unit life depends upon the temperature, operating pressure, number of cycles, loads on members, materials of construction, and exposure to space environments. The critical module which limits life is the valving unit seal or seat. For metal-to-metal sliding seals the materials must resist galling, galvanic corrosion, and scratches from contaminants. If the seal is in a seat configuration, the materials may be required to have sufficient fatigue strength to withstand repeated hammering of the valving element against the seat without adverse effects, such as excess leakage. For soft seals, potential seal blowout, cold flow, extrusion, and feathering limit valving unit life. A distinction should be made between seal failure resulting from permanent set during long storage and that resulting from cold flow under pressure.

The application of a particular valve is usually the factor which determines the life requirement of the valving unit. For example, a booster rocket propellant pre-valve which is required to open only once is a far different application than the attitude control system propellant valve which may be called upon to function through thousands of cycles.

To attempt to design unnecessary life into a valving unit may result in a module which is less reliable for its particular function. For example, the soft seal material which may provide the best leakage characteristics for a poppet valve seat may be capable of withstanding only a few cycles, but could be the best selection for some applications.

Long life requirements may take the form of very large numbers of cycles (as in aircraft and some test facility and spacecraft applications), or long total elapsed time, either between installation and operation (as with weapon systems), or between initial cycling or exposure to critical environment and final required functioning (as in spacecraft). In any of these cases, the manner as well as the magnitude of the valve life requirement must be considered in selecting the appropriate valving unit type, configuration, and materials.

6.2.2.14 WEAR. Reference 36-13 emphasizes the relationship between wear and valve life, and points out that most valve wear problems are associated with the valving element and seat. It is further suggested that the size of the wear particle generated between the seat and valving element is one parameter influencing leakage in valving units, and indicates that harder materials are more successful in limiting leakage in metal-to-metal valve closures. It is also noted that while minimum friction forces in a vacuum environment may be obtained by sliding a hard material on a softer material which has been plated to a harder material, the softer material will produce larger wear particles that may influence leakage. The size of wear particles, as determined by Rabinowicz, produced by various materials rubbing against like materials is presented in Table 6.2.2.14a; the effect of various environments on the size of copper wear particles, as determined by Elliott, is shown in Table 6.2.2.14b. According to Reference 36-13, Rabinowicz has shown that copper wear particles are essentially independent of surface velocity, but that wear particle size tends to increase with load, ranging from less than 44μ to over 500μ in diameter. It is also mentioned that while extensive tests of nylon-on-nylon showed no wear, Teflon-on-Teflon produced fibrous particles with diameters ranging from 40μ to 150μ, with the average taken as 90μ.

Three valving unit wear minimizing techniques which have been used to date according to Reference 36-13 are:

Table 6.2.2.14a. Size of Wear Particles and Related Functions Under Standardized Conditions of Ambient Atmosphere
(Reference 36-13)

Metal	p	W	W/p	d	W/pd
Lead	4×10^{8}	440	110×10^{-8}	270×10^{-4}	42×10^{-6}
Tin	6	540	90	120	75
Bismuth	12	375	31	50	62
Woods alloy	16	400?	25	400	6.2
Cadmium	23	600	26	320	8
Aluminum	30	900	30	140	21
Zinc	30	750	25	440	5.6
Antimony	45	380	85	400	22
Copper	60	1100	18	250	7.3
Brass	120	700?	5.8	100	5.8
Mild Steel	200	1000	5	60	8.3
Iron (oxide)	2000?	600?	0.3	1	30
Aluminum (oxide)	2000	900	0.45	1	45
Teflon	4	15?	3.8	90	4.2
Nylon	20	30?	1.5	?	?
Babbitt	30	400	13	350	3.7
Silver	80	920	11.5	330	3.5
Nickel	260	1650	6.3	35	18
Glass	550	200?	0.36	1	36

p = Penetration hardness, $dynes/cm^2$
W = Work of adhesion of the system, $ergs/cm^2$
d = Diameter of the average wear particle, cm

a) *Double series valves.* A hard-seat valve is closed first and opened last to protect a soft-seat valve (used for tight sealing) from erosion and wear problems associated with throttling. This approach has been used in test facilities at NASA's Langley Research Center and a variation has also been used on some spacecraft propulsion systems where soft-seat shutoff valves are placed in series with nonseating throttling control valves.

b) *Retractable valve seats.* The seat is retracted from the valving element surface before valving element rotates or slides, then is returned to the sealing position after valving element motion has stopped. This technique is typified by the 17-inch diameter ball valves manufactured by AiResearch for propellant suction line shutoff on Saturn launch vehicles.

c) *Poppet translation prior to rotation.* Similar in principle to the retractable valve seat, but entailing motion of only the valving element, this approach is typified by the ball poppet valves manufactured by Orbit Valve Company, Tulsa, Oklahoma.

6.2.2.15 ENVIRONMENTAL CONDITIONS. As discussed in detail in Section 13, environmental conditions include:

- Pressure
- Acceleration, shock, and vibration
- Atmosphere
- Temperature
- Space environments
- Corrosion.

The considerations discussed in Section 13 apply to valving units as well as other modules and should be carefully

Table 6.2.2.14b. Wear of Copper in Various Environments
(Reference 36-13)

Environment	Average Fragment Diameter (microns)
Atmospheres	
Nitrogen	480
Helium	380
Carbon dioxide	300
Dry air	224
Oxygen	201
Laboratory air	177
Wet air	144
Lubricants	
Cetane	12.0
Silicone DC 200	9.5
Ucon LB-70X	9.5
Palmitic acid in Cetane	8.0

evaluated. Pressure, acceleration, shock, vibration, and temperature may be expected to influence the valving unit configuration. (Particular attention is directed to the effect of acceleration fields upon poppet orientation and compensation, as treated in Sub-Topic 13.3.3.) In addition to absolute temperatures, thermal distortion and differential expansion resulting from temperature gradients must be considered. Pressure must be considered not only from the standpoint of those strains which must be determined in cases where critical tolerances exist, but also in terms of exposure to cyclical pressure variations with respect to possible fatigue failure.

On the other hand, atmosphere, temperature, space environment, and corrosion will particularly influence the selection of valving unit materials. Usually the most significant manifestation of environmental conditions upon valving unit material choice is the feasibility of using soft seats or seals for a particular application. Usually radiation or temperature are the environmental factors which may preclude the use of soft seats or seals in a particular valving unit. Temperature extremes can be governed by the flowing medium temperature as well as the ambient temperature. Material selection usually must be determined from the worst conditions; where exposure time at temperature extremes is limited to prevent valving unit parts from reaching a high steady-state temperature, consideration should be given to the use of thermal sinks. Space radiation and vacuum can have adverse effects on the sealing of the valving unit, as pointed out in Reference 131-6. Many materials (particularly polymers) degrade in the presence of radiation. Vacuum exposure enhances the chance of pressure bonding, particularly for metal-to-metal sealing (cold welding), and galling. The degree to which these effects may occur depends upon the design of the valve and the length of time in the environment. Cold welding, if it is going to occur, will take place in a relatively short period of time (usually measured in minutes), whereas radiation effects in a space environment are usually significant only after relatively long durations. Table 13.6.8 lists the time required to accomplish several representative space missions.

6.2.2.16 MATERIALS. The selection of construction materials for the seal assembly and valving element must be based upon practically all of the preceding factors. The function of the valving unit initially determines the severity of the material selection problem. Flow modulation or flow diversion functions may usually be satisfied with all-metal valving units using a relatively large variety of metals. The sealing requirements of the shutoff function, on the other hand, when combined with other application factors, may present a most significant problem in the selection of materials. This discussion will therefore emphasize the seat and valving element material requirements associated with the sealing problem in shutoff valving units.

Valving units for shutoff applications may use either hard or soft seals, packless valving units seldom being used for leak-tight shutoff. Although a hydraulic system, for example, might employ a spool or piston valve to perform a shutoff function with a relatively high allowable leakage, such an application is not considered to represent a shutoff application for purposes of this discussion. Hard seals normally imply metal seats and metal valving elements, usually with both parts hard and finished. Soft seals normally imply a plastic or elastomer, usually mated with a hard metal. Common aerospace practice has been to use soft-seal valves for shutoff applications wherever practicable, largely because of difficulties encountered in the development of reliable, low-leakage, hard-seal valves. In some spacecraft applications, the desirability or necessity of using soft seals has resulted in revisions of valve configuration to provide for shielding of the seat from the radiation environment.

Increasing use of highly reactive propellants such as fluorine, which is not compatible with any presently available polymers, has increased the demand for all-metal valving units. Section 12, "Materials," includes a discussion of fluids as well as structural and seal materials, and indicates the compatibility of various materials with fluids. If no plastic or elastomer is suitable for a particular valving unit application, the metal-to-metal alternatives available may be summarized as follows:

a) Hard metal valving elements and seats with fine finishes and no scrubbing action, usually limited to poppet valves (discussed in detail in Detailed Topic 6.2.3.11).

b) Hard flat metal valving elements and seats with fine finishes and pure scrubbing action, wherein valving element and seat remain constantly in contact. This approach is most commonly found in all-metal linear or rotary slide valving units for fluid power applications.

c) Precision mating pistons and cylinders, seldom applicable to aerospace designs other than fluid power applications because of limitations regarding lubrication, temperature, and manufacturing tolerances. (An approximation of this approach is found in hydraulic system spool and piston valving units with relatively large allowable leakage rates.)

d) Piston rings, used as seals on axial cylindrical applications such as sleeve valves and on rotary applications as seals for butterfly, ball, or plug valves. Acceptable leakage rates are usually obtained only by the use of overlapping rings or rings with stepped gaps (Reference V-135).

e) Expandable thin-wall cylindrical metal seats sealing against conical or spherical poppets through shearing or scrubbing action (See Figures 6.2.3.13a and b).

Hardness is the common demoninator in design of metal-to-metal seating interfaces, a hard material affording the formation of a relatively smooth surface. Alternative (a) usually incorporates both hard seats and valving elements to obtain the required surface finish and resistance to marring. The other four techniques involve rubbing contact; this usually dictates unlike hardnesses if friction and wear are to be minimized, unless the fluid involved is a good lubricant.

In very general terms, the surface energy approach to friction and wear as explained by Rabinowicz (References 491-1 and 19-233) indicates that:

a) With good lubrication, hard similar metals yield low friction coefficients and small wear particles.

b) With no lubrication, dissimilar metals yield lower friction coefficients and smaller wear particles than similar metals, and are primarily dependent upon the characteristics of the softer material.

c) Good lubrication will reduce the surface energy of adhesion (and hence friction and wear) of nonmetals to approximately 25 percent of the unlubricated values, but will reduce that of similar metals to about 5 percent of the unlubricated value.

Friction and wear data useful in the selection of mating materials for valving units may be found in Sub-Section 12.7.

6.2.2.17 PARAMETRIC INTERACTIONS. Serious evaluation of the preceding design and selection factors reveals that practically any design feature chosen on the basis of one of these parameters affects one or more other parameters, an interaction effect which should be considered when evaluating any particular design parameter or factor. Many valve failures result from simple design oversights which could have been avoided had the interrelationships of environment, materials, stress, and dynamics been considered.

In evaluating the interactions between the various design and selection factors, a good approach is to group these parameters into three categories:

1) Specified or fixed parameters, usually including the fluid medium, pressure, temperature, and flow rate

2) Required or desired design goals, usually including pressure drop, leakage, response time, weight, and valve life

3) Design variables, usually including materials (in addition to type and configuration), but for each specific case probably including several other design factors such as flow versus stroke, response time, actuation force, and size.

Such an evaluation, in addition to showing the relative significance of the design requirements or goals, is useful because of all of those factors which may be varied in arriving at a valving unit design are identified. In the absence of such an evaluation, it is very easy to overlook the fact that one particular parameter may be varied in order to facilitate satisfying the more critical design requirements. For example, the lack of critical pressure drop or flow capacity requirements may permit the use of a relatively small valving unit of high pressure drop, but with which it is far easier to satisfy leakage, actuation force, weight, and envelope requirements or goals.

6.2.2.18 DESIGN OPTIMIZATION. In spite of their high degree of sophistication, very few of the valves produced for aerospace applications represent optimized designs. An optimized design for a valving unit is the one design that is best for a given application. The extensive design analysis and the numerous design-fabrication-test analysis iterations required to develop an optimum valving unit usually entail cost and time expenditures which are unreasonable for most applications. It is apparent from the large number of design and selection factors and their interactions that any valving unit design must represent a series of compromises if all design requirements, as well as the more general cost and reliability considerations, are to be satisfied. However, instances occur in which optimization, or at least an approximation of optimization, of one or more factors is justified. Those factors which most often lend themselves to such treatment are:

a) Pressure drop and flow capacity

b) Leakage

c) Response time

d) Actuation forces

e) Weight

f) Envelope

g) Valve life.

Optimization of a design to achieve particular levels of such parameters must usually be limited by constraints imposed by cost and/or reliability. These constraints are much the same for valving units as for other modules and include consideration of the following:

a) The tradeoff between the high cost of developing the least expensive valving unit that will meet all requirements, and the cost of using a somewhat overdesigned and more expensive unit whose more liberal design allowances significantly reduce development costs. The total number of valving units required is usually of paramount importance in such cases.
b) The tradeoff between the cost of developing the lightest weight valving unit that will meet all requirements and the cost to the system of accepting the weight penalty of a slightly heavier design.
c) The tradeoff between the cost of developing a valving unit with minimum pressure drop and the cost of using a higher upstream or lower downstream pressure. This type of tradeoff study can become very complex when considering spacecraft systems.
d) The tradeoff between the reliability associated with the use of redundant seals on a particular valving unit and the use of redundant valving units or redundant valves, or even a more reliable nonredundant valving unit.
e) The cost and reliability implications associated with deviating from a valving unit type with which experience has been obtained.

An excellent example of an appropriate use of valving unit optimization may be found in the power industry. The design of large (500 to 1000 megawatt) steam turbo generators by Allis-Chalmers Mfg. Co., Milwaukee, Wisconsin, required modification of the angle poppet (linear plug) stop valves and throttling valves of existing 400mw turbines to accommodate the higher steam flow rates. Five alternatives were available:

a) Increase pressure drop through the valve
b) Increase valve size
c) Increase the number of parallel flows
d) Change the basic flow pattern or valving unit type
e) Optimize the flow path to minimize pressure drop through the valve.

The last approach was chosen on the basis of cost, reliability, and such technical considerations as actuator power requirements, thermal stress, flow control, leakage, and overall system performance. Several hundred model tests were conducted on two basic valve types (similar in configuration, but with and without strainers). The optimized valve configurations were found to have a total pressure loss, including entrance, valving unit, and exit of less than one velocity head ($K = 0.65$) as compared with former values of more than two velocity heads. The analysis was sufficiently comprehensive to permit separating entrance, valving unit, and exit losses. In addition, a significant reduction in the size of the valving unit was achieved.

The application of valving unit optimization techniques to aerospace valve designs may be expected to be equally successful, especially in view of the stringent demands of spacecraft propulsion systems.

6.2.3 Design Data

This sub-topic is divided into valving unit types, with information concerning valving unit configurations, sealing, actuation forces, and flow characteristics presented for each type. Parametric data is presented where available, and reference is made to sources from which additional or more detailed data may be obtained.

Sub-Sections 5.1 through 5.9 of the handbook include related data, some of which is reiterated in this sub-topic.

6.2.3.1 BUTTERFLY. The butterfly valving unit consists of a relatively flat valving element (usually referred to as a *disc*, but also known as a *damper*, *gate*, *butterfly*, *butterfly plate*, *valve plate vane*, *flapper*, or *blade*) which may be rotated within the valve body or housing to obstruct or permit flow. The valve seal may be located either on the movable valving element or may be attached to the housing. Sub-Topic 5.2.4 describes the operation, performance characteristics, applications, and limitations of butterfly valves for shutoff service. Detailed Topic 5.3.5.5 describes flow and actuation torque characteristics of butterfly valves for flow control service.

Configuration. The configuration of a butterfly valving unit may be described in terms of the following basic parameters:

a) *Port shape.* The great majority of butterfly valves use circular ports, usually of the same diameter as the line; however rectangular ports with rectangular valving elements are found in low pressure gas ducting (Reference 26-86). Particular flow characteristic requirements can result in unusual port and disc shapes bearing little resemblance to the ordinary butterfly valve, such as the V-port butterfly valve described in Reference 144-1.
b) *Disc seating angle.* The closed position of the disc is frequently chosen to be at a disc seating angle, α_o (the angle between the plane of the disc and a plane perpendicular to the port centerline) between 0 and 30 degrees in order to facilitate sealing and to avoid the necessity for placing stops in the port which would interfere with flow in the open position.
c) *Shaft location.* As illustrated in Figures 5.2.4.1 a, b, and c, the shaft may either be straight (in the plane of the disc); canted (shaft centerline passing through the center of the disc at an angle, β); or eccentric (shaft centerline parallel to plane of the disc, but offset from the plane of the disc and/or the centerline of the port).
d) *Disc shape.* The plan view of the disc is primarily defined by the port shape and disc seating angle. For a circular port and 0-degree seating angle ($\alpha_o = 0$) the disc will be circular, whereas if a positive seating angle is used the disc must be elliptical, with the major axis of the ellipse equal to the port diameter divided by cos α_o. Disc shape is also constrained by the shaft location. It is apparent, for example, that a rectangular

disc and port cannot be employed with a canted shaft. It is in the shape of the disc cross section, however, that the greatest number of variations may be found. The examples described in the remainder of Detailed Topic 6.2.3.1 illustrate some of the disc shapes used to provide particular design characteristics.

The relationship between the disc rotation, α, disc seating angle, α_0, and shaft angle, β, is explained in Detailed Topic 5.2.4.1 and may be expressed by the following equation:

$$\alpha = 90 \text{ deg} - \alpha_0 + \sin^{-1} \tan^2 \beta \quad \textbf{(Eq 6.2.3.1a)}$$

where

α = shaft rotation required to change disc position from the seated position to that position in which the plane of the disc is parallel to the port axis, degrees

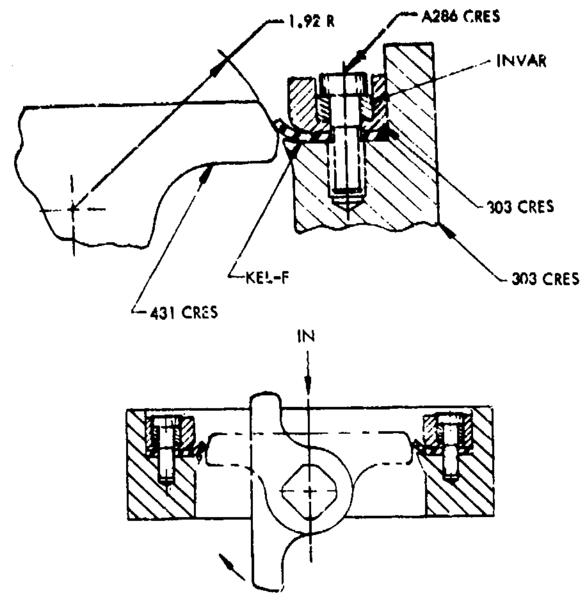
α_0 = disc seating angle (the angle between plane of the disc in the closed position and a plane perpendicular to the port centerline) measured in the plane of the port centerline, degrees

β = shaft offset angle (the angle between the centerline of the shaft and the plane of the disc), degrees

Sealing. The design concepts described below are representative of sealing techniques which have been used successfully in various aerospace butterfly valve applications. The number of approaches appears to be unlimited, but little in the way of parametric data is available to aid the designer. The scrubbing action with which many butterfly shutoff valves seat requires application of the dynamic seal techniques discussed in Sub-Section 6.4. It is important that butterfly valve seats be designed to withstand unshielded exposure to the flow stream when the valve is open, whether the seal is located on the housing or the disc.

Figures 6.2.3.1a and b illustrate two seat designs for butterfly valving units of the type which have been used in diameters up to 5 inches for both cryogenic and storable liquid rocket engines. Both designs use Kel-F lip seals and eccentric shafts. The 1000-psi design shown in Figure 6.2.3.1a uses a radial seal interference of 0.01 in./in. of the inside diameter of the seal, when a 0.075 in. thick seal is used (Reference 35-12). The tendency for the seal to be damaged by deflecting down into the path of the disc while closing presents a development problem with both designs.

Figure 6.2.3.1c shows a floating two-piece convoluted seal for aircraft use with hot (750°F) engine bleed air. This seal is formed to an involute profile, so that contact with the radiused edge of the disc is always in a direction normal to the seal surface. As with the eccentric shaft designs shown in Figures 6.2.3.1a and b, the slight shaft eccentricity of this design serves to result in a plugging action as the valve closes, somewhat approximating the linear translation of a poppet against its seat. Note that the seal does not protrude into the flow stream, although it is exposed.



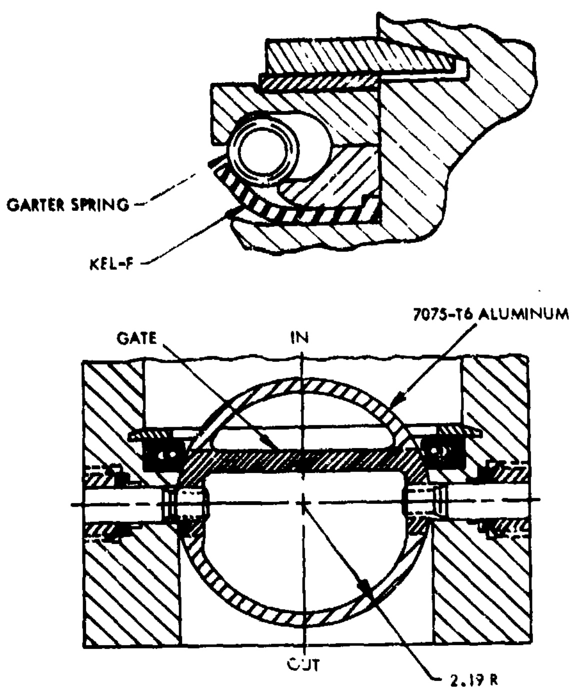
DESIGN PARAMETERS

FLOW MEDIA: LIQUID OXYGEN
OPERATING PRESSURE: 1000 PSIG
OPERATING TEMPERATURE: -423 TO 160 F
SIZE: 3 IN. LINE

Figure 6.2.3.1a. High-Pressure Cryogenic Butterfly Valve
(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-88)

Another approach to the use of an all metal valving unit with a convolution seal is shown in Figure 6.2.3.1d where the seal is located on the disc rather than the body. Normally, valving units such as this seat at an angle, α_0, of about 5 degrees, the circular convolution seal deforming to the required elliptical shape elastically. The shaft of this design is canted to an angle of 10 to 15 degrees to provide an uninterrupted seal and seating surface. The convolution seal is simply formed in the periphery of a thin disc which in turn is clamped between two backup plates to form the valving element. As with the convolution seal shown in Figure 6.2.3.1c, the seal is oriented with the open side upstream to provide some degree of pressure actuation. The design illustrated in Figure 6.2.3.1d has been used in 2-inch and larger sizes for pressures to 260 psig and temperatures of 900°F.

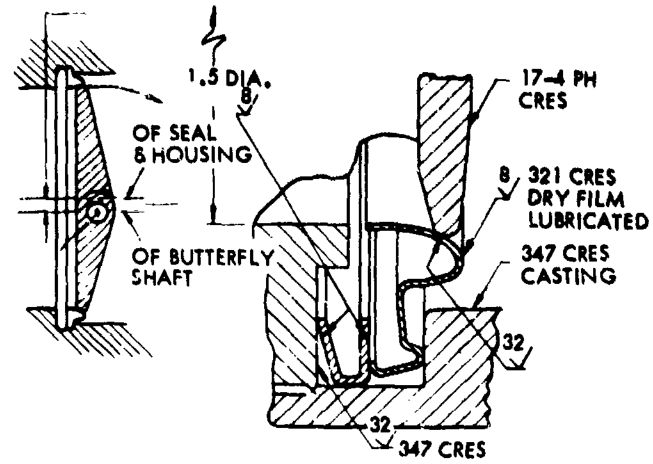
Among the simplest butterfly valve seals is an elastomeric sleeve on the inside of the body which forms a seat against which the disc closes. Usually the disc is round and of a diameter sufficiently larger than the sleeve ID to provide the desired seal interference. Either canted or straight shafts are commonly used with valve seating angles from 0 to 3 degrees. For applications below a 2-inch diameter, the disc is sometimes cantilevered on a single shaft (usually canted) and is referred to as a *spoon valve.*



DESIGN PARAMETERS

FLOW MEDIA: LIQUID OXYGEN
OPERATING PRESSURE: 40 PSIG
OPERATING TEMPERATURE: -320 TO +160 F
SIZE: 41/2 IN. LINE

Figure 6.2.3.1b. Low-Pressure Cryogenic Butterfly Valve
(Reference 35-12; Clary Dynamics Corporation design, p. B-92)



DESIGN PARAMETERS

FLOW MEDIA: AIR
OPERATING PRESSURE: 7 TO 205 PSIG
OPERATING TEMPERATURE: -75 TO +250 F
FLUID TEMPERATURE: -75 TO +750 F
SIZE: 1-1/2 IN.

Figure 6.2.3.1c. Butterfly Valve with Floating Metal Seal
(Reference 35-12; Aero-Corry Division of Aeroflow Dynamics, Incorporated design, p. B-94)

An approach to butterfly valve sealing which eliminates scrubbing between disc and seat, greatly reducing torque requirements, is the pressure-actuated cylindrical diaphragm seal illustrated in Figure 6.2.3.1e. Reference 68-1 presents a comprehensive design analysis of this concept which has been used successfully in both cryogenic and high-temperature applications. The thin-walled cylindrical sealing band, usually of nickel alloy such as Inconel X, is butt-welded peripherally into the valve body. Pressurization of the sealing band actuation cavity forces the sealing band into contact with the edge of the disc to effect a seal. Where temperature and compatibility permit, plastic or elastomer inserts may be used to improve sealing characteristics. It is interesting to note that for some applications where sealing is required only at high temperature, advantage may be taken of thermal expansion effects by using actuation pressures equal to or lower than line pressure, permitting use of the line pressure itself as the actuating fluid. The inflatable-band concept dictates the use of a canted shaft to permit a continuous seal, the angle required being a function of sealing band width and shaft diameter.

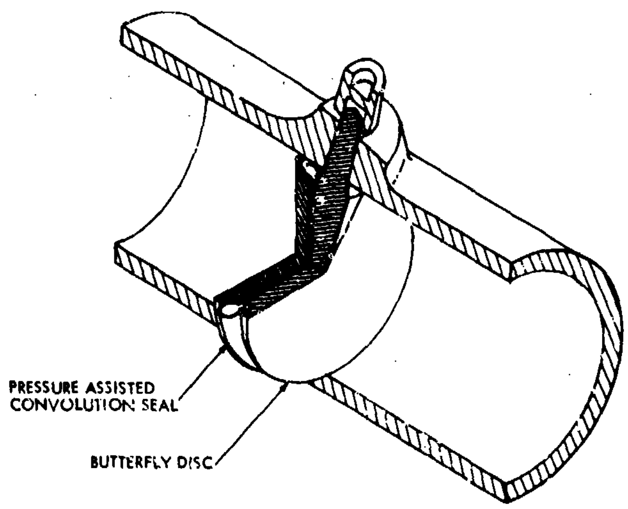
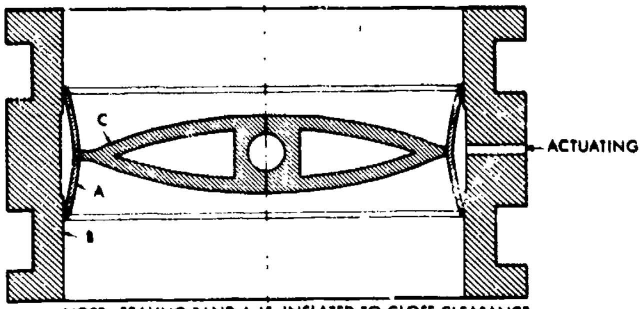


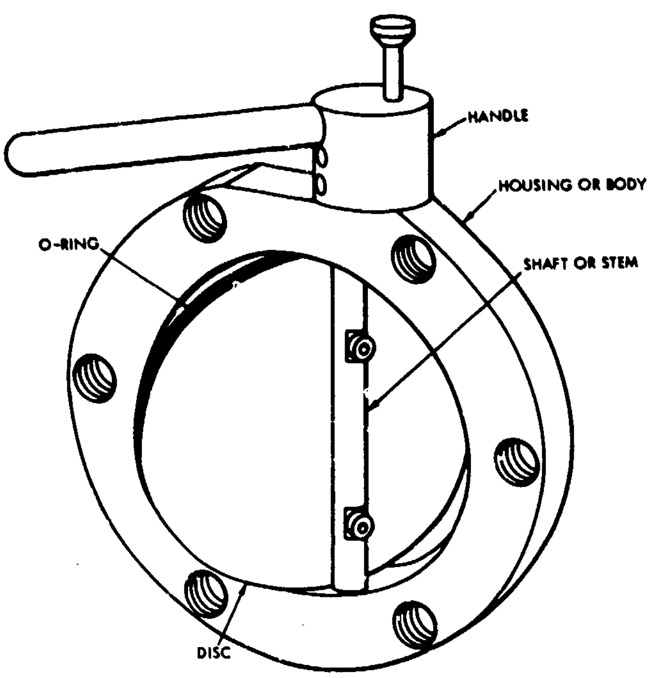
Figure 6.2.3.1d. Butterfly Valve with Metal Convolution Seal on Disc
(Courtesy of AiResearch Mfg. Division, The Garrett Corporation, Phoenix, Arizona)



NOTE: SEALING BAND A IS INFLATED TO CLOSE CLEARANCE BETWEEN VALVE BODY B AND VALVE DISC C

Figure 6.2.3.1e. Pressure-Actuated Cylindrical Diaphragm Seal

(Adapted from ASME Paper 59-SA-1, C. Y. Neou, Copyright 1958, American Society of Mechanical Engineers, New York, New York)



NOTE:

VALVE BODY INSIDE DIAMETER	3.752 IN.
DISC OUTSIDE DIAMETER	3.700 IN.
DISC O-RING GROOVE DIAMETER	3.410 IN.
O-RING CROSS SECTION DIAMETER	0.210 IN.
O-RING SQUEEZE	0.039 IN.
DISC THICKNESS	3/8 IN.
VALVE STEM DIAMETER	3/8 IN.
VALVE STEM OFFSET FROM CENTER LINES	5/16 IN.

Figure 6.2.3.1f. High-Vacuum Butterfly Valve with O-Ring Seal

(Reprinted with permission from "Review of Scientific Instruments," October 1955, vol. 26, no. 10, R. E. Holland and F. P. Mooring, Copyright 1955, American Institute of Physics, New York, New York)

Still another approach to the all-metal butterfly valve seal is the simple expedient of installing one or more piston rings in a groove machined in the disc periphery (References V-135 and 112-1).

In some high-velocity applications, and where divided rings are used (in straight shaft rather than canted configurations), the rings are pinned in place. Piston ring butterfly valve seals have been used successfully in both hot air (aircraft) and cryogenic service.

A similar principle uses an elastomer O-ring in the disc periphery. Figure 6.2.3.1f shows such a hand-operated valve for high-vacuum service as well as dimensions of a typical valving unit. An eccentric shaft is used to accommodate the continuous seal.

Actuation Forces. The actuation force required to position the butterfly disc is usually expressed as the torque required to overcome seal friction, bearing friction, and fluid forces. For any given application, the sum of these forces may be expected to result in a difference between the torques required to open and to close the valve.

Seal friction forces are a function of the valving unit configuration, seal design, seal lubrication, and fluid pressure (in the case of pressure-actuated seals such as lip seals and convolution seals), as well as the friction coefficient of the rubbing materials. Seal friction is a maximum at the seated position, although some friction may be present throughout the entire stroke of some soft-seat valves as a result of rubbing contact near the shaft. Eccentric-shaft designs which provide a nearly linear plugging action upon seating will usually have a lower seal friction than straight-shaft designs. Similarly, discs which seat at a relatively high α_0 will normally have a higher final closing torque requirement and a lower opening or breakaway torque requirement than the disc which seats at $\alpha_0 = 0$.

Packless butterfly valving units, or those which use retractable seats, such as the inflatable diaphragm seal (Figure 6.2.3.1e), may have no seal friction force at all.

Bearing friction forces are a function of the bearing design, bearing materials, lubrication, and the magnitude of the thrust vector transmitted by the shaft. The thrust vector is in turn dependent upon disc angle, flow or pressure forces, and unbalanced sealing forces. Although in some instances bearing friction forces may account for 50 percent of the total actuation force, in the great majority of applications bearing friction forces are appreciably lower than either seal friction or fluid forces and, unlike these other forces, bearing friction is essentially the same in either the opening or closing direction.

Fluid forces are relatively complex and are a function of the following factors:

a) Shaft eccentricity
b) Pressure differential
c) Disc shape

d) **Disc position, α**
e) **Fluid properties**
f) **Inlet passage geometry**
g) **Exit passage geometry.**

Shaft eccentricity and static pressure differential establish torque due to fluid forces in the closed position. In any disc position the torque transmitted by the disc is actually a function of the static pressure distribution across both faces of the disc, but as the disc opens this pressure distribution is dictated by the fluid dynamic condition, as illustrated in Figure 6.2.3.1g. This fluid dynamic torque acts in a direction to close the valve if the disc is symmetrical and located in a straight section of line.

Fluid dynamic torque resulting from compressible flow past a symmetrical disc can be approximated by the following empirical relationship:

$$T = p_1 d^3 c_1 c_2 \qquad \text{(Eq 6.2.3.1b)}$$

where

T = torque, in-lb$_f$

p_1 = static upstream pressure, lb$_f$/in^2

d = disc diameter, in.

c_1 = an empirically determined function of α
For the specific case of air flowing past a symmetrical disc, Reference 408-1 gives the following values for c_1

$$c_1 = 0.01942\,\alpha^2 \qquad \text{if } \alpha < 1.047$$

$$= 0.01942\,(25.66\,\alpha - 13.43 - 11.25\,\alpha^2) \qquad \text{if } \alpha > 1.047$$

α = angular rotation from closed position, radians

$$c_2 = 1 - \left(\frac{\dfrac{p_1}{p_c}\dfrac{p_2}{p_1} - 1}{\dfrac{p_1}{p_c} - 1}\right)^2 \qquad \text{if } p_2 \geq p_c$$

$$= 1 \qquad \text{if } p_2 < p_c$$

p_2 = static downstream pressure, lb$_f$/in^2

p_c = static downstream pressure for critical flow, lb$_f$/in^2

For specific butterfly valve configurations it is common practice to measure the torque coefficient, C_T, experimentally at different positions and use this in the expression

$$T = C_T\, p_{t_1} d^3 \qquad \text{(Eq 6.2.3.1c)}$$

where

T = torque, in-lb$_f$

C_T = torque coefficient, dimensionless

p_{t_1} = total upstream pressure, lb$_f$/in^2

d = disc diameter, in.

For a specific butterfly valve, the torque coefficient, C_T, of Equation (6.2.3.1c) is usually presented as a function of both disc angle α and pressure ratio across the valve. In some instances, particularly where the pressure ratio does

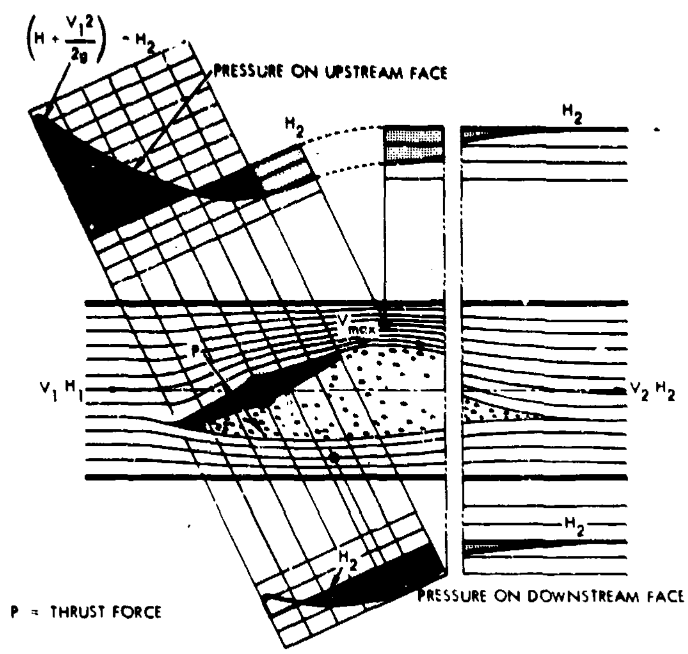
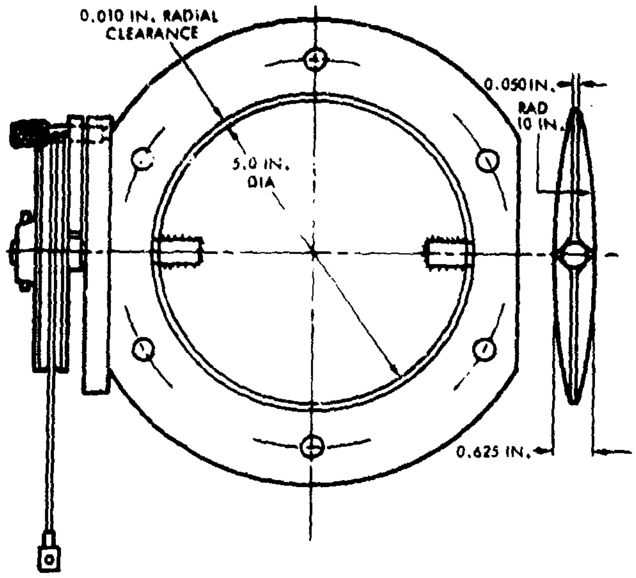


Figure 6.2.3.1g. Diagram of Pressure Generated Around a Butterfly Valve Disc
(Adapted from "Water Power," December 1951, vol. 3, no. 12, D. Gaden, Copyright 1951, Tothill Press Ltd., London, England)

not exceed the critical, C_T is presented only as a function of α and the static pressure differential across the valve Δp is substituted for p_{t_1} in Equation (6.2.3.1c). Reference 201-1 presents detailed results of tests performed with air upon a typical, packless, lenticular-disc, straight-shaft butterfly valving unit which shows the effect of significant changes in pressure ratio, disc position, and passage geometry. Figures 6.2.3.1h, i, and j show these results. It is interesting to note that with no inlet or exit pipe (see Figure 6.2.3.1i) there is a strong torque, tending to open the valve at high pressure ratios when disc angle α is 30 degrees or more. These to characteristics are of major significance when it is ed to use a butterfly valving unit at high pressure atios and/or in a location other than in a straight line. Although no data is available to illustrate the point, it may be expected that the orientation of bends in the inlet and exit passages relative to disc shaft orientation may influence torque and flow characteristics. The inlet pipe configuration used for Figure 6.2.3.1j includes a double bend (crank-shaped bend) to accommodate a 4½-inch offset within 18 inches of length, but this bend is in the plane of the disc shaft and its influence will therefore be expected to be minimal. Reference 273-2 gives the following experimental results showing the effect on torque coefficient of a 90-degree bend of



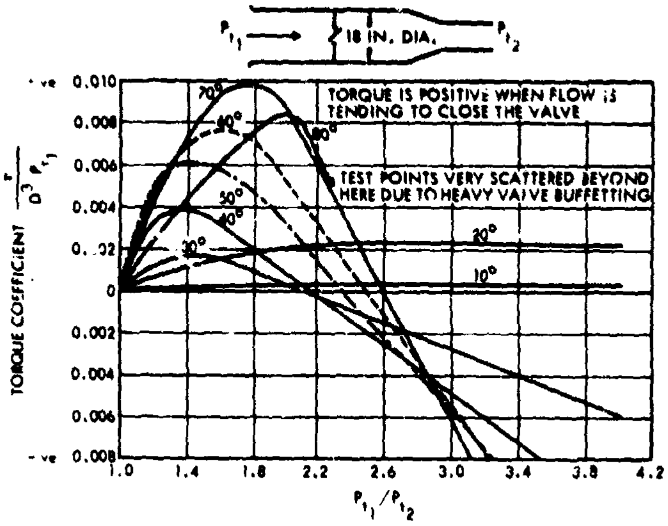


Figure 6.2.3.1i. Torque Characteristics of Butterfly Valve in Bulkhead

(Adapted, by permission of the Controller of Her Britannic Majesty's Stationery Office, from NGTE Memo No. 304, July 1957, A. H. Robinson, Copyright 1957, Great Britain National Gas Turbine Establishment, Pyestock, Hants, England)

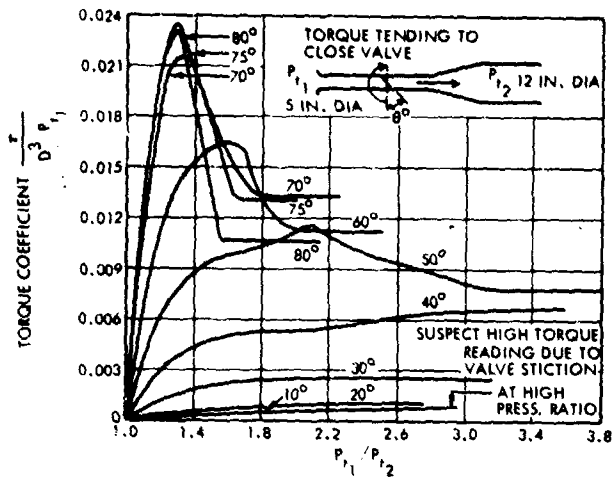


Figure 6.2.3.1h. Torque Characteristics of Butterfly Valve in Pipe

(Adapted, by permission of the Controller of Her Britannic Majesty's Stationery Office, from NGTE Memo No. 304, July 1957, A. H. Robinson, Copyright 1957, Great Britain National Gas Turbine Establishment, Pyestock, Hants, England)

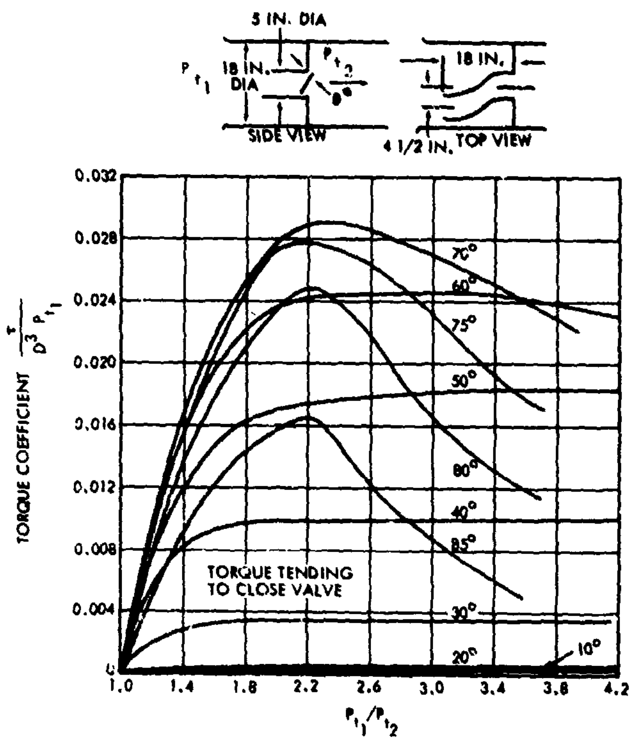


Figure 6.2.3.1j. Torque Characteristics of Butterfly Valve in Bulkhead with Inlet Pipe

(Adapted, by permission of the Controller of Her Britannic Majesty's Stationery Office, from NGTE Memo No. 304, July 1957, A. H. Robinson, Copyright 1957, Great Britain National Gas Turbine Establishment, Pyestock, Hants, England)

radius D (one pipe diameter) in a plane perpendicular to that of the disc shaft:

	Length of Straight Line Ahead of Disc	Percent Maximum C_T
No bend in line	10D	100
Bend in line	10D	94
Bend in line	6D	87

These tests (performed with airfoil-shaped discs) indicate that some airfoil shaped discs have a negative (opening) torque coefficient in the full-open position (α = 83 degrees to 90 degrees). Identical tests with flat and lenticular discs showed zero torque coefficient when full open and a positive (closing) value at all other positions except full closed when the value again falls to zero, in accordance with accepted theory.

The total external torque required to position the disc is the algebraic sum of the torque resulting from seal friction, bearing friction, and fluid forces, and at any disc angle will usually be greater in the opening direction than in the closing direction (see Figure 6.2.3.1k). It should be noted that most experimental data on butterfly valve torque (see references at end of this sub-section) are concerned with fluid force torque and the experimental valves used were packless units with low-friction bearings. For this reason most of the available experimental torque data (such as Figures 6.2.3.1h, i, and j) on various butterfly valving unit configurations show only one torque coefficient value for given values of α and Δp, with no indication of directional variations between opening and closing torques such as shown in Figure 6.2.3.1k. Therefore, careful consideration should be given to these valve/actuator interactions as treated under Detailed Topic 5.3.5.5 as well as to the details of valving unit design treated in this detailed topic.

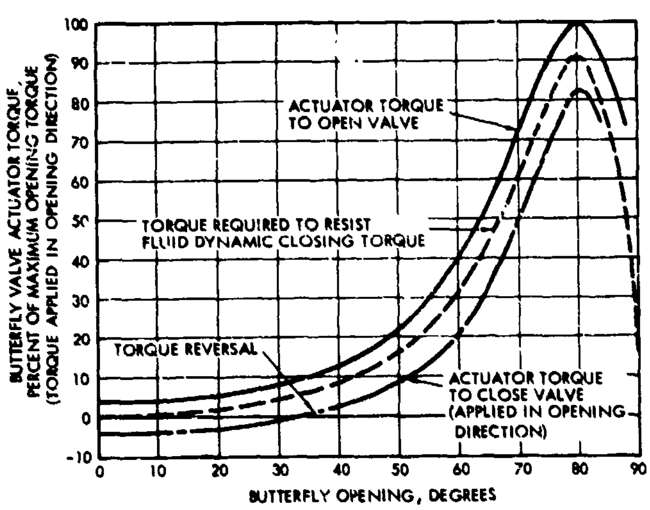


Figure 6.2.3.1k. Fluid Dynamic Torque Curve for a Typical Butterfly Valve
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

In more critical aerospace applications, such as the rocket propellant valving units shown in Figures 6.2.3.1a and b, the thick discs and large eccentric offsets frequently employed may be expected to result in torque characteristics that differ appreciably from those using more conventional disc configurations. As the disc shape becomes more complex, it becomes even more difficult to analyze torques theoretically or to correlate data from conventional disc shapes. In such cases it will frequently be found profitable to measure the torque characteristics of the design under consideration experimentally, preferably using full-scale models or actual prototype hardware.

Flow Characteristics. A discussion of butterfly valve flow characteristics is presented in Detailed Topic 5.3.5.5; Figures 6.2.3.1l and m illustrate the basic flow characteristic and present resistance coefficients for conventional butterfly valves. This information is adequate for a basic appreciation of butterfly valve flow characteristics and the influence of disc angle and thickness, but the valving unit designer concerned with meeting specific flow requirements should also consider such factors as valve seating angle, α_0, the effects of adjacent ducting, and the total upstream and downstream pressure. Although the sources of flow characteristic data referenced at the end of this sub-section have used various flow media and units, a number have expressed flow in terms of some form of the discharge coefficient, C_D.

$$Q = C_D A_0 \sqrt{2g \frac{\Delta P}{w}} \qquad \text{(Eq 3.8.1f)}$$

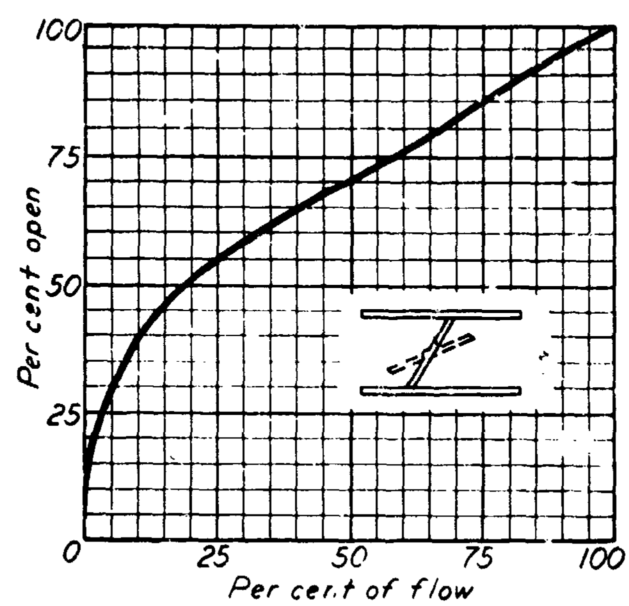


Figure 6.2.3.1l. Butterfly Valve Flow Characteristics
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

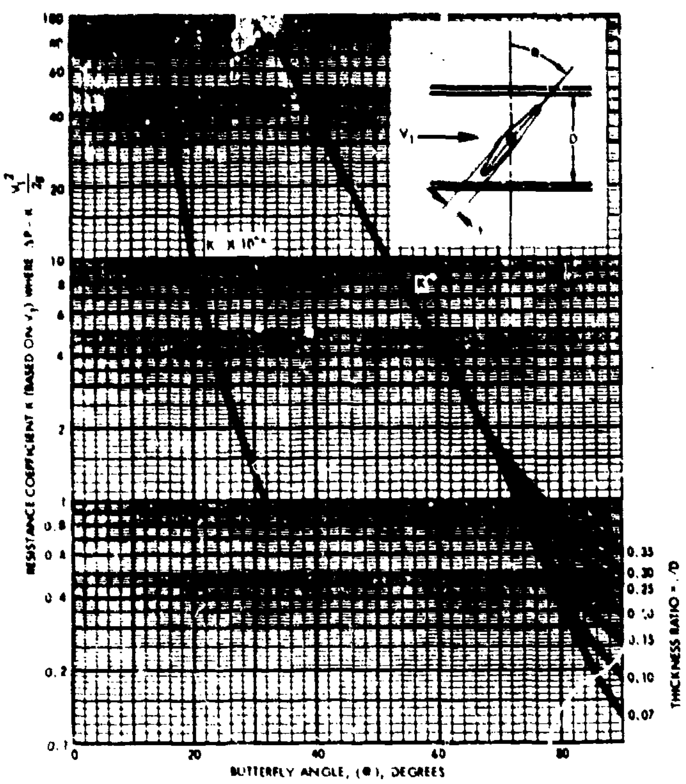


Figure 6.2.3.1m. Butterfly Valve Resistance Coefficient as a Function of Disc Angle
(Adapted from "Machine Design," December 1957, vol. 30, no. 26, D. Dar ie, Copyright 1958, Penton Publishing Company, Cleveland, Ohio)

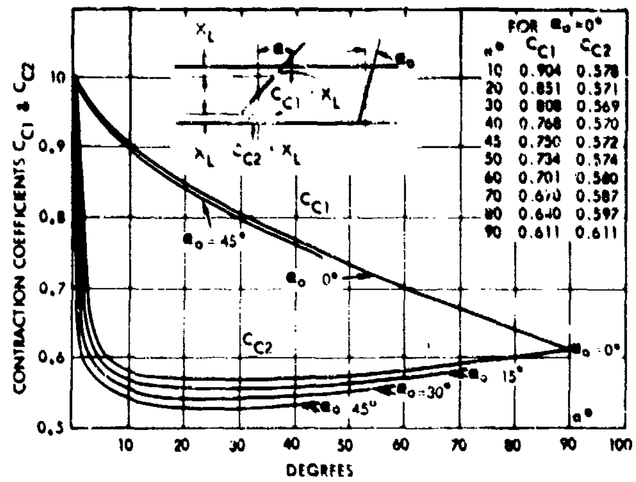


Figure 6.2.3.1n. Butterfly Valve Contraction Coefficients
(Adapted from ASME Paper 60-WA-105, T. Sarpkaya, Copyright 1960, American Society of Mechanical Engineers, New York, New York)

where

Q = volumetric flow rate, ft³/sec

C_D = discharge coefficient, dimensionless

A_0 = equivalent orifice area (usually taken as cross-section area of the line), ft²

ΔP = pressure drop across valve measured between points where upstream and downstream diameters are equal, lb_f/ft^2

w = specific weight, lb_f/ft^3

As explained in Detailed Topic 6.2.2.4, C_D is usually determined experimentally; it has, however, been expressed analytically for butterfly valves by Sarpkaya (Reference 68-4) using free-streamline analysis methods. In the case of free discharge into the atmosphere

(Eq 6.2.3.1d)

$$C_D = \frac{\pi}{4\sqrt{2}}(C_{c_1} + C_{c_2})\left(1 - \frac{\cos\alpha}{\cos\alpha_0}\right)$$

For axisymmetric continuous flow in a pipe of constant diameter

(Eq 6.2.3.1e)

$$C_D = \frac{\frac{\pi}{4\sqrt{2}}(C_{c_1} + C_{c_2})\left(1 - \frac{\cos\alpha}{\cos\alpha_0}\right)}{1 - \frac{1}{2}(C_{c_1} + C_{c_2})\left(1 - \frac{\cos\alpha}{\cos\alpha_0}\right)}$$

where

C_{c_1}, C_{c_2} = contraction coefficient (see Figure 6.2.3.1n)

α = angle of opening

α_0 = angle of complete closure (seating angle)

It is apparent that if the line diverges or is other than cylindrical in the immediate vicinity of the valving unit, the contraction coefficient may be expected to vary significantly. The effects of variations in the inlet and exit conditions may be observed in the results of Robinson's tests with air at pressure ratios up to and exceeding the critical (Figures 6.2.3.1o, p, and q). It should be noted that a flow coefficient, rather than a discharge coefficient, is plotted in these curves. This flow coefficient is based on air and assumes downstream pressure to be atmospheric. Note the use of *total* pressure rather than static pressure.

$$\text{Flow Coefficient} = \frac{\dot{m}\sqrt{T}}{A\,P_{t_1}} \qquad \text{(Eq 6.2.3.1f)}$$

where

$\dot{m}$ = air flow rate, lb_m/sec

T = absolute temperature, °K

P_{t_1} = total pressure upstream of valve, lb_f/in^2

A = cross-sectional area of valve, in²
(Note: not the projected flow area)

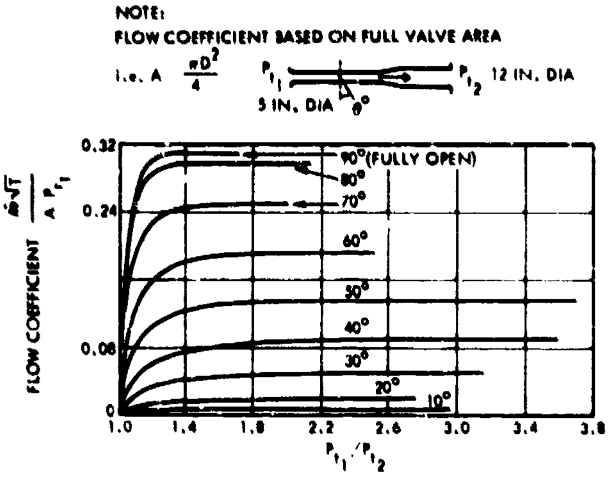


Figure 6.2.3.1o. Flow Characteristics of Butterfly Valve in Pipe

(Adapted, by permission of the Controller of Her Britannic Majesty's Stationery Office, from NGTE Memo No. 304, July 1957, A. H. Robinson, Copyright 1957, Great Britain National Gas Turbine Establishment, Pyestock, Hants, England)

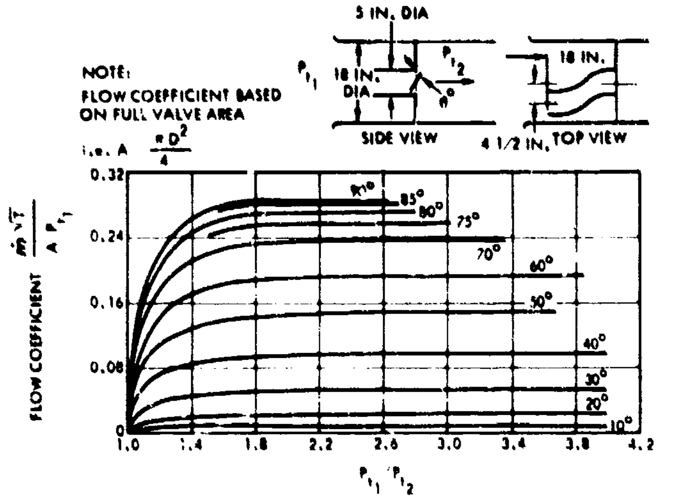


Figure 6.2.3.1q. Flow Characteristics of Butterfly Valve in Bulkhead with Inlet Pipe

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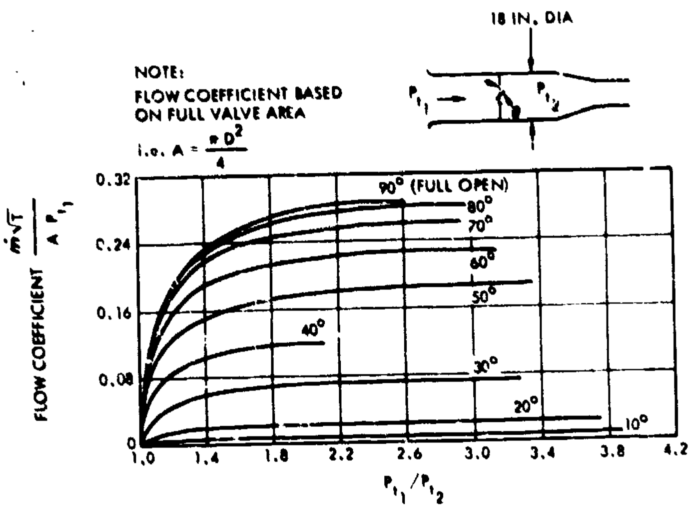


Figure 6.2.3.1p. Flow Characteristics of Butterfly Valve in Bulkhead

(Adapted, by permission of the Controller of Her Britannic Majesty's Stationery Office, from NGTE Memo No. 304, July 1957, A. H. Robinson, Copyright 1957, Great Britain National Gas Turbine Establishment, Pyestock, Hants, England)

Finally, it is necessary to consider the influence of the specific values of upstream and downstream pressure and the flow characteristics of a butterfly valving unit. It is customary to divide butterfly valving unit applications into two classifications. The first class of valves includes those discharging to a downstream pressure which is greater than atmospheric due to some resistance further downstream. The second class of valves includes those with a downstream pressure equal to or less than atmospheric, and whose delivery under a given upstream pressure depends only upon the angle of the disc. Figures 6.2.3.1r and s present the flow coefficients obtained by various investigators with butterfly valves of the two classifications. The flow coefficients in this instance, C_Q and C_Q'', are simply a variation of the discharge coefficient C_D adaptation of the orifice equation, Equation 3.8.1f.

For valves discharging to pressures above atmospheric (first class of valves)

$$Q = C_Q D^2 \sqrt{\frac{\Delta P}{\rho}} \qquad \text{(Eq 6.2.3.1g)}$$

where

Q = flow rate, ft^3/sec

C_Q = flow coefficient, empirical

D = valve diameter, in.

ΔP = static pressure differential, lb_f/in^2

ρ = density, lb_m/in^3

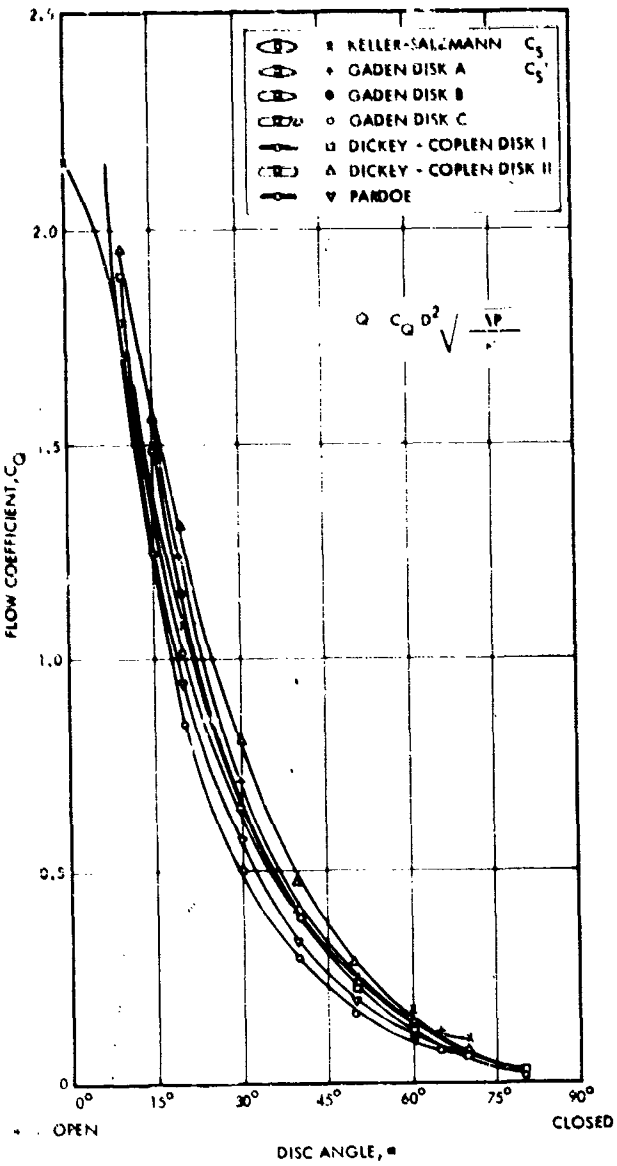


Figure 6.2.3.1r. Flow Coefficients for First Class of Butterfly Valves

(Adapted from "Instruments," August 1951, vol. 24, no. 8, S. D. Cohn, Instruments Publishing Co., Inc., Pittsburgh, Pennsylvania)

For valves discharging to atmospheric pressure or below (second classification)

$$Q = C_Q'' D^2 \sqrt{\frac{\Delta P'}{\rho}} \qquad \text{(Eq 6.2.3.1h)}$$

where

C_Q'' = flow coefficient, empirical

$\Delta P'$ = total pressure differential, lb_f/in^2

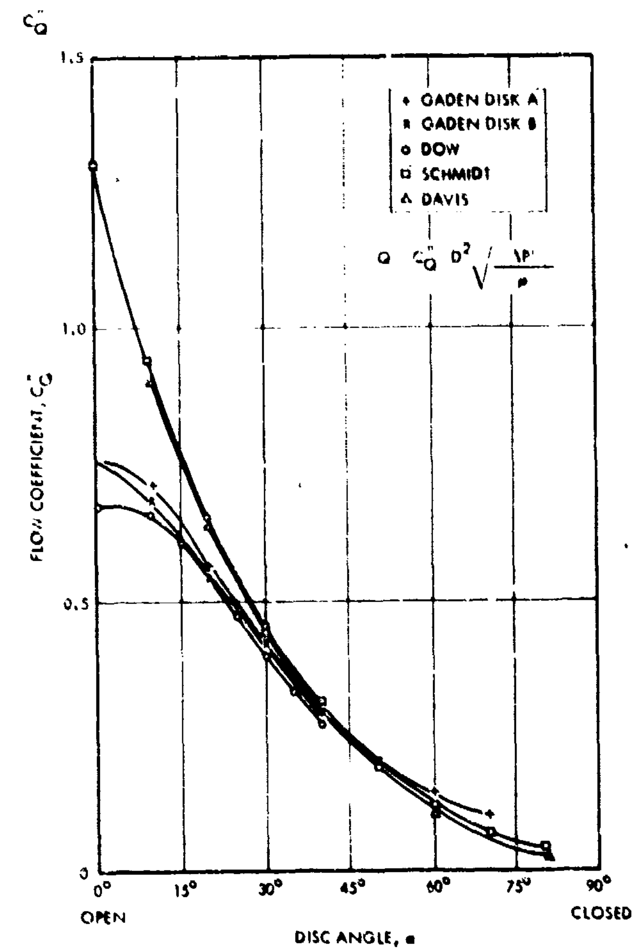


Figure 6.2.3.1s. Flow Coefficients for Second Class of Butterfly Valves

(Adapted from "Instruments," August 1951, vol. 24, no. 8, S. D. Cohn, Instruments Publishing Co., Inc., Pittsburgh, Pennsylvania)

6.2.3.2 BALL. The ball valving unit is somewhat similar to the butterfly in that rotation of the valving element within the housing either permits or obstructs flow. The ball valving element is spherical, however, and accommodates flow directly through a hole in the ball, rather than requiring the fluid to flow around the valving element. Shutoff ball valves are described in Sub-Topic 5.2.3 and ball valves suitable for either shutoff or flow diversion are discussed in Detailed Topic 5.8.5.1.

Specific definition of ball valving unit configuration usually is a function of the following parameters:

a) *Bore direction.* Although straight-through porting of the ball is most common, multidirectional balls are used in multiple-passage valve applications. Figure 6.2.3.2a illustrates both a ball with a 90-degree turn bore and a multiple-passage ball.

b) *Valving element support.* The ball can be either fixed or floating as shown in Figures 6.2.3.2b and c, respectively. The fixed-ball configuration uses either bearings on both sides of the ball or two sets of bearings on one side of the ball to provide cantilever support, the former arrangement being much more common. The floating-ball configuration relies upon the valve seat for bearing support and is usually found in lower cost, less critical applications.

c) *Valving element shape.* The simplest ball valving units are solid spheres with a cylindrical bore, but a great many variations of this simple configuration are used. Flats are frequently machined on one or more sides of

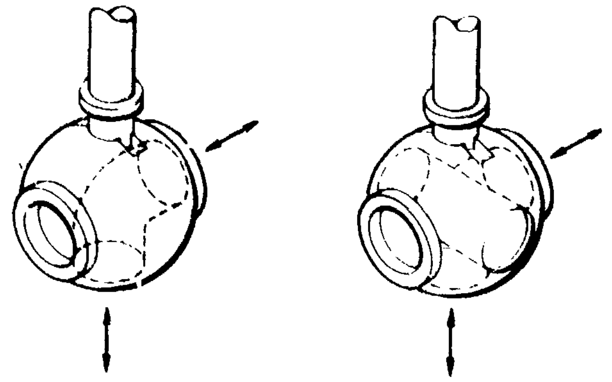


Figure 6.2.3.2a. Multiported Ball Valves
(Courtesy of Pacific Valves, Inc., Long Beach, California)

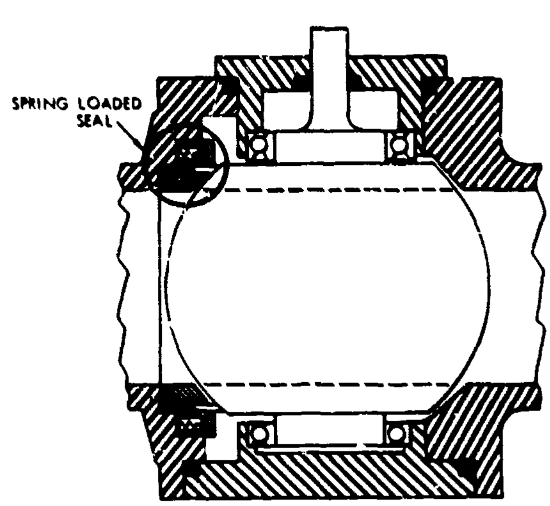


Figure 6.2.3.2b. Full-Flow Ball Valve with Fixed Ball

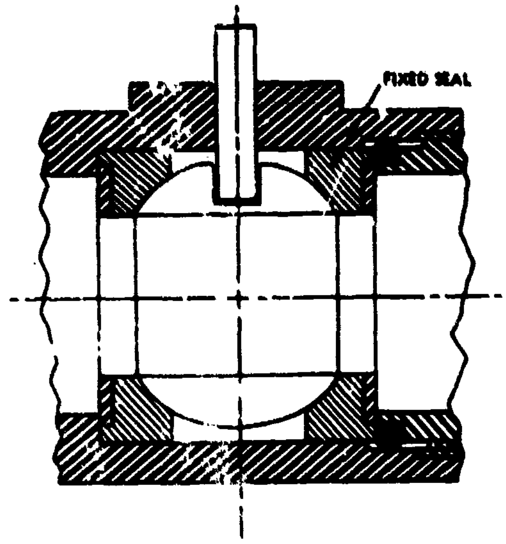


Figure 6.2.3.2c. Reduced-Port Ball Valve with Floating Ball

the ball, as illustrated in Figures 6.2.3.2d and e. This may be done to permit flow around as well as through the valving element in the partially closed position, reduce envelope requirements, reduce torque requirements, and/or reduce weight. In the case of the plastic ball valving unit of the type shown in Figure 6.2.3.2a, the flats also serve to add to the resiliency of the seal portion of the ball. Figure 6.2.3.2b also illustrates how the ball sides may be machined to accommodate ball or roller bearings and thereby reduce overall valve envelope dimensions. An extreme variation of ball shape is found where hollow-ball valving elements consisting of only about ¼ of the total sphere area are used: these are often called *visor valves.*

d) *Bore-to-ball diameter ratio.* The ratio of the diameter of the hole through the ball to the ball diameter will determine the minimum angle of rotation of the ball necessary to effect valve closure. Figure 6.2.3.2f shows this relation graphically for cylindrical ports whose centerline is coincident with the center of the ball. For example, a ½-inch hole in a 1-inch ball requires 60 degrees of rotation of the ball to effect closure. While the curve in Figure 6.2.3.2f shows the minimum rotation of the ball to close the valve, sealing considerations may necessitate further rotation.

e) *Bore-to-line diameter ratio.* The size of the bore, or hole, in the valving element, relative to diameter of the line is frequently used in describing the valving unit. In order to reduce weight, envelope, and actuator requirements, it is common practice to take advantage of the inherent low pressure drop characteristic of the ball valve and install a small ball valving unit in a venturi section as illustrated in Figure 6.2.3.2g. Small-bore ball valving units are often used without venturi sections in industrial applications where cost or length constraints are more important than pressure-drop considerations (Figure 6.2.3.2c). Such a small-bore ball valve is usually referred to as having *reduced trim* or as a *reduced port* valve, as opposed to the *full-port* valve whose bore is essentially the same diameter as the line.

Sealing. Ball valves, particularly those of the fixed-ball type, most commonly seal by means of plastic or elastomeric seals, often spring-loaded, located in the valve

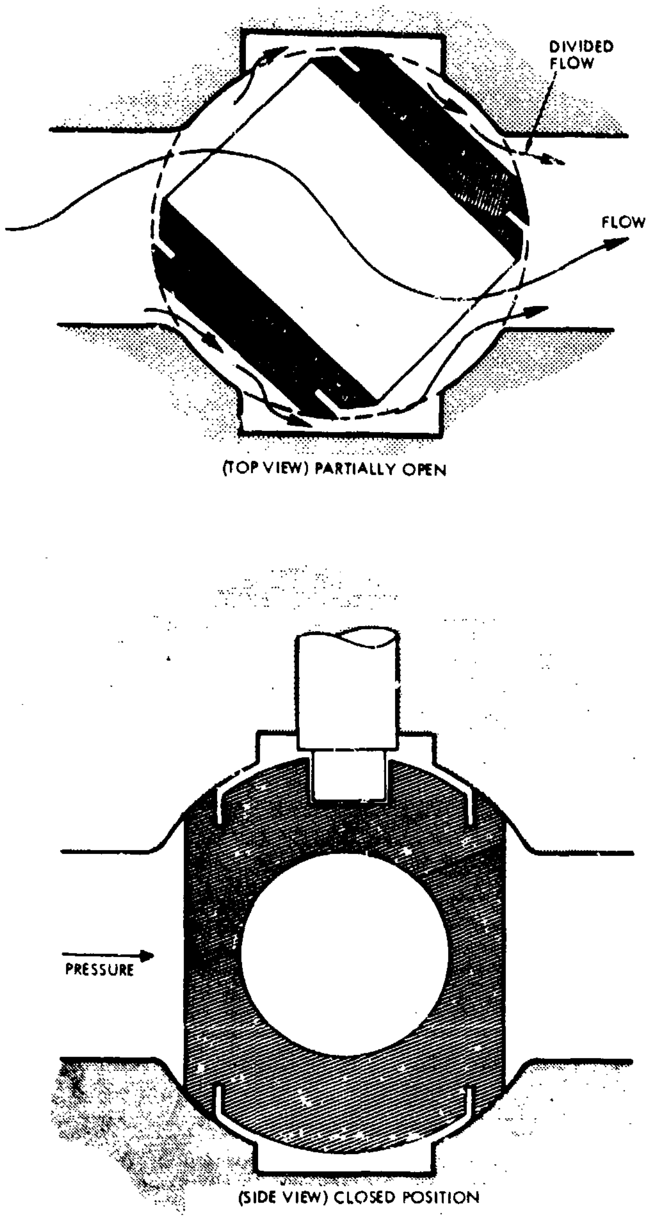


Figure 6.2.3.2d. Plastic Ball Valving Element with Integral Seals

(Adapted from "Product Engineering," 21 December 1964, vol. 35, no. 26, R. E. Sanctuary, Copyright 1964, McGraw-Hill Publishing Company, Inc., New York, New York)

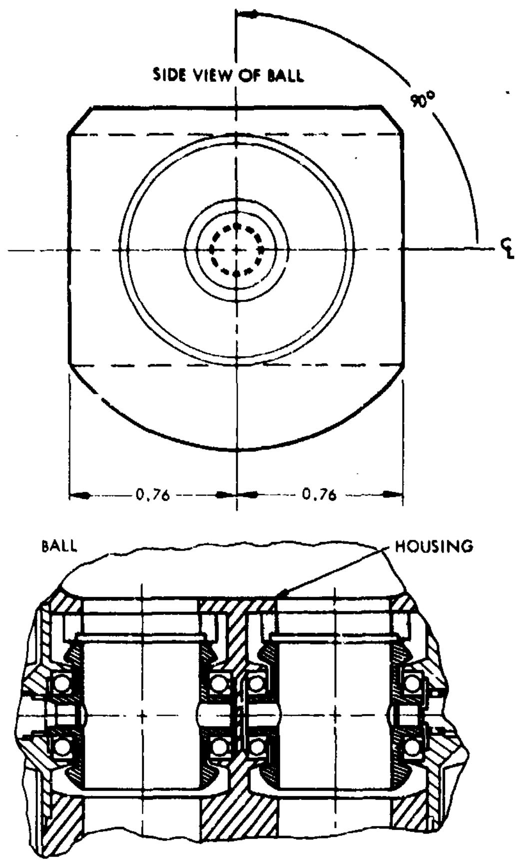


Figure 6.2.3.2e. Multiple Ball Valving Units in Common Housing

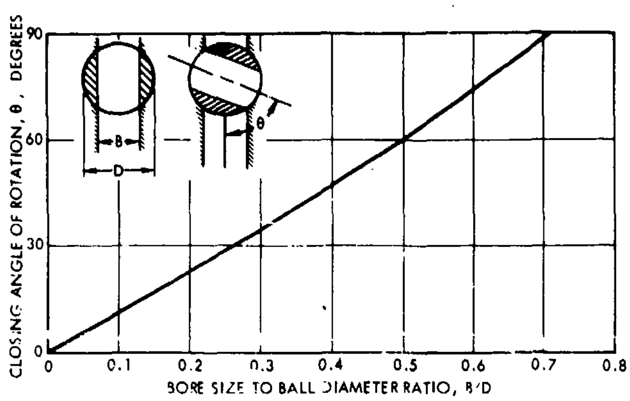
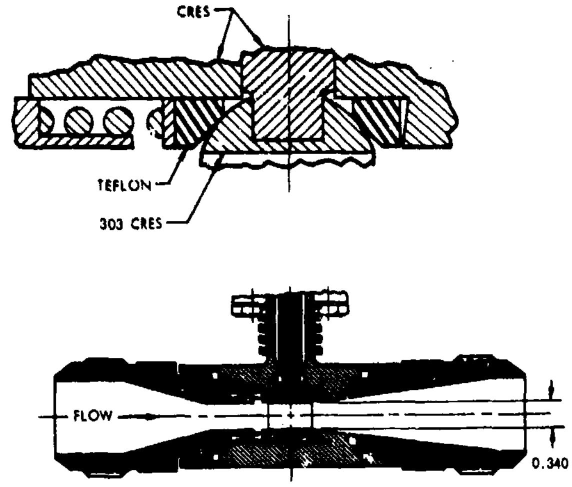


Figure 6.2.3.2f. Angle of Rotation Required to Close a Ball Valve



DESIGN PARAMETERS

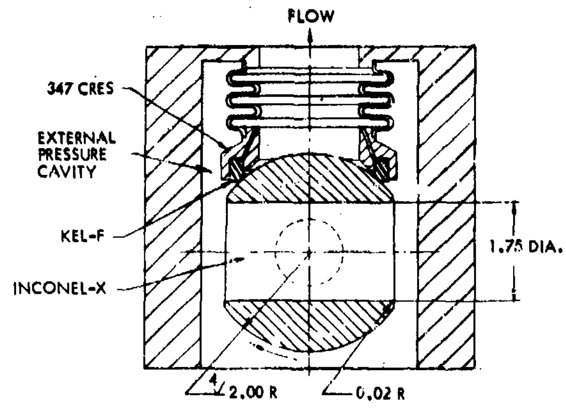
FLOW MEDIA: LIQUID HYDROGEN OR
LIQUID OXYGEN
OPERATING PRESSURE: 50 PSIG
OPERATING TEMPERATURE: -423 to 160 F
SIZE: 0.340-IN. HOLE
TUBE SIZE: 1 IN.

Figure 6.2.3.2g. Cryogenic Ball Valve in a Venturi Section
(Reference 35-12; Clary Dynamics Corporation design, p. B-82)

housing around the inlet and/or outlet ports. Figures 6.2.3.2b and e illustrate such a seal around one port, whereas Figure 6.2.3.2g shows a fixed Teflon seal around the outlet port, and a spring-loaded Teflon seal around the inlet port. Figure 6.2.3.2h illustrates the use of a bellows whose effective pressure diameter is slightly smaller than the ball-contact diameter of the Kel-F seal, to provide pressure actuation in addition to the spring force of the bellows. Note the conical seal configuration, which results in initial line contact between the ball and seal until pressure and spring force cause the plastic seal to deform, increasing the contact area. Ball valves for rocket engine propellant shutoff, such as those shown in Figures 6.2.3.2e and 6.2.3.2h normally specify a 4-microinch (A A) ball finish usually on a hardened or coated surface, and ball sphericity in the order of 0.0005 inch, while the mating plastic seal has a 16-microinch (A A) surface. In addition, the edge of the bore is usually shaped to avoid cutting the seal, as shown in Figure 6.2.3.1i.

Teflon (plain or reinforced) is most commonly employed as the seat material for aerospace applications although ball valves are also produced with seals of Kel-F, nylon, Delrin, Buna-N, Butyl, Neoprene, silicone, and Viton-A.

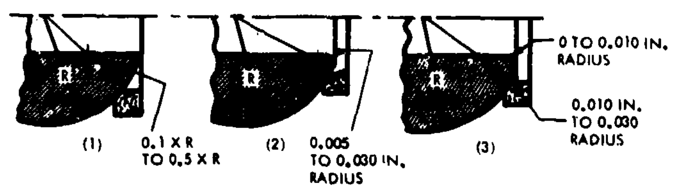
Metal or graphite seats are occasionally used to meet critical requirements, as in the case of the 12-inch diameter, 30,000 psi, 600°F ball valve described in Reference 73-100, which employs an aluminum-bronze seal ring. Figure 6.2.3.2j illustrates some of the temperature and pressure regimes wherein various seal materials are employed.



DESIGN PARAMETERS

FLOW MEDIA: LIQUID OXYGEN
OPERATING PRESSURE: 1800 PSIG
OPERATING TEMPERATURE: -320 TO 160 F
SIZE: 1.75 IN. BORE

Figure 6.2.3.2h. High-Pressure Cryogenic Ball Valve
(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-80)



NOTE THAT ARRANGEMENT (2) PERMITS
USE OF SEAL WITH SMALLEST ID

Figure 6.2.3.2i. Shaped Bore Edges to Minimize Seal Damage by Ball

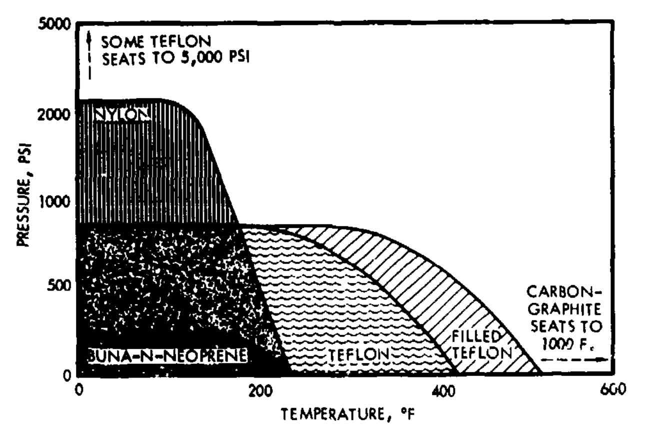


Figure 6.2.3.2j. Temperature-Pressure Rating for Common Ball Valve Seal Materials
(Reprinted with permission from "Chemical Engineering," 13 May 1963, vol. 70, no. 10, D. S. Antrim, Copyright 1963, McGraw-Hill Publishing Company, Inc., New York, New York)

Other variations in the sealing techniques employed in ball valving units include:

a) Use of an eccentric cam on the actuator shaft to translate the valving element (ball or visor segment) axially away from the fixed seat prior to rotating the valving element (References 74-22 and 36-13.) (This also provides an axial poppet-type seating action upon closing and has recently been used with success on advanced liquid rocket engines.)

b) Use of cams to retract seal(s) from ball prior to rotating the ball

c) Permitting seals to rotate slightly within the housing each time the ball is turned, to distribute seal wear around the seal periphery (seat wear is concentrated at the point where flow is initiated upon opening or pinched off upon closing) (Reference 74-22)

d) Inclining seals at an angle in the housing to obtain a wedge-type action similar to that found in rotary-plug valves (Reference 74-22)

e) Use of a plastic ball with integral seals which seal against the housing, as illustrated in Figure 6.2.3.2d

f) Use of O-rings as seals for both fixed-ball and floating-ball valving units.

Actuation Forces. Actuation forces for ball valves must overcome bearing friction, seal friction, and fluid forces, as in the case of butterfly valves. Bearing and seal friction forces are combined in floating-ball designs, but the following discussion will treat the fixed-ball configuration.

Bearing Friction Forces. Because ball valves are pressure-unbalanced, bearing friction torque is dependent upon the bearing friction characteristics and the thrust transmitted through the bearing. Maximum thrust usually occurs when the valve is closed and is a function of effective seat diameter and pressure differential across the valving element. Where bearing loads are light enough to permit the use of rolling-element bearings, as in Figure 6.2.3.2e, bearing friction may be significantly reduced. (Compare the 0.0011 friction coefficient for roller bearings from Table 6.8.3.1c with the 0.04 value for Teflon from Table 12.7.) Rolling-element bearings also provide precise location of the valving element relative to seals and ports. Reference 34-11 gives the following rule-of-thumb expression for ball valve bearing friction torque:

$$\tau_B = 1.75 \times 10^{-3} \Delta p^{\frac{3}{2}} d^4 \mu_B \qquad \text{(Eq 6.2.3.2a)}$$

where

τ_B = bearing friction torque, in-lb_f

Δp = pressure differential across ball, lb_f/in^2

d = seal diameter (assumed equal to port diameter), in.

μ_B = bearing friction coefficient, dimensionless

Figure 6.2.3.2k relates ball valve size, pressure, and bearing friction torque for valves using sleeve bearings with a friction coefficient of 0.15.

Seal Friction Forces. **The torque required to overcome seal friction in ball valving units is more difficult to analyze than bearing friction torque and is a function of the following:**

a) Seal type (degree of pressure-actuation)

b) Percent of ball surface swept by seal (the use of flats on the ball reduces seal friction during part of stroke)

c) Seat diameter/ball diameter ratio

d) Seat and ball materials

e) Shaft seal friction characteristics.

Reference 34-11 presents the following rule-of-thumb expression for estimating seal friction torque:

$$\tau_s = 0.625 \Delta p d^3 \mu_s \qquad \text{(Eq 6.2.3.2b)}$$

where

τ_s = seal friction torque, in/lb_f

Δp = static pressure differential across seal, lb_f/in^2

d = seal diameter, in.

μ_s = seal (seat) friction coefficient, dimensionless

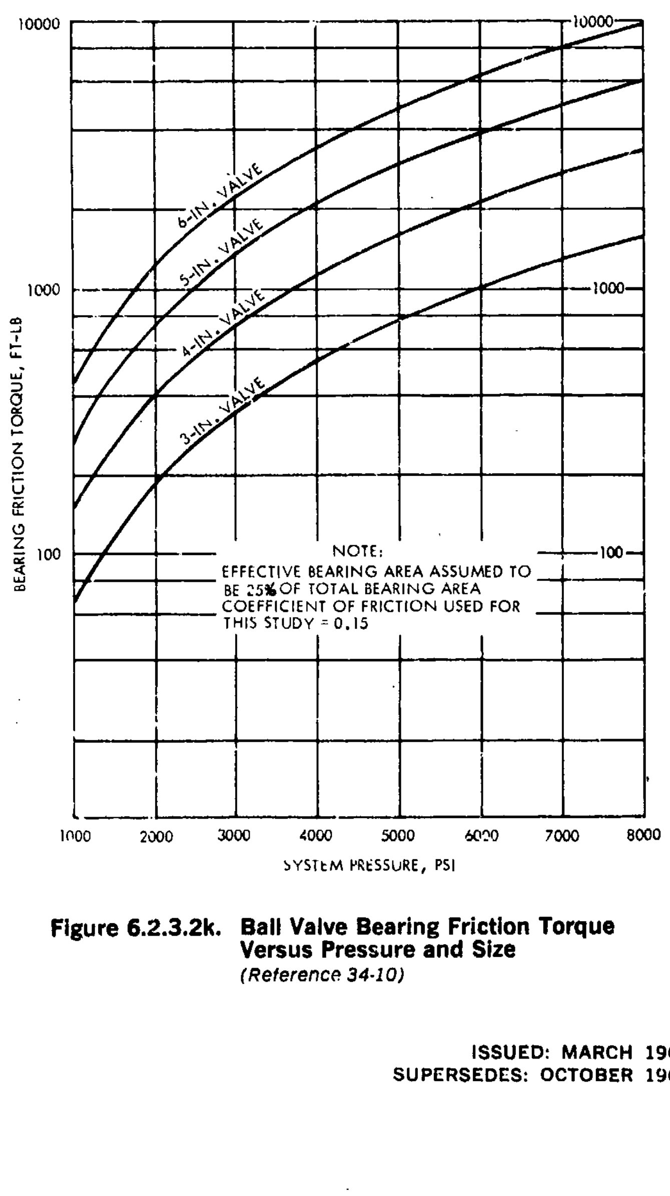


Figure 6.2.3.2k. Ball Valve Bearing Friction Torque Versus Pressure and Size
(Reference 34-10)

ISSUED: MARCH 1967
SUPERSEDES: OCTOBER 1965

Fluid Forces. The torque required to compensate for fluid forces (other than the static pressure differential which affects bearing and seal forces) is usually small in ball valving units. The fluid dynamic torque acts in a direction to close the valve (similar to a butterfly valve) and reaches a maximum between 60 degrees and 80 degrees open.

Figure 6.2.3.21 shows the torque characteristics of the valving unit illustrated in Figure 6.2.3.2d. It may be seen that fluid dynamic forces are apparently negligible compared with bearing and seal friction for this particular design.

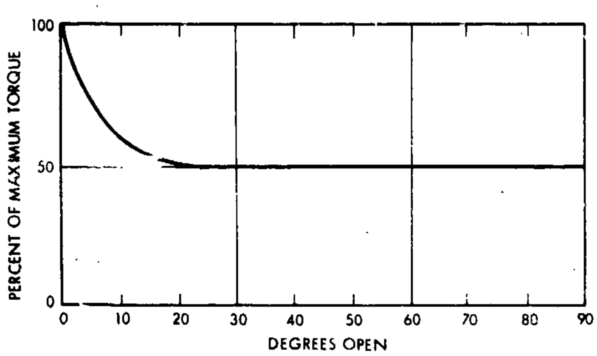


Figure 6.2.3.2l. Torque Characteristics for Ball Valve Illustrated in Figure 6.2.3.2d
(Adapted from "Product Engineering," 21 December 1964, vol. 35, no. 26, R. E. Sanctuary, Copyright 1964, McGraw-Hill Publishing Company, Inc., New York, New York)

Flow Characteristics. The flow characteristic of primary interest in most ball valve applications is the extremely low pressure drop in the full-open position. With a full-port ball and close-fitting seals on both inlet and outlet, resistance to flow is essentially equal to that of an equal length of line. Flow resistance in the partially-open position is a function of (a) the angle between the axis of the ball bore and the axis of the ports, or ball position; (b) the bore diameter/ball diameter ratio; (c) the seal configuration, particularly seal diameter; and (d) the presence of flats on the ball sides which permit flow around the ball (see Figure 6.2.3.2d).

Figure 6.2.3.2m relates flow to valving element position for a typical ball valving unit. Although the flow characteristic may be seen to approximate an equal percentage or parabolic characteristic (Figure 6.2.2.2a), ball valves are not particularly well suited for throttling service because of the risk of damage to seals if the valving element is left in an intermediate position. In critical shutoff applications, such as rocket engine propellant valves, where the flow characteristic influences the engine start and shutdown transients, knowledge of the flow characteristic is essential even though throttling is not involved.

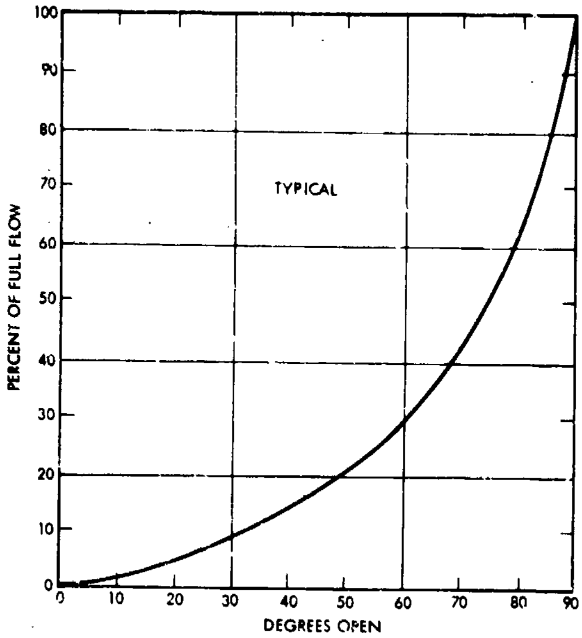


Figure 6.2.3.2m. Ball Valve Flow Characteristics
(Reference 34-10)

6.2.3.3 ROTARY PLUG. Like the ball valve, the rotary-plug valving element rotates within the housing to permit flow through a hole in the plug. Sub-Topic 5.2.11 describes the three basic types of rotary-plug valves as the simple cock, the lubricated plug, and the non-lubricated plug. Detailed Topic 5.3.5.7 treats the use of rotary-plug valves in flow-control service.

Configuration. The conical shape of the plug, or valving element, of a rotary plug valving unit usually dictates a larger envelope and greater weight than a ball valving element of comparable port area. The rotary-plug valving unit is closely comparable to the floating-ball valving unit and its configuration may be described in terms of the following parameters: (a) port shape (both plug and housing), (b) cone angle, and (c) seat type. Port shape is usually round (giving flow performance similar to a ball valving unit), rectangular (to provide maximum flow area with minimum cone width), or V-shaped (for flow control). Cone angle significantly influences operating torque, sealing efficiency, and plug-drop (as the seat wears); the optimum angle for a specific design application usually is determined experimentally during development. Figure 6.2.3.3a illustrates the effect of plug angle or plug drop for a non-lubricated plug with a Teflon seat. Seat type or sealing technique has the greatest effect on plug valve configuration when provisions are incorporated to lift the plug from the seat to reduce friction loads while rotating the plug.

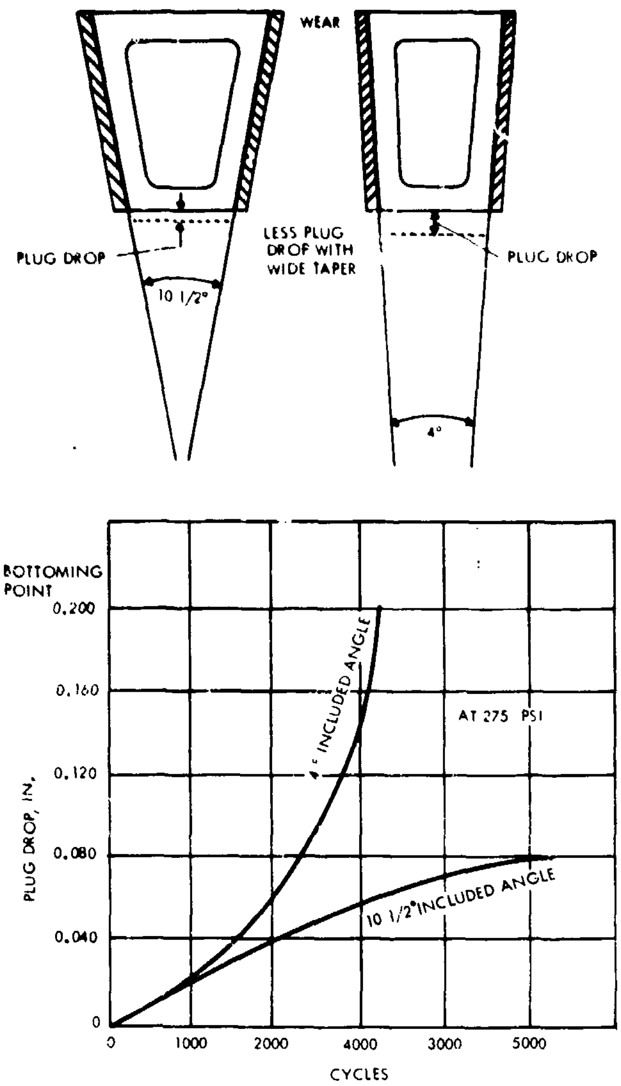


Figure 6.2.3.3a. Rotary Plug Cone Angle
(Adapted from "Product Engineering," 28 September 1964, vol. 35, no. 20, H. O'Connor and J. Hawley, Copyright 1964, McGraw-Hill Publishing Company, Inc., New York, New York)

Sealing. Sealing techniques are discussed in Sub-Topic 5.2.11 for each of the three types of rotary-plug valves. Rather than using spring-loaded or pressure-actuated seals (as in fixed-ball valves), or permitting fluid pressure to force the valving element into the seat (as with floating-ball valves), plug valves usually utilize a force normal to the port axis to force the plug into the conical seat. This force must be sufficient to overcome the unbalanced pressure force tending to lift the plug from the seat in the open position as well as to provide adequate seating stress in the closed position. The greater the included angle of the plug taper, the greater will be this force requirement. Surface finish requirements tend to be slightly less severe for plug valves than for ball valves. The Teflon seat design shown in Figure 6.2.3.3a was found to work satisfactorily with a plug finish of 8.10 microinches (rms) and to provide a satisfactory static seal in a housing of approximately 63 microinches (rms) surface finish (Reference 19-235).

Actuation Forces. Rotary-plug valving units are generally considered to have the highest actuating force requirements of any of the common valve types (References 112-8 and 160-18). Because the valving element is usually supported by the seat, bearing and seal friction is combined and generally completely overshadows fluid-dynamic forces. Actuation torque requirements then become primarily a function of cone angle, axial force on plug, and friction coefficient between plug and seat.

Reference 19-235 indicates that the change in cone angle from 4 degrees to 10½ degrees, shown in Figure 6.2.3.3a, reduced torque requirements by 40 percent. Bearing and seal friction loads can be limited to the loads produced by fluid pressure forcing the plug against the downstream port, if the plug is lifted slightly, prior to rotation. The reduced friction load is then comparable to that of a free-floating ball valving unit. If shafts and bearings are provided on both ends of the plug, it is possible to raise the plug completely away from the seat and reduce loads to bearing friction and fluid dynamic forces. If lifting forces are obtained only by a cam action associated with rotation of the plug, the actuation force requirement will initially be greater than if no lift were provided (this lift technique is normally used to reduce seat wear rather than to reduce actuating torques). Reduction in actuation torque requires lifting the plug prior to commencing plug rotation, as provided for in some lubricated plug valves.

Flow Characteristics. Plug valve flow characteristics are somewhat more readily tailored to flow-control requirements than are ball valves. A compromise is entailed between the very low full-open pressure drop of the full-port valve with the characteristic shown in Figure 6.2.3.3b, and any other port shape, such as the V-port shown in Figure 6.2.3.3c, which has a nearly parabolic flow characteristic. The use of any design other than the full port may be expected to result in a somewhat higher full-open resistance. If the plug is designed to drop as the seat wears, it is apparent that precise matching of the holes in the plug and the seat cannot be maintained.

6.2.3.4 GATE. Gate valves, described in Sub-Topic 5.2.7, are typified by a flat or wedge-shaped valving element (gate) which is retracted along a line perpendicular to the housing port centerline and stored in a bonnet or cavity in one side of the housing.

Gate valving units are closely related to blade, linear slide, and rotary slide valving units because all are essentially based upon the principle of a flat plate operating in a plane normal to the port axis. The distinction between the four types is based upon the type of valving element motion (linear or rotary) and whether the valving element contains a flow passage. These factors are illustrated in Figure 6.2.3.4a.

Configuration. The four elements which define the configuration of a gate valving unit are:

a) Port shape, usually the same as the cross section of the line or duct

b) Gate edge shape (bottom or throttling edge), straight or curved

c) Gate cross-section, flat or wedge shaped

d) Double or single seat.

Circular ports are by far the most common, although rectangular gates are found in some applications. Wedge-shaped gates, such as that pictured in Figure 6.2.3.4b, are usually employed with double seats in the majority of common applications. Valves with flat gates are often referred to as *parallel-slide* valves.

Sealing. Sealing of the gate valving element against its seat or seats may be accomplished by:

a) Using fluid pressure to force the gate against the downstream seat

b) Forcing a wedge-shaped gate between two seats

c) Forcing the seat against the gate after the gate has been moved to the closed position.

Actually, any combination of the three techniques can be employed. When only one seat is used, it is almost always the downstream seat in order to exploit the fluid pressure force. Reference 36-13 describes a 10-foot diameter gate valve for high-vacuum service which uses eight equally-

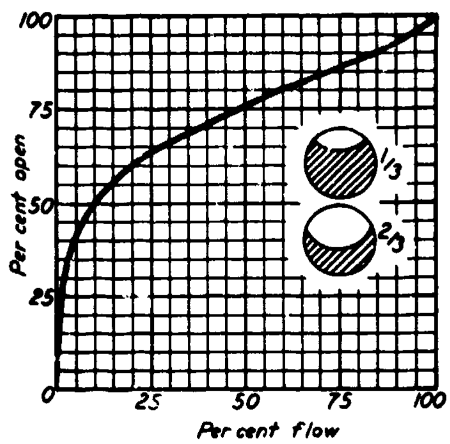


Figure 6.2.3.3b. Flow Characteristics of Rotary Plug Control Valve with Circular Ports

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

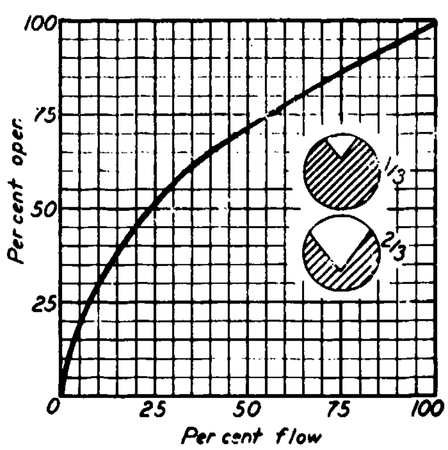


Figure 6.2.3.3c. Flow Characteristics of Rotary Plug Control Valve with V-Shaped Ports

(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

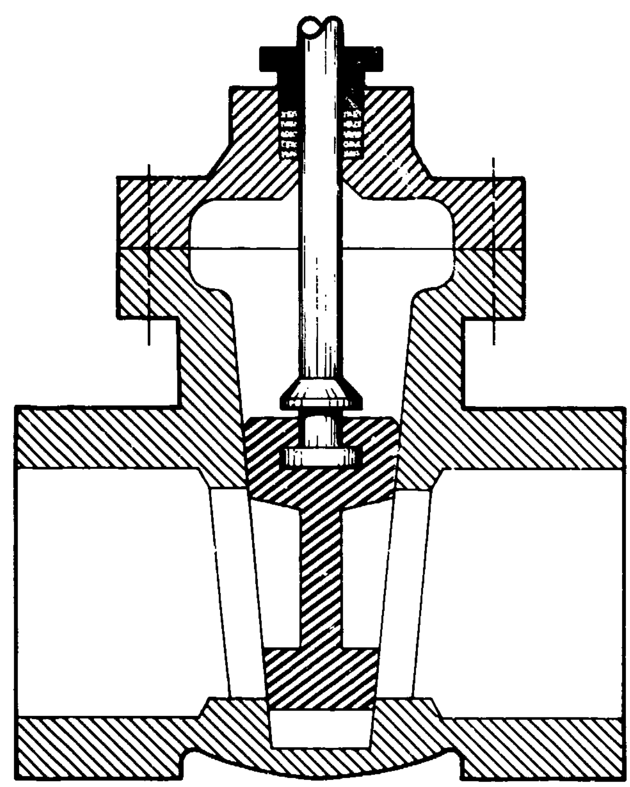
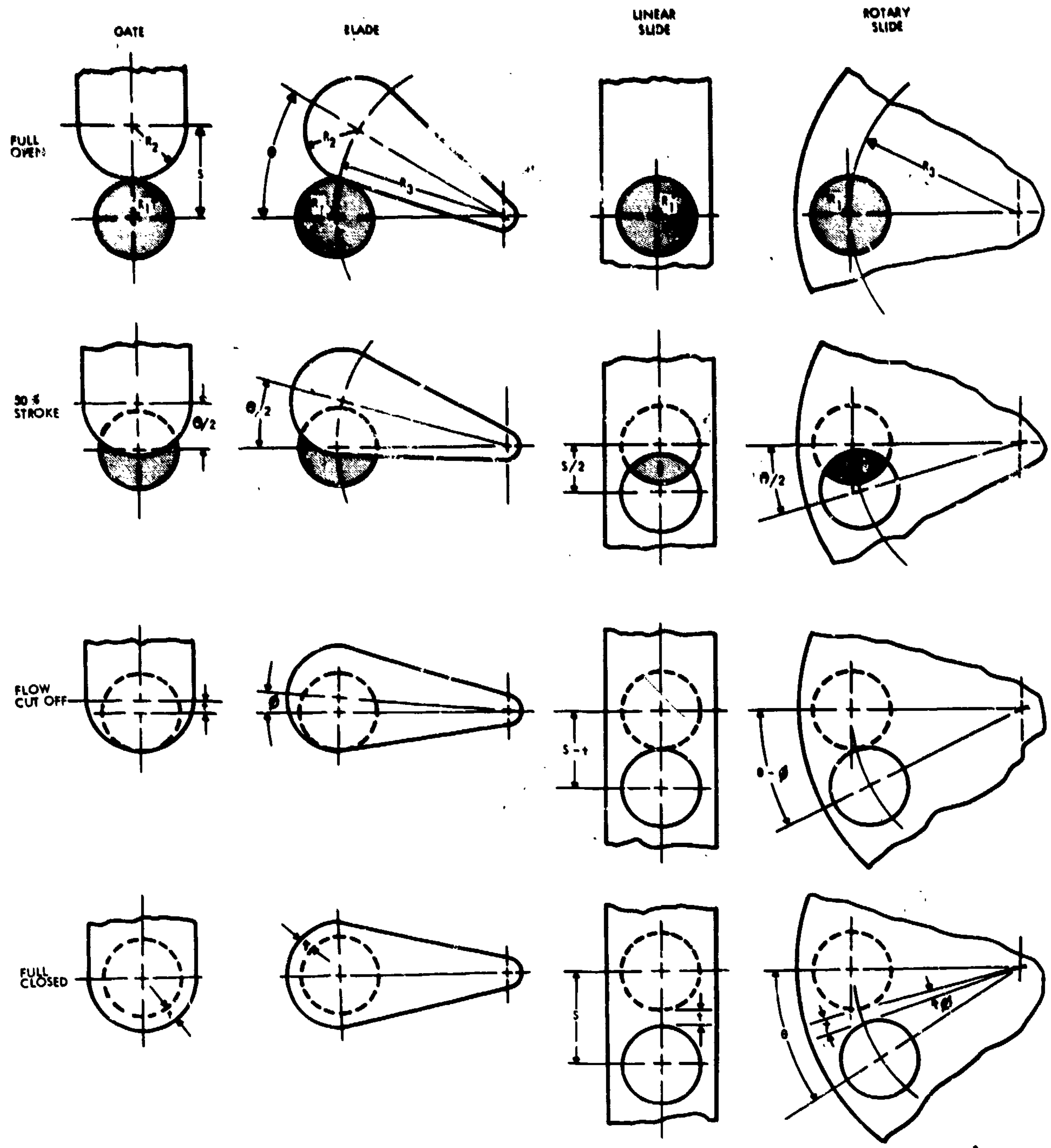


Figure 6.2.3.4b. Gate Valve (Solid Gate)



GATE	BLADE	LINEAR SLIDE	ROTARY SLIDE
$S = 2R_1 + t$	$\theta \approx 2 \text{ ARC SIN } \frac{R_1 + \frac{t}{2}}{R_3}$	$S = 2R_1 + t$	$\theta \approx 2 \text{ ARC SIN } \frac{R_1 + \frac{t}{2}}{R_3}$
	$\phi \approx \text{ ARC SIN } \frac{t}{R_3}$		$\phi \approx \text{ ARC SIN } \frac{t}{R_3}$

NOTE:
DEADBAND STROKE REQUIRED BEYOND FLOW CUTOFF POSITION IS NECESSARY TO PROVIDE MINIMUM SEAL LEAK PATH OF LENGTH t.

$R_1 = \frac{\text{PORT DIA.}}{2}$, IN. $\quad t$ = MINIMUM SEAL LEAK PATH, IN.

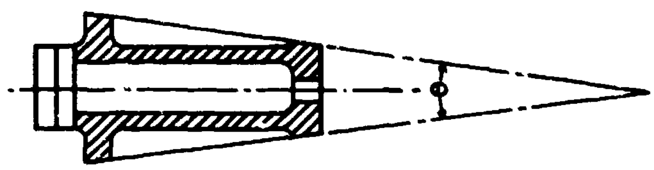
$R_2 = R_1 + t$, IN. $\quad \theta$ = STROKE, DEGREES

S = STROKE, IN. $\quad \phi$ = DEADBAND STROKE, DEGREES

Figure 6.2.3.4a. Distinction Between Gate, Blade, Linear Slide, and Rotary Slide Valving Units spaced pistons to force the gate axially against double concentric O-ring seals in the valve seat after the valve has been closed. With wedge-shaped gates, the wedge angle and the applied force will determine the seating force in a manner similar to the cone angle of a rotary plug valve. Table 6.2.3.4 shows that smaller wedge angles are normally used with larger conventional gate valves to minimize actuation force requirements. It should be noted that both the seat and the gate mating surfaces are fully exposed to the fluid when the gate is in the open position, although only the seat need be exposed to high-velocity flow. Figure 6.2.3.4c illustrates techniques employed to secure seats in the housing of wedge-gate valves.

Table 6.2.3.4. Wedge Angles for Gate Valves

(Adapted with permission from "The Design of Valves and Fittings," G. H. Pearson, Copyright 1964, Sir Isaac Pitman and Sons, Ltd., London, England)



Size of Valve (in.)	Wedge Angle, θ (deg)	Size of Valve (in.)	Wedge Angle, θ (deg)
1/2	10	14	7-1/2
3/4	10	16	6-1/2
1	10	18	6-1/2
1-1/2	10	20	6
2	10	22	6
2-1/2	10	24	5-3/4
3	10	30	5-3/4
4	10	36	5-1/4
5	9	40	5-1/4
6	9	48	5
7	9	52	5
8	8	60	5
9	8	72	4-3/4
10	8	84	4-1/2
12	7-1/2	96	4

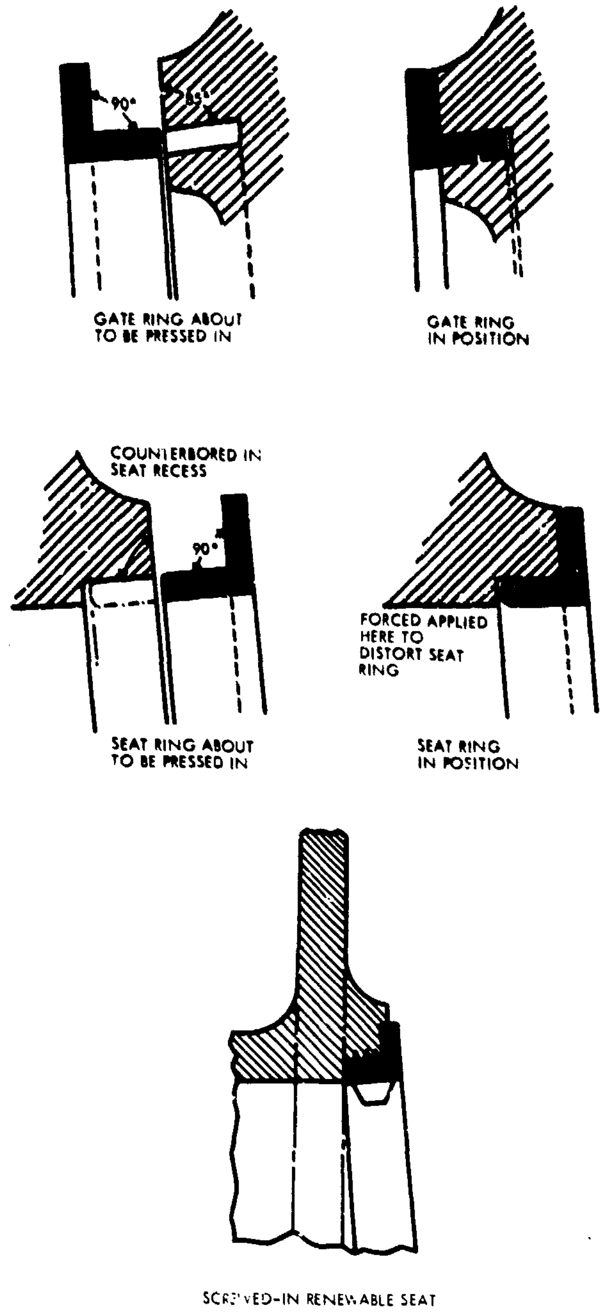


Figure 6.2.3.4c. Methods of Securing Seats in Wedge-Gate Valves

(Adapted with permission from "The Design of Valves and Fittings," G. H. Pearson, Copyright 1964, Sir Isaac Pitman and Sons, Ltd., London, England)

Actuation Forces. For flat or single-seat gate valves the actuation force is primarily a function of pressure forces holding the gate against the seat, and the friction coefficient between the gate and the seat. If pressure differential across the gate is constant under all conditions, naturally the highest actuation force requirement will be in the closed position. Should it be desired to estimate the momentum force acting on a partially-open gate valving element, Reference 68-64 presents the results of a two-dimensional analysis which may be representative of square gate valves.

With wedge-shaped double-seat gate valves, maximum actuation forces are almost always associated with initial opening of the valve. The actual force required to crack open such a gate valve is very difficult to calculate, as it is a function of:

a) The wedge angle

b) The speed or inertia with which the gate was seated

c) Seating stress variations resulting from differential thermal expansion

d) Friction coefficient between seats and gate.

It is common practice in such wedge-shaped gate valves to design a significant backlash into the operator mechanism in order to permit the development of a substantial inertial force in the operator upon opening, to assist in breaking the gate loose from the seats. A more detailed analysis of the forces acting on a wedge-shaped gate may be found in Reference 196-1.

Flow Characteristics. Table 3.9.5.2 presents the following range of resistance coefficients, K (as used in Equation 3.9.5.2a) for circular, wedge-shaped gate valving units as a function of stroke.

Gate Position	K
Fully open	0.19
Half open	2.0
Quarter open	17.0

6.2.3.5 BLADE. Blade valving units, shown in Figure 6.2.3.5, are closely related to the gate valving units shown in Figure 6.2.3.4a, the two primary differences being (a) the valving element rotates about a pivot, and (b) only flat type valving elements are used, whether single or double seats are employed.

With the exception of those features associated with wedge-shaped gates, the preceding discussion of gate valving units may be considered applicable to blade valves. It is apparent from Figure 6.2.3.4a that the flow characteristic of the blade valve may be altered somewhat by modifying the shape of the valving element leading edge. It may also be seen that the conventional blade shape shown in Figure 6.2.3.4a will result in a slightly greater reduction in flow area for a given percentage of stroke than will the curved-edge gate valve with the same seal width.

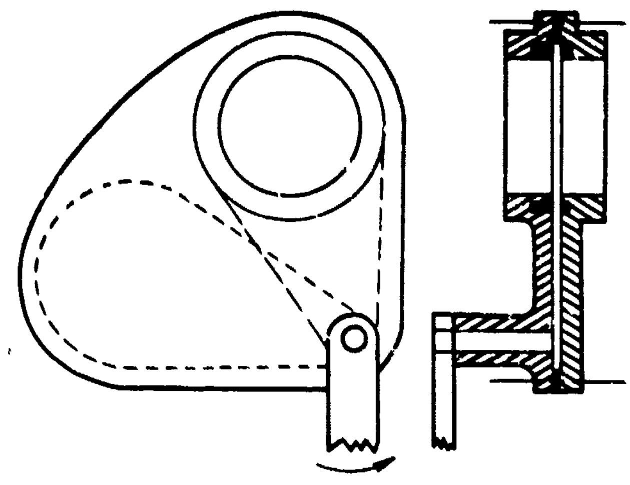


Figure 6.2.3.5. Blade Valve

6.2.3.6 LINEAR SLIDE. Linear slide valving units are similar to gates except that the valving element (slide or plate) contains a hole or port which aligns with the port in the housing in the open position. Linear slide valves are limited in application largely to:

a) Requirements for two or more valving units in close proximity to operate simultaneously, in which case a common valving element plate with multiple ports is used

b) Requirements for simple switching between full-open, closed, and various discrete orifice sizes, wherein the valving element plate may contain one or more precisely machined and calibrated orifices which may readily be positioned across the housing port

Sealing and actuation force techniques are very similar to flat-gate and blade valving units, with the advantage that the seal of the linear-slide valving unit is shielded in the full-open position as well as the full-closed position, as with a ball valve. By matching valving element ports to housing ports, excellent full-open pressure-drop characteristics, comparable to ball valving units, may be obtained. A valving element port shape other than one matching the housing ports may be used to obtain some degree of control over the flow characteristic, but the same throttling limitations associated with gate valves as well as the loss of the excellent full-open pressure-drop properties discourage this practice.

6.2.3.7 ROTARY SLIDE. Rotary slide valving units, as illustrated in Figure 6.2.3.4a, exhibit the basic characteristics of linear slide valves except that rotary motion of the valving element is employed. Rotary slide valves bear a relationship to linear slide valves similar to that relating blade and gate valves with one minor exception: the shape of the valving element port in slide valves is usually identical, resulting in similar flow characteristics, whereas blade and gate valving units may exhibit a somewhat different flow area versus stroke characteristics, as shown in Figure 6.2.3.4a. Rotary slide valves are discussed in Sub-Topic 5.2.12. Figures 6.2.3.7a and b illustrate two rotary slide valve designs, each of which employs two identical valving units. Although a basic feature of the rotary slide valving unit is straight-through flow through the valving unit, the valve designs shown in Figures 6.2.3.7a and b entail several changes in flow direction in order to achieve a series redundancy for shutoff purposes and, in the case of Figure 6.2.3.7a, to provide for pressure balancing. As this example illustrates, rotary slide valves lend themselves to sealing by means of a very close fit between the valving element (disc, rotor, slide, or plate) and the seat or housing. A high degree of flatness and surface finish, and spring-loaded seats are generally used.

6.2.3.8 SLEEVE. Sleeve valves are often classified with spool and piston valves, as discussed in Detailed Topic 5.3.5.6, because the three types share two common characteristics: a cylinder is moved within a cylinder to expose a port, thereby permitting flow, and the fluid is usually required to negotiate a turn. In a sleeve valving unit either the inner or outer cylinder may be the valving element, although the internal valving element as illustrated in Figure 6.2.3.8a is by far the more common. Piston and spool valving units usually contain solid valving elements, spools being essentially identical to pistons except that spools have two or more lands. In small-size applications, such as regulators and hydraulic controls, are found single-land valving elements which are hollow for pressure balancing (Figure 5.5.6f) and therefore fit the sleeve description, but which by convention are referred to as piston valving units.

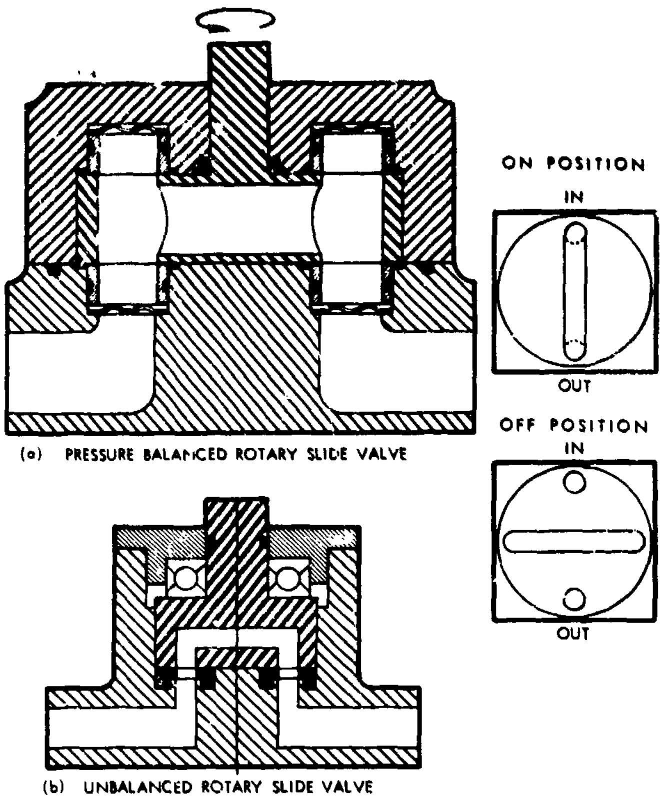


Figure 6.2.3.7a,b. Two-Way Rotary Slide Valves

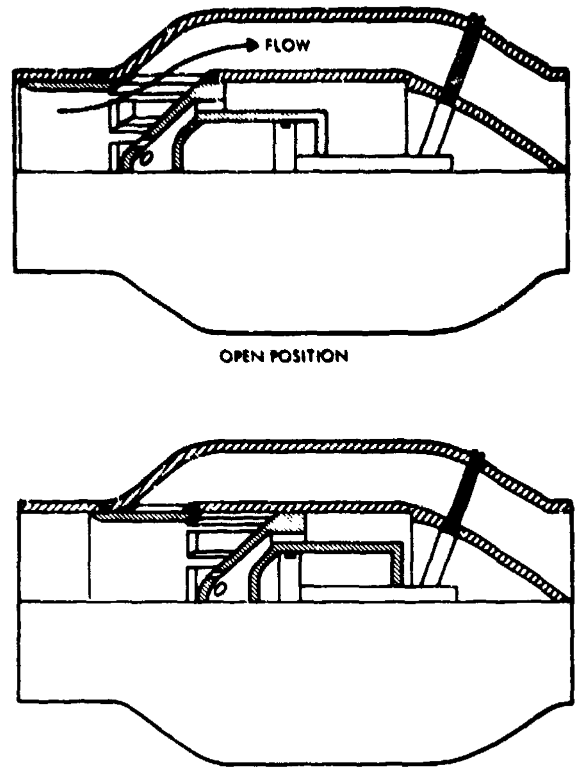


Figure 6.2.3.8a. Sleeve Valve with Seals in Housing
(Adapted from Reference 34-10)

Configuration. Both linear (axial) and rotary sleeve valves have been designed, but rotary sleeve valves are seldom used because of sealing difficulties and the fact that less than one-half of the sleeve circumference may be used for porting. For this reason, only the linear sleeve valving unit will be treated in this discussion. The parameters which define the configuration of a sleeve valving unit are:

a) Sleeve diameter

b) Stroke (port length)

c) Port shape (rectangular, circular, annular, V-port, etc.)

d) Seal type and location (in valving element, in housing, or packless)

e) Streamlining or special tailoring for flow (such as the flow guide and chamfered ports in Figure 6.2.3.8a)

f) Sleeve length/diameter ratio (at least 1.5 to avoid cocking, unless otherwise guided)

g) Flow direction through port (radially inward or radially outward).

Unlike the valving units which interpose the valving element across the natural flow path of the line (butterfly, ball, plug, gate, etc.) the sleeve valving unit requires that the fluid change direction. This feature necessitates a flow passage external to the nominal line diameter, but also permits the use of totally-enclosed actuators which are not feasible for the preceding valving unit types.

Sealing. The inability of dynamic seals to perform as low leakage static seals has been the most significant factor limiting the application of the sleeve valve. In small sizes (1 inch or below) the close tolerance (packless sealing) techniques associated with spool valves are usually employed. In larger sizes, various dynamic seal approaches discussed in Sub-Section 6.4 have been employed. The polymeric lip seal illustrated in Figure 6.4.4.1d has been used by Aerojet-General Corporation in 10-inch diameter sleeve valves for cryogenic propellant service (Reference 34-10). Other approaches have included the use of multiple-element precision piston rings for sleeve valving units. A major consideration in sleeve valving unit design is the choice of seal location (housing or valving element) and the detail design of the seal and its mating surface. A tradeoff must be made between the disadvantage of having a seal in the valving element pass across the port (Figure 6.2.3.8b) and that of permitting a seal in the housing to be fully exposed to high-velocity flow after the sleeve has been retracted (Figure 6.2.3.8a).

Actuation Forces. The sleeve valving unit is inherently pressure-balanced and therefore requires a minimal actuation force. The forces which must be considered include:

a) Pressure unbalance resulting from actuator shaft area

b) Seal friction (a function of pressure with pressure-actuated dynamic seals)

c) Fluid momentum force against flow guide, if guide is attached to sleeve. (This does not apply if flow guide is fixed.)

Pressure unbalance forces due to the actuator shaft area and seal friction forces may be analyzed by conventional techniques, but the fluid dynamic forces acting on a valving element flow guide may be appreciable in magnitude

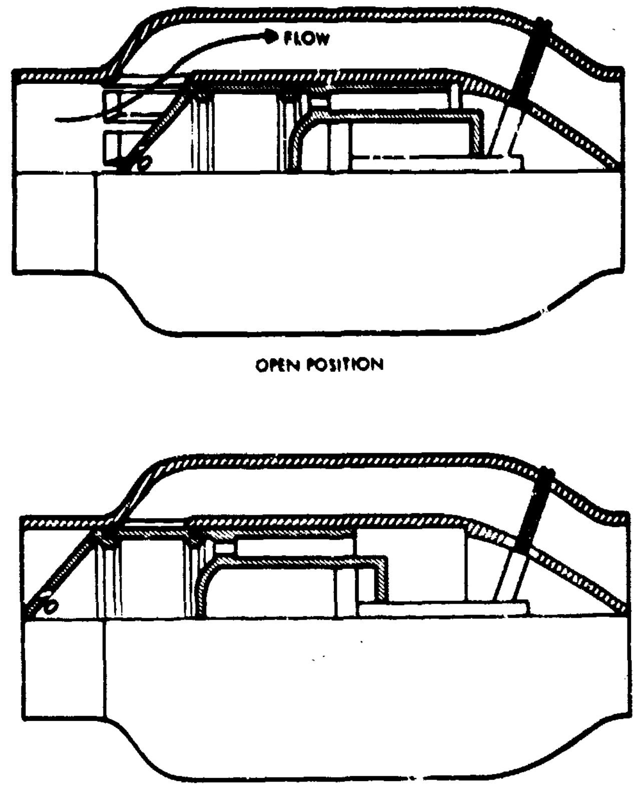


Figure 6.2.3.8b. Sleeve Valve with Seals in Sleeve
(Adapted from Reference 34-10)

and difficult to analyze. For steady-state operation in any given sleeve position, the force will be based upon the difference between the net static pressure acting on the flow-passage side of the flow guide and the static pressure acting on the internal, or shielded side of the flow guide.

This internal static pressure will, of course, be a function of the location of the pressure-balancing holes (as illustrated in Figure 6.2.3.8c) as well as the configuration of the flow guide and the position of the sleeve. The latter two factors will establish the pressure distribution on the flow-passage side of the flow guide.

It should also be noted that sleeve valving units, like spool and poppet valving units, are linear motion devices and therefore are influenced by acceleration forces acting along their axis.

Flow Characteristics. One of the major advantages of the sleeve valving unit is the wide range of flow characteristics which may be obtained by tailoring the port shape and/or the flow guide configuration. If rectangular or

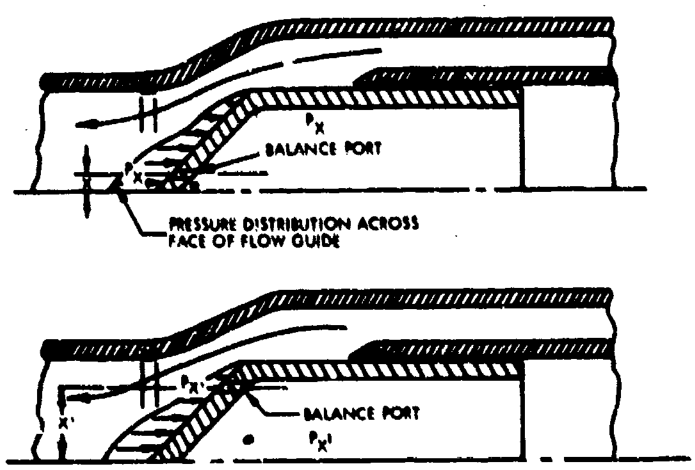


Figure 6.2.3.8c. Effect of Balance-Hole Location on Sleeve Valve Flow Guide

annular ports are used, flow guide configurations may be varied in a manner closely akin to that used with poppet valving units (see Detailed Topic 6.2.3.11). The simplicity of obtaining a particular characteristic by using a shaped port is offset by the longer port and stroke required for a given maximum flow rate. It may be difficult to design for minimum full-open pressure drop if shaped ports are employed. Reference 34-10 indicates that in-line sleeve valves may be designed with a resistance coefficient, K, of only about 0.5, using annular or rectangular streamlined porting and a flow guide.

As with poppet valving units, there is no precise value of maximum stroke (or port length) for a sleeve valving unit; the valve selected for a particular design is usually a compromise between desired flow characteristics plus size and weight considerations. Where minimum flow resistance is important, however, it is advantageous to maintain constant fluid velocity through the valve by careful design of the flow guide, port, and housing, thus avoiding expansion and contraction losses (see Figure 6.2.3.8d).

6.2.3.9 SPOOL. Spool valving units, because of their wide application in servo control systems, have been the subject of much analytical and experimental effort in recent years. The data presented below will provide an understanding of basic design techniques, but the designer faced with a critical design problem should consult a text devoted to the subject, such as Reference 45-1 and 42-1.

Configuration. The spool valving unit consists of a solid cylindrical valving element or spool, having two or more lands (a spool with only one land is referred to as a piston) which fit closely within the bore of the housing. The valv-

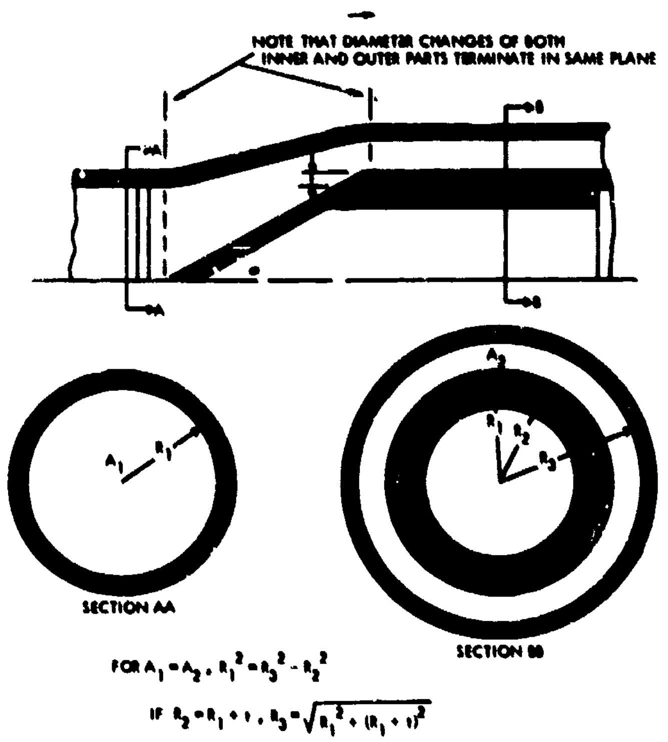


Figure 6.2.3.8d. Sleeve Valve Proportions for Minimum Pressure Drop

ing unit is characteristically packless and controls flow through ports in the housing by translation of the valving element. The parameters which define the configuration of a spool valving unit include:

a) Number of spool lands

b) Number of housing ports

c) Shape of housing ports, usually rectangular with straight walls and sharp edges although circular ports are not uncommon. (Annular ports are most desirable but relatively difficult to fabricate.)

d) Shape of spool grooves or inter-land reliefs (usually straight-walled with sharp edges, except where reaction force compensation is incorporated)

e) Width and spacing of lands and ports

f) Spool land/port overlap or underlap; related to item (e) in Figure 6.2.3.9a

g) Clearance between spool lands and bore

h) Stroke.

Figure 6.2.3.9b presents recommended basic dimensional proportions for a typical 4-way spool valving unit.

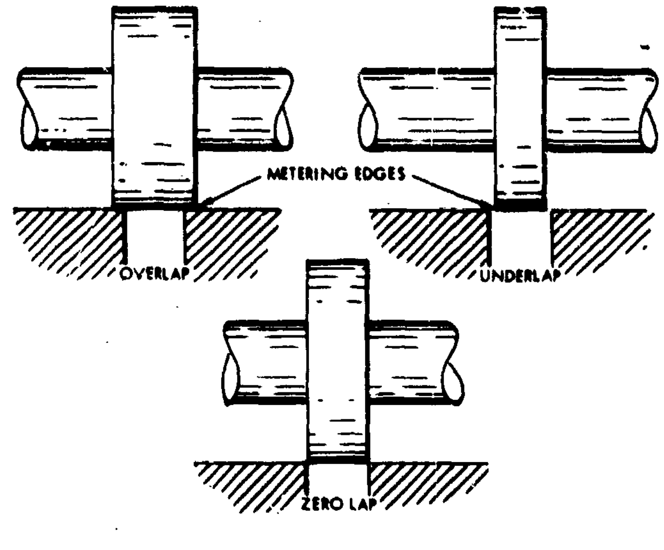


Figure 6.2.3.9a. Spool Lands Showing Overlap, Underlap, and Zero Lap

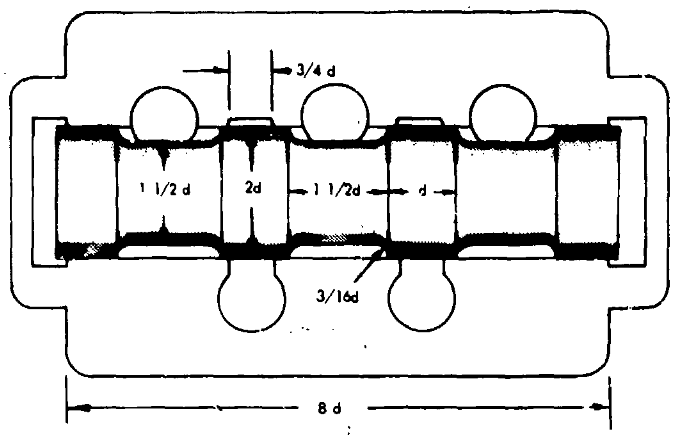


Figure 6.2.3.9b. Dimensional Proportions for a 4-Way Spool Valving Unit

(Adapted from "Hydraulics and Pneumatics," July 1962, vol. 15, no. 7, L. Dodge, Copyright 1962, The Industrial Publishing Corporation, Cleveland, Ohio)

Sealing. Spool v applications do not usually require tight sealing and it is expected that some leakage will occur. With 4-way hydraulic spool valves in the nul or central position, it is reasonable to anticipate leakage flows approaching one in^3/sec with slightly overlapping lands. With zero overlap or underlapping spools, much higher leakage rates must be expected. Leakage is critical to the performance of servovalves, but is tolerable if it is consistent and controlled. Towards this end a minimum practical radial clearance of about 0.00005 inch may be considered for light hydraulic service (Reference 45-1). The difficulties associated with tolerances commensurate with this order of magnitude, especially on cylinder bore, may present such severe manufacturing problems that greater radial clearances must be used.

The spool valving unit does not lend itself to, nor do its usual applications dictate, the use of dynamic seals, since most such seals do not lend themselves to the precise flow passage dimensional control possible with hard, sharp-edged, packless spools and ports.

Actuation Forces. The actuation force required for a spool valving unit is a function of fluid forces, bearing friction forces, mass of the valving element, and acceleration forces. Acceleration forces due to vibration or acceleration of the entire system are directly associated with spool mass, and may in some applications present design problems or constraints upon design parameters such as sensitivity. Bearing friction forces are dependent upon the friction coefficient between spool and bore and the radial forces holding the spool against the bore. Reference 73-60 suggests that a conservative estimate of the maximum friction force on a spool valving element may be obtained from the expression

$$F_f = 0.0075\pi d^2 p \qquad \text{(Eq 6.2.3.9a)}$$

where

F_f = static friction force, lb_f
d = housing bore diameter, in.
p = pressure, psig

Figure 6.2.3.9c illustrates two techniques whereby spool lands may be grooved to minimize the stick-slip tendency which can hamper smooth operation.

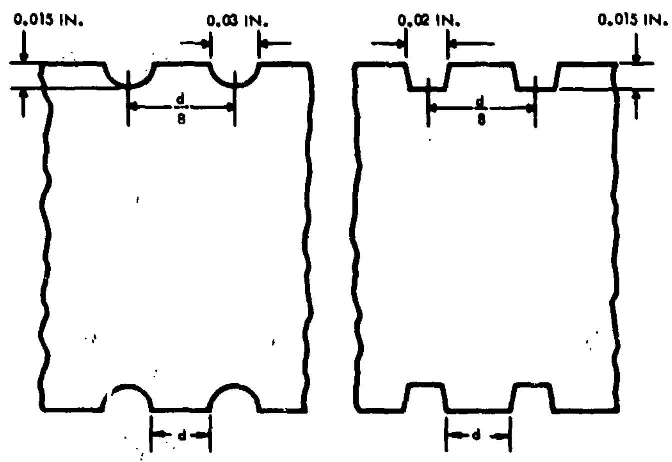


Figure 6.2.3.9c. Spool Land Grooves for Preventing Stick-Slip

(Adapted from "Hydraulics and Pneumatics," July 1962, vol. 15, no. 7, L. Dodge, Copyright 1962, The Industrial Publishing Corporation, Cleveland, Ohio)

In most spool valve applications fluid forces are far greater than acceleration or bearing friction forces and have therefore been the subject of extensive study. Forces due to static pressure differentials are simple to calculate by conventional means, but fluid dynamic forces are a complex function of port and land geometry, pressure drop across the port, and radial clearance between spool and bore. The angle θ (Figure 6.2.3.9d) between the axis of the stream and the spool axis has been found analytically and experimentally to be 69 degrees for small values of valve opening x and zero radical clearance. Blackburn (Referance 45-1) gives the following equation for calculating the axial force on a valving unit with perfectly sharp corners and rectangular ports

$$F_A = 2\, C_q\, w\, \Delta p \cos\theta \sqrt{x^2 + C_r^2} \qquad \text{(Eq 6.2.3.9b)}$$

where

F_A = axial fluid force, lb_f

w = peripheral width of port, in.

$\Delta p = P_1 - P_2$ (Figure 6.2.3.9d), lb_f/in^2

x = axial length of port, in.

C_r = radial clearance between spool and bore, in.

$\cos\theta$ = obtained from Figure 6.2.3.9e

C_q = discharge coefficient $= \dfrac{Q}{wx\sqrt{(2\Delta p)/\rho}}$

where

Q = total rate of flow, in^3/sec

ρ = density of fluid, lb_m/in^3

Figure 6.2.3.9f illustrates the effect of rounding port and spool land edges to a radius, r. With rounded edges, for given x and C_r, θ will be less than 69 degrees and, as would be expected from the greater horizontal component, the axial force on the spool is greater than with the ideal sharp-edged valve. For large values of x, θ approaches 69 degrees; for negative values of x, θ approaches zero as x approaches $-r$.

A wide variety of port and land configurations may be utilized to compensate for these axial fluid dynamic forces. Specific designs are presented in References 6-36, 6-159, 26-29, 42-1, and 45-1.

Flow Characteristics. Flow characteristics of spool valving units are a function of spool and port configurations and may be tailored to a large variety of requirements. Sub-Section 5.6 discusses some of these requirements. Most spool-valve applications require pressure versus flow characteristics, rather than the stroke versus flow characteristics used to describe the other valving units treated in this detailed topic. Reference 19-229 indicates that representative 4-way spool valves may be expected to have a total flow resistance, K, of 3 to 5.5 in the fully open position.

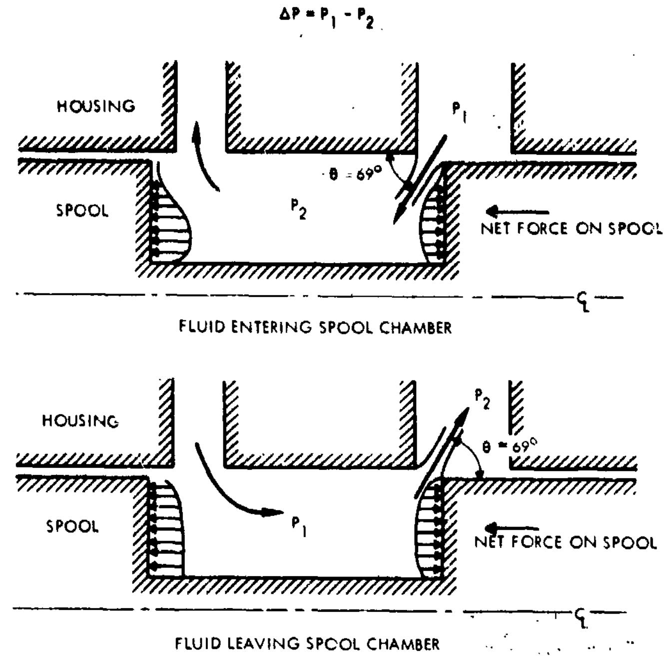


Figure 6.2.3.9d. Flow Force with Sharp-Edged Annular Ports and Square Spool Lands

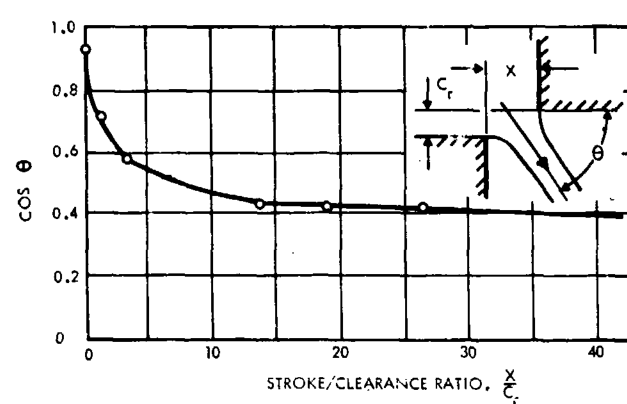


Figure 6.2.3.9e. Effect of Radial Clearance on Cos θ

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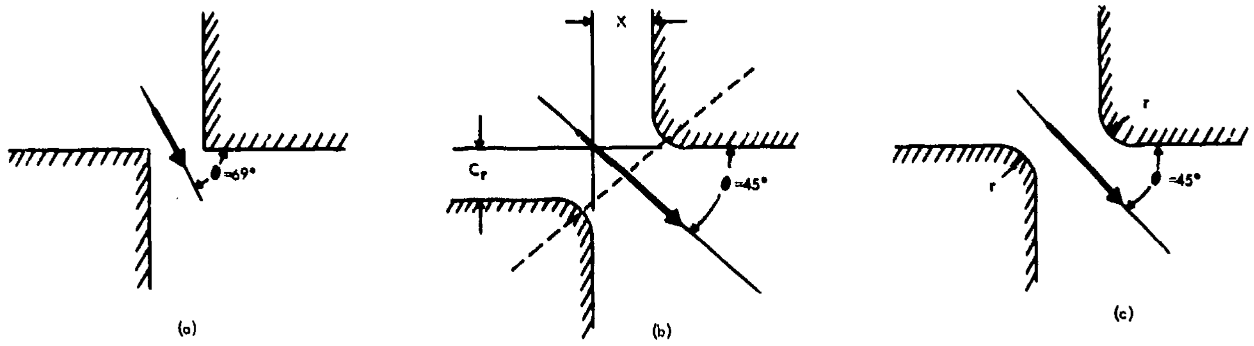
6.2.3.10 PISTON. As discussed under "Sleeve Valving Units," Detailed Topic 6.2.3.8, the term *piston* is frequently

Figure 6.2.3.9f. Effect of Rounded Land Edges on θ
(Reprinted with permission from "Fluid Power Control," J. F. Blackburn, G. Reethof, and J. L. Shearer, MIT Press, Cambridge, Massachusetts, Copyright 1950)



applied to sleeve and spool valving units. There is little in the way of standard terminology in this regard; most manufacturers use the particular convention with which they are most familiar. For the purpose of this treatment of valving units, however, the solid piston valving unit is considered to be equivalent to a spool valving unit with a single land, and the hollow or pressure-balanced piston is considered the equivalent of the sleeve valving unit. In those instances where a piston seals by seating the crown of the piston against a seat, it is considered to be a poppet valving unit.

6.2.3.11 POPPET. The poppet is probably the most widely used of any of the valving units, to which the distribution of functions performed by the various valving units as given in Table 6.2.1 attests. Poppet valves for shutoff and control service are discussed in Sub-Topics 5.2.5 and 5.3.5, respectively.

Configuration. The poppet valving unit is characterized by the translation of the poppet (valving element) away from the seat in a direction that is initially perpendicular to the plane of the seat. This motion eliminates the necessity for sliding contact between seat and poppet although some designs obtain varying degrees of scrubbing action for sealing purposes. Table 6.2.3.11 lists the primary configuration-defining parameters of the most common seat and poppet combinations. A large variety of seat and poppet configurations have been evolved for specific purposes, and the essential elements of representative design approaches will be discussed as they apply to sealing, actuation force, or flow characteristic considerations.

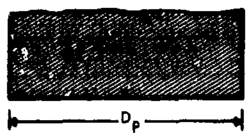
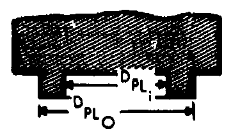
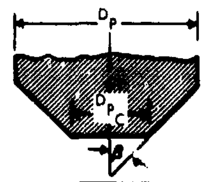
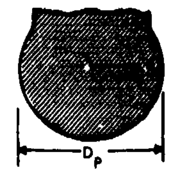
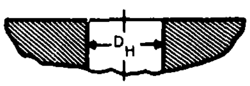
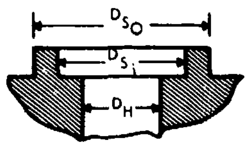
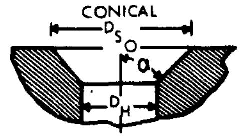
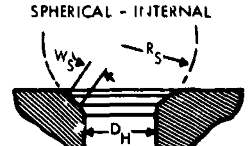
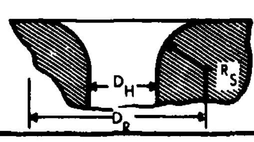
Sealing. The sealing reliability of poppets and seats has resulted in the use of this valving unit type in some of the most demanding applications, where no other valving unit has proven satisfactory. The internal combustion engine's conical poppets, which are subjected to severe functional and thermal cycling, and the large spherical-poppet valves used in steam turbine throttling service are examples of applications wherein the poppet valve is virtually unchallenged. These applications have required hard seats, whereas the majority of aerospace applications to date have used soft seats, primarily in order to obtain sufficiently low leakage rates. Several examples of both hard and soft seat designs are presented in the following paragraphs, followed by a brief summary of recent analyses of hard seat and poppet sealing theory.

Hard Seats. The recent use of highly reactive rocket propellants, and requirements for controlling the flow of hot gases have resulted in increased interest in metal-to-metal seat poppet valves for aerospace applications. Typical of the hard materials used for seats and poppets are 440C stainless steel, stellites, and carbides. Hard metal seats and poppets have been used in aircraft hydraulic systems for many years, but usually with relatively high permissible leak rates of comparatively viscous hydraulic fluid.

The small, dual concentric seat and flat disc, or poppet, of the valving unit illustrated in Figure 6.2.3.11a is an example of such a concept which has recently been applied to more rigorous gas sealing applications, and which has achieved leakage rates in the order of 1×10^{-4} cc/sec of gaseous helium at 500 psi. In this particular design, the outer knife-edge ring serves as a bumper to protect the inner knife-edge ring (seat) from damage as well as to align the floating poppet against the seat. This valving unit is normally produced in very small sizes (about ¼-inch port diameter), using seat widths optically lapped to thicknesses in the order of 0.010 inch. A similar approach is employed in the unit illustrated in Figure 6.2.3.11b, which uses an interrupted outer ring and a seat and poppet whose sealing surfaces are lapped to a flatness of less than one light band.

Hard conical seats and poppets are also found in small sizes, as illustrated in Figures 6.2.3.11c and 6.2.3.11d. It

Table 6.2.3.11. Primary Configuration-Determining Parameters for Common Seat and Poppet Combinations

POPPET / SEAT	FLAT 	FLAT WITH RAISED SEAL LAND 	CONICAL 	SPHERICAL 
FLAT 	$D_P > D_H$	$D_{P_{L_I}} > D_H$	$D_{P_C} < D_H < D_P$	$D_P > D_H$
FLAT WITH RAISED SEAL LAND 	$D_P \geq D_{S_O}$	SELDOM USED $D_{PL_O} \approx D_{S_O}$ $D_{PL_I} \approx D_{S_I}$	NOT USED	NOT USED
CONICAL 	SELDOM USED $D_P > D_{S_O}$	SELDOM USED $D_{P_{L_I}} > D_{S_O}$	$D_P > D_H$ USUALLY $\alpha \geq \beta$ SELDOM $\alpha < \beta$	USUALLY $D_H \cos^{-1}\alpha < D_P < D_{S_O}$ SELDOM $D_P > D_{S_O}$ SELDOM $D_H < D_P < D_H \cos^{-1}\alpha$
SPHERICAL - INTERNAL 	NOT USED	NOT USED	NOT USED	$D_P = 2R_S$
SPHERICAL - EXTERNAL 	SELDOM USED $D_P > D_R$	SELDOM USED $D_{P_{L_I}} > D_R$	$D_P > D_H$	$D_H < D_P < D_R$

is interesting to note that in one case both seat and poppet are machined to the same angle, whereas in the other example different angles are used, and the seat and poppet are lapped together to form a seating area somewhat wider than the initial line contact.

Three spherical poppet design approaches using hard seats are illustrated in Figures 6.2.3.11e, f, and g. It is particularly interesting to note the higher port diameter/ball diameter ratio of the sapphire valving unit shown in Figure 6.2.3.11g, with its relatively high sealing stress.

To date, hard-seat poppet valving units for aerospace applications have been limited to relatively small sizes, largely because of difficulty in meeting stringent leakage requirements. Reference 498-1 describes the difficulties encountered and solutions employed in the development of small, spherically-lapped poppet valving units, and other design data are available pertaining to hard-seat poppet valving units for aerospace applications from the recent studies described in References 35-12 and 35-14, which are discussed in the following paragraphs under sealing theory. Reference 35-14 includes an analysis of the bounce or ringing which occurs when a hard poppet impacts on a hard seat.

Soft Seats. A wide variety of soft-seat design concepts have been evolved, and a few representative examples are described below. It is primarily with soft-seat designs that poppet valving units have been applied to larger sizes for

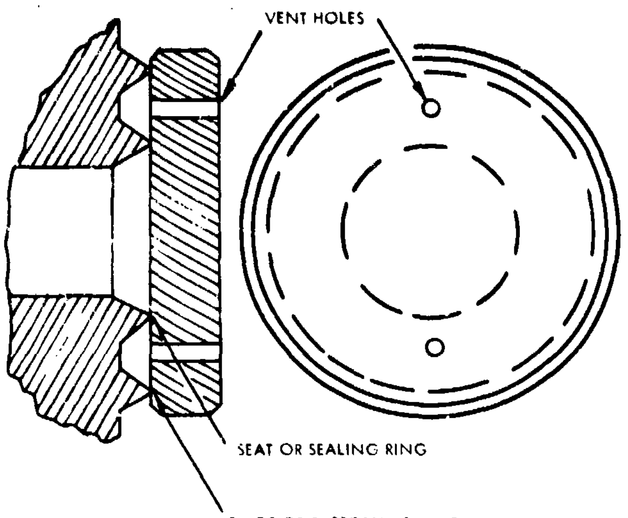
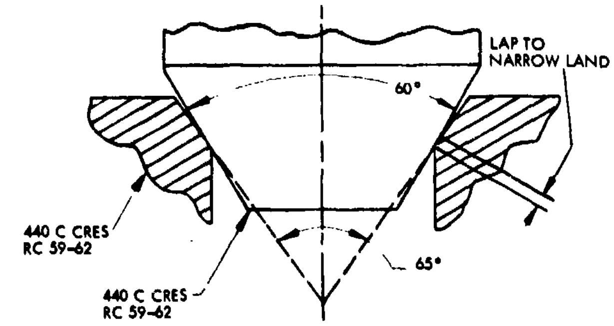


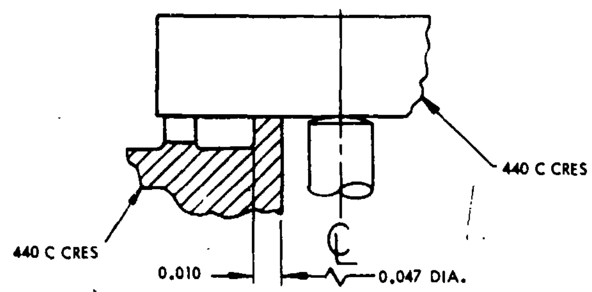
Figure 6.2.3.11a. Flat Disc Hard Seat and Poppet Concept
(Courtesy of Bertea Products, Pasadena 6, California)



DESIGN PARAMETERS

FLOW MEDIA: HYDRAULIC OIL (MIL-0-5606)
OPERATING PRESSURE: 300 TO 500 PSI
OPERATING TEMPERATURE: -65 TO +160 F
TUBE SIZE: 3/4 IN.

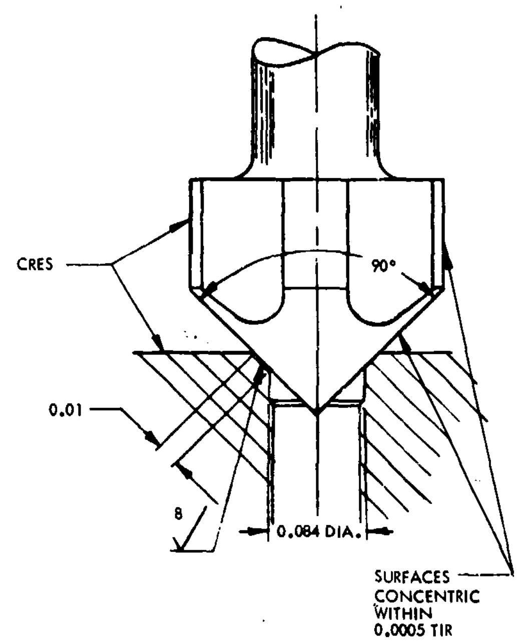
Figure 6.2.3.11c. Hard Conical Poppet Seat Design for Hydraulic Relief Valve
(Reference 35-12; Benbow Manufacturing Corporation design, p. B-10)



DESIGN PARAMETERS

FLOW MEDIA: AIR, NITROGEN, HELIUM
OPERATING PRESSURE: 750 PSIG
OPERATING TEMPERATURE: -320 TO +160 F
CAPACITY: 0.030 EQUIVALENT ORIFICE

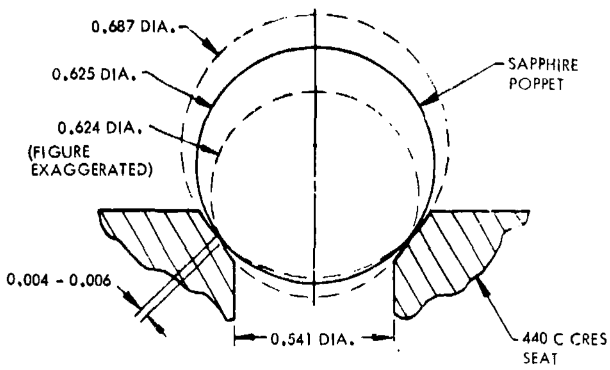
Figure 6.2.3.11b. Hard Flat Poppet Seat Design
(Reference 35-12; Weston Hydraulics, Limited design, p. B-58)



DESIGN PARAMETERS

FLOW MEDIA: NITROGEN TETROXIDE, UDMH, HYDRAZINE
OPERATING PRESSURE: 300 PSIA
OPERATING TEMPERATURE: -65 TO +165 F
CAPACITY: 0.07 IN. ORIFICE

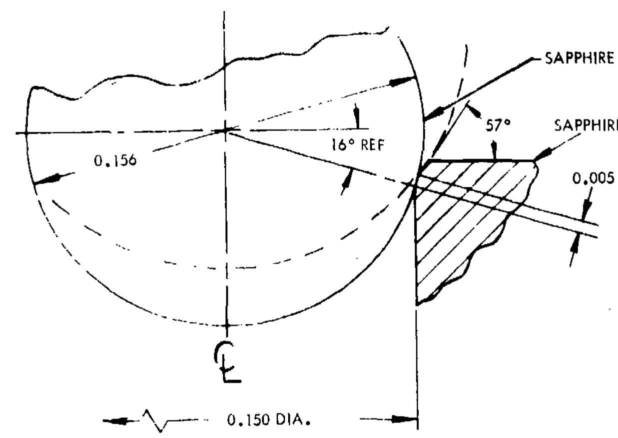
Figure 6.2.3.11d. Hard Conical Poppet Seat Design for Propellant Solenoid Valve
(Reference 35-12; Minneapolis-Honeywell Regulator Company, Aeronautical Division design, p. B-16)



DESIGN PARAMETERS

FLOW MEDIA: AIR, NITROGEN, HELIUM
CRACKING PRESSURE: 50 TO 100 PSIG
OPERATING TEMPERATURE: -450 TO +300 F
CAPACITY: 0.12 IN. EQUIVALENT ORIFICE

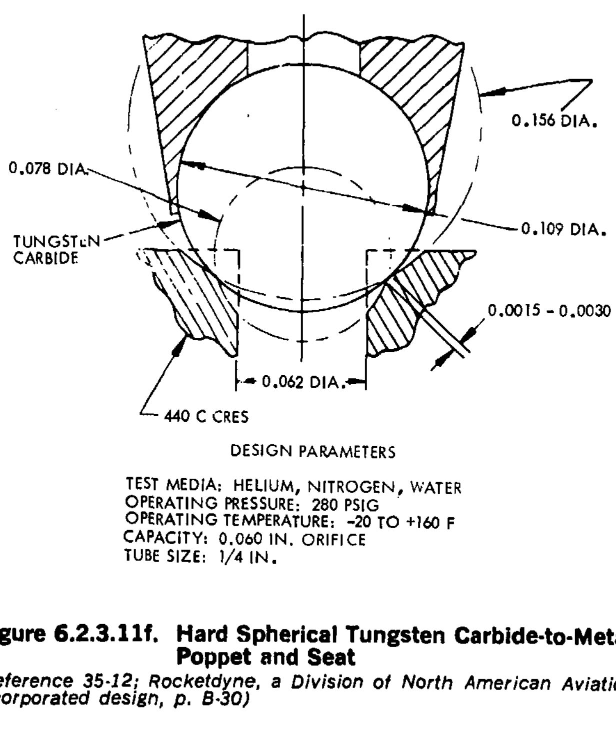
Figure 6.2.3.11e. Hard Spherical Sapphire-to-Metal Poppet and Seat
(Reference 35-12; Frebank Company design, p. B-26)



DESIGN PARAMETERS

FLOW MEDIA: HELIUM & FLUORINE GAS COMBINATION
OPERATING PRESSURE: 280 TO 290 PSIG
(CRACK, FULL FLOW & RESEAT)
OPERATING TEMPERATURE: -30 TO +100 F
CAPACITY: 0.0084 IN. EQUIVALENT ORIFICE
TUBE SIZE: 1/4 IN.

Figure 6.2.3.11g. Hard Spherical Sapphire-on-Sapphire Poppet and Seat
(Reference 35-12; Whittaker Controls Division of Telecomputing Corporation design, p. B-46)



DESIGN PARAMETERS

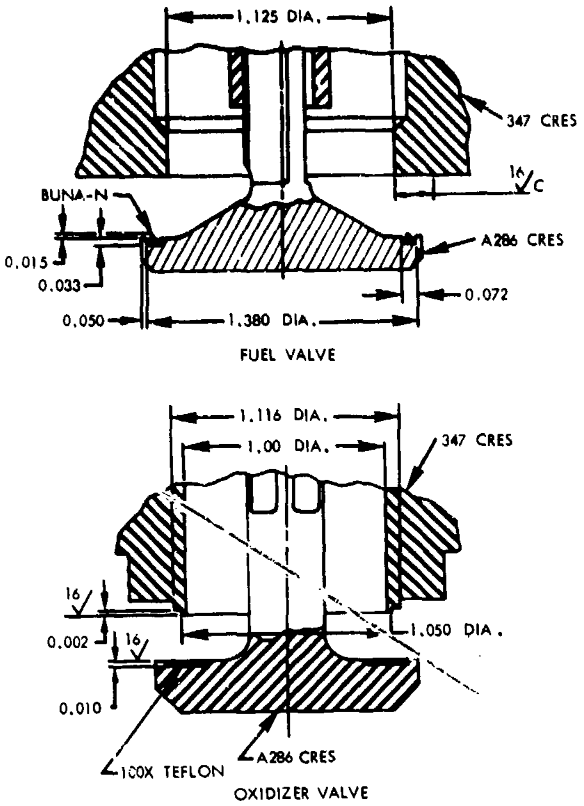
TEST MEDIA: HELIUM, NITROGEN, WATER
OPERATING PRESSURE: 280 PSIG
OPERATING TEMPERATURE: -20 TO +160 F
CAPACITY: 0.060 IN. ORIFICE
TUBE SIZE: 1/4 IN.

Figure 6.2.3.11f. Hard Spherical Tungsten Carbide-to-Metal Poppet and Seat
(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-30)

aerospace applications. Soft-seat poppet valving units commonly employ either a relatively hard poppet or seat with the mating part of comparatively soft material.

Among the materials used for soft seats or poppets are plastics such as Teflon, Kel-F, nylon, and Mylar; elastomers such as Buna N, Viton, and silicone, and relatively soft metals such as copper and aluminum. Figures 6.2.3.11h and 6.2.3.11i illustrate typical soft-seat designs for flat poppets and seats. It is interesting to note that the soft sealing element may be placed in either the seat or poppet surface. These examples illustrate three techniques for obtaining relatively high initial seating stress. Figure 6.2.3.11h demonstrates the use of narrow raised lands on either the hard or soft surface, whereas Figure 6.2.3.11i shows the use of a slight conical taper on the soft valving-element seal. Another approach, not illustrated, is to use a flat, sharp-edged valving element whose outside diameter is only slightly greater than the internal diameter of the soft flat seat, seating stress being limited by providing a positive stop which prevents excessive interference or deformation of the soft seat by the valving element.

Conical poppet sealing techniques may be seen in the examples shown in Figures 6.2.3.11j, k, and l. Perhaps the simplest approach is that of the hard conical poppet seating in a sharp-edged hole in the soft seat material, as shown in Figure 6.2.3.11j. A positive limit is set on seating stress by the metal seat or stop, which limits compression of the Teflon seat in the design approach of Figure 6.2.3.11k. Development of the larger pressure-actuated lip-seal design shown in Figure 6.2.3.11l took place during a state-of-the-art improvement program for extreme environment components.

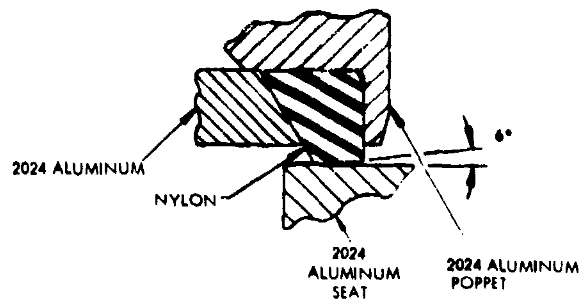


DESIGN PARAMETERS

FLOW MEDIA: LIQUID NITROGEN, OXYGEN, RP-1
OPERATING PRESSURE: 800 PSIG
OPERATING TEMPERATURE: OXIDIZER VALVE: -320 TO +160F
FUEL VALVE: -40 TO +160F
CAPACITY: OXIDIZER PORT: 0.61 -IN. ORIFICE
FUEL PORT: 0.94-IN. ORIFICE

Figure 6.2.3.11h. Flat Poppet with Soft Seats for Gas Generator Control Valves

(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-60)

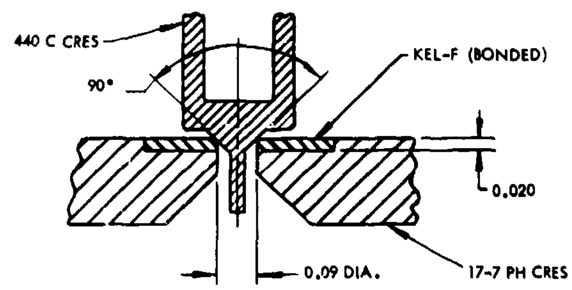


DESIGN PARAMETERS

FLOW MEDIA: AIR, NITROGEN, HELIUM
CRACKING PRESSURE: 300 TO 1000 PSIG
OPERATING TEMPERATURE: -30 TO +160 F
CAPACITY: 0.7 IN. EQUIVALENT ORIFICE
TUBE SIZE: 1 IN.

Figure 6.2.3.11i. Flat Poppet with Soft Seat for Pressure Relief Valve

(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-50)



DESIGN PARAMETERS

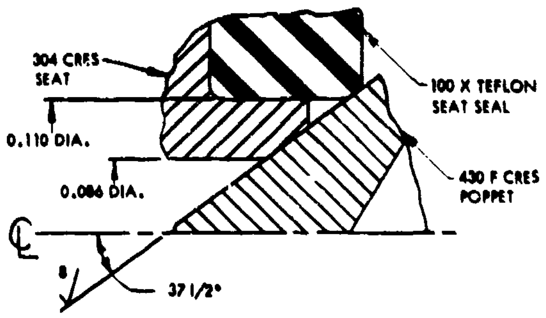
FLOW MEDIA: NITROGEN
INLET PRESSURE: 390 TO 500 PSIG
OPERATING TEMPERATURE: -30 TO +120 F
CAPACITY: 0.05 IN. ORIFICE (MAIN VALVE)
TUBE SIZE: 1/4 IN. INLET, 3/8 IN.

Figure 6.2.3.11j. Conical Poppet and Soft Seat for Pressure Regulator Valve

(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-8)

Spherical poppets are used with soft seats of various configurations, the more common being conical seats and sharp-edged holes in flat seats. Figure 6.2.3.11m shows a high-pressure design for a relief valve application wherein only a small edge of the Teflon seat is exposed to the valving element. The surrounding seat metal serves both to retain the Teflon and to limit its deformation by the poppet. Pitfalls which must be guarded against when designing for soft seals include seal blowout and cold flow. Elastomeric seals exposed to the high velocity gas flow which occurs in the near-seated position are particularly susceptible to blowout. Minimizing exposure of the seal to the fluid stream aids in avoiding this problem (see Subsection 6.4). Cold flow is minimized by confining the seal particularly in the loaded condition.

Poppet and seat sealing theory. The following discussion of the sealing and leakage aspects of the poppet valving unit is based almost entirely on Reference 35-14, "Poppet and Seat Design Data for Aerospace Valves," Technical Report AFRPL-TR-66-147, by G. F. Tellier of Rocketdyne. This document, issued July 1966, describes the results of a detailed analytical and experimental study performed to provide fundamental sealing data on metal-to-metal seat and poppet valving units. It is the most comprehensive recent work on the subject of sealing between seats and poppets,

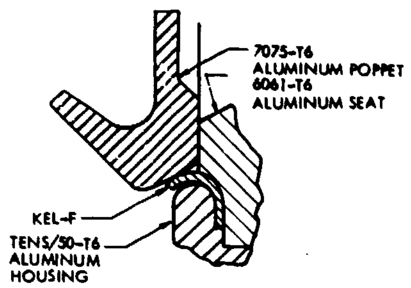


DESIGN PARAMETERS

FLOW MEDIA: NITROGEN TETROXIDE, UDMH, HYDRAZINE
OPERATING PRESSURE: 325 PSIG
OPERATING TEMPERATURE: -65 TO +160 F
CAPACITY: 0.06 IN. ORIFICE
TUBE SIZE: 1/4 IN.

Figure 6.2.3.11k. Conical Poppet and Soft Seat for Solenoid Valve

(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-14)



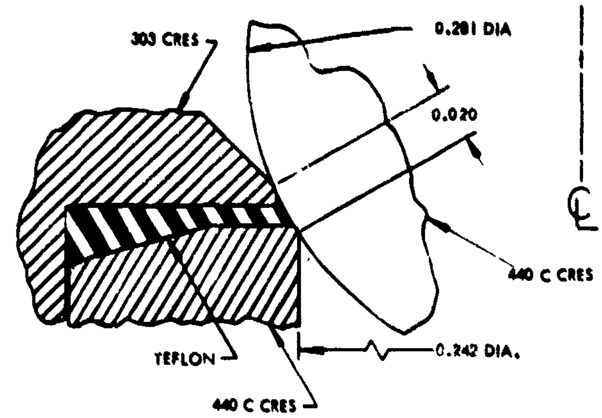
DESIGN PARAMETERS

FLOW MEDIA: AIR, NITROGEN, HELIUM, LIQUID OXYGEN
OPERATING PRESSURE: ADJUSTABLE 30 TO 60 PSIG
OPERATING TEMPERATURE: -300 TO +160 F
SIZE: 4-IN. LINE

Figure 6.2.3.11l. Conical Poppet Tank Vent Valve with Lip Seal

(Reference 35-12; Rocketdyne, a Division of North American Aviation, Incorporated design, p. B-20)

and deserves study by any engineer charged with the responsibility of designing a hard-seat poppet valving unit. Although the discussion that follows treats only the hard-seat poppet valving unit, a number of the principles may be applied to soft-seat poppet valving units. In addition, the significance of material properties and the character of the interfacing surfaces is treated in some detail under "Static Seals," Sub-Section 6.3. Other sources of information pertinent to seat and poppet sealing are References 35-12, 46-32, 46-33, and 152-4.



DESIGN PARAMETERS

FLOW MEDIA: HELIUM
OPERATING PRESSURE: 4500 TO 4650 PSIG
OPERATING TEMPERATURE: -30 TO +160F
CAPACITY: 0.0021 IN. EQUIVALENT ORIFICE
TUBE SIZE: 1/4 IN.

Figure 6.2.3.11m. High-Pressure Spherical Poppet Soft-Seat Valving Unit

(Reference 35-12; Aqualite Corporation design, p. B-44)

The following basic elements of the seat and poppet sealing problem will be summarized:

a) Leakage flow

b) Leakage path

c) Seating surface texture deformation

d) Contact stress distribution

e) Configuration considerations

f) Seating surface fabrication

g) Seating surface inspection

h) Model test results.

a) **Leakage Flow.** Established gas flow equations can be applied to valve seat leakage throughout the entire flow spectrum. For a given fluid and equivalent parallel plate closure, the established flow regimes are:

   a) Nozzle

   b) Turbulent channel (similar to nozzle but with a correction for friction effects)

   c) Laminar

   d) Transition

   e) Molecular

   f) Diffusion laws.

Although the applicable range for nozzle, turbulent channel, and laminar flows may be determined by established criteria for a measurable gap (h_o), the applicability of theoretical criteria for high-pressure transition and molecular flow is uncertain due to the very small gap (see Figure 6.2.2.2b). However, since the molecular component is theoretically indicated at higher flow levels than experimentally determined, it provides a factor of safety commensurate with the lack of data. For the 1-inch diameter model shown in Figure 6.2.2.2c, it was feasible to apply the laminar flow equation for leakages ranging from 10^{4} to 10^{-1} scim, despite the narrow theoretical laminar flow range shown in Figure 6.2.2.2b.

Examination of flow regime criteria shows that valve seat leakage is usually laminar. This is basically a function of the small gap (h_o) relative to land width (L) necessary in most valve seats for obtaining effective closure. The near impossibility of nozzle or orifice flow constituting leakage flow past a closed poppet and seat was demonstrated in Reference 34-14. Here it was shown that to achieve leakage rates of the order of 100 scim, the seat land width of a 1-inch diameter poppet valve could be no wider than 0.0001 inch ($L = 0.0001$ inch) for orifice flow to exist. Other practical considerations limit the probability of attaining pure molecular level of leakage with most metal-to-metal seating configurations.

Liquid leakage, although not evaluated in Reference 35-14, has been demonstrated in the literature to follow a pattern similar to the above. At very low levels of leakage, the effects of surface tension may be a significant leakage limiting factor; however, no correlative data is available. Therefore, the application of gas-liquid conversions based on the laminar flow equation should provide a conservative prediction for liquid leakage (see Equation 6.3.2.3).

Leakage flow comparison experiments with nitrogen, helium, argon, and hydrogen gases have proved the correlative accuracy of the laminar flow equation. Consequently, for volumetrically measured gas leakage, viscosity is the controlling parameter. For this reason it should be noted that there is little difference in the leakage rates of various gases through a given leak path in the laminar flow regime.

b) **Leakage Path.** The leakage path created by two opposed annular surfaces under a normal load is a function of the relative dimensions of various form and surface texture errors. Land crown, out-of-flatness, out-of-parallel or out-of-roundness, and taper constitute the usual form errors. Surface texture errors are roughness, waviness, nodules, pits, and scratches. Under light loads the leakage gap will be controlled by errors causing material to protrude above some average plane. The partial load effect of each of these errors will be submerged in the total load in proportion to their relative size. At higher loads, the predominant flow path will be through roughness, scratches, and interconnecting pits.

Approximation of these gap leakages may be determined by integration of height variation to determine equivalent parallel plate heights (h_e). Since the loading effects of composite form errors cannot be precisely separated, the assessment of leakage from test results for a known seat geometry and surface texture will allow a conclusion as to the probable cause for a given leakage level. These data should provide a sound basis for a practical compromise between geometry, surface texture, and leakage requirements.

Tests have shown that scratch leakage is relatively independent of load. Assuming a V-shaped scratch model, the equivalent laminar flow parallel plate height for this configuration is 0.63 h, where (h) is the scratch depth. Evaluation of numerous models has shown that all lapped surfaces have scratches, and scratch leakage can be calculated, providing accurate measurements are made of depth, width, length, and quantity. In practice, many seating surfaces are remade to remove visible scratches, although these scratches may make an insignificant contribution to total leakage. Application of suitable scratch criteria to engineering drawings can reduce this waste.

With the lapped valve seat, the significant leakage path is usually through the minute interstices between contacting roughness asperities. (This is assuming that sufficient load exists to flatten nodules.) Where the surface roughness lay of opposed surfaces is multidirectional or a crossing of unidirectional scratches, the controlling leakage parameter is the sum of the average PTV (peak-to-valley) heights for each surface.

With opposed surfaces having circular lay, the leakage path is through the gap between intersections caused by lay irregularities and eccentricity. Since the PTV height sum (for both surfaces) adds only at each real contact intersection (which also blocks flow), the radial gap is only one-half of the PTV sum of the cross lay roughness heights between intersections. Consequently, circular lay surfaces under just enough load to contact the roughness level will have approximately one-eighth the leakage of similarly loaded unidirectional or multidirectional lay surfaces.

c) **Seating Surface Texture Deformation.** The analytical and experimental work of Reference 35-14 has resulted in the conclusion that valve seating is essentially a totally elastic process of forcing two relatively flat surface textures into intimate contact. Plasticity may play an important role on the initial few contacts, particularly for the less hard metals; however, after the seat is set, very little plastic flow takes place on subsequent cycles. This conclusion is somewhat in conflict with the five regimes of sealing (Reference 46-19) described in Detailed Topic 6.3.2.1 as well as the general conclusion adhered to by many researchers that metal-bearing area is developed by plastic asperity flow (Bowden and Tabor theory). It has been generally recognized that under certain conditions elasticity contributes significantly to bearing area under load. For hard smooth materials it has been conclusively demonstrated in Reference 35-14 that interfacial elasticity is the predominant mechanism of closure, although there are exceptions which must be recognized.

With some valves having relatively large geometric errors or very light loads, the situation of complete land contact may not be achieved. Other than the special cases presented in Reference 35-14, deformation analyses for errors such as land taper and conical poppet tilt have not been formulated due to their extreme complexity. Seat land taper which results in theoretical line contact at zero load is exceedingly difficult to assess because of the unknown contact width and stress variables. Below a certain load, a tapered seat land will not seal as well as a seat land with full land width contact. This is due to unavoidable edge roughness and waviness combined with the relatively narrow real contact land. As load is increased, the tapered seat land seals more effectively than the full contact land due to increased real land width and higher contact stress at the initially contacted corner. The potential of edge plastic flow and cutting of the opposed surface (if overlapping) poses a danger in seat land configurations having significant taper, due to the possibility of recontacting with the plastically raised metal edges overlapping. This situation is somewhat improved with valving units that accurately control the axial and radial alignment of poppet and seat, or have the overlapping member substantially harder (> 20 percent higher Rockwell C) than the narrower land part.

From the test results of Reference 35-14 it is concluded that all surface textures have a number of identifiable geometrical errors which affect the load deformation characteristic. It would be an oversimplification to assume that the roughness parameter alone is the significant topographical variable. The finest surface with insufficient load to flatten a waviness component would have excessive leakage compared with roughness expectations. Results of stress-leakage tests (Reference 35-14) have indicated that for lapped models an apparent contact stress between 300 and 600 psi was required to establish leakage equivalent to the sinusoidal analytical model idealized by the recorded PTV inspection data. Below this load level, the effects of waviness, nodules, and variable edge conditions predominate and, except in specific measured instances, their effect on the stress-leakage characteristic is undefined.

Roughness deformability is a function of the relative profile sharpness. The sharpness characteristic is broadly described by a PTV average slope angle. Similar to a corrugated tin roof, the sharper the angle, the more rigid the structure. Deformability is also controlled by such elastic material properties as modulus and Poisson's ratio. It follows that under equal loading and test conditions, a "rough" surface of shallow profile could leak less than a "fine" surface having a sharp or pitted texture. Naturally, the rougher surface will leak more below a given load level.

Seating surfaces with nearly concentric circular lay roughness profiles are more readily deformed than comparable multidirectional surfaces. This increased deformability, combined with a smaller effective radial flow gap (1/2 PTV sum of each surface), explains the observed reduced leakage with circular lay surfaces. From comparisons of unidirectional and circular lay surfaces, it has been concluded that comparable leakage rates may be obtained, with circular lay surfaces having two times the roughness of unidirectional lay surfaces for apparent stress of 1000 psi, and four times for 10,000 psi. While these data are considered conservative for the models tested, it must be emphasized that the evaluation is only on a roughness basis and does not include other geometric variables.

d) **Contact Stress Distribution.** Valve seats are most often designed so that one seating surface overlaps the other. With small corner radii, edge-bearing stresses are much greater than the apparent seating stress. This effectively reduces the available load apportioned to the center of the land which, consequently, results in less asperity deformation in this area. In essence, the peripheral edges of the seat may become the predominant sealing areas, with the land center acting as a trough. Under these conditions it may be hypothesized that edge radial scratches can conduct significant leakage even though they do not bridge the land. Furthermore, with sharp edges the maximum contact stress is developed in areas of minimum geometric regularity, since discontinuities are more prevalent near machined boundaries (i.e., corners).

From the previous conclusion it follows that the crowned and/or edge-dubbed seat land configuration produces the optimum contact stress distribution for most effective sealing. This is evident particularly where sharp edge discontinuities may cause plastic deformation. A more subtle advantage lies in the distribution of effective load over a wider land width in the central land area, where the finish is likely to be more uniform. Since the load is concentrated at the land center, edge scratches not bridging the land may be neglected.

The contact stress and deformation analyses for crowned and dubbed seat lands found in Reference 35-14 are the only known analytical approaches to define real geometric bearing area. In lieu of an analytical approach to taper deformation, it is suggested that estimates of taper-deformed land width may be approximated by geometric comparisons between the crowned and tapered seat land. Due to the unrestrained edge, a greater deformation and wider land width will be produced with the taper than with the crown. Contact stresses also will be higher. Until proven equations are developed for the taper conditions, the crown analysis will provide a limiting design criteria.

e) **Seating Surface Fabrication.** Turned and ground models have demonstrated the great potentialities of these methods in producing seating surfaces having circular lay with no radial scratches. With machines designed specifically to produce accurate geometry and fine finish, it is expected that either method could be developed to maintain flatness (or contour) and surface roughness errors below 10 micro-inches over most seating lands. However, until such machines and techniques are developed, lapping will remain the only process for almost automatically obtaining accurate and smooth conformal geometry.

A high degree of conformity is more readily obtained with flat lapped surfaces than with either conical or spherical geometries. This is because of the directly comparable single plane geometry of the flat surface which allows simple inspection with optical and stylus type surface measurement equipment.

Like flat surfaces, spherical contours are easily obtained by lapping because of the constantly changing path between the lapping parts, which naturally generates a round surface at any section. Spherical seating, however, involves the additional radius parameter which can result in serious taper error if the seat and poppet radii are significantly different. The radius of a spherical seat land cannot be measured with sufficient accurecy to determine this error. Therefore, analytical methods must be employed to ensure seating conformity. It is possible that accurate conformance between poppet and seat may be obtained by match lapping the poppet and seat with a suitably fine compound.

Unlike flat and spherical surfaces, the conical geometry is not self-generating and must be obtained through an external datum plane such as the ways of a lathe. Moreover, the difficulties attendant with measuring differential seating angle (taper) make this the least desirable geometry for most applications. Like spherical surfaces, accurate conformance may possibly be obtained by match lapping; however, the additional problems of poppet ridges and severe crowning due to axis wobble must be considered.

All metal seating surfaces should be polished following geometric finishing to remove what might be termed "the roughness on the roughness." This process, accomplished at low speed with a soft lap such as paper, should remove no more than 10 percent of the PTV roughness. Polishing produces a specular surface which reveals surface flaws and scratches. It further reduces the sharpness of real asperity contacts, thus yielding improved deformability and lower asperity stress. The result is less leakage and longer life.

f) **Surface Inspection.** The secret to fabricating fine surfaces lies in the ability to measure them. The interference microscope provides this capability down to about 1 microinch PTV. When the controlling parameters are defined and can be observed and measured, the correlation of cause (roughness) and effect (leakage) automatically points to the direction for improvement.

The basic limitation in microinterferometric interpretation of surface texture is the horizontal resolving power of the ordinary microscope. When surface discontinuities (pits, scratches, etc.) become too narrow to follow a single interference band into and out of a given defect, the interference technique cannot be used for accurate depth measurement. With the Leitz instrument and Polaroid photos used in the effort described in Reference 35-14, horizontal resolutions were made down to about 20 microinches. Unfortunately, with the finer valve seats, much significant surface texture lies below optical resolution. For these cases the interference method may be used on a comparative basis (as with roughness comparison masters).

Curved surfaces present an additional variable which must be considered when measuring surface texture variables with the interference microscope. Since interference band width depends upon a relatively small angle between a flat reference mirror and the viewed object, conical and spherical surfaces must be rotated to examine any given area. With conical surfaces, gross geometry may be measured along the cone axis. However, since spherical surfaces curve continuously, a similar comparison is not possible with a flat reference and only texture variables may be assessed.

With simple flat surfaces, the optical flat is most suitable for gross-geometry measurements. Where very small or complex shapes and a combination of dimensions are involved, the ultimate measuring system is comprised of the precision surface or spindle, electronic gage, and profile data recorder. When this equipment is supplemented by low-power interference and plain microscope measurements, seating geometry may be accurately assessed and correlated with leakage measurements.

It is highly recommended that anyone concerned with the inspection of poppet valving unit seating surfaces be familiar with References 35-14, 46-32, and 152-4.

g) **Configuration Selection Criteria.** Selection of flat, conical, and spherical geometry must be based upon specific application requirements. This is best accomplished by a critical examination of the advantages and disadvantages inherent in design, fabrication, inspection, and performance.

Conical or spherical seating has several basic advantages over the flat surface. The most apparent is the mechanical load advantage effect which allows a working contact stress to be developed with reduced loads. Since the land is developed from the intersection of right angle surfaces, a narrow width is easily obtainable. Once seated, the conical and spherical configurations have a force component which resists lateral motion due to vibration, often a serious disadvantage in flat seating since high-frequency vibration can cause failure in minutes if the poppet is permitted to scrub laterally over the seat.

An inherent disadvantage of conical and spherical geometry is the slippage which takes place during seating. Since conformance is measured in millionths of an inch, poppet and seat axes cannot be installed or guided with sufficient accuracy to avoid this wearing shear. Moreover, with large loads or small included angles, elastic entry of the poppet into the seat will add to the wear problem.

Whereas spherical seats have the advantage of perfect alignment once seated, conical seating has the additional potential error of axis tilt. Alignment moments for the tilted cone are a complex function of the load application point and interfacial friction. Unless the seating load is applied below the seating line (toward the apex) and perfectly axial, conical seats are probably not self-aligning (see Figure 6.2.3.11v). The conical configuration, therefore, must have generally a narrow land and higher seating loads to reduce the gap caused by axis tilt.

From a performance viewpoint, the flat configuration is the natural choice for lowest leakage combined with maximum life. Spherical seating offers the load and lateral retention advantages, but with greater wear potential. Conical seating should probably be selected on an economical basis or where experience has indicated adequate performance.

h) **Conclusions from Model Test Results.** While indicating a general agreement with simplified seating analyses, model tests (Reference 35-14) have demonstrated the dependence of leakage at any given load upon the occurrence and distribution of a multiplicity of surface variables.

The majority of tested flat models indicated that roughness and roughness lay were the significant load-leakage geometric parameters between 500 and 20,000 psi apparent seat stress. Analysis of these data has indicated that a correlation of the roughness PTV parameter and the laminar flow equation is capable of predicting leakage change with load within a factor of about two over this stress range. This estimate is derived from the correlation of two empirical equations based upon data from 16 flat model tests. Because leakage over a limited span follows a nearly constant inverse relationship with apparent stress,

$$Q \sim \frac{1}{S^n}$$

these relationships may be combined in a single equation for leakage between 500 and 20,000 psi apparent stress. For multi- or unidirectional lay surfaces

(Eq. 6.2.3.11a-1)

$$Q_u = \frac{100\, D_s H^3 \,(p_1^2 - p_2^2)}{\mu L T S^{2/3}}$$

and for circular lay surfaces

(Eq. 6.2.3.11a-2)

$$Q_c = \frac{2 \times 10^4 D_s h^3 \,(p_1^2 - p_2^2)}{\mu L T S^{3/2}}$$

where:

Q_u = unidirectional lay leakage, std in^3/min (scim)

Q_c = circular lay leakage, std in^3/min (scim)

D_s = mean seat diameter, in.

H = peak-to-valley height for two surfaces, in.

h = peak-to-valley height for one surface (opposing surfaces assumed identical), in.

p_1 = upstream pressure, lb$_f$/in^2

p_2 = downstream pressure, lb$_f$/in^2

μ = absolute viscosity, lb$_f$ min/in^2

L = radial seat land width, in.

T = gas temperature, °R

S = apparent seat stress, lb$_f$/in^2

Leakage values computed directly from roughness measurements should be accurate within a factor of 10, although circular lay surfaces may have much lower leakage than predicted due to the conservative slope employed in correlation. Combining the above empirical equations in unidirectional lay roughness (h_u) leakage and circular lay roughness (h_c) leakage results in

(Eq. 6.2.3.11a-3)

$$\frac{Q_u}{Q_c} = 0.04\, S^{5/6} \left(\frac{h_u}{h_c}\right)^3$$

Thus, unidirectional lay leakage at light loads (500 to 1000 psi apparent stress, S) will be about eight times that for an equal circular lay surface. At 20,000 psi the ratio is approximately 150:1. For a given leakage requirement, surfaces with circular lay may be considerably more rough than with unidirectional lay, depending upon the load as previously indicated.

Evaluation of circular lay eccentricity has led to the tentative conclusion that moderate amounts of lay eccentricity do not significantly affect leakage. Due to the small dimensions involved, numerical evaluation of this parameter has not been performed.

Because of the overriding influence of roughness, no direct comparison of the material parameter could be made. Where relatively large deformations take place to effect sealing, it is concluded that the more elastic material will require less load. The apparent flatten stress equation (based upon Hertz contact) indicates a direct relationship between flatten stress and elastic modulus (E). In view of many other factors, however, the selection of a material for seating should probably not be influenced by this parameter. Materials and processes investigated for seat and poppet applications in the course of the effort reported in Reference 35-14 included 440-C steel, 17-4 PH steel, tungsten carbide, and aluminum with processing including turning, grinding, lapping, polishing, anodizing, liquid honing, passivating, and gold plating. Other recent investigations (Reference 81-5) have evaluated the possible advantages of using Berylco Nickel 440 for both seats and poppets.

Combined with circular lay, the crowned land surface designed to develop an adequate land width is the optimum seating configuration. The advantage of this configuration is particularly evident at low loads where high contact stresses are developed to decrease roughness height. In some cases, however, it may be determined that the load geometry relationship is such that crown dimensions cannot be measured (i.e., less than 1 microinch fall-off) for the desired land width. The effect of adequate corner dub then becomes more critical and should be applied keeping the land essentially flat.

Cycle testing has shown that seat degradation is closely related to the peak load and interfacial motion occurring during impact. The basic wear mode observed was one of fretting. Where the land was sharp, fracture and plastic flow occurred at the edge boundary. By making flat poppet and seat-opposing surfaces of similar geometry and material, lateral differential deformation (and thus fretting) was greatly reduced. Edge damage was eliminated by land crowning and dub-off.

The poppets which were cycle tested experienced a 25-microinch maximum lateral axis motion relative to the seat. Under these conditions it was apparent from the results that a certain amount of impact was beneficial to seating. This was attributed to the plastic deformation of nodules, high asperities, and sharp edges which intermeshed with the matched opposing topography. The net effect was a reduction in leakage with cycling, even though surface roughness and contaminant-induced flaws increased. Combined with previous findings, it is concluded that optimum seating is provided by the flat dubbed or crowned configuration restrained to have no lateral motion (as might be obtained by a flexure mounted poppet).

i) **General Conclusions.** Other than the gross-geometry parameters such as land width, seat diameter, and general configuration, valve closures are not designed in the full sense of the word. Engineering drawings do not, for the most part, specify the real controlling parameters, but only point the way toward a fabrication process. Few valve poppet and seat drawings specify the 1/4- to 4-microinch AA finishes, and 1- to 50-microinch flatness (or roundness) necessary to meet performance requirements. Scratches and pits are uncontrolled. Consequently, metal valve seats and poppets are presently evaluated primarily by leakage test results rather than by conformance with drawing finish and flatness (or roundness) specifications. The limitations of present inspection tools, combined with the prodigious amount of detail inspection required, preclude specifying exact surface profiles on engineering drawings. Nevertheless, in many instances the ultimate performance of a seat is dependent upon a fabrication process or control completely uncalled for by the drawing.

To control the reliability and performance of low leakage components, all geometric parameters should be dimensionally specified to the level dictated by performance. Where such controls exceed measurement capabilities, or would be uneconomical to dimensionally prove, either a final or comparative leakage test should be allowed in lieu of these controls. As shown in Reference 35-14, the leakage test is cubically more sensitive to seating gap than are direct measurements. Therefore, with some dimensions measured to ensure basic geometry, the more difficult assessments as embodied in surface texture may be indirectly measured as a composite group by means of the leakage test. With the information provided, comparative tests may be established under a variety of conditions to ensure proper functioning in the final assembly. The most important aspect in defining such tests would be to ensure loading and geometric similarity with the final assembly.

Actuation Forces. The forces acting on poppet valving elements have been the subject of numerous investigations, as may be seen from the list of references at the end of this sub-section. The static condition wherein the valve is closed may be treated separately from the open condition with its associated flow forces, but failure to appreciate the transient condition between the two states can result in the chattering phenomenon discussed in the following paragraphs and in Detailed Topic 5.5.7.2.

Parameters defining the static forces acting on the poppet in the closed position include:

a) Seat land diameter and width

b) Pressure differential across valve (magnitude and direction)

c) External force either seating the poppet or tending to open it.

Figure 6.2.3.11n illustrates the influence of geometry on the force balance of a seated flat, conical, or spherical poppet. The calculation of the net pressure force acting on the poppet is very straightforward, with the exception of determining the effect of pressure over the seat land area. For gross force balance analysis it is common practice to assume either the ID or OD as the effective sealing diameter, based upon known geometry or a worst tolerance condition. Once the poppet has lifted sufficiently for laminar flow to be established, the effective diameter may be accurately calculated as described below. In the case of the model shown in Figure 6.2.2.2c, flowing gaseous nitrogen, a lift of from 10 to 100 microinches was required to establish laminar flow (Figure 6.2.2.2b).

The curves presented in Figure 6.2.3.11o show fluid flow force as a function of stroke for conical, spherical, and flat poppet valving units with flow in either direction. Unlike rotary-element or gate-type valving units, poppet valving units have no clearly defined limit of stroke, for it is possible that the poppet may be translated so far away from the seat that any further motion will have absolutely no effect upon flow through the port. This feature is evident in the qualitative plots of stroke versus percent of maximum force for typical poppet configurations and simple seats given in Figures 6.2.3.11p, q, r, and s. The slight reduction in force in the near-seated position of the overlapping flat poppet (see Figure 6.2.3.11p), resulting from reduced static pressure developed by high-velocity flow across the seat land, is eliminated in the cup-shaped poppet (see Figure 6.2.3.11q)

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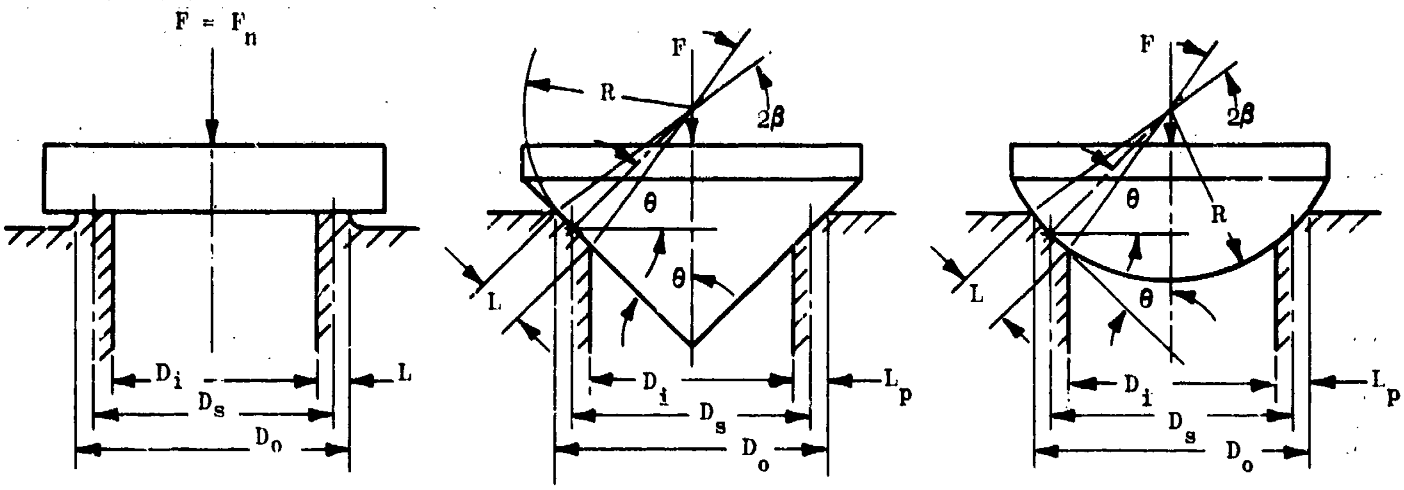
SUPERSEDES: OCTOBER 1965



$$L = 1/2\ (D_o - D_i)$$
$$D_s = 1/2\ (D_o + D_i) = D_i + L$$
$$A_s = \pi D_s L$$
$$S \equiv \frac{F}{A_s} \equiv \frac{F}{\pi D_s L}$$

$$L_p = 1/2\ (D_o - D_i) \equiv L \sin\theta$$
$$L = 2R \tan\beta \cong 2\beta R$$
$$D_s = D_i + L_p$$
$$A_s = \pi D_s L$$
$$A_{sp} = \pi D_s L_p$$
$$F_n = F/\sin\theta$$
$$F_r = F/\tan\theta$$
$$S \equiv \frac{F}{A_{sp}} \equiv \frac{F_n}{A_s}$$

$$L = 2\beta R$$
$$L_p = 1/2\ (D_o - D_i) \cong L \sin\theta$$
$$D_s = 2R \cos\theta$$
$$A_s = \pi D_s L$$
$$A_{sp} = \pi D_s L_p$$
$$S \equiv \frac{F_n}{A_s} \equiv \frac{F}{A_{sp}}$$

L = total radial seat land width, in.

D_o = outer seat diameter, in.

D_i = inner seat diameter, in.

D_s = mean seat diameter, in.

$A_s = \pi D_s L$ = apparent seat land contact area, in^2

S = total normal apparent seat contact stress, lb/in^2

F = total axial seat load, lb_f

L_p = total radial projected seat land width, in.

R = spherical or cylindrical radius, in.

β = angle illustrated, radians

F_n = total normal seat load, lb_f

$A_{sp} = \pi D_s L_p$, apparent seat land normal projected contact area, in^2

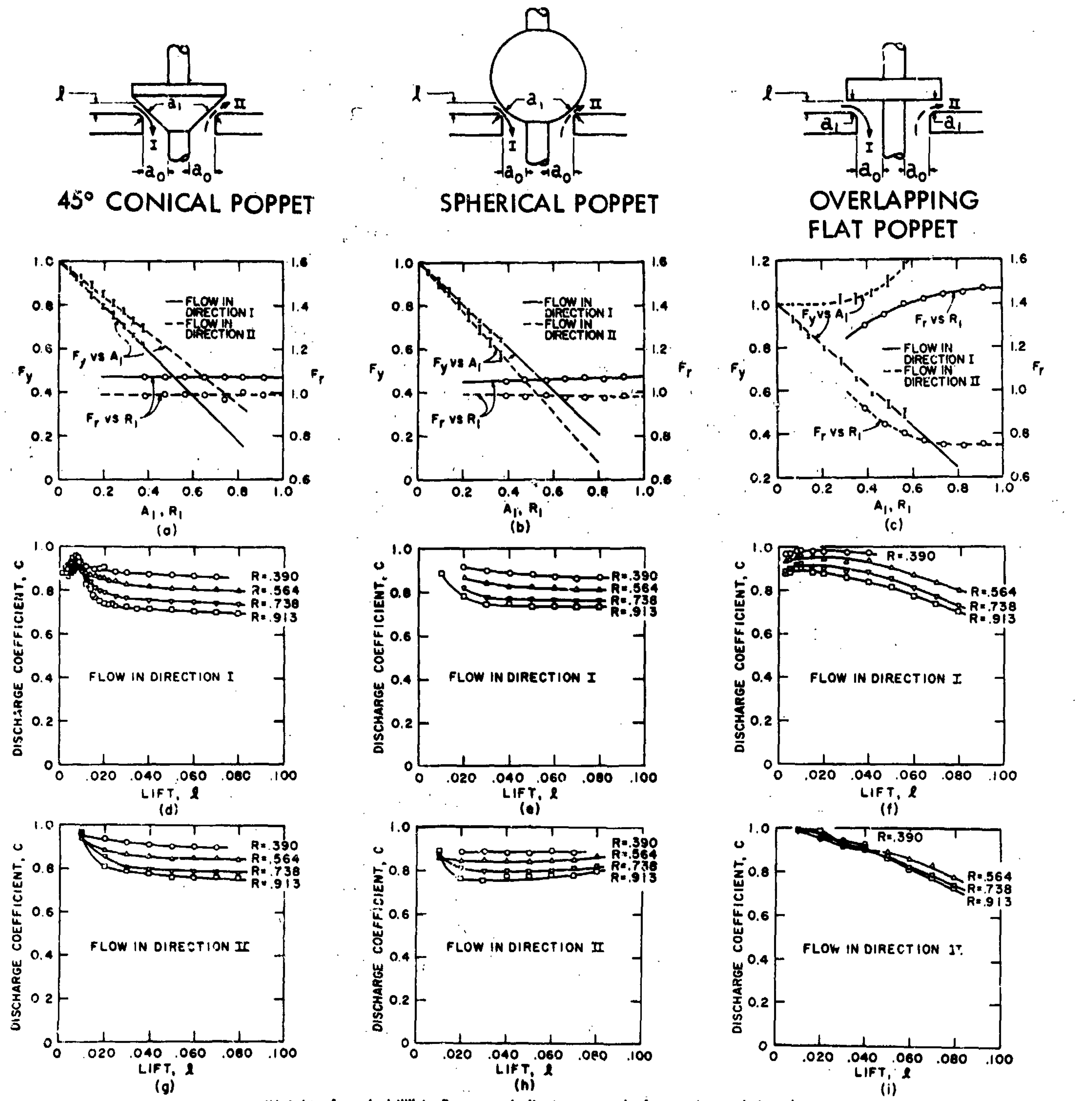
θ = poppet and seat included half (seating) angle, radians

Figure 6.2.3.11n. Flat, Conical, and Spherical Seating Equations

$\bar{a}$ = seating area, in^2
a_1 = valve opening area, in^2
a_o = annular orifice area, in^2
f = measured flow force, lb_f

f_o = force on seated poppet, lb_f $= \bar{a}(P_1 - P_2)$, lb_f
P_1 = upstream pressure, lb_f/in^2
P_2 = downstream pressure, lb_f/in^2

A_1 = valve opening area ratio $= a_1/a_o$, dimensionless
R_1 = pressure ratio $= P/P_1 = P_2/P_1$, dimensionless
C_1 = discharge coefficient, dimensionless
F_y = flow force ratio $= f/f_o$, dimensionless
F_r = flow force function $= f_o/a_o\,(P_1 - P_2)$, dimensionless



(Height of symbol "I" in F_y curves indicates spread of experimental data.)

(See Sub-Topic 7.4.8 for complete discussion of these curves)

Figure 6.2.3.11o. Force and Flow Characteristics for Three Typical Poppets
(Adapted from "ASME Trans. Series D, Journal of Basic Engineering," June 1961, vol. 83, no. 2, D. H. Tsai and E. C. Cassidy, Copyright 1961, American Society of Mechanical Engineers, New York, New York)

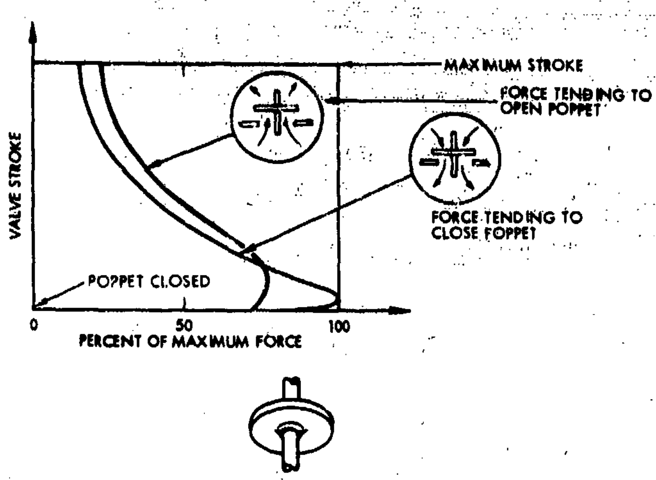


Figure 6.2.3.11p. Overlapping Flat Poppet Flow Forces
(Adapted from "Instruments," December 1950, vol. 23, no. 12, R. A. Rockwell, Copyright 1950, Instruments Publishing Co., Inc., Pittsburgh, Pennsylvania)

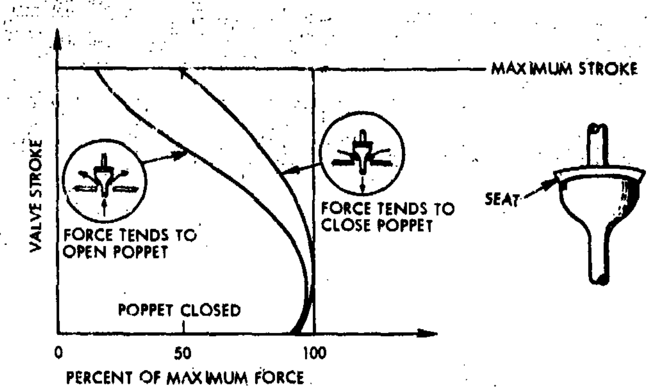


Figure 6.2.3.11r. Parabolic Plug Poppet Flow Forces
(Adapted from "Instruments," December 1950, vol. 23, no. 12, R. A. Rockwell, Copyright 1950, Instruments Publishing Co., Inc., Pittsburgh, Pennsylvania)

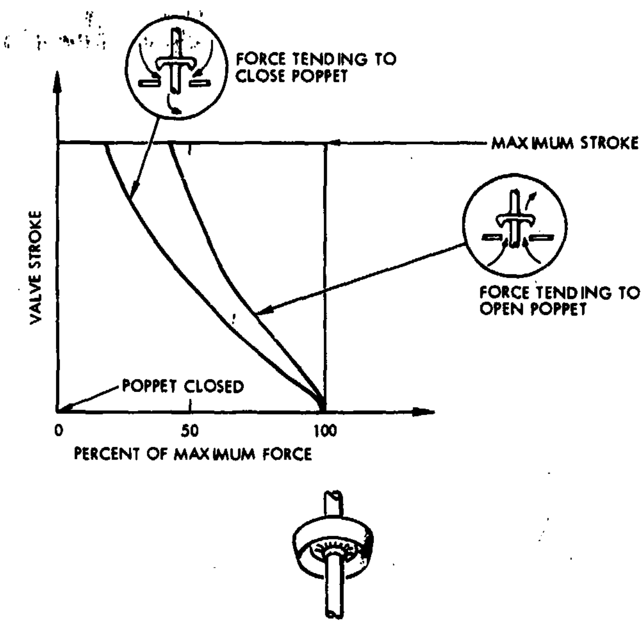


Figure 6.2.3.11q. Cup-Shaped Poppet Flow Forces
(Adapted from "Instruments," December 1950, vol. 23, no. 12, R. A. Rockwell, Copyright 1950, Instruments Publishing Co., Inc., Pittsburgh, Pennsylvania)

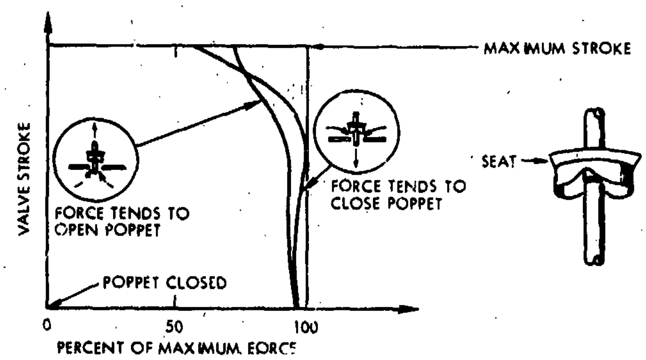


Figure 6.2.3.11s. V-Port Plug Poppet Flow Forces
(Adapted from "Instruments," December 1950, vol. 23, no. 12, R. A. Rockwell, Copyright 1950, Instruments Publishing Co., Inc., Pittsburgh, Pennsylvania)

with knife-edged seat lands. The tailoring of actuation forces as a function of stroke demonstrated by the parabolic plug poppet (see Figure 6.2.3.11r) and the V-port plug (see Figure 6.2.3.11s) may be compared with flow guide and port contouring of sleeve valving units. The plug version of poppet valving units usually uses a conical, flat, or spherical seat and a corresponding seat ring on the poppet above the plug for sealing purposes. The plug or contoured skirt exists primarily to provide desired flow characteristics, but also affects actuation force, as shown. Reference 26-59 indicates the significance of the housing configuration by concluding that smaller downstream chambers result in higher flow forces. This is logical because, for a given poppet, a smaller housing will require higher fluid velocities and hence greater fluid dynamic drag on the poppet.

Static pressure distribution across a valve seat is of interest to the valve designer in determining the flow forces associated with a particular configuration. These forces determine such external performance parameters as cracking and reseat pressures in relief valves, or influence stability in the case of regulators. The theoretical aspects of pressure distribution for the various flow regimes as described in Reference 35-14 are discussed in the following paragraphs.

Nozzle Flow Regime. There has been a considerable amount of analytical work conducted on converging nozzles, particularly for compressible flow. However, the average poppet valve in the open position has essentially constant cross section along the flow path and hence is analogous to a short-tube orifice. The pressure profile along such an orifice configuration is similar to the converging nozzle because the flow separates from the walls after entering the seating separation and exits with the stream contracted (vena contracta). Unfortunately, this analysis is basically qualitative, and the determination of the pressure profile in this configuration must be determined experimentally. Figure 6.2.3.11t illustrates how the static pressure distribution across the seat land always tends to open the valve, regardless of the direction of flow. (Note, however, that with outward flow and OD sealing, this net static pressure distribution will be less than when the valve was closed.)

Turbulent Channel Flow Regime. For the incompressible fluid, the pressure drop in turbulent flow regime is a direct function of the length of the path; therefore, the pressure profile is a straight line across the seat land. This results in an effective seat diameter location at the land midpoint for the case of flow discharging to a negligible (atmospheric) downstream pressure.

The pressure distribution for a compressible gas is somewhat more complex. In general, the pressure profile along the channel approximates a straight line for the higher Reynolds numbers. However, as the valve model closes (corresponding to a decrease in parallel plate channel height, h_c, and Reynolds number), the pressure profile progressively approaches the parabolic shape found in the laminar flow regime.

Laminar Flow Regime. Unlike the analysis of pressure distribution in the nozzle and turbulent channel regimes, the laminar consideration is straightforward. A general equation for pressure profile may be derived from the laminar flow equation. For parallel and tapered plates, the pressure at any distance across the seat land is:

(Eq. 6.2.3.11a-4)

$$p_x = p_1 \left\{ 1 - \beta \left[1 - \left(\frac{p_2}{p_1} \right)^n \right] \right\}^{1/n}$$

where p_x = static pressure at distance x from upstream edge of seat land, lb_f/in^2

p_1 = upstream static pressure, lb_f/in^2

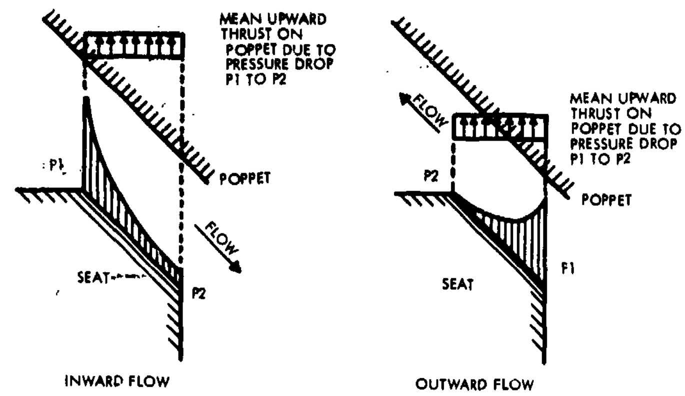


Figure 6.2.3.11t. Pressure Distribution Across a Conical Poppet Valve Seat

(Adapted from "Fluid Pressure Mechanisms," H. G. Conway, Copyright 1958, Sir Isaac Pitman and Sons, Ltd., London, England)

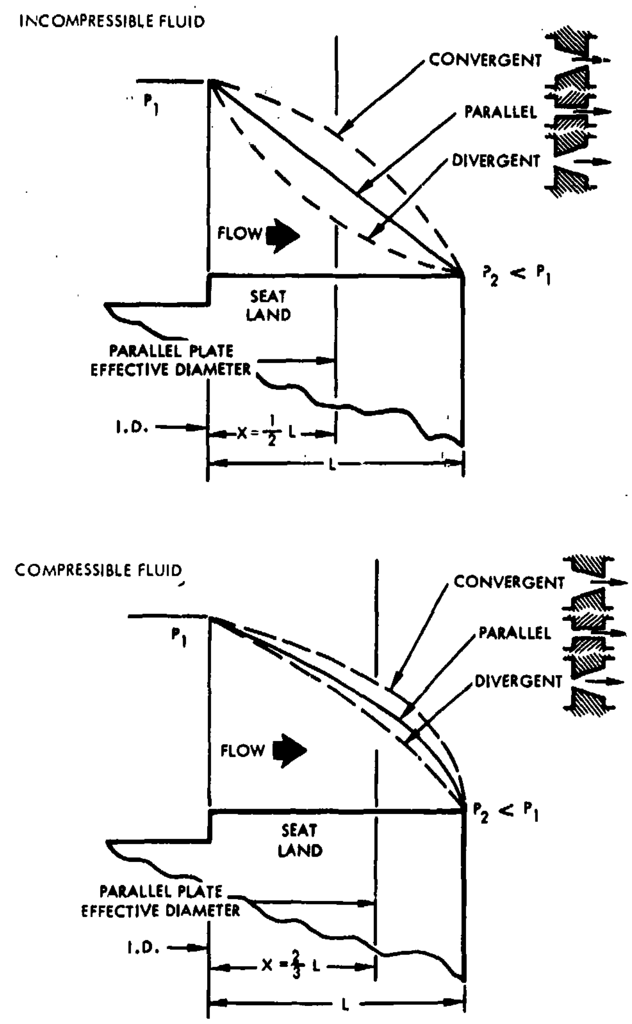


Figure 6.2.3.11u. Laminar Flow Pressure Profiles

(Reference 35-14)

p_2 = downstream static pressure, lb_f/in^2

n = 1 for incompressible flow, 2 for compressible flow

β = pressure profile factor, determined as appropriate from Equations 6.2.3.11a-5, -6, or -7

a) For parallel plates

$$\beta = \beta_{px} = \frac{x}{L} \qquad \text{(Eq. 6.2.3.11a-5)}$$

where x = distance from upstream edge of seat land, in.
L = length of seat land, in.

b) For converging plates (minimum clearance at downstream edge of seat land)

(Eq. 6.2.3.11a-6)

$$\beta = \beta_c = \frac{x}{L}\,\frac{\left[2\left(1+\frac{h_p}{h_o}\right)-\frac{x}{L}\right]\left(\frac{h_p}{h_o}\right)^2}{\left[2\left(1+\frac{h_p}{h_o}\right)-1\right]\left(1+\frac{h_p}{h_o}-\frac{x}{L}\right)^2}$$

where h_p = parallel plate channel height (minimum distance between seat and poppet), in.

h_o = taper height, in. ($h_p + h_o$ = gap at widest opening, in.)

c) For diverging plates

(Eq. 6.2.3.11a-7)

$$\beta = \beta_d = \frac{x}{L}\,\frac{\left(2\frac{h_p}{h_o}+\frac{x}{L}\right)\left(1+\frac{h_p}{h_o}\right)^2}{\left(\frac{h_p}{h_o}+1\right)\left(\frac{h_p}{h_o}+\frac{x}{L}\right)^2}$$

The pressure profile described by the parallel plate equation is linear for incompressible flow and parabolic for compressible flow. Seat land taper causes the profile to be biased in the direction of the narrow opening, as shown in Figure 6.2.3.11u.

Integration of the parallel plate profile equation (Equation 6.2.3.11a-5) for average pressure gives the following for incompressible flow:

$$p_i = \frac{p_1 + p_2}{2} \qquad \text{(Eq. 6.2.3.11a-8)}$$

where p_i = average static pressure across seat land for compressible flow, lb_f/in^2

For compressible flow:

(Eq. 6.2.3.11a-9)

$$p_c = \frac{2}{3}\,\frac{p_1^3 - p_2^3}{p_1^2 - p_2^2}$$

where p_c = average static pressure across seat land for compressible flow, lb_f/in^2

For high-pressure valves leaking to vacuum or atmosphere, p_2 may be neglected, thus the average pressure reduces to:

$$p_i \cong \frac{p_1}{2} \qquad \text{(Eq. 6.2.3.11a-10)}$$

and

$$p_c \cong \frac{2}{3}p_1 \qquad \text{(Eq. 6.2.3.11a-11)}$$

It follows that for narrow seat lands (L small relative to the ID), the effective seat diameter for laminar compressible flow is located at two-thirds the distance across the seat land and the effective diameter is given by:

$$D_e = D_s + \frac{1}{3}L \qquad \text{(Eq. 6.2.3.11a-12)}$$

where D_e = effective seat diameter or effective pressure balance diameter, in.

D_s = mean set diameter, in.

L = length of seat land, in.

For most valve seats, the land is relatively narrow and the ID circumference is very nearly equal to the OD circumference. Thus, the effect of radially spreading flow (divergency) may be neglected and the above equations are applicable. When the seat land width becomes sufficiently large with respect to the ID, radial divergence must be considered. Hence, for parallel plates, Equation 6.2.3.11a-5 becomes

(Eq. 6.2.3.11a-13)

$$\beta = \beta_p = \frac{\ell n \frac{R}{R_i}}{\ell n \frac{R_o}{R_i}}$$

where R = R_i + x, in.

R_i = inside seat radius, in.

R_o = outside seat radius, in.

The effect of radial divergence is to straighten out the parabolic curve for laminar compressible flow and, for the extreme case, reverse the curve so that the effective diameter is less than the land midpoint.

Transition and Molecular Flow Regimes. In Reference 35-14, detail consideration was not given to the transition and molecular pressure distribution because these flow regimes almost always occur under highly stressed seating conditions. Therefore, the force resulting from small differences in the effective seat area is negligible relative to the total seat force.

One further actuation force consideration is instability. As discussed in Detailed Topic 5.5.7.2, instability of a poppet valving unit is called "chatter," and is the repeated hammering of the valving element against its seat. Chattering occurs frequently in relief valves, check valves, and control valves which have poppet valving element configurations. Chattering must be distinguished from purely aerodynamic and hydrodynamic instability such as fluttering of a butterfly valving element. For chattering to occur, the valving element must be connected to an energy storing device such as a mechanical spring, or a pneumatic or hydraulic actuator which acts as a fluid spring. The spring contributes the restoring force which repeatedly reseats the valving element.

Three primary factors influencing valving element stability are seat land area, fluid flow forces, and point of load application. The effect of seat land width on pressure forces as related to valve chatter (reducing seat land width reduces chattering tendencies) is discussed in Detailed Topic 5.5.7.2. The pressure force increase resulting from admission of fluid to the seat unbalanced area (the actual seating surface) always tends to open the valve, regardless of which direction the fluid is flowing. Closely associated with seat land area is the effective seat diameter. According to Reference 35-14, both experiment and theory have demonstrated a significant variation in effective seat diameter within the near-seated region. From turbulent channel to laminar flow (Figure 6.2.2.2b), the pressure profile varies so that the effective seat diameter changes from near midland to two-thirds across the seat land width (in the flow direction), as shown previously in Figure 6.2.3.11u. Where the land pressure force is a significant part of the total pressure force, pressure profile variations will contribute to instability (chatter) in pressure-sensitive devices because the transition between the flow regimes normally takes place within a few ten thousandths of an inch of stroke.

In poppet valving units where the load from the valve stem is imparted to a hinged valving element which is free to "heel" about a point on the seat, as shown in Figure 6.2.3.11v, it may be postulated that the valving element will be unconditionally stable only if the point of load application is below the level of the valve seat contacting forces. If the point of load application is above the level of the valve seat contacting forces, the valving element is only conditionally stable. Thus, if any offset occurs it becomes aggravated rather than corrected. This concept is discussed by Pearson in Reference 196-1. A detailed analytical study of swing check valve disc motion during flow reversals may be found in Reference 96-2.

Flow Characteristics. Flow characteristics for various poppet valving units are treated in Sub-Topic 5.3.5. Conical, spherical, and flat poppet configurations are shown in Figure 6.2.3.11o, with the corresponding discharge coefficients as a function of stroke given for flow in either direction for each type. As mentioned under the discussion of actuation forces, the addition of a contoured skirt or plug to the poppet valving element may be used to achieve almost any flow versus stroke characteristic desired.

For a simple flat poppet, valving unit flow area is related to stroke by the expression

$$A_o = \pi D_i S \qquad \text{(Eq 6.2.3.11a)}$$

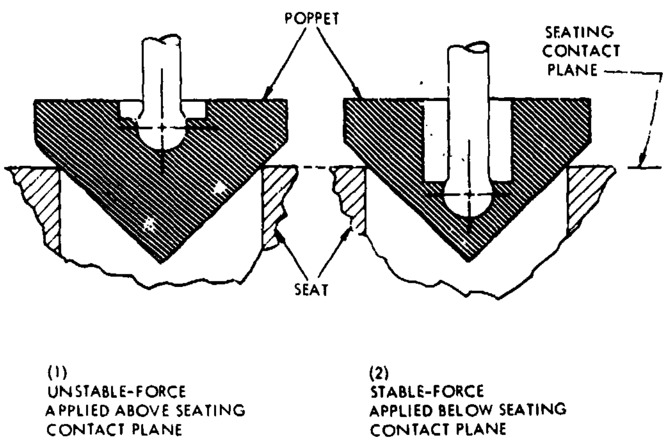


Figure 6.2.3.11v. Hinged Valving Element Stability

where

A_o = valving unit flow area, in²
D_i = internal seat diameter, in.
S = stroke, in.

The stroke required to achieve a valving unit flow area equivalent to that of the port is

$$S = \frac{D_i}{4} \qquad \text{(Eq 6.2.3.11b)}$$

An approximation of resistance coefficient for a flat poppet valving unit may be obtained from the following expression (Reference 19-229):

$$K \approx 1.3 + 0.2\left(\frac{A_i}{A_o}\right)^2 \qquad \text{(Eq 6.2.3.11c)}$$

where

A_i = internal port area, in²
$= \pi \frac{D_i^2}{4}$

For a conical poppet, the valving unit flow area/stroke relationship becomes a function of the cone angle, as seen in Figure 6.2.3.11v and the following equation

$$A_o = \pi S\left(D_i \tan\frac{\theta}{2} - S\tan^2\frac{\theta}{2}\right) \qquad \text{(Eq 6.2.3.11d)}$$

where

A_o = valving unit flow area, in²
S = stroke, in.
D_i = internal seat diameter, in.
θ = poppet cone angle, degrees (Figure 6.2.3.11w)

An approximation of flow resistance between 10 and 100 percent of stroke may be obtained from the expression (Reference 19-229)

$$K \approx 0.5 + 0.15\left(\frac{A_i}{A_o}\right)^2 \qquad \text{(Eq 6.2.3.11e)}$$

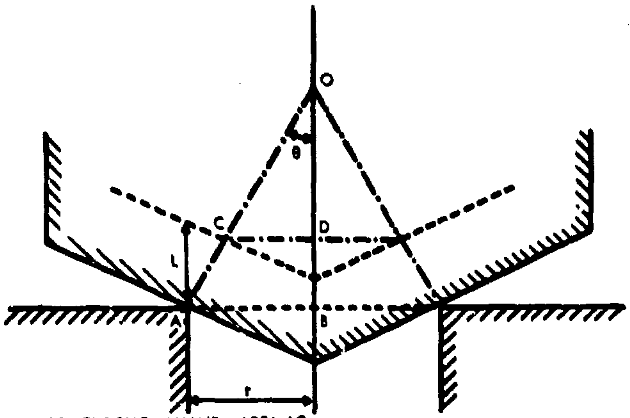
where

A_i = internal port area, in²
A_o = valving unit flow area, in²

For spherical poppets, the area versus stroke relationship is slightly more complex, being dependent upon the poppet and seat diameters illustrated in Figure 6.2.3.11x and the equation

(Eq 6.2.3.11f)

$$A_o = \frac{\pi D_i\,(S^2 - D_p S\cos\theta)}{2\sqrt{\frac{D_p^2}{4} + S^2 - D_p S\cos\theta}}$$



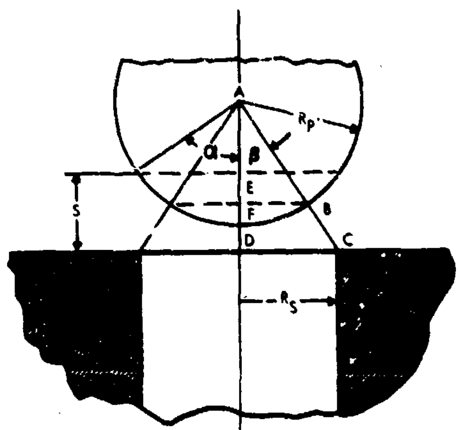
AREA THROUGH VALVE = AREA AC,
= AREA OF CONE OAB − AREA OF CONE OCD.

$$\therefore = L\frac{1 \pm \sqrt{1-\sin\theta}}{\cos\theta\,\sin\theta}r$$

IF θ = 60°, L = 1.48r = 74 PER CENT OF THROAT DIAMETER
= 45°, L = 0.92r = 46 PER CENT OF THROAT DIAMETER
= 30°, L = 0.67r = 34 PER CENT OF THROAT DIAMETER
= 0°, L = 0.5r = 25 PER CENT OF THROAT DIAMETER

Figure 6.2.3.11w. Stroke Required for a Conical Poppet Valving Unit

(Adapted from "Fluid Pressure Mechanisms," H. G. Conway, Copyright 1958, Sir Isaac Pitman and Sons, Ltd., London, England)



VALVE AREA = AREA OF CONE FRUSTRUM BC

$$A = \pi\left(\sqrt{\left(\sqrt{R_P^2 - R_S^2} + S\right)^2 + R_S^2} - R_P\right)\left(R_S + \frac{R_P R_S}{\sqrt{\left(\sqrt{R_P^2 - R_S^2} + S\right)^2 + R_S^2}}\right)$$

Figure 6.2.3.11x. Stroke Required for a Spherical Poppet Valving Unit

where

A_s = valving unit flow area, in²
D_i = internal seat diameter, in.
D_p = diameter of poppet, in.
S = stroke, in.

$$\cos\theta = \cos\left(180 \text{ deg} - \text{arc} \sin \frac{D_i}{D_p}\right)$$

For the specific instance of $D_p = 1.3\ D_i$, A_s may be approximated by

$$A_s \approx 0.75\ \pi\ D_i\ S \qquad \text{(Eq 6.2.3.11g)}$$

Flow resistance may also be approximated by (Equation) 6.2.3.11e.

Cavitating venturi poppet valving units, discussed in Detailed Topic 5.3.5.3, provide unique flow characteristics, as seen in Figure 6.2.3.11y. It is apparent from Figure 6.2.3.11y that flow rate in the cavitating region is dependent only upon stroke, whereas in the noncavitating region flow rate is a function of both pressure drop and stroke. The flow rate through a cavitating venturi valve may be expressed as

$$Q = C_D A_o V_o = C_D A_o \sqrt{\frac{2g}{w}\ (P - P_v)} \qquad \text{(Eq 5.3.5.3b)}$$

where

Q = flow rate, ft³/sec
C_D = discharge coefficient (assumed to be 0.93, although values of 0.80 to 0.85 are frequently found), dimensionless
A_o = throat area, ft² (with conical poppets this may be calculated as shown in Figure 6.2.3.11v)
V_o = theoretical throat velocity, ft/sec
g = gravitational constant, ft/sec²
w = specific weight, lb_f/ft^3
P = total pressure, lb_f/ft^2
P_v = fluid vapor pressure, lb_f/ft^2

As explained in Detailed Topic 5.3.5.3., best recovery is experienced with divergence angles of 5 to 6 degrees. Similarly, recent studies (Reference 131-26) have shown that the pintle or poppet flowguide downstream of the throat should have an included half-angle of approximately 8 degrees or less.

6.2.3.12 FLEXIBLE VALVING ELEMENTS. Of the several flexible valving element concepts described in Sub-Topics 5.2.8 (diaphragm) and 5.2.9 (pinch and flexible sleeve), only the pinch valve has been receiving increased attention for aerospace applications. The solenoid valve design illustrated in Figure 6.2.3.12a is presently being developed for food-handling service in bio-satellites. Only the pinch valve valving unit will be discussed in this detailed topic, but Reference 196-1 presents excellent detailed design data on diaphragm valves, particularly those of the Saunders Patent configuration.

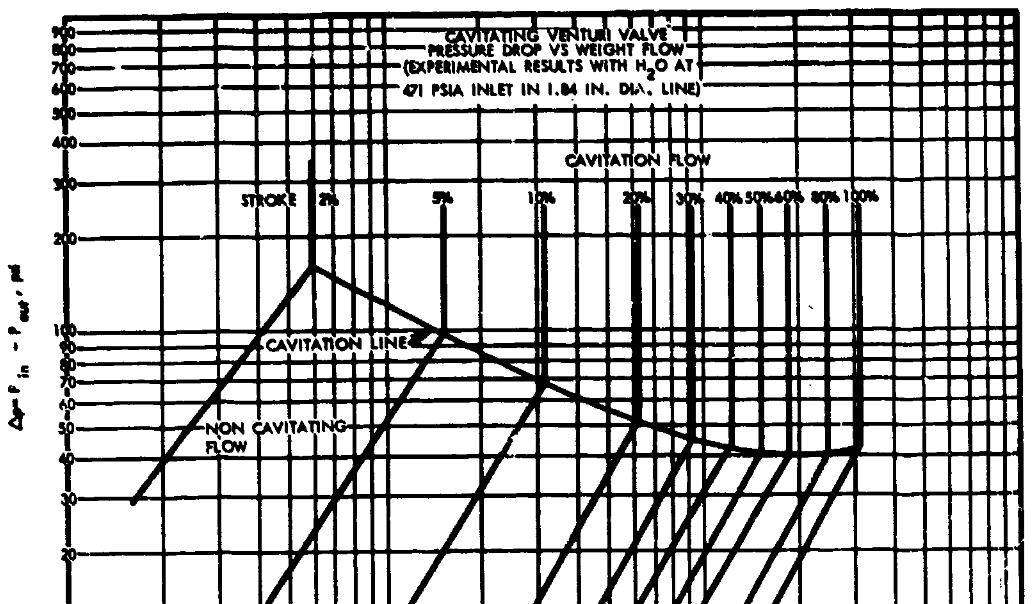


Figure 6.2.3.11y. Cavitating Venturi Poppet Valve Flow, Stroke, and Pressure Drop Characteristics
(Reference 131-26)

Configuration. The configuration of a pinch valving unit may be described in terms of the following parameters:

a) Tube diameter

b) Tube wall thickness

c) Pinch bar shape, if used (Figure 6.2.3.12b)

d) One or two moving pinch bars, if used

e) Restricted or unrestricted cavity (Figure 6.2.3.12c)

f) Length of unsupported tube, unrestricted cavity

g) Means of retaining tube in valve body.

As shown in Figures 6.2.3.12d and e, pinch valves may be actuated either by mechanical (pinch bar) means, or by pressurizing a cavity between the tube and the valve housing. Figure 6.2.3.12b shows three of the many alternate pinch bar configurations which may be used. Both pinch bars may move, as shown in Figure 6.2.3.12a, or one may be fixed, as shown in Figure 6.2.3.12d. If both pinch bars move, the restricted cavity (see Figure 6.2.3.12c), wherein the width of the pinched tube is not permitted to exceed its outside diameter, may be used. If one pinch bar is fixed, either the fixed pinch bar must be placed at the tube

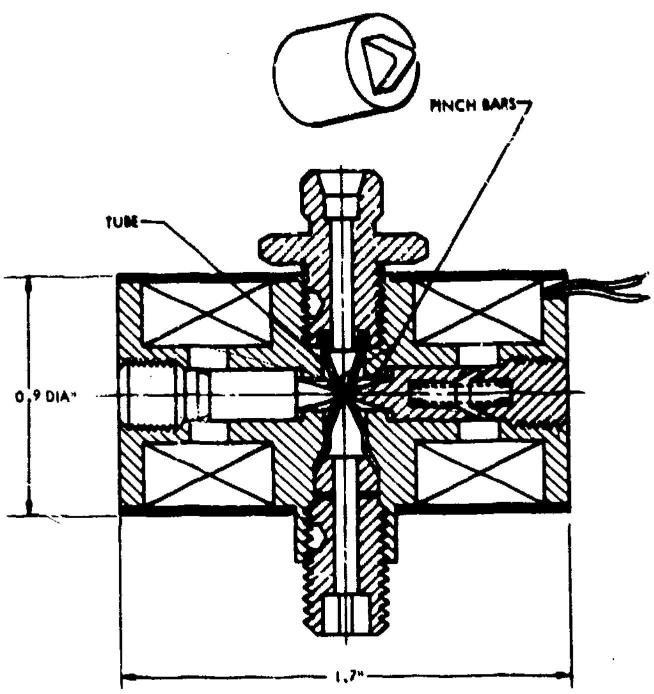


Figure 6.2.3.12a. Solenoid-Actuated Pinch Valve
(Courtesy of Eckel Valve Co., San Fernando, California)

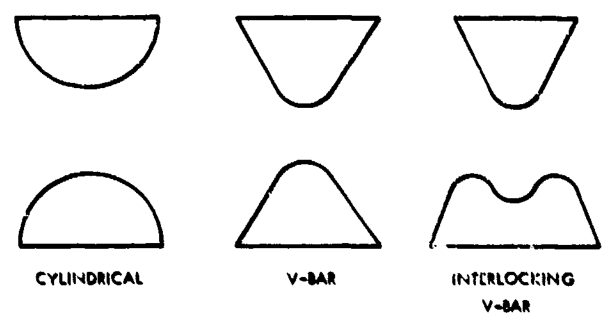


Figure 6.2.3.12b. Pinch Bar Shapes

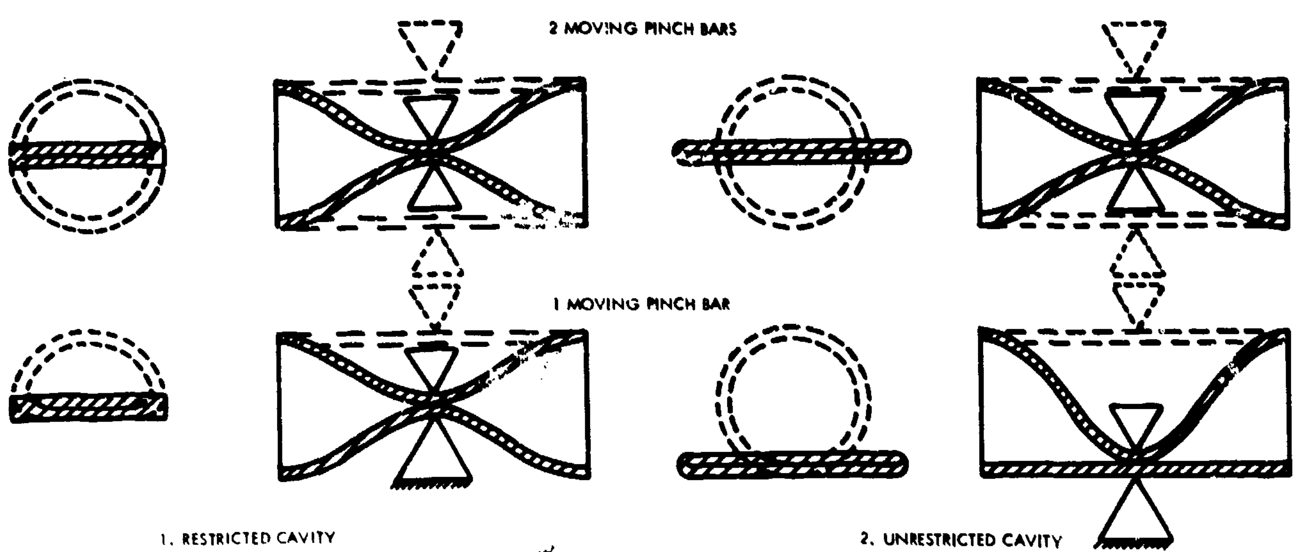


Figure 6.2.3.12c. Pinch Valve Port Area

centerline (see Figure 6.2.3.12c-1), or an unrestricted cavity must be provided (see Figure 6.2.3.12c-2).

The restricted cavity is usually simpler to fabricate, but tube wall thickness and structural characteristics must be chosen with care to avoid buckling and folding of the tube wall when the valve is closed. Thin-walled tubes are more likely to buckle than are thick-walled tubes, although the restricted cavity design shown in Figure 6.2.3.12a has performed satisfactorily with tube walls as thin as 0.007 inch. The unrestricted cavity is used with both pressure-actuated and pinch bar-actuated valves, and must be carefully designed to avoid stress concentrations or abrasion of the tube where it enters the cavity. As with most soft-seat valving units, a major design problem is retention of the tube in the valve housing. The most successful approach employed to date on the design in Figure 6.2.3.12a employs conical ferrules similar to those used with hose fittings (see Sub-Section 5.13).

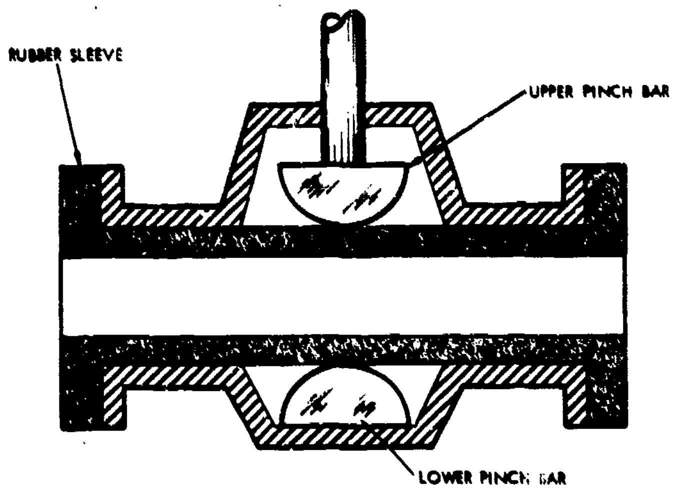


Figure 6.2.3.12d. Pinch Valve, Mechanical Actuation

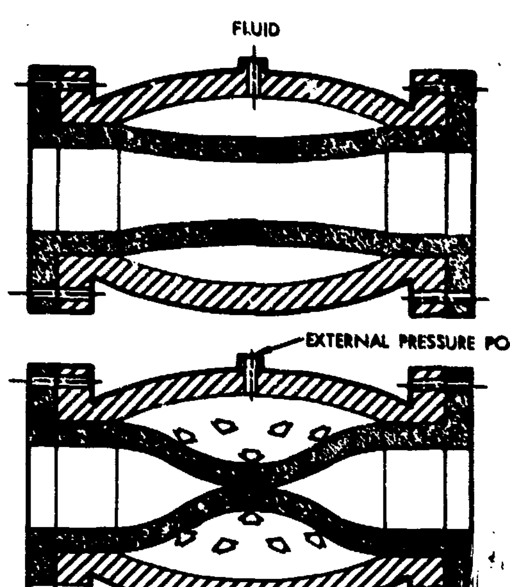


Figure 6.2.3.12e. Pinch Valve, Pressure Actuation
(Courtesy of Red Jacket Company, Inc., Carnegie, Pennsylvania)

Sealing. The force required to effect complete closure of a pinch valving unit is very difficult to treat analytically as compared with a simple diaphragm valve. [illegible] (Reference 196-1) recommends a sealing stress of 1¼ P, between the diaphragm and wier of a Saunders valve. The sealing stress in the pinched tube will not be distributed evenly across the width of the seal interface, but may be expected to be less at the outer edges where bending and compressive stresses act most strongly against the sealing force. While no quantitative data is available for verification, there are indications that the interlocking V-bar sealing method (see Figure 6.2.3.12b-3) lends itself to sealing with lower actuation force than do the simpler pinch-bar configurations.

Actuation Forces. The force required to effect closure of a pinch-bar valving unit is a function of the following:

a) Tube diameter
b) Tube wall thickness
c) Tube material
d) Pressure ratio between flowing fluid and cavity (P_i/P_c)
e) Pinch bar shape
f) Restricted or unrestricted cavity
g) Length of unsupported tube, unrestricted cavity.

Lowest actuation force requirements for a given tube diameter, tube material, and pressure ratio (P_i/P_c) may be expected from a thin-wall tube actuated by two opposing V-bar pinch bars in an unrestricted cavity of sufficient length to minimize longitudinal stress in the sides of the tube (which must stretch slightly along the longitudinal centerline when pinched off).

Flow Characteristics. Figure 6.2.3.12f shows the flow versus stroke characteristic of a typical pinch valve with two moving pinch bars. Such a pinch valve has an area versus stroke relationship approximated by a circle whose area is varied as illustrated in Figure 6.2.3.12g. The gradual transition provided by the tube shape in a throttled position results in smooth flow, as through a venturi. Pressure-actuated pinch valves lack the positive control of tube shape and therefore are used primarily in shutoff rather than throttling service.

6.2.3.13 UNCONVENTIONAL CONCEPTS. The designs included in this detailed topic represent valving unit approaches which have received recent attention for aerospace applications, but which are sufficiently unconventional to preclude their treatment in any of the preceding classifications.

All-Metal Cylindrical Seat Poppet Valving Unit. This valving unit concept (see Figure 6.2.3.15a) has been employed in liquid fluorine service and obtains its sealing force by deforming the seating surface only within its elastic limit. Initial cycling of the valve may result in some plastic deformation of the thin cylindrical seat, but on subsequent actuations the stroke of the conical or spherical poppet is limited, so that only elastic deformation occurs.

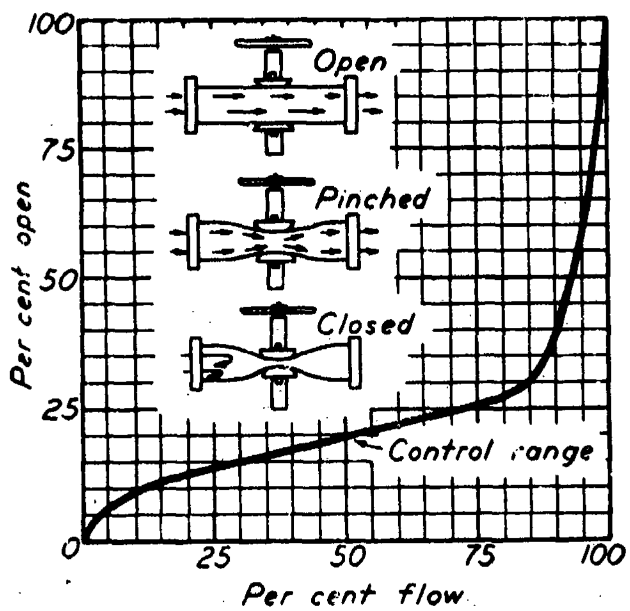


Figure 6.2.3.12f. Pinch Valve Flow Characteristics
(Adapted from "Control Valves," Beard, Copyright 1957, Instruments Publishing Company, Pittsburgh, Pennsylvania)

An alternate approach is to provide a rigid cylindrical backup ring to limit radial seat expansion. Obviously excessive closing force will result in compressive and/or tensile yielding of the seat, even with the backup ring, unless provision is made for limiting stroke. Pressure is normally applied only to the outside of the thin cylindrical seat so that increased pressure differential across the closed valving unit serves to increase sealing stress and maintain a leak-tight seal. Unlike the conventional hard-seat conical poppet valving unit, the deformation of the thin seat takes place simultaneously with a significant linear scrubbing action between valving element and seat, which helps to clean the sealing interface and effect a tight seal. This scrubbing action may be expected to yield wear particles which might interfere with the proper operation of some systems (Reference 131-6). The size of such wear particles is largely dependent upon the seat and poppet materials used, as discussed in Detailed Topic 6.2.2.12. Because of stiction, if narrow conical poppet cone angles or low poppet diameter/seat diameter ratios for spherical poppets are used, the actuation force required to break loose the poppet from the seat on opening may be expected to be significantly greater than that required to open a comparable hard-seat poppet valve. Otherwise, the actuation forces and flow characteristics of conical or spherical poppet valving units may be considered equally applicable to the cylindrical seat concept.

Cone Labyrinth Valving Unit. The scrubbing action, elastic deformation, poppet action, and all-metal construction of

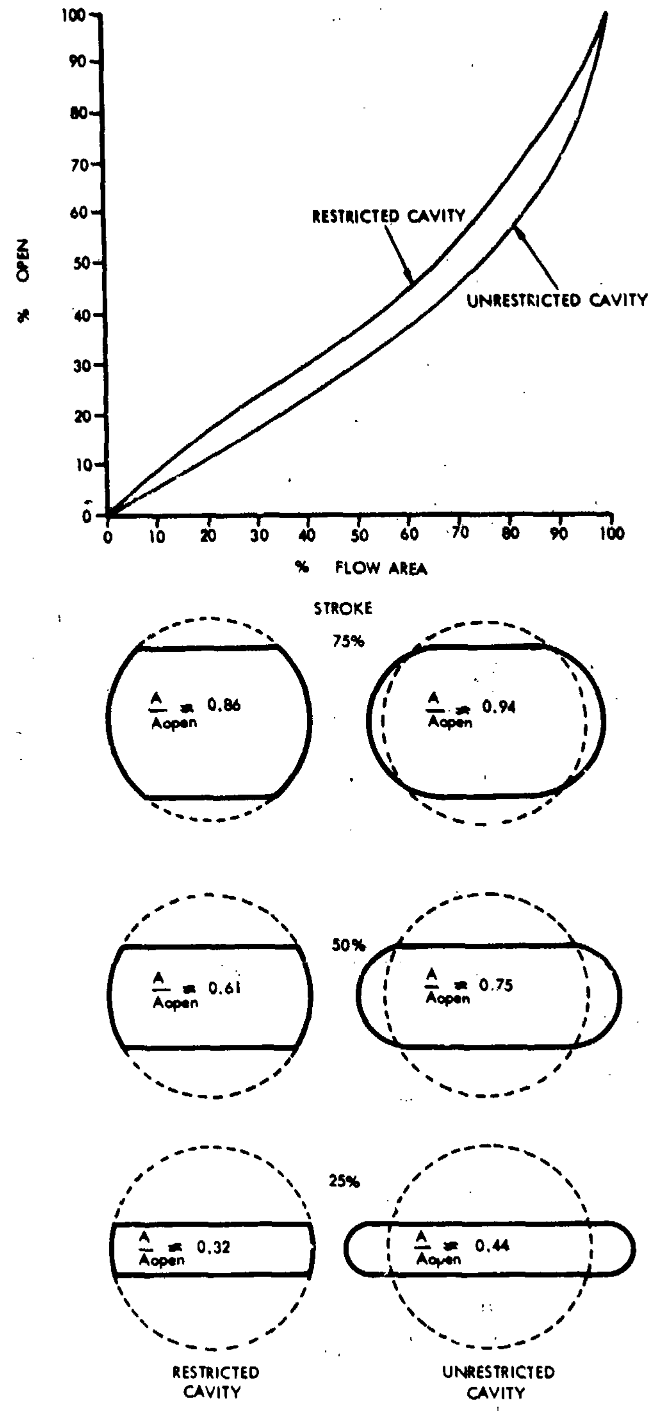


Figure 6.2.3.12g. Pinch Valve Flow Area Change with Stroke

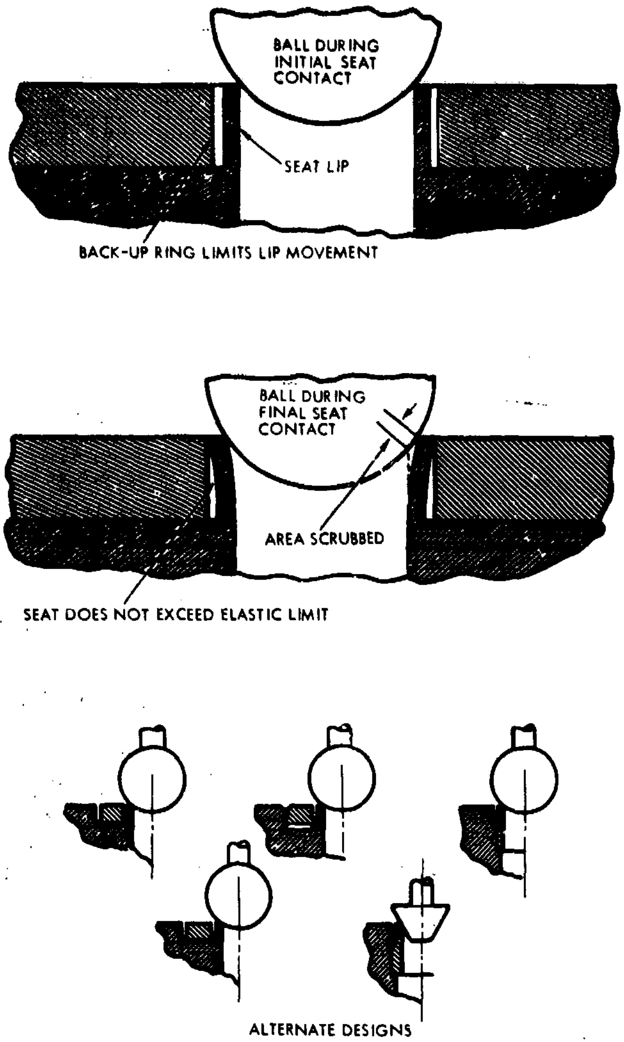


Figure 6.2.3.13a. All Metal Cylindrical-Seat Poppet Valving Unit
(Adapted from Reference 36-13)

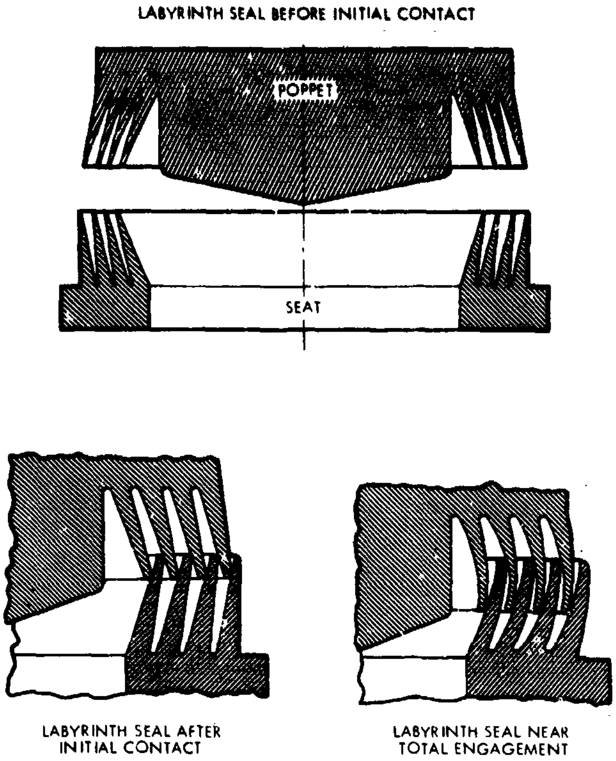


Figure 6.2.3.13b. Cone Labyrinth Valving Unit
(Courtesy Smirra Developments Company, Los Angeles, California)

the cylindrical-seat valving unit are further exploited in this unique design (see Figure 6.2.3.13b). The multiple lands of the cone labyrinth valve provide the additional advantages of sealing redundancy and sensitive flow control in the near-closed position. As in the case of the cylindrical-seat valving unit, breakaway force requirements are a function of the specific design and may be either very high or quite low. In addition to the material properties of the seat and valving element, actuation forces and flow characteristics are both dependent upon the relatively complex configuration, whose definition is a function of:

a) Number of poppet lands
b) Number of seat lands
c) Diameters of lands
d) Height of lands
e) Internal and external cone angles of seat lands
f) Internal and external cone angles of poppet lands
g) Tip radius or tip thickness of lands
h) Position and land thickness at point of initial seating contact
i) Length of stroke after initial contact (scrubbing action).

The labyrinth seal action afforded by the multiple lands in the near-closed position provides a flow characteristic such as that illustrated in Figure 6.2.3.13c. As with the conventional labyrinth seal, the distribution of the total pressure drop over a number of flow-restricting annular orifices (or approximations thereof) extends the range of laminar flow and minimizes erosive cavitation.

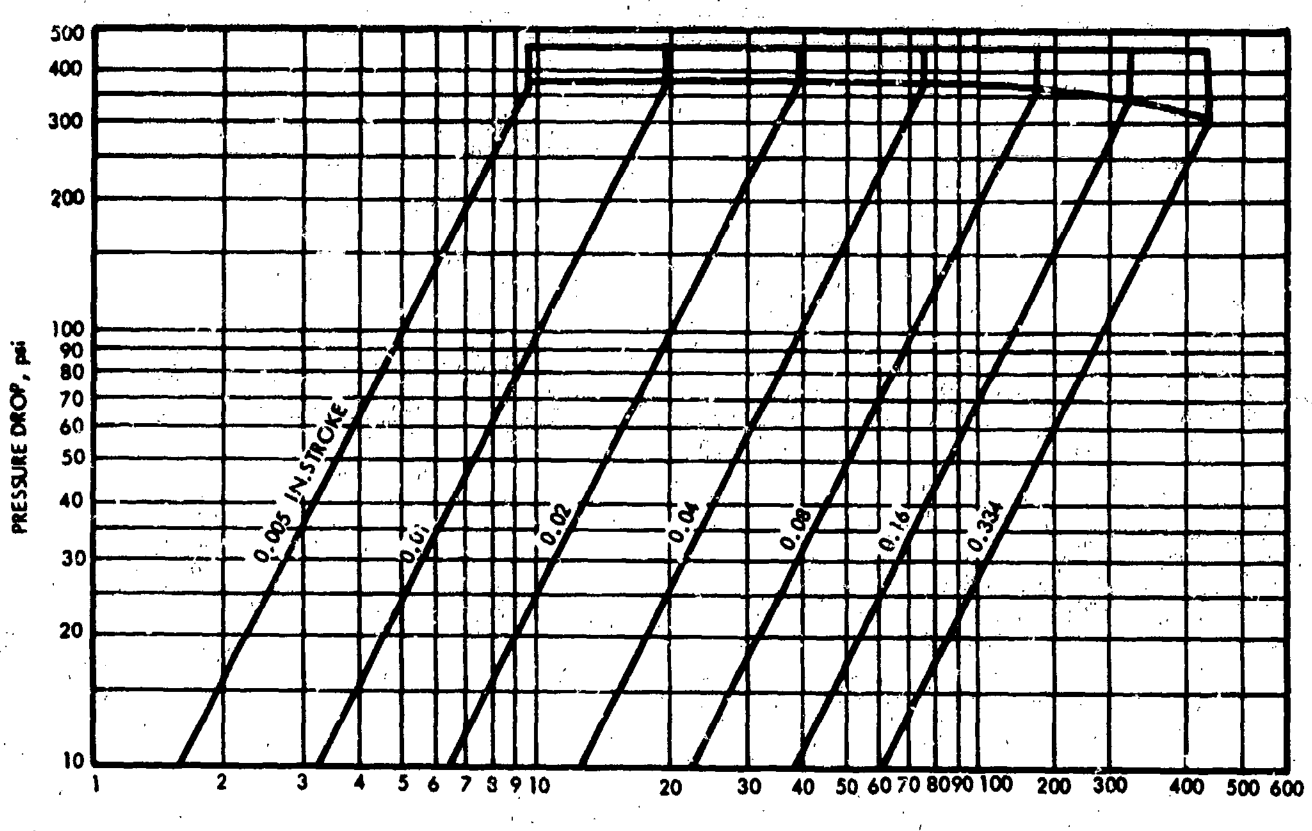


Figure 6.2.3.13c. Cone Labyrinth Valve Flow Characteristics
(Courtesy Smirra Developments Company, Los Angeles, California)

The cone labyrinth valving unit may be designed to seat with the lands "bottomed out" or at any point after initial contact. Among the considerations in this element of the design are the degree of scrubbing action desired and the degree of fluid contamination anticipated. The scrubbing action makes the design very insensitive to contamination (impurities tend to be scraped away rather than crushed or imbedded), but "bottoming out" of the lands may be undesirable in contaminated-fluid applications.

Conical Disc Poppet With Spherical Seat. Figure 6.2.3.13d illustrates a unique design which provides for seating of the poppet despite misalignment of the poppet stem. The thin conical wall of the poppet disc diminishes in thickness uniformly towards its periphery, thereby maintaining a uniform level of stress. The conical wall is always perpendicular to the tangent of the spherical seat at the point of contact, whether the poppet stem is in axial alignment or not. Deformation is kept within the elastic limits of both poppet and seat.

Curtain-Flap Valving Unit. Figure 6.2.3.13e illustrates a unique design approach for providing proportional flow control. One end of the Teflon curtain is secured to the inside of the stationary hollow cylinder and the other is secured to the rotatable valve stem. As the valve stem

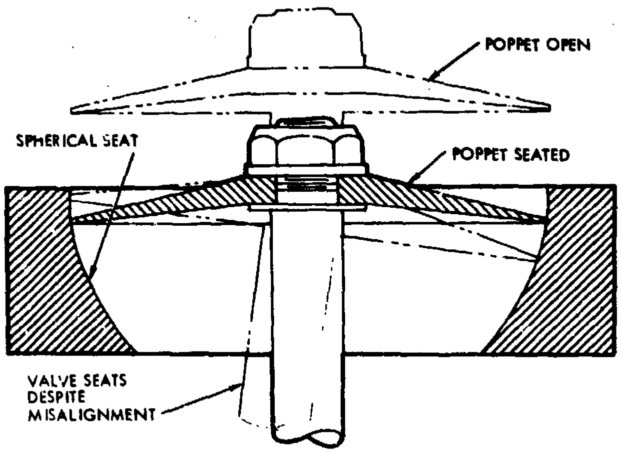


Figure 6.2.3.13d. Thin Conical Poppet and Spherical Seat
(Reference 36-13)

rotates and winds up the Teflon curtain, small ports in the hollow cylinder are exposed, and fluid which has entered the cylinder axially is permitted to flow out radially. Since these ports must be small enough so that the Teflon will not extrude through the holes, numerous small holes are used. The ports are arranged as shown in Figure 6.2.3.13e, with the angle θ and separation distance x chosen to provide proportional flow control. If structural conditions permit, narrow circumferential slots may be considered as an alternative to the multiple circular discharge ports. Similarly, any specific flow versus stroke characteristic may be obtained by tailoring the axial and circumferential spacing of the circular ports.

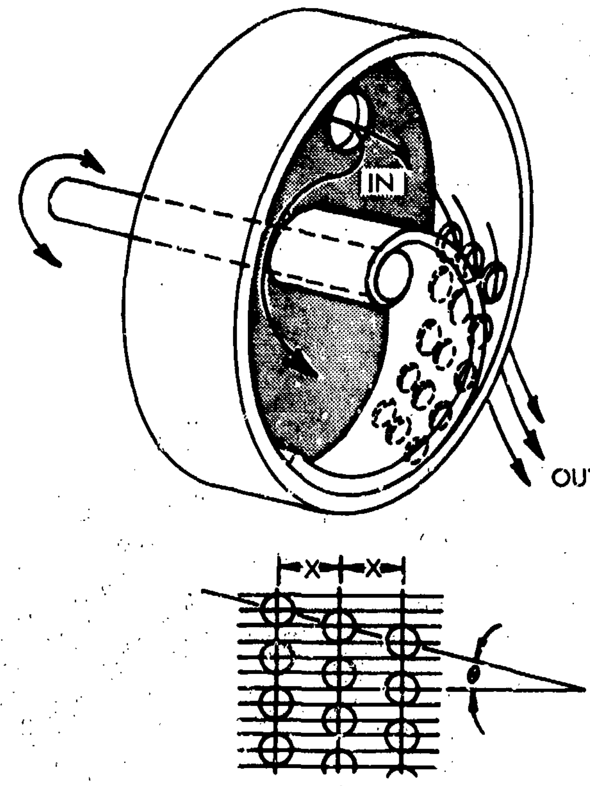


Figure 6.2.3.13e. Curtain Flap Proportional Valving Unit
(Reference 36-13)

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Butterfly Valving Units:

Configuration

23-38	144-1
26-86	160-18
34-10	167-1
35-12	201-1
71-3	497-1

Sealing

68-16	195-7
71-3	V-135
112-1	

Actuation Forces

1-42	193-24
26-86	195-7
68-4	201-1
144-1	408-1
160-18	497-1
165-3	

Flow Characteristics

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19-229	167-1
23-38	193-24
26-86	195-7
34-10	201-1
68-4	205-4
144-1	273-2
160-18	408-1
165-2	497-1

Miscellaneous

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131-6

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Ball Valving Units:

Configuration
19-231 73-100
34-10 74-22
35-12 454-3
36-13

Sealing
19-231 73-100
35-12 74-22
36-13

Actuation Forces
34-10 131-6
73-100

Miscellaneous
131-6
454-3

Rotary Plug Valving Units
19-235 160-18
112-8 196-1

Gate Valving Units
19-229 68-64
23-38 112-8
36-13 196-1

Rotary Slide Valving Units
19-229 144-1
36-13

Sleeve Valving Units
19-229 34-11
34-10

Spool Valving Units:

Configuration
6-36 42-1
6-159 45-1
23-19

Actuation Forces
26-29 68-6
42-1 73-60
45-1 169-9

Flow Characteristics
6-36 42-1
19-229 45-1
26-29 73-60

Piston Valving Units
36-13 45-1
169-9 144-1
42-1

Poppet Valving Units:

Configuration
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26-48 144-1
35-12 196-1
36-13 256-1
71-1 35-14*

Sealing
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35-12, 14* 498-1
71-1 46-32*
131-6 152-4*

Actuation Forces
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131-26 256-1
144-1 354-1
165-7 498-1
194-7 35-14*

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194-6

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144-1

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Contamination Sensitivity
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131-6 35-14*

Valve Life
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131-6

Environmental Conditions
36-13
131-6

Valve Rangeability
165-36*

ISSUED: MARCH 1967
SUPERSEDES: OCTOBER 1965

6.3 STATIC SEALS

6.3.1 Introduction

Static seals are elements common to most fluid components. Their importance in fluid component design cannot be overemphasized, as seal leakage constitutes a major cause of component malfunctions. Elimination of static seal leakage would result in a significant overall increase in fluid component reliability. The purpose of this sub-section is to discuss the various factors which must be considered in the design and/or selection of static seals, and to present design and application data for various specific static seal types used in aerospace fluid component applications.

A *static seal* may be defined as a device used to prevent leakage of a fluid through a mechanical joint in which there is no relative motion of the mating surfaces other than that induced by changes in the operating environment. The *seal gland* is the adjacent structure which either wholly or partially confines a seal.

Although sealants and hermetic seals (welded and brazed joints) can be broadly classed as static seals, this sub-section is limited to seals which can be considered as mechanical devices. Welded and brazed joints are treated respectively in Sub-Topics 5.12.6 and 5.12.5.

6.3.1.1 SEAL CLASSIFICATION. Static seals can be classified in a variety of ways; four commonly used classifications are given in Table 6.3.1.1.

Table 6.3.1.1. Seal Classifications

Method of Loading
- Pressure-energized
- Temperature-energized
- Mechanically-preloaded

Material
- Metallic
- Nonmetallic
- Combination

Application
- Cryogenic
- Vacuum
- High temperature
- High pressure

Configuration
- Flat gaskets
- O-Rings
- Flexible shapes
- Solid shapes

6.3.1.2 SEALING MECHANISM. The degree to which seal leakage is eliminated depends on the capability for achieving continuous mating of the sealing interfaces. Seal and gland mating can be accomplished (1) by making one surface relatively soft and compliant so that it will readily conform to the harder surface irregularities, (2) by providing sufficient sealing load to plastically deform the softer of the two sealing surfaces, or (3) by carefully finishing both surfaces so that good mating can be achieved without high sealing loads. The stress at the sealing interface required to achieve good mating depends both on the physical properties and the surface finishes of the mating materials. The materials utilized normally define a minimum seating stress level required to achieve satisfactory mating. Resilience is another important characteristic of a static seal, since it is the resilience of the seal which insures that adequate sealing stress is maintained, in spite of relative movement of the joint surfaces caused by changes in the operating environment.

Understanding of how a static seal works has been limited, and only recently have basic studies in sealing technology been conducted (References 46-17, 46-27, 152-2, and 152-4).

6.3.1.3 DEFINITIONS. The following definitions will apply to sealing terminology used throughout this sub-section.

Seal Load. The seal load is the total force applied normal to the seal interface expressed in pounds.

Unit Seal Load. The unit seal load is the load distributed over a given seal length. This term is usually limited to use in circular seal shapes and may be determined by dividing the seal load by the nominal length of the seal interface. The units are commonly expressed in pounds-per-inch of seal length.

Apparent Seal Stress. The apparent seal stress is determined by dividing the seal load by the total area of the seal interface. The units are designated in pounds-per-square-inch. Most seal stress values are given as the apparent seal stress.

Actual Seal Stress. Due to irregularities in the seal interface surfaces, the actual area of contact will be less than the total interface area. The actual seal stress is determined by dividing the seal load by the actual contact area and is expressed in pounds-per-square-inch. Since the actual area of contact is difficult to determine, the actual seal stress is normally limited to use in theoretical analysis.

Seating Stress. Seating stress is the applied apparent seal stress required to establish mating of the sealing surfaces.

Zero Leakage. Zero leakage must be defined in terms consistent with the sensitivity of the technique used for measuring leakage. Where leakage is measured by a helium mass spectrometer, a definition of zero leakage which is gaining acceptance for aerospace applications is 1×10^{-6} standard cubic centimeters (scc) per second (i.e., 0.08 scc/day or 2.6 scc/month).

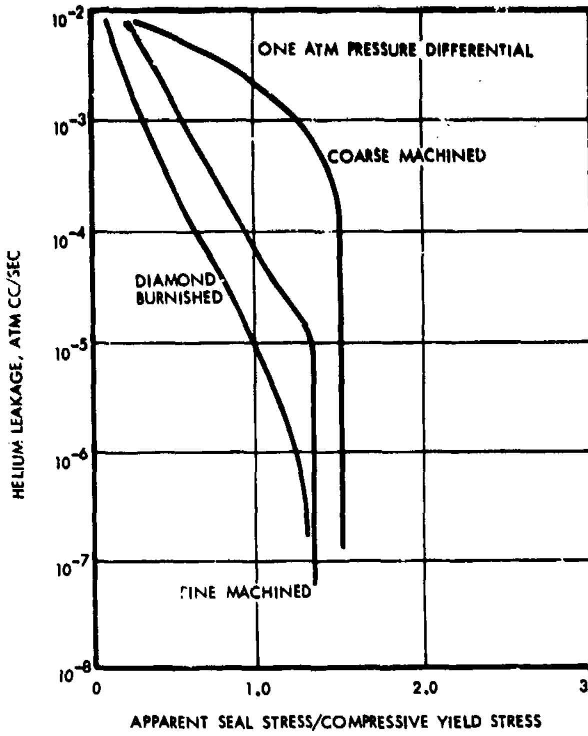


Figure 6.3.2.1a. Typical Leakage Rate Curves for Metal-to-Metal Sealing
(Adapted with permission from a paper by F. O. Rathbun presented at the SAE Aerospace Fluid Power Systems and Equipment Conference, May 1965)

6.3.2 Design and Selection Factors

The designer and user of static seals must be aware of the various interrelated factors which influence seal performance. Basic considerations such as material properties, sealing interface conditions, and seal load; use considerations such as reusability; and environmental factors such as pressure, temperature, and fluid media are all interrelated parameters which must properly be accounted for if satisfactory sealing is to be achieved. In this sub-topic these various factors are discussed with the intent of providing the fluid component designer a basis for proper static seal selection.

6.3.2.1 SEALING INTERFACE. Two major factors which influence the extent of mating at the seal interface are (1) the intital topography of the mating surfaces, and (2) the interface deformation resulting from seal load. In general, the better the finish of the mating surfaces, the lower the stress required for achieving a seal. Figure 6.3.2.1a presents leak test data from Reference 23-56, illustrating the effects of both surface finish and seal stress on helium leakage through metal-to-metal seals. Stress is plotted as the ratio of the apparent seal stress to the compressive yield stress of the seal material (the compressive yield stress was measured on the actual seal during load in the test rig). By normalizing the seal stress with respect to the material yield stress, several different metals used as seals were found to fall on the same leakage curve when finished in the same manner (e.g., fine machined or coarse machined). In addition to the type of metal-to-metal sealing illustrated in Figure 6.3.2.1a, wherein effective sealing is obtained after compressive yielding has occurred, the following discussion of surface deformation also treats static seals which utilize only elastic deformation. The latter type of seal includes both those which incorporate an elastomeric seal material and those which utilize superfinished metal surfaces. Seal hysteresis, which is a measure of the degree to which a sealing interface can be unloaded after initial seating without affecting leakage, is discussed for sealing in both the plastic and elastic deformation ranges.

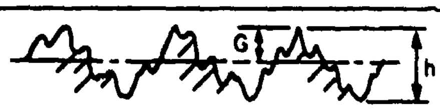
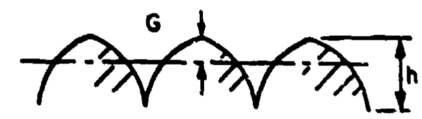
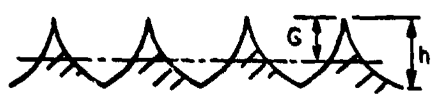
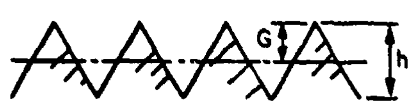
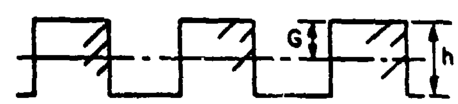
Surface Topography. Surface topography determines the degree of interface mating between a seal and gland prior to loading. Material properties as well as processing techniques determine surface characteristics. The surfaces of parts molded from elastomers and plastics are determined to a large extent by the characteristics of the material during the molding and cooling processes. Surface irregularities in the mold cavity will of course be repeated in the molded part. Machining operations used on metals and some plastic parts generate surface irregularities resulting from the tearing or cutting action of cutting tools or abrasive grains, coupled with gross irregularities associated with inaccuracies in the machine tools used (i.e., chatter, tool wear, deflection, etc.).

Regardless of how the surface characteristics are evolved, the profiles in all instances consist of a series of peaks and valleys which deviate in a more or less irregular fashion above and below a mean surface. Although a majority of surfaces appear to have random topography and consist of many irregular protrusions, there are characteristics that may be used to describe the surface. These characteristics are generally broadly classified as either surface roughness or waviness. Roughness takes into account the finer irregularities caused by the cutting tool and the machine tool feed, while waviness is the wider irregularity resulting from machine or work deflection, vibration, heat treatment, etc. To distinguish between roughness and waviness, a roughness width cutoff must be established in order to specify the maximum width in inches of surface irregularity to be included in the measurement of roughness height. The roughness width cutoff must always be greater than the actual roughness width of the predominant pattern of surface roughness. The roughness characteristic of machined parts will consist of two components: (a) the basic pattern which is formed by successive grooves, or scratch marks, in manufacturing the part, and (b) the irregularities within each groove, which are formed by the rupturing of the material when the chip is torn from the part in machining. Roughness height is measured as the arithmetical average (AA) value, also called the centerline average (CLA) expressed in microinches. Since 1955, the AA system has largely replaced the previously used root mean square (RMS) system. Table 6.3.2.1a compares AA and RMS measurements for several mathematically described surface geometries.

Figure 6.3.2.1b depicts the relationship of the symbols commonly utilized to describe the limits of surface characteristics as described in ASA B-46.1 (Reference 68-71). The lay noted in Figure 6.3.2.1b is the direction of the predominant surface pattern produced by tool marks, and is dependent on the production methods used. Lay is defined by the symbols, as specified in Table 6.3.2.1b. The direction of lay will have a significant effect on the leakage rate if the dominant direction is across the seal face.

Table 6.3.2.1c shows the typical applications of surface roughness values which may be obtained by various manufacturing methods. As a general rule, elastomer seals will require a surface finish on the gland interface equal to 32 microinches, plastics 16 to 32 microinches, and metals 4 to 16 microinches, the latter depending primarily upon lay (see Detailed Topic 6.2.3.11).

Table 6.3.2.1a. Average Height Value for Various Wave Forms

WAVE FORM: h = 1		AA	rms	$\frac{h}{AA}$	$\frac{h}{rms}$	$\frac{G}{h}$	$\frac{G}{AA}$	$\frac{rms}{AA}$
UNIFORMLY RANDOM		0.2	0.25	5.0	4.0	0.5	2.5	1.25
ROUND CRESTED PARABOLIC		0.256	0.298	3.91	3.36	0.333	1.29	1.16
SHARP CRESTED PARABOLIC		0.256	0.298	3.91	3.36	0.667	2.60	1.16
SINUSOIDAL		0.318	0.353	3.14	2.83	0.5	1.57	1.11
SAW TOOTH		0.25	0.289	4.0	3.46	0.5	2.0	1.16
SQUARE		0.5	0.5	2.0	2.0	0.5	1.0	1.0

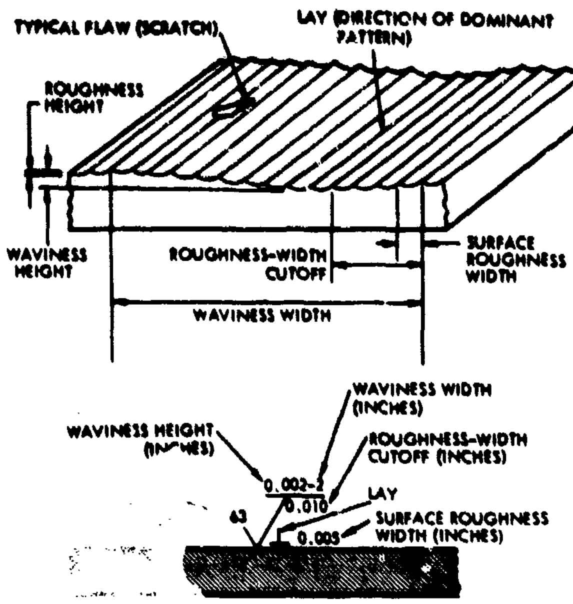


Figure 6.3.2.1b. Relation of Symbols to Surface Characteristics

([illegible] Texture," ASA B46.1-1962, with the permission [illegible] erican Society of Mechanical Engineers, United [illegible] East 47th Street, New York 17, New York)

In addition [illegible] .age surface characteristics, the allowable number and magnitude of flaws present in a surface must be considered. Flaws are such defects as cracks, blow holes, checks, ridges, scratches, etc., which appear at one place only, or at widely varying intervals in a surface. The effects of flaws are not normally included in roughness height measurement, and must be separately defined and limited.

Interface Deformation and Hysteresis. The fact that surface deformation is a necessary factor for low cost static seal designs is illustrated in Figure 6.3.2.1c, which shows the AA surface finishes that can be achieved by various manufacturing processes as compared to the effective height of a leak passage required to reduce helium leakage to 10^{-4} scc/sec. The nature and degree of deformation at the seal interface depends upon the seal and gland materials involved. In the case of soft polymeric seal materials, a relatively high degree of seal deformation will occur at the interfaces in the elastic strain regime. In the case of hard materials, the deformation is more complex, since it involves a significant amount of plastic, as well as elastic strain of surface asperities. The degree to which a seal can be unloaded after initial seating and still continue to seal is dependent upon the relative amount of plastic deformation. The more plastic deformation obtained, the less sensitive the system to stress removal.

Leakage flow past measured hard metal surfaces has been correlated with analytical equations and is reported in Reference 35-12. (See also Detailed Topic 6.2.3.11u.)

The following paragraphs discuss the influence of the material property in sealing.

Sealing with Elastomeric Materials. Elastomers deform to mate with rigid surfaces by large elastic deformations. The apparent seal stress required to achieve surface mating is extremely small. Often apparent stresses as low as 500 psi are adequate to effect a seal. Since the deformation of the seal is almost entirely elastic, the initially-applied seating load must be maintained. Thus, an additional load margin must be applied to allow for strain relaxation during the life of the seal.

Sealing with Soft Plastic Materials. Like elastomers, soft plastic materials are readily deformed against hard metal surfaces, thereby minimizing the importance of good surface finishes for static sealing. The required loads applied to the plastics are generally such that the plastic will conform to the surface of the stronger material at the microscopic level. The deformation in plastics is of viscoelastic nature; thus, without proper constraint, the phenomenon of cold flow can exist, resulting in an unsatisfactory design due to sealing stress relaxation. Prevention of bulk flow of the plastic material requires adequate containment of the seal by the gland. In general, minimum apparent seating stress levels of approximately 0.7 times the yield strength of the plastic used will be sufficient to effect a seal.

Metal-to-Metal Compression Sealing. (Reference 46-19). Plastic deformation plays an important role in metal-to-metal sealing. When two soft metal surfaces are brought together and compressed under a normal load, five separate regimes of material flow are experienced. (For hard metals see Detailed Topic 6.2.3.11.)

Regime I. Initial contact between two surfaces having some asperity distribution results in plastic deformation at some high asperities under low apparent seal stress. There is little overall increase in actual contact area and little decrease in leakage in this regime.

Regime II. As the load increases, the asperities of average height come into contact and deform plastically. A rapid increase in actual contact area with nominal increases in normal stress is experienced. Leakage in this regime decreases with stress more rapidly than in Regime I. At the end of Regime II, the apparent seal stress is near the yield strength of the weaker material.

Regime III. Plastic flow of the asperities continues and "pile-up" of material between asperities begins. Strain hardening and bulk flow of the softer material begins in this regime.

Regime IV. Bulk flow of the seal material increases the actual contact area by shearing along the surface and by physically increasing the apparent area of contact. The amount of bulk flow is dictated to a large degree by the actual seal stress and the strain hardenability of the seal material.

Regime V. The normal stress begins to cause bulk flow of the harder material. This regime should be avoided if it is

Table 6.3.2.1b. Lay Identification Symbols

(Extracted from "Surface Texture," ASA B46.1-1962, with the permission of the publisher, The American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York 17, New York)

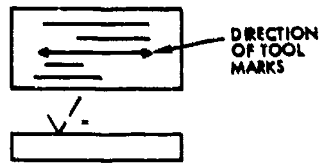
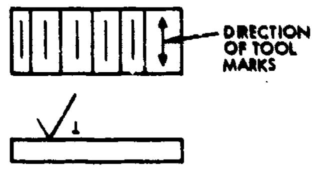
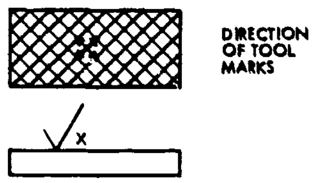
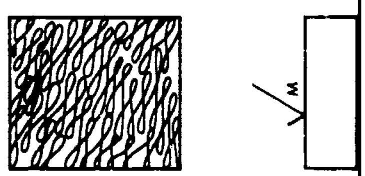
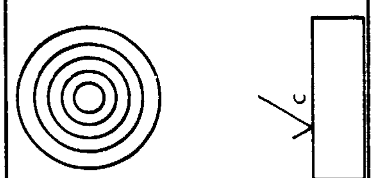
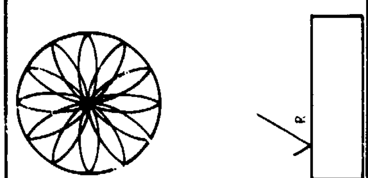
Lay Symbols		
Lay Symbol	Designation	Example
=	Lay parallel	
⊥	Lay perpendicular	
X	Lay angular in both directions	
M	Lay multidirectional	
C	Lay approximately circular	
R	Lay approximately radial	

Table 6.3.2.1c. General Application of Surface Roughness
(From MIL-STD-10A)

ROUGHNESS HEIGHT RATING	GENERAL APPLICATION OF ROUGHNESS HEIGHT RATINGS
1,000 √	Very rough, low grade surface, resulting from sand casting, torch or saw cutting, chipping, or rough forgings. Machine operations are not required as appearance is not objectionable. This finish, rarely specified, is suitable for unmachined clearance areas on machinery, jigs, and other rough construction items.
500 √	Very rough, low grade surfaces, where smoothness is of no object, resulting from heavy cuts and coarse feeds in milling, turning, shaping, boring, and from very rough filing, rough disc grinding, and snagging. This surface is suitable for clearance areas on machinery, jigs, and fixtures. This surface roughness may be obtained by natural processes of sand casting or rough forging.
250 √	Coarse production surfaces, for unimportant clearance and cleanup operations, resulting from very coarse surface grind, rough file, disc grind, and from rapid feeds in turning, milling, shaping, drilling, boring, grinding, etc., where definite tool marks are not objectionable. This roughness may also be produced on the natural surfaces of forgings, permanent mold castings, extrusions, and rolled surfaces. Surfaces with this roughness value can be produced very economically and are used to a great extent on parts where stress requirements, appearance, conditions of operations, and design permit.
125 √	This is the roughest surface recommended for parts subject to loads, vibration, and high stress. This surface roughness is also permitted for bearing surfaces when the motion is slow and the loads are light or infrequent, but are not to be specified for fast rotating shafts, axles, and parts subject to severe vibration or extreme tension. This surface is a medium, commercial machine finish in which relatively high speeds and fine feeds are used in taking light cuts with well-sharpened tools, and may be economically produced on lathes, milling machines, shapers, grinders, etc. The surface finish may also be obtained on permanent mold castings, die castings, extrusions, and rolled surfaces.
63 √	A good machine finish, produced under controlled production procedures using relatively high speeds and fine feeds in taking light cuts with well-sharpened cutters. This surface value may be specified where close fits are required and may be used for all stressed parts, except for fast rotating shafts, axles, and parts subject to severe vibration or extreme tension. This surface roughness is satisfactory for bearing surfaces when the motion is slow and the loads are light or infrequent. This surface roughness may also be obtained on extrusions, rolled surfaces, die castings, and permanent mold castings when rigidly controlled.
32 √	A high-grade machine finish, requiring close control when produced by lathes, shapers, milling machines, etc., but relatively easy to produce by centerless, cylindrical, or surface grinders. This surface may be specified in parts where stress concentration is present. This finish is satisfactory for bearing surfaces when motion is not continuous and loads are light. When finer finishes than this are specified, production costs rise rapidly; therefore, such finishes must be analyzed carefully by the engineer or designer. Also, processes such as extruding, rolling, or die casting may produce a comparable surface roughness when such processes are rigidly controlled.
16 √	A high quality surface, produced by fine cylindrical grinding, emery buffing, coarse honing, or lapping. A surface of this value is specified where

Table 6.3.2.1c. General Application of Surface Roughness (Continued)

ROUGHNESS HEIGHT RATING	GENERAL APPLICATION OF ROUGHNESS HEIGHT RATINGS
	smoothness is of primary importance for proper functioning of the part, such as rapidly rotating shaft bearing, heavily loaded bearings, and extreme tension members.
8√	Very fine surfaces produced by special finishing operations such as honing, lapping, or buffing. Surfaces refined to this degree are specified where packings and rings must slide across the direction of the surface grain, maintaining or withstanding pressures; the interior honed surfaces of hydraulic cylinders are an example. Finishes of this value may also be required in precision gages and instrument work, on sensitive valve surfaces or on rapidly rotating shafts, and on bearings where lubrication is not dependable.
4√	Refined surfaces produced by special finishing operations such as honing, lapping, and buffing. This surface roughness value should be specified only when the requirements of design makes it mandatory as the cost of manufacturing is extremely high. Surfaces refined to this degree are required in instrument work, gage work, and where packings and rings must slide across the direction of surface grain, such as on chrome plated piston rods, etc. where lubrication is not dependable.
2√ 1√ ½√ ¼√	Very refined surfaces produced only by the finest of modern honing, lapping, buffing, and superfinishing equipment. These surfaces may have a satin or highly polished appearance, depending on the finishing operation and material. Finishes of this type are only specified when design requirements make it mandatory, since the cost of manufacturing is extremely high. Surfaces refined to this degree are specified on fine or very sensitive instrument parts or other laboratory items, and certain gage surfaces, such as precision gage blocks.

intended that the joint be opened and then resealed. The original asperities on the gland do not suffer great deformation prior to and even during this regime, due to the containment of the asperities by the seal material mated with them.

Although dependent on surface finish, mating of metal-to-metal surfaces generally requires a seating stress of two to three times the yield strength of the softer material.

Metal-to-Metal Sealing by Shear Deformation. The yielding necessary for the production of seal between two material surfaces is not caused by a high compressive stress alone, but by a combination of stresses existing in the material to be deformed. Stress can be applied in a manner that causes the soft material to strain in shear, thereby requiring less load to achieve a plastic stress condition at the sealing interface. Knife edges, shear O-rings, and other geometries which cause high stress concentration factors can be used to accomplish this.

Superfinish Metal-to-Metal Sealing. Recent experimental work (Reference 46-32) has shown that elastic sealing with metals is possible by mating superfinished surfaces. The two most significant characteristics of this type of metal-to-metal static seal are the absence of plastic deformation and the corresponding use of low apparent seal stresses. In addition to the high cost of producing superfinished surfaces and the danger of surface damage during handling and assembly, the greatest problem associated with the use of such seals is the difficulty of inspecting the surface. Recent studies (References 35-12, 46-32, and 152-4) have added to the technology of topography identification however, and development work is continuing on superfinished metal-to-metal static seals.

Hysteresis. Seal hysteresis is related to the relative effects of permanent interface deformation compared with elastic deformation. A seal that undergoes significant plastic deformation will exhibit relatively high hysteresis, i.e., after achieving the initial seating stress required to reduce leakage to an acceptable level, the seal load can be significantly relaxed without influencing leakage. The curves in Figure 6.3.2.1d illustrate differences in hysteresis between sealing elastically (A) and plastically (B).

6.3.2.2 MATERIALS. Any solid material can potentially be used as a seal; however, the ideal seal material should have the following properties:

a) Capability of storing elastic energy to allow the seal to maintain interface loads under conditions of gland deformation (resiliency or low elastic modulus)
b) Capability of containing high differential pressures without deformation (high yield strength)
c) Capability of deforming and conforming to minute surface asperities (soft)
d) Resistance to damage during handling or operation such as cuts, nicks, and scratches (hard)

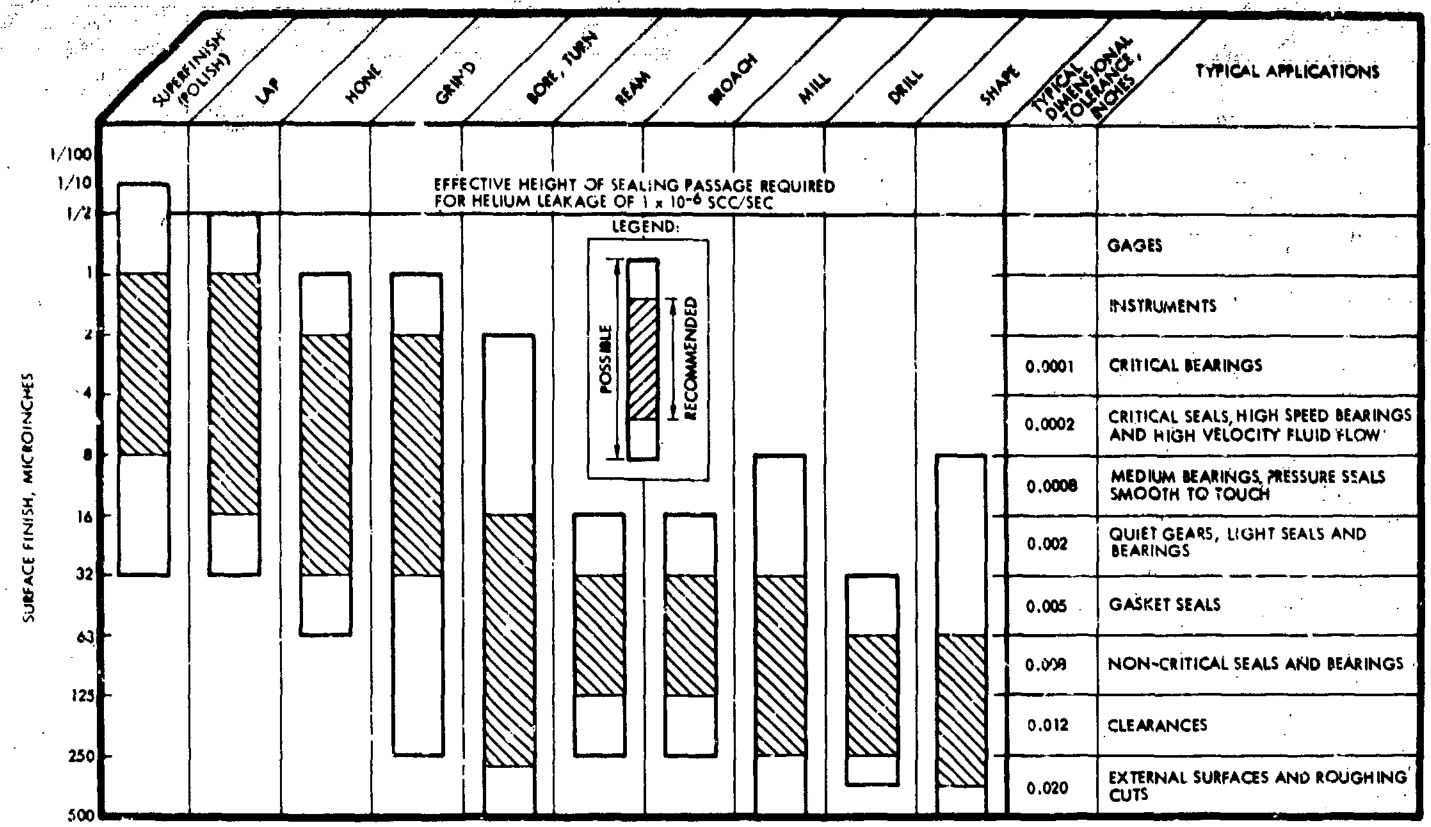


Figure 6.3.2.1c. Typical Surface Finishes Produced by Various Manufacturing Techniques
(References, 46-32, V-356)

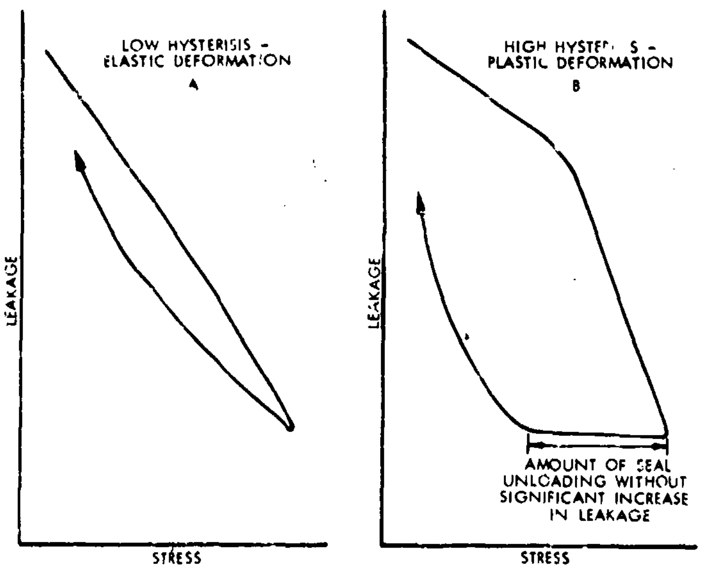


Figure 6.3.2.1d. Stress-Leakage Curves Showing Seal Hysteresis
(Adapted with permission from a paper by F. O. Rathbun presented at the SAE Aerospace Fluid Power Systems and Equipment Conference, May 1965)

e) Resistance to creep or stress relaxation as a result of applied load or elevated temperatures (stable and inert)
f) Simplicity of fabrication (low cost)
g) Capacity for unlimited reuse
h) Chemical compatibility with the sealed media
i) Resistance to fluid flow through the seal body (low porosity or low permeability).

Obviously no single material will meet all the above requirements. Selection must therefore be based on the best combination of properties for any given application. Combination of various materials is often utilized to obtain desirable properties not found in any one material, for example, the reinforcing of Teflon to reduce cold flow, and plating of metals to combine strength with a soft interface.

Nonmetallic Materials. When temperature, strength, and compatibility considerations permit, nonmetallic materials, particularly elastomers, are preferred to metals. Elastomeric materials offer good resiliency, excellent conforming properties (which minimize the need for a smooth gland finish), are reusable, and are low in cost.

Nonmetallic materials used for seals include both plastics and elastomers. The most commonly used plastics are Kel-F, nylon, Mylar, and Teflon. Elastomers include Buna-N, butyl, polysulfide, silicone, and fluorocarbon rubbers. Reinforcing materials used for adding to the strength of elastomers include nylon, rayon, glass fiber, and asbestos. Properties of elastomers and plastic materials are given respectively in Tables 12.3.1 and 12.3.2.

Temperature-limiting factors which must be considered in the application of nonmetallic seal materials are loss of ductility at extreme low temperatures (see Table 13.5.2.1) and loss of strength or decomposition at high temperatures (see Table 13.5.2e). In general, elastomers are limited to a maximum steady-state service temperature of 300 to 450°F, although for short duration an exposure to considerably higher temperatures can be tolerated.

Nonmetallics are more susceptible than metals to permanent set, creep, and stress relaxation, all being properties of seal materials which will affect the capability of the seal to maintain contact with the housing. Certain plastics, including Teflon, are particularly subject to creep and cold flow under load. Seal glands must be designed to confine the seal to minimize this permanent deformation. Permanent set becomes a problem when a seal is subjected to temperature cycling. Where permanent deformation has occurred at elevated temperatures, thermal contraction caused by cooling to a lower temperature may result in separation of the seal interfaces.

Space environments such as vacuum and radiation can have detrimental effects on nonmetallic materials, as discussed in Sub-Section 13.6. The relative decomposition of elastomers by evaporation of the volatile components, particularly plasticizers, in a vacuum atmosphere, is shown in Table 13.6.2.2; the radiation tolerance levels for nonmetallics are shown in Table 13.6.3.7b.

The permeation rate of contained media through nonmetallic materials is an important consideration for low leakage sealing applications, since the permeability of the material may be considered a built-in leak. Permeation rates for typical gases through nonmetallic materials are given in Table 12.6b. Caution must be exercised when using these data, because permeation rates will change widely with variations of differential pressure across the seal material and with changes in temperature.

Chemical compatibility must be given careful consideration when selecting nonmetallics for propellant seals. Fluorocarbon plastics such as Teflon and Kel-F exhibit the best chemical inertness, but are often inadequate when subjected to propellants. General materials compatibility data are presented in Sub-Section 12.5, but it is recommended that maximum use be made of the material referenced in the bibliography in order to determine compatibility limitations.

The incompatibility of polymers with fluids is commonly measured in terms of percent volume swell of an unconfined sample exposed to a particular liquid. The actual amount of swell experienced in typical usage where the seal is contained and only partially exposed to the fluid will usually be much lower. The swell of seal materials, however, should not be considered detrimental to static seal operation unless deformation of the housing is induced or extrusion of seal material from the gland occurs. Through appropriate design, the increase in seal loads resulting from swell can be used to advantage.

Different polymer compounds exhibit various amounts of shrinkage during the post-mold cooling operation. The differences in mold shrinkage rate on commercially available seals are compensated for by a corresponding increase in mold size. Selection of a non-standard polymer as an O-ring material may require a special mold to achieve a desired standard O-ring size, depending on shrinkage considerations. Silicons and Vitons will result in parts which are from three to five percent smaller than the conventional synthetic rubbers made from the same mold.

Metallic Materials. Metal seal materials are normally used for high-pressure, high-temperature, and/or corrosive fluid applications. The most commonly used materials for metallic seals are steels, copper alloys, aluminum alloys, Monel, Inconel, gold, silver, and nickel.

Selection of metallic seal materials must include consideration of the thermal environment, since the levels of creep strength, fatigue strength, and notch sensitivity of metallic materials may be adversely affected by extremes of temperature. Typical creep strength properties of metallic materials at elevated temperatures are given in Table 13.5.2.2a.

Differences in thermal expansion, if not compensated for, can result in seal failure through structurally damaging loads or loads relaxation. Coefficients of thermal expansion are given in Table 13.5.2.3a.

The material of the seal must be chemically compatible with the contained media and with the surrounding environment. Material combinations which would promote corrosion must be avoided, particularly those combinations of materials which would promote galvanic action, oxidation, or stress corrosion. The relative galvanic series of various materials is shown in Table 13.7.3.1b.

In metallic seals, plastic flow of one of the materials at the seal interface is usually required to obtain an effective seal. Softer materials with a desirable high plastic strain range will usually have a low ultimate strength level. To balance the requirements of high strength in the seal body and good plastic flow under low stress at the interface, high strength materials are often plated with a thin coating of a softer material. Plating materials and their recommended applications are given in Table 6.3.2.2. Liquid metals such as mercury and gallium have also been applied as seal coatings. The liquid metals readily fill voids between mating surfaces yet are viscous enough to resist flow under high pressure differentials (Reference 131-23).

Table 6.3.2.2. Standard Platings and Coatings for Metallic Seals

(Adapted from "Metal V-Seal," Cat. 8800, 1964, Parker Seal Company, Culver City, California)

Plating or Coating	Temperature Range (°F)	General Recommendations
Silver	- 325 to + 1650	Excellent general purpose plating for high temperature resistance, but generally less suitable for cryogenic temperatures than gold or Teflon. Excellent chemical and radiation resistance.
Gold	- 423 to + 1850	Similar to silver but somewhat better resistance to certain corrosive fluids. Improved high and low temperature resistance but higher in cost than silver.
Teflon (TFE)	- 423 to + 500	Excellent coating for applications up to + 500°F. Excellent chemical resistance. Particularly suitable for cryogenic applications.
Teflon (FEP)	- 423 to + 400	Similar to Teflon (TFE) but somewhat softer and more dense. High temperature resistance lower than Teflon (TFE).
Kel-F	- 423 to + 300	Similar to Teflon, but more resilient and plastic at low temperatures, and generally higher in cost than Teflon.
Platinum	- 423 to + 3100	Highest temperature-resistant plating. Normally limited to use with ultra high temperature base metals such as TZM.
Nickel (soft)	- 325 to + 2500	High temperature-resistant plating but slight sacrifice in softness and ductility compared to other platings.
Lead	- 65 to + 450	Very soft plating but limited temperature resistance. Excellent radiation resistance.
Indium	- 320 to + 300	Very soft plating but limited temperature resistance. Suitable for cryogenic applications.
Aluminum	- 423 to + 900	Compatible with most oxidizers and fuels, but extremely costly as a plating material. Particularly suitable for liquid and gaseous fluorine.
Tin (pure)	- 32 to + 350	Very ductile, but limited temperature resistance. Usage limited to a few corrosive chemicals.
Copper	- 423 to + 1900	Suitable for vacuum applications; resistant to fluorine and certain other corrosive chemicals.

6.3.2.3 SEALED MEDIA. The primary effect of the sealed media on the seal performance is compatibility with the seal and gland materials. Compatibility data of seal materials with various fluids are given in Sub-Section 12.5.

The properties of the media being sealed affect the leakage rate. The maximum size of the leak path which can be tolerated will be determined by various properties of the contained media, such as viscosity, density, surface tension, and phase state (i.e., gaseous, liquid, or mixed). The flow of liquid through a very small passage is related to the ability of the liquid to wet the surfaces, and the higher the liquid surface tension the lower its wetting tendancies. Therefore, due to the phenomenon of surface tension, a leak path for gases may be impervious to liquid leakage. This significant difference between gas and liquid sealing must be appreciated when testing with substitute fluids. A helium leakage check following a hydrostatic proof test may be very misleading by virtue of residual water actually sealing against gas leakage. Eventual evaporation will destroy this "liquid seal." On the other hand, requiring that a component intended for liquid service be leak tested with either gaseous nitrogen or helium may result in unnecessary added cost because of overdesign.

The difficult problem of measuring liquid leakage, however, particularly with highly volatile and toxic liquids such as commonly used in aerospace applications, has resulted in the rather widely accepted practice of specifying leakage allowance for liquid components in terms of either nitrogen or helium gas.

If surface tension effects are ignored, and in those cases where leakage flow of both the gaseous and liquid media are laminar (Equations 3.11.3a and b) the flow relationship may be expressed as

(Eq. 6.3.2.3a)

$$Q_l = Q_g \frac{2\mu_g P_s T_g}{\mu_l (P_{1g} + P_{2g}) T_s}$$

where

Q_l = volumetric flow rate of liquid, cc/sec

Q_g = volumetric flow rate of gas, std cc/sec (sccs)

μ_g = absolute viscosity of gas, lb_f sec/in^2

μ_l = absolute viscosity of liquid, lb_f sec/in^2

P_s = standard atmospheric pressure, psia

P_{1g}, P_{2g} = upstream and downstream gas pressures respectively, lb_f/in^2

T_l = absolute temperature of gas, °R

T_s = standard absolute temperature, °R

Various studies are currently being made of gas/liquid leakage correlation in the molecular and transition flow regimes, but at present no reliable correlations have been established. In addition to the material presented in Sub-Section 3.11, further information on gas/liquid leakage correlation may be obtained from References 12-10, 36-30, 46-7, 46-33, 131-28, and 565-1.

6.3.2.4 GLAND DESIGN. The gland, which is that portion of the housing structure surrounding the seal, serves to position and/or contain the seal, in addition to providing the seal mating surfaces. It is through the gland that preload is applied to the seal. (The subject of preload is treated in detail under Detailed Topic 5.12.3.2.) The following paragraphs discuss basic gland design types and the gland design and manufacturing considerations related to seal contact area and machining tolerances.

Gland Design Classification. Gland designs fall into three categories: unconfined, semi-confined, and fully confined. Typical examples are shown in Figure 6.3.2.4.

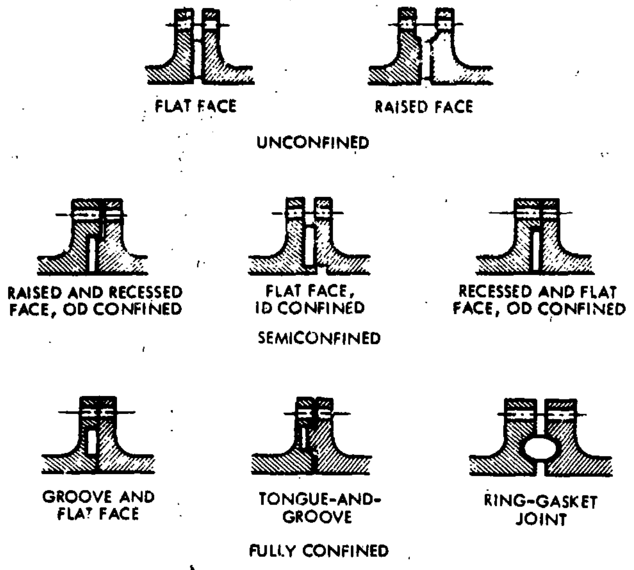


Figure 6.3.2.4. Gland Design Classification

Unconfined. In an unconfined gland design, contact between the seal and the gland occurs only at the sealing interface; the seal is free to deform normal to the seal load. Non-circular flat gaskets are usually limited to this design. The gland faces may be flat, allowing gasket width and deformation of the gaskets to determine the contact area, or they may have a raised face to limit the contact area. The amount of compression of the seal is not limited by the gland design. Most seals utilizing an unconfined gland design are series loaded, i.e., all the load transmitted between the gland faces is carried by the seal. The unconfined gland does not, in itself, directly provide a means of positioning the seal; however, the seal may be located by sizing the seal outside diameter to coincide with either the inside perimeter of the bolt pattern or the outside diameter of the gland.

Semi-confined. In a semi-confined gland design, freedom of the seal to deform normal to the applied load is restricted in one direction. Seal deformation or pressure differential may result in an interface load on the restraining surface, but normally it is only used for positioning the seal rather than to aid in sealing. This type of gland design is the most common for static seals and is used for nearly all circular gasket applications. The restricting surface may either be on the inside or the outside diameter of the seal. The amount of compression of the seal may be limited by designing metal-to-metal seating of the gland flanges. Normally, in the semi-confined gland the depth of the recessed face is equal to or less than the height of the raised face, in order to prevent the flanges coming together metal-to-metal when the gasket is compressed. Metal seating of the gland flanges will result in parallel seal loading, i.e., a division of the preload between the gland and the seal. The relationship between series and parallel loading and preload is described in Detailed Topic 5.12.3.2.

Fully Confined. A fully confined gland design provides contact on all sides of the seal. The fully confined groove and flat face gland configuration provides a fixed seal compression and is normally limited in use to O-rings. Fluid pressure energizes the seal in either direction. The tongue and groove configuration is a fully confined gland used to prevent blowout in applications where extreme high pressure is being sealed. Flat gaskets are commonly used in tongue and groove gland designs. The tongue depth determines whether parallel loading limits the amount of seal compression or whether series loading is applied. The ring joint is a fully confined gland design used for solid metal shape gaskets or for hollow metal O-rings. This is one of the most expensive gland designs, because of the very close concentricity control required between the gland flanges.

Seal Contact Area. The apparent seating stress is a function of the sealing area and the applied seal load. By selecting a seal or designing the gland with a small contact area, i.e., narrow interface, high stresses may be attained even though the total applied load is relatively small. Care must be exercised when using narrow seal gland interfaces to prevent damage during handling or installation. This design is quite susceptible to failure resulting from scratches or flaws across the sealing face. Extreme cleanliness at installation and careful handling are mandatory for this type seal. The gland design will, to a large extent, determine the effective load applied to the seal. Data on the effective seal width of gaskets as affected by gland design are presented in Detailed Topic 6.3.3.2.

Tolerances. The installed relationship of the seal interfaces is influenced by tolerance buildup accumulated in the machine tool production of seals and seal components. Table 6.3.2.4 outlines some of the standards of accuracy which can be obtained from various regular machine tools in use. Acceptable limits on misalignment of seal interfaces, and critical tolerances on positioning, flatness, etc., that can be allowed, will vary with seal types and also with seal installation. As shown in Table 6.3.3.1, these limitations can vary from 0.0001 to 0.010 inch.

6.3.2.5 LEAKAGE ALLOWANCE. Of primary consideration in a seal design is the determination of the acceptable amount of leakage which can be tolerated at any seal interface. Each design must be examined with respect to the total system and mission requirements in order to arrive at an acceptable leakage rate that can be permitted for each of the seal joints. Since reduction of the leakage rate requires a reduction in the size of the leak path (which in turn necessitates higher seal loads, smoother surface

Table 6.3.2.4. Standards of Accuracy for Machine Tools

Machine	Flatness per 12 Inches	Average Roughness (microinches)	Spindle Runout Total Indicator Reading
Milling Machine			
Regular	0 to 0.001	32	0 to 0.0005
Special	0 to 0.0005	8	0 to 0.0001
Surface Grinder			
Regular	0 to 0.0001	-	-
Special	0 to 0.000025	16	-
Lapping machine	0 to 0.000011 (one light band)	2	-
Lathe	0 to 0.002	32 to 125	0 to 0.0001

finishes, or softer seal materials) attempts at achieving an overly restrictive leakage rate could result in an excessively complicated design. Because of the need for close manufacturing control on items such as surfaces finishes, critical tolerances, flatness, etc., such designs may also become very expensive. In the case of metallic gaskets, the higher seal loads required could result in a weight penalty due to the associated requirement for more rigid glands.

An extremely low leakage rate may be below the permeation rate of many nonmetallic materials, thus eliminating their consideration as a seal material. Determination of an acceptable leakage rate will depend on the conditions under which the seal will function. Factors which determine acceptable leakage rates include fluid loss allowances, system performance, corrosion considerations, and hazards associated with contaminating the environment.

Zero Leakage. In establishing a leakage rate specification, a common tendency is to state that the leakage shall be zero; however, it should be recognized that an absolute seal is impossible to obtain. Gaseous leakage inspection of a part by immersion may show a zero bubble count over an extended period of time, while inspection of the same part by a mass spectrometer method may show a relatively high leakage rate. Leakage by diffusion through the solid walls of the component or by permeation through the seal material is generally well below zero leakage as determined by a bubble test, although it may still be readily detectable by mass spectrometer measurements. Zero leakage as defined in Detailed Topic 6.3.1.3 is 1×10^{-6} atm cc/sec of He. The sensitivity range of various leakage measurement techniques is given in Table 5.17.7.1.

"Zero leakage" has received various definitions for specific applications or programs in recent years. Bubble-tight leakage has frequently been specified for the high-pressure components of launch-vehicle rocket engines (which operate at pressure for only a few minutes), whereas spacecraft systems requirements dictate leakage rates measurable only by mass-spectrometer techniques. Reference 36-14 indicates that the current NASA approach is to specify a maximum leakage of 1×10^{-6} scc/sec of gaseous helium for spacecraft applications and 1×10^{-4} scc/sec for launch-vehicle applications.

Leakage Flow. Total leakage will be a combination of two factors: (1) permeation or diffusion through the gland and/or seal, and (2) flow at the seal interface. (For definition and coverage of these factors see Sub-Topic 3.11.5). Analytical methods for determining the flow rate at the interface surface have been restricted to flow paths which are relatively simple in shape and easy to define, as discussed in Sub-Section 3.11. In actual practice, flow paths are seldom simple or of a uniform cross section. Although interface flow is predominantly laminar, in very small passages flow is complicated by transition from laminar to molecular, and quite often two-phase flow will exist. To determine or predict an absolute value of leakage rate for most applications, measurement of actual performance by testing will be required. (See also Detailed Topic 6.2.3.11 and Reference 35-14.)

6.3.2.6 REUSABILITY. The degree to which seal components may be reused after disassembly is an economic consideration, particularly for separable connector applications. In general, when plastic deformation of one or both of the components is required to effect a seal, the components will not be reusable; if the deformation is in the elastic range, the components will probably be reusable. In those areas where plastic deformation is required, consideration should be given in material selection to limiting the plastic deformation to one component only. The component that is most easily replaced (normally the seal element) should be designed or selected on the basis of plastic deformation occurring in the component, with little, if any, occurring in the mating gland interface. The relative relationship of plastic deformation of two interfaces will govern the number of times that the seal can be replaced without a major rework of the component.

Nonmetallic seals, particularly elastomers, are generally reusable; however, because of their low cost this type of seal is often replaced when an installation is disassembled in order to improve the reliability of the seal joint.

Soft metal plating is often used on metallic seals in order to reduce the contact stress required for sealing. Because of this plating, many of these seals may be reused if the base structure is not plastically deformed.

6.3.2.7 TEMPERATURE. In addition to the points discussed in Detailed Topic 13.5.2.3, it should be noted that the selection of seal materials with appropriate coefficients of thermal expansion can increase the seal load under temperature extremes as well as decrease it. In some cryogenic applications, use of a material such as Invar has been utilized in overcoming the thermal expansion differential normally associated with the elastomer and the metallic components. Thermal expansion data for typical seal and gland material are shown in Table 6.3.2.7.

In the design of a seal joint, care must be taken to ensure that the required seal load is maintained throughout the operating environment of the seal. Two temperature conditions which could affect the stability of the seal load are

Table 6.3.2.7. Linear Thermal Expansion of Typical Seal and Gland Materials

Material	Contraction +75 to −65°F (in./ft)	Expansion +75 to +375°F (in./ft)	Coefficient of Expansion (in./in.°F)
Teflon	0.093	0.148	5.5×10^{-5}
Nitrile, general purpose	0.218	0.468	13.0×10^{-5}
Chloroprene	0.190	0.407	11.3×10^{-5}
Viton-A	0.130	0.277	9.0×10^{-5}
Kel-F	0.064	0.137	3.5×10^{-5}
Silicone	0.252	0.540	15.0×10^{-5}
High-temperature aluminum, 2017	0.022	0.047	13.0×10^{-6}
Stainless steel, type 302	0.016	0.035	9.6×10^{-6}
Steel, mild	0.011	0.024	6.7×10^{-6}
Invar	0.001	0.002	6.0×10^{-7}

the steady state effects at temperature extremes and the transition effects during temperature change. One of the chief problems associated with the thermal gradients is the possibility of a momentary shifting of the sealing surface, which may destroy the surface mating achieved in original assembly. This would result in a shift of the plastically-deformed interface and a corresponding change in the leakage path.

6.3.2.8 PRESSURE. Design of the structure surrounding the seal element must take into consideration the effects of continued operation at pressure extremes as well as the possibility of pressure surge effects. The surrounding structure must be sufficiently rigid to ensure that any deformation of one of the seal interfaces will not allow the seal load to drop below that value required for the application.

The differential pressure across the seal is a factor in establishing the maximum allowable leak path and its related apparent seating stress. The higher the differential pressure, the higher the leakage rate will be through a given size leakpath.

Unloading of an O-ring seal will occasionally occur when pressure surges cause the O-ring to roll in the groove. When the pressure decreases, the resilience of the O-ring causes it to return to the original position. Use of a narrower groove may be necessary if this condition occurs.

The magnitude of pressure differential across the seal will have a significant effect on the seal load required at the seal interface in mechanically preloaded seals. As a general rule, the seating stress is made at least as high as the maximum fluid pressure. Another method of determining the minimum seating stress required at the interface is to multiply the pressure differential by a gasket, m. This factor, discussed in Detailed Topic 6.3.3.2, is an empirically determined number which is dependent on the seal configuration and seal materials.

The pressure-energized seals utilize increases in the magnitude of pressure differential to increase the seal load. In O-ring design, the elastomer acts as a highly viscous liquid with high surface tension, and transmits the increased pressure load to the sealing surface. In flexible metallic seals, the interface contact area is designed much smaller than the area of the flexible member exposed to the contained fluid. Increase in fluid pressure results in correspondingly higher loads at the contact interface.

The rate of pressure change must be considered when using elastomers in a high temperature pneumatic system. The gases absorbed by the seal material will expand and the forces generated can rupture the material if the decompression rates are high. Typical failure data is described in Reference 106-3.

6.3.2.9 RESILIENCY. Factors such as axial loads, bending, and thermal gradients which contribute to deflections tending to separate static seal interfaces are discussed under "Threaded Connectors" in Sub-Section 5.12.

The resiliency of the seal is a measure of its capability to accommodate flange deflection without reducing seal loads to a level where the allowable leakage rate is exceeded. Resiliency may be expressed as the ratio of energy given up on recovery from deformation to the energy required to produce the deformation.

In nonmetallic designs, particularly those utilizing elastomers, the seal load due to deflection of the seal during installation in the housing results in a significant portion of the stress necessary for sealing. The natural resilience of nonmetallic materials, the effective pressure actuation, and the initial deflection (for O-rings this is usually 10 to 20 percent of the cross-section width) act to reduce the problem of increased leakage due to flange or housing deformation.

In metallic designs, there is little or no resilience in the material itself. A measure of resilience may be provided by designing the seal so a flexible member is deformed elastically in a cantiliver type of deflection by the housing during installation.

In parallel loaded seal joints, a measure of resiliency can be designed into the gland by preloading the gland to a load level beyond that required at the sealing interface. Relaxation of this preload from the imposed factors must occur before the seal loads will begin to diminish (see Detailed Topic 5.12.3.2).

6.3.2.10 COST. In many aerospace applications cost is not normally a dominant factor in selection of components, because of the limited quantities involved and the premium placed on performance. Since in the application of seals commercially available designs are often easily adapted, the use of special design seals should be avoided wherever possible. Cost factors for commercial seals will vary by several orders of magnitude, as shown in Table 6.3.2.10.

Table 6.3.2.10. Cost Magnitude Factor for Commercial Seals

Seal Type	Factor*
Plastic O-ring	2 - 3
Metal O-ring	5 - 10
Metal gasket	2 - 20
Elastomer gasket	1 - 3
Metal shapes	10 - 125
Plastic shapes	5 - 25
Special design	50 - 1000

*Based on cost of elastomer O-ring = 1

6.3.3 Design Data

This sub-topic presents design data on specific types of static seals. Much of the data presented is empirical, owing to the general lack of proven analytical techniques available for relating parameters such as surface conditions, material properties, and applied stresses to seal leakage. Much work remains to be done in the development of static seal parametric data and analysis techniques. Available data is presented in this sub-topic on gaskets, O-rings, spring loaded seals, radial metallic seals, boss seals, shear seals, seals for cryogenic applications, and seals for vacuum service.

6.3.3.1 STATIC SEAL COMPARISON CHART. Table 6.3.3.1 presents comparative data on a variety of static seals used in fluid system applications, compiled from Reference 34-13 and from the various manufacturers' literature.

6.3.3.2 GASKETS. Gaskets form the bulk of the mechanically preloaded seals. A gasket is a solid shape, usually flat and of rectangular cross section, which is clamped between two members of the gland to provide the seal. As a rough guide to determine whether a gasket should be metallic or nonmetallic, multiply the operating pressure differential in psi by the operating temperature in degrees F. If the result exceeds 250,000 only metallic gaskets should be used (Reference 49-35). Use of nonmetallic gaskets may be further limited by extreme temperatures. O-rings, because of their wide usage, are considered as a special kind of gasket and are covered in Detailed Topic 6.3.3.5.

In determining the minimum gasket apparent seating stress required, a series of empirical relationships have been developed by the American Society of Mechanical Engineers (Reference 68-70). These relationships have generally proved satisfactory, however where allowable leakage rates are low, higher stress levels may be required. The minimum gasket unit seal load, F_a, required to seat the gasket is expressed as:

$$F_a = bY \qquad \text{(Eq 6.3.3.2a)}$$

where

F_a = unit seal load, lb_f/in. of gasket length
b = effective gasket seat width, in.
Y = minimum design seating stress, lb_f/in^2

At operating conditions, the effect of the contained pressures is considered with the use of a gasket factor, m, as follows:

$$F_p = bm\,\Delta p \qquad \text{(Eq 6.3.3.2b)}$$

where

F_p = unit seal load based on operating pressure, lb_f/in. gasket length
b = effective gasket seat width, in.
m = gasket factor
Δp = pressure differential across seal, lb_f/in^2

Typical values for effective seating width, b, are shown in Table 6.3.3.2a for various gland and seal interface conditions.

Values for Y and m for various gasket materials and configurations are presented in Tables 6.3.3.2b, c, and d.

F_p should be used as the design seal load value if it exceeds F_a.

Nonmetallic Gaskets. Typical seating stress required for nonmetallic gaskets is shown in Table 6.3.3.2b. Using seating stress values and the compression curves of Figure 6.3.3.2a, the installed thickness of the gasket can be determined for various seal loads applied. In an unconfined flange design, to provide the highest interface stress for a given applied load, the gasket material should be as thin as possible. Additional load is required in thick gaskets due to the load lost in bulk deformation and cold flow. On the other hand, if a high degree of resiliency is required of the gasket, this characteristic is achieved through the use of thicker gaskets. Due to the high deformation available in elastomer materials, simple gasket shapes can be adapted to irregular designs, eliminating the need for molded shapes. Typical examples are shown in Figure 6.3.3.2b.

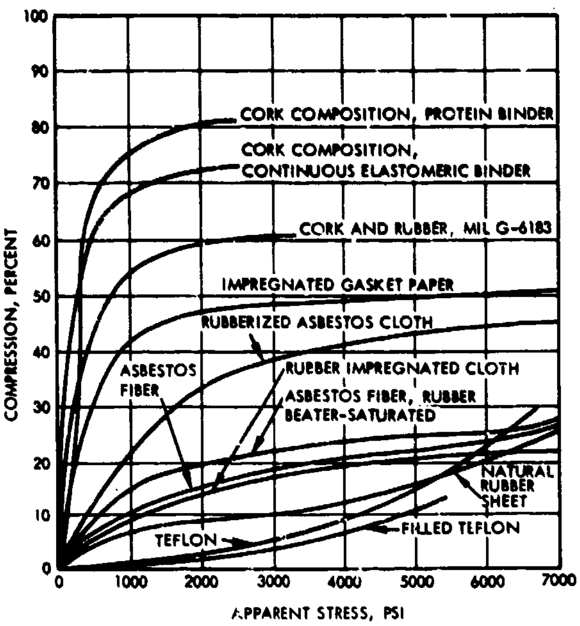


Figure 6.3.3.2a. Compression for Varying Seal Stress for Common Nonmetallic Gasket Materials

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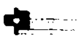
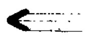
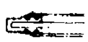
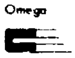
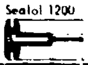
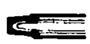
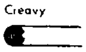
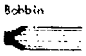
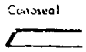
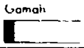
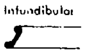
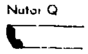
Table 6.3.3.1. Static Seal Comparison Chart
(Reference 34-13)

Seal Type	Configuration or Trade Name	Manufacturer	Base Material	Coating	Gland Design Requirements: Surface Finish (microinches)	Flatness (in.)	Allowable Separation (in.)	Unit Seating Load lb/in	Deflection Limit Required	Max. Pressure Limit Without Special Gland or Seal Design (psi)	Relative Cost: Gland	Seal	Remarks
Gasket	Flat	Various	Elastomers and plastics	None	32-64	0.010	0.010	See Table 6.3.3.2b	No		Low	Low	Most common moderate temperature flat gasket seal; permeable.
	Flat, plain, and grooved	Various	All metals	None	8-32	0.001	0.0000	See Table 6.3.3.2c	No		Low	Low	Gland must be designed to assure no distortion or deflection.
	Corrugated	Various	All metals	None	64	0.002	0.005	See Table 6.3.3.2c	No	500	Low	Low	
	Solid round	Various	All metals	None	32	0.001	0.000	See Table 6.3.3.2c	No	15,000	Low	Low	Gland must be designed to assure no distortion or deflection.
	Solid shapes	Various	All metals	None	32	0.001	0.000	See Table 6.3.3.2c	No	15,000	High	Moderate	Gland must be designed to assure no distortion or deflection.
	Jacketed	Various	See Detailed Topic 6.3.3.2	None	64-125	0.005	0.005-0.010	See Table 6.3.3.2d	No*	**	Low	Moderate	
	Spiral wound	Johns Manville, Flexitallic, Garlock	See Detailed Topic 6.3.3.2	None	64-125	0.0002 per [illegible]	0.0005	See Table 6.3.3.2d	Yes	**	Low	Low	
	Molded shapes	Parker Seal Co., Stillman Rubber Company	Elastomer seal shape molded to metal base	None	32-64	0.003	0.003	25-30	No		Low	High	Used in multiple O-ring installations where insufficient gland room is available.
	O-ring	Various	Elastomers and plastics	None	32-64	0.004	0.004	5-200	Yes	1500	Moderate	Low	Most common moderate temperature seal; permeable.
	Hollow O-ring	United Aircraft Products Inc., Advanced Products [illegible] Mfg. Co., Parker Seal Co.	Various metals	See Table 6.3.2.2	8-32	0.0002 per circ. in. 0.001 max.	See Table 6.3.3.4c	See Table 6.3.3.4c	Yes	5000	Moderate	Moderate	Sensitive to radial scratches.
Flexible Metals	[illegible]	Pressure Science, Inc.	Various metal alloys	Silver, gold, Teflon	16-32	0.0002 in.	0.002	200	Yes		Moderate	Moderate	Sensitive to radial scratches.
	[illegible]	Pressure Science, Inc.	Various metal alloys	Silver, gold, Teflon	16-32	0.001	0.006	100-300	Yes		Moderate	Moderate	Higher resiliency than O-ring.
	K Seal	Harrison Mfg. [illegible]	Stainless alloys	See Table 6.3.2.2	8-32	0.0002 in.	0.002	30-60	No	**	High to moderate	High	Sensitive to handling damage; requires critical gland finish control.
	[illegible]	[illegible] Engineering	Stainless alloys	See Table 6.3.2.2	2-32	.0005	0.002	20-100	No	**	High to moderate	High	Same as K Seal.
	Big Edge	[illegible]	Various stainless steels	None	16	0.001	0.002	20-100	Yes		Moderate	High	Same as K Seal.
	[illegible]	Aerospace [illegible]	Various metal alloys	See Table 6.3.2.2	8-64	0.002	0.002	20-80	Yes	5000	High to moderate	High	Same as K Seal.

* [illegible]
** Dependent on seal diameter and cross section.

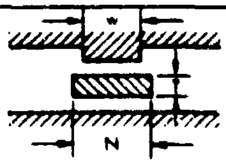
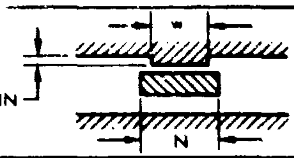
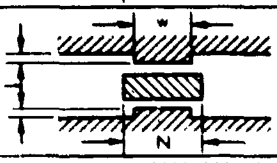
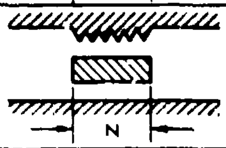
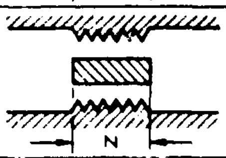
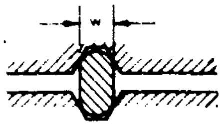
Table 6.3.3.1. Static Seal Comparison Chart (Continued)

Seal Type	Configuration or Trade Name	Manufacturer	Base Material	Coating	Gland Design Requirements: Surface Finish (microinches)	Flatness (in.)	Allowable Separation (in.)	Unit Sealing Limit (lbs/in.)	Deflection Limit Required	Maximum Press Limit Without Special Gland or Seal Design (psi)	Relative Cost: Gland	Seal	Remarks
Flexible Metallic (Cont'd.)	Pressure Lock Nastat 	Koppers Co., Navan Prod.	Stainless steel	None	8-32	0.0015	0.002	220-550	Yes	1500	High to moderate	High	Same as K Seal, relatively high unit seal load required.
	Hydrodyne 	Donaldson Co.	Various metal alloys	See Table 6.3.2.2	8	0.0002 in.	0.002	[illegible]	Yes	[illegible]	High	High	Same as K Seal.
	Vee 	Parker Seal Co., Hi-Temp Rings, Tetrafluor, Inc.	Various metal alloys	See Table 6.3.2.2	16-32	0.0005 in.	0.008	100-250	Yes		Moderate	Moderate	Similar to K Seal.
	Naflex 	Navan Prod.	Inconel 718, 4340 Stainless	Nickel Plated	32 (Circular lay)	0.0006 in.	0.012	200-300	No	3000	Low	High	Similar to K Seal, increased flange separation capability resulting from heavier plating thickness
	Pneutlex 	Del Mfg. Co.	Stainless and Al. alloys	See Table 6.3.2.2	16	0.0002 in.	0.002-0.003	60	Yes		Moderate	Moderate	
	Bar-X 	Wiggins Oil Tool Co.	Stainless alloys	See Table 6.3.2.2	16	0.0005	0.002	250	Yes		Moderate	High	Similar to K Seal.
	Nucoseal 	National Utilities	Stainless or Al. alloys	None	64	0.005	0.005	20-80	Yes		Low	High	
	Omega 	Servotronics	Various	See Table 6.3.2.2					No				
	Sealol 1200 	Sealol, Inc.	Stainless or Al. alloys	Various plating: gold, silver, etc.	32	0.050	0.050	Very low	No	10,000	High	High	Excellent flange separation capability may justify high cost for some applications
Plastic Spring-Loaded	Bal-Seal 	Bal-Seal Mfg. Co.	Teflon jacket over stainless coil spring	Not applicable	32	0.005	0.010	50-100	Yes	1200	Moderate	Moderate	Permeable, LH_2 brittleness.
	Omniseal 	Aeroquip Corp.	Teflon jacket over flat stainless helical spring	Not applicable	63	0.005	0.020	50-100	Yes	1200	Moderate	Moderate	Permeable, LH_2 brittleness.
	Raco 	Raco Mfg. Co.	Teflon jacket over stainless finger spring	Not applicable	32	0.005	0.015	50-100	Yes	1200	Moderate	Moderate	Permeable, LH_2 brittleness.
	Creavy 	Haskel Engineering	Teflon tube over stainless steel coil spring	Not applicable	32	0.005	0.010	50-100	Yes	1500	Moderate	Moderate	Permeable, LH_2 brittleness.
	Tec Ring 	Tec Seal Corp.	Teflon jacket	Not applicable	32	0.005	0.010	50-100	Yes	1200	Moderate	Moderate	Permeable, LH_2 brittleness.
Radial Metallic or Toggle	Bobbin 	Battelle Institute	Stainless alloys	None	32	0.020	0.005	500	Yes		High	High	Requires special gland.
	Conoseal 	Aeroquip Corp.	Stainless or Al. alloy	None	32	0.020	0.020	500-600	Yes		High	Moderate	Requires special gland.
	Gamah 	Gamah Corp.	Stainless or Al. alloy	None	32	0.010	0.005	500	Yes		High	High	Requires special gland.
Metallic Boss	Infundibular 	Futurecraft	Stainless alloys	None	32	0.001	0.0005	1000-2350	No		Moderate	High	Will fit standard and 10050 boss.
	Nutor Q 	Navan Prod.	Stainless alloys	None	32	0.001	0.0005	1000	No		Moderate	High	Will fit standard and 10050 boss.

Total load applied should not crush gasket.
Dependent on seal diameter and cross section.

Table 6.3.3.2a. Effective Gasket Width, Annular Ring Gaskets

(Adapted from "Rules for the Construction of Unfired Pressure Vessels," 1962 Edition, ASME Boiler and Pressure Vessel Code, Section VIII, American Society of Mechanical Engineers, 345 East 47th Street, New York 17, New York)

FACING SKETCH (EXAGGERATED)	BASIC GASKET SEATING WIDTH, b_o	
	COLUMN I	COLUMN II
1a 	N	N
1b  $w \gtrless N$	$w + T$; $\left(\frac{w+N}{2}\ max\right)$	$w + T$; $\left(\frac{w+N}{2}\ max\right)$
2  1/64" NUBBIN; $w \leq \frac{N}{2}$	$\frac{w+N}{2}$	$\frac{w+3N}{4}$
3  1/64" NUBBIN; $w \leq \frac{N}{2}$	w; (N min)	$w + N$; $\left(\frac{3N}{4}\ min\right)$
4* 	$\frac{3N}{4}$	$\frac{7N}{8}$
5* 	$\frac{N}{2}$	$\frac{3N}{4}$
6 	$\frac{w}{4}$	

Effective Gasket Seating Width, b:

$b = b_o$ when $b_o \leq 1/4$ in.

$b = \frac{\sqrt{b_o}}{2}$ when $b_o > 1/4$ in.

*Where serrations exceed 1/64 in. depth and 1/32 in. width spacing

Table 6.3.3.2b. Minimum Seating Stress for Nonmetallic Gaskets

(Adapted from the following: "Select and Apply Gaskets Effectively," by J. J. Whalen, in Chemical Engineering, vol. 69, no. 20, 1 October 1962, Copyright 1962 by McGraw-Hill Inc.; and "Rules for the Construction of Unfired Pressure Vessels," 1962 Edition, ASME Boiler and Pressure Vessel Code, Section VIII, American Society of Engineers, 345 East 47th Street, New York 17, New York; also from Reference 36-12)

Gasket Material	Thickness (in.)	Gasket Factor (m)	Minimum Design Seating Stress (psi) (Y)*	Minimum Seating Stress (psi) (Y)**	Sketch	Ref. Table 6.3.3.2a Use Facing Sketch	Ref. Table 6.3.3.2a Use Column
Rubber without fabric or a high percentage of asbestos fiber:							
Below 75 Shore Durometer	1/32 and up	0.50	0	175			
75 or higher Shore Durometer	1/32 and up	1.00	200	200			
				SBR Binder / CR Binder			
Asbestos with a suitable binder for the operating conditions	1/8	2.00	1600	1200 / 1500		1(a,b) 4, 5	II
	1/16	2.75	3700	1600 / 2000			
	1/32	3.50	6500	2000 / 2500			
	1/64			3000 / 3750			
Rubber with cotton fabric insertion	1/32	1.25	400	400			
Rubber with asbestos fabric insertion, with or without wire reinforcement	4 ply			1800			
	3 ply	2.25	2200	2100			
	2 ply	2.50	2900	2500			
	1 ply						
Fluorocarbon Polymer (TFE)							
Virgin	1/64			14000			
	1/32			6500			
	1/16			3700			
	1/8			1600			
Glass-filled	1/64			14000			
	1/32			11000			
	1/16			6000			
	1/8			3000			
Asbestos cloth (impregnated)	3/32			1600			
Fluorocarbon Polymer (FEP)							
Virgin	3/32			2000		1(a,b) 4, 5	II
Encapsulated glass fabric***	3/32	1.31		3000			

*Data obtained from "ASME Code for Unfired Pressure Vessels," 1962
**Data obtained from "Chemical Engineering," October 1, 1962
***Data obtained from Proceedings Conference on Leak-Tight Connectors, NASA MSFC, March 1964

Table 6.3.3.2c. Minimum Seating Stress for Combination Gaskets

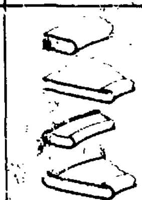
(Adapted from "Rules for the Construction of Unfired Pressure Vessels," 1962 Edition, ASME Boiler and Pressure Vessel Code, Section VIII, American Society of Engineers, 345 East 47th Street, New York 17, New York; also from Reference 1-292)

Gasket Configuration	Data from "ASME Code for Unfired Pressure Vessels," 1962			Data from "Machine Design Seals Book," June 1964			Sketch	Ref. Table 6.3.3.2a	
	Material	Factor (m)	Min. Seating Stress (Y)	Material	Thickness	Min. Seating Stress, psi (Y)		Use Facing Sketch	Use Column
Solid flat metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	4.00 4.75 5.50 6.00 6.50	8800 13000 18000 21800 26000	Aluminum Copper Soft steel (iron) Monel Stainless steel	1/8	16,000 36,000 55,000 65,000 75,000		1(a,b) 2,3,4,5	I
				Aluminum Copper Soft steel (iron) Monel Stainless steel	1/32 and 1/16	20,000 45,000 68,000 81,250 93,750			
Grooved flat metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	3.25 3.50 3.75 3.75 4.25	5500 6500 7600 9000 10100	Aluminum Copper Soft steel (iron) Monel Stainless steel	1/8-in. pitch, all thicknesses	25,000 35,000 55,000 65,000 75,000		1(a,b) 2,3	II
				Aluminum Copper Soft steel (iron) Monel Stainless steel	1/16-in. pitch, all thicknesses	30,000 40,000 60,000 70,000 80,000			
				Aluminum Copper Soft steel (iron) Monel Stainless steel	1/32-in. pitch, all thicknesses	35,000 45,000 65,000 80,000 95,000			
Corrugated metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.75 3.00 3.25 3.50 3.75	3700 4500 5500 6500 7600	Aluminum Copper Soft steel (iron) Monel Stainless steel	1/8	1,500 2,000 4,000 4,500 6,000		1(a,b)	
Solid round				Aluminum Copper Soft steel (iron) Stainless steel	Any diameter	1300 lb/circular in. 1300 lb/circular in. 4500 lb/circular in. 6000 lb/circular in.		6	*
Solid round wrapped				Aluminum jacket Aluminum cores Aluminum jacket Stainless steel cores Stainless steel jacket Stainless steel cores	Any diameter	1500 lb/circular in. 1500 lb/circular in. 6000 lb/circular in.		1(a,b) 2,3	
Solid shapes	Iron or soft steel Monel or 4-6% chrome Stainless steels	5.50 6.00 6.50	18000 21800 26000					6	I

*Effective seat width not required

Table 6.3.3.2d. Minimum Apparent Seal Stress for Combination Gaskets

(Adapted from "Rules for the Construction of Unfired Pressure Vessels," 1962 Edition, ASME Boiler and Pressure Vessel Code, Section VIII, American Society of Engineers, 345 East 47th Street, New York 17, New York; also from Reference 1-292)

Gasket Configuration	Data from "ASME Code for Unfired Pressure Vessels," 1962			Data from "Machine Design Seals Book," June 1964			Sketch	Ref. Table 6.3.3.2 a	
	Material	Gasket Factor (m)	Min. Seating Stress (Y)	Material	Gasket Thickness (in.)	Min. Seating Stress (psi) (Y)		Use Facing Sketch	Use Column
Corrugated metal, asbestos inserted	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.50 2.75 3.00 3.25 3.50	2900 3700 4500 5500 6500	Aluminum Copper Soft steel (iron) Monel Stainless steel	1/8	2,000 2,500 3,000 3,500 4,000		1(a,b)	
Corrugated metal jacketed, asbestos filled				Lead Aluminum Copper Soft steel (iron) Monel Stainless steel	Approximately 9/64	500 1,000 2,500 3,500 4,500 6,000			II
Flat, metal jacketed, asbestos filled	Soft aluminum Soft copper or brass Iron or soft steel Monel 4-6% chrome Stainless steels	3.25 3.50 3.75 3.50 3.75 3.75	5500 6500 7600 8000 9000 9000	Lead Aluminum Copper Soft steel (iron) Nickel Monel Stainless steel	1/8	500 2,500 4,000 6,000 8,000 7,500 10,000		1a, 1b, 2*	
Spiral-wound metal, asbestos filled	Carbon, Stainless, or Monel	2.50 3.00	2900 4500	Carbon steel Stainless steel Stainless steel Stainless steel	1/8 3/16 1/8 3/16	2,500-15,000 2,500-15,000 3,000-30,000 3,000-30,000		1(a,b)	

*The surface of a gasket having a lap should not be against the nubbin

When using nonmetallic gaskets, particularly composites, care should be taken to avoid applying a seal load high enough to crush the gasket material. Crushing may cause loss of resiliency of the seal material and result in leakage with movement of the seal interface.

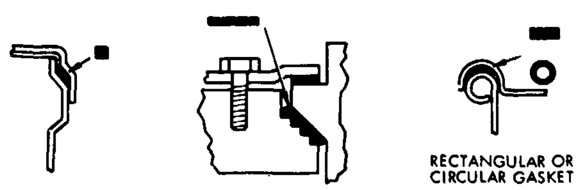


Figure 6.3.3.2b. Special Cavity Joints for Nonmetallic Gaskets

(Courtesy of Armstrong Cork Company, Lancaster, Pennsylvania)

Metallic Gaskets. Metallic gaskets fall into several basic groups:

a) Flat
b) Corrugated
c) Round cross-section
d) Special heavy cross-section.

Typical apparent seating stress values are shown in Table 6.3.3.2c.

The plain, solid, flat gasket is the most common form of metallic gasket, and is used where compressibility is not required to compensate for flange surface finish, warpage, or misalignment, and where sufficient clamping force is available for the metal selected. Generally, the width should be at least metal thickness plus 50 percent. There are no limitations on flat gasket dimensions beyond the width of metal sheet commercially available and handling practicability.

For best service, plain gaskets should be used between flanges with concentric serrated surfaces. Serrated or grooved metal gaskets may be used when pressure (radial strength), temperature, or the highly corrosive nature of the confined fluid necessitate the use of a metal gasket, and the seal load is not sufficient to seal a plain gasket.

Corrugated gaskets are a line contact seal. Multiple corrugations concentric with the ID provide some degree of resilience to the seal. A minimum of three corrugations is desirable to provide stability during compression and to provide redundant sealing. In full-face gaskets the addition of one corrugation outside the bolt circle will equalize the seal load and may be helpful in preventing flange distortion. A slight flat inside the inner corrugation and beyond the outer corrugation will help stiffen the gasket.

Corrugated gaskets are best suited for smooth-faced, complex or noncircular, low-pressure (500 psi) applications. They are available in metal thicknesses 0.010 to 0.031 inch, with corrugation pitches of 0.045 to 0.250 inch. Overall gasket thickness is 40 to 50 percent of corrugation pitch.

Round cross-section gaskets are usually made from round wire and used with specially designed grooved flanges. A single V-groove in one flange of the gland may be used provided the volume of the groove is less than the gasket. For use in an unconfined gland, the round gasket may be wrapped with a jacket of the same or a softer material. The outer edge of the jacket provides a means of centering on the flange bolts. An alternate configuration has two concentric metal ring cores enclosed and spaced by the jacket. The inner wire cross-section is normally about 0.020 inch heavier than the outer wire. This assures full initial seal load at the gasket ID, but the outer ring restricts the amount of flange bending. Bolt holes are usually in the connecting web or jacket.

Special heavy, solid, cross-section shapes have been developed for use in high pressure (1000 to 10,000 psi) piping systems and pressure vessels. Very high seal loads can be obtained with moderate bolt loads. The most common designs, which are used only in ring-gasket joints are the oval and octagonal shape.

Combination Gaskets. The available combination metallic-nonmetallic gaskets may be grouped as follows:

a) Corrugated-coated, filled
b) Jacketed-filled
c) Spiral wound
d) Molded shapes.

Typical apparent seating stress values are shown in Table 6.3.3.2d.

Coating a basic metallic corrugated gasket with a nonmetallic, or filling the corrugations with asbestos or Teflon cord, increases the usable pressure range and allows the use of rougher surface finish of the flanges. Where the metal corrugation is steel or a metal of similar hardness, coating reduces the minimum required apparent seating stress. Metal thickness is 65 to 75 percent of pitch, where pitch is the distance between corrugation peaks.

Metal-jacketed, soft-filler gaskets consist of a soft compressible filler, usually asbestos millboard, encased partly or wholly in a metal packet. For corrosive applications, Teflon may be used as the filler for temperatures up to 500°F. For temperatures over 500°F, metallic fillers may be used if completely encased, as in the double jacket or corrugated jacket types. The metal-jacketed gaskets are used in several different cross-sections (Figure 6.3.3.2c), each of which is best suited for particular applications. The one-piece French type is used for narrow circular applications requiring a positive unbroken metal face across the full width. The minimum gasket width is gasket

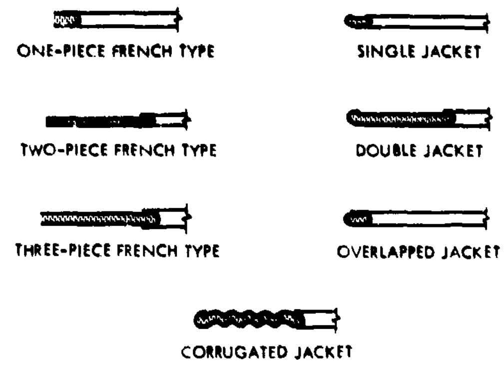


Figure 6.3.3.2c. Metal-Jacketed Gasket Configurations

thickness multiplied by 1.5. The two- or three-piece French types are used for wide or irregular shapes not requiring protection of the filler material or additional flange support at the outer edge. The tooling for these is less costly than for the one-piece type.

The single jacket design is used for relatively narrow applications similar to the French type, but is generally less costly. Noncircular as well as circular shapes can be obtained. When complete protection of the filler material is required, the double jacket configuration should be used. For widths of less than 5/32 inch, the overlapped jacket design should be used; where increased resiliency is required, the corrugated jacketed configuration may be used. This corrugated design may be used for circular and moderately noncircular shapes in widths one-half inch and wider. The corrugations provide redundant seals.

In those configurations where the metal jacket is overlapped (Figure 6.3.3.2d) the inner metal lap should not be specified indented and flush because metal interface flow will be impaired on compression. The inner lap will provide the primary seal and the outer lap, if any, will provide a secondary seal. This type of gasket is normally supplied in standard thicknesses of 1/16, 3/32, and 1/8 inch and requires 20 to 30 percent compression. The maximum width of the lap that can be formed without wrinkling or cracking is limited by the metal thickness, gasket thickness, and gasket diameter. Table 6.3.3.2e presents typical values.

Table 6.3.3.2e. Metal Thickness for Jacketed Gaskets
(Adapted from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Gasket Inside Diameter (in.)	Sheet Metal Thickness (in.)
Up to 4-1/2	0.008 - 0.012
4-1/2 to 8	0.010 - 0.019
Over 8	0.015 - 0.025

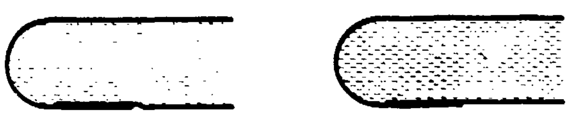


Figure 6.3.3.2d. Lap Configuration

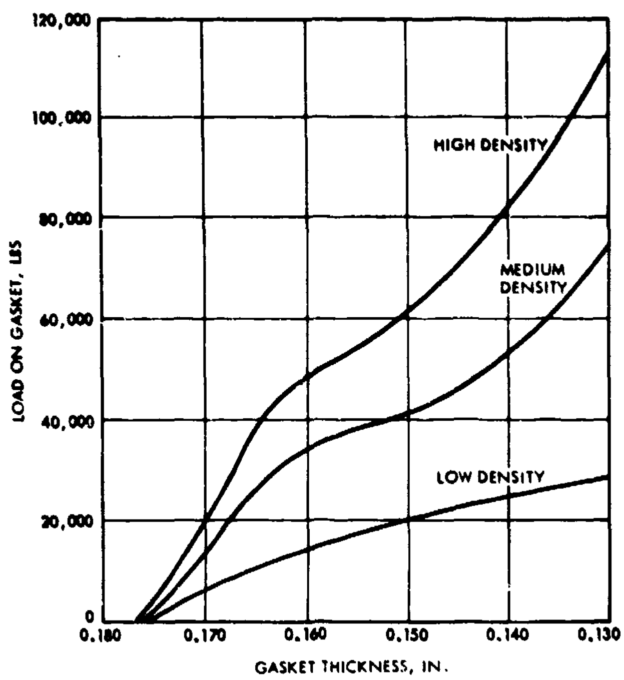


Figure 6.3.3.2e. Typical Compression Curves for 0.175-Inch Thick Spiral-Wound Gasket

(Reprinted with permission from "Johns-Manville Metallic Gasket Catalog," Johns-Manville, New York 16, New York)

Spiral-wound gaskets consist of V-shaped, preformed plies of metal, wound in a spiral with a soft material separation such as asbestos paper or Teflon. For temperatures above 1000°F, ceramic fiber fillers have been used. Because of the V-shape, these gaskets have the best resilience of the combination material gaskets. Density of construction (ratio of metal-to-filler) may be varied to control compressibility for specific gland loadings (Figure 6.3.3.2e). Spiral-wound gaskets work best when compressed to a specific thickness. For 1/8-inch thick gaskets, a compressed thickness of 0.100 ±0.005 and for 3/16-inch thick gaskets a compressed thickness of 0.130 ±0.005 is preferred. Gland design must assure that both the inner and outer layers of metal plies are under compression. This gasket is limited to circular or moderately oval shapes where the major axis is not more than twice the minor axis.

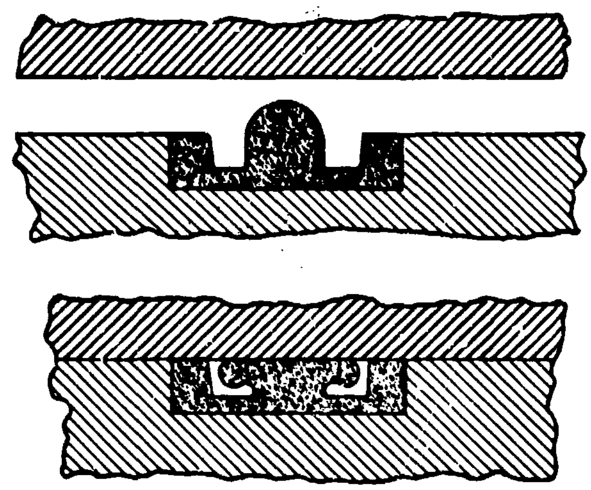


Figure 6.3.3.2f. Operation of Molded Shape Gasket

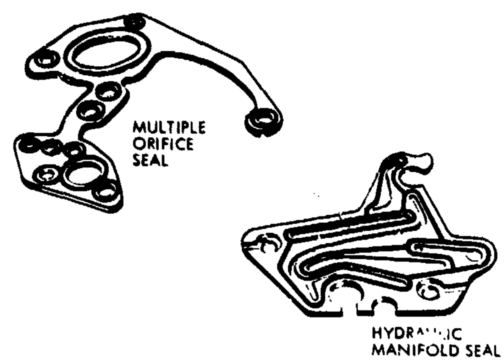


Figure 6.3.3.2g. Typical Configurations Available with Molded Shape Gaskets

(Courtesy of Parker Seal Company, Culver City, California)

Molded shape gaskets are specially manufactured gasket seals designed to provide some of the advantages of O-ring sealing in applications where standard O-rings cannot be used. Such a design is an elastomer molded into a grooved metal base, normally aluminum. On installation, the elastomer is deformed to fill the groove and provide a seal load on the mating flange, as shown in Figure 6.3.3.2f. The elastomer seal is fully confined between the gland flange and the grooved metal base. The unit seal load will be determined by the seal design and usually will be on the order of 40 lbs/inch of seal length. The advantage of this type of seal is that there is practically no limit to the complexity of the seal shape outline (Figure 6.3.3.2g). Gland design should be sufficient to assure a maximum of 0.002-inch flange separation during operation. Typical dimensional space requirements for this gasket are shown in Figure 6.3.3.2h. A design variation which provides flexibility for conforming to a curved surface is shown in Figure 6.3.3.2i.

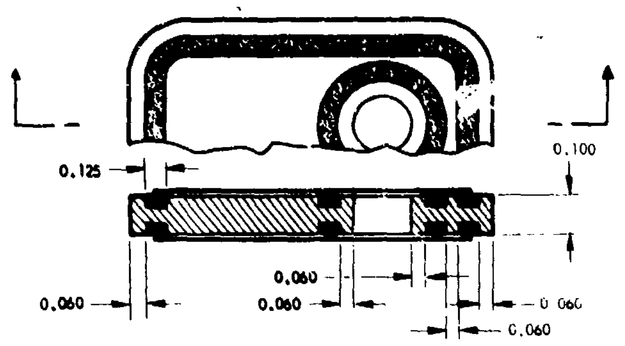


Figure 6.3.3.2h. Dimensional Requirements for Molded Shape Gasket

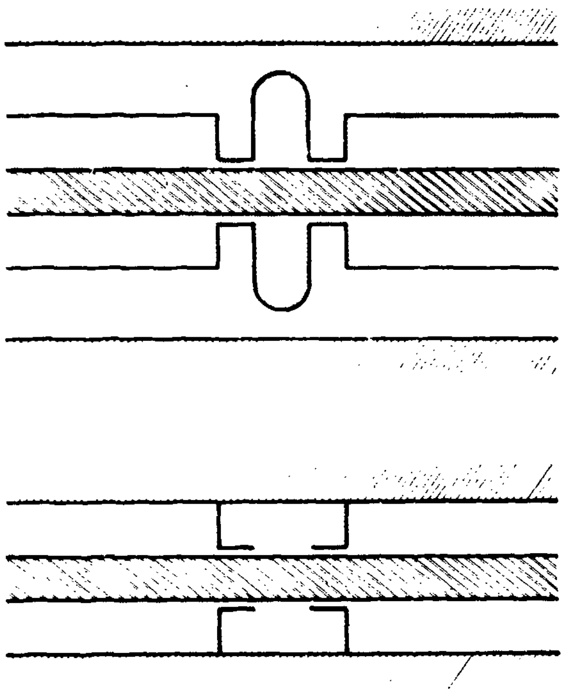


Figure 6.3.3.2i. Flexible Molded Gasket
(Courtesy of Stillman Rubber Products, North Olmstead, Ohio)

6.3.3.3 NONMETALLIC O-RINGS. A nonmetallic O-ring may function either as a mechanically preloaded seal or as a pressure-energized seal, as illustrated in Figure 6.3.3.3a. Because of its wide usage, its application has become standardized as to sizes and installation requirements; in fact, many of the all-metal pressure-energized seals are designed for use in standard O-ring grooves. The advantages offered by the O-ring as a seal include:

a) Low initial cost
b) Adaptability to limited space
c) Easy installation
d) Sealing in both directions
e) High resiliency.

Material compatibility is treated in Sub-Section 12.5 and in SAE AIR 786, "Elastomer Compatibility Considerations Relative to O-Ring and Sealant Selection."

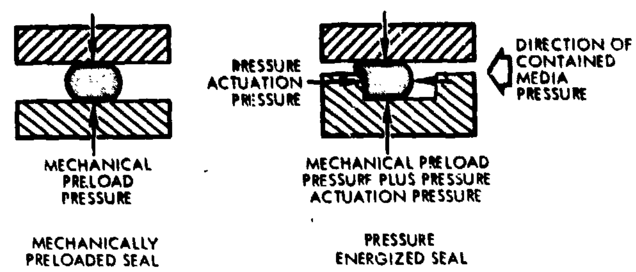


Figure 6.3.3.3a. O-Rings as Mechanically Preloaded and Pressure-Energized Seals

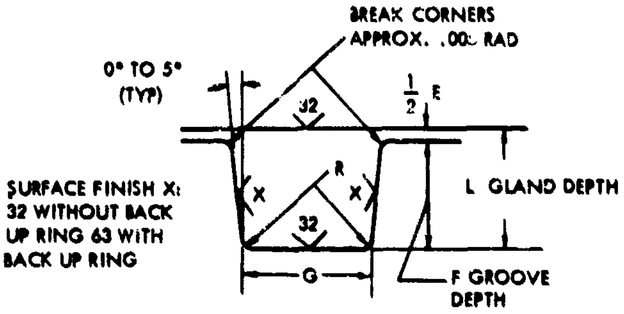
GROOVE DESIGN—ELASTOMER O-RINGS. An elastomer O-ring can be installed in any shape gland; however, the rectangular groove shape provides a more even distribution of stresses in the O-ring and is the preferred design. Dimensions for standard O-ring grooves are shown in Table 6.3.3.3a. The width of the groove is designed to allow for a 15 percent volume swell of the O-ring material. Under zero pressure differential, the flat sealing surface caused by diametrical squeeze is 40 to 45 percent of the O-ring cross section. As the differential pressure increases, the O-ring deforms and the contact width increases to 70 to 80 percent of the original O-ring cross section. The groove design should provide for an installed stretch of the O-ring inside diameter of not more than 5 percent. Where extremely long service life is the most important consideration, installed stretch should be limited to 2 percent. Excessive stretch will result in loss of compression (Figure 6.3.3.3b).

Radial clearances should never exceed one half of the minimum squeeze, even when pressures do not require close fit. Under excessive clearance, loss of sealing contact is possible (Figure 6.3.3.3c).

In some flange joints, the use of a triangular groove may be used to reduce costs. Typical dimensions are shown in Table 6.3.3.3b.

Table 6.3.3.3a. Standard Static Seal Elastomer O-Ring Groove Dimensions

(Reprinted with permission from "Parker O-Ring Handbook," Cat. 5700, November 1964, Parker Seal Company, Culver City, California)



GLAND DETAIL

W Cross Section (in.)		L Gland Depth (in.)	Squeeze		E Diametral Clearance (in.)		G Groove Width (in.)		R Groove Radius (in.)	Eccentricity Max. (b) (in.)
Nominal	Actual		Actual (in.)	%	800 psi Max. (a)(c)	1500 psi Max. (a)(c)	Fuel and Engine Oil Uses	Vacuum and Gases		
1/16	0.070 ±0.003	0.050 to 0.054	0.013 to 0.023	19 to 32	0.002 to 0.010	0.002 to 0.005	0.401 to 0.107	0.082 to 0.088	0.005 to 0.015	0.002
3/22	0.103 ±0.003	0.074 to 0.080	0.020 to 0.032	20 to 30	0.002 to 0.010	0.002 to 0.005	0.136 to 0.142	0.117 to 0.123	0.005 to 0.015	0.002
1/8	0.139 ±0.004	0.101 to 0.107	0.028 to 0.042	20 to 30	0.003 to 0.011	0.003 to 0.006	0.177 to 0.187	0.157 to 0.163	0.010 to 0.025	0.003
3/16	0.210 ±0.005	0.152 to 0.162	0.043 to 0.063	21 to 30	0.003 to 0.011	0.003 to 0.006	0.270 to 0.290	0.247 to 0.253	0.020 to 0.035	0.004
1/4	0.275 ±0.006	0.201 to 0.211	0.058 to 0.080	21 to 29	0.004 to 0.012	0.004 to 0.007	0.342 to 0.362	0.322 to 0.328	0.020 to 0.035	0.005

(a) Clearance gap must be held to a minimum consistent with design requirements for temperature range variation

(b) Total indicator reading between groove and adjacent bearing surface

(c) Reduce maximum clearance 50% when using silicone O-rings

(d) For face seals, E = O, therefore, F = L

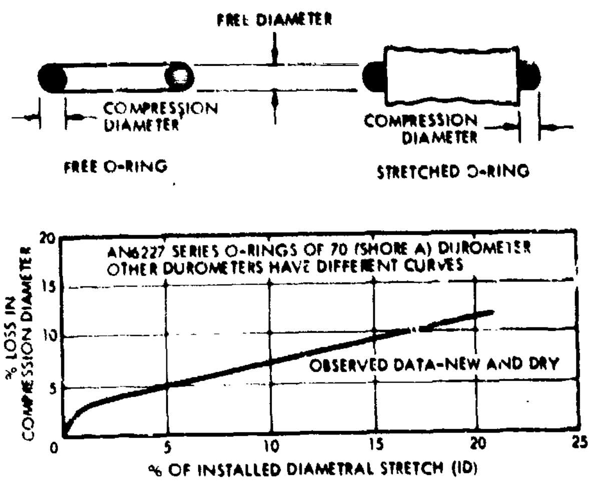


Figure 6.3.3.3b. Loss in Compression Diameter Due to Stretch

(Adapted from "Parker O-Ring Handbook," Cat. 5700, November 1964, Copyright 1964, Parker Seal Company, Culver City, California)

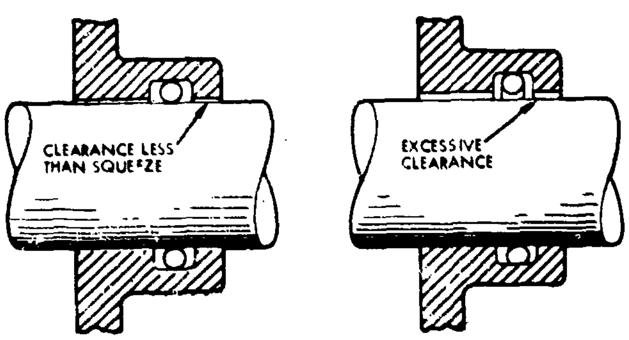
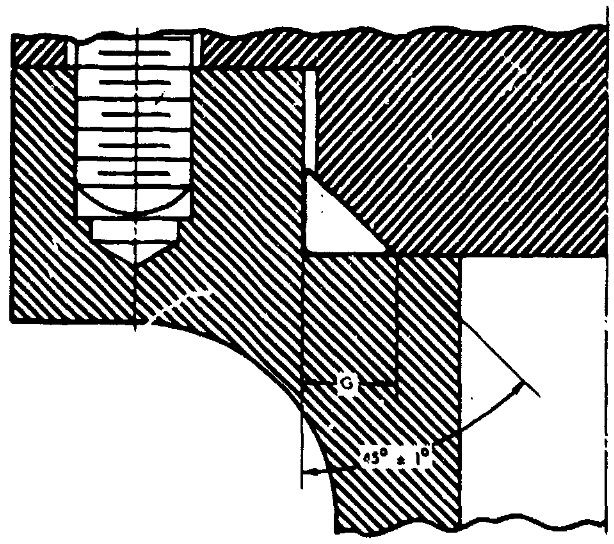


Figure 6.3.3.3c. Loss of Seal Due to Excessive Clearance

(Reprinted with permission from "Product Engineering," September 1951, vol. 22, no. 9, Copyright 1951, McGraw-Hill Publishing Co., New York 36, New York)

Table 6.3.3.3b. Gland Dimensions for Triangular Grooves

(Reprinted with permission from "Precision Rubber Engineering Handbook," 8th Ed., Copyright 1963, Precision Rubber Products, Dayton, Ohio)



O-Ring Cross Section		G Gland Depth
Nominal	Actual	
1/16	0.070 ±0.003	0.095 + 0.003 - 0.000
3/32	0.103 ± 0.003	0.137 + 0.005 - 0.000
1/8	0.139 ± 0.004	0.186 + 0.007 - 0.000
3/16	0.210 ± 0.005	0.279 + 0.010 - 0.000
1/4	0.275 ± 0.006	0.371 + 0.015 - 0.000

Installation Requirements. An O-ring groove design can apply compression on the inside diameter, outside diameter, or axially, as illustrated in Figure 6.3.3.3d. Expansion of the inside diameter to reach the groove should not exceed 25 percent. If this amount is exceeded, sufficient time must be allowed for the O-ring to return to its normal diameter before closing the groove. Closure of the O-ring gland should not pinch the O-ring at the groove corners. Typical designs to preclude pinch are shown in Figure 6.3.3.3e. To aid in installing an O-ring in the groove, a thin coating of lubricant, usually silicone grease, may be applied to the O-ring.

Anti-Extrusion Rings. The extent of extrusion of the O-ring into the clearance gap depends on the differential pressure, the hardness of the O-ring, and the maximum diametral clearance considering thermal expansion of the metals. To prevent extrusion of the O-ring, thin backup rings of metal, leather, or plastic (usually Teflon or nylon) can be added to the groove. If a single backup ring is used it must be installed on the low pressure side of the seal. Where the O-ring may be required to seal against pressure in both directions two backup rings should be used. To prevent the possibility of incorrect installation, the use of two backup rings should be considered in both cases. Installation of backup rings requires an increase in the groove width, as shown in Table 6.3.3.3c. Refer to Figure 6.3.3.3f to determine if the use of backup rings is required.

Plastic O-Rings. The most common plastic O-ring materials, Teflon and Kel-F, are used primarily for applica-

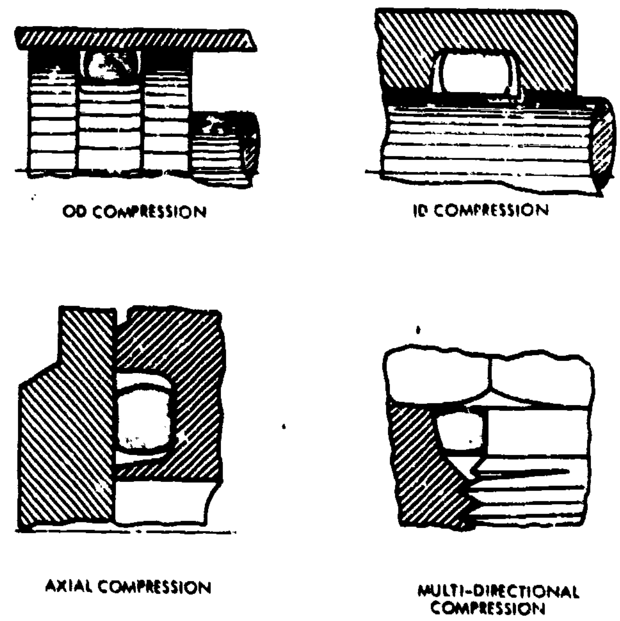


Figure 6.3.3.3d. O-Ring Installation Requirements

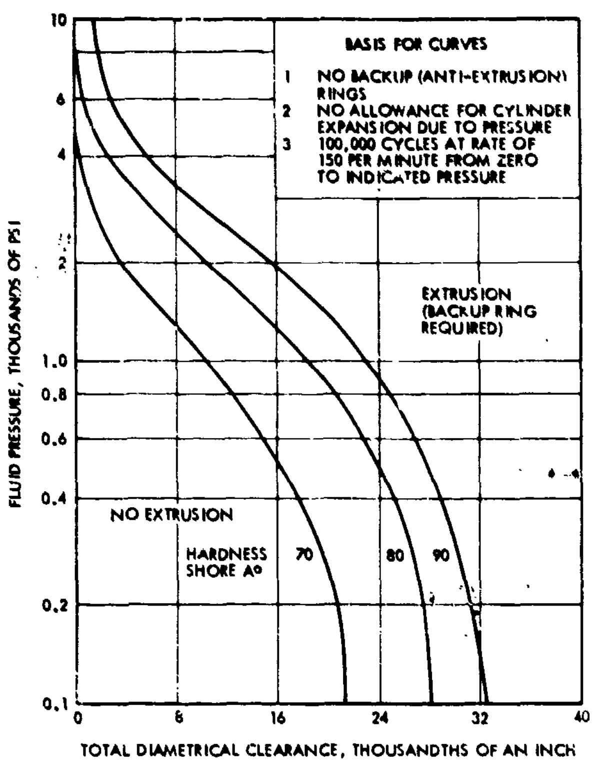


Figure 6.3.3.3f. Requirements for Backup Rings
(Reprinted with permission from "Parker O-Ring Handbook," Cat. 5700, November 1964, Copyright 1964, Parker Seal Company, Culver City, California)

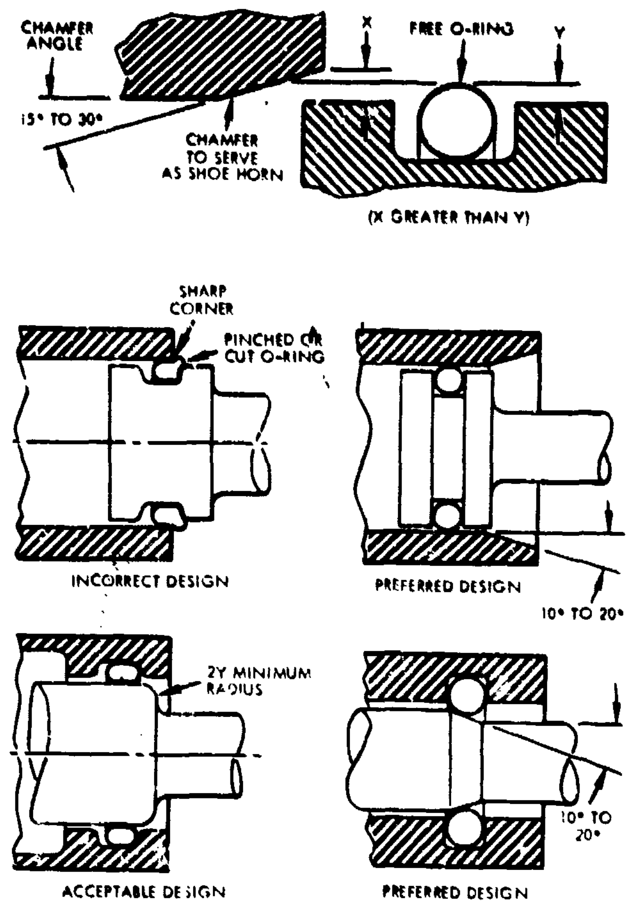


Figure 6.3.3.3e. Typical Static O-Ring Gland Designs
(Courtesy of Seals Eastern Inc., Red Bank, New Jersey)

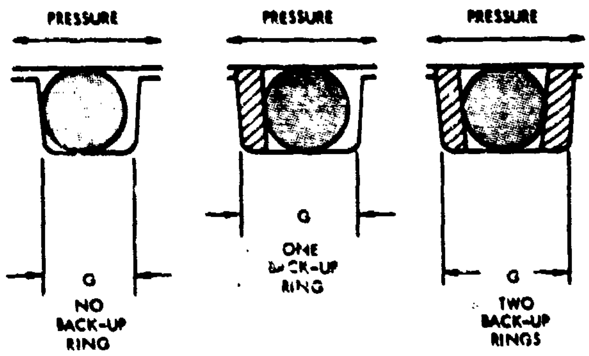
tions where elastomers are not compatible with the sealed media. The groove dimensions for elastomer O-rings can be used for Teflon O-rings, although experience has shown that effective sealing has been achieved with squeeze reduced to as low as 5 percent.

6.3.3.4 HOLLOW METALLIC O-RINGS. The primary advantages of metallic O-rings over polymeric O-rings are: (1) non-extrusion at high pressures, (2) no deterioration during vacuum conditions, and (3) use at elevated temperatures. Three basic types are available: plain, self-energizing, and pressure-filled (Figure 6.3.3.4a).

Plain Metallic O-Rings. These are used for fully confined or semi-confined designs. When used in fully-confined ring joints, standard metallic O-rings are useful at temperatures from −420°F to +800°F and pressures up to 1000 psi. Standard metallic O-rings will not seal in semi-confined designs to the same high temperatures and pressures as a pressure-filled O-ring, but they are more economical.

Table 6.3.3.3c. Groove Width Change for Backup Ring Installation (Refer to Parker Handbook OR5700 and Compound Manual C5702 for complete installation data)

(Adapted with permission from "Parker O-Ring Handbook," Cat. 5700, November 1964, Parker Seal Company, Culver City, California)



W Cross Section		G Groove Width		
Nominal	Actual	No Back-Up Rings	One Back-Up Ring	Two Back-Up Rings
1/16	0.070 ±0.003	0.101 to 0.107	0.138 to 0.143	0.205 to 0.210
3/32	0.103 ±0.003	0.136 to 0.142	0.171 to 0.176	0.238 to 0.243
1/8	0.139 ±0.004	0.177 to 0.187	0.208 to 0.213	0.275 to 0.280
3/16	0.210 ±0.005	0.270 to 0.290	0.311 to 0.316	0.410 to 0.415
1/4	0.275 ±0.006	0.342 to 0.362	0.408 to 0.413	0.538 to 0.543

Self-Energizing Metallic O-Rings. These are used exclusively for semi-confined designs. The inner periphery is vented by small holes, therefore pressure inside the ring is the same as the pressure in the system. Since sealing occurs at two upper and lower points, increasing the internal pressure increases the sealing load. This configuration is useful at pressures above 1000 psi.

Pressure-Filled Metallic O-Rings. These are used for fully confined or semi-confined designs. The ring is filled with an inert gas, usually at about 600 psi. At elevated temperatures the gas pressure increases, offsetting the inherent loss of strength in the tubing at high temperatures, and actually increasing the resilience. This ring cannot support the pressures that the self-energizing ring can endure, but it is useful in the temperature range above 800 F.

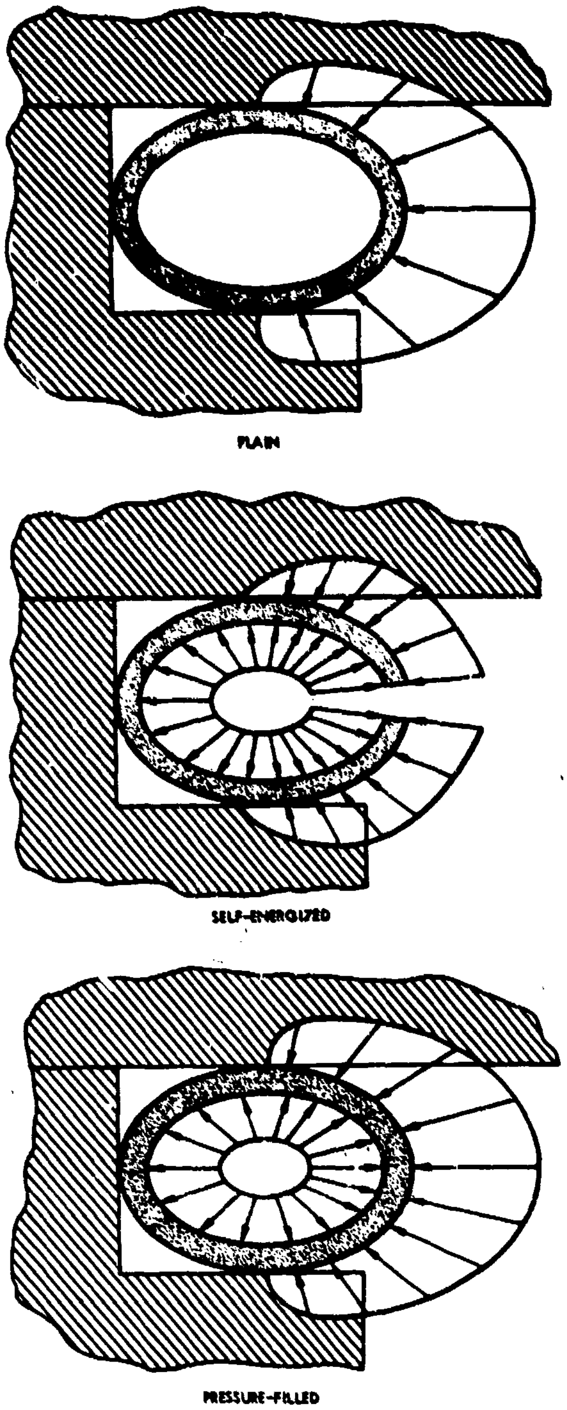


Figure 6.3.3.4a. Types of Hollow Metallic O-Rings
(Courtesy of United Aircraft Products, Inc., Anaheim, California)

Typical groove dimensions for round cross-section O-rings are shown in Table 6.3.3.4a. The semi-confined groove is the most common, although fully confined grooves may be used (Figure 6.3.3.4b). Sealing is accomplished on the two flattened faces. Applications for sealing on the inside or outside diameter require extremely critical gland design and are not recommended. Wall thickness of the tubing will depend on the fluid and pressure to be sealed. Liquids may be sealed with thin-walled tubes, but gases will usually require heavy-wall rings. It is suggested that the smallest cross section available be used in the outside diameter when sealing pressure is in excess of 1000 psi; for maximum resilience and minimum flange loading the largest cross section should be used. Heavy-walled tubing should be used for sealing extremely high pressures. Standard wall thicknesses and tubing outside diameters are shown in Table 6.3.3.4b. O-rings may be obtained with any of the coatings shown in Table 6.3.2.2. Standard coatings include Teflon for service up to 450°F, and silverplate for elevated temperature service. The squeeze for sealing will vary from 20 to 30 percent of the O-ring cross section. The seal load required will vary with material and wall thickness, as shown in Table 6.3.3.4c. Normally, surface finish requirements are 16 microinches (AA), with all tool marks circumferential, when sealing liquids, and 8 microinches when sealing gases.

6.3.3.5 FLEXIBLE METALLIC SEALS. To provide resiliency in a seal in environments beyond the usable range of elastomers, a group of flexible metallic seals has been developed. These seals form the major part of the pressure-actuated class of seals. As illustrated in Table 6.3.3.1, a wide variety of seal shapes are available, each one a specialty for a particular seal manufacturer.

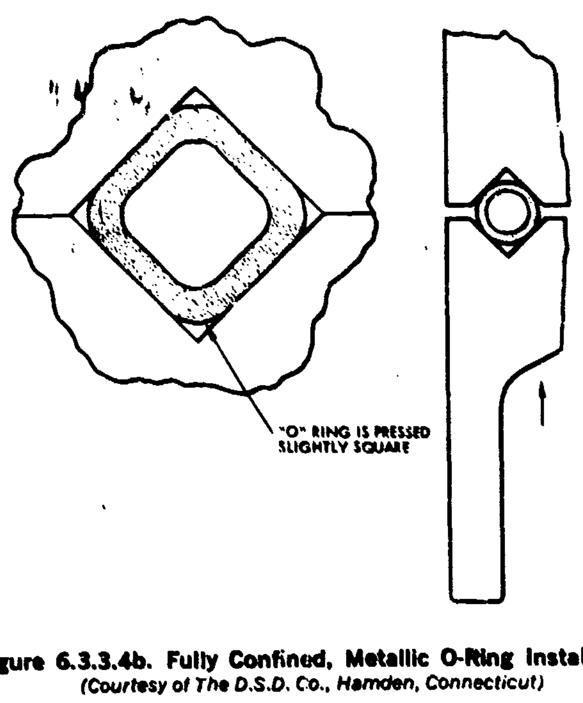
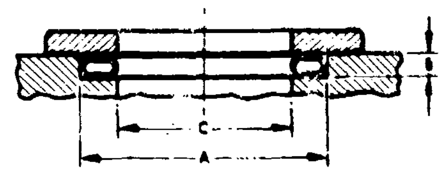


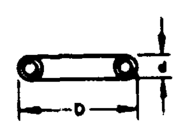
Figure 6.3.3.4b. Fully Confined, Metallic O-Ring Installation
(Courtesy of The D.S.D. Co., Hamden, Connecticut)

Table 6.3.3.4a. Metallic O-Ring Groove Dimensions
(Adapted from "Metal Seals Specification Manual," The Advanced Products Company, North Haven, Connecticut)



Groove Dimensions		
A Diameter*	B Diameter	C Diameter
Ring OD + 0.005/0.009	0.023 0.027	Ring OD - 0.90
+ 0.005/0.010	0.042/0.047	- 0.150
+ 0.005/0.010	0.074 0.079	- 0.220
+ 0.005 0.010	0.105 0.110	- 0.290
+ 0.005 0.010	0.125 0.130	- 0.375
+ 0.008/0.016	0.150 0.155	- 0.450
+ 0.008/0.016	0.200 0.205	- 0.600
+ 0.010/0.020	0.300 0.305	- 0.900
+ 0.010 0.020	0.400 0.405	- 1.200

*For standard plating or coating thickness (0.001 0.0015) add 0.005 inch to A diameters. For other plating or coating thicknesses, add 2 × maximum thickness to A diameters.



Nominal O-Ring Dimensions		
Tubing Diameter	O-Ring Diameter	D Diameter (Unplated)
0.035	0.032 0.041	Ring OD + 0.000 0.005
0.062	0.059 0.070	+ 0.000 0.005
0.094	0.092 0.104	+ 0.000 0.005
0.125	0.123 0.135	+ 0.000 0.005
0.156	0.132 0.157	+ 0.000 0.005
0.188	0.134 0.000	+ 0.000 0.007
0.250	0.245 0.265	+ 0.000 0.008
0.375	0.376 0.348	+ 0.000 0.009
0.500	0.495 0.503	+ 0.000 0.010

Table 6.3.3.4b. Standard Dimensions of Hollow Metallic O-Rings

Tubing OD (in.)	Ring OD (in.)	Wall Thickness (in.) Thin-Wall Rings	Heavy-Wall Rings
1/32	0.25 to 3.00	0.005	0.010
1/16	0.50 to 6.00	0.010	0.012
3/32	1.00 to 20.00	0.010	0.018
1/8	2.00 to 30.00	0.010	0.020
5/32	4.00 to 50.00	0.016	0.025
3/16	6.00 to *	0.020	0.032
1/4	9.00 to *	0.020	0.035
3/8	12.00 to *	0.035	0.049

* To any reasonable diameter. Parts have been fabricated up to 14 feet in diameter.

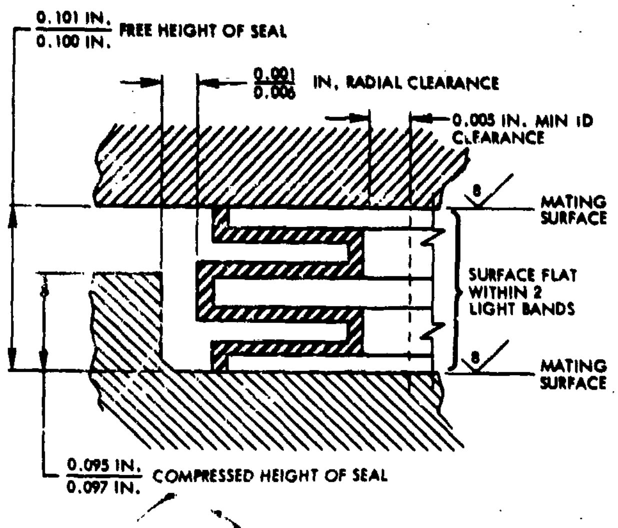


Figure 6.3.3.5a. Typical Flexible Metallic Seal Installation Requirements

(Courtesy of Hydrodyne Division, Donaldson Co. Inc., North Hollywood, California)

Recommended installation requirements will vary with the manufacturer (Figure 6.3.3.5a), but all seals must have some means of limiting the installed deflection of the flexible leg. In some cases, such as the K-seal and the Haskel seal, this limit is provided by the seal body. In other designs the capability must be added to the gland design, as shown in Figure 6.3.3.5b. As is typical with metallic seals, use is limited to face sealing.

The usable temperature range for this type of seal will be determined by the coating applied (Table 6.3.2.2).

A detailed analysis of the stresses encountered in flexible metallic seals is given in References 46-25, 152-2, and 152-4.

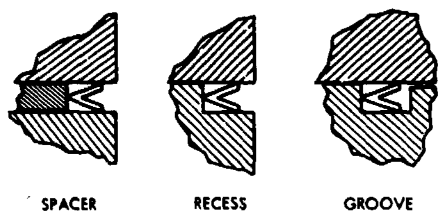


Figure 6.3.3.5b. Gland Design for Limiting Initial Seal Deflection

6.3.3.6 PLASTIC SPRING-LOADED SEALS. Plastic spring-loaded seals were developed to provide improved resiliency and, in some instances, pressure energization when sealing with plastic materials. They compliment the flexible metallic seals and are particularly useful within the temperature and compatibility limits of plastic materials. The basic construction consists of a plastic (usually Teflon) cover over a metallic spring core. The spring core, in addition to providing initial seal load at zero pressure, tends to compensate for dimensional changes resulting from tolerance variations, thermal expansion or contraction, and cold flow of the plastic. A list of available seals in this category is included in Table 6.3.3.1.

In axial sealing applications, designs are available which will function satisfactorily in a standard O-ring groove. For applications where the seal interface is radial, consideration must be given to the installation of the seal; stretch must be limited to prevent damage to the spring. Where possible, a split gland design is preferred. Typical gland designs are shown in Figure 6.3.3.6.

6.3.3.7 RADIAL OR TOGGLE METALLIC SEALS. This class of seals is a special configuration in which axial load is converted to a radial interface load through a toggle action within the seal structure. The seal gland is designed to confine the outer rim of the seal in such a way that radial interference occurs upon application of an axial load during assembly. Due to the critical relationship between the seal and gland, the gland design for seals in this category are rigidly controlled by the seal manufacturer and in many cases are considered proprietary. Typical examples of radial metallic seals are shown in Figure 6.3.3.7a.

One advantage of the radial metallic seal is the effect of force magnification to provide the seal load required for

Table 6.3.3.4c. Average Natural Resiliency and Seating Force of Metal O-Rings

(Reprinted with permission from "Product Engineering," November 1962, vol. 33, no. 26, J. N. Andrews, McGraw-Hill Publishing Company, New York 36, New York)

Basic Tubing OD (in.)	Wall Thickness (in.)	Tubing Material	Elastic* Spring Back (in.)	Unit Seal Load of O-Ring (lb per linear in.)
1/32	0.005	Stainless steel	0.002	300
	0.010	Stainless steel	0.002	400
	0.012	Stainless steel	0.002	800
	0.005	Inconel	0.0015	200
	0.005	Inconel X	0.001	300
1/16	0.005	Stainless steel	0.003	200
	0.010	Stainless steel	0.002	500
	0.012	Stainless steel	0.002	800
	0.014	Stainless steel	0.0025	1300
	0.016	Stainless steel	0.003	1500
	0.006	Inconel	0.002	300
	0.010	Inconel	0.002	420
	0.012	Inconel	0.002	550
	0.014	Inconel	0.002	1100
	0.010	Inconel X	0.002	550
	0.012	Inconel X	0.002	700
	0.010	Mild steel	0.002	400
	0.014	Mild steel	0.002	850
	0.010	Aluminum	0.0015	200
	0.014	Aluminum	0.0015	350
	0.010	Monel	0.002	450
	0.010	Copper	0.001	150
	0.012	Copper	0.001	250
	0.014	Copper	0.001	350
3/32	0.007	Stainless steel	0.002	200
	0.010	Stainless steel	0.002	350
	0.012	Stainless steel	0.002	425
	0.018	Stainless steel	0.0035	1100
	0.007	Inconel	0.0025	150
	0.010	Inconel	0.0025	250
	0.012	Inconel	0.002	350
	0.018	Inconel	0.0025	1000
	0.010	Inconel X	0.0025	300
	0.010	Mild steel	0.002	200
	0.012	Mild steel	0.002	250
	0.018	Mild steel	0.002	950
	0.010	Aluminum	0.002	200
	0.012	Aluminum	0.002	175
	0.018	Aluminum	0.002	350
	0.010	Monel	0.002	200
	0.012	Copper annealed	0.001	200
	0.018	Copper hard	0.004	500
	0.012	Tantalum annealed	0.002	650
1/8	0.010	Stainless steel	0.003	210
	0.012	Stainless steel	0.003	320
	0.020	Stainless steel	0.004	1000
	0.010	Inconel	0.004	250
	0.012	Inconel	0.002	300
	0.020	Inconel	0.0035	1000
	0.025	Inconel	0.004	1400
	0.020	Inconel X	0.004	800
	0.025	Inconel X	0.004	1600
	0.010	Mild steel	0.002	250
	0.020	Mild steel	0.002	700
	0.010	Aluminum	0.002	75
	0.012	Aluminum	0.002	100
	0.020	Aluminum	0.002	220
	0.025	Aluminum	0.002	280
	0.010	Monel	0.003	250
	0.018	Copper	0.002	500
	0.030	Copper	0.002	800
5/32	0.010	Stainless steel	0.004	150
	0.012	Stainless steel	0.004	160
	0.016	Stainless steel	0.003	300
	0.025	Stainless steel	0.003	1000
	0.010	Inconel	0.003	150
	0.025	Inconel X	0.002	950
3/16	0.010	Stainless steel	0.005	150
	0.012	Stainless steel	0.004	175
	0.020	Stainless steel	0.004	450
	0.032	Stainless steel	0.005	2300
	0.020	Inconel	0.004	600
1/4	0.010	Stainless steel	0.006	75
	0.012	Stainless steel	0.006	90
	0.020	Stainless steel	0.006	350
	0.035	Stainless steel	0.006	1100
	0.049	Stainless steel	0.007	2500
	0.035	Aluminum	0.003	250
5/16	0.050	Stainless steel	0.005	2000
3/8	0.035	Stainless steel	0.005	500
	0.049	Stainless steel	0.005	1750
1/2	0.080	Stainless steel	0.008	3300
	0.120	Stainless steel	0.007	7600

*Allowable flange separation

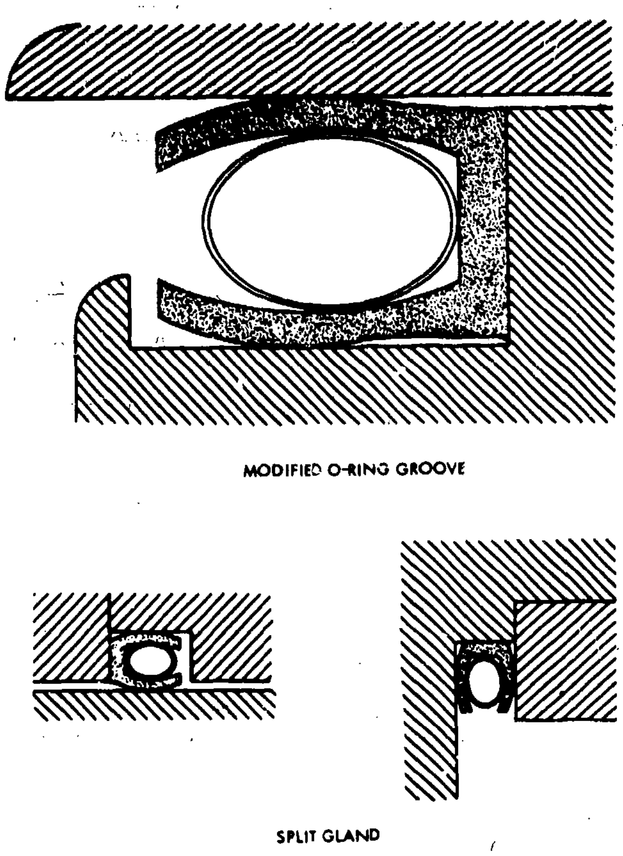


Figure 6.3.3.6. Typical Gland Design for Plastic Spring-Loaded Seals
(Courtesy of Aeroquip Corporation, Aircraft Division, Jackson, Michigan)

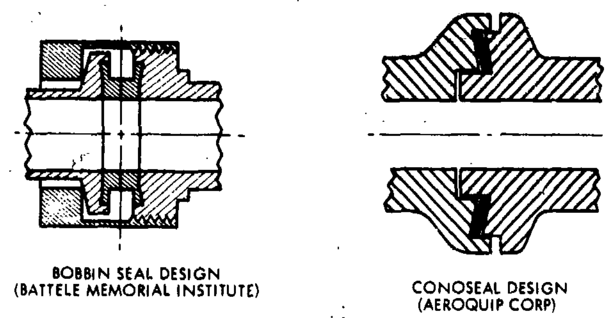


Figure 6.3.3.7a. Toggle or Radial Metallic Seal Gland Designs
(Reference 44-14)
(Courtesy of Aeroquip Corp., Marman Division, Los Angeles 64, California)

metal-to-metal sealing, as illustrated in Figure 6.3.3.7b. Assuming that the seal materials are the same, and that contact areas A_1 and A_2 are equal, then seal load R must approximately be equal to seal load F_2 in order to obtain similar sealing characteristics. However, applied load F_1 is approximately equal to R sin θ. In practical limits, the axial force required to seal a radial metallic seal can be one-fifth the force required to seal the flat gasket.

Various configurations of radial or toggle seals are shown in Table 6.3.3.1.

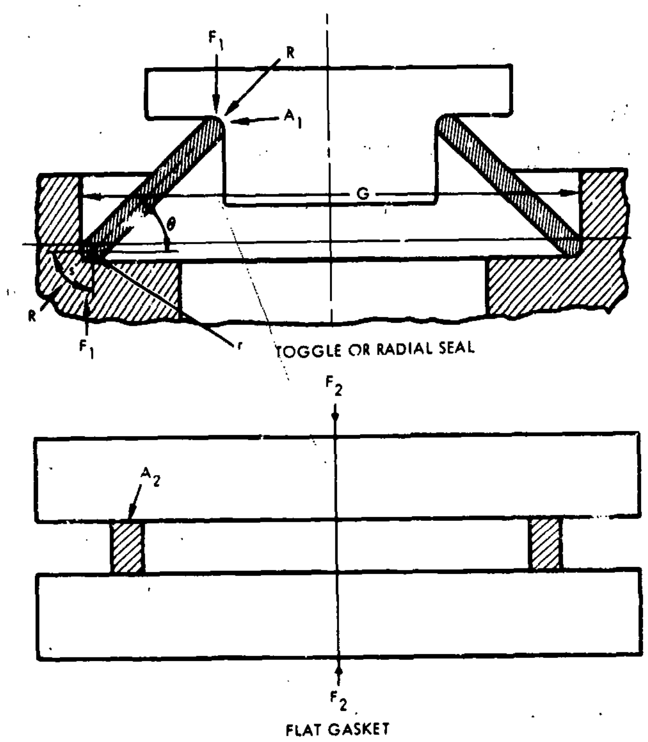


Figure 6.3.3.7b. Force Relationships Between Flat Gasket and Toggle or Radial Metallic Seals
(Reference 44-14)

6.3.3.8 **METALLIC BOSS SEALS.** In order to increase the useful temperature range of the SAE and MS standard fittings for intercomponent tubing connections, a group of metallic seals has been developed which may be used in place of the conventional O-ring. A typical example is shown in Figure 6.3.3.8a. Available seal designs are shown in Table 6.3.3.1.

By modifying the standard gland design, nearly any configuration of the flexible metallic seals may be used, as shown in Figure 6.3.3.8b.

6.3.3.9 **METALLIC SHEAR SEALS.** Metal-to-metal sealing by shear deformation requires that the interface stress impose a strain in shear on the softer of the interfaces. By designing the gland to impose a high stress concentration on the seal through the use of sharp corners or knife edges, a higher level of mating can be accomplished for a given applied force than is achieved with compression stress.

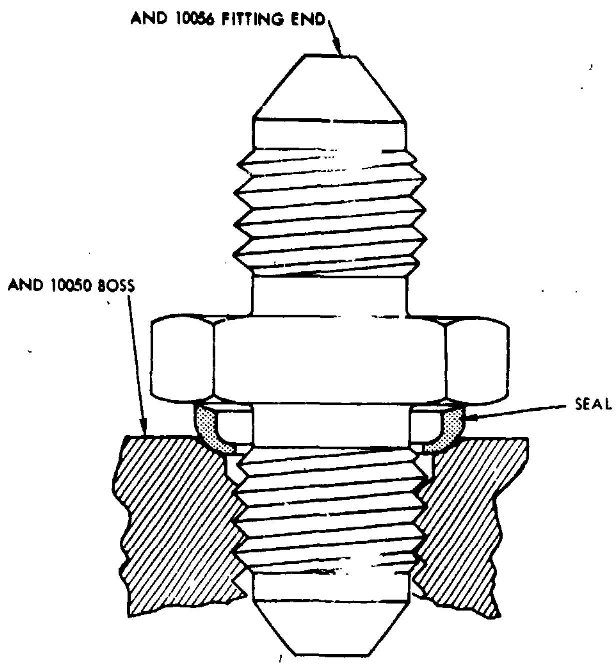


Figure 6.3.3.8a. Metallic Boss Seal
(Courtesy of Navan Products Inc., El Segundo, California)

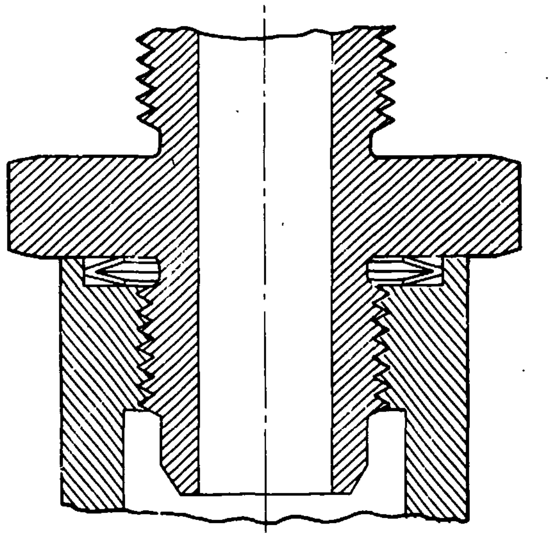


Figure 6.3.3.8b. Standard Boss, Modified for a Metallic V-Seal
(Reference 496-1)

The most common configurations of shear seals are the O-ring and flat gasket. The materials used are limited to soft metals such as copper, aluminum, nickel, etc. In order to minimize damage to the gland, the yield strength of the gland material should be at least three times higher than the seal material.

Typical gland designs are shown in Figure 6.3.3.9. The metal O-ring gland design requires a close control on location to insure proper positioning of the ring, particularly in large diameters. The modified O-ring gland design, which provides a confined parallel-load configuration, also provides positioning control during assembly. Flat gaskets may be used with either a stepped gland or a knife-edge design. By designing the seal and gland so that the sealed pressure loads the seal on the knife edges, pressure energization can be utilized.

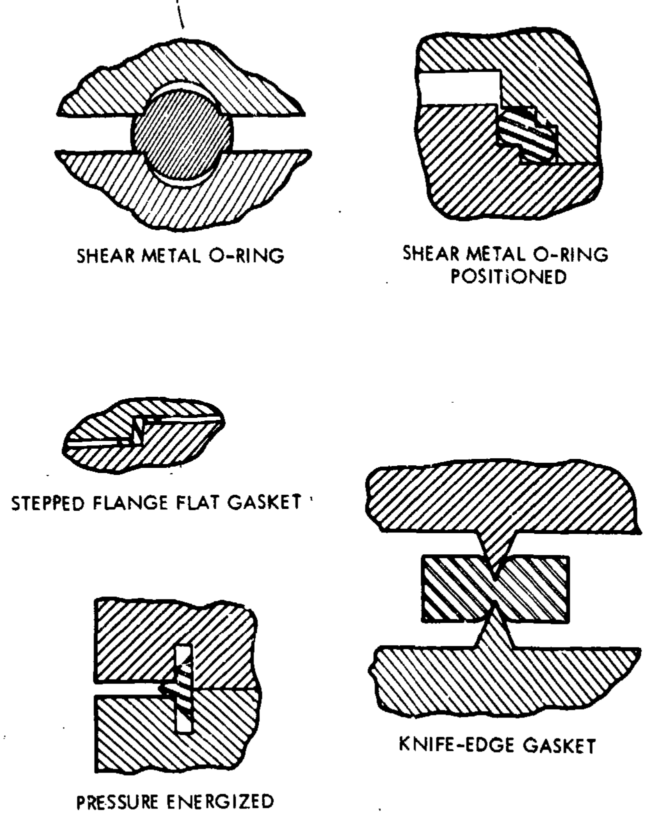


Figure 6.3.3.9. Gland Configurations for Metallic Shear Seals
(References 46-29 and 82-14)

6.3.3.10 CRYOGENIC SEALS. Most of the basic types of seals may be used at cryogenic temperatures; however, special design considerations may be necessary because of the changes encountered in material properties.

Gaskets. The use of soft plastic gaskets has been limited to reinforced or laminated Teflon, while thin gaskets made from hard plastics, such as Mylar, may be used with high stress loading. Soft metals such as copper and aluminum may be used with high shear-stress loading. Typical flange designs used to achieve such loading are shown in Figure 6.3.3.10a. The 3/16-inch radiused ring height is 80 percent of the gasket thickness, and the V-ring height is 60 to 70

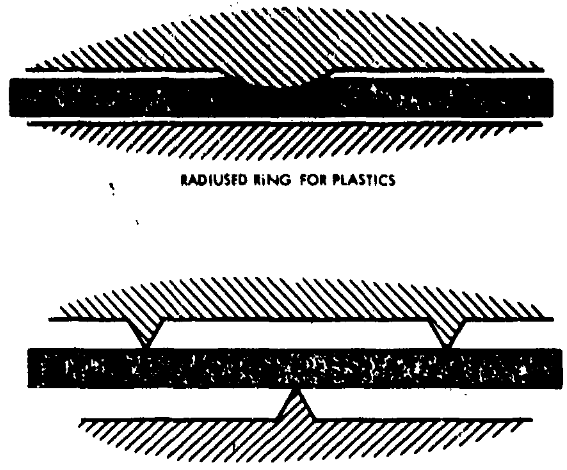


Figure 6.3.3.10a. Gasket Gland Designs for Cryogenic Service
(Reference 82-14)

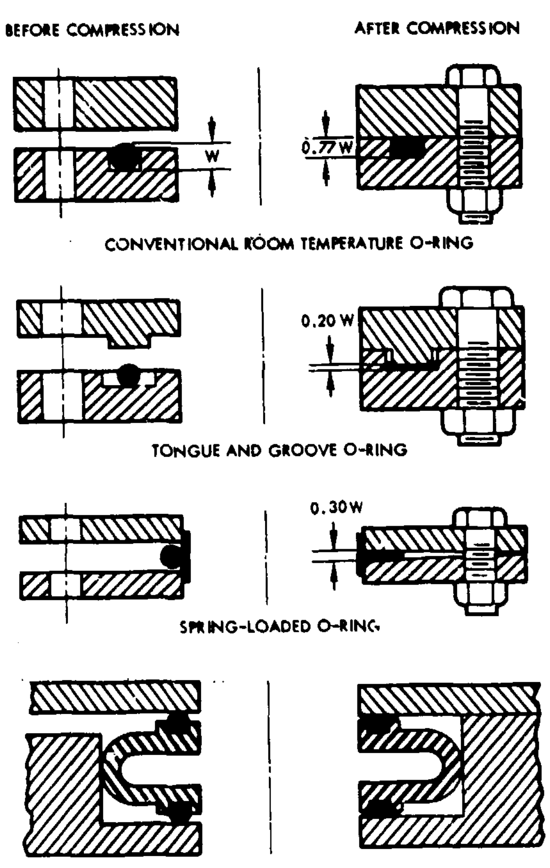


Figure 6.3.3.10b. Elastomeric Designs for Cryogenic Service
(Reference 82-14)

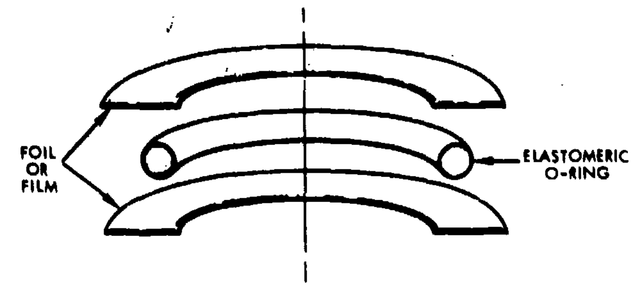


Figure 6.3.3.10c. Sandwich Seal Configuration
(Reference 82-14)

percent of the gasket thickness. Surface finish on the raised portions is critical, i.e., small scratches will probably result in leaks. The conical and spiral-wound gaskets may be used in cryogenic hard-metal gasket applications. A unique glass fabric-Teflon encapsulated composite gasket which exhibits good resilience at cryogenic temperatures is described in Reference 36-12.

Elastomer O-rings. Two installation techniques have been developed for using elastomer O-rings in cryogenic applications. The first is to increase the installed squeeze to 70 to 80 percent of the O-ring cross section (Figure 6.3.3.10b) to overcome the differential expansion problem. The second is to design the tongue and groove joint to provide 5 percent bulk compression, resulting in low-level leakage rates (Reference 82-14). Installing the O-ring between thin foil films, or coating the O-ring with a material such as indium, which remains ductile at cryogenic temperatures, has also proven effective (see Figure 6.3.3.10c).

Metallic O-Rings. Metallic O-rings, either solid or hollow, will function as cryogenic seals. The best results are achieved using soft-metal solid rings such as copper, indium, or lead. Because of the cold-flow tendency of indium, a confined gland design is required.

Flexible Metallic Seals. Any flexible metal seal design, if coated with a suitable material, may be used as a cryogenic seal.

Temperature Actuation. Several cryogenic seal designs have been developed which utilize the difference in thermal expansion of the seal materials to increase seating loads at cryogenic temperatures. The metal elastomer design shown in Figure 6.3.3.10d exploits the very low thermal expansion coefficient of Invar to radially load the elastomer seal as the aluminum or stainless steel flanges expand (Reference 92-2). The bimetallic design shown in Figure 6.3.3.10e utilizes the difference in the thermal expansion coefficients of aluminum and stainless steel to provide the wedge-action force which loads the seal (Reference 126-3). Metallic O-rings may be temperature-actuated by using a restraining ring with a different expansion coefficient to increase the seal load with a change in temperature (Reference 51-5).

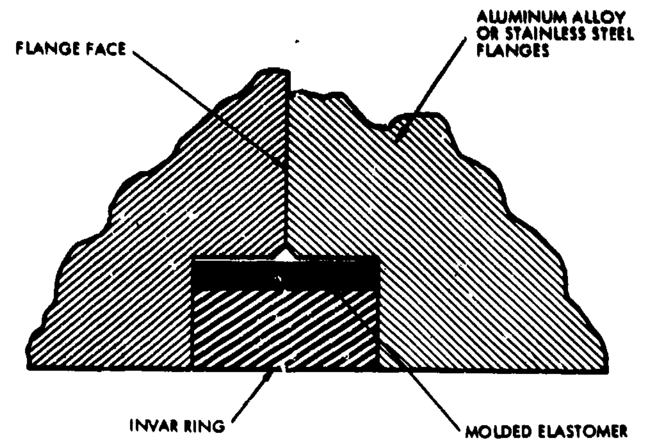


Figure 6.3.3.10d. Composite Temperature-Actuated Seal
(Reprinted with permission from "Jet Propulsion," July 1955, vol. 25, no. 7, S. E. Logan, Copyright 1955, American Institute of Aeronautics and Astronautics, New York, New York)

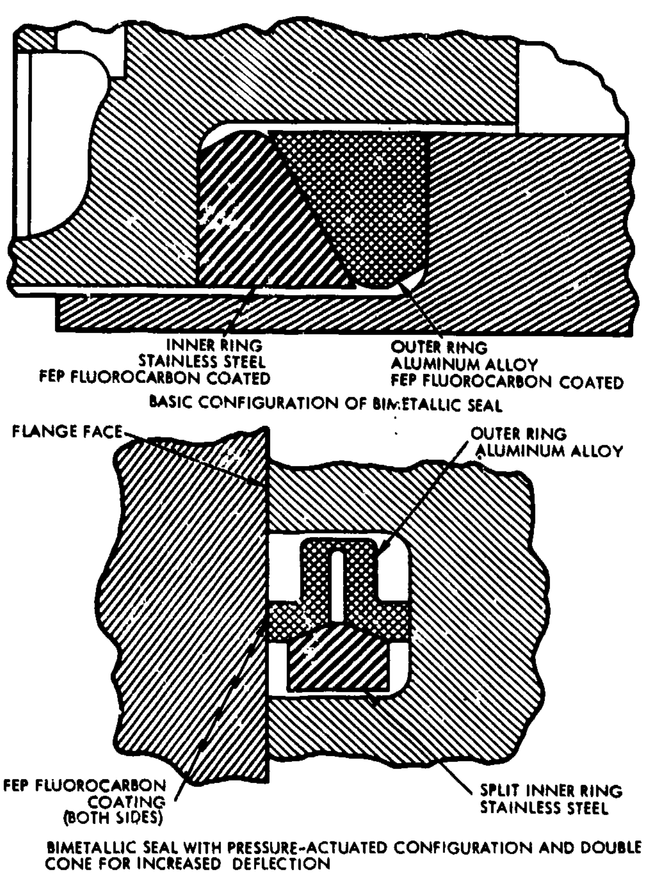


Figure 6.3.3.10e. Bimetallic Temperature-Actuated Seals
(Reference 36-12)

6.3.3.11 VACUUM SEALS. Seals for vacuum service are normally limited to gaskets subjected to shear loading or elastomer O-rings. Many seals for cryogenic application are also required to function as vacuum seals where vacuum insulation techniques are used. The gasket gland designs shown in Figure 6.3.3.10a are typical examples. Flat metal gaskets are usually limited to soft metals with one-half hard copper being the most common. Other metals used include aluminum and indium. Common nonmetallic gaskets are Mylar, Teflon, and Viton.

Elastomer O-Rings. When outgassing or permeation do not prohibit their use, O-rings are an effective vacuum seal; however, the gland design must be altered slightly to minimize the possibility of the O-ring moving after installation. Most elastomers will not reseat readily under a vacuum condition because of a lack of sufficient differential pressure. Also, most elastomers will shrink when exposed to a vacuum because of outgassing; therefore, the gland should be almost completely filled by the O-ring. An excess of O-ring cross section should be limited to 1 percent to avoid extruding the O-ring upon installation. Standard gland dimensions may be used by applying the groove width for vacuum and gases shown in Table 6.3.3.3a.

Very high squeeze can be used with silicone rubber because it does not deteriorate excessively as some other rubbers do under the constant high stress in the vacuum assembly.

When silicone compounds are used in vacuum systems, a hardness of 60 durometer is normally used. However, it is sometimes desirable to use a softer material when extreme sealing difficulties are encountered and the other operational factors (such as high pressure) are not prohibitive.

It is permissible, and sometimes very desirable, to use two or more O-rings in the same or separate grooves for vacuum seals. With multiple O-rings in the same groove, the groove width must be modified accordingly. Problems due to pressure trapping normally do not arise, since the maximum pressure differential is atmospheric and the leakage is compressible gas.

Flexible Metallic Seals. The use of a flexible metal seal requires a reverse seal design to provide differential pressure actuation. The use of this type of seal will be limited by the leakage rate. The differential pressure usually is not sufficient to provide required interface stress levels without the use of Teflon coatings.

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41-7 46-32
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46-23 46-33*

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50-9 V-319
496-1 V-324
V-39 V-325
V-57 V-326
V-70 V-330
V-81 V-344
V-143 V-355
V-180 V-354
V-229 44-24*

Boss Seals
6-191 V-232
151-6 V-326

Vacuum Seals
1-142 46-33*

6.4 DYNAMIC SEALS

6.4.1 Introduction

A dynamic seal is a mechanical device used to minimize, or reduce to an acceptable level, leakage of a fluid from one region to another when there is relative motion between the sealing interfaces. In normal dynamic seal operation, relative motion will exist between a static or stationary interface and a moving interface. The basic types of motion which may occur singly or in combination are:

a) Sliding motion, in which the direction of motion is perpendicular to or across the seal interface. The contact interface is continuously changing location on the gland. If the motion occurs in both directions it is called reciprocating.

b) Rotary motion, in which the direction of motion is along the seal interface. The contact interface is limited to one location on the gland. Where reversal of motion occurs, it is termed oscillating.

The motion may be continuous, but in most instances it will be intermittent with varying periods of time, during which the seal will function in a static condition. A special case of intermittent action, in which the interfaces are separated and remated, occurs in some valve closure seals. Typical applications of valve closure seals are shown in Sub-Section 6.2, "Valving Units."

In addition to meeting the sealing requirements, a dynamic seal must also be designed for acceptable friction and wear at the interface. No escape of the sealed fluid is desirable from a sealing standpoint. However, in order to get reasonable friction and wear characteristics at the seal interface with the current state-of-the-art of dynamic seal design, a small amount of leakage must be tolerated for lubrication. Results of one of the latest studies on dynamic sealing are described in Reference 152-4. In some dynamic seal applications it is possible only to minimize leakage, not prevent it. Friction coefficients for various material combinations are given in Sub-Section 12.7.

If leakage cannot be tolerated, the use of bellows or diaphragms can be used in certain applications to replace conventional dynamic seals. The use of bellows or diaphragms will be controlled by the physical limitations of the design, such as length of stroke required, power available to move the seal, i.e., bellows compression force, and by the envelope available. Application of bellows is described in Sub-Section 6.6 and diaphragms in Sub-Section 6.7.

Dynamic seals considered in this sub-section are limited primarily to seals used in valve applications, including such typical applications as valve actuator seals involving relatively low rotary and linear speeds. These paragraphs do not include a discussion of dynamic seals which are primarily intended for high speed rotary applications, such as pumps and turbines.

A discussion of the application of face or mechanical seals is covered in References 112-12 and 46-27; positive clearance seals such as labryinth and bushing seals are described in References 19-72 and 152-3. The following sub-topics consider factors which influence gland and seal element design and selection, and present data on specific dynamic seal designs.

6.4.2 Dynamic Seal Classification

Common methods of classifying dynamic seals include type of motion, method of loading, material application, and configuration. Some of these classifications are shown in Table 6.4.2.

6.4.3 Design and Selection Factors

Primary factors which must be considered in dynamic seal design and selection involve relative balance between allowable leakage and permissible wear of the seal and gland. Influencing factors include material properties, interface conditions, compatibility with the sealed media, and environmental effects such as operating temperature and pressure.

6.4.3.1 SEALING INTERFACE. The effectiveness of a dynamic seal design is determined by the capability of the seal interfaces to maintain good mating over the seal operating environment. The mechanisms of sealing described in Detailed Topic 6.3.2.1 are applicable to dynamic seals; however, the extent of deformation at the interface will be less. The contact surface stress is

Table 6.4.2. Dynamic Seal Classifications

Type of Motion
- Sliding
- Reciprocating
- Rotating
- Oscillating

Method of Loading
- Mechanically preloaded
- Pressure-energized
- Positive clearance

Method of Sealing
- Radial
- Face

Material
- Metallic
- Nonmetallic
- Combination

Application
- High pressure
- Vacuum
- High temperature
- Cryogenic

Configuration
- O-ring
- Lip
- Mechanical
- Labyrinth
- Bushing
- Piston ring
- Compression packing
- Molded packing

considerably lower than in a comparable static seal, due to load limitations imposed by wear considerations. Therefore, the effect of gross surface irregularities such as waviness, out-of-roundness, etc., may be the dominant factor in establishing leakage paths. The general subject of surface topography is covered in Detailed Topic 6.3.2.1 under "Static Seals," and is applicable also to dynamic seals. In dynamic seal applications where lubrication is required, gland surface finishes finer than 10 microinches have shown insignificant increase in seal life. In fact, a surface finish which is too smooth (in the range of 2 microinches) cannot support a lubrication film, and premature seal failure may occur (Reference 1-292).

In high pressure dynamic seals, abrasion and tearing of the interface materials are primary reasons for seal failure. The rate and amount of abrasion are difficult to predict in dynamic seal performance, because of changing conditions such as lubrication and friction encountered during operation. These conditions change during the life of the seal. Characteristics which influence the rate of wear include surface finish and hardness, and frictional characteristics at the interface.

Gland Design. In a dynamic seal design installation, all structural members which directly or indirectly influence the performance of the seal element are considered part of the gland. For example, the gland of the sliding seal on a piston will include both the piston and the adjacent cylinder. The primary functions of the gland are to contain the seal element and to provide, either wholly or partially, the contact loading at the seal interface necessary to produce the required sealing stress. In positive clearance type dynamic seals, where there is no relative load between the seal interfaces, the purpose and function of the housing is primarily to contain and support the seal.

Whenever possible, the two surfaces of the gland being sealed should not come in contact; bearing areas should be outside the sealing area. When this is not possible, the part of the gland containing the seal should be a soft bearing material which will not score or mark the part of the gland which moves across the seal.

In dynamic seal applications where the seal mates with a shaft, shaft hardness, smoothness, and material are factors that must be considered. The usual recommendation for shaft hardness is a Rockwell C of at least 50. If the shaft is one of the soft metals, brass or aluminum for example, it is advisable to press a hardened steel ring on the shaft to serve as a seal running surface. Hard, dense chrome plate electro-deposited on the shaft will also serve satisfactorily.

Maximum seal efficiency and life are obtained with a finely finished gland surface, usually in the 10 to 20 microinch range. Polished or ground finishes with concentric marks are preferred in rotary applications. Direction of finishing marks and spiral lead are important; when a finish lead is present, it should be in the direction that guides the sealed media inward.

Friction. The seal friction force which must be overcome in actuating the moving member is a function of the coefficient of friction at the seal interface. Prediction of the coefficient of friction to be expected in actual operation is complicated because of its dependence upon several operating parameters, including rate of movement, the time at rest prior to movement (which influences breakout friction), operating temperature, reaction sensitivity of the surfaces to the contained media, surface geometries, and the amount of wear to be expected.

One of the major factors influencing friction at the seal interface and wear life of the seal components is the relative surface finish of the interfaces. The machinability of metallic seal materials therefore becomes a matter for consideration. Relative surface conditions which can be expected from various manufacturing operations are shown in Table 6.3.2.1c. Other material properties which influence the friction coefficient of seal materials are the relative surface hardness and type of surface film on the seal interfaces. In general, the softer metals exhibit a higher coefficient of friction.

When motion is intermittent, the amount of time the seal is at rest and functioning as a static seal will have a significant effect on the degree of mating and level of friction loads, particularly in plastics and elastomers. As shown in Figure 6.4.3.1, the levels of static (breakout) friction may be three times that of dynamic (running) friction. Friction load may be decreased by decreasing unit load, differential pressure, surface finish (of metals), or durometer (of elastomers), and by increasing hardness (of metals) or running speed, as well as by the use of lubrication. Typical values of static and dynamic friction coefficients are given in Sub-Section 12.7.

The magnitude of frictional load opposing movement of the seal may be expressed as:

$$F = \mu(w_1 s + w_2 P)$$

where

- F = Frictional load, lbs/linear inch of seal contact
- μ = Coefficient of friction
- w_1 = Width of seal in contact as installed, in.
- w_2 = Width of seal exposed to pressure energizing, in.
- s = Installed seat stress, psi
- P = Operating differential pressure across seal, psi

Wear. Wear generally takes two basic forms:

a) The removal of solid materials from rubbing surfaces; i.e., the loose wear particles which are formed when they leave the sliding system. The particles generated by this type of wear are a major source of contamination of the sealed media.

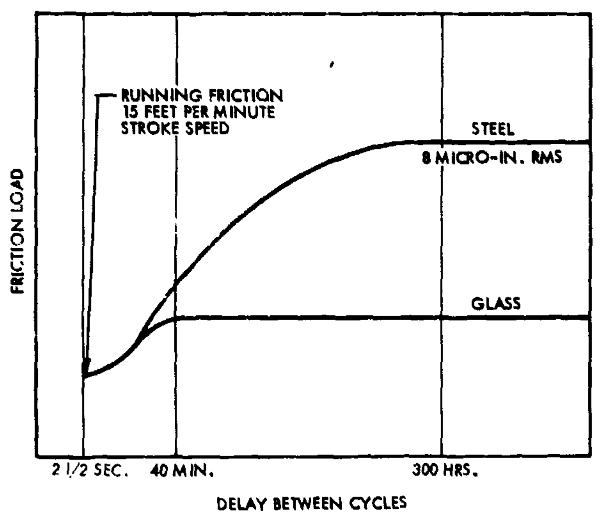

Figure 6.4.3.1. Change of Breakout Friction with Time at Rest for an Elastomer O-Ring

(Reprinted with permission from "Parker O-Ring Handbook," Cat. 5700, 1964, Parker Seal Company, Culver City, California, and Cleveland, Ohio)

b) Solid materials removed from one sliding surface and transferred to the other.

The latter type of wear may be acceptable if the material transfer results in smooth sliding surfaces despite the fact that transfer of small particles occurs. For specific applications, meaningful numbers of frictional resistance and wear can best be obtained by testing an assembled seal in the environment expected in use. A detailed discussion of the theory of the wear process may be found in References 494-1 and 495-1.

Typical wear test data for various combinations of materials and operating conditions are discussed in References 46-27 and 152-3. The wear phenomena which may occur in parts can generally be classified in one of the following four major groups:

1) *Adhesive or Galling Wear.* Adhesive wear is the most fundamental of the several types of wear and generally will persist when all other types of wear have been eliminated. When two solid surfaces are placed in contact, the interface load causes plastic deformation of the surface asperities in contact. In those regions where the intimate contact occurs, strong adhesion takes place and, in the case of metals, becomes in effect a continuous weld, often referred to as cold welding. Relative movement between the two surfaces results in the breaking of this adhesive bond, resulting in transfer of material from one part to the other, or a generation of loose wear particles.

2) *Abrasive or Cutting Wear.* In abrasive wear, loose particles are plowed or gouged out of the base material rather than pulled out, as in adhesive wear. A typical example of this type of wear would be a hard rough surface plowing through a soft material. However, the most probable source of this type of wear which would be encountered by the fluid component designer is where an abrasive particle, either from external generation (atmosphere or system fluid) or from internal generation (oxide fragments or wear particles), is caught between two sliding surfaces. A second case would be where fluid streams are impinging upon solid surfaces.

3) *Corrosive Wear.* Corrosive wear requires the occurrence both of corrosion of the base metal and the presence of some form of adhesive or abrasive wear. Corrosive wear is essentially adhesive or abrasive wear intensified by continuous corrosion of the base metal which is being exposed through the rubbing wear process.

4) *Surface Fatigue.* Cyclic reversal of stresses between two sliding surfaces, or repeated stresses from the rolling contact of two bodies, can result in fatigue of the base metal, further resulting in local pitting or flaking of the seal interface. This wear process is not gradual, but will commence suddenly, after many contact cycles, removing relatively large particles from the surface.

Each design application will have significant wear types which may occur simultaneously and interact. Adhesive and corrosive wear, for example, will often be found in combination in sliding seal surfaces. The application of a mild form of corrosive wear is sometimes used in sliding seals as a means of reducing total wear. One example of this is lubricant additives which mildly corrode the contact area, forming a sulfide, oxide, or phosphate coating. This resulting nonmetallic anti-welding film prevents galling or seizing of the bare metal surfaces.

In general, the design approach should be such that the item which will be most susceptible to deformation, wear, or tearing during operation is the least expensive and most easily replaced component. This will normally be the seal.

Lubrication. Many premature seal failures, particularly in pneumatic and vacuum applications, can be attributed to inadequate lubrication or lack of lubrication, resulting in high friction at the interface, excessive wear of the seal and rapid heat buildup. Lubrication can be provided from several different sources; quite often the liquid being sealed will provide adequate lubrication.

In sliding applications, relative motion is normal to the plane of the seals, thus carrying a film of fluid across the working face and extending the contact area. Some lubrication is therefore insured in liquid systems.

In a rotary application, the seal load tends to squeeze out the lubricating fluid at the contact face. Continuous operation against the same small area of the gland can induce excessive localized heating. If severe enough, this action can damage the seal and possibly the shaft, impairing sealing.

Typical seal lubricants for various temperature ranges are listed in Reference 1-50, and a more detailed discussion of the mechanisms of lubrication and types of lubricants is presented in Detailed Topic 6.8.2.2. Lubricants especially suited for use with elastomers are covered in Detailed Topic 6.4.5.3. The effects of various lubricants on the friction coefficients of different material combinations are tabulated in Sub-Section 12.7.

To reduce frictional resistance in those areas where lubricants cannot be used, either coating or impregnating the base metals is often utilized. Coatings may take the following forms:

a) Chemical reaction on the surfaces of the base metals, such as hard anodizing of aluminum

b) Application of a soft metal film, such as silver, nickel, or indium

c) Application of a polymer film, such as Teflon

d) Application of hard coatings (such as tungsten carbide or aluminum oxides) and platings (such as chromium).

See Table 12.7 for actual friction coefficient values for such coatings. Using gland alloy materials with a high content of nickel or silver will often produce a solid lubricating film at the interface.

6.4.3.2 MATERIALS. The factors required in selection of materials for dynamic seals will be the same as for static seals (see Detailed Topic 6.3.2.2), except that friction and wear at the interface are prime considerations, in addition to good interface mating.

6.4.3.3 CONTAMINATION. Lip seals are often used to protect other components, such as bearings, from extreme operating environment conditions and contamination. In general, lip seals with auxiliary lips are satisfactory in moderately severe conditions, while metallic wipers may be required for extremely dirty conditions (Figure 6.4.3.3). In seal design where excessive wear occurs, the seal can become a contamination generator. Special consideration may be required in seal material selection when sealed media contamination cannot be tolerated.

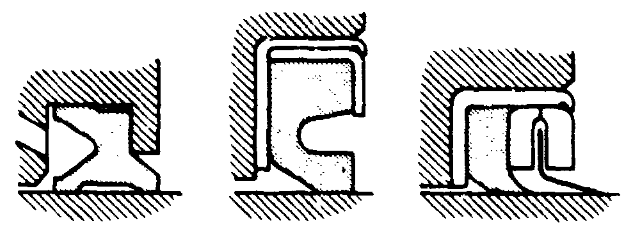

Figure 6.4.3.3 Wiper Configurations
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6.4.3.4 FLUID COMPATIBILITY. The swell of polymeric seal materials when in contact with various liquids is more serious in dynamic seal design than in static seal design. In a static seal design, the increase in stress resulting from the change in volume of the seal materials is not detrimental as long as gross deformations do not occur in the seal housing. In dynamic seal design, however, the seal interface is a balance between seal stress required for the allowed leakage rate and load limits which will make the wear characteristics or abrasion of the seal material intolerable. Increased seal loads resulting from swell can drastically increase the frictional load and rate of wear of the seal interface or, under extreme conditions, may even prevent operation. As a general rule, materials for dynamic applications should be limited to a volume swell of 15 percent.

6.4.3.5 TEMPERATURE. In addition to the thermal environment, temperature problems with dynamic seals can result from heat generated at the seal interface by rubbing friction. The temperatures attained, aside from material selection considerations, will require consideration of the following factors:

a) A change in the interface relationship resulting from thermal expansion. A decrease in the interface loading will result in increased leakage, while an increase in the loads will increase the frictional forces and wear of the interface. Thermal expansion comparison of typical seal and gland materials is shown in Table 6.3.2.7.

b) Vaporization of liquid surrounding the seal interface, which will destroy the interfacial liquid film and change the lubrication characteristics.

Life and performance factors can be improved if the design provides for dissipating excessive heat. This is particularly true for dynamic seals which operate continuously and/or operate at high speeds.

6.4.3.6 PRESSURE. In general, the allowable speed of a given seal will decrease as the pressure of the contained fluid increases. (Allowable speed depends not only on pressure, but also on temperature, gland surface finish, deflection, and endplay, and upon the actual lubrication that reaches the seal.)

Seal design for sliding motion should provide, where possible, application of the fluid pressure in the opposite direction of the friction force to improve seal life and reduce extrusion in packing installations. By placing the seal in the opposite part of the gland, the relation of friction force to fluid pressure will be reversed, as illustrated in Figure 6.4.3.6a.

In those applications utilizing pressure-energized seals, special consideration must be given to the possibility of reversal of the direction of fluid pressure or vacuum. Unless the seal functions bi-directionally, as is the case with an O-ring, these conditions can lead to introduction of dirt or other contaminants into the system. A typical back-to-back dual installation of unidirectional seals is shown in Figure 6.4.3.6b. Proper venting of the gland to prevent pressure buildup, or vacuum caused by temperature changes, may be required.

6.4.3.7 RESILIENCY AND RATE OF RETURN. Resiliency of a seal is its ability to return to its original shape after release of an applied load. It may also be defined as the ratio of energy given up on recovery from deformation to the energy required to produce the deformation. Seal resilience is a function of geometry as well as material properties and is limited to deformation in the elastic range. In simple shapes, material properties may be of greater significance. The elastic range of metals is usually associated with very low strain levels, but in nonmetallic materials the property of recovery is applicable over a very wide strain level. Resiliency is a very important consideration in seal design in that it determines the capability of the seal to maintain contact under conditions of deflection of the mating gland interface.

A related property which is important in dynamic seal design is the rate of return after compression, or the speed with which the material returns to its original shape after being deformed. While total displacement in valve operation is generally small, actuation time is often quite fast and relatively high speeds can be attained. Rate of return can therefore be important when the seal must follow and maintain contact with an uneven surface.

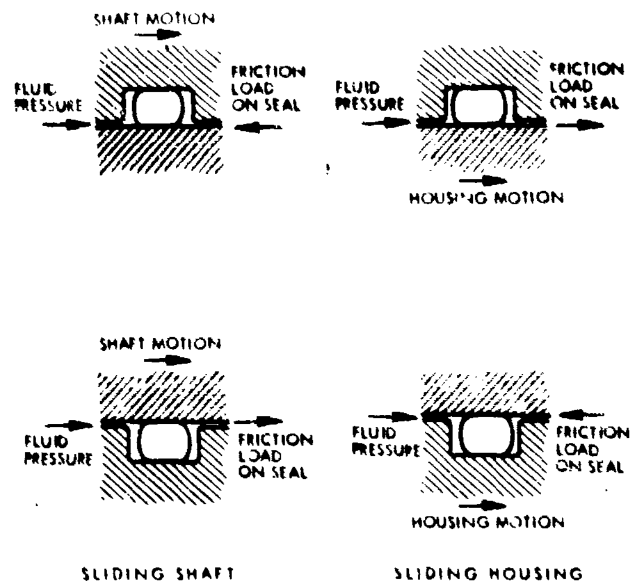

Figure 6.4.3.6a. Relation Between Fluid Pressure and Frictional Forces in Sliding Seals

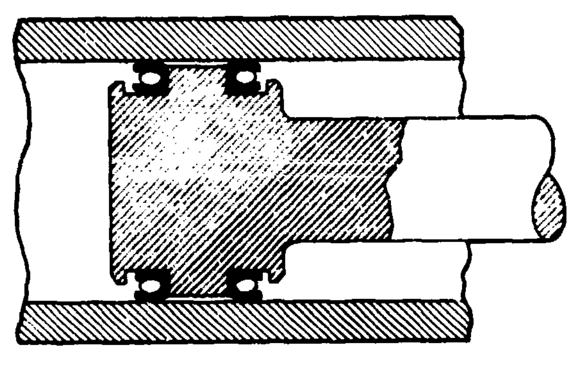

Figure 6.4.3.6b. Dual Seal Installation for Bi-Directional Pressure Capabilities

(Courtesy of Aeroquip Corporation, Jackson, Michigan)

6.4.4 Design Data

The following presentation of dynamic seal design data includes parametric data and examples of representative design approaches and techniques. The sub-topic is divided into the basic dynamic seal types. Reference 152-4 is recommended for a comprehensive analytical treatment of the dynamic sealing problem.

6.4.4.1 LIP SEALS. Lip seals consist of relatively flexible nonmetallic sealing elements which have an interference fit with the moving part of the gland. The sealing element, usually positioned by a metal case, will be either a plastic or an elastomer, and elastomers may be coated with Teflon to reduce friction. To augment the seal loads and maintain continuous contact when sealing low viscosity fluids, the seal element is often spring-loaded. Most lip seals are also pressure-energized.

Pressure sensitivity of lip seals is inherent in their design. Typical action of forces on a lip seal are shown in Figure 6.4.4.1a. In order to have equilibrium at the lip, the spring force S and the flexing moment M must counterbalance three forces: (1) F_R, the force caused by shaft runout, (2) I, the inertia force, and (3) the opposing components of the unit pressure, P. Generally, all these various forces can be resolved into two forces, P_H and P_V, acting horizontally and vertically. P_H tends to displace the sealing element along the shaft and with high enough pressure may blow the lip out. In the usual sealing element configuration, however, the vertical force P_V is larger and tends to flatten the sealing element on the gland well before such blowout occurs. An excessive value of P_V will create a high seal load, which will result in a wide contact width. Excessive heat generation from the increased friction load may harden and degrade the sealing element material. Resulting cracks in the contact surface become leak channels. Using a harder compound, shortening and thickening the seal element, or using an abrasion resistant compound, can improve the performance, but such changes will result in lower runout tolerance, higher initial heat generation, and generally shorter life.

Primary characteristics of the lip seal are small space requirements and ease of installation; wide variety of standard sizes, materials, and lip combinations; good sealing capability with many types of fluids; good exclusion of dirt or contaminants; and relatively low cost.

The lip seal is most commonly used in rotary applications, frequently for retention of lubricant in rolling element bearing applications. Generally distinguished as an individual type, the wiper ring (Figure 6.4.3.3) is especially designed to exclude foreign matter either in rotating or sliding installations. The external lip seal (Figure 6.4.4.1b) is seldom used on a rotating shaft, because of the effects of centrifugal force. Lip seals may be used in high pressure applications with special gland designs which limit deformation. For example, the seal shown in Figure 6.4.4.1c was designed for a 2000 psi cryogenic application involving sliding motion.

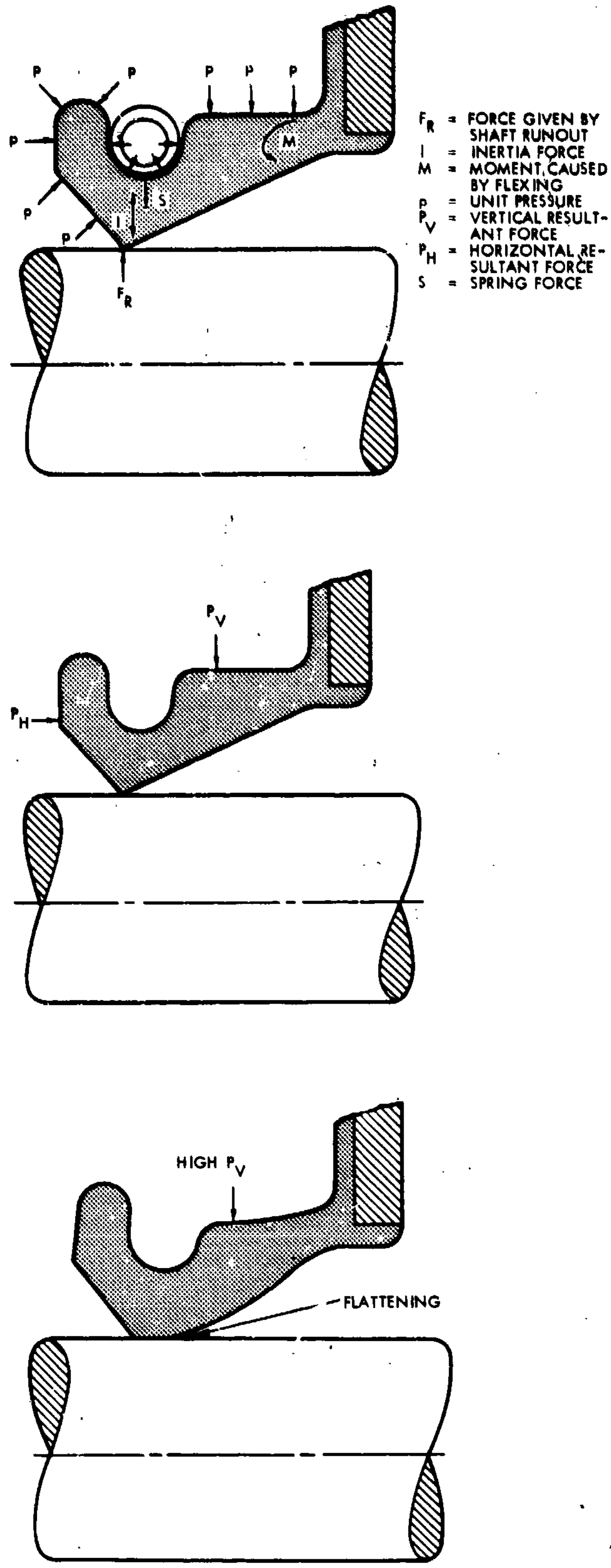

Figure 6.4.4.1a. Action of Forces on a Lip Seal
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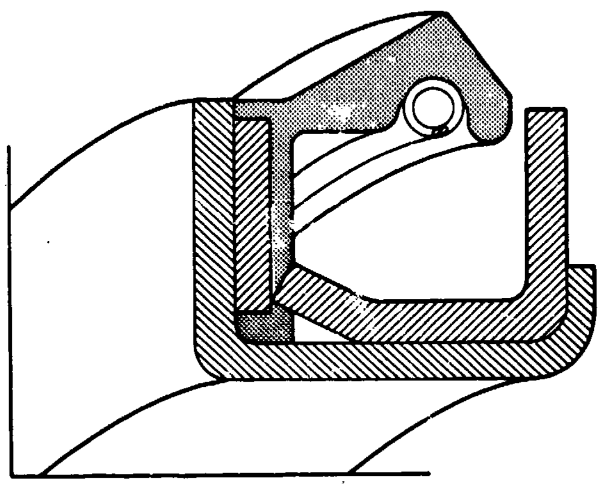

Figure 6.4.4.1b. External-Type Lip Seal
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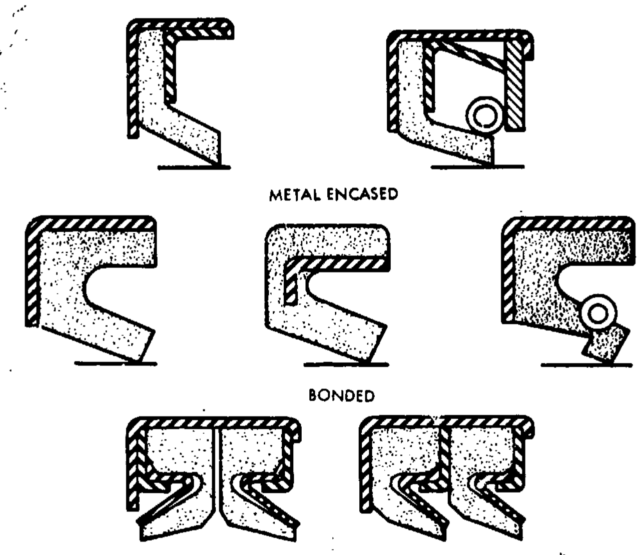


Figure 6.4.4.1d. Lip Seal Configurations
(Reprinted with permission from "Machine Design," 31 October 1957, vol. 29, no. 22, J. B. Holt and W. S. Miller, Copyright 1957. The Penton Publishing Company, Cleveland, Ohio)

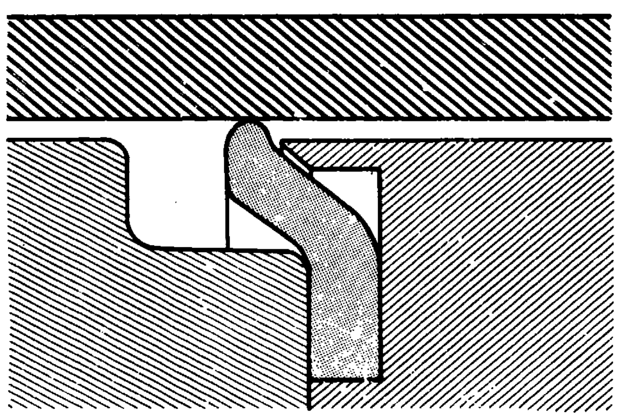


Figure 6.4.4.1c. Gland Configuration for High Pressure Lip Seal
(Reference 34-10)

Lip Seal Construction. Commercially available lip seals may be categorized into three basic classes according to construction (Figure 6.4.4.1d):

a) *Cased seals,* in which the sealing element is retained in a metal case which is pressed into the gland.

b) *Bonded seals,* in which the sealing element is permanently bonded to a flat washer or to a formed metal case. Use of the bonded seal is often recommended where operating conditions require increased seal flexibility, or where installation space is limited by external design requirements.

c) *Dual-element or combination seals,* which are available for unusually severe service, or where liquids are present on both sides of the seal. Other combination seals provide a sealing action in one direction, plus a wiping action in the reverse direction. Seals of all categories can be provided with spring-tension elements, either garter-spring or finger type, for use where either shaft speed or eccentricity demand higher unit sealing pressures.

6.4.4.2 MOLDED SHAPE PACKINGS. There are five major forms of molded nonmetallic packings used in dynamic sealing, characterized by their particular shape and commonly known as:

a) O-rings

b) V-rings

c) U-rings

d) Cup

e) Hat or flange.

With the exception of the O-ring, molded shape packings are normally used in sliding applications. (O-rings, because of their wide usage, are discussed separately in Detailed Topic 6.4.4.3.) Where limitations of the O-ring preclude its use, the various shape packings may be used in specific applications. Molded shape packings seal by pressure energization except in extremely low pressure applications, in which case seal resiliency and gland preload provide the necessary seal load. Elastomeric packings are normally limited to a pressure differential of 1,500 psi, but may be used at higher pressures through the use of backup rings. Teflon, because of its low resiliency, is normally limited to use in applications where compatibility is a problem.

The minimum clearance between moving and stationary parts of the gland necessary to prevent extrusion of the packing is a function of the contained pressure. Some recommended clearances are shown in Table 6.4.4.2a. Where compatibility will permit, elastomeric materials can be fabric-reinforced to provide greater strength and improved extrusion resistance. The clearances are applicable to all packing shapes, except O-rings, and may be used whether the seal is inside-packed or outside-packed. In an inside-packed installation, the moving seal interface is on the outside diameter of the seal; in an outside-packed installation, the moving seal interface is on the inside.

The V-ring and the U-ring packings are considered balanced, and may be installed inside-packed or outside-packed. A balanced packing will seal both the inside and the outside diameter, with equal pressure being applied on the side walls of the gland. The cup and hat packings are unbalanced in that sealing occurs on only one diameter. The cup packing is used in inside-packed installations and the hat (which is a reverse configuration of the cup packing) is used in outside-packed installations.

V-Ring Packings. V-ring (chevron) packings are installed in sets, each set consisting of a number of rings and a male and female adapter. As more rings are added, greater protection against extrusion from pressure surges is provided, however, more rings mean more friction force. The number of rings recommended, which represents a compromise between friction and extrusion factors, depends on the pressure and the packing material (Table 6.4.4.2b). Where installation becomes a problem, split V-rings can be used with the joints staggered, however this reduces the sealing effectiveness and life.

Table 6.4.4.2a. Reinforced Packings for Elastomer and Plastic Packings

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

	Cylinder Diameter (in.)	Diametral Clearance (in.)		
		Under 500 psi	500 - 3000 psi	Over 3000 psi
Reinforced Packings	Under 3	0.006	0.004	0.003
	3 to 8	0.008	0.006	0.004
	8 to 10	0.010	0.008	0.005
	10 to 12	0.012	0.010	0.006
	12 to 16	0.014	0.012	0.007
	16 to 24	0.016	0.014	0.008
Elastomer and Plastic Packings	Under 2	0.006	0.005	
	2 to 5	0.008	0.006	
	5 to 8	0.010	0.008	
	8 to 10	0.012	0.010	
	10 to 12	0.014	0.012	
	12 to 16	0.016	0.014	

* For inside-packed installations, clearance is between the rod or piston and cylinder wall. For outside-packed installations, clearance is between back support ring and shaft.

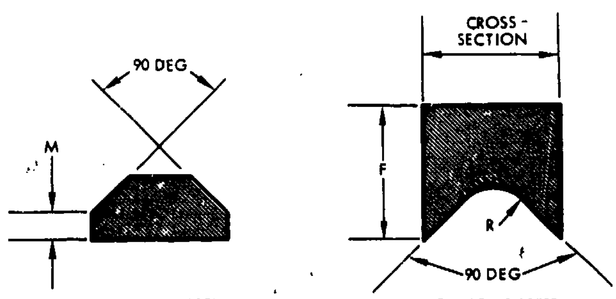
Table 6.4.4.2b. Recommended Number of Packings per Set, Based on Solid Rings

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Pressure (psi)	Elastomer and Plastic (Number of rings per set)	Reinforced
Up to 500	3	3
500 to 15,000	4	4
1500 to 3000	5	4
3000 to 5000	5	5
5000 to 10,000	-	6
10,000 and over	-	-

Table 6.4.4.2d. Dimensions for V-Ring Adapters

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)



MALE ADAPTER FEMALE ADAPTER

Cross-Section	Dimensions (in.) F	M	R
1/4	1/4	1/8	1/16
5/16	5/16	1/8	7/64
3/8	3/8	1/8	1/8
7/16	7/16	1/8	5/32
1/2	1/2	1/8	5/32

Table 6.4.4.2c. Representative Standard Nominal Sizes for V-Ring Packings

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Cross-Section	Nominal ID (in.)	Nominal OD (in.)	Dimensions Thickness ±0.010 (in.)	"V" Radius R (in.)
1/4	1/4	3/4	0.083	5/64
1/4	1/2	1	0.083	5/64
1/4	3/4	1-1/4	0.083	5/64
1/4	1	1-1/2	0.083	5/64
1/4	1-1/4	1-3/4	0.083	5/64
5/16	1-3/8	2	0.140	1/8
5/16	1-7/8	2-1/2	0.140	1/8
5/16	2-3/8	3	0.140	1/8
3/8	2-3/4	3-1/2	0.156	9/64
3/8	3-1/4	4	0.156	9/64
3/8	3-3/4	4-1/2	0.156	9/64
7/16	4-3/4	5-5/8	0.197	3/16
1/2	5-1/2	6-1/2	0.197	3/16
1/2	6	7	0.197	3/16
1/2	7	8	0.197	3/16
1/2	8	9	0.197	3/16
1/2	9	10	0.197	3/16
1/2	11	12	0.197	3/16
1/2	13	14	0.197	3/16
1/2	15	16	0.197	3/16

The standard angle for V-rings is 90 degrees. Typical dimensions for V-rings are shown in Table 6.4.4.2c and for adapters in Table 6.4.4.2d. The adapters may be separate pieces or may be integral with the gland. To prevent overtightening and to ensure automatic, controlled wear take-up, as well as to allow a fixed gland length, spring biasing can be used (Figure 6.4.4.2a). A spring load of 5 pounds per linear inch of mean ring circumference is usually recommended. A single spring is used for small diameters, and multiple springs are used in large diameters.

Table 6.4.4.2e. Representative Standard Nominal Sizes for Cup Packings

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Dimensions				
OD (in.)	Height (in.)	Hole OD (in.)	Thickness, E (in.) Min.	Thickness, E (in.) Max.
1/2	1/4	To suit	1/32	1/16
3/4	5/16		1/32	3/32
1	7/16		1/16	1/8
1-1/2	1/2		3/32	1/8
2	1/2		3/32	5/32
2-1/2	1/2		3/32	5/32
3	5/8		1/8	3/16
3-1/2	5/8		1/8	3/16
4	5/8		1/8	3/16
5	3/4		1/8	3/16
6	3/4		1/8	3/16
7	3/4		1/8	3/16
8*	1		1/8	3/16
9*	1		1/8	3/16
10*	1		1/8	3/16
12*	1-1/4		1/8	3/16

* For reinforced cup packings
E Max. = 1/4 inch

Table 6.4.4.2f. Representative Standard Nominal Sizes for U-Ring Packings

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Cross-Section (in.)	Nominal ID (in.)	Nominal OD (in.)	Dimensions: Ring Height (in.)	Dimensions: Thickness, E (in.)
1/4	1/2	1	5/16	1/16
3/8	1	1-3/4	3/8	3/32
3/8	1-1/2	2-1/4	3/8	3/32
1/2	2	3	7/16	1/8
1/2	2-1/2	3-1/2	1/2	1/8
1/2	3	4	1/2	1/8
5/8	4	5-1/4	5/8	5/32
5/8	5	6-1/4	5/8	5/32
3/4	6	7-1/2	3/4	3/16
3/4	7	8-1/2	3/4	3/16
3/4	8	9-1/2	3/4	3/16
3/4	9	10-1/2	3/4	3/16
3/4	10	11-1/2	3/4	3/16
3/4	12	13-1/2	1	3/16
3/4	15	16-1/2	1	3/16

U-Ring Packings. U-ring packings, a combination of the cup and flange design, are balanced packings which seal on both the OD and ID. They are completely automatic in action, with low frictional characteristics at relatively low pressures, and are not stacked in nested sets.

Standard U-ring dimensions are shown in Table 6.4.4.2f, and typical gland designs are shown in Figure 6.4.4.2d.

Cup Packings. The cup packing is the most common of the lip-type, pressure-energized packings. A lip seal has less strain energy than a compression seal such as an O-ring, but unlike a compression seal the lip seal obtains additional contact pressure from the applied fluid pressure. Correspondingly, for relatively low differential pressure, the larger portion of the frictional load is provided by the internal energy of the seal itself, and overall frictional force is low. As differential pressure increases, the system pressure becomes the dominant component of contact pressure, and the larger contact force of the lip seal produces greater total friction load. Thus, in low-pressure systems where friction is important, cup packings may be preferred to O-rings. Cup packings have another advantage in their ability to withstand foreign-particle contamination in situations where the operating fluid cannot be kept clean.

In cup packings, sealing may occur at the lip at low pressure, but normally proper sealing requires expansion of the heel or shoulder of the lip to contact the gland. Wear occurs primarily at the heel, with very little occurring at the lip. Excessive installation loads on the bottom of the cup will crush the material and cause the lips to "toe in," impairing sealing and increasing friction and wear. Compression should normally be limited to 25 percent of the material thickness. Standard dimensions for cup packings are shown in Table 6.4.4.2e.

When the internal pressures are high, the lip will maintain contact with the gland. For low pressures, and for

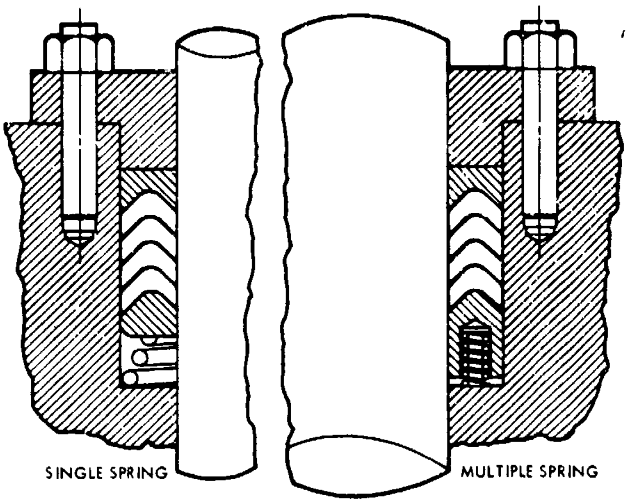


Figure 6.4.4.2a. V-Ring Packings

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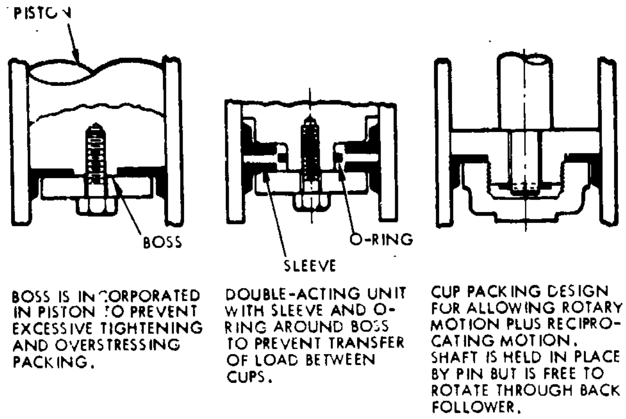

Figure 6.4.4.2c. Cup Packings

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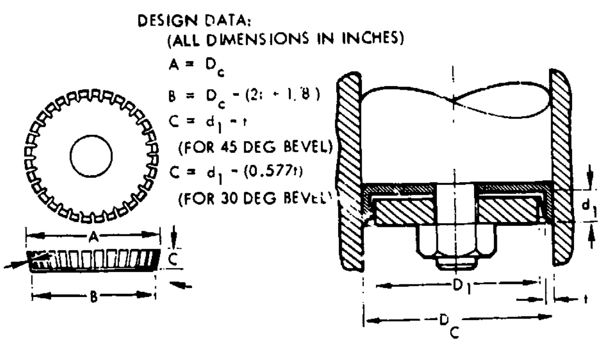

Figure 6.4.4.2b. Spring Expander Design for Cup Packings

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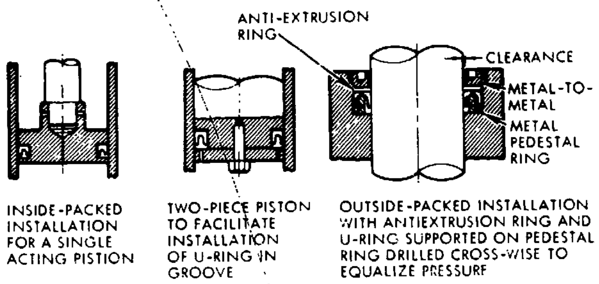

Figure 6.4.4.2d. U-Ring Packing Installations

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conditions of pressure surges, an expander may be required to maintain the load on the lip circumference. The finger spring expander shown in Figure 6.4.4.2b is the most common type used. The metal thickness will range from 0.010 inch for small sizes up to 0.020 inch for large sizes. Typical cup packing installations are shown in Figure 6.4.4.2c.

Flange Packing. The flange packing design is the reverse of the cup packing design. It is used for outside-packed installations usually when there is insufficient space for a V-ring or U-ring. The installation methods for cup packings are applicable to flange packings.

6.4.4.3 **O-RINGS.** An O-ring is a molded shape packing which, because of its wide acceptance, has become standardized as to sizes and installation. The advantages of the O-ring as a packing are:

a) Low initial cost

b) Small installation envelope

c) Simple installation

d) Bi-directional sealing

e) High resiliency

f) No adjustment requirement

g) Relatively low friction.

Groove Design. The standard O-ring groove for dynamic seal application is rectangular in shape (dimensions are shown in Table 6.4.4.3a). When O-rings are used as dynamic seals, the clearance gap must be kept small in order to minimize extrusion of the O-ring into the clearance on pressure actuation. When the seal moves, pieces of the O-ring which have extruded into the clearance will be cut off or nibbled away as the O-ring turns inside the groove and the extruded section is pinched. If required, diametral clearances for O-rings can be varied with O-ring hardness and pressure, as shown in Table 6.4.4.3b. Care must be exercised when using harder compounds to compensate for higher pressure, as this may result in shorter life. For example, increase in hardness of a compound from 70 to 90 durometer is accompanied by a loss of elasticity and stress life, and an increase of compression set. These factors hasten the onset of fatigue failure and increase friction in dynamic-sealing applications. Therefore, it is preferable to obtain high-pressure extrusion resistance by using closer fits or suitable backup rings.

The groove volume is approximately 15 percent greater than the O-ring volume, thus allowing the O-ring to roll, facilitating assembly, and permitting swelling of the O-ring under fluid action. The gland depth allows for approximately 14 percent squeeze, but may be varied to meet specific requirements of sealing and wear. When squeeze is reduced, care must be taken to assure that the minimum squeeze is always greater than one-half the maximum diametral clearance, to avoid loss of the seal in an eccentric gland condition. As in the case of static seals, gland design should provide for an installed stretch of not more than 5 percent of the O-ring inside diameter.

Finish of the gland surface over which the O-ring moves should be very smooth; a finish of 10 to 16 microinches AA is recommended for dynamic applications. Finish should be ground concentrically for rotary sealing and longitudinally for sliding sealing. Finishes smoother than 5 microinches AA should be avoided in reciprocating applications, since the O-ring tends to remove lubricant from the gland as it moves in one direction, resulting in inadequate lubrication on the return stroke.

In low pressure (200 psi) pneumatic applications, the use of a floating gland design should be considered when tolerance accumulations on runout and eccentricity become excessive. The object of this design, shown in Figure 6.4.4.3a, is to allow a small amount of floating to offset misalignment. Typical gland dimensions and applications are described in References V-70 and V-74.

For sliding applications, the placement of the groove in the gland should be such that the friction loads oppose the pressure loads (see Figure 6.4.3.6a). If the friction load of the moving gland across the O-ring is in the same direction as the pressure loads, the O-ring will tend to

These figures relate to Table 6.4.4.3a (opposite page)

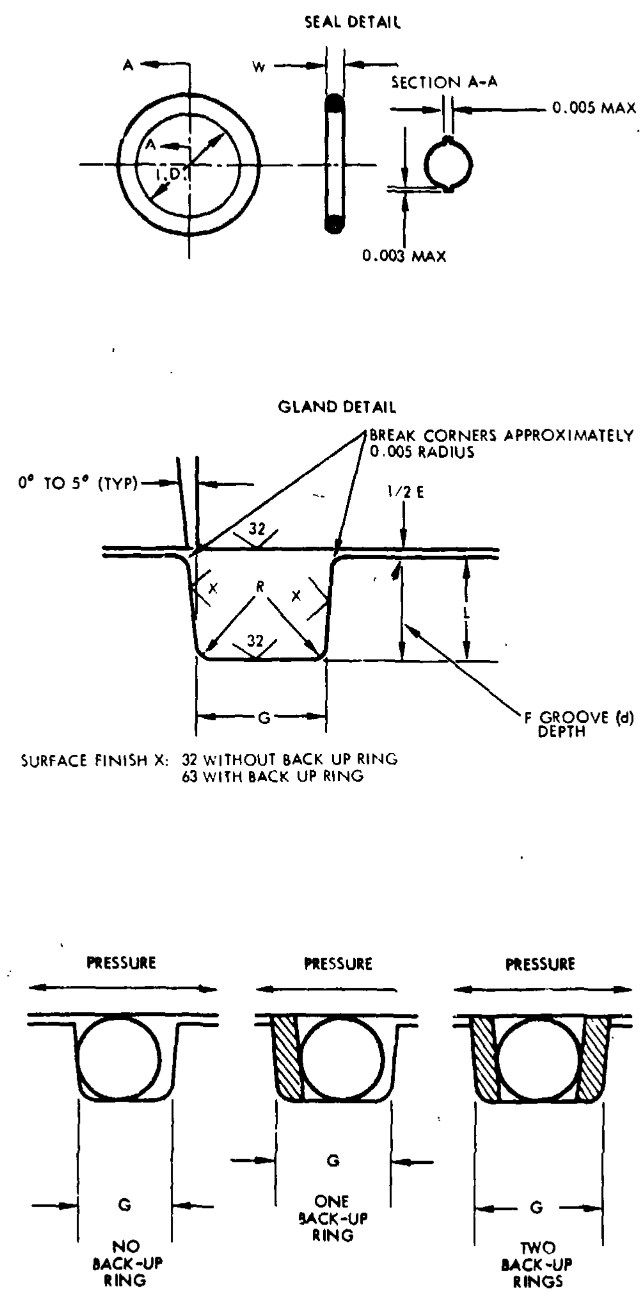


Table 6.4.4.3a. Standard Dynamic Seal Elastomer O-Ring Gland Dimensions

(Adapted from "Parker O-Ring Handbook," Cat. 5700, November 1964, Copyright 1964, Parker Seal Company, Culver City, California)

W Cross-Section		L	Squeeze		E	G +0.010 -0.000			R	
Nominal	Actual	Gland Depth	Actual	%	Diametrical Clearance Max. (b)	No Back-up Rings	One Back-up Ring	Two Back-Up Rings	Groove Radius	Eccentricity Max. (a)
3/64	0.040 ±0.003	0.031 to 0.032	0.005 to 0.012	13.5 to 28	0.004	0.063	-	-	0.005 to 0.015	0.002
3/64	0.050 ±0.003	0.040 to 0.041	0.006 to 0.013	13 to 25	0.004	0.073	-	-	0.005 to 0.015	0.002
1/16	0.060 ±0.003	0.048 to 0.049	0.008 to 0.015	14 to 24	0.004	0.083	-	-	0.005 to 0.015	0.002
1/16	0.070 ±0.003	0.056 to 0.058	0.010 to 0.017	15 to 23.5	0.004	0.094	0.149	0.207	0.005 to 0.015	0.002
3/32	0.103 ±0.003	0.089 to 0.091	0.010 to 0.017	10 to 16	0.005	0.141	0.183	0.245	0.005 to 0.015	0.002
1/8	0.139 ±0.004	0.121 to 0.123	0.0115 to 0.0215	8 to 15	0.006	0.188	0.225	0.304	0.010 to 0.025	0.003
3/16	0.210 ±0.005	0.186 to 0.188	0.017 to 0.029	8.3 to 13.5	0.007	0.281	0.334	0.424	0.020 to 0.030	0.004
1/4	0.275 ±0.006	0.238 to 0.241	0.028 to 0.0425	10.5 to 15	0.010	0.375	0.440	0.579	0.020 to 0.030	0.005

(a) Total indicator reading between groove and adjacent bearing surface.

(b) Clearance gap must be held to a minimum consistent with design requirements for temperature range variation.

be dragged into the clearance gap more readily and extrude at much lower (30 to 40 percent) than normal pressures.

Installation Requirements. The requirements for O-ring installation and gland closure for static seals are also applicable to dynamic seal installation. Lead-in chamfers should be provided, as shown in Figure 6.3.3.3c, to avoid cutting of the O-ring. In some instances of sliding action it will be necessary for an O-ring to cross over a port or hole. In order to avoid being cut, torn, or pinched by the sharp edge of the cross port, undercutting or chamfering as shown in Figure 6.4.4.3b is required.

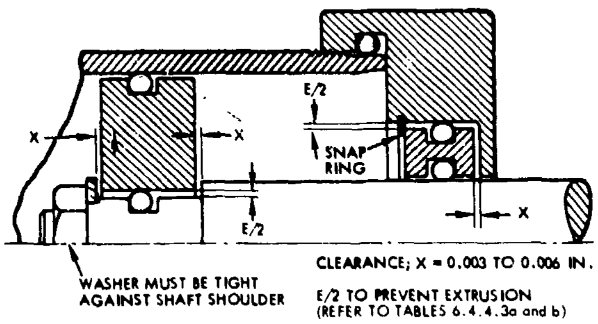


Figure 6.4.4.3a. Floating Gland
(Courtesy of Parker Seal Company, Culver City, California)

Anti-Extrusion Rings. In cases where the recommended clearances shown in Table 6.4.4.3b cannot be achieved, or where pressures are higher than noted, anti-extrusion backup rings may be installed, as described in Detailed Topic 6.3.3.3. These rings may be made of plastic or metal, with the width of the groove controlled to prevent the backup rings from rolling. Variation in the shape of the backup rings can result in better support for the O-ring, plus minimized distortion, resulting in increased life capability. A typical design is described in Reference 6-118.

Friction Reduction. In O-ring installations, the break-out or starting friction can be as high as three times the running friction. To reduce the amount of friction loads with O-rings under high pressure, slipper rings are frequently provided, as shown in Figure 6.4.4.3c. In cross-section, these rings may be U-shaped, L-shaped, or plain bands, and are usually made of Teflon. Since Teflon has a low coefficient of friction, breakaway force is low and running friction is almost negligible, even in the absence of lubrication. The free-sliding slipper rings thus permit full use of the sealing resilience of the O-rings at higher working pressures. Because of the relative stiffness of Teflon, care must be taken during assembly to prevent marring the slipper ring. In small sizes, it may be necessary to provide a split gland design to allow installation. A second method, presently in use, combines the low friction characteristics of Teflon with the resiliency of elastomers. It consists of applying a thin coating of Teflon directly to the O-ring, on either one or both sides.

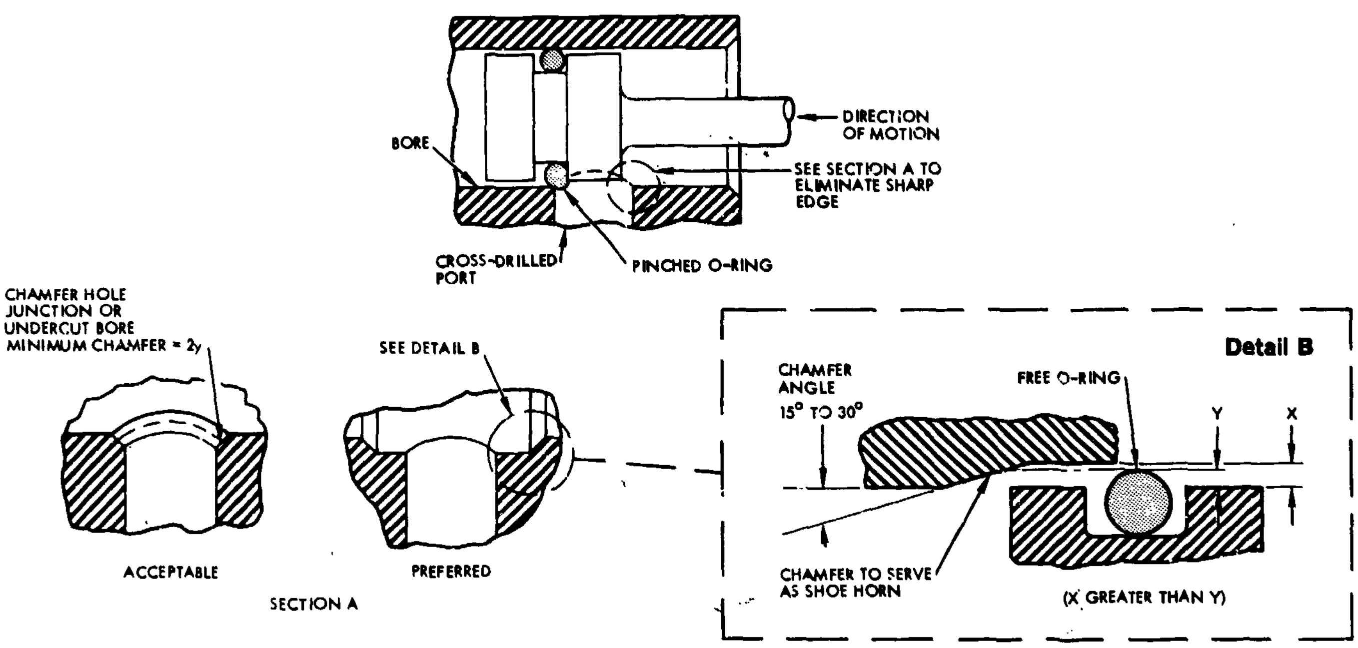


Figure 6.4.4.3b. Gland Chamfer Techniques to Prevent O-Ring Damage

Table 6.4.4.3b. Diametral Clearance for O-Rings

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Hardness (Shore A Durometer)	Ring Cross Section W (in.)	Clearance (in.)							
		Maximum Pressure (psi)							
		250	500	1000	1500	2000	2500	3000	5000
60	0.070	0.007	0.005	0.004	0.002	-	-	-	-
	0.103	0.009	0.007	0.005	0.003	-	-	-	-
	0.139	0.011	0.009	0.006	0.004	-	-	-	-
	0.210	0.012	0.010	0.007	0.005	-	-	-	-
	0.275	0.014	0.012	0.008	0.006	-	-	-	-
70	0.070	-	0.008	0.006	0.004	0.002	0.001	-	-
	0.103	-	0.010	0.007	0.005	0.003	0.0015	-	-
	0.139	-	0.012	0.009	0.006	0.004	0.002	-	-
	0.210	-	0.014	0.010	0.007	0.004	0.0025	-	-
	0.275	-	0.016	0.012	0.008	0.0045	0.0025	-	-
80	0.070	-	0.010	0.008	0.005	0.004	0.003	0.002	-
	0.103	-	0.012	0.010	0.007	0.005	0.004	0.003	-
	0.139	-	0.016	0.012	0.008	0.006	0.005	0.004	-
	0.210	-	0.018	0.014	0.010	0.007	0.006	0.0045	-
	0.275	-	0.020	0.016	0.012	0.008	0.007	0.005	-
90	0.070	-	0.014	0.012	0.010	0.008	0.006	0.005	0.003
	0.103	-	0.016	0.014	0.012	0.009	0.007	0.006	0.004
	0.139	-	0.018	0.016	0.014	0.010	0.008	0.007	0.005
	0.210	-	0.020	0.018	0.015	0.012	0.010	0.008	0.006
	0.275	-	0.020	0.018	0.016	0.012	0.010	0.008	0.006

Generally, the liquid being sealed will provide adequate lubrication. For certain applications, however, particularly in pneumatic and vacuum systems, proper lubrication must be arranged. The optimum lubricant for any application depends on the operating conditions (temperature, fluid, pressure, type of seal) and should be compatible with the system fluid and the O-ring compound. Table 6.4.4.3c lists some commonly used lubricants for O-rings.

Sliding Applications. O-rings are used in rotating service, but find their best application with sliding or reciprocating shafts.

If properly designed, an O-ring will slide during all but a small part of its stroke. Under static conditions, the pressure of the O-ring tends to squeeze the lubricant film out, leaving rubber-to-metal contact. When motion starts, the O-ring rolls slightly, allowing a film of fluid to form between the O-ring and the gland. Starting friction would be much greater without this rolling action. Rolling motion will then change to sliding motion. When segments of the O-ring tend to slide while other sections are rolling, twisting of the seal results. When torsion becomes excessive, spiral failure can occur. Some of the

Table 6.4.4.3c. O-Ring Lubricants

(Reprinted with permission from "Parker O-Ring Handbook," Cat. 5700, November 1964, Copyright 1964, Parker Seal Company, Culver City, California)

Name	Manufacturer	Type	Temperature Range-°F(°C)	Seal Use	Best Service With (Base Rubber)
Petrolatum*	Many	Petroleum base	-20 to +180 (-29 to + 82) (may vary between sources)	Assembly of petroleum system components (fuel and hydraulic)	Nitrile, Neoprene, Viton
Parker O-Lube**	Parker Seal Co., Culver City, California Cleveland, Ohio	Barium grease	-20 to +300 (-29 to +149)	Pneumatic (low pressure—200 psi max)	Nitrile, Neoprene, Viton and most other petroleum oil resistant rubbers
DC200*	Dow Corning Dow Corning Corp., Midland, Michigan	Silicon oil (200,000 centistrokes only)	-65 to +440 (-64 to +227)	Pneumatic-High pressures and speeds	Some Nitrile, Neoprene, Viton (will cause shrinkage of some compounds)
Celvacene	Consolidated Vacuum Co., Rochester, New York	Cellulose ester & castor oil	-40 to +266 (-40 to +130)	Vacuum Static 1×10^{-7} Torr	Silicone, Nitrile, Neoprene, Viton, Butyl, Ethylene-Propylene
Versilube	General Electric Waterford, New York	Silicone grease	-100 to +400 (- 73 to +204)	Pneumatic 3000 psi and high speed	Nitrile, Neoprene, Viton (will shrink some compounds and will dissolve some silicone rubber)
DC55 (MIL-G-4343)	Dow Corning Corp., Midland, Michigan	Silicone grease	-65 to +400 (-54 to +204)	Vacuum, EXTREME TEMPERATURE Lub.	Nitrile, Butyl, Ethylene-Propylene, Viton, Neoprene
Apiezon N	Made In England		+60 to +85 (+15 to +30)	Vacuum 1×10^{-8}	Nitrile, Butyl, Ethylene-Propylene, Viton, Neoprene
Fluorolube	Hooker Chemical Corporation, Niagara Falls, New York	Fluorocarbon fluid	-65 to +400 (-54 to +204)	Oxygen service	Silicone, Nitrile, Neoprene. Ethylene-Propylene, Butyl

*May be used in systems having micronic filters.
**Will not pass through micronic filters. (If no mark appears the micronic filter data was unavailable.)

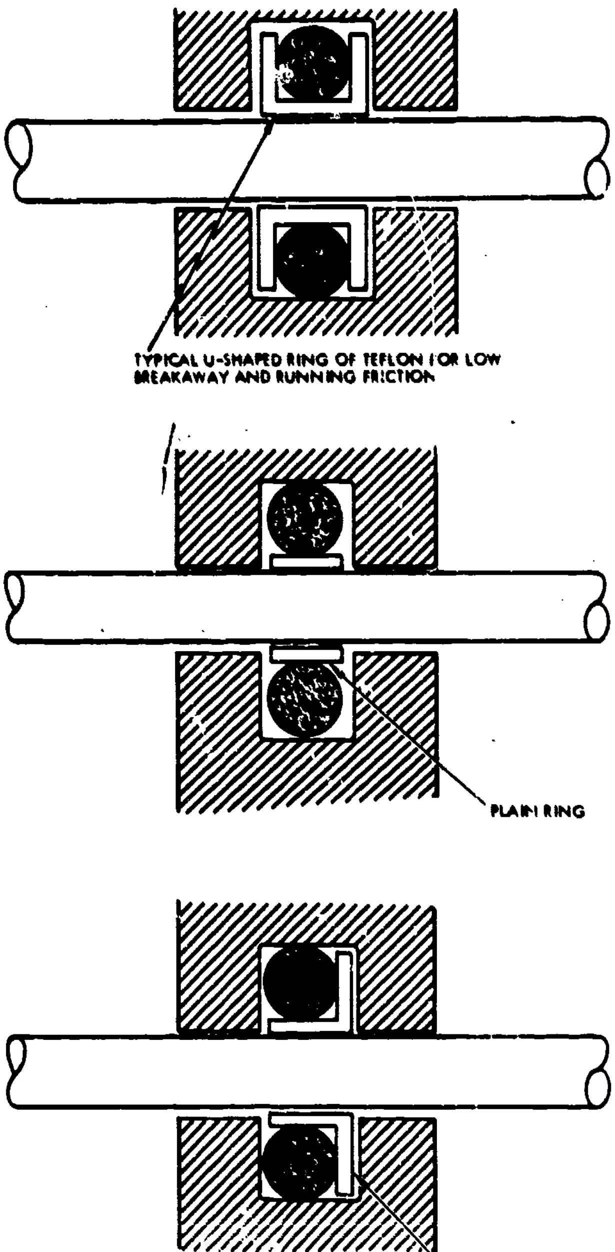


Figure 6.4.4.3c. Typical Slipper Ring Installations
(Reprinted with permission from "Assembly Engineering," January 1965, G. M. Hunter, Copyright 1965, Hitchcock Publishing Company, Wheaton, Illinois)

factors which contribute to spiral failure are listed, in order of their relative importance (Reference V-70).

a) *Speed of stroke.* At speeds of less than 1 foot per minute, the sliding friction is nearly equal to breakout friction and at low pressures extreme twisting can occur.

b) *Lack of Lubrication.* Spotty lubrication or lack of lubrication will induce rolling.

c) *Pressure Differential.* Spiral failure seldom occurs when the pressure differential is in excess of 400 psi. Spiral failure is more likely to occur if the pressure and friction load are in the same direction.

d) *Excessive Squeeze.* More rolling force is created on the O-ring cross-section with an increase in squeeze.

e) *Length of Stroke.* The longer the stroke, the greater the probability that eccentricity, bending, and side loads, will induce factors which contribute to spiral failure. O-rings are not recommended when the stroke exceeds 12 inches.

f) *Excessive Groove Width.* Groove width should be between 25 and 50 percent greater than the O-ring cross-section. Reducing the groove width will control the amount of roll that can occur, but may also result in an increased frictional load on the O-ring, with a resulting decrease in seal life.

To reduce the probability of spiral failure, special seal designs have been developed, the most common being a variation in the cross-section shape. Most of the configurations shown in Figure 6.4.4.3d were designed to operate in a standard O-ring groove. The D-ring and

Figure 6.4.4.3d. Designs to Reduce Spiral Failure

T-ring designs require the use of backup rings, even at low pressures. The Delta-ring design is not subject to spiral failure, but the friction load is higher than the conventional O-ring design and the expected life is relatively short. This design has limited application. The square-shaped ring with four rounded lobes can be used in a conventional O-ring groove for both sliding and rotating motion. The lobed ring is installed with less squeeze than the O-ring, and will have less movement in the groove and reduced breakaway friction. One additional method of reducing spiral failures is the use of a composite elastomer-metal O-ring as described in Reference 77-8. A coil spring is molded inside the O-ring to increase the torsional strength of the seal and still retain a high measure of resiliency.

One other item which should be considered in sliding O-ring applications is the possibility of packing blowout, which can occur when the clearance gap becomes larger as the seal moves. The contained pressure acting on the O-ring causes it to rise and slide into the increased

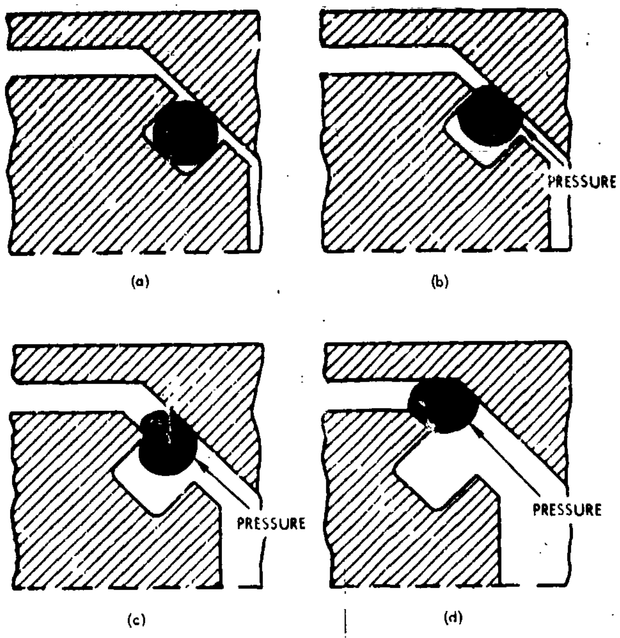


Figure 6.4.4.3e. O-Ring Blowout Action

clearance area, as shown in Figure 6.4.4.3e. The O-ring will continue to seal the opening until it is completely stretched out of the groove. This is a typical failure when O-rings are used as valve seats. Several design methods are available either to retain the O-ring physically or to remove the high pressure differential at the moment of opening, as shown in Figure 6.4.4.3f. Typical dimensions for a dovetail groove are shown in Table 6.4.4.3d.

Rotary Applications. Rotary sealing with O-rings is limited by both pressure and surface speed. No definite limits are set, but generally 600 fpm (Reference 1-292) is considered a maximum reasonable rubbing speed. When speed exceeds 50 fpm, special attention must be directed toward gland design (Reference V-338). With proper gland design, speeds may go as high as 1500 fpm (Reference V-70) at very low pressure. As speed is decreased, pressure can be increased, the practical limit being a static seal installation at about 1500 psi pressure. For rotary applications it is desirable to use an O-ring with the smallest cross-section available for the diameter required. Maximum cross-section diameters are shown in Table 6.4.4.3e.

In addition to the temperature effects of the operating environment discussed in Detailed Topic 6.4.3.6, rotary O-ring applications must also take into account the heat generated during operation. When rubber is stretched and then heated, it tries to contract, a phenomenon known as the Gow-Joule effect. When using an O-ring to seal a rotating shaft, the general tendency would be to make the O-ring a little smaller than the shaft, so that it will

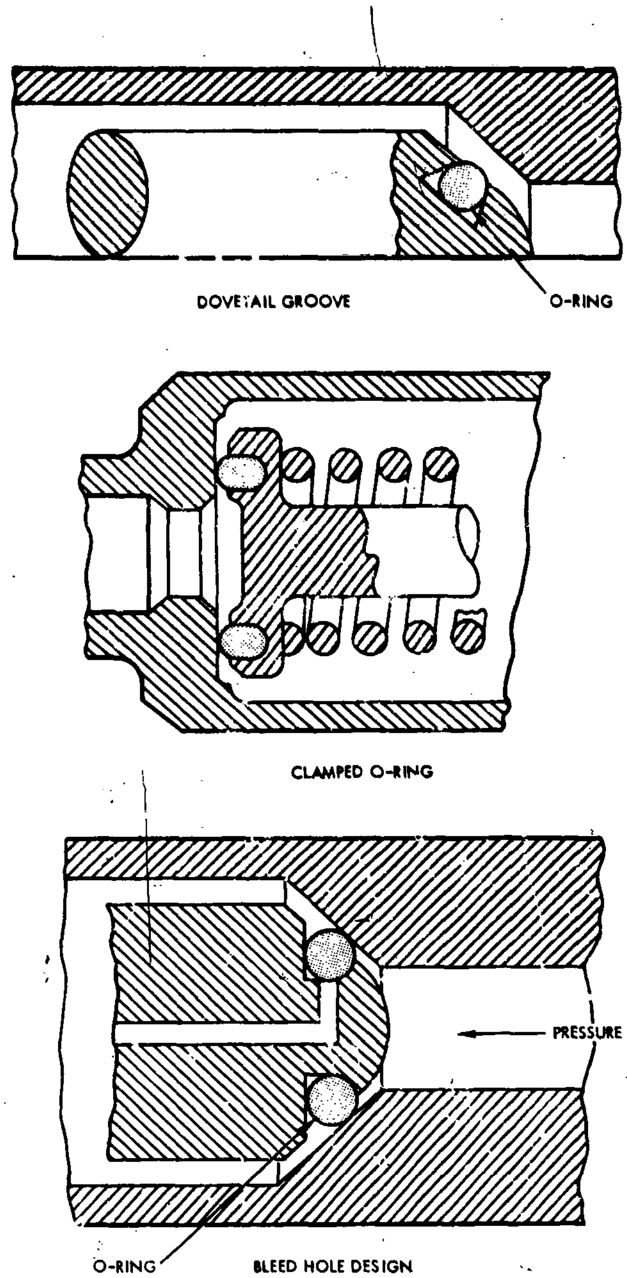
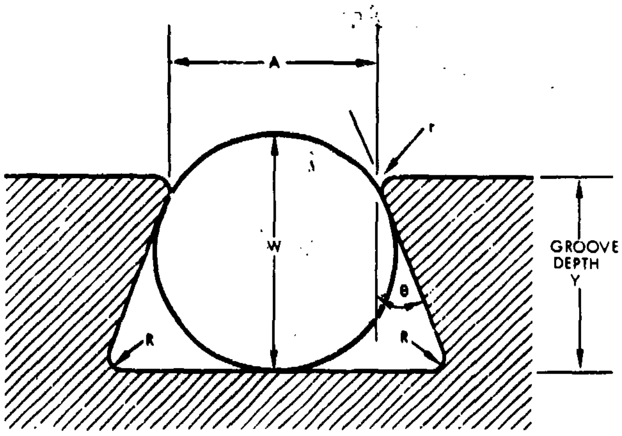


Figure 6.4.4.3f. Gland Designs to Prevent O-Ring Blowout

provide interference and give a better seal; however, in operation friction heats the O-ring. Under the Gow-Joule effect most elastomers will contract, resulting in more friction and more heat. To avoid this problem, the O-ring should be made oversized and the seal obtained by compressing the ring against the shaft. Normally the inside diameter of the O-ring is sized 5 percent larger than the shaft diameter. At surface speeds below 200 fpm this effect is sometimes overcome by plastic flow, but it should be considered in all rotary applications.



$\theta = 24° \pm 1°$

Table 6.4.4.3d. Standard Dovetail Groove Sizes for O-Ring Seals

(Adapted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

O-Ring W	A Groove Length Sharp Edge Tolerance ±0.002	A Groove Length Rounded Tolerance ±0.002	Y Groove Depth ±0.000-0.002	Radius r	Radius R
0.070 ±0.003	0.057	0.063	0.052	0.005	1/64
0.103 ±0.003	0.085	0.090	0.083	0.010	1/64
0.139 ±0.004	0.115	0.120	0.115	0.010	1/32
0.210 ±0.005	0.160	0.170	0.180	0.015	1/32
0.275 ±0.006	0.220	0.235	0.234	0.015	1/16

Table 6.4.4.3e. O-Ring Cross-Section for Rotary Seals

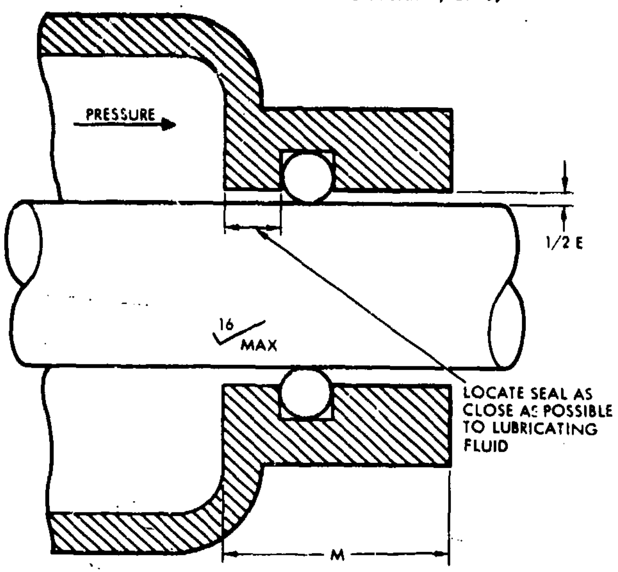
Shaft Diameter (in.)	Max. Speed (FPM)	Max. Cross-Section (in.)
1/8 - 9/32	350	0.103
3/8 - 11/16	450	0.103
3/4 - 1-1/4	600	0.070

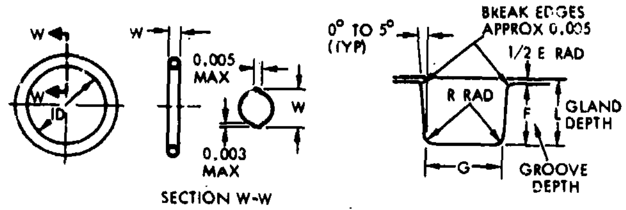
The proper amount of squeeze for rotary O-ring seals has to be determined by compromise. It is desirable to have only enough squeeze to make the seal contact the shaft. This keeps the unit seal load and area of contact to a minimum. High unit loading causes more friction, however more than bare contact is needed to offset the effects of expansion and compression set. A cross-sectional squeeze of 2 to 5 percent is considered a good balance for most rotary O-ring applications when using O-rings with a cross-section up to 0.139 inch. To avoid the effects of centrifugal force on the interface, the O-ring groove should not be installed in a rotating shaft. The groove width should be ten percent larger than the O-ring cross-section to prevent bunching of the seal. Typical groove dimensions are shown in Table 6.4.4.3f.

Plastic O-Rings. The most common O-ring plastic used in dynamic applications is Teflon. Teflon O-rings can be used in standard elastomer O-ring glands, although the cross-section squeeze may be reduced to 5 percent in

Table 6.4.4.3f. Rotary O-Ring Gland Dimensions

(Adapted with permission from "Parker O-Ring Handbook," Cat. 5700, 1964, Parker Seal Company, Culver City, California, and Cleveland, Ohio)





W Cross-Section		L	G	E(a)		M	R
Nominal	Actual	Gland Depth	Groove Width	Diametrical Clearance	Eccentricity Max. (b)	Bearing Length Min. (a)	Groove Radius
1/16	0.070 ±0.003	0.065 to 0.067	0.066 to 0.070	0.012 to 0.016	0.002	0.700	0.005 to 0.015
3/32	0.103 ±0.003	0.097 to 0.099	0.096 to 0.100	0.012 to 0.016	0.002	1.030	0.005 to 0.015
1/8	0.139 ±0.004	0.133 to 0.135	0.132 to 0.136	0.016 to 0.020	0.003	1.390	0.010 to 0.025

(a) If clearance (extrusion gap) must be reduced for higher pressures, bearing length M must be no less than the minimum figures given. Clearances given are based on the use of 80 Shore Durometer minimum O-ring for 800 psi max.

(b) Total indicator reading between groove OD, shaft, and adjacent bearing surface.

larger cross-sections. Teflon O-rings are normally limited to rotary speeds of 600 fpm.

When a small diameter O-ring of Teflon is to be installed in an external groove, the stretch required to install the ring may permanently distort it, so that it will not fully snap back into position. If the assembly is heated to 600°F, plastic memory will cause the ring to return to its original size. Heating small rings prior to installation, i.e., immersing them in boiling water, will often make them sufficiently flexible to assemble.

Since one of the primary difficulties with Teflon O-rings is their lack of resiliency, special designs have been developed which utilize a metal spring to provide resiliency. All of the spring-loaded plastic seals described in Detailed Topic 6.3.3.6 except the Raco seal may be used in rotary applications.

6.4.4.4 COMPRESSION PACKINGS. Compression packings consist of twisted, woven or braided strands of plastic fibers, asbestos, or soft metals packed between shaft and housing. These are often combined with a binder and a lubricant. In practice, the packing is usually placed in one or more coils in a bore and compressed with a flanged unit (Figure 6.4.4.4a).

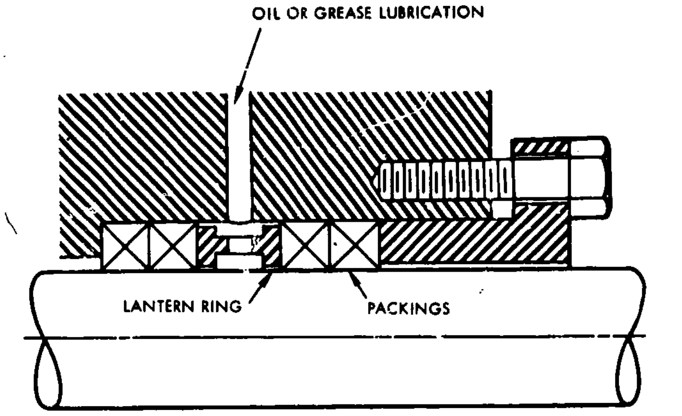


Figure 6.4.4.4a. Typical Low Pressure Compression Packing Design

(Reprinted with permission from "Product Engineering," March 1961, vol. 32, no. 12, Copyright 1961, V. L. Peickii and D. A. Christenson, McGraw-Hill Publishing Company, New York)

Compression packing is used at relatively low speed, and in high-pressure and temperature applications. Pressures in the 10,000-100,000 psi range have been successfully contained by compression-type packings. Despite the disadvantages of high friction and rapid shaft wear, the compression seal still finds extensive use where pressures and temperatures are very high or where sealed shaft diameters are large.

Although compression-type packings have been applied to rotating shafts, they are usually limited to high-pressure reciprocating installations. An unlubricated packing in operation acts much like a bearing: in a typical installation the fluid flows through the slight clearance and acts as a lubricant for the packing. When the gland is overtightened, the fluid flow stops and the packing runs dry. Unless lubrication is provided by some other means, the packing may heat, harden, and score the shaft, as with bearing failures.

Compression packings have a tendency to score the shaft more than most other types of contact seals, because foreign particles tend to enter and become imbedded in the packing. Cleanliness is necessary, and the shaft should be as hard and smooth as is practical. Separate wear rings can be installed where scoring is likely to damage an expensive shaft. As the packing wears, provisions must be made for adjusting the load imposed by the gland. A limited amount of automatic adjustment may be achieved by spring loading the packing, as shown in Figure 6.4.4.4b.

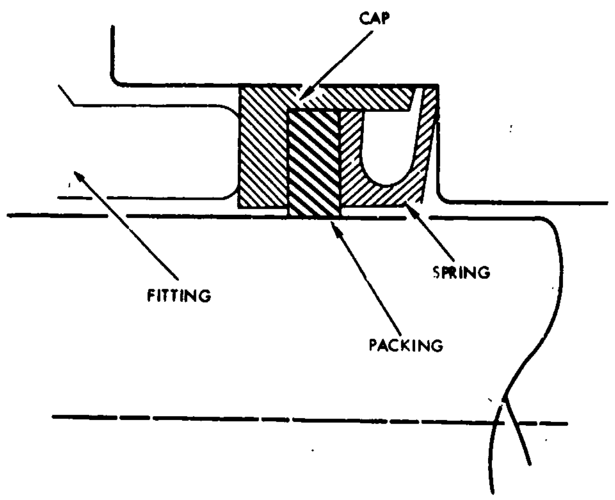


Figure 6.4.4.4b. Spring-Loaded Compression Packing

(Courtesy of Del Manufacturing Company, Los Angeles, California)

The recommended cross-section of the packing is related to the shaft size, as shown in Table 6.4.4.4. When rings are stacked to provide high-pressure capability, the first few rings on the high-pressure side will cause the maximum load, as shown in Figure 6.4.4.4c. Mixing packing styles by alternating rings with those of a more dense or metallic packing will help to reduce extrusion.

6.4.4.5 SPLIT RING SEALS. This category, forming the largest group of high temperature sliding seals, can be divided into two groups: (1) expanding split rings (piston rings) where dynamic sealing is on the outside diameter, and (2) contracting split rings (rod seals) where dynamic

Table 6.4.4.4. Recommended Packing Sizes for Various Shaft Diameters

(Reprinted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

Shaft Diameter (in.)	Packing Size (in.)
1/2 to 5/8	5/16
11/16 to 1 - 1/2	3/8
1 - 9/16 to 2	7/16
2 - 1/16 to 2 - 1/2	1/2
2 - 9/16 to 3	9/16
3 - 1/16 to 4	5/8

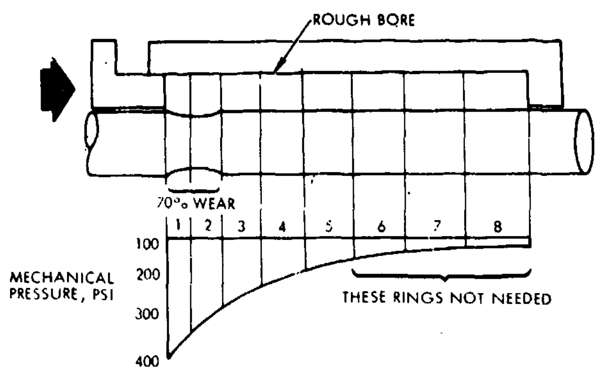


Figure 6.4.4.4c. Load Distribution in Multiple Packing Installation

(Reprinted with permission from "Machine Design Seals Reference Issue," June 1964, vol. 36, no. 14, Copyright 1964, The Penton Publishing Company, Cleveland, Ohio)

sealing is on the inside diameter. Most split ring seals are metallic, with Teflon rings often used to reduce friction where lubrication is not possible.

There is no initial interference in the split ring; sealing is accomplished by an exact fit between the seal and the gland. Sealing contact stresses at the dynamic interface are provided by two means. First, the initial dimensions of the ring are selected for radial contraction or expansion. When the ring is installed, the presence of a radial gap, which is required for installation, prohibits the contraction or expansion from resulting in interference. The ring, however, has a tendency to return to its free state shape and produces a radial contact load on the gland wall. The second source of seal interface load is provided by fluid pressure acting radially against the ring. Secondary or static sealing is accomplished by the action of the contained medium pressure forcing the ring against the side of the groove. Close dimensional control is required over flatness and squareness of both the ring and the sides of the grooves in relation to the axis of sliding for adequate sealing. To minimize the leakage effect of the radial gap, two or more rings can be placed in the same groove with gaps staggered, or various gap shapes can be used, as shown in Figure 6.4.4.5a. Because of the low contact stresses, frictional wear will be minimized. This type seal has the lowest wear of the sliding seal application. To provide sealing capability in both directions, a two-piece design, as shown in Figure 6.4.4.5b, may be used.

Installation. In high pressure installations, pressure balancing may be required to reduce friction load. This pressure balancing is accomplished by adding a groove in the wear surface, which is vented to the high pressure area. The ungrooved land should be limited to a minimum of 0.030 inch axial length to avoid a considerable sacrifice in sealing ability. For low pressure applications, spring loading may be added to increase the interface loads, as shown in Figure 6.4.4.5c.

When a ring is installed, it must often pass over a diameter larger than its own inside diameter. The amount of increase in the gap must be limited to avoid fracturing the ring. The stress level attained may be determined by:

$$S = \frac{0.4815\,E\,\frac{\Delta G}{d}}{\left(\frac{D}{d} - 1\right)^2}$$

where

S = Stress, psi

E = Modulus of elasticity, psi

ΔG = Change in ring gap, in.

d = Radial ring wall dimension, in.

D = Ring outside diameter, in.

Tension of contracting rings is limited, in comparison to that possible in expanding rings. It is therefore often necessary to resort to spring loading to ensure the best conformity of seal ring to the rod. A built-up housing is usually necessary to accommodate contracting seals.

6.4.4.6 SPECIAL SEALING CONCEPTS. In order to attain leakage rates in dynamic seals comparable to those achieved in static seals, considerable effort has been expended in developing methods of sealing where the principle of operation does not depend on contact stress. Some of the investigations are discussed in the following paragraphs.

Freeze Seal. The freeze seal was originally developed for systems having high temperature liquid metals. The operating principle consists of forming a barrier between the liquid metal system and the outside by freezing a slug of the metal in a narrow passage. Frozen metal is held in the narrow passage, which is usually an annulus, by friction and adhesion. In a rotating freeze seal, the frozen metal is continuously sheared at the shaft surface and a thin, high-viscosity liquid film is maintained at the shear interface. Heat generated by film friction is removed continuously from the frozen area by a liquid or gas coolant. This type seal has proved very satisfactory in nuclear systems involving liquid sodium or potassium. A typical configuration is shown in Figure 6.4.4.6, and further information may be found in Reference 19-234.

Fusion Seal. In applications where motion is intermittent and actuation time is not critical, application of a seal similar to the fusion seal described in Detailed Topic 5.2.10.4 may be used to advantage.

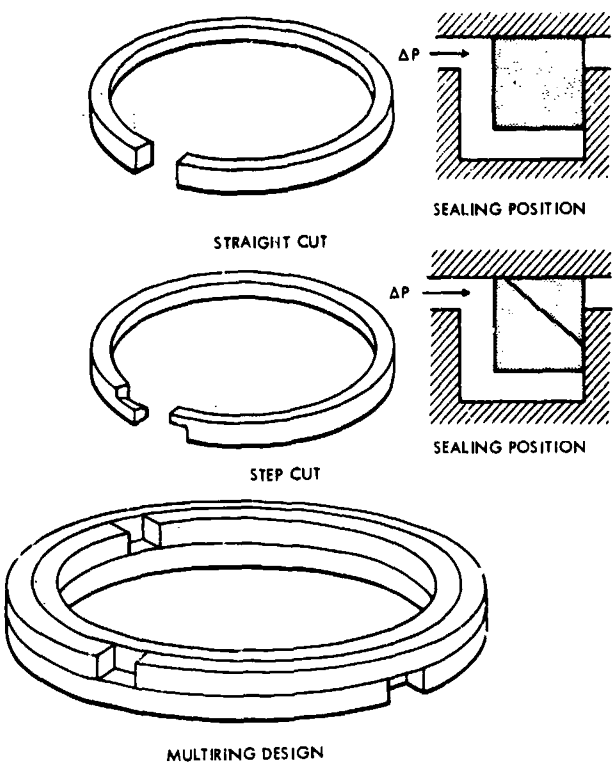


Figure 6.4.4.5a. Piston Ring Seal Gap Configurations
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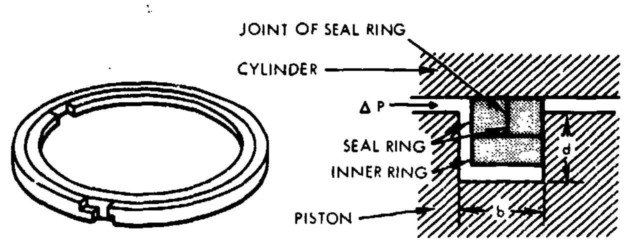

Figure 6.4.4.5b. Two-Piece Ring for Bi-Directional Sealing
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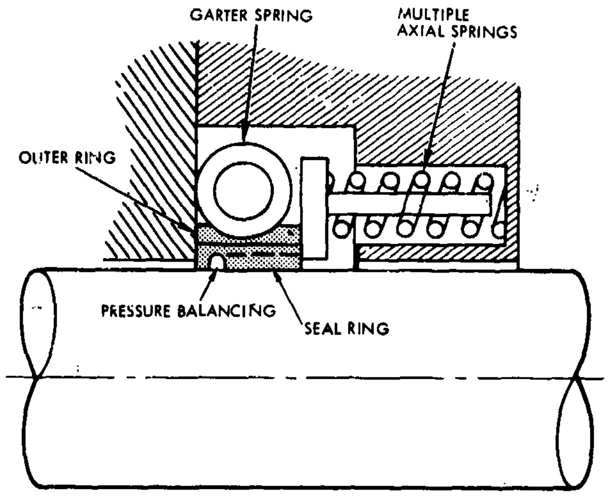

Figure 6.4.4.5c. Two-Piece Rod Seal with Pressure Balancing and Spring Loading
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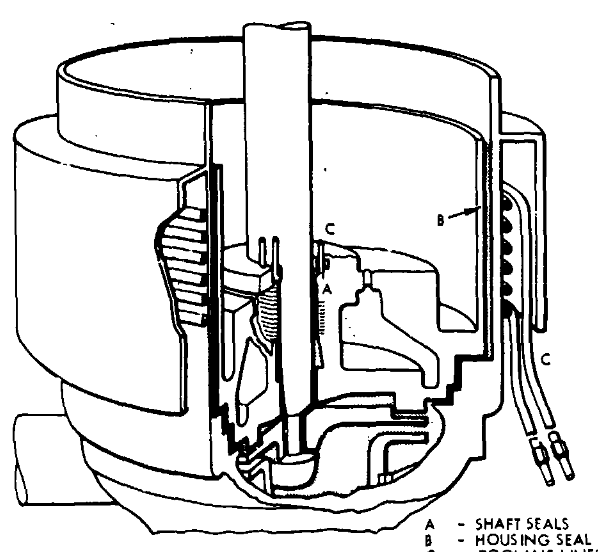

Figure 6.4.4.6. Freeze Seal Configuration
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Dynamic Sealing Theory

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135-1	

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1-[illegible]	[illegible]2-20
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106-6	[illegible]8
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6.5 SPRINGS

6.5.1 Introduction

A spring is any elastic device which, as its primary function, deflects under a load but regains its original shape after release of the load.

Functions of a spring as a fluid component include:

a) Calibrated force — springs with accurate force tolerances which act as force reference elements, typically used in pressure regulators, pressure switches, safety and relief valves, and many pressure sensitive devices.
b) Non-critical force — springs with wide force tolerances which may be dynamically operated as poppet loading springs or static in application as mechanical retainers.
c) Energy storage — springs designed to do work by alternate loading and releasing. Firing springs for ballistic devices and operating springs for quick shutoff valves are typical examples.
d) Vibration and shock isolation — highly damped mechanical springs or various forms of elastomeric springs frequently employed in isolation mounting of delicate pressure sensing devices.

Spring forms can be classified into four groups:

a) Mechanical springs based on the elastic properties of solids.
b) Liquid springs employing the bulk compressive characteristics of a confined liquid.
c) Pneumatic springs using the compressive characteristics of gases.
d) Elastomeric springs based on the elastic properties of rubber and other non-metals.

By far the most common form is the metal mechanical spring. A type of mechanical spring will fit virtually all requirements for loads and spring rates. Weight, volume, or other considerations may indicate the use of a liquid or pneumatic spring to be advantageous. Liquid springs give smaller envelope volumes for high force, high rate applications. Pneumatic springs are lighter for high displacement energy requirements. Elastomeric springs find applications for use on shock and vibration mounts where inherent high internal friction provides a useful damping function.

Treatment of spring design within this section is primarily directed to the fluid components designer. The objective is not to duplicate material which may be found in many good handbooks on mechanical design. It is rather to give the designer information helpful to the solution of spring problems common to fluid components. Useful sources of design data are referenced throughout the text. A particularly informative and comprehensive book on spring design is Reference 416-1.

6.5.2 Design Factors

The most difficult requirements for springs designed for flight duty fluid components include:

a) Meeting narrow tolerances for calibrated force output
b) Obtaining compatibility with severe environments
c) Achieving minimum weight and volume
d) Providing an assured operating life.

The optimum design for a given application must of necessity be a compromise of these requirements and others. The majority of applications fall within the capabilities of common helically wound wire springs. However, mechanical springs of special design may have advantages. For example, non-linear springs can at times be used to provide special load characteristics. The Belleville spring is a good example which has been applied to the extreme of producing a force reversal. Special springs often come into play for use as mechanical retainers and seal loading devices. Some forms of metal seals are essentially a complex form of spring.

A typical design approach useful for fluid component design should include the following steps:

a) Establish the general configuration. The type of spring best suited to a particular application is frequently somewhat obvious. The first thought is to use the simple mechanical helical compression spring. Occasionally envelope limitation dictates the use of a flat spring, Belleville, or other spring forms.
b) Determine the material. With consideration of the spring configuration established, the range of available materials must be reviewed and a selection made. The nature of environments and the breadth of acceptable tolerances combine to define the boundaries. For non-critical applications involving large production, material and fabrication costs may be a dominant factor, however, this is rarely true in aerospace flight appli-

cations where quantities are relatively low and performance is at a premium. Temperature extremes create the most severe material limitation. Availability of the required stock must be established for less common materials to avoid procurement delays.

c) Finalize the design. With steps (a) and (b) having established the boundaries, the detail design must be completed. If any latitude remains in either configuration or materials, some design exploration will be useful. At this stage, stress levels, weight and envelopes, fatigue and life, requirements and tolerances must be determined and resolved into a single configuration.

The following sections provide information and data useful in selecting a spring configuration.

6.5.2.1 MATERIALS. Any material capable of storing energy by elastic deformation can potentially be used to produce a spring. Solids, liquids, and gases have, in practice, found use in spring applications. By far the most frequently employed materials are high strength metals.

Metals. Metals with yield strengths as low as 40,000 psi are commonly considered usable as spring materials. In application, a number of characteristics are desirable and should be considered in determining the suitability of a metal. They include:

a) High yield strength
b) High elastic modulus
c) High fatigue strength
d) Good formability
e) Consistent metallurgical properties
f) Ability to meet severe environmental conditions.

As with other design elements, increasingly severe performance and environmental requirements for springs have resulted in a need for better materials. Consequently, the designer has been provided with an increasing selection of specialized materials in many available forms. The earlier, specially developed, carbon spring steels still find use in by far the largest proportion of applications and are the lowest in cost. Carbon steels are capable of excellent performance for high duty use requiring millions of load cycles. Valve springs in internal combustion engines are the most common example.

The most common requirements precluding the use of carbon steel are temperatures exceeding 300°F and resistance to corrosion. The stainless steels provide corrosion resistance and some alloys are usable at temperatures up to 600°F. Nickel alloys have been developed to be serviceable as high as 1000°F. Inconel X is the most common metal of this type. Presently, a number of super alloys are under investigation capable of temperatures exceeding 1000°F. Most of these are cobalt or nickel-cobalt alloys. A survey of these materials is presented in Reference 47-8.

A number of materials are available which have been developed to give high performance in a single characteristic such as constant elastic modulus over a given temperature range, low hysteresis and drift, non-magnetic properties, and high electrical conductivity. Table 6.5.2.1a presents typical materials to meet specifically desired spring characteristics.

Strength and elastic properties given as nominal for many materials are subject to wide variation in apparent usable strength. This is shown in the numerical ranges given in Table 6.5.2.1b for some alloys.

Factors influencing allowable stress are:

a) Variation in original alloy composition
b) Cold working, heat treatment, and tempering of the stock materials
c) Stock shape and size
d) Spring shape
e) Cold work, heat treatment, tempering, and stress relieving of the finished spring
f) Spring duty requirements involving fatigue and temperature extremes.

Table 6.5.2.1a. Materials for Special Spring Characteristics
(Reference 48-4)

CHARACTERISTICS	MATERIAL
Constant modulus	Ni-Span C alloy 902 Iso-elastic
Non-magnetic	Beryllium copper 172 Inconel X — 750 Phosphor Bronze 510 Monel K — 700
High electrical conductivity	Beryllium copper 10 Beryllium copper 172 Phosphor Bronze 510
Low hysteresis and drift	Beryllium copper 172 Ni-Span C alloy 902 Iso-elastic
Complex forming	Annealed spring steel Beryllium copper 172
Low cost	SAE 1050, 1075, 1095 Music wire
Stress carrying ability	SAE 1095 Music wire
Precision of manufacture and reliability	SAE 1095 Music wire
Very low temperatures	Beryllium copper 172 Inconel X — 750
Special corrosion resistance	Monel 400 Monel K — 500 Phosphor Bronze 510 Titanium AISI 301, 302, 17-4 PH Inconel X — 750

Table 6.5.2.1b. Strength and Elastic Properties of Spring Materials

MATERIAL	TENSILE PROPERTIES: ULTIMATE STRENGTH PSI x 10^3	TENSILE PROPERTIES: YIELD STRENGTH PSI x 10^3	TENSILE PROPERTIES: ELASTIC MODULUS, E PSI x 10^6	TORSIONAL PROPERTIES: ULTIMATE STRENGTH PSI x 10^3	TORSIONAL PROPERTIES: YIELD STRENGTH PSI x 10^3	TORSIONAL PROPERTIES: MODULUS OF RIGIDITY, G PSI x 10^6
Carbon Steels: Hard drawn spring wire SAE 1065	150 to 300	100 to 200	30	120 to 220	75 to 130	11.5
Oil tempered wire SAE 1065	155 to 300	120 to 250	30	115 to 200	80 to 130	11.5
Music wire SAE 1085	250 to 500	150 to 350	30	150 to 300	90 to 180	11.5
Valve spring wire ASTM A 230-47	200 to 230	—	30	—	—	11.5
Clock spring steel SAE 1095	180 to 340	150 to 260	30	not used	not used	not used
Flat spring steel SAE 1074	160 to 320	125 to 280	30	not used	not used	not used
Flat spring steel SAE 1050	160 to 280	120 to 180	30	not used	not used	not used
Alloy steels: Chromium-vanadium SAE 6150	200 to 250	180 to 230	30	140 to 175	100 to 130	11.5
Chromium-silicon SAE 9254	250 to 325	220 to 300	30	160 to 200	130 to 160	11.5
Silicon manganese SAE 9260	200 to 250	180 to 230	30	140 to 175	100 to 130	11.5
Stainless Steels: Anstenitic alloy AISI 302	160 to 330	60 to 260	28	120 to 240	45 to 140	10
Anstenitic alloy AISI 316	170 to 250	130 to 300	28	120 to 220	80 to 130	11
Precipitation hardening 17-7PH	—	60 to 200	29	—	—	11
18-8 type	160 to 330	60 to 260	28	120 to 240	45 to 140	10
Nickel alloys: Monel	120 to 165	85 to 125	26	85 to 110	50 to 70	9.5
K Monel	115 to 185	85 to 120	26	—	—	—
Duranickel	125 to 205	80 to 140	30	85 to 145	50 to 85	11
Inconel	140 to 185	110 to 140	31	95 to 130	55 to 80	11
Inconel X	130 to 220	90 to 150	31	90 to 155	50 to 90	11.5
Ni-Span C	200	110	27.5	—	—	9.8
Iso-Elastic (John Chatillon and Sons)	170	90 to 100 (working stress)	26	—	40 to 60 (working stress)	3.2

Table 6.5.2.1b. Strength and Elastic Properties of Spring Materials (Continued)

MATERIAL	TENSILE PROPERTIES:			TORSIONAL PROPERTIES:		
	ULTIMATE STRENGTH PSI x 10^3	YIELD STRENGTH PSI x 10^3	ELASTIC MODULUS, E PSI x 10^6	ULTIMATE STRENGTH PSI x 10^3	YIELD STRENGTH PSI x 10^3	MODULUS OF RIGIDITY, G PSI x 10^6
Nickel-cobalt alloys: René 41	—	—	—	—	—	—
Refractaloy 26	—	—	—	—	—	—
Waspaloy	—	—	—	—	—	—
Cobalt-based alloys: Elgiloy	300	200	29.5	—	—	—
Copper-based alloys: Spring brass CABRA 260	100 to 130	40 to 60	15	45 to 90	30 to 60	5.5
Phosphor bronze CABRA 510	100 to 150	60 to 110	15	80 to 105	50 to 85	6.25
Silicon bronze CABRA 655	100 to 150	60 to 110	15	80 to 105	50 to 85	6.25
Tin bronze CABRA 42[illegible]	—	45 to 75	16	—	—	—
Beryllium copper CABRA 172	160 to 200	100 to 150	16 to 18.5	100 to 130	65 to 95	6 to 7
Zirconium copper CABRA 150	48 to 59	39 to 53	19.3	—	—	—
Chromium copper CABRA 182	59 to 70	49 to 62	18	—	—	—
Copper-nickel CABRA 706	61 to 75	56 to 67	18	—	—	—
Nickel silver CABRA 770	92 to 125	60 to 90	18	—	—	—

Because of the extreme forming involved in manufacture of wire and flat sheet, a great amount of cold working and grain redistribution takes place. The effects of this processing vary with the size and shape of the stock produced, and influence the product even after heat treatment. In fabrication of the spring, forming of the stock again produces cold working which can be either beneficial or detrimental to apparent strength depending on whether it adds or subtracts from the stresses induced by the designed spring loading. Commonly used spring design equations in many instances do not establish peak stresses that will be developed. General practice is to apply a safety factor to allowable stress values to compensate for such uncertainties. Metallurgical processing of stock materials and the formed spring is a major factor in determining the characteristics of the finished product. The effects of processing may overshadow the importance of the initial material selection. For optimum quality springs, material processing must be tightly controlled from the mill throughout manufacturing to the finished product. Variations in material analysis, pre-treatment before final forming, and post forming treatments all affect the results. When performance must be pushed to the limit because of weight or space limitations, testing is required. Guides to material selection for springs can be found in References 19-92, 48-2, and 147-11.

Stock Dimensions. The availability of a desired material in the required stock dimension must be verified to insure quick delivery of a new design. In general, wires are stocked to standard gauges. Spring steel is generally available to all the Washburn and Moen gauge sizes. Some alloy steels may require special order and can be sized to requirements. Music wire and stainless steel wire are available in any decimal size. Non-ferrous alloy wires are generally

Table 6.5.2.1c. Decimal Equivalents of Standard Gauges
(Courtesy of Associated Spring Corporation, Bristol, Connecticut)

No. of Gauge	Washburn & Moen	Brown & Sharpe	Birmingham or Stubbs	U.S. Standard for Plate (Iron & Steel)	Stubbs Steel Wire	Imperial Wire Gauge	Morse Twist Drill and Steel Wire	Wood and Machine Screws	Amer. S. & W. Piano & Music Wire	No. of Gauge	Fractional Parts of an Inch with Decimal Equiv.
											1/64 = .01563
											1/32 = .03125
7-0										7-0	3/64 = .04688
6-0						.464				6-0	1/16 = .0625
5-0						.432			.005	5-0	5/64 = .07813
4-0	.394	.460	.454			.400			.006	4-0	3/32 = .09375
3-0	.362	.410	.425			.372		.032	.007	3-0	7/64 = .10938
2-0	.331	.365	.380			.348		.045	.008	2-0	1/8 = .125
0	.307	.325	.340	.313		.324		.058	.009	0	9/64 = .14063
1	.283	.289	.300	.281	.227	.300	.228	.071	.010	1	5/32 = .15625
2	.263	.258	.284	.265	.219	.276	.221	.084	.011	2	11/64 = .17188
3	.244	.229	.259	.250	.212	.252	.213	.097	.012	3	3/16 = .1875
4	.225	.204	.238	.234	.207	.232	.209	.110	.013	4	13/64 = .20313
5	.207	.182	.220	.219	.204	.212	.206	.124	.014	5	7/32 = .21875
6	.192	.162	.203	.203	.201	.192	.204	.137	.016	6	15/64 = .23438
7	.177	.144	.180	.188	.199	.176	.201	.150	.018	7	1/4 = .25
8	.162	.128	.165	.172	.197	.160	.199	.163	.020	8	17/64 = .26563
9	.148	.114	.148	.156	.194	.144	.196	.176	.022	9	9/32 = .28125
10	.135	.102	.134	.141	.191	.128	.194	.189	.024	10	19/64 = .29688
11	.120	.091	.120	.125	.188	.116	.191	.203	.026	11	5/16 = .3125
12	.105	.081	.109	.109	.185	.104	.189	.216	.029	12	21/64 = .32813
13	.092	.072	.095	.094	.182	.092	.185	.229	.031	13	11/32 = .34375
14	.080	.064	.083	.078	.180	.080	.182	.242	.033	14	23/64 = .35938
15	.072	.057	.072	.070	.178	.072	.180	.255	.035	15	3/8 = .375
16	.063	.051	.065	.063	.175	.064	.177	.268	.037	16	25/64 = .39063
17	.054	.045	.058	.056	.172	.056	.173	.282	.039	17	13/32 = .40625
18	.047	.040	.049	.050	.168	.048	.170	.295	.041	18	27/64 = .42188
19	.041	.036	.042	.044	.164	.040	.166	.308	.043	19	7/16 = .4375
20	.035	.032	.035	.038	.161	.036	.161	.321	.045	20	29/64 = .45313
21	.032	.028	.032	.034	.157	.032	.159	.334	.047	21	15/32 = .46875
22	.028	.025	.028	.031	.155	.028	.157	.347	.049	22	31/64 = .48438
23	.025	.023	.025	.028	.153	.024	.154	.360	.051	23	1/2 = .5
24	.023	.020	.022	.025	.151	.022	.152	.374	.055	24	33/64 = .51563
25	.020	.018	.020	.022	.148	.020	.150	.387	.059	25	17/32 = .53125
26	.018	.016	.018	.019	.146	.018	.147	.400	.063	26	35/64 = .54688
27	.0173	.0141	.016	.017	.143	.0164	.144	.413	.067	27	9/16 = .5625
28	.0162	.0126	.014	.016	.139	.0149	.141	.426	.071	28	37/64 = .57813
29	.015	.0112	.013	.014	.134	.0136	.136	.439	.075	29	19/32 = .59375
30	.014	.010	.012	.013	.127	.0124	.129	.453	.080	30	39/64 = .60938
31	.0132	.0089	.010	.011	.120	.0116	.120	.466	.085	31	5/8 = .625
32	.0128	.0079	.009	.010	.115	.0108	.116	.479	.090	32	41/64 = .64063
33	.0118	.007	.008	.009	.112	.010	.113	.492	.095	33	21/32 = .65625
34	.0104	.0063	.007	.0086	.110	.0092	.111	.505	.100	34	43/64 = .67188
35	.0095	.0056	.005	.0078	.108	.0084	.110	.518	.106	35	11/16 = .6875
36	.009	.005	.004	.007	.106	.0076	.1065	.532	.112	36	45/64 = .70313
37		.0044		.0066	.103	.0068	.104	.545	.118	37	23/32 = .71875
38		.0039		.0062	.101	.006	.1015	.558	.124	38	47/64 = .73438
39		.0035			.099	.0052	.0995	.571	.130	39	3/4 = .75
40		.0031			.097	.0048	.098	.584	.138	40	49/64 = .76563
41					.095		.096	.597	.146	41	25/32 = .78125
42					.092		.094	.611	.154	42	51/64 = .79688
43					.088		.089	.624	.162	43	13/16 = .8125
44					.085		.086	.637	.170	44	53/64 = .82813
45					.081		.082	.650	.180	45	27/32 = .84375
46					.079		.081	.663		46	55/64 = .85938
47					.077		.079	.676		47	7/8 = .875
48					.075		.076	.690		48	57/64 = .89063
49					.072		.073	.703		49	29/32 = .90625
50					.069		.070	.716		50	59/64 = .92188
											15/16 = .9375
											61/64 = .95313
											31/32 = .96875
											63/64 = .98438
											1 = 1.000

It is preferable to specify flat spring steel in decimals instead of gauge numbers to avoid all error of misunderstanding. When gauge numbers must be used in specifying flat spring steel the Birmingham gauge should be used.
These tables are theoretically correct but variations must be expected in practice.

available to the Brown and Sharpe gauge sizes. See Table 6.5.2.1c for decimal equivalents of these standards. Flat spring material is generally available in a variety of thicknesses and strip widths.

6.5.2.2 FORCE STABILITY. In many fluid components and instruments the output of a continuously loaded spring provides a calibrated reference force. To maintain a constant force over a period of time and under extremes of temperature is primarily a metallurgical requirement.

Factors affecting force stability are:

a) Dissipation of internal stresses
b) Creep
c) Changes in elastic and rigidity moduli with temperature
d) Thermal expansion
e) Corrosion
f) Hysteresis.

6.5.2.3 STRESS RELAXATION AND CREEP. Internal residual stresses are produced by cold working during drawing or rolling of the stock material, or in the forming and pre-setting operations of spring manufacture. The distribution of the residual stress is generally not beneficial to the working of the spring. With time, these stresses will relax, causing a decrease in the load the spring can bear under constant deflection. A stress-relieving process is usually induced prior to application and involves heating the spring to a pre-determined temperature for one-half hour or less. Stress-relieving improves metallurgical properties and makes the spring more resistant to intergranular corrosion and fatigue. Permanent relaxation occurs because of gradual relief of applied and residual stresses. The applied stresses will relieve if the spring is continuously loaded at temperature. The residual stresses relieve at temperature even though the spring is relaxed.

Pre-setting is done after stress relieving. This process involves loading the spring to a stress level above the normal working stress for a number of cycles. The effect is to redistribute stresses within the working section by local yielding in the highest stressed areas. The resulting residual stresses in this case act to reduce peak stresses developed during normal operation. Consequently, they make the spring more load stable.

Tests conducted over a ten year period (Reference 65-33) have shown that room temperature coil spring load relaxation and compression set can be predicted based on a 168 hour (one week) test. Equilibrium load and compression set can be empirically predicted by doubling load loss and compression set values measured at the end of one week with the test spring at room temperature under service load.

A phenomenon known as anelastic flow which causes a temporary relaxation is also experienced. The loss in output force from this effect is fully recoverable, if the spring is unloaded and the temperature level is maintained. The time for recovery approximately equals the period of loading. Figure 6.5.2.3a shows this behavior for Inconel X. Anelastic flow can be minimized by heat setting. This treatment involves heating the spring to a temperature higher than the expected operating temperature while held under load. A part of the relaxation expected in service is thereby eliminated in advance. If unloaded at temperature, the spring will recover force with time just as if the

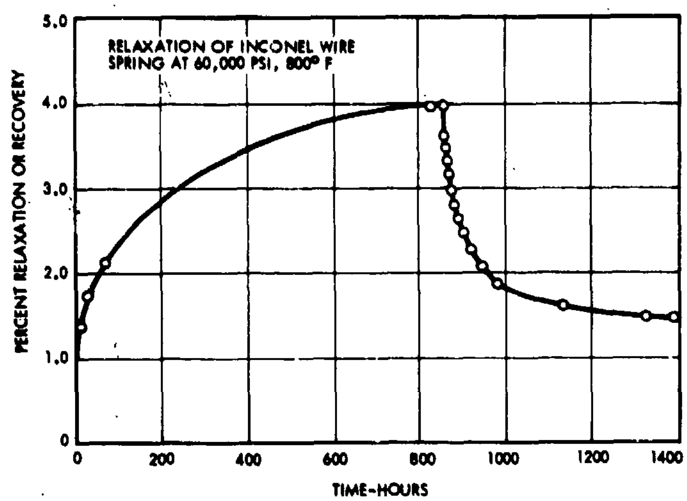


Figure 6.5.2.3a. Anelastic Flow Behavior of Inconel Coil Spring at 800°F and Stressed to 60,000 Psi

(From "Electromechanical Components and Systems Design," W. R. Johnson, vol. 2, no. 9, August-September 1958, Copyright 1958, Benhill Publishing Corp., Brookline, Massachusetts)

relaxation had taken place under operation. Figure 6.5.2.3b gives the effects of relaxation with time for Inconel X.

The continuous deflection of a constantly stressed spring over long periods of time (many hours at least) is known as creep. This behavior is most apparent at elevated temperatures but can be significant at normal temperatures after operation or storage under load for months or years. As a rule, it may be expected that creep expressed as a percentage of the initial deflection will be greater than the percentage loss in load, providing the initial stresses are the same, since with creep the load continually decreases.

The effect of temperature on stress relaxation and creep is depicted in Figure 6.5.2.3c. At the indicated creep rate, the spring will continue to deflect under constant load, and stress relaxation will proceed asymptotically to a defined load loss.

6.5.2.4 ELASTIC MODULUS AND THERMAL EXPANSION. For most materials the elastic and rigidity moduli decrease as temperature increases. With many materials this effect is very non-linear, and some materials exhibit abrupt changes in behavior over a temperature range. The elastic moduli of several typical spring materials are shown as a function of temperature in Figure 6.5.2.4. Data on moduli of rigidity and elasticity for a number of materials are presented in Reference 82-8. Elastic modulus changes are often the predominant cause of spring load changes and necessitate the use of compensating methods for many applications.

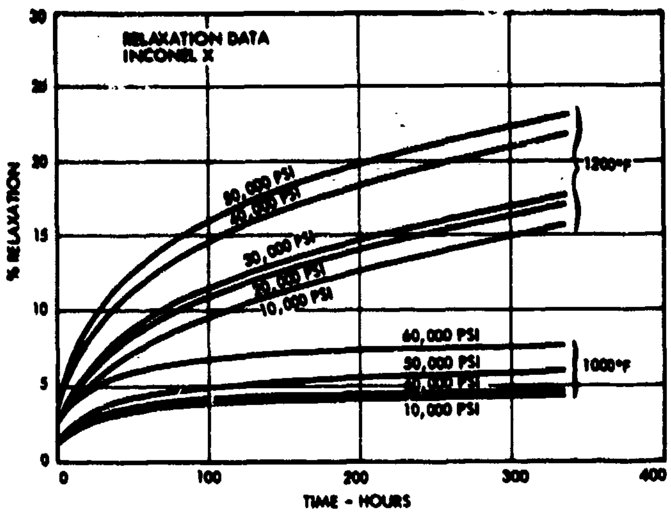


Figure 6.5.2.3b. Inconel X Relaxation Data

(From "Electromechanical Components and Systems Design," W. R. Johnson, vol. 2, no. 9, August-September 1958, Copyright 1958, Benhill Publishing Corp., Brookline, Massachusetts)

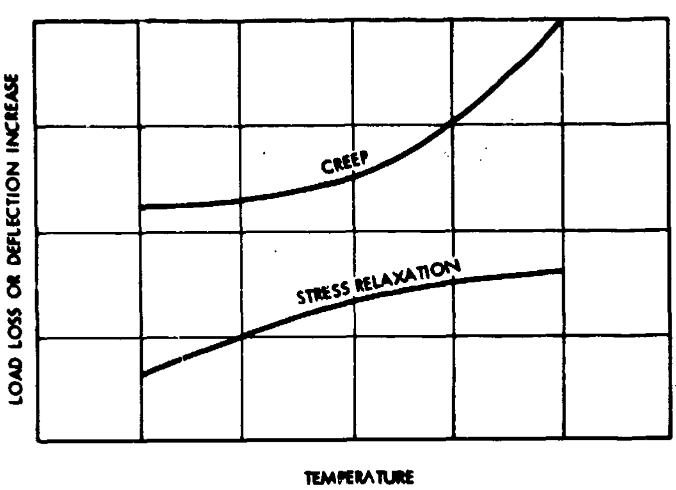


Figure 6.5.2.3c. Effect of Temperature on Creep or Stress Relaxation

Thermal expansion is generally of minor significance to spring force changes except in very high rate applications. The significance of thermal dimensional changes is determined by the material and configurations of the spring and supporting structure. In general, the higher the spring rate involved and the longer the load paths through the structure, the greater the effect of thermal expansion for given sets of materials and temperature range. By contrast, the effect of temperature change on spring force due to changes in elastic modulus of the spring material are essentially independent of configuration. The force change is proportional to the percentage change of elastic modulus within the given temperature change.

The effect of temperature change on the force of calibrated springs must be considered for any mechanism requiring

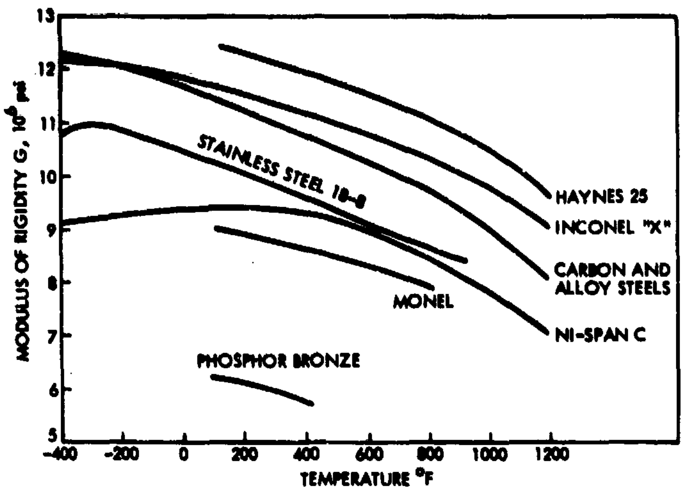


Figure 6.5.2.4. Modulus of Rigidity for Several Metals as Functions of Temperature

(Data from References 82-8 and 416-1)

very narrow force tolerances. In fluid components, the most common application is in pressure-sensing elements of such devices as pressure regulators, flow controllers, or pressure switches.

Preliminary design calculations will determine if temperature effects are significant and, if required, a compensation method can be established. Because of the complexity of thermal effects when temperature gradients exist, temperature compensation often requires refinement during development testing.

6.5.2.5 HYSTERESIS. Internal hysteresis limits the repeatability of a given load point with increasing or decreasing force transients. Hysteresis for most materials does not exceed 0.2 percent of total deflection in a helical spring design. Ni-Span C and Iso-elastic alloys provide very low hysteresis in the order of one tenth of this value. The apparent spring hysteresis in many installations is caused by friction in the mounting bearing surfaces or between spring elements. Such friction must be eliminated before internal hysteresis can be considered significant.

6.5.2.6 FATIGUE. The effect of metal fatigue should be considered for springs required to operate in excess of 1000 cycles under a significant load. Springs requiring 1000 to 500,000 maximum load cycles can be considered medium duty springs. Some reduction in allowable stress below values for static applications is likely to be required. Springs demanding cycle life upward from 500,000 cycles can be classed as high duty springs and require special care in design and fabrication.

Establishing allowable stresses for endurance loaded springs is complex because of the interaction of material, material treatment, shape factors such as spring index for helical springs, and difficulty in obtaining useful data. The working stress range as well as peak working stress must be considered in evaluating fatigue life requirements. Most available specific data is for helical springs. The following design concept refers to helical coil springs.

Springs for static loading and up to 1000 life cycles can be simply designed based on uncorrected (Wahl factor) stresses. Figures 6.5.2.6a and 6.5.2.6b provide stress data for compression and tension springs, respectively.

For medium duty applications, corrected stresses should be used in design and the fatigue characteristics of the material should be considered in establishing a working stress (Figure 6.5.2.6c).

For design of high duty springs, corrected stresses should be used and in addition the stress range must be considered. Figure 6.5.2.6d is a typical endurance diagram which can be used to establish allowable maximum stresses. The stress range is calculated for the application based on corrected stresses. The upper and lower values obtained are applied to a vertical line on the chart. The lower value will lie on the minimum stress line. The upper value cannot exceed the maximum stress value given for the material in question. For high duty applications where failure would be hazardous or damaging, testing of the design is mandatory.

The ability of a material to resist fatigue failure is primarily related to its surface structure in highly stressed areas. Fatigue failures originate at the surface and are only secondarily related to the bulk character of the material. The focal center of failure is always at a microscopic discontinuity in the surface intergranular structure.

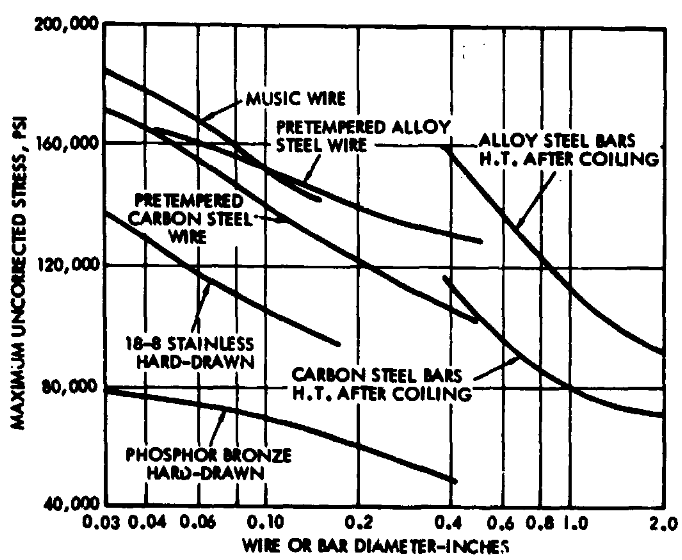


Figure 6.5.2.6a. Stresses in Compression Springs for Ordnance Applications Where Space is Limited and Only Short Life is Required

(From Kent's "Mechanical Engineers' Handbook," Design and Production Volume, 12th ed., Copyright 1950, John Wiley and Sons, Inc., New York, New York)

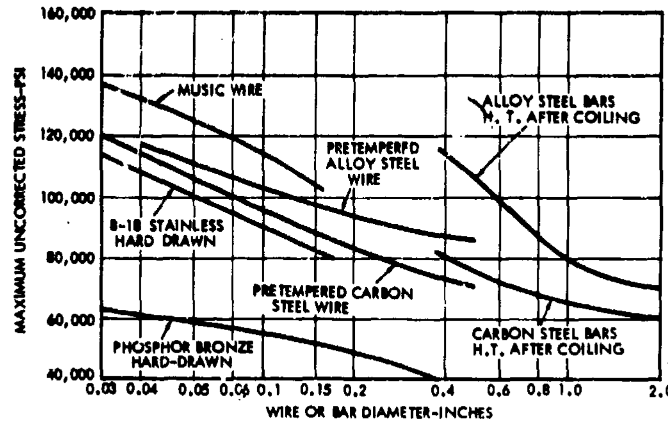


Figure 6.5.2.6b. Stresses for Tension Springs in Ordnance Applications Where Space is Limited and Where Only Short Life is Required

(From Kent's "Mechanical Engineers' Handbook," Design and Production Volume, 12th ed., Copyright 1950, John Wiley and Sons, Inc., New York, New York)

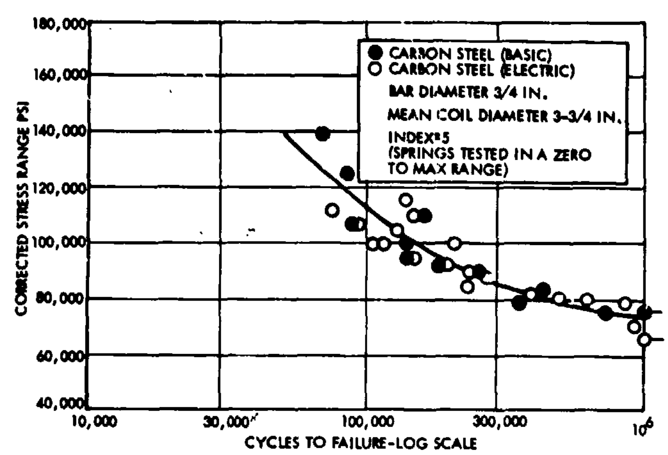


Figure 6.5.2.6c. Results of Fatigue Tests of Hot-Wound Heavy Duty Helical Springs

(From Kent's "Mechanical Engineers' Handbook," Design and Production Volume, 12th ed., Copyright 1950, John Wiley and Sons, Inc., New York, New York)

Factors determining the susceptibility of a material to fatigue are:

a) The intergranular characteristic of the material surface and presence of flaws.
b) Stress concentration effects due to rough finish tooling marks and "designed in" stress risers.
c) The inherent ability of a material to compensate for high concentrated stresses by local yielding.

A fine grained metal surface provides the best resistance to fatigue failure. Proper heat treatment is of utmost im-

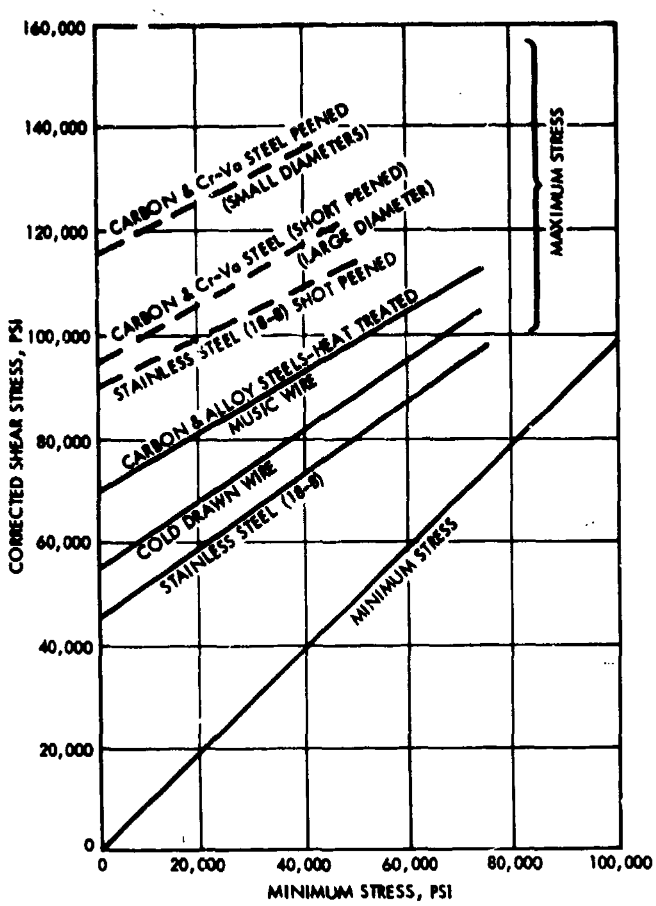


Figure 6.5.2.6d. Approximate Endurance Diagram for Helical Compression Springs
(Reprinted from Kent's "Mechanical Engineers' Handbook," Design and Production Volume, 12th ed., Copyright 1950; John Wiley and Sons, Inc., New York, New York)

portance in obtaining a good surface free of cracks and roughness. Surface decarburization must be avoided, and materials must be inherently free of inclusions, voids, and other surface imperfections. Stress concentrations caused by either tool marks or design configuration are to be avoided in high duty springs. Grooves are especially damaging when transverse to the direction of loading. Stress concentration factors should be applied to allowable stresses when sharp corners or severe bends cannot be avoided. Flat springs of complex design and end loops of extension springs require special care in design. Chemical surface treatments for cleaning and plating must be carefully controlled to avoid hydrogen embrittlement.

Shot peening is frequently employed to increase fatigue life. The process produces cold work in the metal surface, improving the surface properties. In addition, residual compressive stresses are established at the surface, resulting in a reduction in net applied tensile stress. Shot peening can be applied to spring wires of one-sixteenth diameter and larger. The results of shot peening are lost at temperatures high enough to stress relieve. The process is, therefore, not useful for springs to be subjected to elevated temperatures. Shot peening is not equally effective on all materials; applied to corrosion-resistant alloys it can create corrosion sensitive areas capable of reducing fatigue strength. Figure 6.5.2.6e gives an example of the increase in allowable stress range possible by use of shot peening for some common materials. The initial stress is the stress resulting from winding the coil spring.

The bulk characteristics of metal, although not necessarily related to its surface character, do play a part in fatigue strength. High strength, brittle metals are more susceptive to fatigue failure. In part, this is due to a greater sensitivity to heat treating difficulties causing surface cracks. The more commonly significant factor involves the inability of such materials to redistribute local high stresses by yielding with a resulting increase in the significance of stress concentration.

6.5.2.7 CORROSION. Minor surface penetration due to corrosion can cause extensive percentage loss in the working cross section of small springs. In addition, the outer surfaces invariably carry the highest stresses and are most sensitive to the surface-roughening effects of corrosion. Fatigue strength can be radically affected by a very small amount of corrosion.

Plating of carbon steel springs can be considered under mildly corrosive conditions for larger springs (section over one-quarter inch) where the stresses are relatively low and changes in force or failure would not be cata-

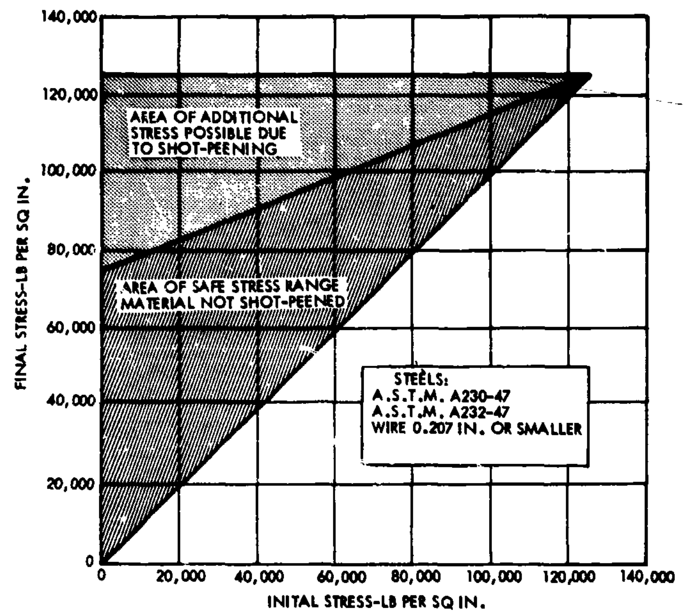


Figure 6.5.2.6e. Stress Range Increase Due to Shot Peening
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 22)

strophic. When strengths higher than those available with fully corrosion-resistant materials must be obtained, the use of semi-corrosion resistant materials and plating is the best choice. Plating on springs must be well bonded and able to deflect with the outer surface of the spring. Thick, brittle coatings are to be avoided. Care must be exercised in the plating of springs to insure that the cleaning and plating process do not result in hydrogen embrittlement or other adverse effects at the surface of the spring material (see Sub-Section 13.7 for further information on corrosion).

6.5.2.8 DYNAMIC LOADING. The effects of dynamic motion must be considered when a spring is subjected to vibrational or cyclic loading, or when individual high velocity load impulses are involved. For fluid components, dynamic spring problems can arise in the design of high response valves or mechanisms subject to load vibrations or rapid oscillating motions. Vibrational loading can result in severe dynamic spring stresses, if the input frequency is resonant with the primary natural frequency or a harmonic frequency of the spring.

Equations for the natural frequency of compression springs with both ends clamped and with one end free are given in Table 6.5.2.8. Frequency relationships for spring-mass systems are given in Sub-Section 7.3.

To reduce stresses due to resonant vibrations, the natural frequency of the spring should be made as high as possible above the input frequency. When this is not possible, the primary resonant frequency of the spring should be as far as possible from the input frequency. Reducing the pitch of the end coils will tend to keep the spring from resonating. Damping is useful in eliminating resonant vibration. This may be done by use of special friction damping devices working with the spring coils, or by application of springs with inherent high working friction such as volute springs or stacked Belleville springs. For some applications, liquid or pneumatic springs which have high damping can be substituted. Liquid springs can be designed with varying amounts of damping.

Applications involving suddenly applied or released loads require evaluation of such factors as spring velocity under free expansion and conditions which will cause a spring to lift off its base. The reaction time for springs to be used in high response mechanisms must be established. Methods based on energy concepts are simple to apply and are useful in estimating reactions of this type. A rigorous method of analysis based on surge wave theory capable of evaluating many complex modes of dynamic spring loading has been developed in References 19-3 and 19-7. By establishing the velocity of the surge wave traveling through a spring, and by analyzing the interaction of the wave and the boundary forces and velocities acting on the spring, the accurate prediction of spring performance and peak stresses can be established by this method.

Table 6.5.2.8. Natural Frequency of Vibration of Coil Springs

Both Ends of Spring Clamped	One End of Spring Free
General Equation	
$n = \frac{d}{\pi D^2 N}\sqrt{\frac{48.2G}{w}}$	$n = \frac{d}{2\pi D^2 N}\sqrt{\frac{48.2G}{w}}$
Equations for Steel Springs	
$n = \frac{14100d}{D^2N}$	$n = \frac{7050d}{D^2N}$

where

- n = frequency, cycles per second
- N = number of active coils
- D = mean coil diameter, inches
- d = wire diameter, inches
- w = specific weight of material, pounds per cubic inch
- G = torsional modulus of elasticity, psi

6.5.3 Design Data

6.5.3.1 HELICAL COMPRESSION SPRINGS. A compression spring is an open helical coil that offers resistance to a compressive force. The basic design factors for compression springs are the working loads and spring rate, sometimes called scale. Envelope limitations and environmental conditions affect their configuration, stress level, and material selection.

A typical design sequence should include the following steps:

a) Establish requirements for:
   - Loads and tolerances
   - Spring rate
   - Envelope limits
   - Environments
   - Duty characteristics

b) Select material based on environment, loading, and cycle life

c) Establish a workable configuration from equations, spring tables or a spring slide rule

d) Review the assumed design for:
   - Stress when compressed solid
   - Stress at the maximum working load
   - Buckling stability

e) For medium or high duty springs and calibrated springs consider:
   - Stress correction (Wahl) factor
   - Fatigue strength
   - Dynamic loading
   - Manufacturing tolerances

The most common helical spring configuration is made of round wire, with ends ground for squareness. The type of ends available as standard for compression springs are shown in Figure 6.5.3.1a. For special cases, other wire sections may be used. Square wire will provide higher load carrying capacity for a given envelope volume. Sections

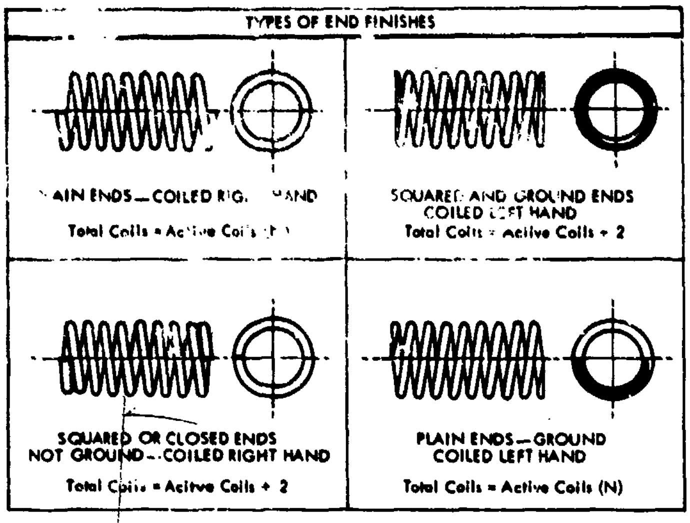


Figure 6.5.3.1a. Types of Ends for Compression Springs
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 18)

other than round are less efficient in torsion and therefore result in higher weight for a given force. Nested springs are generally a better solution to minimize envelope volume.

The common equations for helical compression and extension springs may be found in Table 6.5.3.1a. The Wahl correction factor is applied to stress calculations for medium and high duty springs in order to account for the effects of linear shear and wire curvature. The Wahl factor is larger for springs of low index (ratio of mean coil diameter to wire diameter) (Figure 6.5.3.1b).

The design of a double helical spring by computer methods is presented in Detailed Topic 8.3.3.4. Table 6.5.3.1b provides a quick method of establishing proportions of helical springs, and is directly usable in design of noncritical springs. For high stressed springs, or medium and high duty, stress and deflection of the selected design can be checked by equations.

The table is based upon:

a) Loads developing 100,000 psi torsional stress corrected for shear and curvature (Wahl factor)

b) Shear modulus of 11.5×10^6 psi.

For materials exhibiting other values, the loads and spring scale can be proportionally adjusted by the following relationships:

(Eq 6.5.3.1a)

$$\text{Load required} = \frac{\text{Allowable working stress of materials}}{100{,}000} \times (\text{Load given in Table 6.5.3.1a})$$

Table 6.5.3.1a. Helical Compression and Extension Spring Equations—Round Wire

Uncorrected Stress

$$S = \frac{8FD}{\pi d^3} = \frac{2.55\,FD}{d^3}$$

Corrected Stress $S' = KS$

Spring Index

$$C = \frac{D}{d}$$

Wahl stress correction

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

Deflection

$$\delta = \frac{8FD^3N}{Gd^4}$$

Spring rate

$$k = \frac{F}{\delta} = \frac{Gd^4}{8\,D^3N}$$

Diametral growth (no unwinding) $\Delta D = 0.05\,\frac{P^2 - d^2}{D}$

Diametral growth (unwinding)

$$\Delta D = 0.10\,\frac{P^2 - 0.8\,pd - 0.2d^2}{D^2}.$$

where

F = Load, lb_f
D = Mean coil diameter, inches
d = Wire diameter, inches
S = Uncorrected torsional stress, psi
S' = Corrected stress, psi
δ = Deflection, inches
N = Number of active coils
G = Modulus of rigidity (torsional modulus), psi
K = Wahl correction factor
k = Spring rate, lb_f per inch
C = Spring index
P = Pitch (center to center distance of coils at free height), inches

(Eq 6.5.3.1b)

$$\text{Spring rate required} = \frac{\text{Shear modulus of material}}{11.5 \times 10^6} \times (\text{Spring rate given in Table 6.5.3.1b})$$

The load values given in the table may be converted to those for an uncorrected stress of 100,000 psi by multiplication by the Wahl correction factor in Figure 6.5.3.1b. The allow-

Table 6.5.3.1b. Load and Spring Scale per Coil—Heli

H. F. Ross, Trans. ASME, Oc
(Based on 100,000 psi — corr
(From Kent's "Mechanical Engineers' Handbook," 12 ed., C

Wire Diam., d	Outside Diameter of Coil, in. 1/8	5/32	3/16	7/32	1/4	9/32	5/16	11/32	3/8	13/32	7/16	15/32	1/2	17/32	9/16	5/8	11/16
.010	.305 9.47	← Load in lb at 100,000 psi—corrected stress ← Spring scale per single coil															
.011	.401 14.1	.325 6.90															
.012	.522 20.7	.421 10.0	.350 3.43														
.014	.823 40.3	.669 19.4	.560 10.5	.480 6.45	.422 4.18												
.016	1.21 72.9	.983 34.4	.824 18.4	.714 11.3	.626 7.37	.561 5.05											
.018	1.71 124	1.40 57.4	1.17 30.7	1.01 18.6	.889 12.1	.793 8.30											
.020	2.32 200	1.89 91.3	1.60 48.6	1.39 29.3	1.22 18.9	1.08 12.8	.972 9.06										
.022	3.03 306	[illegible].50 140	2.12 73.9	1.83 44.0	1.60 28.2	1.44 19.3	1.30 13.7	1.18 10.1									
.024	3.90 464	3.22 208	2.72 108	2.36 64.3	2.09 41.4	1.87 28.2	1.68 19.9	1.53 14.6	1.41 11.1	1.31 8.62	1.21 6.67	1.14 5.42					
.026	4.88 678	4.06 299	3.44 155	2.98 91.1	2.63 58.4	2.35 39.6	2.14 28.0	1.94 20.5	1.79 15.5	1.65 12.0	1.54 9.43	1.44 7.55	1.35 6.18	1.28 5.11			
.029	6.61 1160	5.55 496	4.71 253	4.10 149	3.64 94.1	3.25 63.7	2.94 44.6	2.71 32.9	2.50 24.6	2.29 18.9	2.13 14.9	2.00 12.0	1.87 9.75	1.77 8.07	1.66 6.66		
.032		7.30 793	6.25 398	5.47 231	4.84 146	4.35 97.8	3.95 68.0	3.60 49.9	3.33 37.5	3.07 28.7	2.85 22.4	2.67 17.9	2.51 14.8	2.37 12.1	2.24 10.0	2.03 7.27	
.035		9.34 1213	8.10 604	7.10 346	6.32 217	5.66 145	5.14 101	4.72 74.0	4.32 55.0	4.02 42.2	3.73 33.0	3.50 26.3	3.29 21.4	3.08 17.7	2.93 14.7	2.64 10.4	2.41 7.72
.039		12.6 2066	10.9 1009	9.66 572	8.63 354	7.76 234	7.03 162	6.45 118	5.93 87.9	5.52 67.3	5.14 52.8	4.80 41.1	4.52 34.1	4.27 28.0	4.06 23.3	3.66 16.6	3.33 12.2
.043			14.4 1618	12.7 901	11.4 559	10.3 368	9.33 249	8.60 183	7.91 134	7.36 103	6.82 80.0	6.43 63.7	6.07 51.6	5.69 42.6	5.36 34.9	4.89 25.0	4.44 18.3
.047			18.3 2507	16.4 1378	14.7 [illegible]	13.3 547	12.1 373	11.1 270	10.3 199	9.54 151	8.90 117	8.33 93.3	7.83 75.6	7.39 61.9	6.99 51.0	6.35 36.2	5.81 26.6
.051				20.4 2061	18.5 1233	16.8 800	15.3 543	14.2 389	13.0 288	12.1 217	11.3 168	10.6 134	10.0 107	9.45 88.3	8.97 73.0	8.01 51.1	7.37 37.5
.055					22.8 1767	20.8 1143	18.9 768	17.6 553	16.2 401	15.1 304	14.1 234	13.3 187	12.4 149	11.7 122	11.2 101	10.0 70.4	9.23 51.9
.063						30.3 2196	27.9 1461	25.8 1032	23.8 721	22.4 562	20.9 429	19.7 338	18.6 272	17.5 221	16.6 181	15.1 128	13.8 92.6
.071							38.7 2563	36.2 1783	33.4 1295	31.5 972	29.4 741	27.8 580	26.1 463	24.8 375	23.6 307	21.3 215	19.5 156
.080								50.3 3245	47.1 2300	44.1 1705	41.3 1285	39.2 1000	36.9 795	34.5 640	33.4 525	30.2 364	27.8 262
.090									65.2 4075	61.3 2990	57.9 2236	54.8 1734	51.7 1368	49.2 1101	46.9 890	42.9 617	39.0 442
.100										81.8 5018	77.5 3726	73.5 2860	69.7 2248	[illegible] 1800	63.3 1449	58.0 993	53.0 707
.112												100 4950	95.6 3886	91.5 3081	87.4 2462	79.7 1678	73.8 1186
.125													130 6701	124 5232	119 4205	109 2817	101 1977
.135															147 6080	135 4060	125 2820
.148																175 6350	163 4380
.162																	207 6800

Wire Diam., d	Outside Diamet 3/4	13/16	7/8	15/16	1	1 1/16	1 1/8
.039	3.04 9.30	2.82 7.20	← Load in lb at 100,000 p ← Spring scale per single c				
.043	4.10 14.0	3.76 10.8	3.52 8.60				
.047	5.39 20.1	4.93 15.6	4.57 12.4	4.27 9.89	4.00 8.09		
.051	6.78 28.6	6.29 22.1	5.79 17.3	5.44 14.0	5.14 11.4	4.82 9.43	
.055	8.50 39.5	7.83 30.1	7.32 23.9	6.77 19.0	6.40 15.6	6.02 12.9	5.70 10.7
.063	12.7 70.0	11.7 55.7	10.9 42.5	10.2 33.7	9.60 27.5	9.00 22.6	8.55 18.8
.071	18.0 117	16.6 88.9	15.5 70.6	14.5 56.2	13.6 45.3	12.9 37.4	12.2 31.5
.080	25.7 196	23.7 150	22.2 117	20.8 93.7	19.5 75.6	18.3 62.1	17.3 51.4
.090	36.2 329	33.6 230	31.2 195	29.4 155	27.6 125	26.1 103	24.7 85.1
.100	49.2 525	45.6 397	42.6 309	40.1 245	37.6 197	35.5 161	33.5 132
.112	68.1 871	63.5 657	59.3 509	55.7 4[illegible]1	52.2 323	49.6 263	47.0 218
.125	93.8 1436	87.2 1083	81.9 832	76.8 651	72.5 525	68.2 426	65.0 330
.135	116 2035	109 1535	102 1176	96.3 919	90.8 739	85.5 600	81.4 493
.148	151 3166	142 2343	133 1802	125 1403	119 1119	112 896	106 741
.162	195 4887	183 3581	172 2734	162 2112	153 1685	146 1353	139 1114
.177	250 7485	235 5504	222 4150	209 3215	198 2526	188 2035	178 1664
.192		293 8184	277 6128	262 4712	249 3694	236 2950	226 2407
.207			341 8857	322 6765	307 5266	293 4210	279 3419
.225				408 10175	388 7918	371 6246	353 5072
.244					484 11805	461 9257	444 7462
.250						494 10466	474 8389
.283							
.3125							
.343							
.362							
.375							
.394							
.4375							

Spring Scale per Coil—Helical Compression or Extension Springs

H. F. Ross, Trans. ASME, October 1947
(Based on 100,000 psi — corrected stress)
Engineers' Handbook," 12 ed., Copyright 1950, John Wiley and Sons, Inc.)

Outside Diameter of Coil, in.											
7/8	15/16	1	1 1/16	1 1/8	1 3/16	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8
← Load in lb at 100,000 psi—corrected stress ← Spring scale per single coil											
3.52 / 8.60											
4.57 / 12.4	4.27 / 9.89	4.10 / 8.07									
5.79 / 17.3	5.44 / 14.0	5.14 / 11.4	4.82 / 9.43								
7.32 / 23.9	6.77 / 19.0	6.40 / 15.6	6.02 / 12.9	5.70 / 10.7							
10.9 / 42.3	10.2 / 33.7	9.60 / 27.5	9.00 / 22.6	8.55 / 18.8	8.07 / 15[illegible]	[illegible]					
15.5 / 70.6	14.5 / 56.2	13.6 / 45.3	12.9 / 37.4	12.7 / [illegible]	[illegible]	[illegible]	[illegible]	[illegible]			
22.2 / 117	20.8 / 93.7	19.5 / 75.6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
31.2 / 195	29.4 / 155	27.6 / 125	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
42.6 / 309	40.1 / 245	37.6 / 197	35.5 / 161	[illegible]	[illegible] / 111	50.[illegible] / 94.5	[illegible] / 69.0	[illegible]	[illegible]	[illegible]	20.4 / 25.5
59.3 / 509	55.7 / 401	52.2 / 323	49.6 / 263	47.0 / 218	44.6 / 182	42.4 / 152	38.6 / 112	35.8 / 84.5	[illegible] / 65.4	30.7 / 51.7	28.8 / 41.5
81.9 / 832	76.8 / 651	72.5 / 525	68.2 / 426	65.0 / 350	61.7 / 292	58.8 / 246	53.9 / 180	49.4 / 135	45.7 / 104	42.6 / 82.0	39.9 / 65.6
102 / 1176	96.3 / 919	90.8 / 739	85.5 / 600	81.4 / 493	77.2 / 410	73.5 / 344	67.2 / 251	62.2 / 188	57.6 / 145	53.3 / 113	50.0 / 90.7
133 / 1802	125 / 1403	119 / 1119	112 / 896	106 / 741	101 / 611	96.3 / 514	88.0 / 375	81.2 / 279	75.7 / 214	70.5 / 168	65.7 / 133
172 / 2734	162 / 2112	153 / 1685	146 / 1353	139 / 1114	131 / 918	126 / 769	115 / 554	106 / 414	98.5 / 317	91.5 / 247	86.1 / 196
222 / 4150	209 / 3215	198 / 2526	188 / 2035	178 / 1664	170 / [illegible]	163 / 1145	149 / 819	137 / 609	128 / 468	119 / 361	112 / 289
277 / 6128	262 / 4712	249 / 3694	236 / 2950	226 / 2407	215 / 1982	205 / 1640	190 / 1184	174 / 873	162 / 666	151 / 515	142 / 408
341 / 8857	322 / 6765	307 / 5266	293 / 4210	279 / 3419	267 / 2796	255 / 2320	235 / 1658	217 / 1218	202 / 930	188 / 716	177 / 569
	408 / 10175	388 / 7918	371 / 6245	353 / 5072	339 / 4109	323 / 3407	299 / 2415	276 / 1770	258 / 1344	241 / 1043	226 / 822
		484 / 11805	461 / 9257	444 / 7462	426 / 6051	406 / 4988	376 / 3498	349 / 2562	326 / 1935	306 / 1488	287 / 1179
			494 / 10466	474 / [illegible]	455 / 6842	436 / 5640	401 / 3953	375 / 2874	349 / 2169	328 / 1667	307 / 1306
					639 / 12432	614 / 10199	574 / 7086	534 / 5154	499 / 3836	467 / 2919	440 / 2281
							753 / 11444	706 / 8143	662 / 6051	623 / 4611	586 / 3586
							972 / 18101	915 / 12833	858 / 9429	806 / 7139	761 / 5535
								1058 / 16794	997 / 12293	940 / 9207	896 / 716[illegible]
								1166 / 19966	1099 / 14556	1038 / 10903	980 / 8426
									1262 / 18641	1190 / 13967	1127 / 10602
										1[illegible]37 / 23202	1504 / 17715

Wire Diam., d	Outside Diameter of Coil, in.																
	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8	3 1/4	3 3/8	3 1/2	3 3/4	4	4 1/4	4 1/2
.100	19.3 / 20.9	18.1 / 17.2	← Load in lb at 100,000 psi—corrected stress ← Spring scale per single coil														
.112	27.0 / 33.5	25.4 / 27.7	24.1 / 23.1														
.125	37.6 / 53.3	37.7 / 51.3	33.4 / 36.6	31.6 / 30.8	30.2 / 26.2	28.7 / 22.5											
.135	47.2 / 73.9	44.3 / 60.9	41.9 / 50.4	40.0 / 42.5	37.9 / 36.2	36.0 / 31.0	34.5 / 26.6										
.148	62.0 / 109	57.9 / 88.8	55.1 / 74.3	52.6 / 62.8	49.7 / 53.1	47.7 / 45.6	45.3 / 39.1	43.3 / 34.1	41.7 / 29.7	40.0 / 26.2							
.162	80.8 / 161	76.1 / 131	72.0 / 109	68.4 / 90.9	65.1 / 77.9	62.3 / 66.4	59.1 / 56.9	57.2 / 50.0	54.6 / 43.5	52.2 / 38.0	50.5 / 33.5	48.7 / 30.0	46.8 / 26.7				
.177	105 / 234	99.1 / 191	94.0 / 159	89.5 / 133	84.7 / 113	81.1 / 96.7	77.2 / 83.3	74.3 / 72.3	71.0 / 63.2	68.3 / 54.9	65.5 / 48.4	63.0 / 43.0	61.2 / 38.4	57.0 / 30.9			
.192	134 / 331	126 / 269	120 / 226	113 / 186	108 / 160	103 / 135	98.1 / 117	94.1 / 101	90.2 / 88.6	86.9 / 77.2	83.6 / 68.6	80.8 / 60.4	78.0 / 54.2	72.5 / 43.5	68.4 / 35.5		
.207	166 / 457	158 / 375	150 / 310	141 / 258	134 / 218	129 / 187	122 / 160	118 / 139	113 / 121	109 / 107	104 / 93.6	101 / 83.1	97.2 / 74.1	91.2 / 59.5	85.2 / 48.5	80.7 / 40.1	76.0 / 33.5
.225	212 / 659	201 / 536	191 / 445	181 / 371	173 / 314	165 / 266	157 / 229	151 / 198	144 / 173	139 / 151	134 / 134	129 / 118	124 / 104	116 / 83.6	110 / 68.6	103 / 56.5	97.9 / 47.2
.244	269 / 941	255 / 764	241 / 631	229 / 526	218 / 443	208 / 376	199 / [illegible]22	192 / 280	183 / 243	177 / 213	170 / 188	165 / 167	158 / 148	148 / 118	139 / 95.9	132 / 79.4	125 / 66.7
.250	290 / 1051	273 / 853	260 / 700	247 / 586	235 / 492	225 / 419	[illegible]	205 / 310	198 / 270	191 / 237	183 / 208	177 / 184	170 / 163	159 / 131	150 / 106	141 / 87.9	134 / 73.3
.283	415 / 1820	393 / 1474	374 / 1210	355 / 1007	338 / 849	322 / 718	[illegible]	296 / 530	285 / 460	276 / 405	263 / 352	256 / 314	247 / 276	228 / 219	216 / 180	204 / 148	194 / 123
.3125	555 / 2842	525 / 2313	499 / 1892	473 / 1568	453 / 1311	432 / 1085	413 / 947	397 / 816	381 / 705	368 / 617	355 / 541	341 / 478	330 / 424	309 / 338	290 / 273	274 / 224	258 / 187
.343	725 / 4370	684 / 3515	655 / 2869	620 / 2375	593 / 1982	570 / 1675	544 / 1425	52[illegible] / 1226	501 / 1059	483 / 925	466 / 810	451 / 714	433 / 633	408 / 503	382 / 404	362 / 333	341 / 278
.362	844 / 565[illegible]	801 / 4518	765 / 36[illegible]5	729 / 3026	692 / 2535	665 / 2131	634 / 1814	613 / 1555	[illegible] / 134[illegible]	567 / 1171	547 / 1025	525 / 905	508 / 800	478 / 634	452 / 518	426 / 423	403 / 350
.375	932 / 6652	883 / 5281	844 / 4302	804 / 3551	767 / 2960	737 / 2497	704 / 2124	679 / 1820	652 / 1575	[illegible]	605 / 1195	585 / 1053	567 / 933	529 / 739	497 / 597	471 / 492	445 / 406
.394	1068 / 8350	1022 / 6684	974 / 5420	926 / 4476	884 / 3714	849 / 3117	816 / 2656	781 / 2268	750 / 1955	722 / 1704	701 / 1[illegible]97	677 / 1307	651 / 1159	612 / 917	575 / 740	542 / 604	515 / 503
.4375	1433 / 13859	1363 / 11010	1307 / 8861	1249 / 7257	1201 / 6005	1148 / 5013	1102 / 424[illegible]	1063 / 3644	1019 / 3132	980 / 2718	951 / 2367	919 / [illegible]82	888 / 1835	835 / 1450	783 / 1166	738 / 952	704 / 785
.500	2023 / 26645	1976 / 20999	1897 / 16817	1819 / 13718	1742 / 11268	1675 / 9358	1617 / 7880	1555 / 6694	1500 / 5736	1451 / 4971	1396 / 4325	1356 / 3783	1310 / [illegible]	1229 / 2619	1160 / 2097	1092 / 1705	1041 / 1405
.5625			2623 / 29977	2508 / 24232	2426 / 19853	2[illegible] / 16388	2251 / 13751	2177 / 11654	2096 / 9957	2020 / 8341	1956 / 7412	1898 / 6471	1832 / 5686	1727 / [illegible]442	1628 / 3546	1541 / 2868	1468 / 2359
.625					3239 / 33289	3115 / 27445	3010 / 22872	2921 / 19268	2825 / 16386	2722 / 14045	2649 / 12135	2565 / 10551	2491 / 9240	23[illegible] / 7192	2221 / 5710	2101 / 4608	1997 / 3773

B

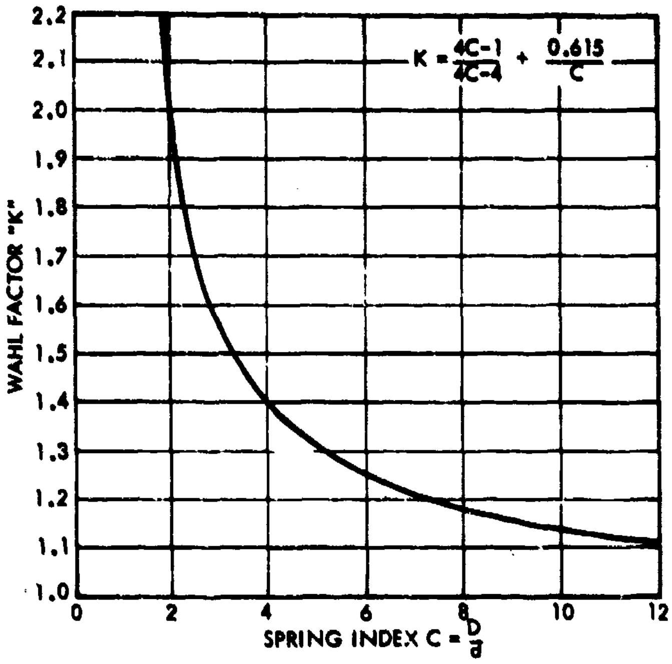


Figure 6.5.3.1b. Wahl Factor for Helical Compression and Extension Springs
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 19)

able working stresses for helical springs is presented in Table 6.5.3.1c for several metals.

If compression springs are made long compared to their diameter, instability may occur due to column action under load. The size of the spring end and the initial squareness of a spring influence the degree of instability. Two types of buckling occur—the first type with fixed ends, the second type with hinged ends. These springs are shown in Figure 6.5.3.1c. The critical deflection at which instability occurs for each type depends on the ratio of free length of the mean diameter (L_o/D), and can be found in Figure 6.5.3.1c. For springs with fixed ends, no buckling will occur when compressed to its solid height if the L_o/D ratio is less than approximately 5.3. For hinged ends, the ratio is reduced to 2.65. An analytical treatment of buckling is presented in Reference 19-9.

The use of guides may be necessary in some designs, when the critical buckling proportions cannot be avoided. The clearance between guide and spring should be kept small, but not so small as to cause rubbing when the spring diameter enlarges during deflection. The amount of diametral growth can be calculated from equations presented in Table 6.5.3.1a. Friction is created by the use of guides and may be unacceptable for designs requiring load reproducibility. Lubrication is generally required. For extremely long springs, several springs can be stacked end-to-end with cylindrical guides between sections. The overall spring rate for springs in series can be calculated by

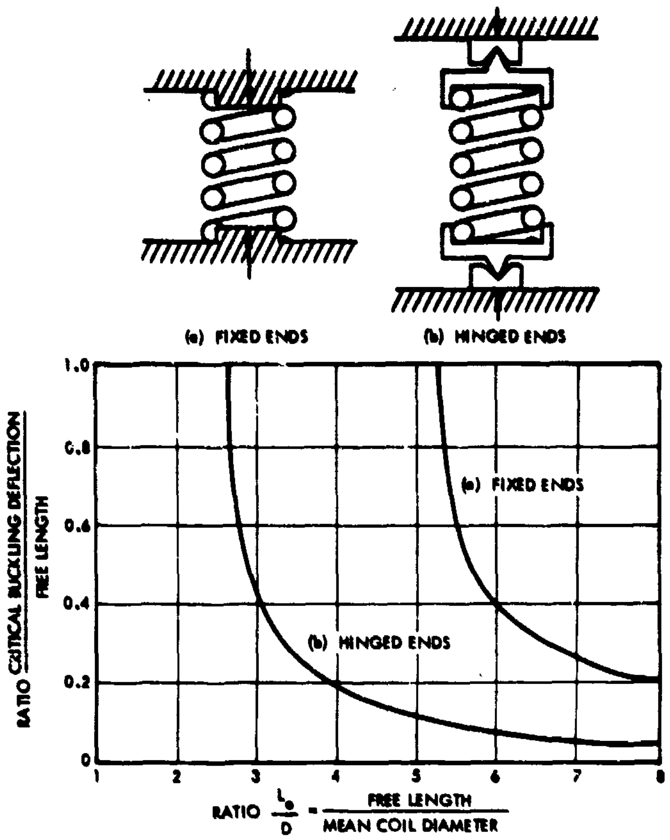


Figure 6.5.3.1c. Buckling of Springs with Either Fixed or Hinged Ends
(Reprinted from Kent's "Mechanical Engineers' Handbook," Design and Production Volume, 12th ed., Copyright 1950. John Wiley and Sons, Inc., New York, New York)

(Eq 6.5.3.1c)

$$\frac{1}{k_T} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \cdots$$

where k_T = total spring rate and $k_1 \cdots$ = individual spring rate

6.5.3.2 HELICAL EXTENSION SPRINGS. An extension spring is a close-coiled helical spring that resists a pulling force. The functional principles and equations which apply to compression springs also apply to extension springs, with an added consideration for initial tension. The initial tension demands that an initial load be applied to the spring before the spring coils start to separate. The initial tension or pre-load in the closed position must be formed into the spring during manufacture. The magnitude depends on the manufacturing technique used. The initial load, F_i, required can be determined from the equation

Table 6.5.3.1c. Allowable Working Stress
(For Occasional Loading)

MATERIAL	WIRE SIZE (IN.)	CORRECTED STRESS (K psi)	UNCORRECTED STRESS (K psi)
Music Wire	0.015	180	150
	0.060	155	129
	0.156	128	107
Chrome-vanadium	0.040	157	131
	0.100	140	117
	0.500	115	96
Carbon valve spring wire	0.020	164	137
	0.100	127	106
	0.500	92	77
18-8 Stainless wire	0.025	122	102
	0.150	93	78
	0.500	70	58
Phosphor-bronze	0.025	77	64
	0.150	65	54
	0.500	46	38
Music wire	0.085	60	—
	0.085-0.185	55	—
Valve spring wire	0.185	48	—
	0.320	42	—
	0.530	36	—
	0.970	32	—
18-8 Stainless wire	0.085	45	—
	0.185	42	—
	0.320	36	—
	0.530	27	—
Phosphor bronze	0.085	30	—
	0.185	27	—
	0.320	24	—
	0.530	18	—

$$F_1 = \frac{2\pi S_t d^3}{16 D} \quad \text{(Eq 6.5.3.2)}$$

where d = wire diameter, in.

D = mean coil diameter, in.

S_t = torsional stress, psi

Figure 6.5.3.2a provides data on stress limitations for initial tension. Extension springs have no definite stop to their deflection as do the solid height of compression springs, and must be protected from being over-extended.

The allowable working stresses for extension springs, applicable to Equations (6.5.3.1b) and (6.5.3.1c) and Table 6.5.3.1b are presented in Table 6.5.3.2.

Many forms of spring ends are used. Figure 6.5.3.2b provides typical end configurations. Loop ends are the least costly. Stress concentrations produced by small radius bends in the end configurations must be considered.

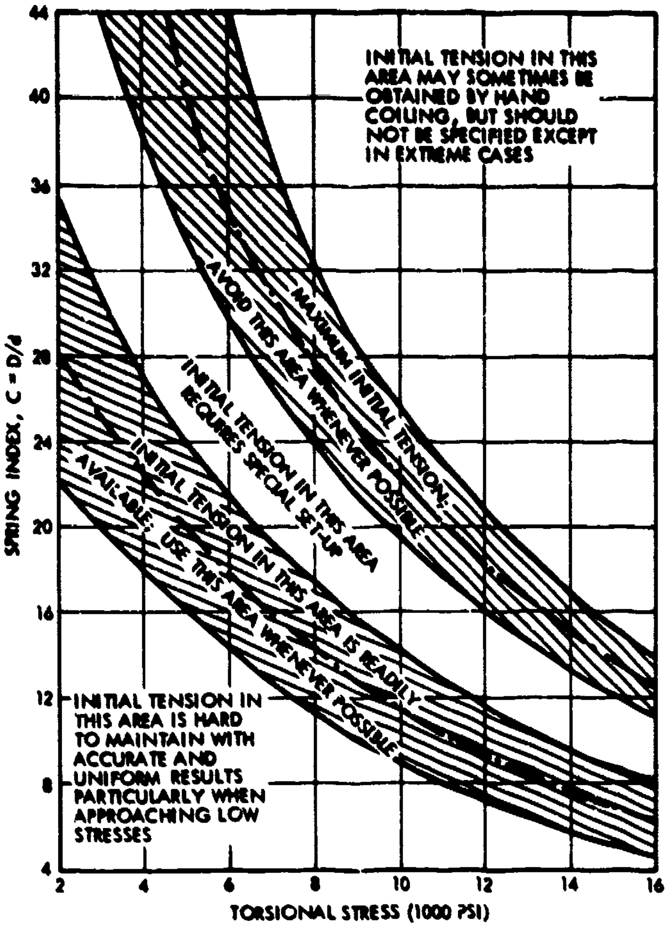


Figure 6.5.3.2a. Stresses Produced By Initial Tension In Extension Springs
(From "Machine Design," H. J. Boll, vol. 30, no. 8, April 17, 1958, Copyright 1958 by the Penton Publishing Company, Cleveland, Ohio)

Table 6.5.3.2. Allowable Working Stress

MATERIAL	WIRE SIZE (In.)	UNCORRECTED STRESS (K psi)
Music wire	0.03	138
	0.05	128
	0.10	117
Pretempered carbon steel wire	0.05	115
	0.10	102
	0.20	92
	0.30	90
18-8 stainless steel wire	0.05	105
	0.01	95
	0.10	90
Phosphor bronze	0.05	60
	0.08	58
	0.10	56
	0.30	44

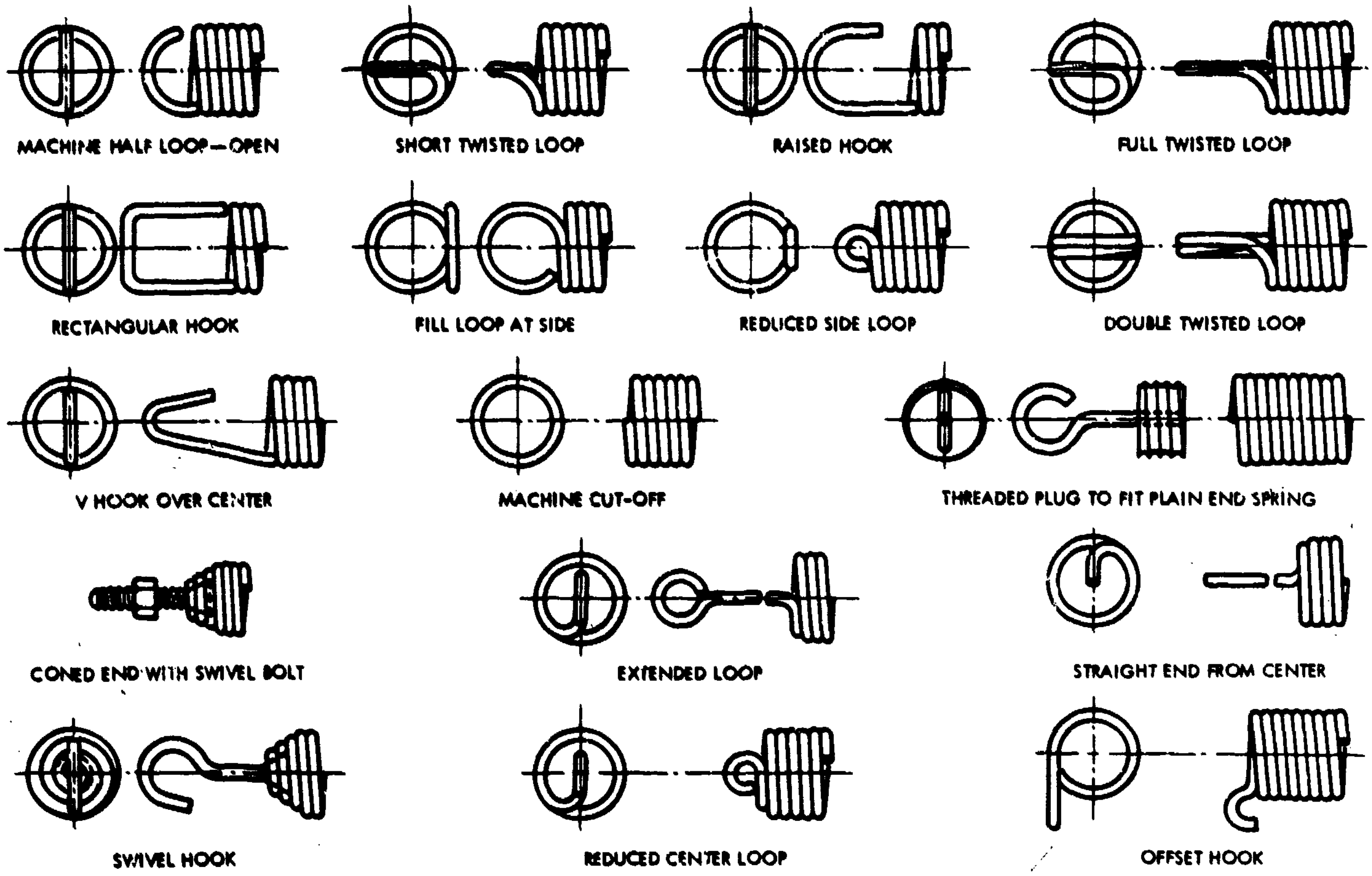


Figure 6.5.3.2b. Typical End Configurations for Extension Springs
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 34)

6.5.3.3 NESTED SPRINGS. Helical compression springs can be nested (Figure 6.5.3.3), to decrease the working stress while keeping constant

a) The load at the compressed height of the spring
b) The load-deflection rate
c) The overall dimensions of height and outside diameter.

Other potential benefits include:

a) Reduction in fatigue stress by reduction of the working stress range. The benefit gained is proportional to the reduction maximum working stress and is a function of a spring index.
b) The use of smaller diameter wire allows higher maximum working stresses.
c) Large spring index tends to reduce the Wahl correction factor.

The conventional approach for design of concentric nested springs is to

a) Assume a ⅔—⅓ load split, respectively, between outer and inner springs for a 2-spring nest; and a load split, outer spring 4/7, middle spring 2/7, and inner spring 1/7 for a 3-spring nest.
b) Assume a mean diameter for the outside or inside coil.
c) Calculate a trial wire size based on uncorrected stress.
d) Determine remaining dimensions by trial and error methods.
e) Re-adjust dimensions until an equal stress distribution is achieved between the springs.

When the Wahl correction factor must be considered, the calculations can be tedious and time consuming. References 1-123 and 19-94 give direct methods for establishing nested designs.

Suggested diametral clearances between springs are: ⅛ inch for outside diameter of smaller springs up to 3 inches; 3/16 inch for outside diameter of smaller spring from 3 to 5 inches; and ¼ inch for smaller springs over 5 inches. Adjacent springs should be coiled in opposite directions to prevent interlocking.

The spring rate for a nested spring system is the sum of the

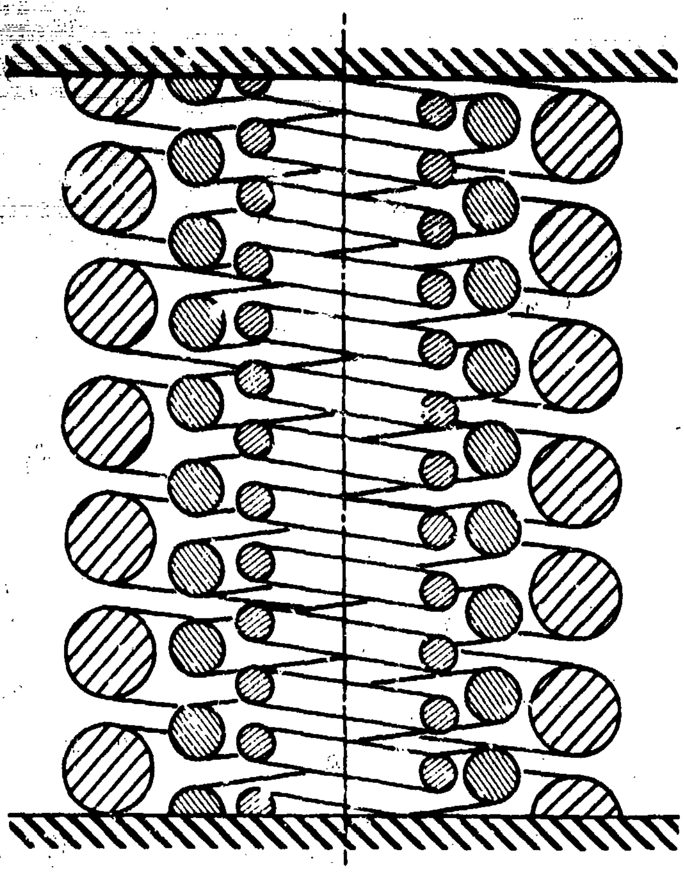


Figure 6.5.3.3. Nested Spring Design
(From "Mechanical Springs," A. M. Wahl, 1963, Second Edition, McGraw-Hill Publishing Company)

spring rates of individual springs. More details of nested spring designs can be found in Reference 19-94.

6.5.3.4 HELICAL TORSION SPRINGS. As the name indicates, torsion springs produce a torsional output. Unlike compression and extension springs, they are not stressed in torsion but in bending. The common equations are given in Table 6.5.3.4. They are based upon the fundamental beam stress and deflection relationships. For high stressed applications or if sharp bends exist, stress correction factors (Figure .5.3.4) should be applied for the effect of curvature in springs of low index.

If a torsion spring is used around an arbor or rod, sufficient decrease must be allowed, because the torsion spring will wind up and decrease its diameter with load. The change in diameter can be calculated by the equations in Table 6.5.3.1b. If the angular deflection is 1/6 turn and the spring has 4 turns, the diametral change is 1/6:4, or approximately four percent.

If the torsion springs are long, buckling can occur at about ⅞ turn for small free length to coil diameter ratios, L_0/D, and about 1.25 turns for long, L_0/D. Axial compressive loads will reduce these values.

Table 6.5.3.4. Helical Torsion Spring Equations

Moment per turn (round wire)	$Ma = \frac{M}{T} = \frac{Ed^4}{10.2DN}$
Moment per turn (rectangular wire)	$Ma = \frac{Ebh^3}{6.6DN}$
Stress (round wire)	$S = \frac{FR\frac{d}{2}}{\frac{\pi d^4}{64}} = \frac{32M}{\pi d^3}$
Stress (rectangular wire)	$S = \frac{FR\frac{h}{2}}{\frac{bh^3}{12}} = \frac{6M}{bh^2}$
Angular deflection	$\theta = \frac{10.2\ MND}{Ed^4}$ (turns)

where

D = Mean coil diameter, in.
d = Wire diameter, in.
E = Modulus of elasticity (Young's modulus), psi
I = Moment of inertia, in^4
Ma = Moment per turn, in-lb_f
M = Moment, in-lb_f
T = Number of spring rotational turns
N = Number of active coils
R = Mean radius lever arm, in.
F = Load, lb_f
S = Stress, psi
b = Width of wire section, in.
h = Thickness of wire section, in.
θ = Angular deflection, turns

6.5.3.5 TUBULAR COILED SPRINGS. The use of tubular material to produce a helical spring provides the following advantages:

a) Lighter weight for a given load capacity, spring rate or energy storage capacity
b) Increased release velocity
c) Higher natural frequency
d) Capability of internal fluid cooling for application in a high temperature environment
e) Usable as a fluid carrying element between spring supported parts.

Although weight is reduced by use of tubular material, the envelope volume is increased for a given load, spring rate, or energy capacity. Torsional crippling is a factor in design which must be considered in determining minimum usable wall thickness. Design of ends and fabrication also present special problems with this form of spring.

Stress and deflection equations for tubular springs are given

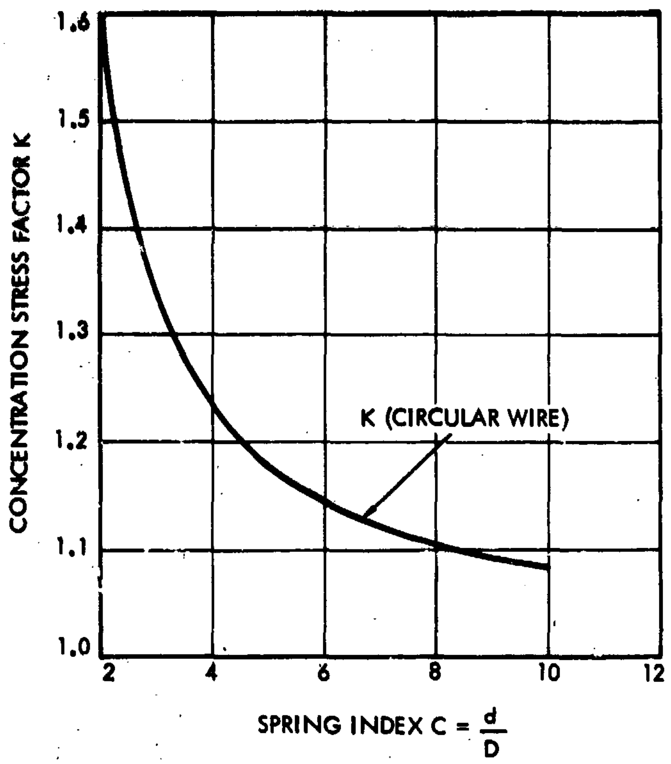


Figure 6.5.3.4. Stress Concentration Factors
(From "Mechanical Springs," A. M. Wahl, 1963, Second Edition, McGraw-Hill Publishing Company)

in Table 6.5.3.5a. As noted, stress calculated by the normal equation for solid wire, including the Wahl correction factor, is usable for design and will yield conservative results. The equation for deflection includes a deflection factor, K_δ, which is significant for thin walled tubular springs.

Relationships for prediction of torsional shear crippling, release velocity, and natural frequency may be found in Reference 1-54. Table 6.5.3.5b provides a weight comparison for a specific application of a tubular spring in place of a nested spring pair.

6.5.3.6 FLAT SPRINGS. In terms of the spring manufacturer, almost any spring made of flat stock can be classed as a flat spring. In general, "U" springs and leaf springs are in this class. Power springs made of coiled flat strips are usually included as a special class although flat spring design principles apply.

Most applications for flat springs involve beam deflection. Some of the common equations for small deflection and single leaf are given in Table 6.5.3.6. For complex shapes, principles of structural stress analysis must be used. As with all sheet metal design, the effect of holes, corners, tool marks, and bends are critical to the life of the part.

Both pre-tempered and annealed stock are used for flat springs. Pre-tempered material is limited in formability

Table 6.5.3.5a. Tubular Section Helical Springs

Stress

$$S_{nom} = \frac{8FD}{\pi d^3(1-B^4)}$$

$$S_{max} = \left[K\,(1+e) + \frac{p^2}{2} \right] \frac{8FD}{\pi d^3\,(1-B^4)}$$

Deflection

$$\delta = \frac{K_\delta F D^3 N}{G d^4 (1-B^4)}$$

$$K_\delta = 1 - \frac{3}{16C^2} + \frac{3B^2}{8C} + 1.27\,P^2$$

where

F = Load, lb_f
k = Tangent of helix angle
B = Ratio of inside diameter to outside diameter
C = Spring index, $\frac{D}{d}$
D = Mean coil diameter, in.
d = Tube outside diameter, in.
S_{nom} = Nominal shear stress, psi
S_{max} = Maximum combined shear stress, psi
p = Tangent of helix angle
e = Load eccentricity ratio to mean coil radius
δ = Axial deflection, in.
G = Modulus of rigidity, psi
N = Number of active coils
K = Wahl factor
K_δ = Deflection factor

Table 6.5.3.5b. Weight Comparison
(From "Machine Design," D. W. Best, vol. 32, no. 4, Copyright 1960 by Penton Publishing Company, Cleveland, Ohio)

	Solid-Spring Nest		Single Tubular Spring*
	Outer Spring	Inner Spring	
d (in	0.500	0.375	0.570
D (in.)	4.15	3.10	4.05
C	8.30	8.27	7.10
Total turns	13	14	11
H (in.)	15.75	13.06	14.12
KS_{nom} (psi) †	109,000	109,000	120,000
F_{max} (lb)	1576		1700
Total weight (lb)	11.75		7.50

*$B = 0.509$
† At Maximum load

Table 6.5.3.6. Flat Spring Equations

Cantilever type

Stress

$$S = \frac{Mc}{I} = \frac{6FL}{bh^2}$$

Deflection

$$\delta = \frac{FL^3}{3EI} = \frac{2SL^2}{3Eh}$$

Elliptical type

Stress

$$S = \frac{Mc}{4I} = \frac{3FL}{2bh^2}$$

Deflection

$$\delta = \frac{FL^3}{48EI} = \frac{SL^2}{6Eh}$$

Rectangular column type

Stress

$$S = 2.63\,\frac{Ehb^2}{L^2}$$

Load to cause buckling

$$F = 0.823\,\frac{Ehb^3}{L^2}$$

Load to cause column
Ends to become parallel

$$F = 1.15\,\frac{Ehb^3}{L^2}$$

Spring rate

$$K = 0.6\,\frac{Ehb^3}{L^3}$$

Where

M = Bending moment, in.-lb_f
C = Distance from neutral axis to outer fiber, in.
I = Moment of inertia, in.^{4}
S = Stress, psi
F = Load, lb_f
L = Spring length, in.
E = Modulus of elasticity, lb_f/in.^{2}
b = Spring thickness, in.
h = Spring width, in.
δ = Deflection, in.

and hardness; hence, it is only usable for designs requiring limited forming and less than maximum strength. Heat treatment of parts formed from annealed stock presents some difficulty because of distortion of thin materials. A final truing operation is often required to hold close tolerances.

6.5.3.7 BELLEVILLE SPRINGS. The term Belleville spring is applied to a class of truncated conical spring washers. Applications fall into two classes:

a) Use in high spring rate applications with small relative deflections where axial space is limited.
b) Applications using the nonlinear and "snap action" characteristics of a load-deflection curve.

Load-deflection characteristics can be obtained by varying the proportions of outside diameter to inside diameter, relative thickness and cone angle. Figure 6.5.3.7a provides typical load-deflection curves for the useful spring range. For $\frac{h}{t}$ ratios less than 0.4 the spring action is linear within 2½ per cent up to the flat position. This is the range useful for application (a) above. If the $\frac{h}{t}$ ratio is greater than 1.41, the load reaches a peak then decreases with further deflection. At a point beyond the flat position, the load increases again. Up to an $\frac{h}{t}$ ratio of 2.83, a positive load is required to deflect the spring. For values greater than 2.83, the load becomes negative at a point beyond the flat position. Loading in the opposite direction must be applied to return the spring. The $\frac{h}{t}$ range between 1.3 and 1.6 is usable for springs providing relatively constant force.

Design equations for Belleville springs are presented in Table 6.5.3 7. More detailed treatment can be found in Reference 416-1. Most methods of design involve a trial and error calculation. In general, they involve calculating or obtaining from a plot stress and deflection constants based on t and the $\frac{D}{d}$ ratio. The values obtained are used in the stress equation to obtain a deflection, h, for a given stress. This is then used in the load equation. If the load value is below that required, the assumed proportions are changed and the calculation repeated. The load at any deflection for various height/thickness ratios is presented in Figure 6.5.3.7c.

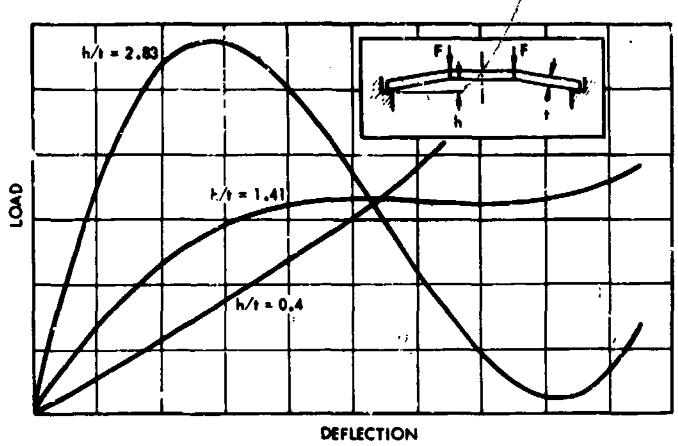


Figure 6.5.3.7a. Typical Belleville Spring Load-Deflection Curves Obtained by Varying h/t

For most applications in the non-linear load regions, some development testing is required to verify calculated deflection. Material characteristics and processing affect performance sufficiently to preclude optimizing a design by calculation. In some cases, for example, peak stresses occurring at the inner edge of the disc can exceed the maximum

Table 6.5.3.7. Belleville Spring Equations

Deflection-Load

$$F = \frac{4E\delta}{(1-\nu^2)KD^2}\left[\left(h - \frac{\delta}{2}\right)(h - \delta)t + t^3\right]$$

Stress (small deflections)

$$S = \frac{4E\delta}{(1-\nu^2)KD^2}\left[C_1\left(h - \frac{\delta}{2}\right) + C_2t\right]$$

Stress (large deflections)

$$S = \frac{4E\delta}{(1-\nu^2)KD^2}\left[C_1\left(h - \frac{\delta}{2}\right) - C_2t\right]$$

F = load, lb_f
E = modulus of elasticity, lb_f/in^2.
ν = Poisson's ratio
K = constant (See Figure 6.5.3.7b)
δ = deflection, in.
D^2 = outside diameter, in.
d = inside diameter, in.
S = stress, lb_f/in^2.
h = free height minus thickness, in.
t = thickness, in.
C_1 = constant (See Figure 6.5.3.7b)
C_2 = constant (See Figure 6.5.3.7b)

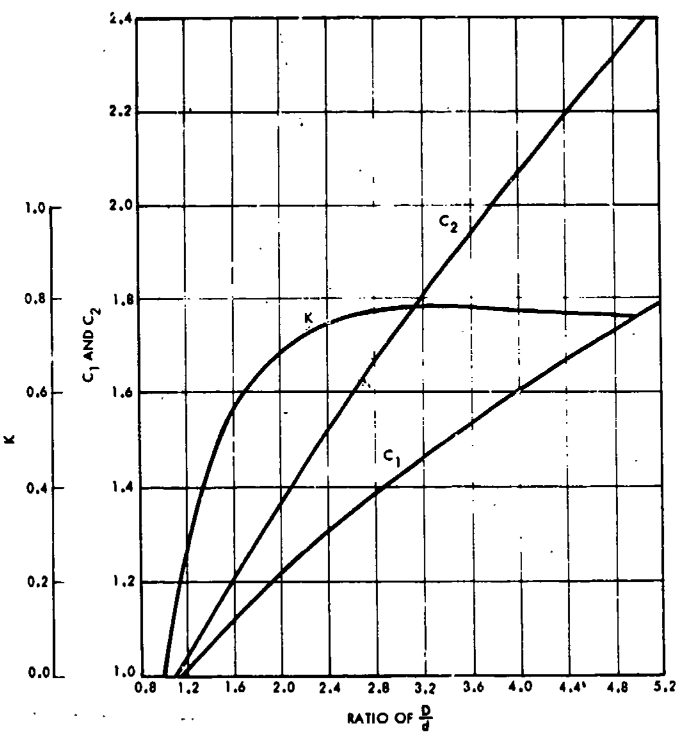


Figure 6.5.3.7b. Stress and Deflection Constants for Belleville Springs
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 72)

allowable yield stresses. The surrounding low stressed area acts to support the higher loaded corner resulting in very little yielding.

Many combinations of Belleville springs are possible, providing a variety of load-deflection characteristics. Figure 6.5.3.7d illustrates several arrangements for series and parallel stacking of Belleville springs. The total spring rate can be calculated in the same manner as compression springs. Friction between parallel discs can be utilized to advantage in applications requiring spring damping; however, techniques for minimizing inter-disc friction must be considered for applications requiring maximum repeatability.

A reduction of friction has been achieved by including spacer rings at the inner and outer edges of the disc. This technique is described in Reference 21-4.

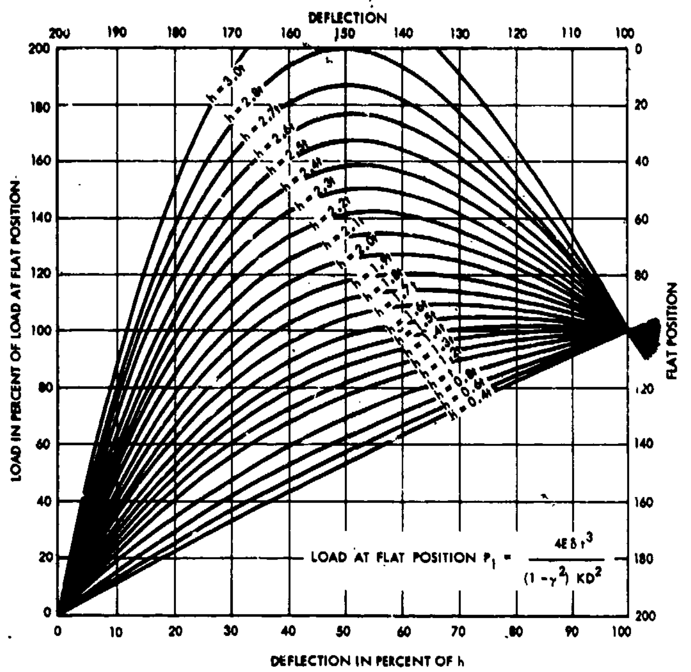


Figure 6.5.3.7c. Belleville Spring Load-Deflection Curves
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 71)

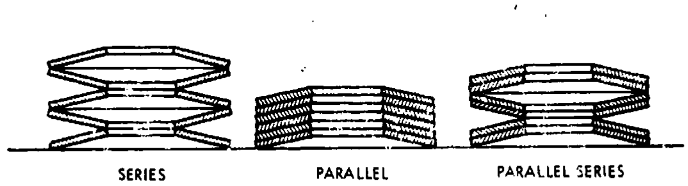


Figure 6.5.3.7d. Combinations of Belleville Springs
(From "Handbook of Mechanical Spring Design," Associated Spring Corporation, Bristol, Connecticut, page 73)

6.5.3.8 MACHINED SPRINGS. A spring device, similar in characteristics to a helical compression spring developed for precision instrument applications, can be fabricated by machining a symmetrically arranged series of slots in a short tubular length of metal. A few designs are illustrated in Figure 6.5.3.8.

Machine springs are adaptable to short stroke, high rate requirements often encountered in pressure and flow regulators and thermostatic devices. They are inherently symmetrical in action, eliminating cocking and torsional effects which create friction in coil springs. End coil stacking or "end effect," which causes non-linearities in coil springs of few turns, is eliminated in the machined spring. In addition, precision dimensions can be held for application in installations allowing very small radial clearance. This class of spring must be custom designed for the specific application. Design calculations would assume a series of curved beams fixed at both ends. Some development is required to obtain a precise load rate (Reference V-27).

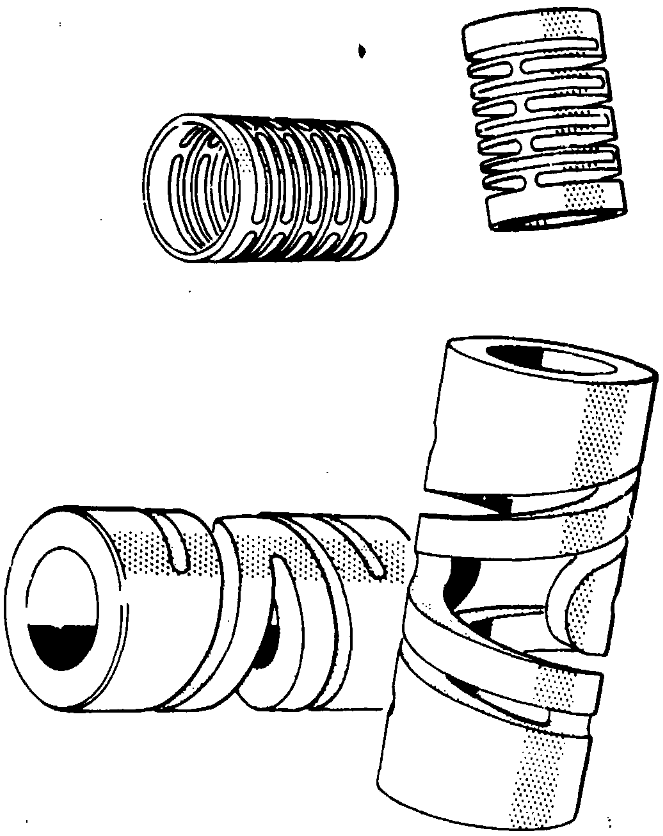


Figure 6.5.3.8. Typical Machined Spring Designs
(Courtesy of Consolidated Controls Corporation, El Segundo, California, holder of patent)

6.5.3.9 NON-METAL SPRINGS. Springs constructed of materials other than metal are discussed as follows:

Plastics and Glass. Springs made of these materials are limited to low load applications. They are most useful for applications requiring large deflections and low spring rates. They exhibit low elastic and torsional moduli and, consequently, form relatively soft springs for the working section involved.

Laminated Fiberglas. Resin bonded glass fiber springs have been used in flat forms. By use of directionally oriented fibers, the elastic properties of the glass have been utilized to a relatively high efficiency. In practice, it has been possible to fabricate flat spring stock with up to 87 percent of the fibers oriented in the working direction. The remainder are cross-plied surface fibers acting to support the position of the working fibers. For flat stock usable in any direction, cross-plied and isotropic laminants are available. The cross-plied laminant uses equal quantities of fibers oriented at right angles. The isotropic form has alternate plies oriented 120° apart.

The prime advantage of fiberglas laminates for spring applications comes from the high volumetric energy absorption (efficiency) exhibited. A comparison of this property with the value for steel is given in Table 6.5.3.9. This indicates an advantage of over twice for the laminated material, and results primarily from the low elastic modulus of the laminant relative to that of steel and a relatively high strength.

Table 6.5.3.9. Spring Materials Compared at Room Temperature
(From "Product Engineering," L. A. Heggernes, vol. 30, no. 46, November 1959, Copyright 1959, McGraw-Hill Publishing Company)

PROPERTIES	65% GLASS—35% EPOXY*			FLAT SPRING STEEL (SAE 1060)
	UNDIRECTIONAL	CROSS-PLIED	ISOTROPIC	
Ultimate flexure strength S, psi × 1000	100	90	66	180
Flexure modulus E, psi × 10^6	4.5	3.0	2.7	30
Volume efficiency, n, in.-lb/in^3 (cantilever: $n = S^2/18E$)	151	135	90	60
Endurance limit, psi × 1000	29	22	15	83
Maximum design stress, psi × 1000	20	15	12	80

*3M's "Scotchply," Type 1000

Other potential advantages of this material include:
a) Corrosion resistance
b) Resistance to fatigue failure
c) High vibration damping characteristics.

Table 6.5.3.11. Comparison of Air Springs
(From "Product Engineering," J. E. Gleck, vol. 33, no. 24, November 1962, Copyright 1962, McGraw-Hill Publishing Company, New York)

TYPE	SPRING RATE (60 psi)	EFFECTIVE AREA (in.²)	STROKE (in.)	NATURAL FREQUENCY
Bellows				
Single convolution	1400	12 to 108	2 to 4½	150 to 240
Two convolution	750	15 to 500	4½ to 10	105 to 120
Three convolution	500	35 to 100	11 to 12½	90 to 95
Rolling sleeve	450	8 to 90	10 to 15	60 to 100
Rolling diaphragm	180	8 to 150	6 to 9½	20 to 100

6.6 BELLOWS

6.6.1 Introduction

A bellows is a flexible, thin-walled, circumferentially corrugated cylinder either with or without integral ends. Bellows may be either metal or nonmetal, but this subsection of the handbook principally treats metal bellows. Metal bellows can be formed, welded, machined, or deposited; several different corrugation patterns are used for either manufacturing method. Single-wall construction is conventional but multiple-ply formed bellows are frequently used, and multiple-ply welded bellows are used occasionally.

A bellows can be compared with a piston because it is frequently used as an actuating element; with a spring because it possesses a spring rate; and with a diaphragm because it is sensitive to pressure and temperature changes and provides a positive seal. Bellows are used to accommodate linear motion, bending or wobble action, and lateral translation.

Bellows serve many different functions, both as independent units or as integral parts of more complex components. This sub-section of the handbook treats bellows alone as a discrete subject and does not cover component design factors such as the restraint devices of a bellows-type flexible fluid coupling. Among the aerospace applications for bellows are:

1) Flexible fluid couplings (Sub-Topic 5.13.5)
2) Accumulators
3) Propellant tank positive expulsion devices
4) Pressure-sensing elements for regulator control pilots
5) Instrumentation pressure gages
6) Valve seals (axial motion) (Figure 6.6.1)
7) Secondary or positive seal elements for rotating shaft face seals
8) Viscous dampers
9) Pressure switch seal/sensing elements (Figure 5.16.3.1d)
10) Volume compensators
11) Linear actuators

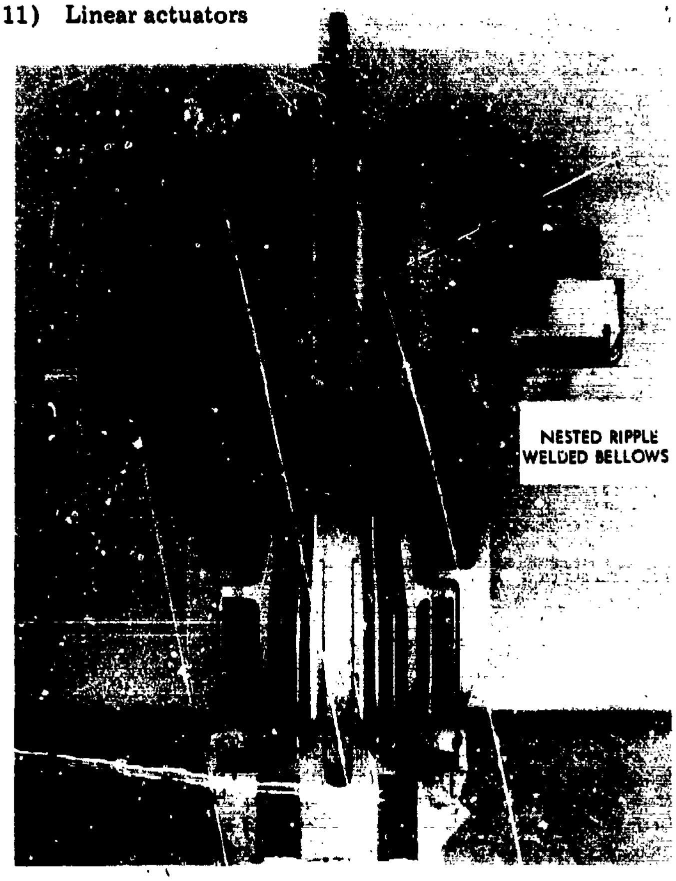


Figure 6.6.1. Nested Ripple Welded Bellow in Flow Control Valve for Apollo Lunar Module Descent Engine

Recent investigations into bellows technology have documented much more design information than was formerly available. These investigations have also resulted in sophisticated computer programs to aid in the design and analysis of bellows. The broadest and most recent of these studies was the development of analytical techniques for the design of bellows and diaphragms conducted by Battelle Memorial Institute for the Air Force Rocket Propulsion Laboratory and reported in Reference 44-37. This effort concentrated on bellows, both formed and welded, which operate primarily within the elastic regime (wherein stresses remain below the yield strength of the material). A comprehensive study of welded bellows operating in the plastic regime for propellant expulsion devices has been performed by Bell Aerosystems Company for NASA (References 81-6, 81-8, 81-9, and 81-10).

An important result of these efforts has been the demonstration that it is extremely difficult to design a bellows precisely on paper and then fabricate an item which will accurately match the specified geometry and performance. For this reason (as well as economic factors such as the cost of tooling for bellows fabrication), in the great majority of cases the fluid component designer should endeavor to adapt an existing bellows configuration to his application rather than to design a completely new bellows. For critical applications the most accurate use of computer techniques is based upon precise knowledge of geometry, usually obtained by sectioning a sample bellows.

References pertaining to various aspects of bellows design and application may be found at the end of this subsection. Two of these references are of particular interest.

Reference 513-3, Analysis of Stresses in Bellows, prepared by Atomics International for the Atomic Energy Commission, treats convoluted, ring-reinforced, and toroidal bellows. SAE ARP 735, Aerospace Vehicle Cryogenic Duct Systems (Reference 23-59) presents an excellent treatment of bellows design as related to flexible joints.

6.6.2 Design and Selection Factors

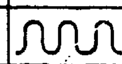
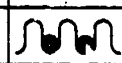
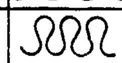
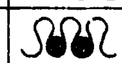
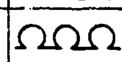
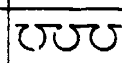
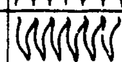
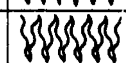
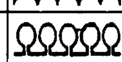
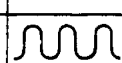
Bellows are ordinarily categorized by three characteristics:

1) Type of material (metal, nonmetallic)
2) Method of manufacture (formed, welded, machined, deposited)
3) Shape of convolutions (toroidal, square, ripple, etc.)

Each method of manufacture and shape of convolution has its own set of advantages and disadvantages as summarized in Table 6.6.2a. In addition, Reference 44-37 discusses these relative merits in detail.

Regardless of the type of convolution shape, the geometry is specified by certain conventional nomenclature, part of which is shown in Figure 6.6.2. Additional nomenclature which is required to describe the overall bellows assembly and the specific convolution shapes is defined as it first occurs in the text and in Table 6.6.2b. The proportions used in bellows construction are limited by the practical problems of fabrication and structural stability. The ratio of free length (length of all convolutions with no forces acting) to inside diameter is generally no greater than 1-1/2 to 1 but many exceptions are made. The span is usually from 3 to 13 percent of the inside diameter, with spans in excess of 25 percent ID being found primarily in very small bellows. Welded bellows are essentially unlimited with respect to span, whereas formed bellows are limited by the necessity for forming convolutions from a cylinder. Torus radii of straight sidewall convolutions (cone angle of 90 degrees) are seldom less than 3 times the wall thickness; however, values as low as 1.8 have been attained.

Table 6.6.2a. Major Bellows Convolutions and Characteristics
(Adapted in part from Reference 44-37)

	CONVOLUTION SHAPE	AXIAL SPRING RATE	LONG STROKE CAPABILITY	RESISTANCE TO DIFFERENTIAL PRESSURE
FORMED				
SEMITOROIDAL		VERY HIGH	VERY POOR	VERY GOOD
U-SHAPED WITH STRAIGHT WALL		MEDIUM	FAIR	FAIR
U-SHAPED, EXTERNAL RING SUPPORT		HIGH	FAIR	VERY GOOD
U-SHAPED, INTERNAL RING SUPPORT		HIGH	FAIR	VERY GOOD
U-SHAPED, EXTERNAL T-RING SUPPORT		HIGH	FAIR	VERY GOOD
S-SHAPED		MEDIUM	FAIR	FAIR
S-SHAPED, EXTERNAL RING SUPPORT		HIGH	FAIR	VERY GOOD
TOROIDAL, EXTERNAL PRESSURE		VERY HIGH	POOR	EXCELLENT
TOROIDAL, INTERNAL PRESSURE		VERY HIGH	POOR	EXCELLENT
NESTING (SINGLE SWEEP)		MEDIUM	GOOD	GOOD
WELDED				
FLAT CONICAL PLATE		MEDIUM	FAIR	GOOD
STEPPED FLAT PLATE		LOW	GOOD	FAIR
SINGLE SWEEP, OR CURVED		MEDIUM	GOOD	GOOD
NESTED RIPPLE, OR CORRUGATED (NESTING)		VERY LOW	EXCELLENT	POOR
CORRUGATED (NON-NESTING), OR RIPPLE		LOW	GOOD	POOR
TOROIDAL		VERY HIGH	POOR	EXCELLENT
DEPOSITED				
U-SHAPED (CAN BE VARIED)		LOW	GOOD	FAIR
MACHINED				
RECTANGULAR		HIGH	FAIR	EXCELLENT

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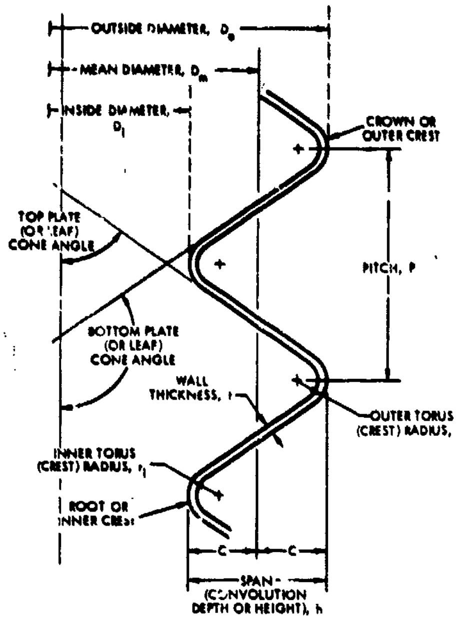

Figure 6.6.2 Generalized Nomenclautre for Bellows Dimensions

Table 6.6.2b. Bellows Symbols and Units

A = effective area, in^2

$A = \pi\left(\frac{D_o + D_i}{4}\right)^2 = 0.1963\,(D_o + D_i)^2$

C_f = free pitch, in.

D = lateral bending stiffness, lb$_f$-in^2

D_i = inside diameter, in.

D_m = mean diameter = $\frac{D_o + D_i}{2}$, in.

D_o = outside diameter, in.

E = Young's modulus of elasticity, psi

F = axial force, lb$_f$

F' = geometry coefficient, dimensionless

f = friction factor

f_b = ratio of friction factor of curved bellows to straight bellows

f_m = natural frequency of bellows in accordion vibration mode, cps

f_L = natural frequency of bellows in bending vibration mode, cps

g = gravitational constant or units conversion, in/sec^2

h = span, or convolution height, in. (NOTE: many bellows manufacturers use h to signify wall thickness)

$h = \frac{D_o - D_i}{2}$

k = bellows spring rate, lb$_f$/in.

K' = empirical fatigue factor

L = length of all convolutions in bellows, in.

ℓ = total live convolution length, in.

m_a = total mass of all active (live) convolutions, lb$_m$

m_c = mass per convolution, lb$_m$

m_e = mass of the liquid in the bellows, lb$_m$

m_{fa} = mass of fluid trapped in active length at rest, lb$_m$

N = expected number of cycles before fatigue failure

N_a = total number of active (or live) convolutions

N_c = total number of convolutions

NP = nested pitch, in.

N_p = number of plies

P = axial load, lb$_f$

p = pressure, psi

Δp = pressure drop, psi

P_{cr} = critical load, lb$_f$

p_{cr} = critical pressure, psi

q = quarter pitch = pitch/4, in.

$\bar{R}$ = effective radius or average radius, in.

R = radius at curvature in single-sweep bellows (leaves), in.

R_e = Reynold's number based on D_i, dimensionless

r = radius of inner and outer convolutions in bellows

r_B = radius of bend for curved bellows, in.

r_o = outside radius, in.

S_a = peak-to-peak amplitude of alternating stress level, psi

t = thickness (wall, leaf, plate or diaphragm), in.

U = distance from crest of convolution to mean radius as measured on the surface of convolution, in.

V = fluid velocity through bellows, in/sec

V_{TOT} = total volume of fluid within bellows, in^3

V_B = radius of curvature (of center line) of bellows, in.

W_m = weight of vibrating bellows, lb$_f$

β = shape factor, dimensionless

δ = total axial displacement of the end from the rest position (change in length from free length), in.

ρ density, lb$_f$/in^3

σ_b = stress in bending, psi

μ = Poisson's ratio, dimensionless

In order to determine the best bellows for a particular application, the designer should consider the following: choice of materials, fatigue life, spring rate, effect of vibration, and method of applying pressure. In most cases it will be wiser to select an existing bellows from a manufacturer's selection rather than to specify fabrication of a new design, as this minimizes development problems and tooling costs which often arise in making new bellows designs. The following detailed topics discuss bellows features and selection considerations.

6.6.2.1 MATERIALS. Bellows are constructed primarily of metal alloys, with nonmetals relegated to a secondary role. The extent to which the service performance of bellows and diaphragms fulfills design predictions is strongly dependent upon the quality of the materials from which they are fabricated and the care taken in their manufacture. Material defects, weld discontinuities, forming irregularities, and postfabrication damage can all result in locally high stresses that may lead to premature failure. The choice of bellows material is dictated by performance and by environmental and manufacturing parameters such as:

a) Compatibility

b) Temperature

c) Life cycles and stress levels (particularly whether within the plastic or elastic deformation regimes)

d) Manufacturing technique.

Compatibility. The bellows material must be compatible both with the fluids to which it will be exposed and with the contacting metals of the final assembly, because the integrity of a bellows is very susceptible to failure by pitting corrosion due to the thin wall thicknesses. Cleaning media and procedures must be considered carefully, since the recesses are relatively inaccessible to cleaning and inspection. Materials compatibility data are presented in Sub-Section 12.5, and corrosion is discussed in Sub-Section 13.7.

Temperature. The useful working temperature range of bellows may extend as low as -460°F and as high as 2000°F. Elevated temperatures generally decrease the spring rate and fatigue life of most materials.

Life and Stress. The pressure range of a bellows is limited by the yield strength of the material, and the cycle life of the bellows is limited by the fatigue life of the material and by the elastic stability of the bellows structure. Long-life bellows are designed to be operated in the elastic range, as in shaft seal and instrumentation bellows. Many aerospace flexible joint bellows are designed to operate in the elastic-plastic stress range, resulting in cycle life in the order of 1000 full-deflection cycles (Reference 23-59). Positive expulsion device bellows require extremely long strokes resulting in significant plastic deformation and cycle life in the order of 100 to 200 full-deflection cycles (References 81-6, 81-8, 81-9, and 81-10). Fatigue life prediction in the plastic regime is difficult, but for elastic deflections the conventional use of S-N curves may be used as described in Detailed Topic 6.6.2.4 below. For elastic-plastic deflection, a strain-range-N relationship may be used.

Manufacturing Technique (Reference 44-37). Materials for formed bellows must be both weldable and formable. Although smaller bellows are usually made from seamless tubing, most formed bellows over an inch in diameter are made from sheet or strip formed into a cylinder and longitudinally seam welded. Welding is also the most common method of end-fitting attachment. Most formed aerospace bellows today are made from one of the 300-series stainless steels. Inconel 718 is being used increasingly because of its higher yield strength and its relative immunity to stress corrosion. Titanium alloys are becoming candidate materials for formed aerospace bellows because of their corrosion resistance and their good strength-to-weight ratios.

Materials for welded bellows need not have the formability of materials for formed bellows. Therefore, in addition to the materials used for formed bellows, a variety of less formable alloys are used for welded bellows. The 300-series stainless steels, Inconel 718, and AM-350 are among the most-used welded-bellows materials.

Materials for deposited bellows are made *in situ* by electroplating or chemical deposition onto machined aluminum mandrels that are later chemically dissolved. The most common material for electrodeposited bellows is nickel, although copper is also used. Chemically deposited bellows can be made from alloys which, though still over 90 percent nickel, contain significant percentages of other strengthening elements. Both types of deposited bellows can be made with composite metal walls consisting of layers of different metals. The deposited-bellows industry is relatively young, and further developments in deposited-bellows materials can be expected.

Additional Materials Consideration. Whether the strip or sheet used in bellows manufacture is purchased to any special tolerances depends upon the end application of the bellows. When the spring rate is not critical (bellows intended for flexible couplings, for example) the customary 10 percent mill-thickness tolerance may be usually considered satisfactory. When the deflection characteristics must be more carefully controlled, materials may be selected from warehouse stock. In this way, thickness may be controlled to within about 5 percent on a given order. Rerolled materials from specialty metal fabricators provide the best commercially obtainable thickness tolerances but are seldom used for making formed bellows. Manufacturers of welded bellows and diaphragms using rerolled materials claim thickness tolerances of ±0.0001 inch. A more commonly quoted tolerance is ±0.00025 inch over a 20-inch-strip width.

Opinions differ among manufacturers as to the desirability of a bright surface finish on the starting material. Some manufacturers claim an improvement in the fatigue life of bellows produced from bright-finished material (No. 2B finish), while others see no difference. Some manufacturers also claim that the bright-finished material, containing more cold work than the dull or matte-finished material (No. 2D finish), is more difficult to form. There is a trend toward the use of bright-finished material.

Typical bellows alloys and their outstanding characteristics are shown in Table 6.6.2.1. Reference 44-37 treats metal bellows materials in detail. Nonmetallic bellows are also made of elastomers such as rubber, Buna N, and Neoprene, or plastics such as Teflon.

Table 6.6.2.1. Typical Bellows Alloys
(Reference 44-37)

Material	Type of Bellows: Formed	Welded	Deposited	Outstanding Characteristics
Copper Alloys				
Brass, Bronze, Beryllium Copper, Zirconium Copper	X		X	High proportional limit, good toughness
Aluminum Alloys				
[illegible]	X			High strength-weight ratios, good toughness at low and intermediate stress levels at temperatures as low as -423 F
Nickel and Copper-nickel Alloys				
Nickel 200, Monels 400, K-500	X		X	Good corrosion resistance, low magnetic permeability
Titanium Alloys				
[illegible]	X			High strength, high strength-weight ratio, good creep strength to [illegible] F
Low Alloy Steels				
[illegible]	X			Extremely high strength, high proportional limit, high fatigue strength, good creep resistance
300-Series Austenitic Stainless Steels				
[illegible]	X	X		Excellent toughness to -423 F, good fatigue and creep strength, good neutron radiation resistance, good corrosion and oxidation resistance
Precipitation Hardening Stainless Steels				
17-4 PH, 17-7 PH, PH 15-7 Mo, AM 350	X	X		High strength, good creep strength, high fatigue strength
Other Iron Base Alloys				
19-9 DL, A-286, Kovar	X	X		High strength, good creep strength, high fatigue strength, plus sealability to hard glass (Kovar)
Nickel Base Alloys — Group I				
Inconels 600, 625, X-750 and 718, Incoloy 825, Hastelloy C	X	X		Good high temperature strength, good fatigue strength, good toughness, good creep resistance, and good strength to rupture
Nickel Base Alloys — Group II				
M-252, Waspaloy, Udimet 700, René 41, René 62		X		Good high temperature strength, good fatigue strength, good toughness, good creep resistance, and good strength to rupture
Cobalt Base Alloys				
L-605		X		Good high temperature strength, good fatigue strength, good toughness, good creep resistance, and good strength to rupture
Refractory Metals				
Columbium		X		Good strength at extreme temperature, high proportional limit, excellent fatigue strength above transition temperature, excellent creep strength
Other Alloys				
Invar, Ni-Span C		X		Zero thermal expansion coefficient or constant elastic modulus

6.6.2.2 LOAD APPLICATION.

When a bellows is subjected to a differential pressure between its interior and exterior, it will change its overall length unless both ends are clamped. Each individual convolution does not lengthen or shorten equally, however, since the convolutions near the ends are restrained somewhat by the rigidity of the parts to which the bellows is attached. By convention, the convolutions are presumed to be totally fixed or totally free, depending on the support configuration. The convolutions presumed to be free are called live or active convolutions and they make up the live or active length of the bellows within which each convolution is presumed to behave identically.

It is preferable to apply the higher pressure to the exterior; this typically forms a more stable configuration, permitting higher pressures, and promoting longer life for any given design. A bellows is an unstable column, and internal pressure creates a column load on the bellows. Even moderate internal pressures may cause squirming (lateral misalignment and deformation) unless the bellows is laterally supported. A geometrically perfect bellows with uniform material properties and no side loads applied would not be susceptible to squirming until the critical axial load was applied. The Euler critical load for an ideal convoluted bellows that is clamped at both ends is:

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$$P_{cr} = \frac{4\pi^2 D}{\ell^2} \qquad \text{(Eq 6.6.2.2a)}$$

where

P_{cr} = Euler critical axial compressive load, lb_f

D = lateral bending stiffness, $lb_f\text{-}in^2$

ℓ = total live convolution length, in.

The lateral bending stiffness, D, is determined empirically; $D = 0.5KR^{-2}$ (Seide formula) is a median value. If the bellows is clamped at its free length, the internal pressure which causes squirming for a perfect bellows is approximately:

$$p_{cr} = \frac{2\pi k}{\ell} \qquad \text{(Eq 6.6.2.2b)}$$

where

p_{cr} = critical pressure, psi

k = axial spring rate, lb_f/in.

ℓ = total live convolution length, in.

Reference 513-5 relates that the critical internal pressure for convoluted bellows clamped at only one end is one-sixteenth that for the case with both ends clamped, and for both ends hinged it is one-fourth as much as for both ends clamped.

Real bellows will squirm at pressures below the calculated values because of geometric and material nonuniformities. The effect of concurrent axial displacement on stability is discussed in Reference 44-37.

Bellows which may be subject to overpressure should be restricted to a reduced stroke by the use of mechanical stops. Mechanical stops can be incorporated into the bellows to prevent extension of the bellows beyond its normal extended length and compression beyond its minimum design height. These stops will inhibit extension into the yield region but will not stop squirming unless lateral support is provided. Reference 23-59 suggests that proof pressures be 1.2 times the design pressure (including transients) or 1.5 times the design pressure, whichever is considered most severe, and that burst pressure be 2.5 times the design pressure. On the basis of the Battelle study (Reference 44-37), it is recommended that burst pressure be replaced by "pressure to produce gross deformation". Although burst pressure values are commonly available, their method of determination varies widely and system failure can occur below the burst pressure because of structural instability.

In order to accommodate higher pressures, bellows may be reinforced with rings that back-up the convolutions or may be made with multi-ply construction. Figures 6.6.2.2a and b show a high pressure, multi-ply bellows in cross-section.

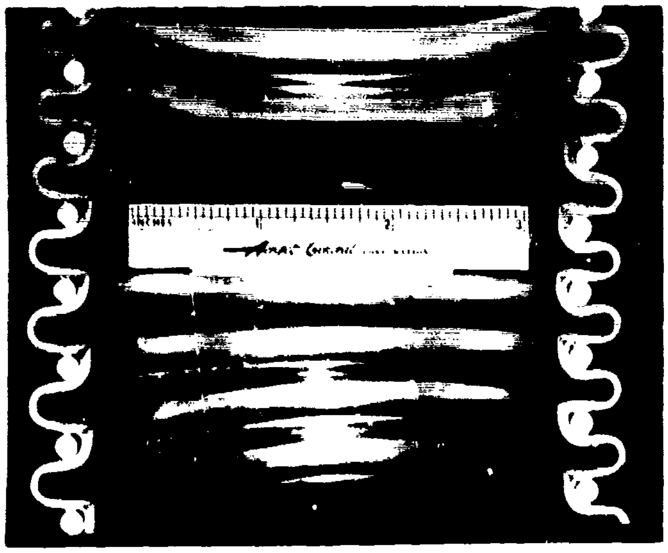


Figure 6.6.2.2a. Experimental 10-ply Bellows for 8400 psi Proof Pressure
(Reference 34-10)

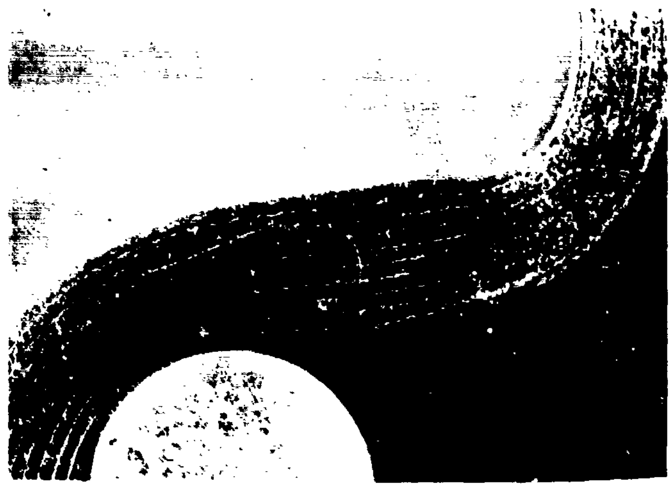


Figure 6.6.2.2b. Convolution Detail of 10-Ply Bellows
(Reference 34-10)

Flexible Joint Deflection. Formed bellows used in flexible fluid couplings are usually designed for operation within the following deflection limitations set forth in Reference 23-59.

a) *Compression Stroke.* 10 to 15 percent of the live length. (Up to 40 percent has been successfully attained; however, fatigue life is greatly shortened.)

b) *Extension Stroke.* 10 percent of the flex section live length. (Up to 40 percent has been successfully attained; however, fatigue life is greatly reduced.)

c) *Angular Deflection.* ±5 degrees maximum, although ±15 degrees has been successfully attained.

d) *Lateral Offset.* 0.02 L_c^2/D maximum, although 0.25 L_c^2/D has been successfully attained. Note that it is poor practice to have one joint accommodate two or more types of deflection simultaneously, although it is often dictated by design considerations.

e) *Torsional Deflection.* Preferably none; however, small rotations can be accommodated by specific design.

For many flexible joint applications the installation of the bellows joint in the final assembly and other handling situations provide the greatest dangers for exceeding specified deflections. With relatively fragile bellows it is good practice to provide positive support until the bellows is firmly fastened in position by means of special tooling often referred to as "hardbacks".

Expulsion Device Bellows Stroke. Propellant tank expulsion bellows differ from most other bellows applications in that there is very low pressure differential across the bellows (usually in the order of 5 psi) and deflections are essentially limited to axial stroke. This axial stroke may be either from fully extended to totally compressed (stack height or solid height) or vice versa. Welded bellows expulsion devices have been developed which consistently demonstrate 100 to 200 full stroke cycles over expansion-to-compression ratios of 10:1 or greater (References 81-6, 81-8, 81-9, and 81-10). Newly-developed nesting-formed bellows of single-sweep configuration have demonstrated 16:1 expansion-to-compression ratios and indicate potential for 20:1 ratios for 500 cycles and as high as 40:1 for 1 cycle (Reference 332-33). At an expansion ratio of 16:1, a formed bellows shows slight circumferential buckling at the convolution crests when expanded (Reference 332-33). Bellows for such expulsion applications usually have guides to preclude buckling.

6.6.2.3 FLEXIBILITY AND SPRING RATE. Bellows spring rate is the ratio of applied force to the resulting deflection. The inverse of the spring rate is called flexibility. More precisely, the spring rate of a bellows or diaphragm can be defined as the average axial force necessary to deflect the bellows a unit distance or as the ratio of a given axial force to the axial deflection caused by that force. The former value is usually supplied by the manufacturer. However, since many bellows exhibit some spring-rate nonlinearity even in the linear region, the value for the latter calculation may be different from that for the former calculation.

Spring rate is one of the important characteristics needed for the selection of bellows and diaphragms. Although it is quite easy to measure the spring rate of a bellows or diaphragm, this has always been one of the most difficult characteristics to predict accurately. The primary reason for this difficulty is that the spring rate in essence is an integration of the stresses in all parts of the bellows or diaphragm. Consequently, in the Battelle studies (Reference 14-57) measured spring rates were expected to be a sensitive indication of the accuracy of the stresses calculated using the NONLIN computing program. It was found that the theoretical spring rate predicted for each lot of experimental bellows was very close to the average measured spring rates for that lot, but that individual specimens varied from the average by several percent.

Flexibility is directly proportional to the number of convolutions per inch of free length. Doubling the number of convolutions per inch of free length halves the spring rate, increasing the flexibility, but decreasing the total possible movement since the solid height is larger (Reference 192-1).

The spring rate, k, of the bellows can be approximated by an empirical relationship (Equation (6.6.2.3a)) if the pressure force is the only acting force; other forces acting on the bellows must be accounted for by additional terms and the effective area must be estimated (see Detailed Topic 6.6.2.6).

$$k = \frac{\text{pressure} \times \text{effective area}}{\text{stroke}} \qquad \text{(Eq 6.6.2.3a)}$$

Like a spring, single-ply bellows spring rate can be approximated from geometry and material property data. The spring rate is related to material thickness; bellows effective diameter; number, shape, and height of convolutions; and modulus of elasticity. Table 6.6.2.3 presents equations to calculate the spring rates for several bellows shapes. The equations can be accurate for constant thickness. However, variations in thickness caused by the fabrication process can produce errors greater than 30 percent. Terms are defined in Table 6.6.2b. A simpler equation (from Reference 31-21) for the spring rate of flat plate bellows which does not take into account the geometric proportions is:

$$k = \frac{\pi E D_m t^3}{N_a h^3} \qquad \text{(Eq 6.6.2.3b)}$$

where

E = Young's modulus of elasticity, psi

D_m = mean diameter, in.

t = leaf or plate thickness, in.

N_a = number of live convolutions

h = span, in.

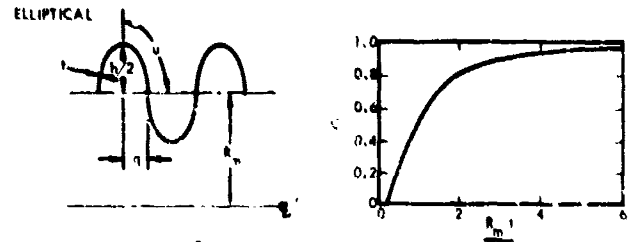
In these equations, the term N_a refers to the number of active or live convolutions which is approximately the total number (N_c) *minus two*. (Bell Aerosystems' detailed analysis of bellows stresses (References 81-6, 81-8, 81-9, and 81-10) indicated that *end effects* were present for a distance of several convolutions from each end, but the summed effect may be estimated for long bellows by this approximation).

Reference 44-37 presents techniques for analyzing spring rate on the basis of comprehensive stress analysis of a completely defined bellows geometry.

SAE ARP 735 (Reference 23-59) gives the spring rate for U-shaped bellows in lateral bending and torsion. Reference 513-5 gives equations for spring rates of single sweep bellows, and both References 513-3 and 513-5 give spring rate equations for U-shaped, toroidal, and ring-reinforced bellows.

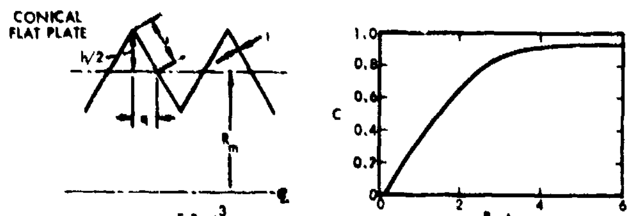
ISSUED: FEBRUARY 1970
SUPERSEDES: MAY 1964

Table 6.6.2.3. Spring Rates for Several Bellows Forms
(From "Machine Design," J. D. Matheny, vol. 34, no 1, January 4, 1962, Copyright 1962, Penton Publishing Company)



$$K = 6.920\,\frac{E R_m t^3}{u h^2}\,\frac{q}{L_c}\left(\frac{1}{C}\right)\left(\frac{q + h/2}{h/2 + 2q}\right)$$

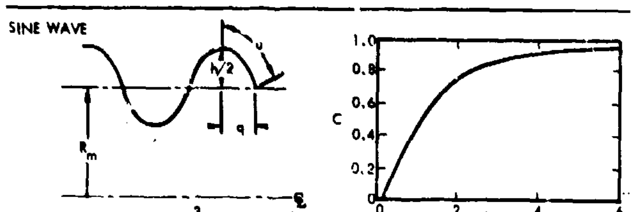
$$m_c = 6.43\,\rho t\,D_m\sqrt{3.3h^2 + 2hq + 3.3q^2}$$



$$K = 6.920\,\frac{E R_m t^3}{u h^2}\,\frac{q}{L_c}\left(\frac{1}{C}\right)$$

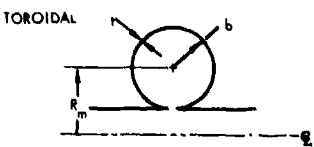
$$m_c = 6.28\rho t\,D_m\sqrt{h^2 + 4q^2}$$

NESTED RIPPLE:
m_c ≈ FLAT CONICAL PLATE + 5 TO 1(DEPENDING UPON RIPPLE D:P



$$K = 6.920\,\frac{E R_m t^3}{u h^2}\,\frac{q}{L_c}\left(\frac{1}{C}\right)\left(\frac{q + h/2}{h/2 + 1.6q}\right)$$

$$m_c = 12.57\,\rho t\,D_m u$$



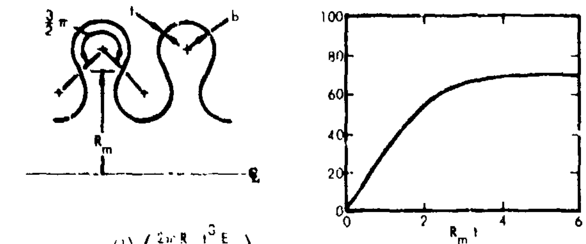
$$K = \frac{E t^2}{3.308r} \qquad 0 < \frac{R_m t}{r^2} \le 1$$

$$m_c = 6.29\pi t\,\rho\left[\frac{2qD_i}{r} - (D_i - r)\sin\theta + (\pi - \theta)(D_i + r\cos\theta)\right]$$

The spring rate of a multi-ply bellows is not precisely equal to the spring rate of one of the plies multipled by the number of plies, due to fabrication inconsistencies, i.e. variation of bellows leaf thickness. Bell Aerosystems (Reference 81-10) found that one two-ply bellows had a spring constant of 225 percent that of one of the plies alone. This is not consistent with data published by the Marman Division of Aeroquip Corporation (v-116) which states that two-ply and three-ply bellows will have spring rates of approximately 71% and 58%, respectively, that of a singly-ply bellows of the same span and total wall thickness for the same design conditions.

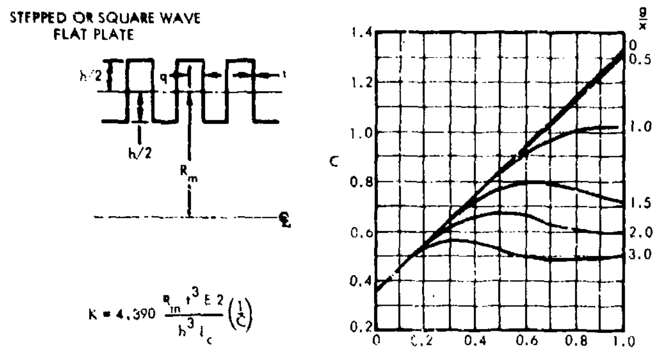
Table 6.6.2.3. Spring Rates for Several Bellows Forms
(From "Machine Design," J. D. Matheny, vol. 34, no 1, January 4, 1962, Copyright 1962, Penton Publishing Company)

'S' TYPE



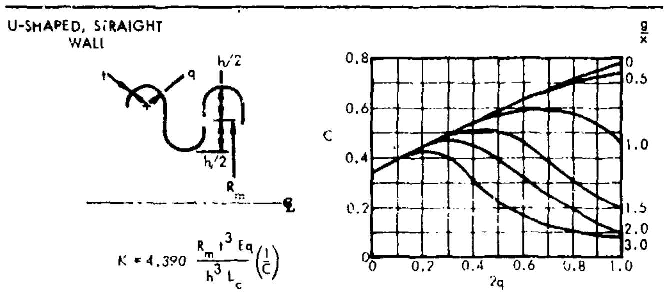
$$K = \left(\frac{1}{C}\right)\left(\frac{2\pi R_m t^3 E}{r^2 L_c (1 - \mu^2)}\right)$$

$$m_c = 29.6\,\rho t r D_m$$



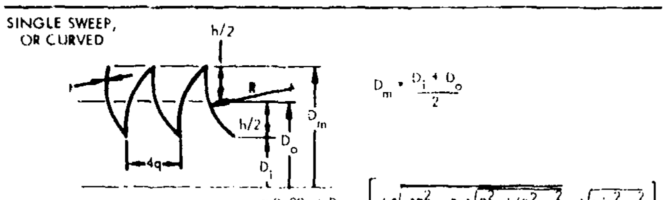
$$K = 4.390\,\frac{R_m t^3 E\, 2}{h^3 L_c}\left(\frac{1}{C}\right)$$

$$m_c = 6.28\rho\left(qD_i + qD_o + \frac{D_o^2}{4} - \frac{D_i^2}{4}\right)$$



$$K = 4.390\,\frac{R_m t^3 E q}{h^3 L_c}\left(\frac{1}{C}\right)$$

$$m_c = 3.14\,\rho t\left(\pi D_i r + \pi D_o r + \frac{D_o^2}{4} + \frac{D_i^2}{4}\right)$$



$$D_m = \frac{D_i + D_o}{2}$$

$$m_c = 2.09\,\rho t\,D_m\left[4\sqrt{2R^2 - R\sqrt{R^2 - h/2^2 - q^2}} - \sqrt{h/2^2 + q^2}\right]$$

6.6.2.4 FATIGUE LIFE AND STRESSES. Fatigue failure is one of the most common types of failures in aerospace bellows and diaphragms. In normal service, the life of a bellows or diaphragm is determined by the cumulative effect of the deflection and pressure stresses (or strains) to which it is subjected. Fatigue life can also be reduced by corrosion, stress concentration at points of geometrical discontinuity, material variations, residual stresses, and heat-affected material. On occasion, unexpected modes of deformation can cause high stresses which result in premature fatigue failure. Common causes of such overstressing are damage during shipment, improper installation, dynamic resonance, excessive stroke, and excessive pressure.

Prior to the program described in Reference 44-37, there has been no method by which the deflection and pressure stresses could be accurately evaluated for formed bellows. Thus, each manufacturer has been required to develop empirical design data based on fatigue tests of the manufacturer's bellows. With sufficient data, it has been possible to construct nomographs relating different fatigue life cycles to different percentages of maximum deflection and maximum pressure for the types of bellows tested. (Maximum deflection and pressure are those values, with appropriate safety factors, which will cause permanent deformation in the bellows.) Such nomographs are available from a number of bellows manufacturers. Reference 73-114 includes graphs for determining the expected life of one manufacturer's nested ripple and flat plate (stepped) welded bellows.

It was demonstrated in Reference 44-37 that the computing program NONLIN could be used to accurately predict the strains in bellows or diaphragms of any shape which remains largely in the elastic state when subjected to either axial deflection or internal pressure. Because of the prevalence of bending in bellows and diaphragms, the surface of an elastic structure can extend significantly into the plastic regime. One of the major tasks in the study was an investigation of the feasibility of using the theoretically predicted maximum strains in a bellows to predict the fatigue life of the bellows under a given cyclic load. The planned approach involved testing formed bellows, welded bellows, and diaphragms to establish their fatigue limits at different levels of cyclic strain as calculated by NONLIN. These experimental results were then to be compared with data obtained from standard fatigue tests made on metal coupons of the same material. If a correlation could be established between the fatigue lives of the bellows and diaphragms at the calculated strain ranges and the fatigue lives of the coupons at the same strain ranges, then the relatively ample coupon fatigue data expressed as strain-range vs. cycles to failure available in the literature could be used, together with NONLIN, to predict fatigue life of bellows and diaphragms. Reference 44-37 gives detailed descriptions of the fatigue tests conducted in the program and the results obtained. These results are summarized briefly here, and some data is presented in Detailed Topic 6.6.3.1.

Formed Bellows. An investigation was carried out for formed bellows made of type 321 stainless steel and Inconel 718. The results of this investigation showed that NONLIN could be used together with coupon data to estimate the minimum fatigue life of a formed bellows subject to the following limitations:

1) The lifetime values estimated from coupon data may be optimistic because of variations of actual bellows from the geometric model on which the computations are based and because of residual stresses and material variations which the theoretical treatment does not include.

2) When such factors have been taken into account by a few fatigue tests on bellows formed by a similar process and from similar material, the stress analysis can be used to provide good interpolations and moderate extrapolations for different loadings and moderately different geometries for fatigue values as low as 10,000 cycles.

Welded Bellows. In contrast to the formed-bellows fatigue tests, no satisfactory correlation could be obtained between the fatigue life and the theoretically predicted maximum strain for welded bellows made of either type 347 stainless steel or AM-350. The strains predicted by NONLIN were satisfactorily accurate, however the tests showed considerable variation in fatigue life for a calculated maximum strain range. Fatigue-life variation was apparently the result of manufacturing variations associated with the welding process (such variations are not accounted for by NONLIN). Significant variations in the fatigue life were observed both for bellows made by different manufacturers and for nominally identical bellows made by the same manufacturer.

As a result of the tests it was concluded that the variability of the fatigue life resulting from manufacturing variations must be experimentally determined for each manufacturer's process. This requires the same type of testing that manufacturers currently perform to establish fatigue nomographs. Thus, although NONLIN can be used to analyze the stresses and strains in welded bellows and to aid in their design, the fatigue life of welded bellows must still be established experimentally. It must be emphasized that experimental determination of welded-bellows fatigue life must be based on a sufficiently large number of tests.

The studies described in Reference 44-37 demonstrated that a combination of compression and pressure may reduce the life of welded bellows significantly. This occurs if the slight ballooning caused by the pressure causes the diaphragms of the bellows to interfere during compression, thereby greatly increasing the stresses at other points of the cross section. Although the analytical prediction of this condition would be very difficult, it can be determined experimentally because the diaphragm interference causes a significant change in the spring rate of the bellows. Since each welded bellows should be used as much in compression as possible to obtain the longest fatigue life, these tests must be conducted to assure that diaphragm interference will not be encountered.

The tilted-edge welded-bellows configuration to be investigated in a follow-on program is expected to experience fatigue failure in the parent material rather than in the weld areas. If this is achieved, the procedure for estimating the fatigue life of formed bellows may be applicable to the tilt-edge configuration.

Exclusive of the computer-dependent technique of Reference 44-37, generalized methods for making accurate predictions of bellows fatigue life are not presently available. The other methods in use are empirical and depend upon properties which are not uniform in real materials at the microscopic scale; so statistical methods are frequently used to resolve the resulting scatter in the data (see References 44-37 and 513-3). ARP 735 (Reference 23-59) quotes an equation for formed bellows life in number of cycles, N, as a function of two empirical factors and the peak-to-peak amplitude of the alternating stress level, S_a;

$$N = \left(\frac{K'}{S_a}\right)^n \qquad \textbf{(Eq 6.6.2.4a)}$$

For type 321 CRES at room temperature, $K' = 1.6 \times 10^6$ and $n = 3.5$. Much longer life may be expected at cryogenic temperatures. References 44-37 and 513-3 also contain discussions, data, and similar empirical equations.

Stresses arise in bellows because of pressure differences across the walls, because of mechanically-applied loads which either deflect or restrain the bellows, and because of thermal effects. Calculating the stress in a bellows is a complicated task because of the complexity of the shapes involved. Computer analysis of different bellows shapes has shown that minor changes in geometry can radically alter the stress patterns. Reference 44-37 gives results which show that tilting the bellows convolutions cross-section near the root and crown welds significantly decreased stresses (but increased spring constants). Thermal stresses are not generally considered in bellows design; Reference 44-37 briefly treats temperature effects. All short-cut methods for calculating stresses in bellows (nomographs and simple equations) are based on simplifications for which all of the assumptions are seldom mentioned. The following relations, therefore, should be approached with caution and with the realization that the literature contains a number of slightly different expressions depending upon the particular assumptions used in the derivations.

Maximum bending stress, σ_b, at the root of a single-ply U-shaped convolution due to axial deflection is:

$$\sigma_b = F'E\left(\frac{t}{h^2}\right)\left(\frac{\delta}{N_a}\right) \qquad \textbf{(Eq 6.6.2.4b)}$$

where

F' = some variable related to a geometrical ratio such as pitch to span, dimensionless

E = Young's modulus of elasticity, psi

t = thickness of leaf, in.

δ = axial displacement from rest position, in.

h = span, in.

N_a = number of active convolutions which contribute (equally) to the deflection.

Reference 446-8 gives values of $F'E$ for a standard (U-shaped) bellows as follows:

$F'E = 41.4 \times 10^6$ psi for type 316 stainless steel

$= 42.7 \times 10^6$ psi for Inconel 625

$= 17.6 \times 10^6$ psi for aluminum alloys

$= 26.3 \times 10^6$ psi for copper

ARP 735 gives the U-shaped bellows stress equation in the same form but

$$F' = \frac{1.5}{\beta(1-\mu^2)} \qquad \textbf{(Eq 6.6.2.4c)}$$

where

β = shape factor (see Figure 6.6.2.4)

μ = Poisson's ratio

Reference 513-3 gives equations and graphs of the required variables to calculate pressure, deflection (axial and torsional), and combined stresses in U-shaped, toroidal, and ring-reinforced bellows. Reference 513-5 contains equations and graphs to calculate axial deflection stresses in U-shaped, toroidal, and single-sweep bellows, plus pressure stresses in single-sweep bellows and stresses due to lateral deflection of U-shaped bellows. Reference 73-114 has nomographs for axial deflection and pressure stresses in flat plate (stepped) and nesting ripple welded bellows; the answers are used in another nomograph to predict life expectancy.

6.6.2.5 DYNAMIC PERFORMANCE. Dynamic performance (of particular interest in instrument applications) involves the response, repeatability (precision), hysteresis, and sensitivity of a bellows. The *response* of a bellows is the time required to move against its own inertia and spring force through its operating stroke. Typical values of response vary from 3 to 50 milliseconds, depending on the bellows geometry. Bellows with large effective areas and low spring rates have faster response times. Bellows that require long deflections are generally slow. Liquid-filled bellows may have response times of one to three seconds.

The *repeatability* or *precision* of a bellows is a measure of the ability of the bellows to return to its original free length. For bellows which operate primarily within the elastic regime (such as instrumentation bellows), repeatability can be approximately 0.05 to 0.5 percent of stroke depending upon the severity of the stresses imposed.

Hysteresis in a bellows is the relationship between the path of force, F, versus displacement, δ, when the bellows is being shortened and the path of force versus displacement when it is being lengthened. This is most clearly seen on a graph where two curves are plotted against linear scales to show an extension stroke and the following compression stroke (Figure 6.6.2.5a). An idealized structure may be limited to proportional stress versus strain (Hookean deformation) in which the work done is fully recoverable (i.e. the structure is fully elastic); in this case the plots of the paths of extension from and contraction to the same initial point will be perfectly superimposed straight lines and there is zero hysteresis. Non-proportionality in a region of pure elasticity will result in a non-linear function but the succeeding reversal of motion will follow exactly the same path so the two curves also would be coincident (i.e., work done in one direction is exactly equal to work done in returning to the same initial point but opposite in sign). Again, this case represents zero hysteresis. Hysteresis is mainly caused by a degree of plasticity which causes non-recoverable losses to occur. To return to initial conditions it is necessary to follow a load-displacement path that forms a loop so the material experiences losses that compensate for each other (i.e., plastic deformation of equal magnitude) at each end of the loop.

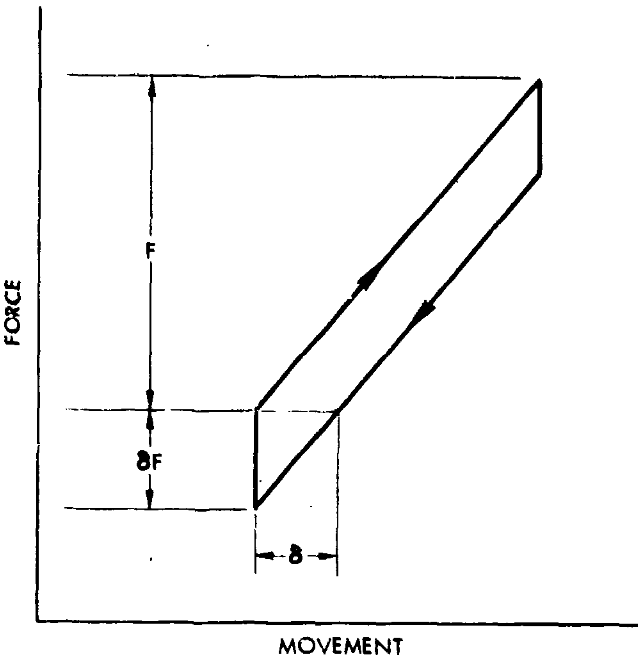


Figure 6.6.2.5a. Bellows Hysteresis
(Theoretical)

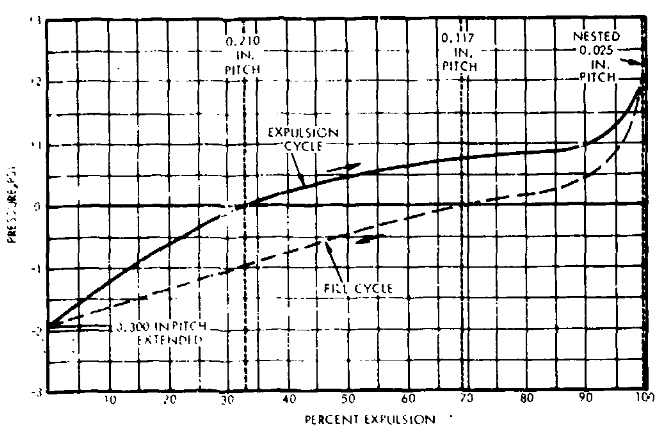


Figure 6.6.2.5b. Welded Expulsion Bellows Hysteresis Curve
(Courtesy of Bell Aerosystems Co., Buffalo, N.Y.)

An actual hysteresis curve for a propellant positive expulsion bellows operating in the plastic regime is shown in Figure 6.6.2.5b. Such plastic-regime bellows demonstrate pronounced hysteresis which must be considered in the design application. Elastic-regime bellows used in high-cycle-life applications demonstrate very little hysteresis over the design operating range of displacement. In elastic-regime bellows hysteresis is commonly ignored and its effect is considered a part of repeatability.

Sensitivity is the smallest pressure change for which a movement of the bellows is measureable and repeatable in a pressure cycle range. Sensitivities as low as 0.01 percent of the design pressure are not uncommon for unrestrained bellows. Friction damping and inertia of attached parts tend to decrease sensitivity.

6.6.2.6 EFFECTIVE AREA AND INTERNAL VOLUME. The effective area is the equivalent surface area on which pressure acts to produce axial force. The effective area of both formed and welded bellows can be approximated to an accuracy of 3 percent by the relationship:

$$A = 0.1963\,(D_o + D_i)^2 \qquad \textbf{(Eq 6.6.2.6)}$$

The volume within a bellows, exclusive of the end plates, is approximately the effective area times the length. However, Reference 81-6 suggests that a closer approximation may be obtained from the following expression:

$$V_{TOT} = 0.262L\,(D_o^2 + D_i\,D_o + D_i^2) \qquad \textbf{(Eq 6.6.2.6a)}$$

6.6.2.7 VIBRATION. The life of a bellows under vibration depends upon the stresses produced, the level and frequency of vibration, and the amount of damping within the system. The bellows should be designed so that the resonant frequency occurs above the operational frequency. Calculation of the exact natural frequency is extremely difficult, but can be estimated for an empty bellows which is free at one end by the following spring formula (Reference V-117):

$$f_n = 3.13\sqrt{\frac{k}{m_1 + \frac{m_2}{3}}} \qquad \textbf{(Eq 6.6.2.7a)}$$

where

f_n = natural frequency, cps

k = spring rate, lb_f/in.

m_1 = mass on free end of bellows, lb_m

m_2 = mass of bellows, lb_m

Structurally Induced Vibration. Bellows can resonate in several modes: longitudinal (accordion), lateral, (transverse or beam), torsional and liquidus. Equations 6.6.2.7b and c below were adapted from work reported by Bell Aerosystems in Reference 81-10. It is important to note that experimental verification of these beam-theory expressions by Battelle Memorial Institute (Reference 44-37) showed good correlation for the accordion mode (Equation 6.6.2.7b), but very poor correlation for the lateral beam mode (Equation 6.6.2.7c). Observed lateral mode frequencies were consistently lower than those calculated by beam theory. The fundamental natural frequency, f_n, of the longitudinal (accordion) mode may be approximated for the condition of less than 1 psi differential pressure and both ends clamped, by the equation

$$f_n = 19.825 \sqrt{\frac{k}{m_{fa} + m_a}} \quad \text{(Eq 6.6.2.7b)}$$

where:

k = total equivalent spring rate of all convolutions, lb_f/in.

m_a = mass of active convolutions, lb_m

m_{fa} = mass of fluid trapped in active length at rest, lb_m

$$m_{fa} = \rho L \left[0.262 \left(D_o^2 + D_o D_i \right) - 0.524 D_i^2\right]$$

where:

D_i = inside diameter, in.

D_o = outside diameter, in.

ρ = density of fluid, lb_m/in^3

L = total length of all convolutions, in.

The fundamental lateral (so-called beam) mode may be approximated for the same conditions by (CAUTION: see text above):

$$f_{n=1} = 2.22 \sqrt{\frac{kD_o^2}{\ell^2 (m_a + m_\ell)}} \quad \text{(Eq 6.6.2.7c)}$$

where:

k = total equivalent axial spring rate of all convolutions, lb_f/in.

D_o = outside diameter, in.

ℓ = Live length of bellows, in.

m_a = mass of active convolutions, lb_m = lb_f-sec^2/in.

m_ℓ = mass of liquid in the bellows, approximately the length times the effective area times fluid density, lb_m = lb_f-sec^2/in.

The above was adapted from Reference 81-10 which also gives a means of calculating the frequency when there is a pressure difference across the wall. Torsional vibration is usually not a problem, but it may be of very high frequency and very low amplitude (estimated as little as 0.00001 inch on the periphery) so that even small clearances on fasteners may allow motion.

The liquidus mode is seldom encountered; however, it may be very severe when present (References 81-10 and 564-11). The liquidus mode occurs in the longitudinal direction when vibration-induced pressure surges in the contained liquid interact with the bellows and is more difficult to analyze. This mode usually does not occur in bellows used in small aerospace components such as valves and regulators but may have to be considered in propellant line and positive expulsion applications.

In Equations (6.6.2.7b) and (6.6.2.7c) the weight of the live convolutions must be supplied. This, of course, varies with the shape of the leaves or plates.

The masses (per convolution) of different shapes of bellows leaves may be approximated, as given in Table 6.6.2.3, if uniform density and wall thickness are assumed and the weld beads, reinforcing rings, etc., are ignored.

Flow-Induced Vibration. The phenomenon of bellows vibration excited by the flowing fluid is usually not a significant consideration in aerospace components. Recent experience with the Saturn 5 launch vehicle indicates that flow-induced vibration can cause difficulties in high-velocity feed system bellows joints. Reference 733-2 describes failures of the stainless expansion bellows with braided reinforcement in the 1/2-inch diameter stainless tubing that feeds liquid hydrogen to the augmented spark igniter (ASI) of the J-2 engines of the S-2 and S-4B stages. These bellows joints performed satisfactorily at the design flow rate of 1.1 lb/sec of liquid hydrogen during many ground tests, but failed in flight. Flow tests in a vacuum chamber resulted in bellows failure in less than 100 seconds due to a 20.8 kilocycle vibration induced by the vortices in the hydrogen flow over the bellows convolutions.

Testing revealed that the -400°F temperature of the liquid hydrogen caused ambient air to liquefy on the outer surface of the bellows, inside the wire braid. The liquid air damped the vibration of the bellows. Under vacuum conditions there was no ambient air to condense, and the undamped resonant vibration reached 500 g, resulting in rapid bellows failure. The high frequency of the bellows vibration was found to be related to resonant frequencies of the individual convolutions rather than that of the entire bellows. Correction of this flow-induced vibration problem consisted of redesigning the hydrogen line with the same tubing diameter but with the bellows deleted. (Reference 733-2.) (The bellows joints were originally incorporated to accommodate assembly/disassembly motion associated with installation of flow calibration orifices).

6.6.2.8 PRESSURE DROP. Pressure drop through bellows and corrugated flexible metal hose is treated in Detailed Topic 3.9.2.2 of this handbook. Pressure losses through internally-restrained bellows joints are largely a function of the restraint device characteristics (see Sub-Topic 5.13.5). References 1-406, 19-243, 23-59, and 766-1 treat pressure losses in bellows joints and flexible metal hose.

6.6.3 Design Data

This sub-topic provides basic information related to each of the general categories of bellows as distinguished by manufacturing technique. Table 6.6.2a summarizes the basic characteristics of typical aerospace bellows. Table 6.6.3 lists the steps involved in producing formed, welded, and deposited bellows.

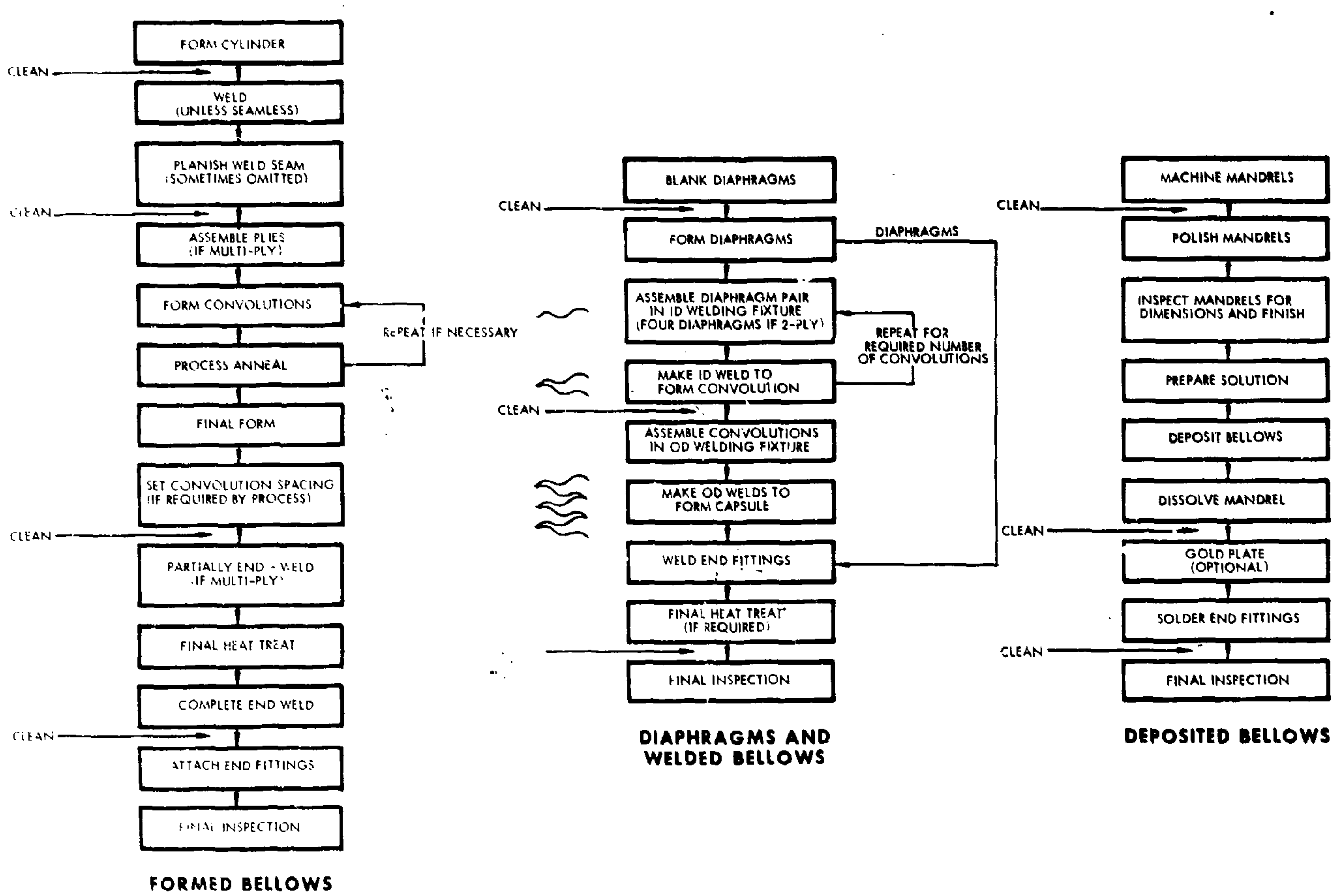
6.6.3.1 FORMED BELLOWS. *Formed metal bellows* (often called *convoluted bellows*) may be either single-ply or multi-ply. Bellows with many plies have been produced; multi-ply construction is used where

a) The deflection forces must be low

b) A long cyclic life is required

c) Resistance to high pressure must be obtained without sacrificing flexibility.

Table 6.6.3. Manufacturing Flow Sheet for Bellows



Figures 6.6.2.2a and b show a 3.5-inch diameter, 8400-psi proof-pressure bellows made of ten plies of 0.008-inch type 347 stainless steel, with root rings. External wire-mesh reinforcement was used with this bellows (Reference 34-10). The irregularities of the internal crests (Figure 6.6.2.2a) are indicative of the difficulties associated with very high pressure bellows.

Formed bellows (both single-ply and multi-ply) are formed from a tube (seamless tubing in small diameters, but usually containing a longitudinal seam weld in sizes over 1-inch diameter), either by hydraulic expansion or rolling. In hydraulic forming, fluid pressure expands the tube outwardly. The inside diameter remains approximately the same as the tube, while cold working is greatest in the flats and the outer crests. An advantage of the process is that a variety of head designs can be formed integrally with the bellows, eliminating welding, brazing, or soldering. Various types of formed bellows end-fitting joints are shown in Figure 6.6.3.1a.

In roll forming, successive rolls or passes deepen and narrow the corrugations. The diameter of the original tube is less than the outer diameter and greater than the inner diameter of the completed bellows; thus, cold-working is imparted to both inner and outer crests, resulting in good elastic and high-cycle fatigue characteristics. However, rolling can introduce surface damage and cover material flaws, and these problems have led to a reluctance to use rolled bellows for critical applications.

Limited testing of some 321 CRES bellows in the as-formed condition and in the annealed-after-forming condition indicated a much shorter life for the annealed specimens. Fatigue data for some formed bellows are shown in Figures 6.6.3.1b, c, d, and e. See also Equation (6.6.2.4a). Detailed treatment of formed bellows design, fabrication, stress, and fatigue life may be found in References 44-37 and 23-59.

Another type of formed bellows which is now used in some aerospace applications is the *nesting formed bellows* (also called *rippled sidewall bellows*). These bellows are usually of the single-sweep or rippled configuration (Figure 6.6.3.1f) and are used in lieu of welded nesting bellows in applications such as propellant expulsion devices. A typical nesting formed bellows is fabricated by hydraulically forming a U-shaped bellows and then reducing the root and crest of each convolution to near zero bend radius with a special machine-die combination in a controlled environment. Reference 332-33 describes a 10-inch diameter nesting formed bellows which was expanded to 40 times the initial compressed length (stack height) before rupture occurred, although edge yielding was observed at an expansion ratio of 25:1.

The stroke of a formed bellows for a given load is directly proportional to the number of active convolutions and inversely proportional to some power of the wall thickness. In one configuration (Reference V-136) under a given load, if the thickness of a one-ply bellows is doubled, the stroke is reduced seven-eighths; if two plies of equal thickness are used, the stroke is reduced by only one-half. Typical sizes and characteristics for stainless steel one-ply, two-ply and three-ply formed bellows are presented in Table 6.6.3.1.

6.6.3.2 WELDED BELLOWS. *Welded bellows* are made up of shaped diaphragms (leaves or plates) alternately welded together at the inner and outer radii (Table 6.6.3).

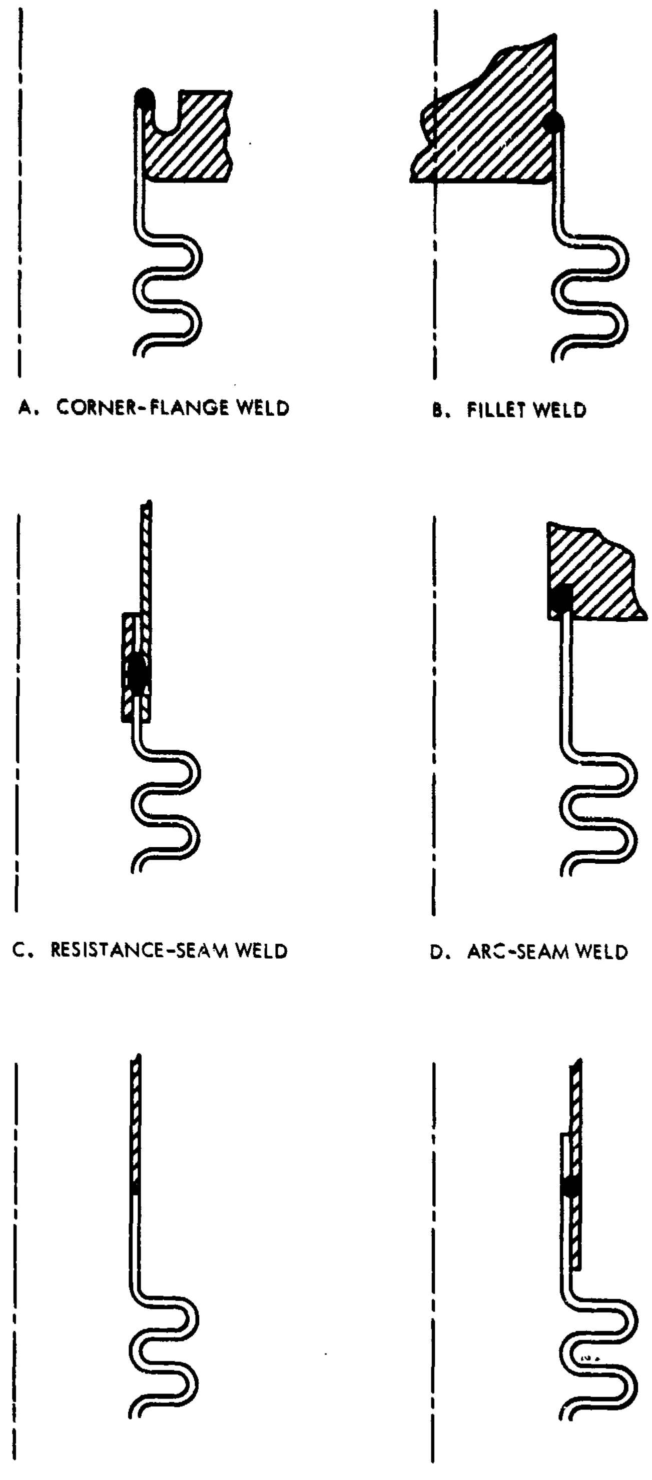


Figure 6.6.3.1a. Typical Formed Bellows End-Fitting Joints

Table 6.6.3.1. Typical Sizes and Characteristics of One, Two, and Three-Ply Convoluted Seamless Stainless Steel Bellows

(Courtesy of Robertshaw Fulton Controls Company)

	Outside Diameter, Inches	Inside Diameter, Inches	Maximum Number of Active Corrugations	Approximate Length Per Active Corrugation, Inches	Effective Area, Square Inches or Linear Volume, Cubic Inches	Spring Rate Per Active Corrugation Lbs. Per Inch	Maximum Stroke Rating Per Active Corrugation, Inches	Maximum Pressure Rating, Lbs. Per Sq. Inch
	D	d	Can be manufactured in any number of active corrugations up to number stated below.	Multiply by number of active corrugations to obtain normal free length of active portion of bellows.	Expressed as cubic inches is approximate volume per inch length of active portion of bellows.	Divide by number of active corrugations to obtain spring rate of active portion of bellows.	Multiply by number of active corrugations to obtain maximum stroke of active portion of bellows.	
STAINLESS STEEL, ONE-PLY	15/32	21/64	14	0.046	0.12	1600	0.004	619
	35/64	23/64	20	0.049	0.16	2230	0.004	612
	21/32	29/64	14	0.065	0.24	1850	0.006	526
	23/32	31/64	11	0.035	0.29	1640	0.009	536
	3/4	1/2	16	0.080	0.31	960	0.011	335
	3/4	31/64	12	0.081	0.30	1000	0.012	364
	31/32	23/32	14	0.068	0.56	4900	0.005	561
	1 1/64	41/64	13	0.087	0.54	198	0.036	115
	1 1/8	47/64	12	0.095	0.69	262	0.033	116
	1 1/8	47/64	10	0.082	0.69	2650	0.013	480
	1 11/32	31/32	12	0.092	1.07	2250	0.015	410
	1 3/8	29/32	9	0.098	1.03	430	0.031	122
	1 1/2	31/32	11	0.114	1.21	3450	0.015	395
	1 1/2	63/64	10	0.140	1.23	210	0.050	81
	1 1/2	1	11	0.137	1.24	1700	0.019	250
	1 19/32	1 7/64	12	0.090	1.45	3130	0.015	343
	1 41/64	1 7/64	8	0.152	1.50	232	0.040	61
	1 3/4	1 7/64	9	0.136	1.62	410	0.060	132
	2	1 11/32	14	0.110	2.22	4000	0.023	395
	2	1 1/2	20	0.144	2.43	394	0.035	72
	2 5/32	1 1/2	15	0.150	2.64	340	0.060	80
	2 23/32	2 13/64	13	0.132	4.77	8500	0.013	392
	2 31/32	2 29/64	7	0.174	5.84	640	0.039	76
	3 31/32	3	7	0.217	9.63	100	0.100	17
	7 3/4	6 7/8	15	0.297	42.00	3400	0.022	49
	9 23/64	8 9/16	14	0.222	63.07	6200	0.018	59
STAINLESS STEEL, TWO-PLY	43/64	7/16	10	0.071	0.24	1340	0.010	526
	3/4	15/32	13	0.083	0.30	910	0.012	328
	15/16	39/64	22	0.080	0.48	750	0.020	291
	1 9/64	49/64	8	0.088	0.72	2100	0.015	438
	1 11/32	7/8	14	0.142	0.98	950	0.032	287
	1 1/2	1	13	0.147	1.24	3300	0.020	500
	1 11/16	1 1/8	19	0.143	1.57	1900	0.027	316
	2	1 21/64	11	0.153	2.20	530	0.061	139
	2 13/32	1 9/16	7	0.212	3.14	375	0.089	98
	2 5/8	2	16	0.229	4.23	13,400	0.009	361
	3 5/8	3	14	0.194	8.65	19,200	0.008	333
	4 5/8	4	16	0.257	14.61	25,000	0.008	319
STAINLESS STEEL, THREE-PLY		3/4	24	0.127	0.70	7630	0.009	968
		1 1/32	12	0.164	1.27	5600	0.011	822
		1 5/8	15	0.156	2.27	2700	0.025	313
		1 15/32	10	0.174	2.58	6500	0.022	566
		1 3/4	10	0.239	3.46	4500	0.025	368
		3 25/32	10	0.238	14.10	4200	0.039	189
		7 7/8	7	0.283	55.36	2400	0.083	96

ISSUED: FEBRUARY 1970
SUPERSEDES: MAY 1964

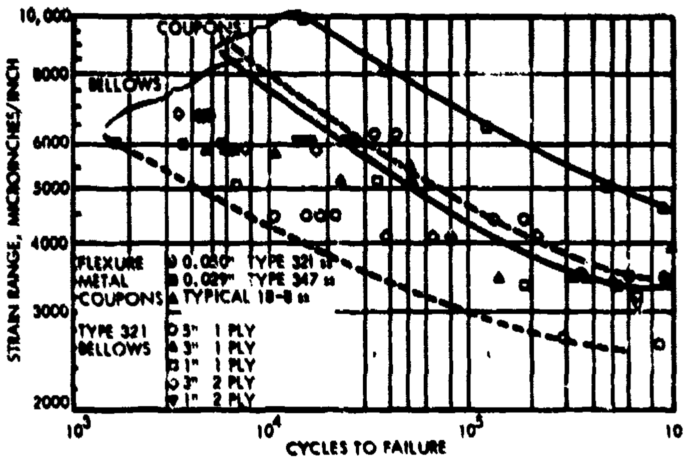


Figure 6.6.3.1b. Bellows Fatigue Data, Stainless Steel
(Reference 44-37)

	Reference	Dm	Type of Motion	Int. Press.	No. Samples	σb/σB
○	Gord. RPT21-56-1	4.287 to 6.766	Axial	10 psig	48	30 to 173
△	Flexon 105410	10.3-10.4	Angular	140 psig	18	2.5 to 6
□	SSP RPT 2175	3.52	Axial	10 psig	27	23 to 49
●	SSP RPT 2074	17.85	Angular	100 psig	2	1.17
■	SSP RPT 2051	19.05	Angular	100 psig	4	2.06

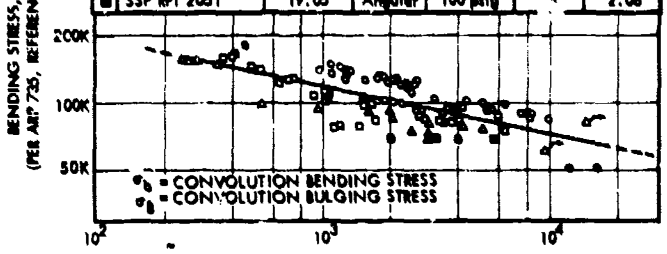


Figure 6.6.3.1c. Bellows Fatigue Data, Type 321 CRES
(Courtesy of Rocketdyne Division of North American Rockwell Corp.)

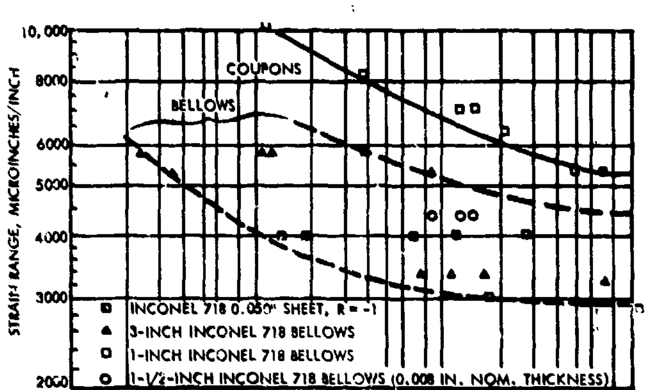


Figure 6.6.3.1d. Bellows Fatigue Data, Inconel 718
(Reference 44-37)

Basic designs of welded bellows are: conical flat plate, stepped flat plate, nesting ripple, single sweep, and toroidal. Each corrugation of a welded bellows is a capsulated diaphragm, as described in Sub-Section 6.7. Primary aerospace applications for welded bellows are valve seals, shaft seals, and positive expulsion devices.

The design of end fittings for welded bellows differs from those for formed types. End fittings can be made from strip, sheet, bar, castings, or forgings. Since welded bellows have precise control of outside and inside diameter dimensions, the diameter of the matching surface of the fitting should be held to within 0.001 inch. Fittings made from strip or sheet are likely to deflect when load is applied to the bellows.

Usually it is better design practice to join the end fitting at the outside diameter rather than at the inside diameter. Figure 6.6.3.2a illustrates good design practice for welding end fittings. Soldering and brazing may also be used. With a nested design, attaching the end fitting at the outside diameter requires clearance for the diaphragm curvature at one end.

Although they are more expensive to manufacture than formed bellows, welded bellows offer three significant advantages over formed bellows:

1) A wider choice of material
2) More deflection per unit length, resulting in shorter assemblies or longer strokes
3) A wider choice of performance characteristics because of a greater variety of convolute dimensions and shapes.

A few companies offer a two-ply welded bellows, but most welded bellows have a single ply. In general, welded bellows are available in sizes from 1/2-inch to 7-inches outside diameter, although bellows in excess of 12 inches in diameter have been produced.

Most welded bellows are of the nested-ripple configuration because this design makes maximum use of the advantages of low spring rate and compactness. However, the other configurations have attractive characteristics for

	Material	Reference	Dm	Type of Motion	Int. Press.	No. Samples	σb / σB
◊	Hastelloy C	SSP RPT 2162	8.935	Combined	50 psi	5	8.7 to 24
♦	Inconel 718	Solar Data	4-12	Axial	10 psi	19	∞ to 50
△	Inconel X	Solar Data	4-12	Axial	10 psi	21	∞ to 1
▲	Inconel X	Solar F-1 Oxid.	18.185	Axial	125 Ext.	2	2.9
○	Hastelloy C	D-K 216-8680	13.075	Combined	56 psi	2	2.9
●	Hastelloy C	D-K 226-1775	5.428	Combined	126 psi	1	2.4
⬭	Hastelloy C	Arrowhead 27"	28.325	Combined	65 psi	1	5.2
○			5.2	Angular	740 psi	4	.97 to 2.4
■			2.2	Axial	20 psi	6	1.1 to 2.4
□			2.2	Axial	20 psi	5	1.8 to 3.7

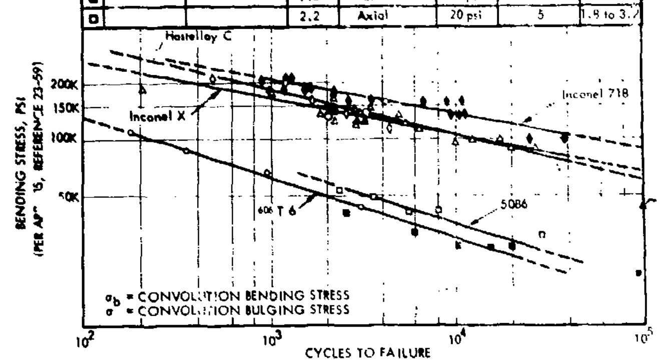


Figure 6.6.3.1e. Bellows Fatigue Data, Aluminum and Nickel - Based Alloys
(Courtesy of Rocketdyne Division of North American Rockwell Corp.)

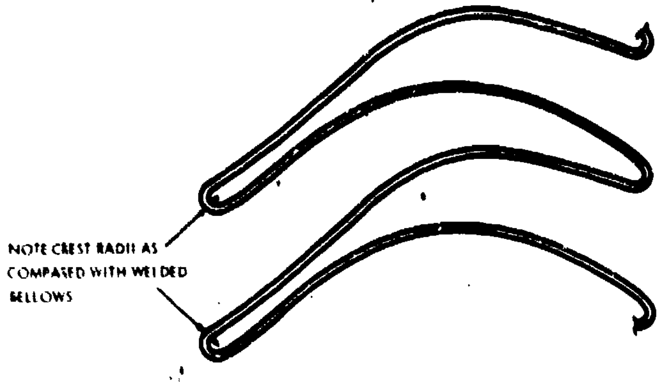


Figure 6.6.3.1f. Convolution Profile of Nesting Formed Bellows at Full Length

certain applications. Despite the impressive welding techniques that have been developed by the manufacturers of welded bellows, the large amount of welding required (approximately 18 inches per convolution in a 3-inch-OD bellows) makes fatigue failure less predictable for welded bellows than for other types of bellows (Reference 44-37).

One potential source of failure unique to welded double-ply bellows is the possibility of omitting one of the diaphragms during convolution assembly or welding the OD of one ply to the wrong convolution (Figure 6.6.3.2b). Both errors have occurred in practice and both are virtually impossible to detect by inspection.

Change in effective area, A, must be considered with welded bellows for certain applications, such as shaft seals. Figure 6.6.3.2c illustrates how pressure can force the diaphragms of some welded bellows configurations into increased contact with a resultant change in mean diameter and effective area. Nesting ripple, single sweep, and conical flat plate configurations are most susceptible to this phenomenon.

The following illustration of how to design a welded bellows with either flat or nested ripple contour is empirically based and uses no sophisticated stress analysis. It is reprinted with permission from "Welded Diaphragm Bellows—Part II," by V. W. Wigotsky, *Design News*, February 27, 1961, copyrighted by Cahners Publishing Company, Inc. (Reference 73-8). Figure numbers have been adapted to the handbook format. Design information presented in the article was provided by the Metal Bellows Corp., Wellesley, Massachusetts.

The design curves and charts were prepared from test data and mathematical analysis and are based upon AM 350 (AMS 5548), a precipitation hardening stainless steel, heat-treated to Rc 42-45 hardness. Curves on spring rate are approximate and serve to orient the designer rather than pinpoint exact values. Curves on life expectancy are conservative. The flat plate and nesting ripple contours generally are used in the zero to 300 psi pressure range and require a similar design approach. The single sweep and torus designs generally are used above 300 psi and require considerably different treatment.

Valve application. The valve poppet is to be attached to a pressure-operated bellows. Maximum pressure differential is 20 psi external to bellows. Assume the effective area must be approximately 0.44 square inches (related to area of valve seat). The valve stroke is one-quarter inch. A pressure of 20 psi will actuate the valve through full stroke and apply 4.4 lb_f on the seat. Linearity of stroke with pressure is desirable but not essential. Life will be 10,000 cycles. The valve operates at room temperature with no corrosive fluids; AM 350 is acceptable. Maximum allowable D_o is 1.031 inches, and it is desirable to make the length as short as possible. Since pressure is low and a small package is desired, the nesting ripple looks satisfactory. Bellows must move one-quarter inch under a 20-psi pressure, and exert a force of 4.4 pounds on the seal.

(Pressure) (Area) = Force

$$\text{Pressure} = \frac{4.4 \text{ lb}}{0.44 \text{ sq in}} = 10 \text{ psi}$$

Therefore, the remaining 10 psi are available for stroking the bellows. The A/K must equal 0.25 in/10 psi; therefore,

$$\frac{A}{k} = \frac{0.44}{k} = \frac{0.25}{10}$$

$$k = 17.6 \text{ lb/in}$$

The longest stroke per convolution will result in the least number of convolutions required and will, therefore, be shorter in overall length and cost less to manufacture. Referring to Figure 6.6.3.2d, showing pitch for nesting ripples, we note that the largest span shown has the longest stroke. Checking span to D_o ratio, if we use 0.350 span, D_o becomes 0.748 + 0.350 = 1.098 (0.748 is mean diameter established by A) for a ratio of 0.350/1.098 or 0.319. This is less than 1 to 3, but just under limit. However, this span is too large, since maximum D_o is 1.031 inches. Considering 0.300-inch span, D_o becomes 0.300 plus 0.748 = 1.048, which is still too large. Trying 0.250-inch span, 0.250 + 0.748 = 0.998, which is acceptable. Ratio of span to D_o is 0.250/0.998 = 0.251 (less than 1 to 3). Maximum deflection (Figure 6.6.3.2d) is 0.046-plus per convolution for 0.003-inch thick diaphragm. Referring to spring rate (Figure 6.6.3.2e), we can expect about 160 lb_f/in/convolution times 0.748 = 120 lb_f/in/convolution; then (120 lb_f/in/conv)/(17.6 lb_f/in) = N = 6.84 conv (use 7). A stroke of 0.046 in/convolution times 7 convolutions = 0.322-inch total stroke possible, which is well over 0.250-inch valve lift. If we consider a smaller span, 0.200-inch with 0.0025-inch thick material, k = 240 lb_f/in/convolution times 0.748 = 179.5 lb_f/in/convolution; which would require

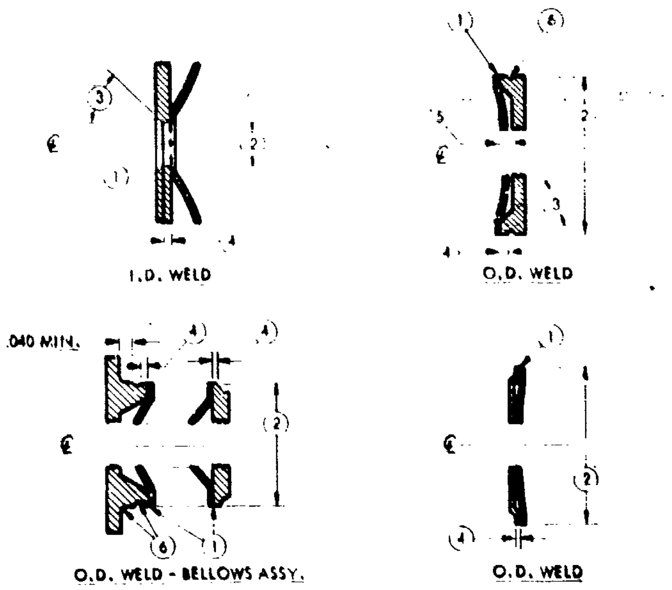
$$\frac{179.5 \text{ lb}_f\text{/in/convolution}}{17.6 \text{ lb}_f\text{/in.}} = 10.2 \text{ convolutions}$$

Therefore, the 0.250-inch span and 0.003-inch thickness seems best.

Let us examine life expectancy. Referring to Figure 6.6.3.2f, life factor (LF) is 3.5. In order to obtain LF for stroke, we determine stroke per convolution to be

$$\frac{0.250 \text{ inch}}{7 \text{ convolutions}} = 0.0357 \text{ in/convolution.}$$

From Figure 6.6.3.2g, we read LF = 6.3. Therefore, 3.5 + 6.3 = 9.8 total LF. Life expectancy (Figure 6.6.3.2h) indicates about 12,000 cycles.



LEGEND

1. WELD AREA
2. .001 F.I.R. CONCENTRICITY BETWEEN BELLOWS AND FLANGE
3. 45° MAXIMUM ALLOWABLE ANGLE
4. TWO TIMES BELLOWS MATERIAL THICKNESS
5. PITCH OF BELLOWS MINIMUM
6. .030 RADIUS MINIMUM ON ALL FILLET

Figure 6.6.3.2a. Methods of Attaching End Fittings to Welded Bellows
(Courtesy of the Belfab Corporation, Daytona Beach, Florida)

From the foregoing we determine the final bellows design specification to be:

Contour = nesting ripple

Material = AM 350

$t = 0.003$ in.

Span = 0.250 in.

$D_o = 0.250 + 0.748 = 0.998$ in.

$D_i = 0.748 - 0.250 = 0.498$ in.

$N = 7$

$$k = \frac{120\ \text{lb}_f\text{/in/convolution}}{7\ \text{convolution}} = 17.1\ \text{lb}_f\text{/in.}$$

Pitch = 0.046-inch maximum stroke + 0.010 NP = 0.056 in.

Free length of bellows = 7 convolutions times 0.056-in. pitch = 0.392 in.

6.6.3.3 MACHINED BELLOWS. Machined bellows are turned or ground from barstock, tubing, or forged rings of most materials used in other types of metallic bellows, as well as of materials not found in sheet stock. High strength, high endurance, heat-treatable tool steels, in addition to high strength, low modulus titanium alloys are among the materials now eligible.

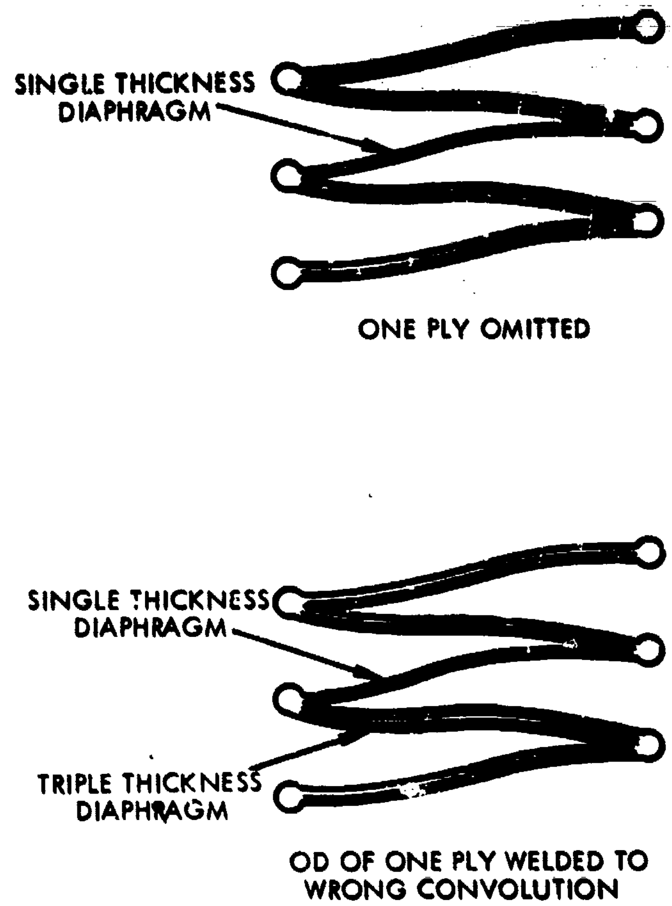


Figure 6.6.3.2b. Potential Fabrication Errors in Assembly of Multi-Ply Welded Bellows

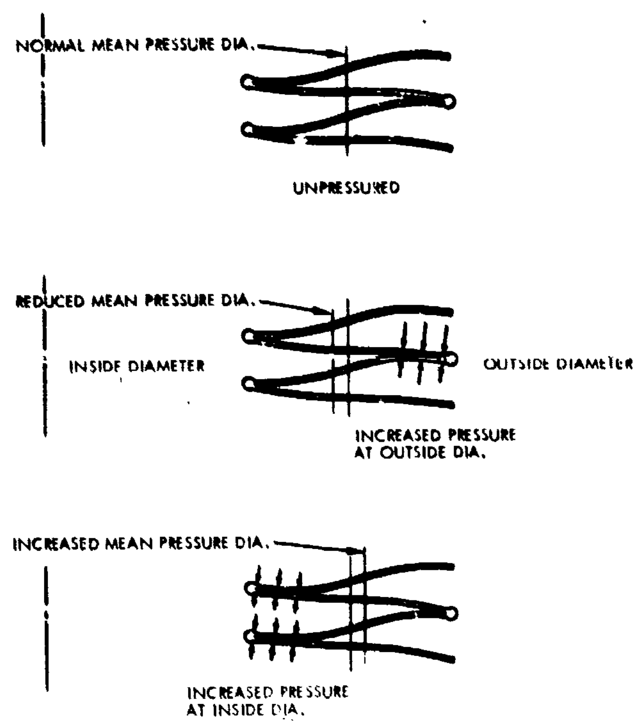


Figure 6.6.3.2c. Change in Bellows Effective Area with Pressure

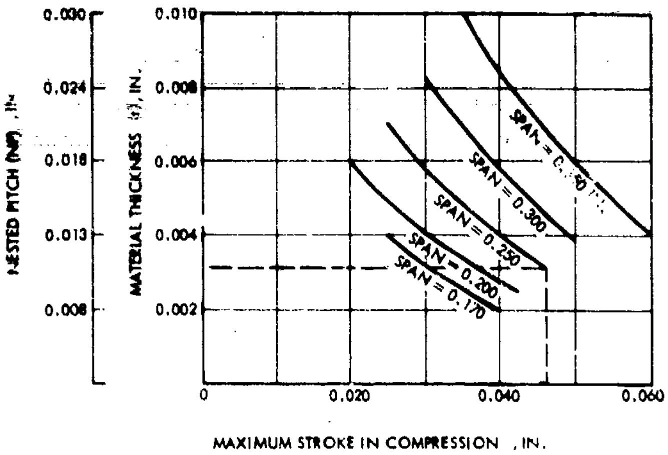


Figure 6.6.3.2d. Pitch for Nested Ripple Bellows of AM 350

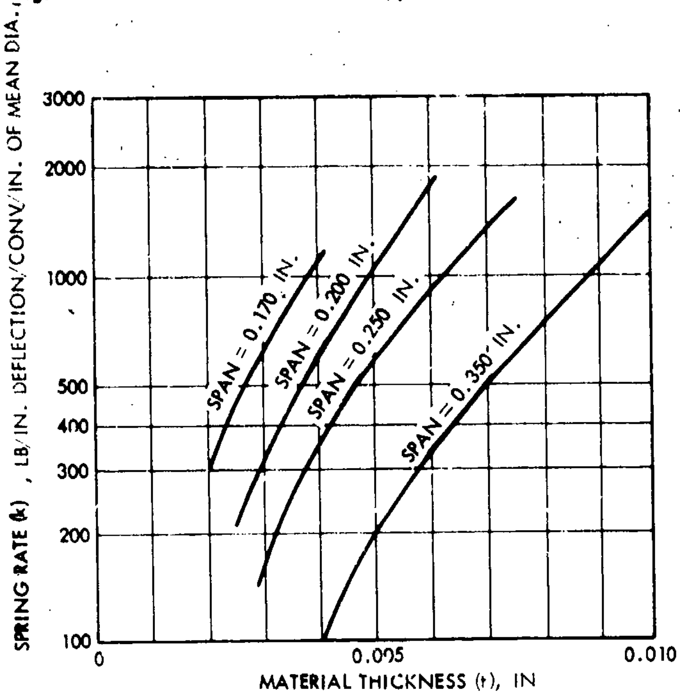


Figure 6.6.3.2e. Pitch for Nested Ripple Bellows of AM 350

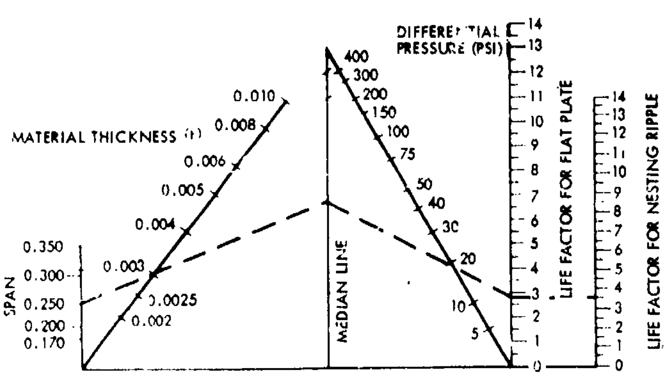


Figure 6.6.3.2f. Pitch for Nested Ripple Bellows of AM 350

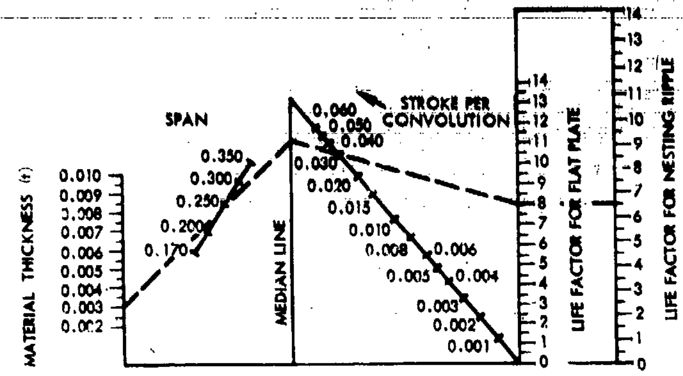


Figure 6.6.3.2g. Pitch for Nested Ripple Bellows of AM 350

The design of machined bellows is highly customized; AM 350 manufacturing of bellows with deep convolutions required for low spring rates is difficult to achieve. Strokes range from one percent of the diameter to nearly 30 percent in special cases. Minimum diameter is approximately 0.250 inch. Machined bellows have been made as large as 60-inches in diameter and for pressures as high as 12,000 psi. Machined bellows maintain accuracy and have high life expectancy with high pressure applied because the effective area can be made uniformly constant throughout the stroke up to the yield point of the material.

6.6.3.4 DEPOSITED BELLOWS. Two kinds of deposited

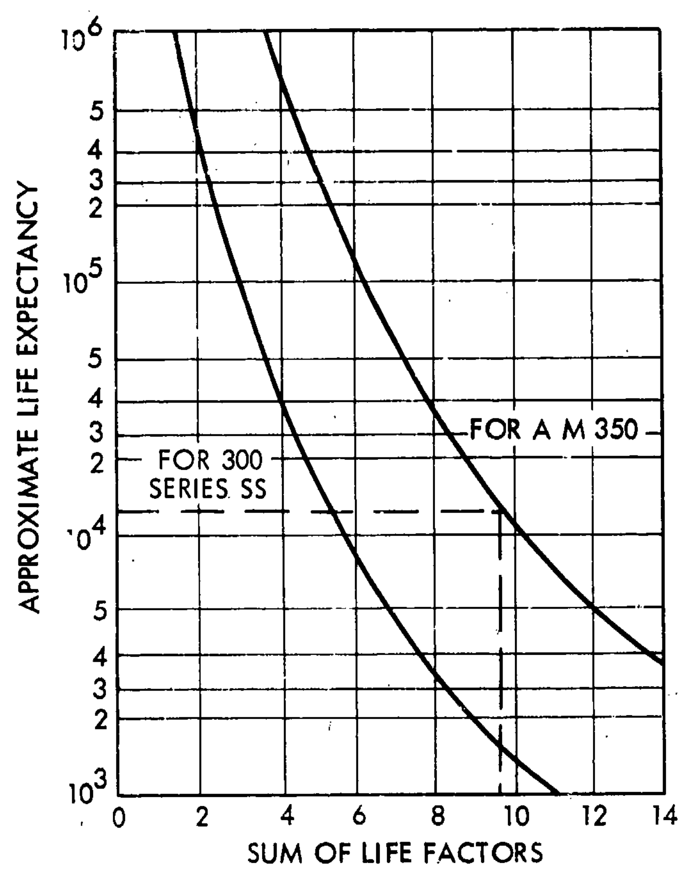


Figure 6.6.3.2h. Pitch for Nested Ripple Bellows of AM 350

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SUPERSEDES: MAY 1964

bellows are commercially available: chemically deposited and electrodeposited. Both methods can be used to produce any shape that can be deposited on a machined mandrel (see Table 6.6.2a). In each method an aluminum mandrel is machined for each bellows, and after the bellows material is deposited, the mandrel is dissolved. The primary advantages of the processes are the ability to produce:

1) Very thin-walled bellows

2) Bellows having no welds

3) Very small bellows

4) Special-shaped bellows.

Chemically deposited bellows are made with wall thicknesses from 0.0003-inch to 0.005-inch, and with diameters from 0.060-inch to 12-inches. Electrodeposited or electroplated bellows are usually produced in nickel or nickel-cobalt alloy. Sizes are available from 0.063-inch to 1.250-inches in diameter, with wall thicknesses varying from 0.0003-inch to 0.006-inch.

REFERENCES

*References Added March 1967
**References Added November 1968

Formed Bellows

1-269, 19-243**, 23-59**, 31-21**, 34-10**, 44-31*, 332-33**, 513-1*, 513-3*, 513-5**

Welded Bellows

1-269, 1-421**, 1-506**, 44-31*, 44-37**, 73-6, 73-8, 73-114**, 73-9, 81-6**, 81-8**, 81-9**, 81-10**, 156-1

Deposited Bellows

44-37**

Non-Metallic Bellows

73-1

Bellows Spring Rate

1-269, 1-421**, 23-59**, 31-21**, 44-37**, 73-8, 73-9, 73-114**, 81-10**

Flexible Joint Bellows

1-506**, 19-243**, 23-59**, 31-21**, 34-10**, 44-37**, 73-6, 513-1*, 513-5**, 621-1**, 733-2**

Expulsion Device (Nesting Bellows)

81-6**, 81-8**, 81-9**, 81-10**, 332-33**

Flow Through Bellows

1-506**, 19-243**, 766-1**

Bellows Stress Analysis

31-21**, 44-31**, 44-37**, 73-8, 73-9, 73-114**, 81-10**, 513-3*, 513-5**, 621-1**

Bellows Dynamic Analysis

44-37**, 81-10**, 564-11**, 733-2**

Manufacturer's Design Data

V-117, V-136, V-278

Miscellaneous

1-421, 19-116, 156-1

6.7 DIAPHRAGMS

6.7.1 Introduction

A diaphragm is a thin dividing membrane which can be used as a seal to prevent the interchange of liquids or gases between two chambers, and also as an actuator to transform pressure into linear motion and force. Used as a seal, the diaphragm acts as a closure in shutoff valves. Typical examples are illustrated in Figures 5.2.7.1b and 5.2.8.1a. Used as an actuator, the diaphragm integrates pressure over an effective area resulting in a force which can be used to position control element in regulators, to actuate micro-switches in pressure switches, or to deform beam members in pressure measuring devices. Typical examples are shown in Figures 5.4.4, 5.15.2.7a, and 5.16.3.1c.

In general, diaphragms are used for converting any one of the quantities (force, pressure, deflection, and volume change) into one or more of the remaining quantities. Some specific diaphragm applications are listed below:

1) Diaphragm-type actuators where the actuator is sealed off from the fluid medium.

2) Pressure switches where the diaphragm actuates a microswitch.

3) Capacitance-type displacement transducers where the displacement of the diaphragm, which is one of two plates of a capacitor, changes the capacitance value.

4) Volume compensators.

5) Temperature sensors and controllers.

6) Pneumatic force-balance force transducers.

7) Acoustic drivers in acoustic instrumentation.

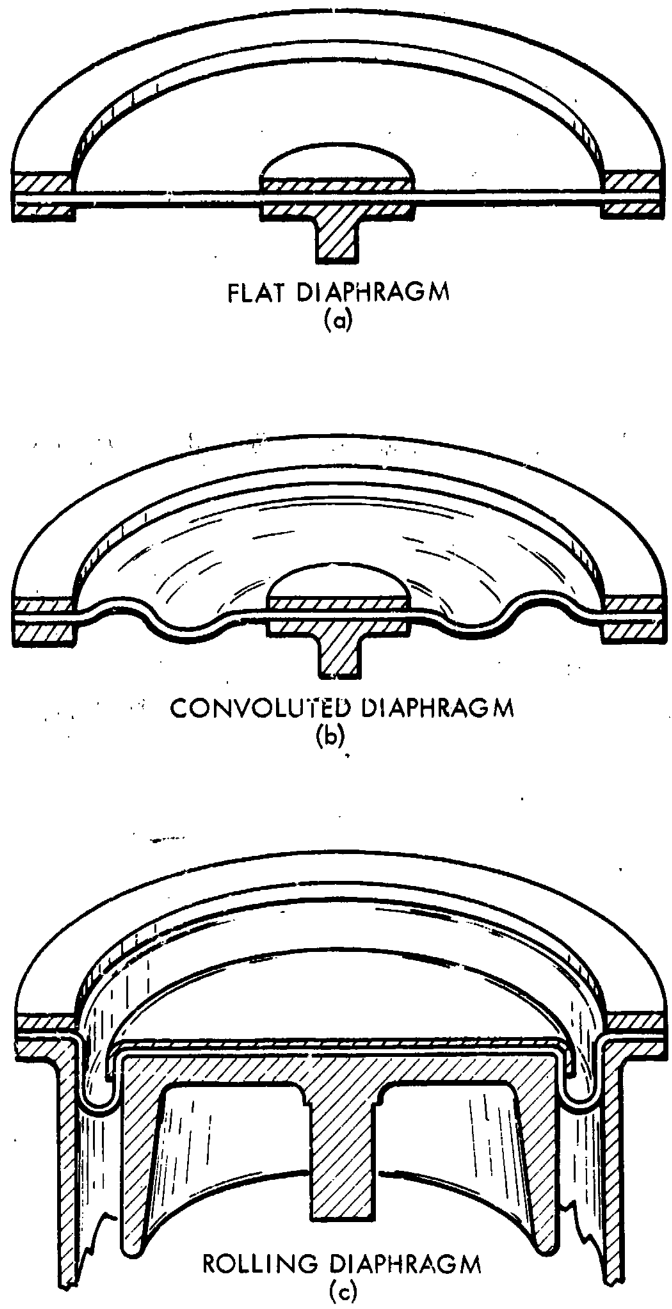

Figure 6.7.1. Three Basic Diaphragm Configurations

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8) Rolling diaphragm expulsion devices in fuel/oxidizer tanks.

Three basic diaphragm forms are *flat*, *convoluted*, and *rolling* (Figure 6.7.1). No universally accepted classification of diaphragms exists. Some references combine flat and convoluted diaphragms into the one category of flat diaphragms which are then categorized as *true flat* and *corrugated flat*. Metallic, convoluted diaphragms are sometimes inverted and joined to form *capsule* elements.

This sub-section discusses design factors and presents design data which will assist the designer in the selection of a particular diaphragm. The reader is also directed to the references listed at the end of this subsection. Table 6.6.3(b) gives the manufacturing steps involved in metal diaphragm fabrication

6.7.2 Design Factors

Factors which must be considered in the selection and design of a particular diaphragm type or configuration for a given application include: choice of materials, fatigue life, stroke, hysteresis, flexibility, and the method of installation. Design considerations influencing these factors are discussed in the following detailed topics.

6.7.2.1 MATERIALS. Diaphragm materials include metal alloys, plastics, elastomers, and elastomer-fabric composites. The most commonly used metal alloys are stainless steel, bronze, brass, and beryllium copper; plastics used for diaphragms include Teflon, Kel-F, and Mylar; fabrics commonly used are cotton, Dacron, nylon, Teflon, or Fiberglas; elastomers include neoprene, Buna-N, butyl, polyacrylics, polysulfide, silicones, and fluorocarbon rubbers. The selection of materials depends upon balancing the following properties: strength, resiliency, formability, compatibility, permeability, and dimensional stability. Table 6.6.2.1 lists metals commonly used in bellows and diaphragms.

The strength of the diaphragm must be sufficient to withstand the tensile stresses imposed by the pressure and must be matched against the resiliency necessary to afford sufficient deflection. Diaphragm stresses and deflections can be calculated using the theory of plates and shells (see Sub-Section 14.10). For convoluted diaphragms with small deflections, the equation for calculating stresses in thin-walled cylinders is approximately correct. Calculations should be based on allowable overpressures from 125 to 150 percent.

Formability is an important consideration for metals. Soft, ductile materials are easily formed, while very hard materials require that manufacturing processes be matched with heat treatment requirements which should include a stress-relieving process. Where deep convolutions are required, it is sometimes necessary that forcing be done in steps, annealing between steps to avoid excessive work hardening. Fabric-elastomers are generally molded and are quite free from formability problems.

The material of the diaphragm must be compatible with the materials, gases, or liquids contacting its surface in terms of chemical compatibility and permeability. For dependable performance, elements of corrosion, galvanic action, heavy oxidation, reduction, or deposition should be avoided. When mounting a metallic diaphragm in a metallic housing or attaching metal components to it, consideration must be given to the possible effects of galvanic couples. A comparison of permeability characteristics of several diaphragm materials is given in Table 3.11.5a, and a comprehensive treatment of permeability for aerospace applications may be found in Reference 152-12.

Diaphragm materials must also be compatible with the thermal environment. At temperatures above 300°F, diaphragm materials for extended service are limited to metals. At cryogenic temperatures, diaphragm materials are limited to metals and a few plastics which include Teflon, Kel-F, and Mylar.

The dimensional stability of a metal diaphragm is adversely affected by creep, which results in an increase in deflection under a constant load. Soft metals tend to creep more than hard metals. In plastic and elastomer diaphragms, cold flow affects dimensional stability. With elastomer-fabric diaphragms, the ratio of elastomer thickness to fabric thickness should be kept to a minimum. Diaphragms with large ratios develop leaks or pull loose after service.

6.7.2.2 FATIGUE LIFE. The *fatigue* or *cycle life* of a diaphragm is a function of the stress level. The fatigue life can be extended by operating at a percentage of the maximum design stress (i.e., a percentage of the maximum allowable stroke).

Fatigue life is determined from S-N curves (stress and number of cycles) for each material (See Sub-Section 14.5 and Figures 6.6.3.1b, c, d, and e). In general, nonmetallic diaphragm materials have good fatigue resistance and are suited to long cyclic life applications, whereas the fatigue life of metal diaphragms is limited, particularly at high stress levels.

In designing a diaphragm, particularly a metal diaphragm, the designer should first estimate the cumulative number of cycles likely to be experienced in the life of the diaphragm and then select the material, dimensions, and heat treatment conditions such that the working stresses are kept below the fatigue strength corresponding to the required cycle life. High stresses resulting from stress concentrations usually determine fatigue life limitations. Diaphragms with section property changes which could cause high stress concentrations should be investigated experimentally by using a stress coat or other strain evaluation technique. Stress concentration levels should be reduced by changing material thickness, diaphragm shape, and/or diaphragm diameter.

Because of the thin materials of construction, diaphragms are very susceptible to corrosion-induced failure. To prevent this, three precautions must be taken.

1) Diaphragm material should be selected so that little, if any, corrosion will occur under normal operating conditions.
2) The manufacturing procedure should be reviewed in detail to determine that corrosion will not be initiated during fabrication and that the finished item will be completely free of corrosive substance.
3) Where the diaphragm is to be used, it should be ascertained that there is no creation or introduction of corrosive agents in the fluid system. Table 13.7.9 gives environments in which stress-corrosion cracking of some alloys has been observed. The compatibility of various diaphragm materials with rocket propellants is listed in Sub-Section 12.5.

6.7.2.3 STROKE, LINEARITY, AND SPRING RATE. The stroke of a diaphragm is its deflection in a direction normal to its mounting plane. Flat, metallic diaphragms have the shortest stroke, while rolling diaphragms have the longest stroke. In general, nonmetallic diaphragms have several times the stroke of metallic diaphragms, and capsulated diaphragms (Detailed Topic 6.7.3.4) have twice the stroke of single convoluted diaphragms.

It is usually desired that the ratio of force to stroke be constant or, in other words, that the pressure-stroke plot be linear. Linearity is frequently the criteria for determining the range of useful displacement of a diaphragm. Typical curves for flat and single convoluted diaphragms are shown in Figure 6.7.2.3.

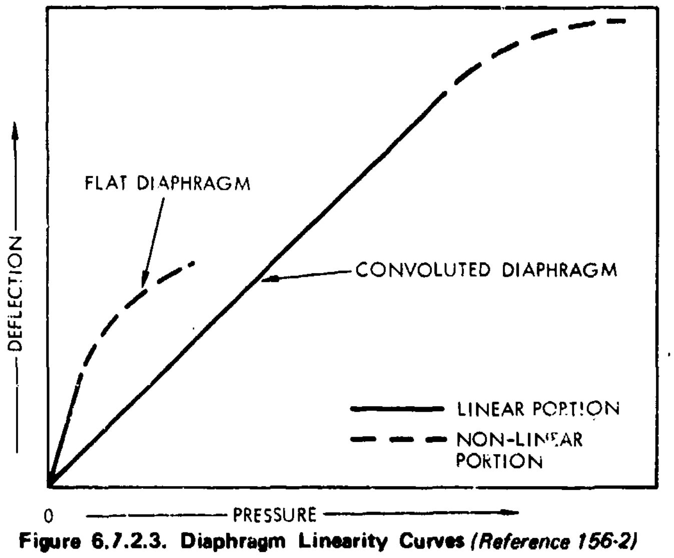


Figure 6.7.2.3. Diaphragm Linearity Curves *(Reference 156-2)*

With a single diaphragm, nonlinearity can be reduced by (1) making the convolutions fewer and deeper, or (2) decreasing the deflection span by adding a parallel spring or making the material thicker. When diaphragms are capsulated, nonlinearity errors can be reduced by matching dissimilar diaphragms, selected so that the nonlinear properties of one cancel the nonlinear properties of the other.

The spring rate or flexibility for compliance of a diaphragm is nonlinear. This is evident from the nonlinear equations of deflections of diaphragms (Detailed Topic 6.7.3.1). For a given deflection it is relatively easy to determine uniform load from deflection equations; however, to determine deflection from a given loading condition, a trial and error method is used. The spring rate is usually easy to measure but is very difficult to predict accurately, especially for convoluted diaphragms.

Table 6.7.2.3 shows the comparison of the theoretically predicted values with the experimentally measured value of the spring rate of a corrugated diaphragm (Reference 44-37).

6.7.2.4 FLEXIBILITY AND SPRING RATE. Flexibility, which is the inverse of spring rate, is the ratio of deflection to applied force. Flexibility is achieved with low modulus materials and with thin sections. The degree of flexibility is limited by the strength required to withstand the pressure forces.

Table 6.7.2.3

Direction of Loading	Theoretical: Calculated Using Linear Model	Theoretical: Calculated Using Nonlinear Model	Experimental
Upward	254	256	212
Downward	250	254	234

6.7.2.5 ANELASTIC EFFECT (HYSTERESIS) AND TEMPERATURE EFFECTS. Terms related to anelastic effect are defined below. (See references 68-88 and 44-37.)

a) *Hysteresis* is the difference between the deflection of a diaphragm at a given load for increasing and decreasing loads.

b) *Drift* is the increase of deflection with time under a constant load.

c) *Aftereffect* is the deflection remaining immediately after removal of the load, i.e., hysteresis at no load.

d) *Recovery* is the decrease of aftereffect with time under no load.

e) *Zero shift* is the permanent deformation at no load. It is the difference in position before loading and sufficiently long after unloading for recovery to occur.

At present, there is no generally accepted basic theory to predict the elastic behavior of diaphragms. However, there are some guidelines for design.

a) Some anelastic behavior generally causes larger errors at high stresses, therefore diaphragms designed to operate at a low peak stress are desirable for precision instruments.

b) The terms defined previously are interrelated. Thus, a diaphragm with small drift would be expected to have a small hysteresis and small aftereffect.

c) Relatively simple tests, such as measurement of drift, may be helpful in quality control to produce units of low hysteresis. However, some precautions must be used in the time schedules for anelastic determinations.

Temperature Effects. In general, an increase in temperature has two effects: it causes an expansion of the metal and, in most cases, a decrease in the modulus of elasticity. The effect of the former is to disturb the neutral position (zero shift in instrumentation). The effect of the decrease in modulus of elasticity is to decrease the spring stiffness of the diaphragm.

The safest way to minimize both effects is to select materials having very low coefficient of thermal expansion and minimum change of modulus of elasticity with temperature change. The zero shift can further be minimized by selecting proper materials for diaphragm and housing to match the coefficient of thermal expansion or by prestretching the diaphragm. The tension force necessary to prestretch the diaphragm will depend upon the magnitude of temperature change. However, prestretching will increase the stiffness and the natural frequency of vibration of the diaphragm. One method of prestretching a flat diaphragm is shown in Figure 6.7.2.5.

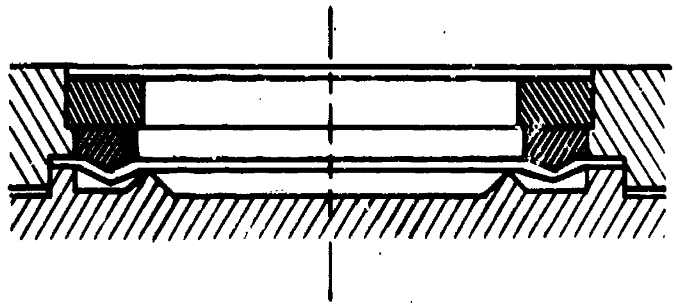


Figure 6.7.2.5. Prestretching Technique for a Flat Diaphragm

6.7.2.6 EFFECTIVE AREA. The effective area of the diaphragm determines the force output of the actuator for given pressure difference. As the actuator strokes, the effective area may change, altering the output force. Referring to Figure 6.7.2.6, a diaphragm is shown in two different positions. If a pressure, p, is applied to the underside of the diaphragm, the effective area being acted on by the pressure is that area normal to the vertical axis, plus a component due to the unsupported section of the diaphragm. The magnitude of the component is inversely proportional to the angle of tangency where it meets the diaphragm plate. In the lower position, the diaphragm is drawn tight and meets the plate at an angle, α. As the diaphragm pad is moved, the angle increases, but the vertical component of the diaphragm's effective area decreases.

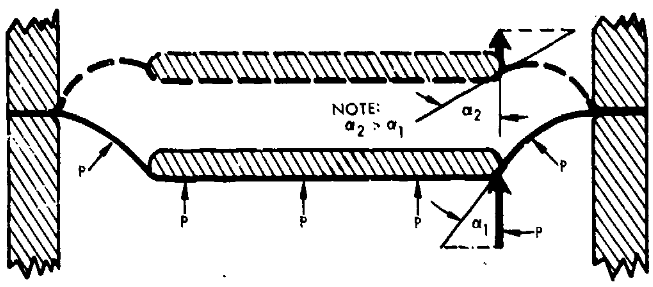


Figure 6.7.2.6. Diaphragm Effective Area Due to Stroke

The effective area change for metallic flat and convoluted diaphragms is small because the stroke is correspondingly small. With nonmetallic flat and convoluted diaphragms which are capable of longer strokes than metallic diaphragms of the same design, the effective area changes are usually large. With rolling diaphragms, the effective area is not changed as long as the 180 degree convolution remains.

6.7.2.7 INSTALLATION. Diaphragms are mounted to the stationary housing at their outer rim, while the center of the diaphragm usually contains an elevated boss or center plate for attachment of the input or output mechanism.

The outer edge of the diaphragm is clamped between flanges which are normally an integral part of the housing. Sometimes a bonding agent is also used to ensure diaphragm retention and sealing. With elastomeric diaphragms, clamping pressures must be closely regulated so that a seal is obtained without extruding the material. Molded beads along the outer perimeter are used to improve the sealing characteristics.

Center plates may be attached by either bolting or cementing. Cementing eliminates the need for a center hole, thus simplifying sealing (Reference 1-130). Bolted center plates are used when the diaphragm will require replacement. With metal diaphragms, bosses which are formed with the diaphragm (integral bosses) or joined to it are used. The integral boss offers predictable performance and installation, but manufacturing techniques and costs may outweigh the advantages. Generous corner radii at the point of clamping should be used to avoid cutting the diaphragm during stroke. Radii from 0.03 to 0.12 inch are good practice. For nonmetal diaphragms, corner radii should be at least twice diaphragm thickness.

6.7.3 Design Data

6.7.3.1 FLAT DIAPHRAGMS. Flat diaphragms are flat circular discs, grouped as either "true flat" or "dish-type" diaphragms. Typical designs are shown in Figure 6.7.3.1a. Flat diaphragms may be either metallic or nonmetallic. The true-flat diaphragm is usually metallic, whereas the dish-type diaphragm is usually nonmetallic. Metallic diaphragms are able to operate at higher pressures and can withstand higher overpressure but have lower fatigue life and shorter stroke than nonmetallic diaphragms.

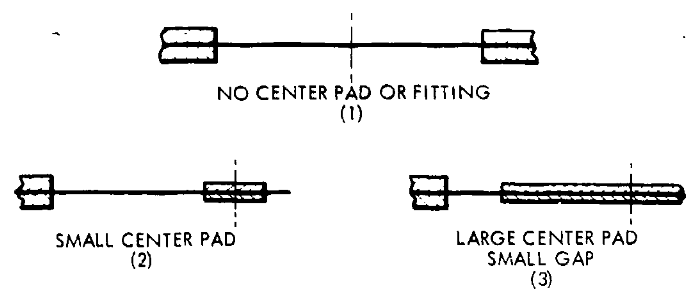


Figure 6.7.3.1a. Flat Diaphragm Designs

(Reprinted from "Diaphragm Characteristics Design and Terminology," F. B. Newell, Copyright 1958, American Society of Mechanical Engineers)

Flat diaphragms have the following advantages over other types:

a) Manufacturing time and costs are less

b) Interchangeability is improved

c) Hysteresis is less

d) Can be operated with reversing pressure.

To obtain linearity, the stroke of a flat metal diaphragm is limited to one-half the material thickness (Reference 156-1). The stroke available for dish-type diaphragms is approximately twice the height of the dish and is limited by the resiliency of the material. Design precautions for dish-type, nonmetallic diaphragms are illustrated in Figure 6.7.3.1b.

The diameter and pressure ranges for typical flat metal diaphragms are presented in Table 6.7.3.1a.

Deflection and Stress in Flat Diaphragms. When the deflec-

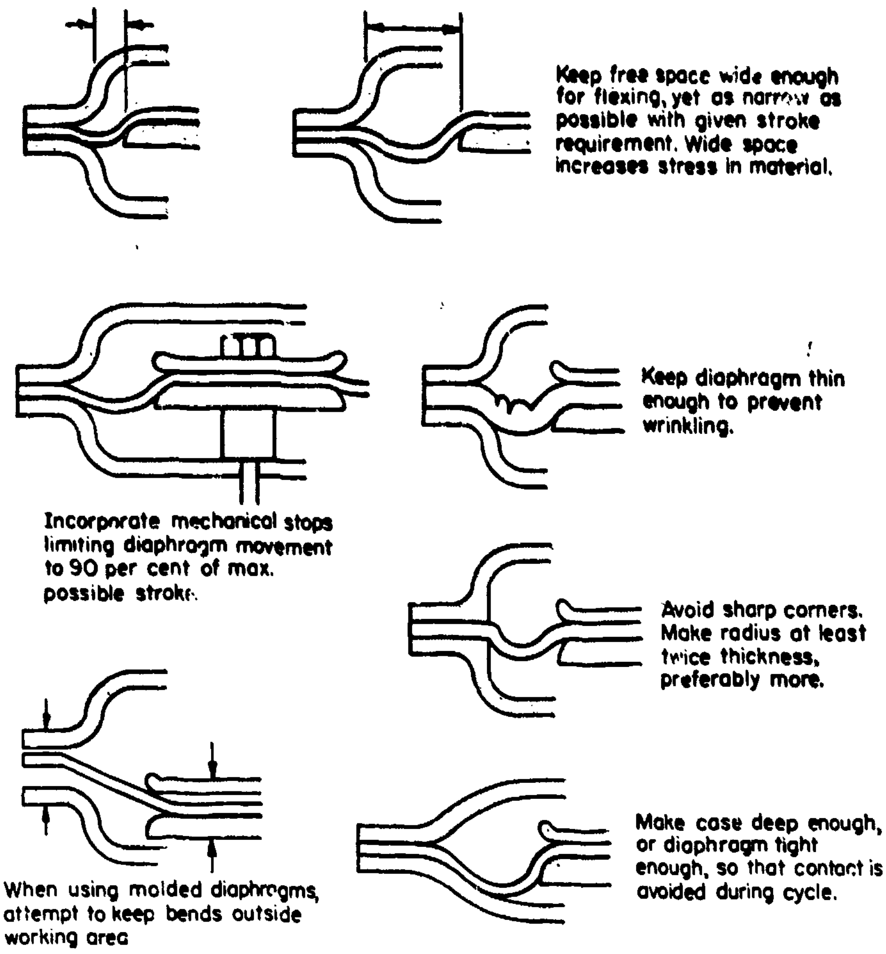


Figure 6.7.3.1b. Flat Diaphragm Design Precautions
(Reprinted from "Machine Design," The Seals Book, January 1961, J. F. Taplin and J. J. Phillips, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

tion becomes larger than about one-half of the thickness of a diaphragm or a thin plate, the middle surface becomes appreciably strained. The stress in the middle surface, called the diaphragm stress, enables the thin plate to carry part of the load as a diaphragm in direct tension. Formulae for the stress and deflection for circular plates are given in Sub-Section 14.10; they should be used whenever the maximum deflection exceeds half the thickness, if accurate results are desired.

Vibration of Flat Diaphragms. To determine analytically the exact natural frequencies of a flat diaphragm of arbitrary shape is quite difficult and laborious. However, by approximate methods such as that of Rayleigh's, the solution can be obtained with an accuracy of a few percent. The natural frequencies of commonly used thin plates (diaphragms) of uniform thickness and small amplitude are given in Table 6.7.3.1b. Formulae for natural frequencies of thin circular diaphragms having a concentrated mass at the center are also given at the bottom of the table. These two formulae would hold for other types of diaphragms provided they are sufficiently thin so that their mass can be neglected in comparison with the one at the center.

Table 6.7.3.1a. Approximate Limits of Pressure Range and Diameter for Flat Metal Diaphragms (Thickness not given, varies as required)

(Reference 156-1)

Diameter (Approx Inch)	Pressure Range (PSI)	
	Minimum	Maximum
0.25	0-10	0-10,000
0.50	0-5	0-10,000
1.00	0-1	0-6,000
2.00	0-0.3	0-6,000
4.00	0-0.1	—

Example: Determine the lowest natural frequency for a circular diaphragm of 2-inch diameter and 0.008-inch thickness with fixed edge. The following values are used:

$$E = 30 \times 10^6 \text{ psi}$$

$$\rho = \frac{0.28 \text{ lb}_f/\text{in}^3}{384 \text{ in/in}^2} = \frac{1}{1370}\left(\frac{\text{lb}_f\text{-sec}^2}{\text{in}^4}\right)$$

$$\mu = 0.28$$

Then

$$\omega_1 = B\sqrt{\frac{E\,t^2}{\rho a^4 (1-\mu^2)}}$$

$$= 11.84\sqrt{30\,(10^6)\left(\frac{64}{10^6}\right)(1370)\left(\frac{1}{16}\right)\left[\frac{1}{1\text{-}0.0785}\right]}$$

$$= 4990 \text{ rad/sec}$$

The exact solution is ω_1 = 4960 rad/sec, percentage of error = 0.6 percent.

6.7.3.2 CONVOLUTED DIAPHRAGMS. A convoluted diaphragm is a corrugated disc which may have one or more circumferential convolutions extending radially from the center. The single convoluted diaphragm can take many forms (Figure 6.7.3.2a). Generally, diaphragm corrugations with more curvature (up to 180 degrees) can withstand higher pressures for the same material thickness. The small angle convolutions are usually about 60 degrees. Stresses in a metal diaphragm with a 60-degree convolution are twice the stress with a 180-degree convolution; and, with a 90-degree convolution are 1.42 times the stress with a 180-convolution. A full semi-circular convolution with a 180-degree curve is the optimum shape (Reference 156-1).

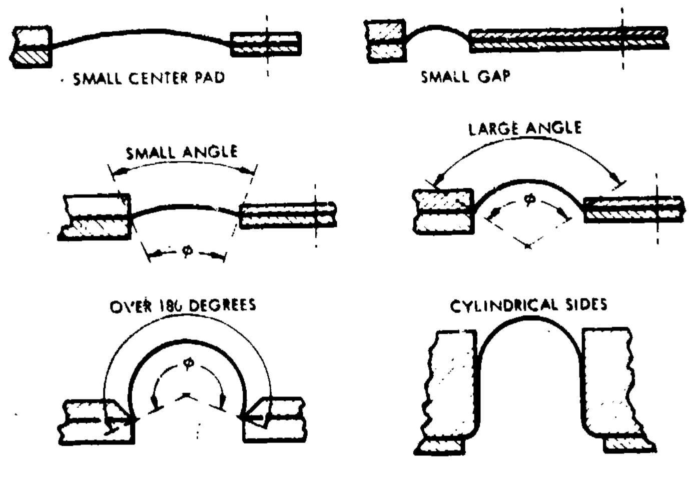


Figure 6.7.3.2a. Convoluted Diaphragm Designs
(Reprinted from "Diaphragm Characteristics Design and Terminology," F. B. Newell, Copyright 1958, American Society of Mechanical Engineers)

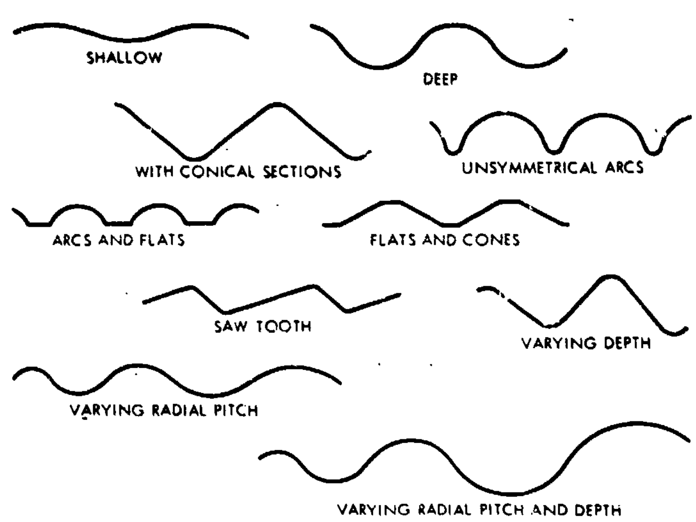


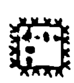
Figure 6.7.3.2b. Multiple-Convoluted Diaphragm Designs
(Reprinted from "Diaphragm Characteristics Design and Terminology," F. B. Newell, Copyright 1958, American Society of Mechanical Engineers)

Table 6.7.3.1b. Natural Frequencies of Thin Flat Plates of Uniform Thickness

(Reprinted with permission from Shock and Vibration Handbook, Volume I, C. M. Harris and C. E. Crede, McGraw-Hill, 1961.)

$$\omega_n = B \sqrt{\frac{Et^2}{\rho d^4 (1-\mu^2)}} \text{ RAD/SEC}$$

E = Young's Modulus, lb_f/in^2
t = Thickness of Plate, in.
ρ = Mass Density, $lb_f\text{-}sec^2/in^4$
d = Diameter of Circular Plate or Side of Square Plate, in.
μ = Poisson's Ratio

Shape of Plate	Diagram	Edge Conditions	Value of B for Mode: 1	2	3	4	5	6	7	8
Circular		Clamped at Edge	11.84	24.61	40.41	46.14	103.12			
Circular		Free	6.09	10.53	14.19	23.80	40.88	44.68	61.38	69.44
Circular		Clamped at Center	4.35	24.26	70.39	138.85				
Circular		Simply Supported at Edge	5.90							
Square		One Edge Clamped — Three Edges Free	1.01	2.47	6.20	7.94	9.01			
Square		All Edges Clamped	10.40	21.21	31.29	38.04	38.22	47.73		
Square		Two Edges Clamped — Two Edges Free	2.01	6.96	7.74	13.89	18.25			
Square		All Edges Free	4.07	5.94	6.91	10.39	17.80	18.85		
Square		One Edge Clamped — Three Edges Simply Supported	6.83	14.94	16.95	24.89	28.99	32.71		
Square		Two Edges Clamped — Two Edges Simply Supported	8.37	15.82	20.03	27.34	29.54	37.31		
Square		All Edges Simply Supported	5.70	14.26	22.82	28.52	37.08	48.49		

Massless Circular Plate with Concentrated Center Mass m in $lb_f\text{-}sec^2/in.$

Clamped Edges  $\omega_n = 4.09 \sqrt{\frac{Eh^3}{md^2 (1-\mu^2)}}$

Simply Supported Edges  $\omega_n = 4.09 \sqrt{\frac{Eh^3}{md^2 (1-\mu)(3+\mu)}}$

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A number of multiple convolution forms are illustrated in Figure 6.7.3.2b. Convoluted diaphragms have the following advantages over flat diaphragms:

a) Essentially constant effective area will be maintained over a longer stroke.

b) Total stroke is longer without permanent deformation.

c) The unit load in the diaphragm is lower for the same pressure due to more effective material area; therefore, it has higher sensitivity than that of a diaphragm of the same size over a large range of pressure.

d) Better endurance life.

e) For a given effective diameter, the overall diameter can be smaller.

f) It has a more stable zero position under no load.

g) A variety of pressure-deflection characteristics may be obtained for a given size diaphragm by using different depths or shapes of corrugations.

A general rule of thumb used in designing metal convoluted diaphragms for linear response within 1 percent or less is to keep the stroke under 2 percent of the diameter. If linearity of more than 1 percent is allowable, deflections up to 3 percent and 4 percent of the diaphragm diameter are permitted (Reference 156-1).

When subjected to pressure, the convolutions near the center of the diaphragm change their shape very little. The convolution which contributes most of the deflection and has the highest stress is near the outermost convolution, since the outside corrugations are stiffened by the connecting flange and do not deflect.

Convoluted diaphragms are used widely in extreme temperature applications. The relative insensitivity to thermal expansion or contraction is due to the ability of the wave form to change shape as required.

General guide points on convoluted diaphragm design are:

a) Travel available on convoluted diaphragms is slightly greater than twice convolution height.

b) Stops should be provided to prevent rupture due to overtravel.

c) Approximately 10 percent excess movement should be allowed in the original design. This excess plus the use of mechanical stops eliminates the danger of diaphragm rupture.

d) Convolutions should be designed so that cross-section and height are equal. Molding problems become greater as height increases.

e) Effective area can be made less variable by providing larger convolutions.

Several methods of assembling or attaching metal convoluted diaphragms to a rigid structure are shown in Figure 6.7.3.2c.

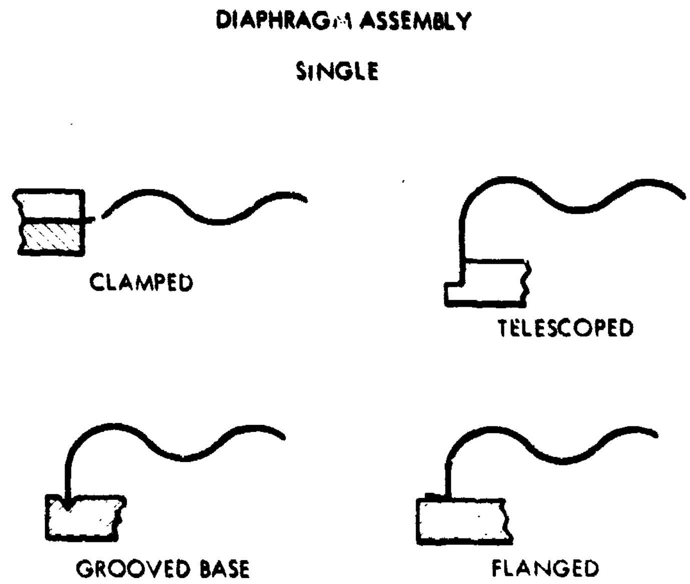


Figure 6.7.3.2c. Methods for Attaching Metal Diaphragms
(Reprinted from "Diaphragm Characteristics Design and Terminology," F. B. Newell, Copyright 1958, American Society of Mechanical Engineers)

Stresses and Deflection in Corrugated Diaphragms. Deflection and stresses in corrugated diaphragms cannot be accurately predicted with linear theory since, for large deflections of several times the thickness, they exhibit highly nonlinear behavior. However, accurate nonlinear calculations are now possible with the use of the NONLIN computer program (Reference 44-37).

An approximate solution of deflection and stress can be obtained under the following assumptions and conditions (Reference 26-216):

a) Load is uniformly distributed.

b) It is assumed that the maximum tangential stress is decisive so that it prevails over radial stress.

c) In deriving the formulae and making a design chart, a trapezoidal approximation of the corrugation with sharp corners is assumed (Figures 6.7.3.2d and e).

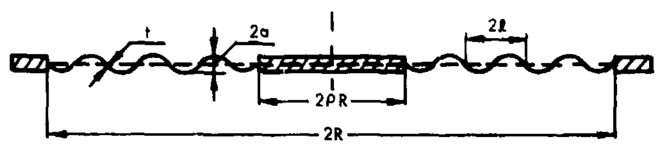


Figure 6.7.3.2d. Corrugated Diaphragm Geometry

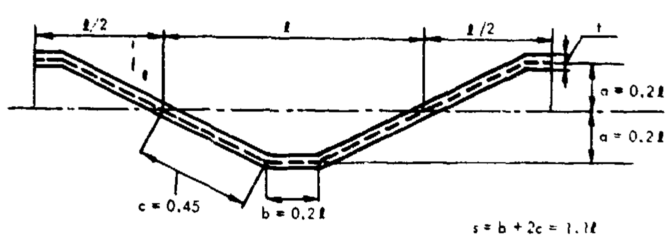
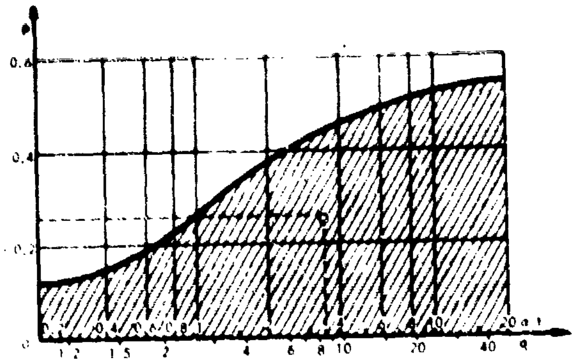


Figure 6.7.3.2e. Transverse Section of a Corrugation with Trapezoidal Profile

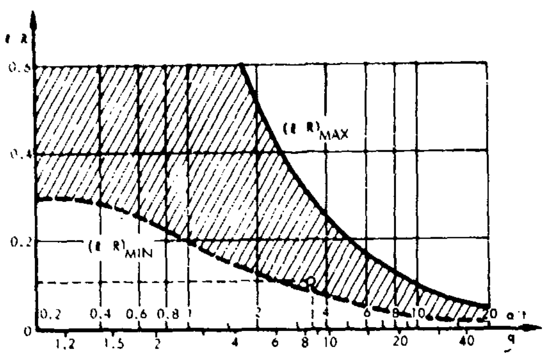
d) There is no rigid central plate. Its effect on the deflection is minor. If the factor ρ falls in the region indicated in Figure 6.7.3.2f, where permissible ρ is plotted against (Q) and (a/t), the design procedure described below should hold to a first order of accuracy.



NOTE:
THIS CURVE APPLIES TO CORRUGATED DIAPHRAGMS IN WHICH THE EFFECT OF A CONTROL PLATE ON RIGIDITY CAN BE DISREGARDED.

Figure 6.7.3.2f. Maximum Acceptable Value of the Ratio for Corrugated Diaphragm
(Adapted with permission from Reference 26-216, "Design of Corrugated Diaphragms" by J. A. Harinx, Transactions of the American Society of Mechanical Engineers, V. 79, No. 1, Jan. 1967, New York.

e) It is assumed that the sheet is sufficiently thick. If the (a/t) value falls in the region shown in Figure 6.7.3.2g for a given value of l/R, this condition is satisfied.



NOTE:
THESE LIMITS APPLY TO CORRUGATED DIAPHRAGMS WHICH ARE TO BE REGARDED AS ADEQUATELY THICK, AND FOR WHICH MAXIMUM TANGENTIAL STRESS IS DECISIVE.

Figure 6.7.3.2g. Acceptable Values of l/R for Corrugated Diaphragms
(Adapted with permission from Reference 26-216, "Design of Corrugated Diaphragms" by J. A. Harinx, Transactions of the American Society of Mechanical Engineers, V. 79, No. 1, Jan. 1967, New York.

The equations required to design the corrugated diaphragm are:

$$p' = \frac{p_{max}}{(1+\Delta)\sqrt{\Delta}}$$

$$f' = \frac{f_{max}}{\sqrt{\Delta}}$$

$$y' = \frac{y_{max}}{\sqrt{\Delta}}$$

$$q = \sqrt{1 + \frac{6a^2}{t^2}}$$

where

t = thickness of the diaphragm sheet material

a = amplitude of the corrugation

R = radius of the diaphragm

l = half of the wavelength of the corrugation (Figure 6.7.3.2e).

Δ = deviation in respect to the linear relation between the pressure and deflection; it is assumed always between 0 and 0.1.

f_{max} = maximum permissible stress

y_{max} = maximum deflection

p_{max} = maximum pressure the diaphragm is subjected to

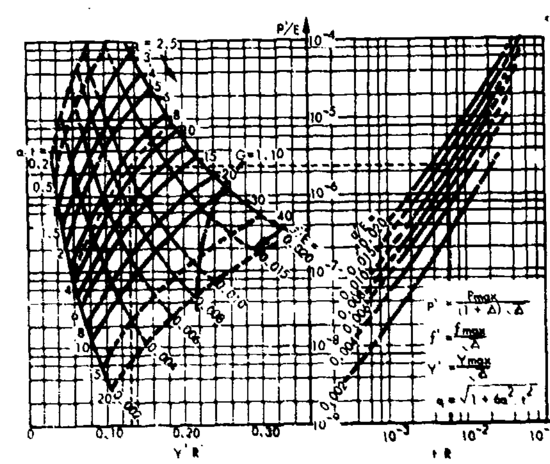
f', y', p' = "reduced" stress, deflection, and pressure to determine the design parameters from the chart.

The above equation along with the chart of Figure 6.7.3.2h can be applied to design a corrugated diaphragm.

Example:

Given: p_{max} = 14.2 psi

y_{max} = 0.158 in.



NOTE:
THIS CHART WAS COMPILED FOR CORRUGATIONS WITH A STANDARD TRAPEZOIDAL PROFILE (FIGURE 6.7.3.2e), BUT MAY BE USED FOR OTHER PROFILES.

Figure 6.7.3.2h. Chart for Design of Corrugated Diaphragms with a Uniformly Distributed Load
(Adapted with permission from Reference 26-216, "Design of Corrugated Diaphragms" by J. A. Harinx, Transactions of the American Society of Mechanical Engineers, V. 79, No. 1, Jan. 1967, New York.

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SUPERSEDES: MAY 1964

$\Delta = 0.10$ in respect to the linear relation between pressure and deflection

$E = 17.4 \times 10^6$ psi

$E' = (1 + \Delta)E = 19.2 \times 10^6$ psi

$f_{max} = 5700$ psi

To determine the dimensions of the diaphragm, reduced quantities are computed.

$$p' = \frac{p_{max}}{(1+\Delta)\sqrt{\Delta}} = \frac{14.2}{(1.1)(0.314)} = 41.2 \text{ psi}$$

$$y' = \frac{y_{max}}{\sqrt{\Delta}} = \frac{0.158}{0.314} = 0.502 \text{ in.}$$

$$f' = \frac{f_{max}}{\sqrt{\Delta}} = \frac{5700}{0.314} = 18{,}200 \text{ psi}$$

$$\therefore \frac{p'}{E} = \frac{41.2}{17.4(10^6)} = 2.35 \times 10^{-6}$$

$$\frac{f'}{E} = 0.0104$$

From Figure 6.7.3.2h, for p'/E, and f'/E the following ratios are read.

$$\frac{t}{R} = 0.0062$$

$$\frac{y'}{R} = 0.132$$

$$\frac{a}{t} = 3.5$$

Since $y' = 0.502$ inch

$$R = \frac{0.502}{0.132} = 3.83 \text{ in.}$$

Also

$$t = 0.0062R = 0.0236 \text{ in.}$$

$$a = 3.5t = 0.083 \text{ in.}$$

$$\ell = 5a = 0.415 \text{ in.}$$

The ratio $\ell/R = 0.108$ at $a/t = 3.5$ lies within the region bounded by two curves in Figure 6.7.3.2d so assumptions (b) and (e) are satisfied.

6.7.3.3 ROLLING DIAPHRAGMS. Rolling diaphragms have full 180 degree convolutions which are maintained during the length of stroke. Materials used are either fabric elastomers or metals. Rolling diaphragms provide long strokes coupled with the dividing membrane features of a standard diaphragm.

Fabric-Elastomer Rolling Diaphragms. The two types of nonmetallic rolling diaphragms are the "top hat" (Figure 6.7.3.3a), which has its convolution hand-formed during assembly, and the convoluted, which has its convolution molded in. Both types have four separate rim configurations or classes as described in Table 6.7.3.3a for top-hat rolling diaphragms. The same shape and classification applies to the molded-convolution diaphragm with the exception that the letter "C" follows the class designation to signify convolution type (1C, 3C, 4C, and 5C). Standard sizes are presented in Table 6.7.3.3b. The advantages of fabric-elastomer rolling diaphragms are:

a) Very low hysteresis — loss

b) Low breakout and moving friction or resistance to motion

c) No spring rate

d) Long stroke

e) Good sensitivity to small pressure variations.

Due to the complex nature of the rolling action, variable supporting wall areas, and possible fabric distortions due to twist, the designer must analyze for several operating conditions. For example, if imposed pressures are reversed, diaphragm wrinkling and sidewall scrubbing as depicted in Figure 6.7.3.3b can cause premature failure. Recommended flange, cylinder, and piston diameters are presented in Table 6.7.3.3c. Diaphragms with molded-in convolutions have only one maximum height for each size range as indicated in Table 6.7.3.3b. The maximum height and stroke are listed in Table 6.7.3.3d.

In the top-hat design, the diaphragm tries to revert to its original shape and may cause sidewall scribbing and diaphragm wrinkling. To prevent scuffing and wrinkling, curved retainer plates (Table 6.7.3.3e) are normally used.

The life of the rolling diaphragm is limited by the flexing elongation that exists when the diaphragm rolls from a smaller diameter piston to a larger cylinder wall. Diaphragms with high elongation are subject to lower life than those diaphragms with low elongation. The percent elongation for standard size fabric-elastomer rolling diaphragms is presented in Table 6.7.3.3f.

Rolling Metal Diaphragms. Recently considerable attention has been devoted to the development of metal or metallic rolling diaphragms, primarily for use in rocket propellant expulsion applications (Figure 6.7.3.3c). Unfortunately, at this time much of the available literature on the subject is either classified or proprietary. Roll-and-peel diaphragm concepts have the advantages of light weight, mechanical simplicity, and high reliability for propellant expulsion applications. State-of-the-art rolling metal diaphragms (also called *rolling bladders* or *rollonets*) consist of prefabricated diaphragms which are inserted into the tank and bonded in place (Reference 308-4). Advanced techniques presently under development include chemical vapor deposition (Reference 761-1) or electrodeposition (Reference 131-34) of the diaphragm material onto the tank wall. Basic in the roll and peel concept for either prefabricated or deposited diaphragms is a requirement for a bond between the diaphragm and tank wall for roll stability.

Rolling diaphragms made of welded or spun sheet metal require relatively close tolerance control because of the requirement for insertion into the tank and provision for a

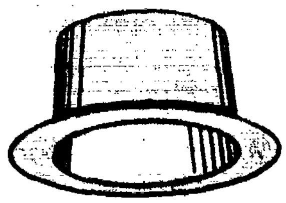


Figure 6.7.3.3a. "Top Hat" Diaphragm Design
(Reprinted from "Machine Design," J. F. Taplin and J. J. Phillips, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

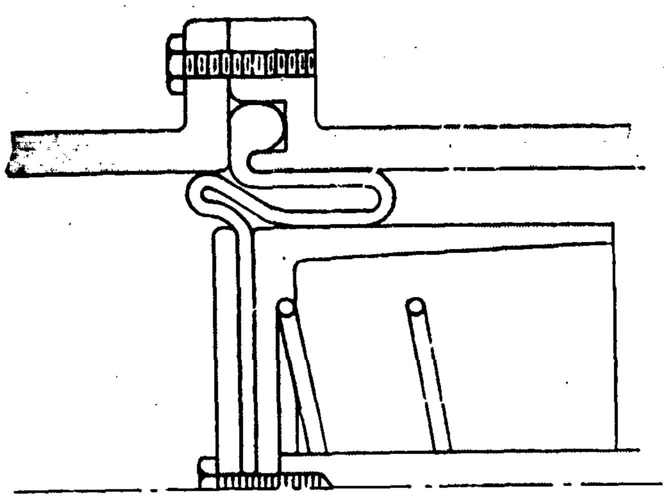


Figure 6.7.3.3b. Problems Resulting from Installation Error
(Reprinted from "Machine Design," J. F. Taplin and J. J. Phillips, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

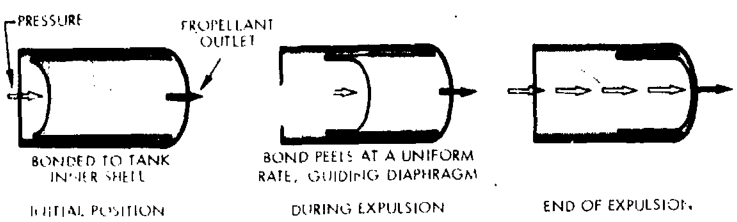
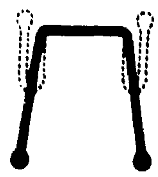
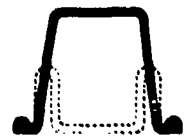


Figure 6.7.3.3c. Rocket Propellant Expulsion Rolling Diaphragm

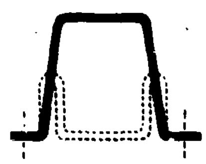
Table 6.7.3.3a. Rolling Diaphragm Configurations
(From "Machine Design," Seals Book, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)



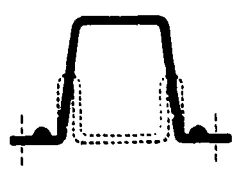
Class 1—Plain Bead
Used where minimum housing OD is required. Square groove should be used, with volume of groove equal to volume of bead. Conventional bolted-flange construction can be completely eliminated, and the diaphragm held by flange retainer plate. Or, it can be used with threaded bevel ring.



Class 3—Beaded Rim
Used when a soft gasket action is required to provide tight pressure sealing against roughly machined or warped flange surfaces. Has a bead around entire periphery of a rim having narrow radial width. Volume of the retaining groove should be equal to the volume bead.



Class 4—Plain Rim
Most commonly used. Applied in applications which have flat mating surfaces between the cylinder and the bonnet. Rim of the diaphragm also serves as a gasket to prevent leakage. The metal flange faces should be flat or serrated (concentric V-grooves spaced 1/32 in. apart with a depth of 0.006 in.) to prevent pull out of rim under high pressure. Flange loading pressure should usually not exceed 1000 psi.



Class 5—Extended Beaded Rim
Used when the working pressures are in excess of 150 psi and when flange clamping surfaces are warped or rough.

Table 6.7.3.3b. Rolling Diaphragm Sizes

(From "Machine Design," Seals Book, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

Cylinder Bore Diam. D_c (in.)	Piston Diam. D_p (in.)	Convolution Width C (in.)	Effective Pressure Area A_e (sq. in.)	Typical Heights, H (in.)				
0.37	0.25	1/16	0.08	0.31	...	...	...	...
0.44	0.31	1/16	0.11	0.31	...	...	...	...
0.50	0.38	1/16	0.15	0.38	...	...	...	...
0.56	0.44	1/16	0.20	0.44	...	...	...	...
0.62	0.50	1/16	0.25	...	0.50	...	...	...
0.69	0.56	1/16	0.31	...	0.56	...	...	...
0.75	0.62	1/16	0.37	...	0.62	...	...	...
0.81	0.69	1/16	0.44	...	...	0.69	...	...
0.87	0.75	1/16	0.52	...	...	0.75	...	...
0.94	0.69	1/16	0.60	...	0.62	0.81	...	...
1.00	0.81	3/32	0.64	0.44	0.62	0.81	1.00	...
1.06	0.87	3/32	0.74	0.44	0.62	0.87	1.06	...
1.12	0.94	3/32	0.83	0.44	0.69	0.94	1.12	...
1.19	1.00	3/32	0.94	0.44	0.50	0.69	1.00	1.19
1.25	1.06	3/32	1.05	0.44	0.50	0.75	1.00	1.25
1.31	1.12	3/32	1.17	0.44	0.56	0.81	1.06	1.31
1.37	1.19	3/32	1.29	0.44	0.56	0.87	1.12	1.37
1.44	1.25	3/32	1.42	0.44	0.62	0.94	1.19	1.44
1.50	1.31	3/32	1.55	0.44	0.62	0.94	1.25	1.50
1.56	1.37	3/32	1.69	0.44	0.69	1.00	1.31	1.56
1.62	1.44	3/32	1.84	0.44	0.69	1.00	1.37	1.62
1.68	1.50	3/32	1.99	0.44	0.75	1.06	1.44	1.68
1.75	1.56	3/32	2.15	0.44	0.75	1.06	1.44	1.75
1.87	1.69	3/32	2.49	0.44	0.81	1.12	1.50	1.87
2.00	1.81	3/32	2.85	0.44	0.81	1.25	1.62	2.00
2.12	1.94	3/32	3.24	0.62	0.87	1.31	1.75	2.12
2.25	2.06	3/32	3.65	0.62	0.94	1.37	1.81	2.25
2.37	2.19	3/32	4.08	0.62	1.00	1.44	1.87	2.37
2.50	2.31	3/32	4.54	0.62	1.06	1.50	2.00	2.50
2.62	2.31	5/32	4.79	...	1.06	1.56	2.12	2.62
2.75	2.44	5/32	5.28	...	1.12	1.62	2.25	2.75
2.87	2.56	5/32	5.80	...	1.12	1.69	2.31	2.87
3.00	2.69	5/32	6.35	...	1.19	1.75	2.37	3.00
3.12	2.81	5/32	6.92	...	1.25	1.87	2.50	3.12
3.25	2.94	5/32	7.51	...	1.31	1.94	2.62	3.25
3.37	3.06	5/32	8.13	1.00	1.37	2.00	2.75	3.37
3.50	3.19	5/32	8.78	1.00	1.44	2.12	2.81	3.50
3.62	3.31	5/32	9.45	1.00	1.50	2.19	2.87	3.62
3.75	3.44	5/32	10.1	1.00	1.50	2.25	3.00	3.75
3.87	3.56	5/32	10.9	1.00	1.56	2.37	3.12	3.87
4.00	3.69	5/32	11.6	1.00	1.62	2.44	3.25	4.00

controlled gap for the bonding agent. Installation of the bonding agent to obtain uniform circumferential peel strength is a requirement for successful operation of rolling diaphragms and necessitates close control of the chemical and physical characteristics of the bond layer.

Electrodeposited rolling diaphragms are fabricated by electrodepositing certain metals such as copper inside the tank (Reference 131-34). The tank configuration in this technique is an important factor since any plating discontinuity at joints within the tank may cause pinhole leaks. Although close tolerance or a special configuration are not required, adequate access into the tank to allow proper preparation and inspection of the substrate is a requirement.

Chemical vapor deposited diaphragms or rollonets are similar in concept. The thermal decomposition of aluminum alkyl causes the aluminum to be deposited on the heated interior of the tank walls to form the diaphragm (Reference 761-1).

6.7.3.4 CAPSULES. A *capsule* is an assembly of two diaphragms sealed together at their outer edges; two or more capsules assembled together are known as a *capsule element*. Principal methods of assembly are soldering, brazing, and welding.

Two capsule designs, the conventional and the nested, are shown in Figure 6.7.3.4. The joining of two diaphragms as a conventional capsule gives double stroke capability. Nested configurations greatly increase the capability to sustain overpressure. In both designs, the nonlinearity of one diaphragm can be compensated by properly forming the adjoining diaphragm to cancel it.

Capsules are useful in pressure ranges of approximately 0.5 to 400 psi.

Table 6.7.3.3c. Dimensions for Rolling Diaphragm Designs

(From "Machine Design," Seals Book, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

Cylinder Bore Diam D_C (in.)	Piston Diam D_P (in.)	Convolution Width C (in.)	Maximum Up-Stroke S_A (in.)	Maximum Down-Stroke S_B (in.)
0.37 to 0.99	$D_C - \frac{1}{8}$	$\frac{1}{16}$	$H - \frac{11}{32}$	$H - \frac{11}{32}$
1.00 to 2.50	$D_C - \frac{3}{16}$	$\frac{3}{32}$	$H -$ [illegible]	$H -$ [illegible]
2.51 to 4.00	$D_C - \frac{5}{16}$	$\frac{5}{32}$	$H - \frac{5}{8}$	$H - \frac{5}{8}$
4.01 to 8.00	$D_C - \frac{1}{2}$	$\frac{1}{4}$	$H -$ [illegible]	$H -$ [illegible]

Classes 1, 3, 4, 5: Min Piston Skirt Length L_P (in.)	Classes 1, 3, 4, 5: Min Cylinder Bore Length L_C (in.)	Classes 1C, 3C, 4C, 5C: Min Piston Skirt Length L_P (in.)	Classes 1C, 3C, 4C, 5C: Min Cylinder Bore Length L_C (in.)	Max Standard Outside Flange Diam: Classes 4, 4C D_F (in.)	Max Standard Outside Flange Diam: Classes 5, 5C D_F (in.)
$\frac{H + S_A}{2}$	$\frac{H + S_B}{2}$	$\frac{5}{32}$	$\frac{5}{32}$	$D_C + 0.75$	$D_C + 1.31$
$\frac{H + S_A}{2}$	$\frac{H + S_B}{2}$	$\frac{7}{32}$	$\frac{7}{32}$	$D_C + 1.0$	$D_C + 1.50$
$\frac{H + S_A}{2}$	$\frac{H + S_B}{2}$	$\frac{11}{32}$	$\frac{11}{32}$	$D_C + 1.5$	$D_C + 2.00$
$\frac{H + S_A}{2}$	$\frac{H + S_B}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$D_C + 2.0$	$D_C + 2.75$

Table 6.7.3.3d. Stroke of Molded Convoluted Diaphragms

(From "Machine Design," Seals Book, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

Cylinder Bore Diam D_C (in.)	Convolution Width, C, (in.)	Max. Stroke S_A or S_B (in.)	Standard Height, K (in.)
0.37 to 0.99	$\frac{1}{16}$	0.075	0.100
1.00 to 2.50	$\frac{3}{32}$	0.085	0.150
2.51 to 4.00	$\frac{5}{32}$	0.150	0.250
4.01 to 8.00	$\frac{1}{4}$	0.250	0.375

Figure 6.7.3.4. Capsule Design

(From WADD-TR-59-743, 1 August 1960, AD-251111, prepared by Gianini Controls Corp., Duarte, California)

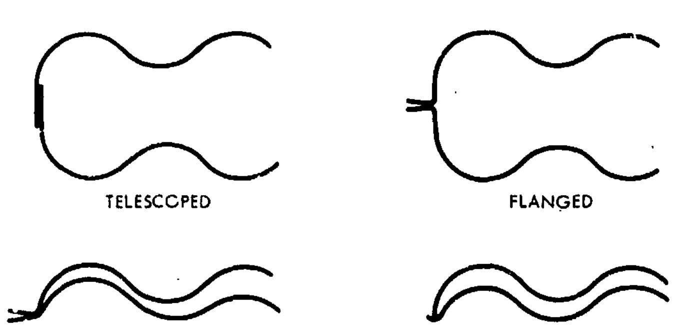


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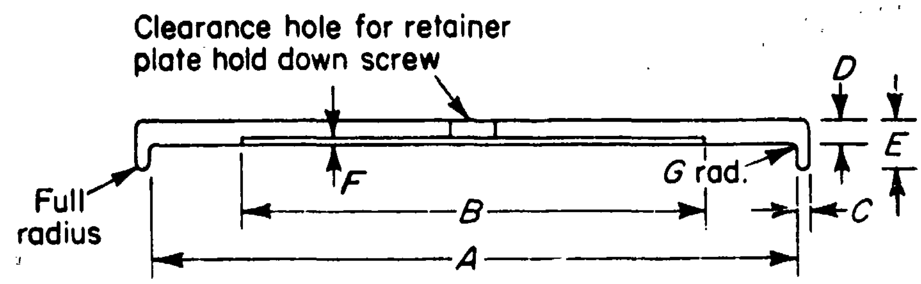
Table 6.7.3.3e. Curved Retainer Plate

(From "Machine Design," Seals Book, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)



Cylinder Bore Diam, D_Q	Dimensions (in.)						
	A	B	C	D	E	F	G
0.37 to 0.99	D_P+2W	Not required	0.025	1/16	1/8	Not required	0.025
1.00 to 2.50	D_P+2W	0.7 D_P	0.032	3/32	1/16	0.010	0.030
2.51 to 4.00	D_P+2W	0.7 D_P	0.032	7/64	7/32	0.015	0.040
4.01 to 8.00	D_P+2W	0.7 D_P	0.060	1/8	1/4	0.015	0.060

All dimensions in inches.

Table 6.7.3.3f. Circumferential Elongation, Rolling Diaphragms

(From "Machine Design," Seals Book, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

Cylinder Bore Diam, D_Q (in.)	Convolution Width, C (in.)	Piston Diam D_P (in.)	Elongation (per cent)
0.37	1/16	0.25	48.1
0.50	1/16	0.38	31.6
0.75	1/16	0.62	19.4
0.94	1/16	0.81	14.8
1.00	3/32	0.81	23.5
1.50	3/32	1.31	14.5
2.00	3/32	1.81	10.5
2.50	3/32	2.31	8.2
2.62	5/32	2.31	13.4
3.00	5/32	2.69	11.5
3.50	5/32	3.19	9.7
4.00	5/32	3.69	8.4
4.12	1/4	3.62	13.8
4.50	1/4	4.00	12.5
5.00	1/4	4.50	11.1
6.00	1/4	5.50	9.1

These elongation values represent a maximum and occur in the fabric of the convolution at a point near the piston head at the full downstroke position.

REFERENCES

*References added March 1967
**References added November 1968

1-130	68-88**	308-4**
26-216*	82-8	380-3*
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44-31*	152-12**	438-1
44-37**	156-1	761-1**
68-37	261-2	

6.8 BEARINGS AND LUBRICATION

6.8.1 Introduction

A bearing is any object, surface, or point that positions or guides one moving part with respect to other parts of an assembly. The application of bearings in fluid components is as broad as the use of the moving parts. For example, the valve's control element is guided and aligned by a bearing surface. Butterfly and ball valves require bearings for rotation, oscillation, or part-turn motion. Bearings for axial linear motion are required for such devices as slides, push-rods or pneumatic and hydraulic piston actuators.

Bearings may be classified as either rolling contact or sliding surface. Rolling contact bearings include the many varied configurations of ball and roller bearings; sliding surface bearings include journal bearings, bushings, and thrust pivots.

The bearings sub-section will discuss only bearings related to valve design. It does not treat bearing problems involving high loads with high rotational speeds. Bearings in valves are usually subjected to more moderate loads with small relative motion, 90° of rotation or fractions of an inch of linear motion. Regardless of the supposed simplicity of valve bearing design, many valves malfunction as a result of galling of the bearing surfaces, necessitating redesign, a change of bearing materials, and an improvement in lubrication.

Rolling contact bearings are designed and fabricated by bearing manufacturers, and are usually specified rather than designed by the valve designers. Sliding contact bearings are usually designed for each specific application. Thus in this section, rolling contact bearings will be considered from aspect of selection, sliding surface bearings from the aspect of design.

6.8.2 Design Factors

6.8.2.1 MATERIALS. Materials for both types of bearings are described.

Rolling Contact Bearing Materials. In rolling contact bearings, the small contact areas between rolling elements and races create high unit stresses which require hardened materials for support. The most common material is SAE 52100 steel, heat treated and stabilized at a relatively low temperature. This material is suitable for temperatures up to 250°F. Higher stabilization temperatures have been used to push the usable temperature up to 350°F. M1 and M10 tool steels have been used at 750°F, and 440 C stainless steel has been used at 1000°F under restricted conditions. For temperatures above 1000°F, stellite is used. In most rolling contact bearings sliding surface contact is created between the rolling element and the retainer. The choice of materials for retainers includes those materials used for sliding surface bearings.

Although the balls and races are primarily made from high alloy steels, non-metals such as sapphire, ceramics, and ceramets have been utilized in special ball bearing applications. Experience to date indicates that their use is limited to slow speeds and light loads. Friction and wear characteristics are presented for many ceramics and ceramets in Reference 93-6.

Sliding Surface Bearing Materials. Materials for sliding surface bearings range from very hard metals and ceramics to soft plastics and elastomers in many combinations. Materials are commonly selected for low friction characteristics, corrosion resistance, and compatibility under defined environmental conditions. Theoretically, any combination of materials could be used as bearing materials, providing an effective lubrication film separates the bearing surfaces during operation. However, in practice, and particularly in valve operation, maintaining an effective lubrication film is not possible due to the varying loads, and the short time interval between startup and shut-down. Solid film lubrication and self-lubricating materials are particularly important in valve design. Table 6.8.2.1a lists and describes the essential characteristics to be considered in choosing bearing materials. Lubricants are discussed in the next section.

Table 6.8.2.1a. Material Selection Factors — Sliding Surface Bearings

(Adapted from the Bearings Book, "Machine Design," vol. 35, no. 14, Copyright 1963 by Penton Publishing Company, Cleveland, Ohio)

Hardness. Bearing materials can be hard or soft. Hard bearing materials provide high load capacity and good wearability. Soft bearing materials provide better antiscoring, conformability, and embeddability properties. Care must be taken to select bearing and shaft materials such that the shaft is harder than the bearing.

Strength. High strength bearing materials simplify design in that the bearing can support itself. Low strength and brittle materials such as babbitts and graphite require support of a stronger material. Compressive and shear strengths are the most important properties for bearing design.

Conformability. Capability to conform to misalignments or deflecting forces is a desirable characteristic. Soft materials with a low modulus of elasticity exhibit favorable conformability.

Load Capacity. Load capacity is a factor indicating the practical load carrying capability of a bearing. It is an empirical summation of strength and other performance properties related to unit load carrying capability and endurance.

Fatigue Resistance. Fatigue resistance is a measure of the capacity of a bearing material to sustain repeated load changes without developing cracks and failing at relatively low stress levels. This factor is important in high velocity reciprocating mechanisms.

Embeddability. The ability to absorb hard foreign particles into the bearing surface, avoiding scoring and wear, is known as embeddability. Softer metals with good anti-seizure properties generally exhibit this characteristic.

Temperature Effects. Thermal expansion and conductivity are important in bearing design. Materials for radial bearings and shafts should have similar coefficients of expansion to minimize clearance changes with temperature variations. Deflection of bearing mountings due to thermal expansion must be considered to avoid misalignment and eccentric loading. High thermal conductivity is desirable to minimize thermal gradients and provide effective heat dissipation. Material transformation and loss of strength must be considered in determining acceptable temperature range.

Corrosion Resistance. Bearings exposed to reactive materials must be selected for inertness in the medium. Galvanic activity must be considered when dissimilar materials are used in contact with an electrolyte. Potential concentration cells should be avoided when possible, since oxidation products of some lubricants can be reactive with certain bearing materials. The use of plating may be desirable for protection.

Cost. The relative importance of material costs is largely determined by the size of equipment. For small bearings, ease of fabrication is more important than material costs. In some installations, a thin layer of an expensive material applied over an inexpensive base can reduce costs.

Compatibility and Wear Resistance. Compatibility refers to the anti-seizure properties of bearing and shaft metal asperities. Metals that are miscible (tend to alloy or be mutually soluble) generally offer poor compatibility and wear resistance. Most immiscible metals or those that form metal compounds offer better anti-seizure properties than miscible metals. Generally, metals with poor anti-seizure properties can be improved by alloying, but metals with good anti-seizure properties cannot be improved. Table 6.8.2.1b examines the affinity of metals as a rating of metal pairs.

Bearing materials can be classified into the following:

- babbitts
- bronzes and copper alloys
- aluminum alloys
- porous metals
- other metals
- plastics
- other non-metals

Many of the physical properties of these materials can be found in Reference 1-132. The characteristics of each class are summarized below.

Babbit Metals. Babbit metals are composed primarily of lead and tin. Compared to other materials they exhibit lower hardness (21-25 Brinell), lower load carrying capacity (800-1500 psi), and very poor fatigue strength. However, they do possess good embeddability and compatibility characteristics under boundary lubrication conditions. Maximum operating temperature is 300°F. In small bushings, babbits are applied as a thin coating over a steel strip. For larger bearings, the babbit is cast on a rigid steel backing. Cast babbit bearings are several times costlier than the coated bushing.

Bronzes and Copper Alloys. Most of these alloys are copper-lead, leaded bronze, tin bronze, and aluminum bronze. The copper-lead and leaded bronze have low strength properties, but good compatibility and anti-scoring properties. As tin, aluminum, and iron are added in greater quantities, the load carrying capacity, high temperature operation, and wear resistance are improved but poor anti-scoring properties result. Hardness and maximum load carrying capacities range from 25-195 Brinell and 2000 to 4500 + psi, respectively. A list of property data for many bronze bearing alloys is given in Reference 1-132.

Table 6.8.2.1b. Seizure Resistance of Pairs of Metals

(From "Materials in Design Engineering," C. L. Goodzeit, vol. 47, no. 6, Copyright 1958, Reinhold Publishing Company, New York, New York)

METAL PAIR	1045 STEEL		ALUMINUM		COPPER		SILVER	
	PAIR TYPE	RATING	PAIR TYPE	RATING	PAIR TYPE	RATING	PAIR TYPE	RATING
Aluminum	M	P	M	P	M	P	M	P
Antimony	C	F-G	C	F-G	M	F	—	—
Barium	I	P	—	—	—	—	—	—
Beryllium	M	P	M	P	M	P	M	P
Bismuth	I	F-G	I	F-G	I	F-G	—	—
Cadmium	I	F-G	I	F-G	—	—	—	—
Calcium	I	P	M	P	M	P	—	—
Carbon	C	F-G	I	P	I	P	I	P
Cerium	M	P	M	P	M	P	—	—
Chromium	M	P	M	P	I	F-G	I	F-G
Cobalt	M	P	M	P	M	P	I	F-G
Columbium	M	P	M	P[b]	M	P[b]	I	F-G
Copper	M	F	M	P	M	P	M	P
Germanium	C	F-G[b]	—	—	—	—	—	—
Gold	M	P	M	P	M	P	M	P
Indium	I	F-G	I	F-G	M	P	M	P
Iridium	M	P	—	—	—	—	—	—
Iron	M	P	M	P	I	P[b]	I	F-G
Lead	I	F-G	I	F-G	I	F-G	I	F-G
Lithium	I	P	—	—	—	—	—	—
Magnesium	I	P	M	P	M	P	M	P
Manganese	M	P	—	—	—	—	—	—
Molybdenum	M	P	M	P[b]	M	P[b]	I	P
Nickel	M	P	M	P	M	P	I	P
Palladium	M	P	—	—	—	—	—	—
Platinum	M	P	M	P	M	P	—	—
Rhodium	M	P	M	P	M	P	—	—
Selenium	C	F-G	C	P	C	F-G	—	—
Silicon	M	P	M	P	M	P	M	P[b]
Silver	I	F-G	M	P	M	P	—	—
Tantalum	M	P	M	P	M	P	—	—
Tellurium	C	F-G	C	F-G	C	F-G	—	—
Thallium	I	F-G	I	F-G	I	F-G	—	—
Thorium	M	P	M	P	M	P	M	P[b]
Tin	C	F-G	M	P	M	P	—	—
Titanium	M	P	M	P	M	P	I	F-G[b]
Tungsten	M	P	M	P	M	P	—	—
Uranium	M	P	M	P	M	P	—	—
Zinc	M	P	M	F	M	P	—	—
Zirconium	M	P	M	P	M	P[b]	M	P[b]

[a]I = immiscible pair, M = miscible pair, C = pair that forms chemical compound.
G = good, F = fair, P = poor. [b]Results are questionable.

Aluminum Alloys. Aluminum alloys exhibit high load carrying capability, superior fatigue strength, high thermal conductivity, good corrosion resistance, and low cost. They lack embeddability, conformability, and exhibit poor compatibility characteristics. Aluminum alloys require hardened shafts and good surface finishes. Typical alloys include a casting alloy (SAE 770) and wrought alloys (SAE 780 and 781).

Porous metals. Porous metal bearings are composed on either bronze or iron. Within each are interlocking pores which comprise 10 to 35 percent of the total volume. In operation, lubricating oil is stored in these voids and passes through the pores to the bearing surface. Load carrying for iron and bronze bearings are typically 8000 and 4500 psi, respectively. Maximum operating temperature is 150°F.

Other Metals. Other metals of particular interest as bearing materials are electo-deposited silver and gold on steel backing. Gold and silver are soft metals which have low shear strengths in all directions. The hard substrates keep the area of real contact between parts small, and the coating of gold or silver keeps the strength of possible cold welded junctions small; together both act to reduce friction and wear. Silver has poor embeddability properties and is sometimes overplated with lead. Uncoated it exhibits excellent fatigue strength and compatibility properties. Methods of coating include vacuum deposition and mechanical burnishing.

At elevated temperature, high temperature metals have been considered as bearing materials, since lubricating characteristics are derived from surface oxidation. Soft oxides with low shear strength facilitate sliding, while hard abrasive oxides have a deleterious effect. Nickel, molybdenum, and Inconel X show decreases in coefficients of friction as great as 70 percent when heated to temperatures greater than 1000°F. Cobalt is a notable exception. A performance summary of these and other metals is presented in References 65-2 and 63-34.

Plastics. Plastics used as bearing materials include phenolics, nylon, and Teflon. Some are used in bulk form, others are filled with such materials as graphite and molybdenum disulfide to increase lubricity and wear resistance. Plastics are often used in valve design because of their corrosion resistance, excellent compatibility characteristics and, in some cases, self-lubricating properties. The operating limits of some plastics is presented in Table 6.8.2.1c.

Phenolics are used as composites with cotton fabric and asbestos, bonded by a phenolic resin. They have excellent strength, shock resistance, resistance to water, acids, and alkalies, but have low thermal conductivity.

Nylon is the most popular plastic bearing material. It has much lower load carrying capacity that the phenolics, but exhibits low friction and requires no lubrication. Nylon will cold flow, which can be minimized by using a well-supported metal sleeve as a backup ring.

Teflon is a self-lubricating material which is used in a variety of forms. Bulk Teflon has a low carrying capacity and a tendency to cold flow. It is usually reinforced by such fillers as glass fibers or graphite. A porous bronze bearing can be impregnated with a mixture of Teflon and lead to provide a thin lubricated surface layer. For slow-speed applications and high load carrying capacity, woven Teflon fabrics impregnated with phenolic resins are used.

Other plastics which are used as bearing materials are Lexan and Delrin. Lexan is similar to nylon but has less tendency to cold flow.

Property data and effects of environments on Teflon and nylon in a variety of forms is presented in Reference 93-6.

Other Non-Metals. Among the non-metals for bearings are carbon-graphite and many types of ceramics and cermets. Graphite possesses self-lubricating properties providing water vapor is present, and is stable at temperatures up to 750°F. Graphite is brittle and can be easily chipped. Ceramics and cermets are excellent for high temperature use, very rigid, highly resistant to wear, hard (as high as 90 Rockwell C), and capable of a fine finish. Such materials include boron nitride ceramics, Pyroceram, aluminum oxide (sapphire), tantelum beryllide, compressed and sintered titanium carbide, or tungsten carbide in a cobalt matrix. Property data for pyroceram and sapphire are given in References 1-66, page 29, and 114-5, page 3, respectively.

6.8.2.2 LUBRICATION. Lubricants provide reduction of friction between rubbing surfaces and dissipation of heat generated by friction within the bearing.

In rolling contact bearings, a thin fluid-film of lubricant separates the rolling elements from the raceway. For high speed and load operation, additional lubricant is often necessary to keep the temperature within safe tolerances. Rolling element bearings can operate with intermittent lubrication for short periods without disastrous effects. For low loads and speeds, dry films or dense chrome plating may be satisfactory for limited operation.

In sliding surface bearings, the use of lubricants is increasingly more necessary, particularly when high loads

Table 6.8.2.1c. Operating Limits for Plastic Bearings

(From Bearings Book, "Machine Design," Copyright 1961 by Penton Publishing Company, Cleveland, Ohio)

PLASTIC	PV FACTOR*	LOAD CAPACITY (PSI)	MAXIMUM TEMP (°F)	MAXIMUM SPEED (FPM)
Phenolic	15000	6000	200	2500
Nylon	3000	1000	200	1000
Teflon	1000	500	500	100
Reinforced Teflon	10000	2500	500	1000
Teflon Fabric	25000	60000	500	50
Delrin	3000	1000	180	10000
Lexan	3000	1000	220	1000

*See Detailed Topic 6.8.3.2 for explanation of use.

are experienced. Lubrication mechanisms for sliding bearings include fully-developed fluid film boundary, mixed film, and dry film. The fully-developed fluid film is used in high speed rotary machinery. In valves the boundary, mixed, and dry film lubrication mechanisms are more common.

Lubrication Mechanisms

Fluid Film Lubrication. The use of fluid lubricants to separate moving elements completely is called fluid film lubrication. Friction in such bearings is essentially that established by the viscosity of the fluid. The coefficient of friction is generally less than 0.001 and values as low as 10^{-6} have been achieved The minimum amount of lubricant is that layer height required to provide clearance of the maximum surface asperities.

The supporting fluid pressure may be hydrostatic or hydrodynamic. Hydrostatic bearings are externally pressurized and make use of a lubricant pumped into the bearing from an external source. Load capacity of such bearings is proportional to the magnitude of the pressure developed within the bearing and the effective pressurized area. If continuous lubricant pressure can be supplied to the bearing during start or stop, metal contact can be eliminated. The life of such a bearing becomes unlimited.

With the self-pressurizing or hydrodynamic-type bearing, lubricant pressure is developed by the motion of the bearing and is dependent upon it. The pressure results from the resistance of the fluid to flow out from converging spaces into which it is forced by relative motion of the parts. The resistance to flow is a function of viscosity of the fluid. In a tapered shoe bearing (Figure 6.8.2.2a) the pressure is developed because of the converging flow passage. In the case of hydrodynamic journal bearings (Figure 6.8.2.2b) the tapered flow path is created when the shaft seeks an eccentric running position.

Gas Lubrication. Air or other gases can be used as fluid film lubricants. The use of gases provides extremely low friction due to low viscosities. In general, such lubrication is limited to bearings with relatively low unit loadings. External pressurization is most commonly used since hydrodynamic pressurization can only be obtained at extremely high rotational speeds and for very low loadings. Gas bearings are more prone to instability than liquid lubricated bearings. Gas bearings are sensitive to high temperatures and exhibit extremely low friction.

Boundary Lubrication. Oils and greases applied in a thin adherent film characterize boundary lubrication. Metal-to-metal contact is not entirely eliminated by this form of lubrication due to the irregularities that exist in surfaces of the finest finish. Intersurface contact takes place when microscopic asperities touch during motion. At these points, the lubricating film is momentarily ruptured and metallic contact occurs. This action (Figure 6.8.2.2c) creates some additional friction and wear due to momentary welding or yielding at the points of contact. Surface finish,

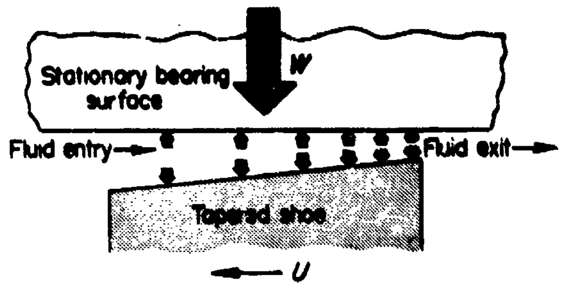


Figure 6.8.2.2a. Tapered Shoe Bearings Illustrating Pressure Increase Due to Converging Flow Passage

(From "Machine Design," Wilcock, D. F., vol. 35, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

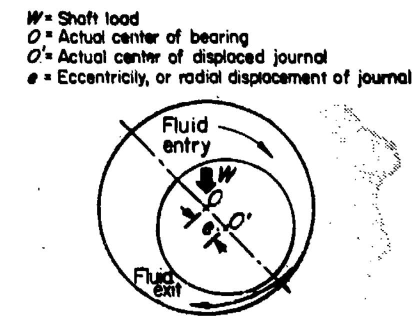


Figure 6.8.2.2b. Hydrodynamic Journal Bearing

(From "Machine Design," Wilcock, D. F., vol. 35, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

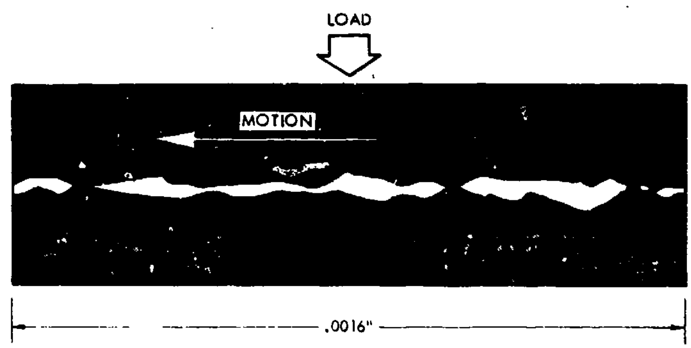


Figure 6.8.2.2c. Cross-section of Ground Steel Surface (8μ inch RMS finish) Magnification 2000 ×

(From "Lubrication Newsletter," no. 5, September 1958, Alpha-Molykote Corp., Stanford, Connecticut)

material combinations, and selection of lubricants are important factors in establishing satisfactory bearing friction and life. Boundary lubrication is accomplished by either of two mechanisms: (1) the lubricants provide strongly adherent molecules, forming extremely tenacous films which act to minimize surface contact, or (2) react chemically to form surface films of low shear strength, enabling contact to occur without damage. The latter mechanism is utilized in E.P. (extreme pressure) additives.

Mixed Film Lubrication. In mixed film lubrication, the load is partially supported by fluid film lubrication, and the balance is supported by boundary lubrication. Hydrodynamic bearings at start or stop, or at a relatively low velocity to load ratio, can be classed as operating with mixed film lubrication. Boundary lubrication requirements must be considered for successful design of hydrodynamic bearings because of potential operation in the mixed film region.

Solid Lubrication. The use of high lubricity solids as lubricants has increased as design requirements for extreme environment applications have evolved. In rocket and space systems, the nature of the environment for working fluid has precluded the application of many liquid or grease lubricants. Solid lubricants either comprise a bulk part of the bearing structure (self-lubricant), or can be applied as a thin bonded surface film or suspension in a lubricating fluid. Self-lubricated bearings exhibit low friction coefficients at the surface. Solid film lubricants have a lamellar crystal structure with a low intercrystalline shear strength and a low coefficient of friction between layers. The lubricant supply must be as deep as the maximum height of surface asperities.

Types of Lubricants. Lubricants are classed as oils, greases, and solids. Solids may take the form of a bulk solid (e.g., Nylon) or of a thin film. More specialized types of lubricants include the use of liquid metals and rocket propellants themselves.

Oils. Originally, the term *oil* referred to slippery hydrocarbon liquids exclusively. However, at present many liquid lubricants of a synthetic nature are also referred to as oils. Several important properties are used to characterize oils. Viscosity, the most important characteristic, influences load carrying capacity, oil film thickness, operating temperature, and the friction that will be developed. In addition to viscosity, the viscosity index is important in many applications. The viscosity index is a measure of the change in viscosity with temperature. It is an important factor when bearings must operate over a broad range of temperature. Other important criteria for liquid lubricants are the pour point, and the flash and fire points. The pour point is the lowest temperature at which a lubricant will flow. The flash point is the temperature at which ignitable vapors are evolved; the fire point is the temperature at which the liquid will burn when ignited.

Mineral oils are the most common type, although many synthetic oils have been developed in recent years possessing better viscosity indices and stability at high temperatures. These synthetics, as general types, include:

polyalkylene glycols
silicones
diesters
organic chlorine compounds
polymer oils
voltol oils

The viscosities and densities of a large group of lubricants contained under these classes are given in Reference 26-130.

Oils possess the following advantages: ease of handling, capability of carrying away dirt and contamination from wear, and capability of transferring heat from the bearing.

Greases. A grease is a semi-solid which is a mixture of an oil and a soap acting as the thickener. Calcium, sodium, and lithium soaps are the most common metallic soaps. Some non-metal and non-washing greases use silicon gel and bemstone as thickener. The most important property is consistency, which is a measure of the hardness or resistance to flow and is related to the ability of a grease to resist being squeezed out of a bearing under pressure. ASTM standards determine consistency by measuring the penetration of a weighted cone in a worked (kneaded) grease sample. Conversion of the depth of penetration into a scale number is established by the National Lubrication Grease Institute.

The relative advantages of greases are retainability, low maintenance, and good sealing capabilities (Reference 1-132).

Lubricant Additives. Substances can be added to oils and greases to improve lubricating ability or alter characteristics for adaptation to special applications. Additives are usually chemical compounds that fortify one or several of the properties of the lubricant.

a) Oxidation Inhibitors. Phosphorous, sulfur, nitrogen, or organic compounds are added to impede formation of gums and acids by slowing the oxidation process.

b) Defoamers. Silicone compounds are added to minimize the formation of foam in high-speed bearing applications.

c) Detergents. Common detergent compounds are added to oils to keep insoluble materials in suspension. This action minimizes sludge buildup on internal surfaces. This additive is most common in lubricants for internal combustion engines.

d) Corrosion Inhibitors. These are usually surface active agents that adhere to metallic surfaces, forming a thin film which will protect against corrosion. Corrosion inhibitors are discussed in more detail under Detailed Topic 13.7.15.2.

e) Pour Point Depressants. These are used with paraffin based oils to reduce the tendency of the wax to gel at low temperatures below the pour point.

f) **E. P. Additives.** Extreme pressure additives are commonly compounds of phosphorous, lead, sulfur, or chlorine. These materials react with metallic bearing surfaces to produce a low shear strength film which prevents metal-to-metal bearing contact. These materials are useful in boundary lubrication for high-speed applications of high unit bearing loading, where bearing temperatures of 300°F and above can be expected. They are commonly used in gear lubricants.

Solid Lubricants. A solid lubricant can be defined as any non-metal exhibiting a coefficient of friction less than 0.4 in contact with a metal (Reference 1-122). Two types of solid lubricants exist. The first type is the solid film lubricant which is a coating, plating, or has a lamellar crystal structure with a low intercrystalline shear strength and a low coefficient of friction between crystals. Typical solid film lubricants are molybdenum disulfide and graphite. Less familiar lubricants are metal oxides, particularly of lead, and the phthalocyanine compound. Lubricity data of the latter two compounds are given in References 63-9 and 47-1. The second type includes the self-lubricating structural materials such as Teflon and nylon, which are discussed in Detailed Topic 6.8.2.1.

A thin, solid film permits the use of two hard materials, but limits the stress level and contact area which results in a low friction force. Such films as molybdenum disulfide shear along low strength crystallographic planes. The degree of friction reduction depends greatly on the lubricant's substrate material. A hard substrate provides a backup for the thin film. If the substrate surface is too smooth, lubricant adhesion will be poor; if it is too rough, friction forces will be high. The wear life of a typical solid film lubricant as a function of substrate roughness is given in Table 6.8.2.2a.

For maximum wear life of thin-film solid lubricants, the substrate surface should be properly prepared. A list of pre-treatments for metals is presented in Table 6.8.2.2b.

Solid film lubricants have been used extensively in missile and aircraft hardware, and many different compounds have been developed using, in particular, MoS_2. Well recognized compounds are molybdenum disulfide and epoxy and molybdenum disulfide, graphite, and sodium silicate. The advantages of solid film lubricants include long term stability, relative insensitivity to high and low temperatures and vacuums, and resistance to extrusion under load. The disadvantages include lack of ability to dissipate frictional heat, slightly higher friction coefficients, and inability to replace lubricant or provide replenishment.

Liquid Metals. At temperatures above 500°F, liquid metals may be effective lubricants in a closed system. Such metals include potassium, rubidium, mercury, and a eutectic alloy of sodium and potassium, designated NaK77.

Liquid metals behave similarly to oils in that their viscosity determines the load carrying capacity and varies inversely with temperature. The viscosity of liquid metals

Table 6.8.2.2a. Wear Life Versus Surface Roughness
(From Bearings Book, "Machine Design," Copyright 1961 by Penton Publishing Company, Cleveland, Ohio)

SURFACE ROUGHNESS (microinches, rms)	ALPHA TESTER WEAR LIFE (hours)
10	58
20	104
30	81
40	51
50	23

Table 6.8.2.2b. Pre-treatment of Substrate Metals
(From Bearings Book, "Machine Design," Copyright 1961 by Penton Publishing Company, Cleveland, Ohio)

METAL SUBSTRATE	PRE-TREATMENT
Steel	Degrease, rougher (vapor blast) phosphate coat
Stainless steel	Degrease, rougher (sand blast)
Aluminum	Degrease, conversion coat (anodize)
Copper Alloys	Degrease, rougher (bright dip)
Magnesium	Degrease, dichromate coat
Titanium	Degrease, rougher (sand blast) fluorophosphate coat
Cadmium or Zinc	Degrease, phosphate coat

is substantially lower than some oils at the same temperature, and only slightly higher than air at temperatures above 1400°F. A viscosity-temperature chart is presented in Reference 65-2, page 113. At elevated temperatures, liquid metals are extremely corrosive, and special protective coatings of bearing metals are required.

Rocket Propellants. The use of the service media as a lubricant is highly desirable for fluid system components. This is particularly true in the case of rocket systems where many propellants are incompatible with oils and greases. Generally, oxidizers are better lubricants than fuels because they will replenish the oxide layer which is removed as wear during operation. Extensive testing of liquid propellants as lubricants is reported in Reference 35-3. A few of the conclusions applicable to design are presented in Table 6.8.2.2c.

6.8.2.3 LOAD, SPEED, AND LIFE. The life of a bearing is a function of its operating load and rotational or linear speed. Operational loads arise from centrifugal forces due to unbalanced rotating masses, power transfer forces transmitted from one shaft to another, and dynamic forces induced by vibration or shock loads. Loads are usually a combination of radial and thrust. In valves, loads are small and may reduce close to zero in space or under zero gravity

Table 6.8.2.2c. Liquid Propellants as Lubricants
(Reference 35-3)

PROPELLANTS	CHARACTERISTICS
Liquid oxygen, liquid hydrogen nitrogen tetroxide. IRFNA	Suitable lubricants for ball bearings 440-C stainless steel is a satisfactory bearing material Glass-filled Teflon is the best ball bearing cage material Satisfactory for low load, short life operation
RP-1	May be used as a ball bearing lubricant High load carrying capacity Non-metallic cage materials recommended are glass filled Teflon and phenolic Bakelite
Hydrazine, UDMH Ethylene Diamine (EDA)	Not recommended as ball bearing lubricants 440-C stainless steel is not a suitable bearing material

NOTE: Tests were conducted at 25,000 RPM with a maximum raceway stress of 220,000 psi and *DN* (Detailed Topic 6.8.2.3) equal to approximately 10^6.

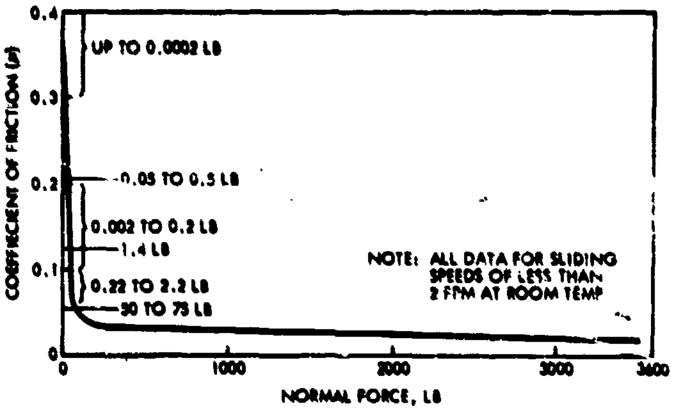


Figure 6.8.2.3. Effect of Load on the Coefficient of Friction of Teflon
(From "Materials in Design Engineering," A Allan and F. Chapman, October 1958, Copyright 1958, Reinhold Publishing Corp., New York)

conditions. Speeds are also not abnormally high. Although actuation times are short, the relative linear or rotational motion is small. However, with the short stroke or rotation, the valve is subjected to frequent startups and shutdowns. If liquid lubricants are used, fully developed fluid film friction will seldom exist, but will be restricted to boundary or mixed film lubrication. If solid lubricants are used, very light loads will result in higher coefficient of friction. The effect of load on the coefficient of friction for Teflon is shown in Figure 6.8.2.3.

Rolling contact bearings should be used for high load, low speed operation. These are excellent where it is necessary to start under full load. In most radial bearings, the fatigue life of the inner race is lower than that of the outer race. Self-pressurized, sliding surface bearings are useful where load increases with rotational speed.

6.8.2.4 TEMPERATURE. Temperature is a major factor in bearing design and selection. The total temperature is the temperature rise due to friction plus the existing ambient temperature. The design of bearings for high temperature use is basically a materials problem. Bearing materials and lubricants which can retain their respective desirable properties at elevated temperatures are necessary. In many cases, the lubricant is used as the heat transfer fluid to carry the heat away from the bearing surface. Sliding surface bearings can prevail upon a larger selection of materials because requirements of hardness and strength are less severe. However, such bearings are limited by changes in surface characteristics and material strength, as well as by lubricant evaporation or gelling caused by high and low temperatures, respectively.

In the moderate range of temperature service (−65°F to 400°F) many liquid and solid lubricants are available, including the MIL oils and greases. In the higher temperature range, many materials and techniques are under investigation to provide lubricants for high temperature service to 6000°F. Most promising have been developments in lubricants that would operate in the temperature range of 400°F to 1000°F. The approach to the problems of high temperature lubrication is not an off-the-shelf selection but rather a careful study of the system itself. The factors to consider for high temperature application involve environments, compatibility with materials, time of operation, and function of the equipment.

The time of operation at elevated temperatures is very important in selecting a lubricant. For instance, a lubricating oil may be subjected to as much as 200°F above its temperature stability point for short periods of time if a cooling system is provided for the lubricants between exposures to the high temperatures.

Most of the development work in high temperature lubrication include the areas of liquid metals, reactive liquids, inert and reactive gases, some organo-metallics, solid inorganic compounds, molten ceramics, graphite and MoS_2. The MoS_2 solid lubricants in powdered form are recommended for high-temperature use up to 750°F. The solid lubricants with a ceramic binder have operated at 1800°F. Reference 172-2 offers an extensive study and analysis of high-temperature (above 700°F) lubrication requirements and problems relating to propulsion systems and space vehicles.

No oils or greases are satisfactory for use at cryogenic temperatures. Only solid films, such as MoS_2 powders, can be used. Low temperature oils, indicated by their pour point, are:

petroleum oils, −40°F
polyester oils, −60°F
dioctylphthalate, −75°F
silicone oils, −130°F

Greases with a petroleum base, ester base, polyglycol base, or silicone base are available for low temperature lubrication. Lithium soaps are often used because they adhere well to metal surfaces and do not harden rapidly.

6.8.2.5 CORROSION AND CONTAMINATION. Common problems are oxidization, dirt, and wear particle contamination. Bearings vary in susceptibility tolerance to particle contamination. Rolling contact bearings generally can tolerate very little contamination of any nature. Oxidation products and contamination caught between rollers and races cause brinelling, which will lead to noise and premature fatigue failure. A continuous supply of particles will cause abrasive wear. In sparsely lubricated bearings, contamination is usually driven from the rolling area which remains relatively clean (Reference 1-132).

Sliding surface bearings can be made tolerant to hard particles by choosing materials with high embeddability. However, load carrying capacity and ability to provide tight support must usually be sacrificed to achieve better embeddability. It is usually more advantageous to provide seals and shields to keep the dirt out of the bearings, or to provide for a circulatory lubricating system with filters to flush dirt away.

6.8.2.6 FRICTION AND WEAR. Friction can be defined as the resistance to relative motion developed between loaded interfacial surfaces. A relative measure of friction is the coefficient of friction which exists for both dynamic and static conditions. For dry surfaces, the coefficient of static or starting friction is always higher than that for dynamic or kinetic friction. As a result, the starting torque requirement of the bearing is usually the most important factor to consider.

Friction coefficients can be determined for fluid-film lubricants on the basis of dynamic viscosity. Coefficients of kinetic friction of solid lubricants must be determined from actual test. Coefficients of more commonly used solid lubricants are presented in Table 6.8.2.6. Additional information can be found in Reference 103-3. Torque requirements for several rotating bearings under both high and low loads are presented as a function of speed in Figure 6.8.2.6a. The effect of sliding speed on the coefficient of friction of a self-lubricating material is shown in Figure 6.8.2.6b. These curves show the nature of the friction curve, which can change considerably with small changes in geometry or viscosity.

The starting torque is high for a fluid film bearing as it goes through the boundary and mixed stages of lubrication. However, when fully developed hydrodynamic lubrication is established, running friction is less than or

Table 6.8.2.6. Coefficients of Kinetic Friction
(References 103-3 and 65-25)

MATERIAL	μ
Molybdenum disulfide (MoS_2)	0.035
Tungsten disulfide (WS_2)	0.035
Graphite	0.045
Teflon (S = 1 ft/min; P = 1500 psi)	0.016
Teflon + 30% MoS_2 (S = 1 ft/min; P = 1500 psi)	0.026
Teflon + 10% graphite (S = 1 ft/min; P = 1500 psi)	0.017
Teflon + 25% glass fiber (S = 20 ft/min; P = 120 psi)	0.230
Teflon + MoS_2 + glass fiber	0.03
Teflon + 40% bronze (S = 20 ft/min; P = 120 psi)	0.240
Teflon + 85% copper (S = 20 ft/min; P = 120 psi)	0.330
Nylon	>0.600
Nylon + 40% MoS_2 (S = 230 ft/min)	0.20
Nylon + 20% graphite (S = 230 ft/min)	0.08
Sapphire	0.13-0.22
Boron nitride	0.60

S = surface speed
P = bearing load

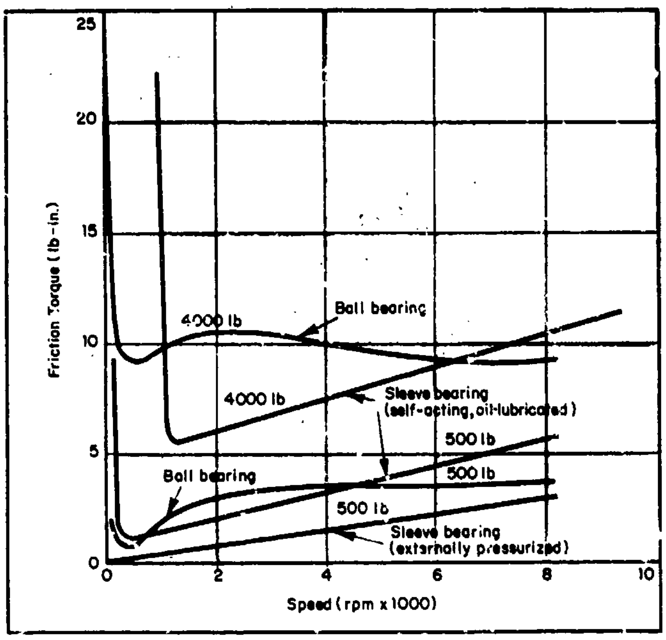


Figure 6.8.2.6a. Friction and Speed Relationships for Different Bearing Types

All be. . ıgs are 40 mm ID. Ball bearing curves are measured experimental. · sleeve bearing curves are calculated. Ball bearings starting torque: .0 lb-in. with 500 lb. radial load; 25 lb-in. with 4000 lb. radial loa' 'ei acting, oil-lubricated sleeve bearing data: L/D = 0.5; C/R = 0.00. ·ubricant viscosity = 2 × 10^{-6} reyns; minimum film thickness = 20 mu in; sliding friction coefficient = 0.15. Starting torque: 60 lb-in. with 500 lb. radial load; 472 lb-in. with 4000 lb. radial load. Externally pressurized sleeve bearing data: L/D = 0.5; C/R = 0.001; lubricant viscosity = 2 × 10^{-6} reyns; land area = recess area.

(From "Machine Design," A. O. DeHart, 1961, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

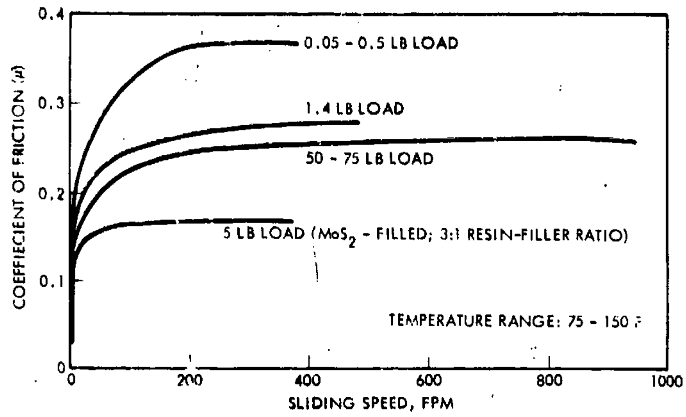


Figure 6.8.2.6b. Effect of Sliding Speed on the Coefficient of Friction of Teflon

(From "Materials in Design Engineering," A. Allan and F. Chapman, October 1958, Copyright 1958, Reinhold Publishing Corp., New York)

comparable to rolling element bearings. For an externally pressurized bearing, the starting torque is zero and the running torque is low. For self-lubricated bearings, torque is quite variable with coefficients of friction ranging from 0.04 to 0.16. In rolling contact bearings running friction is less than starting friction.

The cumulative effect of friction is wear, which may take any of several following forms:

Adhesive wear: adhesion between surface asperities of two contacting materials resulting in formation of loose fragments.

Abrasive Wear: removal of material by the filing action of a hard, rough surface against a softer one.

Corrosive Wear: dislodging of corrosion products.

Surface-Fatigue Wear: crack development in or near the sliding surface.

The sliding surface may give rise to any one or any combination of the several forms of wear. Wear resistance is improved by providing hard materials to reduce penetration of the surface, toughness to prevent breaking off of small particles, and smooth surfaces to reduce asperities. Materials with high stiffness or those that form a protective coating of corrosion products will reduce galling by preventing close atomic contact. Fluid film lubrication will reduce wear, but boundary lubrication is of lesser value. Hard materials will generally reduce cracking, pitting, and spalling (Reference 111-2).

Discussion of the theory of wear is given in References 111-2 and 420-1. Wear coefficients (average wear/mm) for many materials are tabulated in Reference 103-3.

6.8.2.7 SPACE ENVIRONMENTS. The environmental conditions of space, namely vacuum, radiation, and zero-gravity, affect the performance of bearings, particularly in their lubrication. A comparison of a few lubricants for space applications is shown in Table 6.8.2.7a.

Vacuum Effects. Vacuum detrimentally affects bearing materials and lubricants by: (1) failure to replenish the oxide layer on the bearing surface, and, (2) evaporation or sublimation of the lubricant or bearing metal. The lack of an oxide layer allows the metal surfaces to join by cold welding or, if a liquid lubricant is present, to react more readily with the lubricant to form a varnish-like coating on the bearing surfaces. The effect of vacuum on the removal of the oxide layer from metal surfaces is measured by changes in coefficients of friction. The changes in coefficient of friction of aluminum on aluminum and beryllium-copper on beryllium-copper at 10^{-8} torr are shown in Figures 6.8.2.7a and 6.8.2.7b, respectively. Since friction is considered to be the result of shearing of metallic junctions formed by cold welding of asperities, any method that can be devised to contaminate the surfaces will be

Table 6.8.2.7a. Comparison of Lubricants for Bearings in Space

LUBRICANT	TEMPERATURE (°F)	VACUUM	RADIATION	SLIDING CONTACT	ROLLING ELEMENT
Diester-oil-based greases	−65 to 350	P	P	G	E
Mineral-oil-based greases	−40 to 250	P	P	G	E
Silicone-oil-based greases	−100 to 400	P	P	G	E
Molybdenum disulfide	−300 to 750	E	E	E	P
Graphite	−300 to 1200	P	E	G	P
Tungsten disulfide	850	E	E	U	U
Teflon	−300 to 500	G	G*	E	E (as retainer)
Nylon-Bonded coatings with:	−40 to 200	P	G	E	P
Organic binders	−100 to 500	P	P	E	P
Inorganic binders	−300 to 1000	E	E	G	P

*Teflon is relatively resistant to radiation alone; however, if Teflon is in contact with an oxidizer while being irradiated, it rapidly deteriorates.

E = Excellent
G = Good
P = Poor
U = Unknown

helpful in reducing friction. Additional information on cold welding is presented in Detailed Topic 13.6.2.4.

Wear is also aggravated when the oxide layers are removed between the sliding surfaces. Wear particle size formed between the metallic surfaces is believed to be maximum in vacuum. The same methods used to reduce friction will also be helpful in reducing wear.

The rate at which molecules leave the surface of a material varies only as a function of temperature, the vapor pressure of the material, and the exposed surface area. It is independent of the ambient pressure, which determines only the rate that molecules return to the surface by varying the probability of collisions with molecules in the environment and bouncing an evaporating molecule back toward the surface.

Evaporation of any component of the material changes the bulk properties of the material. For example, volatile additives in lubricants will separate, resulting in a highly viscous residue and failure of the lubricant to form metallic soaps which provide lubrication. The best lubricants for vacuum application are the high-temperature-low-vapor-pressure types. The evaporation rates of various oils and greases as a function of temperature at pressures from 8 x 10^{-7} to 2 x 10^{-6} torr are shown in Figure 6.8.2.7c, and evaporation rates of some low vapor pressure oils in vacuum are given in Table 6.8.2.7b. Long chain polymers with high molecular weights have very low vapor pressures and, consequently, low evaporation rates. It should be noted, however, that in some cases radiation may break down long polymer chains into shorter fragments which may have higher vapor pressures.

The evaporation rates of various MoS_2 coatings as a function of temperature at 2 x 10^{-6} torr are shown in Figure 6.8.2.7d. MoS_2 is an effective lubricant in vacuum. Indications are that sulfur is released at the interfaces during sliding and performs the same function for molybdenum disulfide as water vapor does for graphite. However, graphite is not satisfactory as a lubricant in vacuum because the vacuum removes the absorbed film of water, and eventually acts as an abrasive (Reference 131-9). Teflon is an effective dry lubricant having good stability in a vacuum at temperatures up to 212°F.

The composition and amount of gases evolved from the more prominent lubricants subjected to vacuum are presented in tabular form in References 96-4 and 1-266.

Solid film lubricants withstand vacuum conditions better than oils or greases, but are acceptable for ball bearings only for low load and short life applications. The big disadvantage of dry over liquid lubricants is that dry films tend to wear away and cannot be replaced. In cases where liquid lubricants are required, the use of labyrinth seals which maintain high pressure differentials can be effective in reducing losses such that an oil reservoir can provide sufficient lubrication for the mission duration. For long space missions, the reservoir may be unsatisfactory, requiring the use of hermetic sealing. Hermetic sealing requires additional space, weight, and, in some cases, electrical feed-through. Maintenance becomes more difficult.

Radiation Effects. Excessive radiation dosages will result in bearing material and lubricant degradation. The following effects are commonplace:

a) Bearing materials lose strength and surface hardness
b) Some of the liquid lubricants become more viscous, but some of commercial dry films show little effect. In fact, wear life may be improved by irradiation.

Radiation data on various lubricants will not be presented here. See Detailed Topic 13.6.3.7 and reports published by the Radiation Effects Information Center, Battelle Memorial Institute, Columbus 1, Ohio.

Zero-g Effects. Although a zero-g field results in loads due to the weight of all moving parts being reduced to zero,

Table 6.8.2.7b. Evaporation of Low Vapor Pressure Oils in High Vacuum
(Reference 174-5)

TYPE	CHEMICAL NAME	MOLECULAR WEIGHT	TEMPERATURE* AT WHICH EVAPORATION IS: 1000 Å /YEAR °F	10^{-3} CM/YEAR (0.0004 IN./YEAR) °F	10^{-1} CM/YEAR (0.04 IN./YEAR) °F
Hydrocarbon	Alkane, 29-Carbon	414	−60	−20	20
	Alkane, 33-Carbon	460	−50	−10	40
Diester	Di-n-butyl Phthalate	278	−90	−50	0
	Di-isoamyl Phthalate	306	−80	−40	0
	Di-n-amyl Sebacate	343	−50	−10	40
	Di-2-ethylhexyl Phthalate	390	−40	0	50
	Di-2-ethylhexyl Sebacate	426	−22	20	70
Silicone	Permethyldodecasiloxane	900	−40	0	40
	Permethylhexadecasiloxane	1200	30	70	120
	Tetraphenyltetramethyltrisiloxane	490	−10	30	90

*All temperatures are approximate

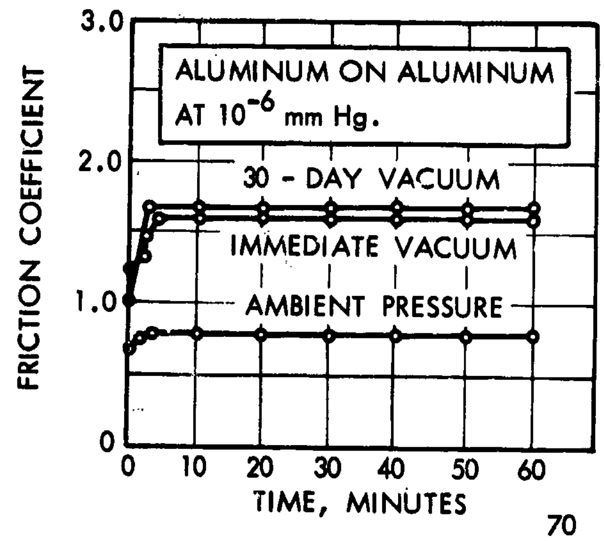


Figure 6.8.2.7a. Coefficient of Friction of Aluminum on Aluminum under 10^{-6} torr
(From "Technical Memorandum 60-7," Litton Systems)

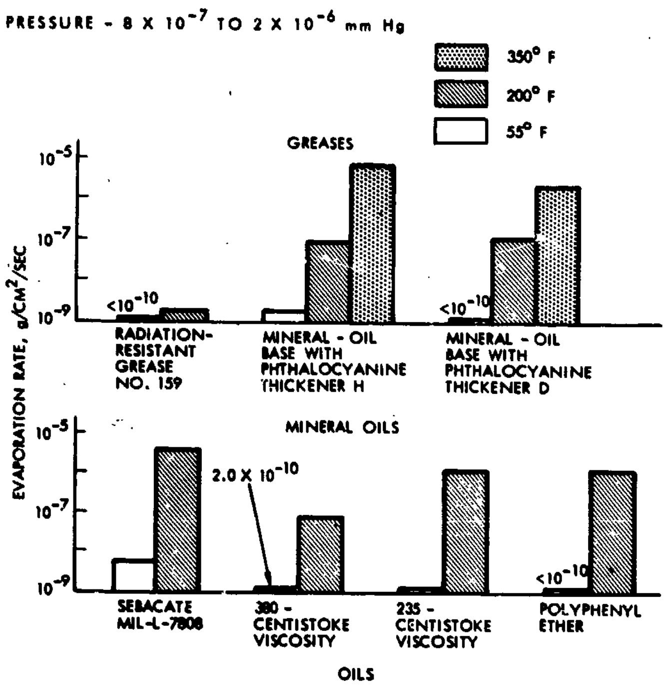


Figure 6.8.2.7c. Evaporation Rates of Various Oils and Greases (P = 8×10^{-7} to 2×10^{-6} torr)
(Reprinted with permission from "ASLE Transactions," Buckley, Swikert, and Johnson, vol. 5, April 1962, Copyright 1962, Academic Press, Inc., New York, New York)

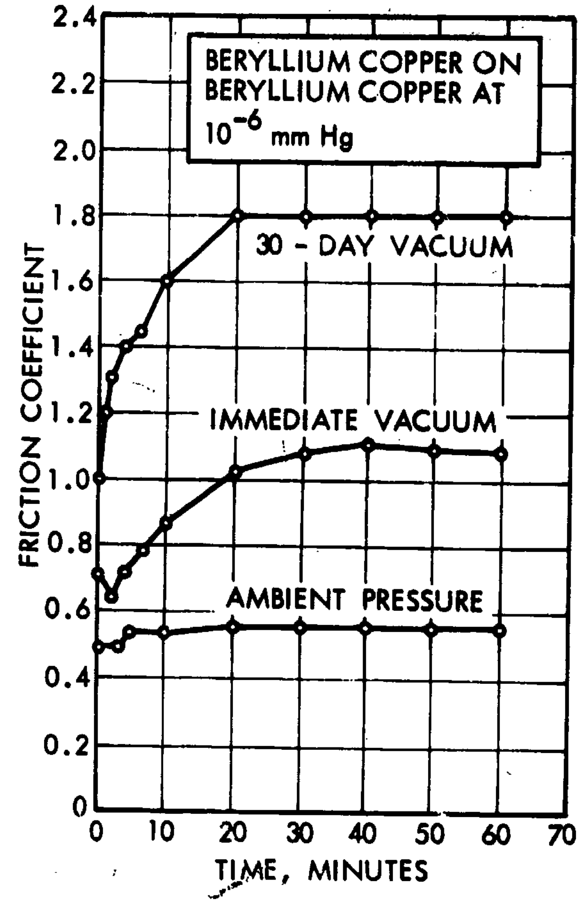


Figure 6.8.2.7b. Coefficient of Friction of Beryllium Copper on Beryllium Copper at 10^{-6} torr
(From "Technical Memorandum 60-7," Litton Systems)

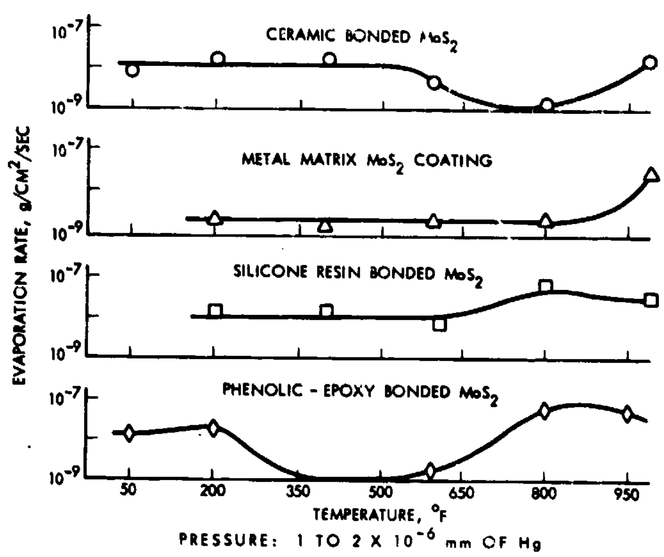


Figure 6.8.2.7d. Evaporation Rates of MoS_2 Coatings in Vacuum (P = 2.0×10^{-6} torr)
(Reprinted with permission from "ASLE Transactions," Buckley, Swikert, and Johnson, vol. 5, April 1962, Copyright 1962, Academic Press, Inc., New York, New York)

the major bearing forces resulting from inertial loads still occur.

At zero-g, gravity fed lubrication systems cannot be used as oils or greases in reservoirs migrate at random. Typical methods to minimize this problem include the use of a pressurized gas blanket over the oil, and centrifugal oil collection.

6.8.3 DESIGN DATA

6.8.3.1 ROLLING ELEMENT BEARINGS*. Design data for single row, deep groove and angular contact, ball bearings are presented is Figure 6.8.3.1a. Other types of rolling contact bearings are seldom used in valve design. Additional design data can be found in References 1-132 and 1-267, as well as in manufacturers' catalogues.

Definitions. The following definitions were adapted from the American Standard B3.11-1959 and Reference 1-267, page 70.

Life: the total number of revolutions before fatigue is evident.

Fatigue (B10) Life: a point selected on the failure distribution curve where 90 percent of the bearings still operate satisfactorily. This life is 20 percent of the average life. A ball bearing failure distribution curve is shown in Figure 6.8.3.1b. Fatigue life is also called *rating life.*

Basic Load Rating: the constant stationary radial load, C, which a group of identical bearings with stationary out ring can endure for 10^6 revolutions of the inner ring.

Equivalent Load: the constant stationary radial load, P, which if applied to a bearing with rotating inner ring and stationary outer ring, would give the same life as that which the bearing will attain under the actual conditions of load and rotation.

Static Load: load acting on a non-rotating bearing.

Basic Static Load Rating: the static radial load, C_o, which corresponds to a permanent deformation of ball or roller and race of 0.0001 of the ball diameter at the most stressed contact. In the angular contact bearings, the basic static load rating relates to the radial component of that load.

Static Equivalent Load: the static radial load, P_o, which would cause the same total permanent deformation at the most heavily stressed ball as that which occurs under actual loading conditions.

Life and Load Equations. The rating life can be calculated by the following equation:

$$L = \left(\frac{C}{P}\right)^b \quad \text{(Eq 6.8.3.1a)}$$

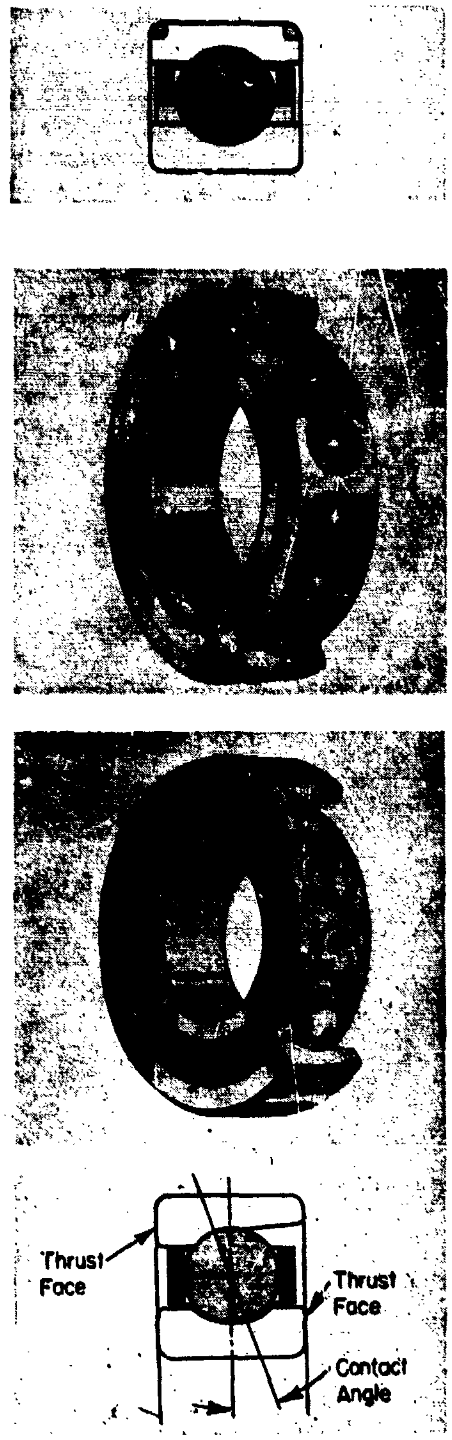


Figure 6.8.3.1a. Radial and Angular Ball Bearings
(From "Machine Design," H. Belanger, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

*Adapted from Reference 1-267

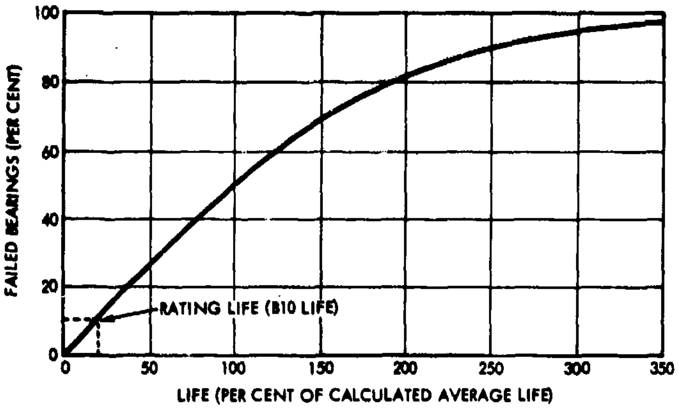


Figure 6.8.3.1b. Typical Failure Distribution Curve for Ball Bearings

(From "Machine Design," vol. 35, no. 14, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

where L = rating life in millions of revolutions
C = basic load rating
P = equivalent load, lb_f
b = exponent

The exponent b varies for different manufacturers. Most manufacturers use $b = 3.0$, which is the value specified in the American Standard B3.11-1959. Other commonly used values are 4.0 and 3.33.

The dynamic and static equivalent loads can be calculated by the following:

Dynamic:

$$P = XVF_R + YF_t \qquad \textbf{(Eq. 6.8.3.1b)}$$

where P = dynamic equivalent load, lb_f
X = radial factor
V = rotation factor
F_R = radial load, lb_f
Y = thrust factor
F_t = thrust load, lb_f

The value of V is equal to unity for inner ring rotating in relation to load, and $V = 1.2$ for inner ring stationary. For dynamic loads, values of X and Y are given in Table 6.8.3.1a. The value of P should be chosen from the largest value obtained from calculations using either X_1, Y_1, X_2Y_2.

Table 6.8.3.1a. Values of X and Y

BEARING TYPE	X_1	Y_1	X_2	Y_2
Radial-Contact	1	0	0.5	1.4
Angular-Contact (shallow angle)	1	1.25	0.45	1.2
Angular-Contact (steep angle)	1	1.75	0.4	0.75

Static:

$$P_o = X_oF_R + Y_oF_t \qquad \textbf{(Eq 6.8.3.1c)}$$

where P_o = static equivalent load, lb_f
X_o = radial factor
F_R = radial load, lb_f
Y_o = thrust factor
F_t = thrust load, lb_f

For static loads, values of X_o and Y_o are given in Table 6.8.3.1b.

Table 6.8.3.1b. Values of X_o and Y_o

BEARING TYPE	X_o	Y_o
Radial Contact	0.6	0.5
Angular Contact		
$\alpha = 20°$	0.5	0.42
$\alpha = 30°$	0.5	0.33
$\alpha = 40°$	0.5	0.26

With intermittent use the total life becomes a summation of the lives under dynamic and static loads. It should be noted that under dynamic bearing conditions higher loads can be carried without fatigue than can be carried under static conditions.

Calculations of Loads from Manufacturers' Data. If catalog load ratings are given in terms of number of hours of life at a given speed, the load can be calculated from the following:

$$P = \frac{H_cN_c}{H_AN_A} \cdot C_c \qquad \textbf{(Eq 6.8.3.1d)}$$

where P = equivalent dynamic, radial load, lb_f
H_c = catalogue rating life, hr
H_A = expected application life, hr
N_c = catalogue speed, rpm
N_A = actual speed, rpm
C_c = catalogue basic radial load rating, lb_f

Speed and life Equations. The relationship of speed and life may be approximated by the equation

$$C = \frac{P\,f_L}{f_s} \qquad \textbf{(Eq 6.8.3.1e)}$$

where C = basic load rating
P = equivalent dynamic load, lb_f
f_s, f_L = speed and life factors, respectively

If C, P, and either speed or life are known, the remaining unknown may be found from Figure 6.8.3.1c

For bearings in oscillating applications, the effective rotating speed is given by

$$\omega = \frac{N\,\theta}{180} \qquad \textbf{(Eq 6.8.3.1f)}$$

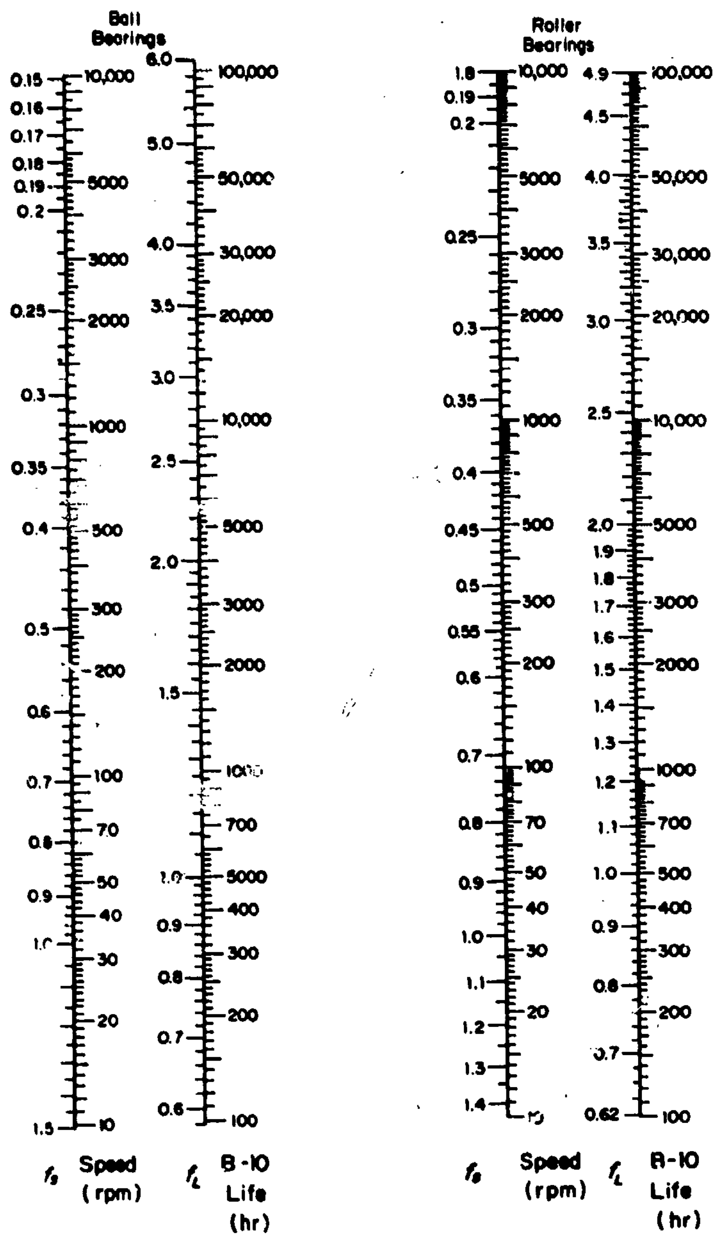


Figure 6.8.3.1c. Correction Factors for Bearing Life and Speed
(From "Machine Design," H. Belanger, 1961, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

where ω = equivalent rotative speed, rpm

N = oscillating speed, oscillations per minute

θ = angle of oscillation

The angle of oscillation is critical because of the chance of fretting corrosion.

Bearing Friction. The bearing frictional torque for ball bearings can be calculated from the following equation:

$$T = \mu P\left(\frac{d}{2}\right) \qquad \text{(Eq 6.8.3.1g)}$$

where T = torque, in-lb_f

P = equivalent load, lb_f

d = bearing bore, in.

μ = coefficient of friction (Table 6.8.3.1c)

Permissible Speed. The speed capability of the bearing depends upon its internal geometry, retainer construction, and type and method of lubrication. Very often the limited speed is defined by the DN number which, in turn, is defined as the bearing bore in millimeters times the speed in RPM. The DN number for radial deep groove bearings with oil as a lubricant is 600,000; for angular contact less than 25° and greater than 25°, 800,000 and 500,000, respectively. Additional values are presented in Table 6.8.3.1d.

Coating Rolling Element Bearings. To provide lubrication for rolling element bearings in a vacuum, the retainers are often coated with a dry-film lubricant. Both the races and retainers can be coated if care is taken to keep the thickness on the races at a minimum in order not to jam the bearings. Film thickness must be taken into account in dimensioning and specifying tolerances on drawings. Allowances must be made for the thickness of the film on surfaces that show a dimension to four decimal places, which usually involves a four-decimal tolerance. If plating is to be used to provide a hard substrate prior to coating with a lubricant, the thickness of the plating should be considered. Selecting balls that are slightly undersized, or selecting races on the high side of the tolerances can help avoid jamming by a thick lubricant film or coating. The coating on the retainer is generally thicker than on the races, since a thicker coating on the retainer provides a reservoir for longer life. However, the coating should not be so thick as to result in poor adhesion to the substrate metal.

Selection of Rolling Element Bearings. The types of bearings discussed are those most commonly used in valve design. However, in special circumstances the more specialized type of bearing will be more applicable. A chart showing many of the different types of bearings, and listing their characteristics is shown in Table 6.8.3.1d.

Table 6.8.3.1c. Typical Coefficients of Bearing Friction
(From Kent's "Mechanical Engineers' Handbook," 12th Ed., Copyright 1950 by John Wiley and Sons)

BEARING TYPE	μ
Self-aligning ball bearings	0.0010
Cylindrical roller bearings (short rollers, flange guided)	0.0011
Thrust ball bearings	0.0013
Single row ball bearings	0.0015
Spherical roller bearings	0.0018

Table 6.8.3.1d. Comparison Chart for Rolling Element Bearings

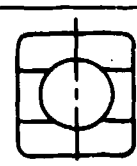
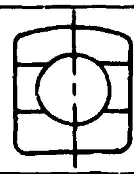
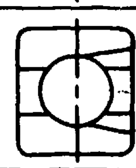
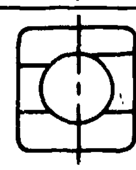
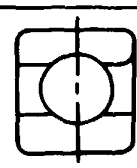
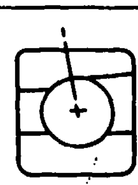
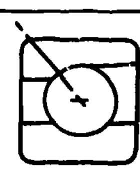
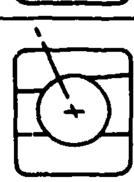
BEARING TYPE	CROSS-SECTIONAL DESIGN	RADIAL CAPACITY	THRUST CAPACITY*	OPERATING SPEED	OPERATE W/ MISALIGNMENT	SPECIAL CONSIDERATIONS
Conrad-type, single row radial		5	5 (2D)	8	3	Adaptable to many seal and shield designs, available in 0.0 sizes from ⅛ to over 12 ft; can be obtained with dirt shields; DN with oil—600,000; with grease—200,000.
Conrad-type, with spherical OD		5	5 (2D)	8	8	DN with oil—300,000; with grease—200,000.
Maximum capacity with loading groove		6	2 (2D)	5	2	More balls can be assembled between the rings than in Conrad; bi-directional thrust capacity is reduced; DN with oil—300,000; with grease—200,000.
Maximum capacity, counter bored construction		6	5 (1D)	8	2	
Maximum capacity, fractured outer ring		6	5 (2D)	8	3	Tight housing fits must be used with high thrust loads.
Angular contact, shallow angle		5	6 (1D)	8	2	Shallow angle $0 \leq \theta < 15$; steep angle $15° \leq \theta < 40°$; radial load capacity and speed is inversely proportional to angle; friction and thrust load directly proportional; DN with oil—800,000; with grease—300,000.
Angular contact, steep angle		4	7 (1D)	5	1	Increased axial and radial rigidity is obtained by preloading; DN with oil—300,000; with grease—200,000.
Angular contact, nonmetallic cage		5	5 (1D)	10	1	

Table 6.8.3.1d. Comparison Chart for Rolling Element Bearings (Continued)

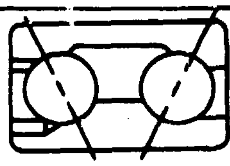
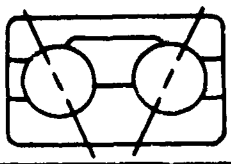
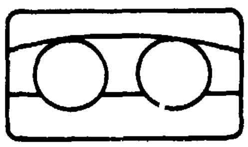
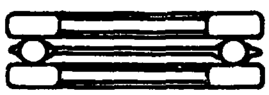
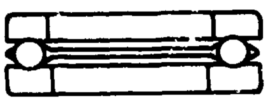
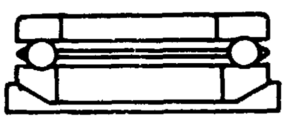
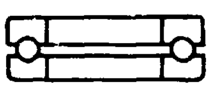
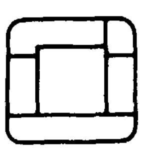
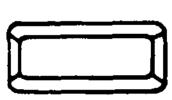
BEARING TYPE	CROSS-SECTIONAL DESIGN	RADIAL CAPACITY	THRUST CAPACITY*	OPERATING SPEED	OPERATE W/ MISALIGNMENT	SPECIAL CONSIDERATIONS
Double-row angular contact, rigid type		7	5 (2D)	5	1	Bearings are usually matched with one low angle for radial capacity, and one steep angle for thrust capacity; require accurate alignment of housing and shaft.
Double-row angular contact, non-rigid type		7	5 (2D)	5	3	
Double-row, internally self-aligning		2	1 (2D)	5	9	Radial and thrust capacity low because of poor raceway conformity to mating curvature of balls; cannot resist binding loads.
Ball thrust, flat race		0	5 (1D)	2	2	Load capacity is low due to race-ball inconformity; operating speeds are low because of centrifugal ball forces.
Ball thrust, grooved race		1	7 (1D)	5	2	Friction forces higher than with flat race.
Self-aligning ball thrust, grooved race		1	7 (1D)	4	8	Compensates for initial static misalignment instead of dynamic misalignment; DN with oil—300,000.
Ball thrust		2	8 (1D)	4	2	Does not require a retainer and can support moderate radial loads with large thrust loads; DN with oil—225,000.
Cylindrical roller (6 types)		8	0	7	2	Available in six lip arrangements (Reference 1-32, page 48). Load capacities are 1½ to 2 times radial ball bearings; requires accurate alignment of housing and shaft; DN with oil—300,000.
Journal roller		9	0	2	1	Wider than roller bearings and can be used without the inner ring; axial retainer must be provided.

Table 6.8.3.1d. Comparison Chart for Rolling Element Bearings (Continued)

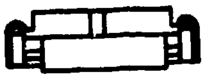
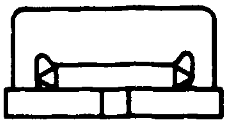
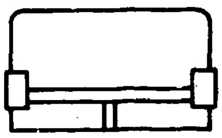
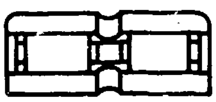
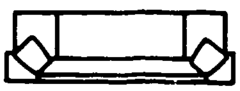
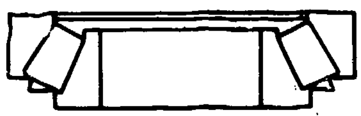
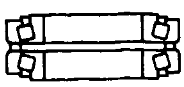
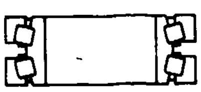
BEARING TYPE	CROSS-SECTIONAL DESIGN	RADIAL CAPACITY	THRUST CAPACITY*	OPERATING SPEED	OPERATE W/ MISALIGNMENT	SPECIAL CONSIDERATIONS
Drawn-cup needle, full complement		6	0	5	1	May be used without inner ring, but cannot resist thrust; axial location must be provided; dimensions are given in inches rather than metric units.
Drawn-cup needle, cage type		5	0	6	3	
Heavy-duty needle, full complement		8	0	5	2	Provides high load capacity in minimum radial space; higher internal friction, lower speed limits, and greater diametral looseness than ball bearings.
Heavy-duty needle, cage type		7	0	6	3	
Heavy-duty needle, center guided rollers		8	0	5	3	
Single row, shallow angle, tapered roller		8	5 (1D)	7	2	Can support heavy, combined radial and axial loads; thrust load capacity and radial load capacity are respectively directly and inversely proportional to the contact angle; accurate alignment of housing and shaft is required.
Single row, steep angle, tapered roller		7	8 (1D)	5	2	
Double row, Type TDO tapered roller		9	7 (2D)	4	1	Main difference is in the direction in which the opposed contact-angle lines of the two rows of rollers diverge; can support thrust loads in both directions.
Double row, Type TDI tapered roller		9	7 (2D)	4	3	

Table 6.8.3.1d. Comparison Chart for Rolling Element Bearings (Continued)

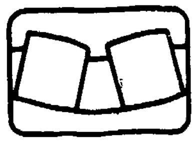
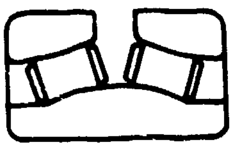
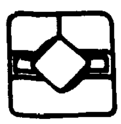
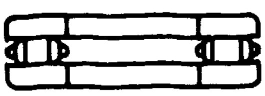
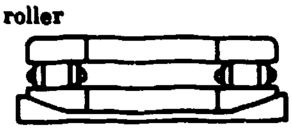
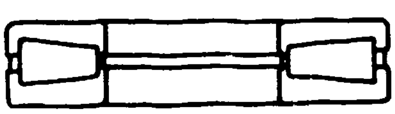
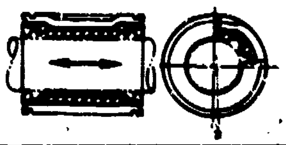
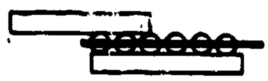
BEARING TYPE	CROSS-SECTIONAL DESIGN	RADIAL CAPACITY	THRUST CAPACITY*	OPERATING SPEED	OPERATE W/ MISALIGNMENT	SPECIAL CONSIDERATIONS
Four row, Type TQO tapered roller		10	8 (2D)	3	1	For heavy thrust loads; (Reference 1-132, pp. 52-56 describes operation). DN with oil—150,000, 250,000; DN with grease—100,000.
Spherical roller, barrel rollers		8	5 (2D)	5	9	
Spherical roller, concave rollers		8	5 (2D)	4	9	
X-type roller		8	8 (2D)	3	5	
Cylindrical roller thrust			9 (1D)	2	1	
Self-aligning cylindrical roller thrust		0	10 (1D)	2	8	For heavy thrust loads; (Reference 1-132, pp. 52-56 describes operation).
Tapered roller thrust		5	9 (1D)	1	9	

Table 6.8.3.1d. Comparison Chart for Rolling Element Bearings (Continued)

BEARING TYPE	CROSS-SECTIONAL DESIGN	RADIAL CAPACITY	THRUST CAPACITY*	OPERATING SPEED	OPERATE W/ MISALIGNMENT	SPECIAL CONSIDERATIONS
Linear ball bearing		4	0	8		Permits very low friction forces; poor conformity of ball to shaft curvature reduces load capacity capability.
Linear roller bearing		10	0	6		Ladder type bearing consisting of two plates separated by a series of caged rollers.

*
0 = none
10 = very high
1D = one direction
2D = two directions

6.8.3.2 SLIDING SURFACE BEARINGS. Three types are discussed as follows:

Metal Bearings. A metal sleeve bearing is a cylindrical casing which encloses and supports a close fitting moving shaft. Many references, Reference 1-3 being noteworthy, present design details for full-rotating journals. However, little information exists for bearings with sliding surfaces, partial rotation, and intermittent operation.

Metal sleeve bearings require lubrication. If liquid lubricants and greases are used in valves, the bearing usually operates under boundary lubrication. More often solid film lubricants or plastic-clad self-lubricating bearings are used.

Shaft Hardness. The shaft should always be harder than the bearing. Recommended shaft hardnesses for several bearings materials are listed in Table 6.8.3.2.

Table 6.8.3.2. Recommended Shaft Hardness
(From Bearing Book, "Machine Design," Copyright 1961 by Penton Publishing Company, Cleveland, Ohio)

BEARING MATERIAL	SHAFT HARDNESS (BRINELL)
Lead base babbit	120
Tin base babbit	120
Copper-lead	500
Lead bronze	300
Phosphore bronze	300
Tin bronze	300
Aluminum-base alloys	250
Cast iron	150
Silver plate	475

In some cases, the designer is left with no alternative but to use identical materials for both shaft and bearing. When this circumstance prevails, lubricants must be used which are sufficiently softer than the shaft or bearing material.

Surface Finish. The roughness of the surface finish and direction of surface finish grooves affect the coefficients of friction, wear life, and load carrying capability.

The effect of surface roughness on friction and wear depends on the type of lubricant used and the surface pretreatment. Using mineral oil as the lubricant the coefficient of friction increases from 0.128 for a 2 super finish to nearly a constant value of 0.36 with a 20 finish. With MoS_2 as a lubricant, optimum wear life occurs at ground surface finish of 20 microinches. Surface treatment with reduced surface roughness improves wear life substantially as shown in Figure 6.8.3.2a.

The load carrying capability can often be improved by reducing the surface roughness. As an example, the load carrying capability can increase by a factor of nine as surface roughness decreases from 150-200 rms to 20-15 rms.

Furthermore, the coefficient of friction may be reduced if surfaces are placed together with grinding marks parallel and sliding motion perpendicular to the grooves.

Coefficient of Friction. Clean, similar metals in sliding contact exhibit large coefficients of friction and will cold weld. Unlubricated surfaces in a vacuum also exhibit such characteristics. Normally, oxides formed on the surface in air promote some lubricity. However, sliding will remove the oxide faster than it can be replaced and will cause seizure. Additional friction reducing materials must be used.

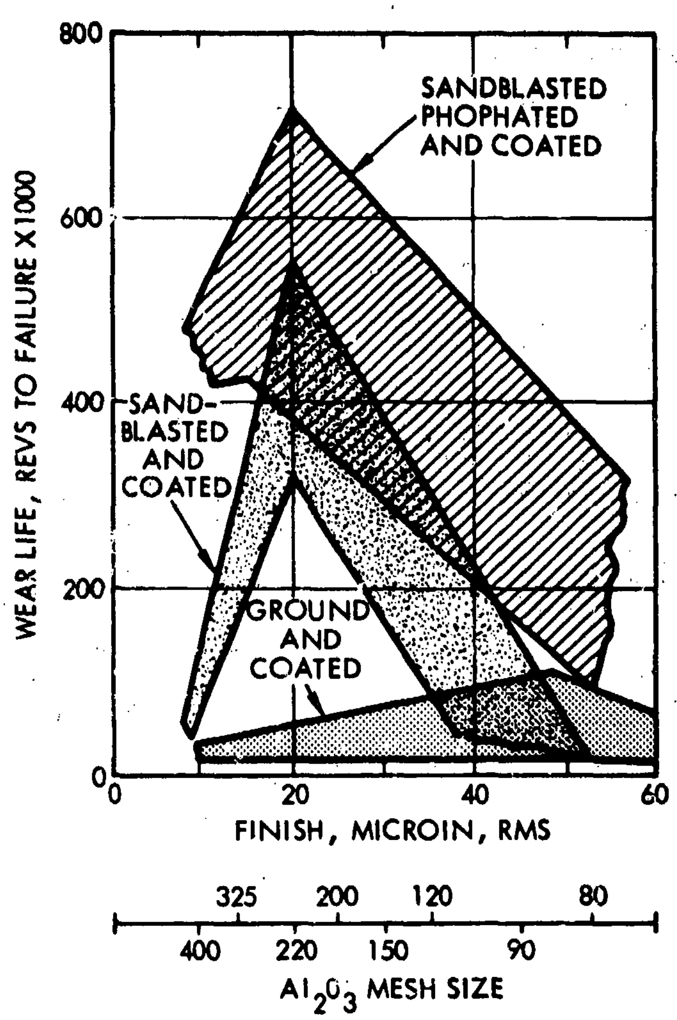


Figure 6.8.3.2a. Effect of Surface Finish on Bearing Wear Life

(Reprinted from "Product Engineering," A. DiSapio, v. 31, no. 36, September 1960, Copyright 1960, McGraw-Hill Publishing Company, Inc., New York)

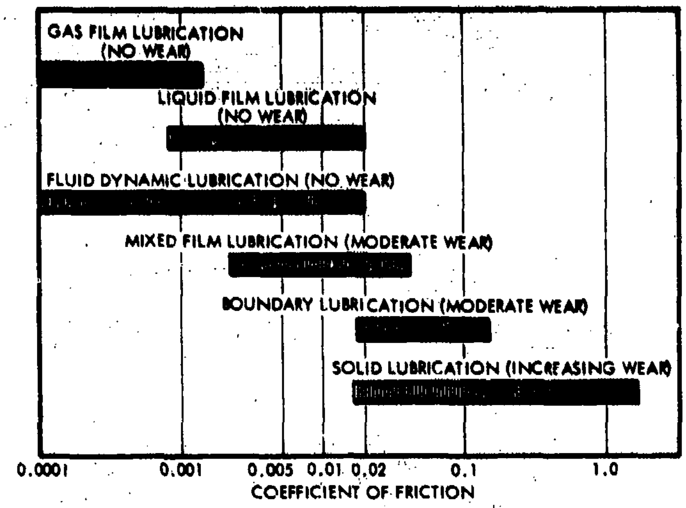


Figure 6.8.3.2b. Lubrication Regimes in Terms of Coefficient of Friction

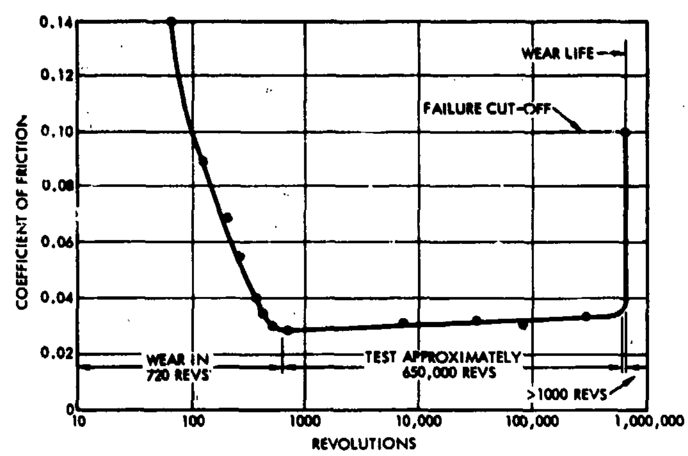


Figure 6.8.3.2c. Typical Wear-In and Wear Life for MoS_2 Solid Film Lubricant

(From "Machine Design," D. F. Wilcock, vol. 35, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

In terms of the lubrication mechanisms, the coefficients of friction for metal bearings with gas lubrication vary between 0.0001 and 0.001; with hydrodynamic liquid lubrication, 0.001 and 0.02. Wear is minimal.

The low value for solid on solid lubrication is approximately 0.02. The range for mixed film lubrication is between 0.005 and 0.1. As friction increases from 0.005, hydrodynamic lubrication decreases and the solid film predominates. A graphical representation is shown in Figure 6.8.3.2b.

Wear-in and Wear Life. Wear-in is the term used to describe the process of smoothing the asperities in contact between two mating surfaces. It is usually noticed as a decrease in the coefficient of friction. The effects of wear-in for poorly lubricated surfaces, are galling, scoring, seizing, and the formation of wear particles. The formation of wear particles and their consequences is not well known, but a summary of advances to date is presented in References 131-6. Wear life usually refers to useful life of lubricants in the bearing. The extent of lubricant life is indicated by a rapid increase in the coefficient of friction. Both terms are measured by total revolutions. Figure 6.8.3.2c illustrates a plot of test results using MoS_2 wear-in and wear life. Wear-in effects are lessened by extreme pressure lubricants. Wear life is affected by surface finish

(Figure 6.8.3.2a) mating surface hardness, humidity and temperature. The effect of these parameters on MoS_2 is presented in Reference 19-218.

Clearances. The clearance between shaft and bearing depends on the operating temperature, load, lubricant thickness, and wear particle size. The wear particle size may set the minimum clearance. For full rotating journal bearings, a clearance of 0.001 times the radius is a practical approximation. Suggested clearances by the American Standards Association are presented in Reference 19-57. Practical experience indicates that diametral clearances rarely need to be greater than 0.002 or less than 0.0005 inch. When such bearings are used for poppet guides or in vibration and acceleration environment, the lower limit clearances should be used.

Plastic Bearings (Adapted from Reference 1-233). The use of plastics eliminates the need for lubrication in many applications, and shows excellent performance with high and low temperatures. Plastic bearings reduce noise level, are lightweight, and can be produced economically by injection molding. Plastics include Nylon and Teflon in bulk and varied filled forms. The maximum operating temperature for most plastics ranges from 200°F to 500°F.

PV Factor. The performance of a plastic bearing can be predicted by the PV factor. For rotary motion, the surface velocity

$$V = \frac{\pi}{12} dw \qquad \textbf{(Eq 6.8.3.2)}$$

where d = shaft diameter, in.

w = shaft speed, rpm

For linear motion, the surface speed is the speed of the sliding surface across the mating surfaces.

The parameter, P, is the total load in pounds divided by the projected area of the bearing surface (bearing I.D. × length). The variation of maximum allowable unit pressure as a linear function of surface velocity for an unlubricated bearing is shown in Figure 6.8.3.2d. With bearing lubrication, the PV factor increases, as illustrated in Figure 6.8.3.2e.

Corrections to PV Factor. With operating temperatures above 75°F or for intermittent operation, correction factors should be applied to the allowable unit pressure, P. Thus the maximum allowable unit pressure can be defined as

$$P = P_a C_t C_c$$

where P_a = maximum allowable unit pressure at 75°F

C_t = thermal correction factor, Figure 6.8.3.2f

C_c = cycling time correction factor, Figure 6.8.3.2g. C_c should be used only if the shutdown period is 3 to 4 times the operating period

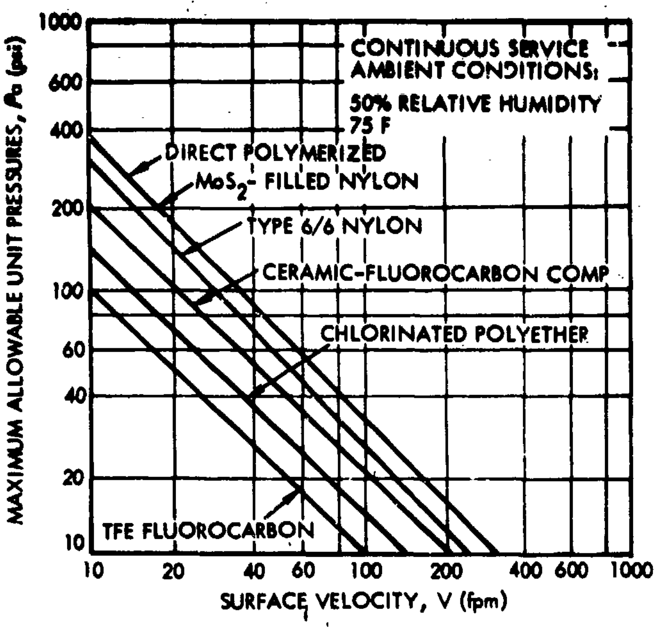


Figure 6.8.3.2d. Variation of Maximum Allowable Unit Pressure with Surface Velocity for Unlubricated Bearings

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

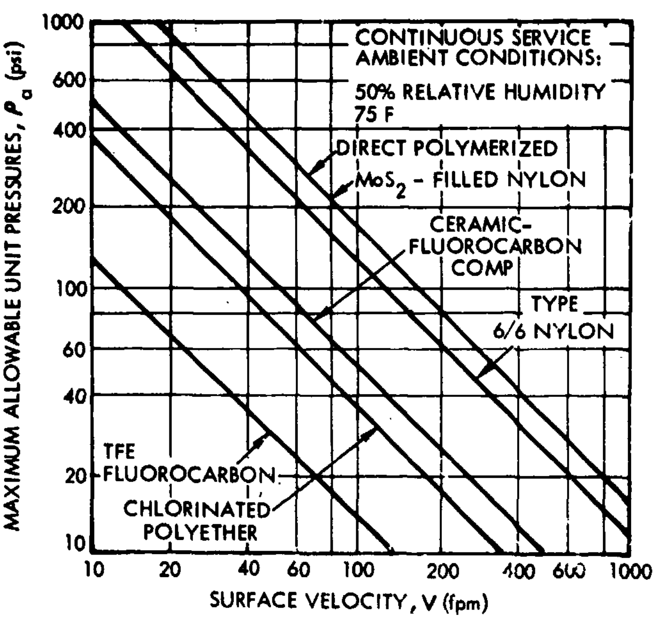


Figure 6.8.3.2e. Variation of Maximum Allowable Unit Pressure with Surface Velocity for Lubricated Bearings

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

Wall Thickness. Recommended wall thickness as a function of shaft diameter is shown in Figure 6.8.3.2h. Impact conditions require substantial increases in wall thickness.

Shape. The bearing shape refers to the length-diameter ratio and has a substantial effect on the coefficient of friction. The coefficient of friction is usually a minimum when the ratio is one, and increases with length due to out-of-roundness and shaft vibration.

Assembly. Plastic bearings are usually press fitted into the bearing housing. The amount of interference (Figure 6.8.3.2i) is dependent upon material and housing diameter. Other means of installation includes mechanical connections (e.g., keys, pins, screws, etc.) and adhesives.

Clearance. Clearance refers to the amount of gap between the shaft and bearing surface, and depends upon the conditions that exist during operation. The basic clearance is called the running clearance, with allowances made to reflect temperature and assembly effects. The total clearance may be computed from the equation

$$C_L = a_1 + (2)^* a_2 + a_3 + a_4$$

where C_L = clearance

a_1 = running clearance (found in Figure 6.8.3.2j)

a_2 = wall thickness allowance in mils, calculated by K_1 (Figure 6.8.3.2k) times bearing wall thickness in inches. Answer is in mils.

a_3 = close-in allowance due to press fit (Figure 6.8.3.2i)

a_4 = thermal allowance of shaft, which can be calculated by K_2 (Figure 6.8.3.2l) times the shaft diameter in inches. Answer in mils.

Carbon-Graphite Bearings (Adapted from Reference 1-153). Provided operation is in the atmosphere, carbon-graphite bearings eliminate the use of lubricants, scoring, or welding and provide good corrosion resistance, low wear, and high hardness. The maximum operating temperature is approximately 750°F.

PV Factor. The PV factor for carbon-graphite is 15000, with a maximum load rating of 600 psi and a maximum surface speed rating of 2500 feet per minute. For lubricated bearings, the PV value increases to 150,000.

Wall thickness. The recommended and absolute minimum allowable wall thickness for carbon-graphite bearings is shown in Figure 6.8.3.2m.

Shape. Most carbon-graphite bearings are straight wall type, since the flanged design with thin walls is less attractive because of the fragility of the carbon. Thin wall sections on large bearings are not desirable. Six inches is regarded as the maximum length for one-piece bushings up to six inches in diameter; larger bearings require seg-

*The coefficient, 2, must be used if the bearing has confined ends.

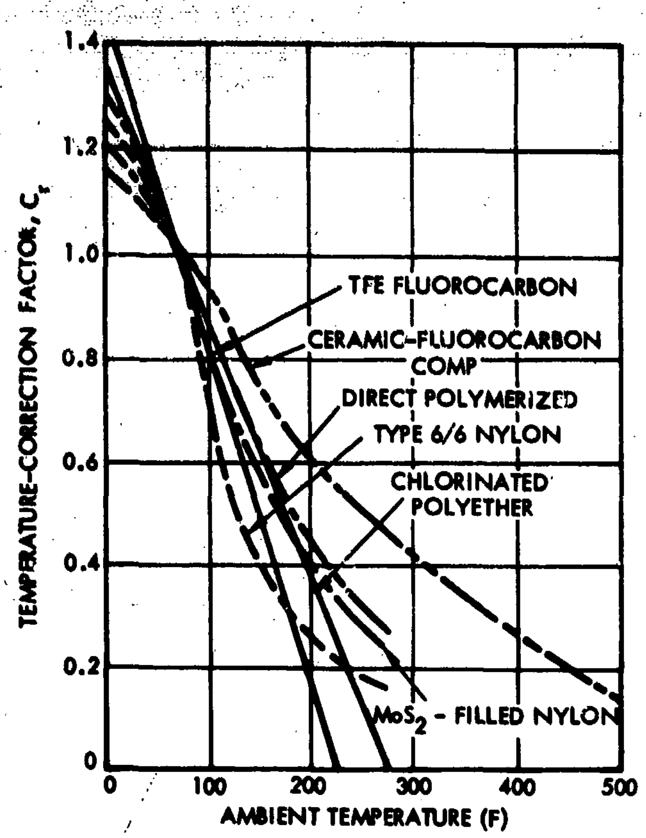


Figure 6.8.3.2f. Temperature Correction Factor, C_t, for Plastic Bearings

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

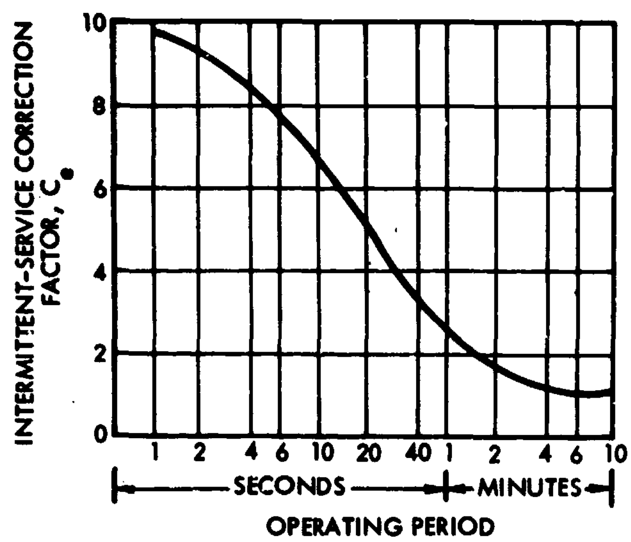


Figure 6.8.3.2g. Intermittent-Service Correction Factor, C_s, for Plastic Bearings

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

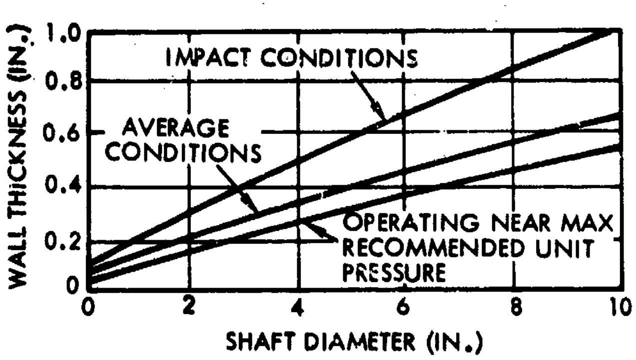


Figure 6.8.3.2h. Recommended Wall Thickness for Plastic Bearings

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

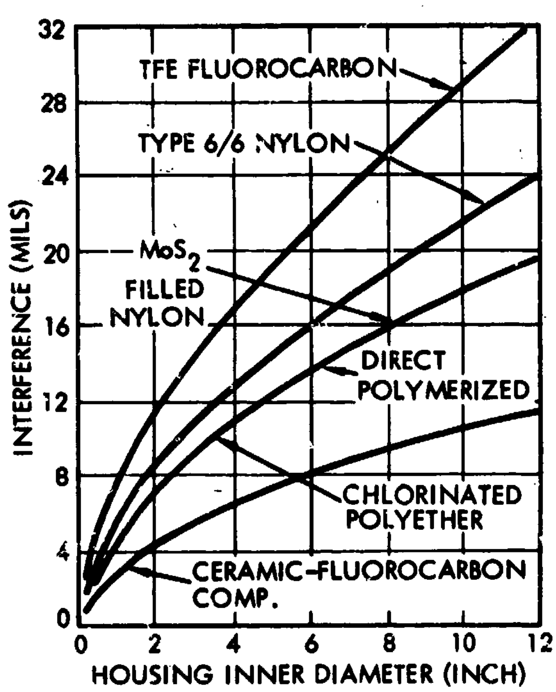


Figure 6.8.3.2i. Recommended Press-Fit Interference for Plastic Bearings

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

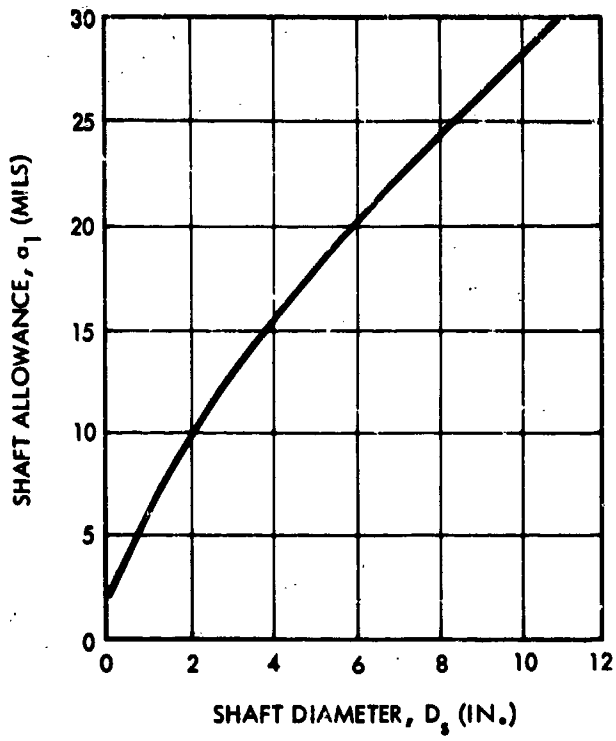


Figure 6.8.3.2j. Shaft Allowance for Various Shaft Diameters

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

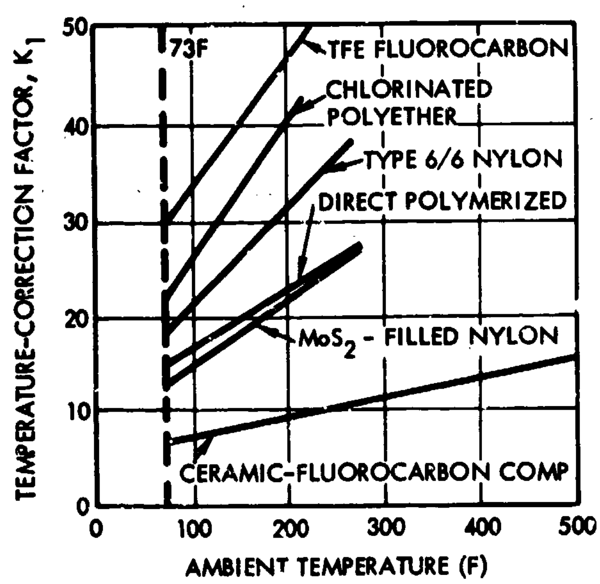


Figure 6.8.3.2k. Temperature Correction Factor for Wall Thickness Allowance Factor for 73°F may be used for all values below 73°F

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

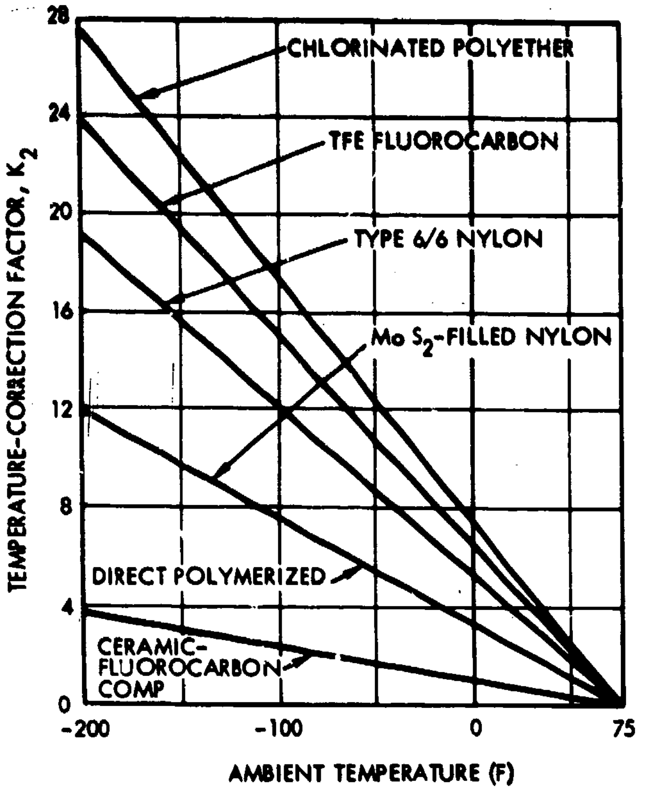


Figure 6.8.3.2l. Temperature Correction Factor for Shaft Allowance

(From "Machine Design," L. M. Rentschler, vol. 34, no. 23, June 1963, Copyright 1963, Penton Publishing Company, Cleveland, Ohio)

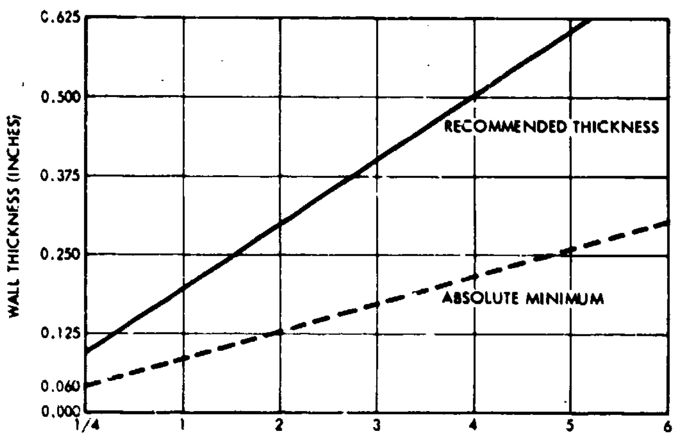


Figure 6.8.3.2m. Recommended and Minimum Wall Thickness for Carbon-Graphite Bearings for Various Shaft Diameters

(From "Machine Design," J. C. Schubert, vol. 26, no. 7, July 1954, Copyright 1954, Penton Publishing Company, Cleveland, Ohio)

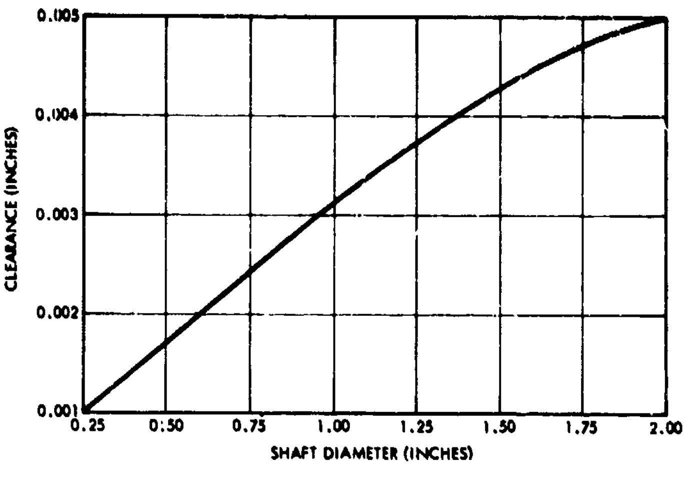


Figure 6.8.3.2n. Recommended Running Clearance for Carbon-Graphite Bearings

(From "Machine Design," J. C. Schubert, vol. 26, no. 7, July 1954, Copyright 1954, Penton Publishing Company, Cleveland, Ohio)

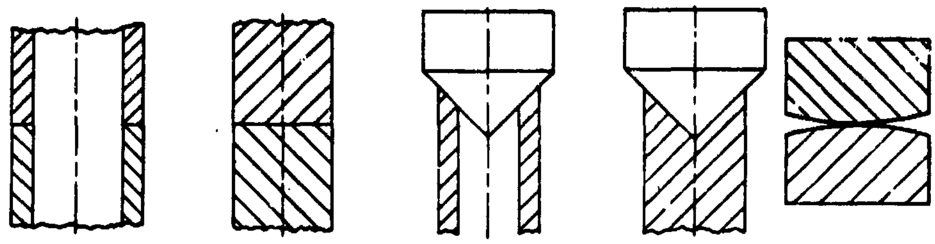
ments. Sharp corners, key slots, and thin sections should be avoided because of the brittleness.

Assembly. Carbon graphite bearings are assembled into the housing with an interference fit, either shrink or press. For press fit the largest recommended interference is 0.005 inch. Larger interferences can be done by the shrink fit. Mechanical-type fasteners and adhesives are not used.

Clearance. The recommended running clearance is shown in Figure 6.8.3.2n. With lubricated bearings, closer running clearances can be used. For high temperature application, correction should be made for diametral changes. The coefficient of thermal expansion is approximately 1.5×10^{-6} inch per inch per °F.

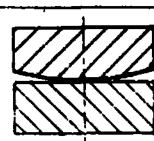
6.8.3.3 PIVOT THRUST BEARINGS. Pivot thrust bearings, sometimes called jewel bearings, offer support to a rotating shaft and can have flat, conical, or spherical surfaces. Hard materials are required. In some instances, high carbon tool steel, glass, sapphire, and even diamonds are used because they are best in terms of low friction, long life, and resistance to shock loading. Some of these materials are costly and difficult to use except in very small and simple configurations.

PV Factor. The PV factor for thrust pivots is less than that for the radial bearings, because they are in constant contact and are not given a chance to cool between contacts. The PV factor depends on the pivot configuration. The equations to determine the PV rating and maximum loads and speed are given in Figure 6.8.3.3.



	Case 1	Case 2	Case 3	Case 4	Case 5*
1. Maximum bearing pressure, p_{max} (psi)	$\frac{0.32F}{a^2 - a_i^2}$	$\frac{0.32F}{a^2}$	$\frac{0.32F}{a^2 - a_i^2}$	$\frac{0.32F}{a^2}$	$\frac{0.48F}{a^2}$
2. Bearing pressure at radius r, p (psi)	Pressure assumed uniform and equal to p_{max}				$p_{max}\frac{(a^2 - r^2)^{1/2}}{a}$
3. Velocity, V (fpm)	$0.524\,rN$				
4. Maximum pressure-velocity product $(pV)_{max}$	$\frac{0.167\,FaN}{a^2 - a_i^2}$	$\frac{0.167\,F}{a}$	$\frac{0.167\,FaN}{a^2 - a_i^2}$	$\frac{0.167\,FN}{a}$	$\frac{0.125\,FN}{a}$
5. Friction torque (lb-in.)	$0.67\,fF\left[\frac{a^3 - a_i^3}{a^2 - a_i^2}\right]$	$0.67\,fFa$	$\frac{0.67\,fF}{\sin\alpha}\left[\frac{a^3 - a_i^3}{a^2 - a_i^2}\right]$	$\frac{0.67\,fFa}{\sin\alpha}$	$0.59\,fFa$
6. Power loss, P (lb-in./sec)	$0.070\,fFN\left[\frac{a^3 - a_i^3}{a^2 - a_i^2}\right]$	$0.070\,fFaN$	$\frac{0.070\,fFN}{\sin\alpha}\left[\frac{a^3 - a_i^3}{a^2 - a_i^2}\right]$	$\frac{0.070\,fFaN}{\sin\alpha}$	$0.062\,fFaN$

$$*a = 0.88\left[\frac{F\left(\frac{1}{E_1} + \frac{1}{E_2}\right)}{\frac{1}{r_1} + \frac{1}{r_2}}\right]^{1/3}$$



Case 6
Identical to Case 5 with $1/r_2 = 0$.

Figure 6.8.3.3. Equations for Pivot Thrust Bearings
(From "Machine Design," F. W. Kinsinan, vol. 33, no. 3, February 1961, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

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1-84	26-130	65-9	131-6
1-122	29-2	65-22	131-9
1-132	35-3	65-23	174-5
1-153	40-4	56-24	400-2
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1-233	62-3	65-26	475-1
1-265	62-4	93-6	V-279
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6.9 ACTUATORS

6.9.1 INTRODUCTION

A valve actuator is a power unit which provides a mechanical operating force to position a valving element; the actuator may be direct-acting or piloted. The direct-acting valve actuators are piston-cylinders, solenoids, electric motors, and servo torque motors, each used independently. Piloting, which involves the use of a small power input to control a larger power source, is common with piston-cylinder actuators which include the electrohydraulic, electropneumatic, pneumatic-hydraulic, or pneumatic-pneumatic combination.

The most basic direct-acting valve actuator is a manually operated handwheel or lever. The handwheel can be attached directly to a rising or non-rising valve stem (see Sub-Section 5.18) or linked to the stem through some mechanical linkage. Unless overhead space is limited, rising stems are sometimes preferred over non-rising stems, because non-rising stem threads which are exposed to the fluid can more readily become contaminated, causing thread wear (Reference 193-3). Handwheel actuators are used for both linear motion (lowering or raising of globe or gate valving elements) or rotary motion (rotation of butterfly, ball plug, or rotary slide valving elements).

Most aerospace valves are powered by remotely controlled, or automatic, actuators. Linear actuators used in aerospace valves include solenoids, piston-cylinders, bellows, and diaphragms. Rotary actuators include electric motors, and servo torque motors. Rotary actuators can also be used to impart linear motion through, for example, a rack and pinion. Linear actuators can be used to impart rotary motion by driving an internally threaded valve stem attachment with a threaded rotating shaft.

The actuator may be either an integral part of the valve, or a separate device mechanically linked to the valve. Solenoids, diaphragms, bellows, piston-cylinders and servo torque motors are usually an integral part of the valve. Electric motors are commonly linked to the valve through a gear train, screw drive, etc.

Valve actuator positioning requirements can be broadly divided into two groups, (1) two position or non-modulating, actuators used for ON-OFF control valves and shutoff valves; and (2) modulating actuators used for positioning control valves. Control valve pneumatic and hydraulic piston-cylinder actuators are commonly used with valve positioners, which provide a feedback loop between the control signal and the stroke of the actuator. Positioners are discussed under "Control Valves" in Sub-Topic 5.36. Bellows and diaphragms, which are used for pressure sensing as well as actuation, are discussed in Sub-Sections 6.6 and 6.7, respectively. The use of explosive charges as a source of pneumatic pressure for valve actuation is discussed in Sub-Section 5.7. The use of servo valve torque motor actuators is discussed in Sub-Topic 5.6.6.

6.9.2 DESIGN AND SELECTION FACTORS

The design or selection of a valve actuator is at best a trade-off among several interrelated factors. Typical mechanical factors which must be considered in the design and selection of a valve actuator include length of stroke desired, locking requirements, speed required, limitation of envelope size and weight, and magnitude of the force which must be provided. If long stroke is required, a solenoid is automatically eliminated. If fast response times are desired, an electric motor, solenoid, or explosive charge should be considered. If high forces must be overcome, either hydraulic or pneumatic pressure must be used. The following detailed topics provide information on the various parameters that must be considered in the design and selection of a valve actuator.

6.9.2.1 POWER SOURCES. The choice of an actuator is limited by readily available power sources. Solenoids, electric motors, and servo torque motors can be used with electric power. Piston-cylinders, diaphragms, and bellows might be selected as the actuator with hydraulic and pneumatic pressure. In a rocket system, power sources available might include high pressure gas used for propellant tank pressurization systems, and hydraulic pressure used for thrust chamber gimballing. Other power sources available include 28 VDC electric power, sometimes converted to AC, and hydraulic pressure of the propellant from the propellant feed system downstream of the turbopump. Also to be considered is the use of hot gas either from gas generators or from thrust chamber bleed. A "one shot" actuator requiring a low power input is an explosive squib. Explosive actuators are described in Sub-Section 5.7. In space vehicles, stored pneumatic pressure, 28 VDC electric power or capsulated explosive charges are more predominantly used for valve actuation than any type of hydraulic fluid.

6.9.2.2 STROKE. The stroke of an actuator is measured in terms of length of travel in the case of linear actuators, or angle of rotation in the case of rotary actuators. Typical strokes for linear valve actuators vary from a few thousandths of an inch (solenoids) to several inches (piston-cylinders). Strokes for rotary actuators vary from angles of approximately 5° (servo torque motors) to several revolutions (electric or hydraulic motors). All actuators, except solenoids, can have unlimited stroke if made large enough. The solenoid is limited in stroke because the movable plunger and the stationary core within the solenoid coil cannot be separated by a large air gap. The greater the air gap, the lesser the effect of the magnetic field and, consequently, the lesser the actuating force. In general, AC solenoids can be used for longer stroke application than the DC type (Reference 19-203). Typical linear solenoid strokes range from 0.010 to 5 inches. The stroke of rotary solenoids normally range from 5° (or less) to 95°.

6.9.2.3 LOCKING. The ability of an actuator to hold a constant position independent of varying forces impressed upon it is called the locking ability. The best type of actuators for maintaining constant position are actuators with screw outputs and piston cylinder actuators utilizing relatively incompressible fluids (e.g. hydraulic oil). The locking ability of hydraulic actuators is called actuator stiffness. For a varying actuator load, air cylinders and electric motors require mechanical locking devices for accurate positioning. Actuator locking mechanisms include toggles or linkages, and pin and ball locks which are spring or piston actuated. Solenoid actuators can be locked in the closed air gap position by the use of a second coil either to hold the plunger (holding coil) or to disengage a locking mechanism such as a detent.

6.9.2.4 SPEED. Speed of an actuator should be considered in terms of initial response, total response time, acceleration, speed, uniformity, and ease of speed adjustment.

The solenoid actuator and the electric motor have higher speeds than other types of actuators. The electric motor actuator has the highest initial acceleration and the best uniformity of response. The uniformity of response is best with induction motors because constant force is produced at every position along its entire length of stroke. A solenoid also responds instantly but the uniformity of response is poor, ranking only slightly above the piston-cylinder actuators. Piston-cylinder actuators and solenoids have the poorest uniformity because the maximum acceleration is not reached until the end of the stroke. The initial response of piston-cylinders is limited by the time lapses in the system valving and/or transmission system. Hydraulic cylinders generally have a faster initial response than pneumatic cylinders. The total response time, particularly for relatively long strokes, is commonly shorter for pneumatic cylinders than for hydraulic cylinders.

The speed of piston-cylinder actuators is easiest to adjust, followed in order by solenoid and electric motor actuators.

6.9.2.5 FORCE OUTPUT. A valve actuator must provide sufficient force to overcome the fluid pressure force, the weight and inertia of the valving element, friction forces, and return spring load (in the case of a single acting actuator). The pressure load is due to the pressure differential acting across the effective area of the valving element. The weight load exists where the unbalanced weight of moving parts must be supported by the actuator, and the inertial load exists due to acceleration of moving parts. Friction forces include viscous friction loads which exist in well-lubricated actuators such as hydraulic cylinders, and static friction loads which are apparent at start-up.

The output force of linear solenoid actuators usually increases with the retracting stroke that closes the air gap between the plunger and the stationary core. The output force of rotary solenoids behaves in a similar manner to linear solenoids except for the helical rotary DC solenoid, which has an almost constant output throughout the stroke.

The electrical induction motor maintains a constant output force if the applied voltage is constant. Fluid cylinders will have a constant output force if the pressure in the cylinder and the effective area are constant.

For electrical actuators, a typical force range is from 0 to 30 pounds. AC solenoids are generally limited to a force of 50 pounds, while some DC solenoids can generate 10,000 pounds of force. Some AC solenoids have been built to exert 100 pounds of force, but these are exceptional cases. For hydraulic cylinders, forces range from 0 to 15,000 pounds with some capable of forces as high as 40,000 pounds. The output force of pneumatic cylinders ranges from 0 to 1500 pounds.

6.9.2.6 RELATIVE SIZE. Relative size for actuators is based on the total energy that can be obtained per actuator volume. The total energy for linear actuation is the output force times the stroke; for rotary actuators, output torque times the angle of rotation (in radians).

High pressure hydraulic and pneumatic cylinders provide the highest ratio of total energy-to-volume. Solenoids and linear motors have a low force-to-volume ratio but require less auxiliary equipment than cylinder actuation. Solenoids require batteries, while cylinder actuators require external pumps, accumulators, and fluid lines. The AC solenoid is usually bulkier than the DC solenoid (Reference 19-203).

6.9.2.7 REPEATABILITY. The ability of the actuator to utilize the same power to produce the same force, speed, and stroke, is called repeatability. Electrical motors have the best repeatability because they have few critical parts and exhibit constant force at every position. Solenoids exhibit good repeatability if the load and ambient temperature are constant. AC solenoids are less repeatable than DC solenoids because of the heat generation and corresponding temperature variation due to hysteresis and eddy currents.

Repeatability of piston-cylinder actuators is difficult to achieve because the fluids are very sensitive to environmental temperature changes, and the pressure drops through valving and connecting lines do not vary linearly with flow rate.

6.9.2.8 ENVIRONMENT. The environments to which a valve actuator can be exposed includes temperature, vacuum, corrosive atmosphere, radiation, vibration, and shock. Environments are discussed in detail in Section 13.0. Temperature variations and extremes will adversely affect performance characteristics such as repeatability and response. With electrical actuators, increases in temperature increase electrical resistance, and also increase the amount of current necessary to obtain a desired output force. Manufacturers usually indicate operating characteristics close to normal room temperature and apply correction factors for different temperatures.

Low temperatures eliminate the use of hydraulic cylinders and oil lubricated screw drives unless heat is supplied. Since high and low temperatures drastically change the characteristics of air cylinders, they should be calibrated at operating temperature.

Special design considerations are required for actuators which must operate under vacuum conditions. Electrical actuators may arc or exhibit a corona discharge if subjected to a partial vacuum. Electrical devices should be encapsulated and supplied with a blanket of helium or nitrogen. Vacuum adversely affects sealing of piston-cylinders, increasing fluid permeation and seal degradation. Screw drives, if not lubricated, will cold weld. For extended periods of vacuum service, solid lubricants should be considered.

Radiation may cause degradation of elastomeric seal materials, electrical insulation materials, and some hydrocarbons in hydraulic fluids.

In general, rotary actuators have greater resistance to shock and acceleration than linear actuators because the rotor (in the case of an electric motor) or the armature (in the case of a solenoid) are very closely balanced. In a linear actuator, a counterbalance must be included to eliminate the tenden[illegible] armature to be jarred out of position. Shock [illegible] actuators can result from a sudden press[illegible] used by the load reacting against th[illegible]

6[illegible] DE[illegible]

6.9.3.1 [illegible] PISTON-CYLINDER ACTUATORS [illegible] provide either linear or rotary [illegible] vide long operating strokes and [illegible] This Detailed Topic will discuss, primarily, the linea[illegible] linder actuator. Linear actuators can be grouped [illegible] piston and rod type, by body types, and by mounting. Table 6.9.3.1 presents several piston-rod types.

Numerous techniques can be used to convert linear piston motion into rotary motion in rotary-cylinder actuators.

A helical spline rotary piston-cylinder actuator is illustrated in Figure 6.9.3.1a. As the piston is prevented from rotating by the guide rod, the shaft rotates as the piston moves. Figure 6.9.3.1b illustrates a double helical spline actuator which eliminates shaft thrust loads by employing left and right hand threads.

Although the most common technique for achieving rotary motion for ball and butterfly valves is by using a linear actuator shaft as a rack driving an external pinion connected to the valve shaft, there are piston-rack rotary actuators having the pinion located within the actuator cylinder. The design illustrated in Figure 6.9.3.1c uses two piston-racks. As pressure moves the piston-rack, the pinion shaft (90° to line of piston motion) rotates. Maximum cylinder pressure for helical-spline actuators is approximately 350 psi, while for the piston-rack-pinion actuator cylinder pressure ranges from 1000 to 3000 psi. Several variations of these actuators use double chambers and pistons which are discussed in Reference 1-261. References 1-67 and 1-81 also discuss rotary piston-cylinders.

Rod Bearing to Reduce Rod Sagging. When the piston rod is fully extended, only the length of the piston and the rod bearing can resist any turning moment due to the weight of the rod or side loads. Deflection of the rod may cause binding on the return stroke. Although there are no set dimensional standards for clearance allotment, one general rule for piston and cylinder actuators with a single rod bearing is

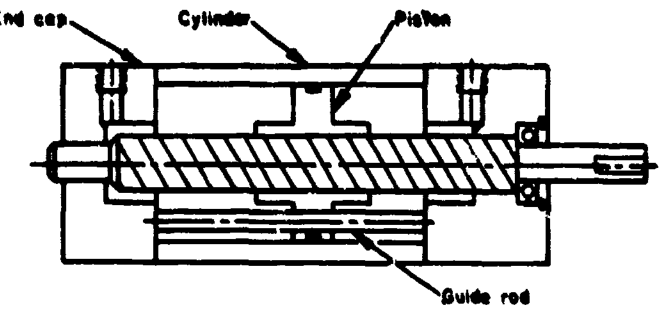


Figure 6.9.3.1a. Helical-Spline Rotary Piston-Cylinder Actuator
(From "Machine Design," C. R. Johnson, Copyright 1962, Penton Publishing Company, Cleveland, Ohio)

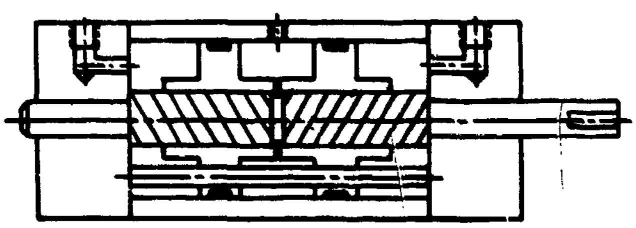


Figure 6.9.3.1b. Double Helical-Spline Rotary Piston-Cylinder Actuator
(From "Machine Design," C. R. Johnson, Copyright 1962, Penton Publishing Company, Cleveland, Ohio)

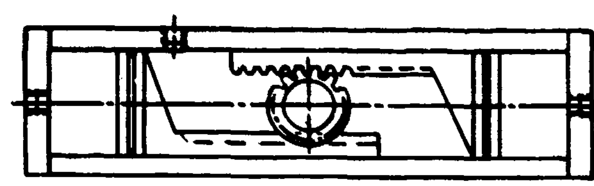
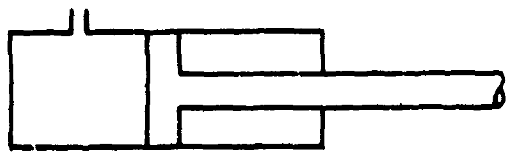
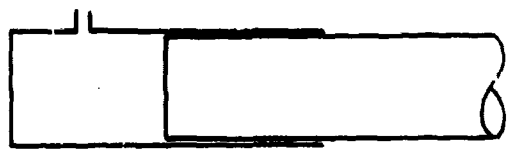
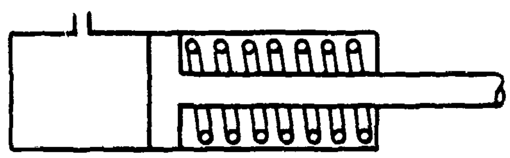
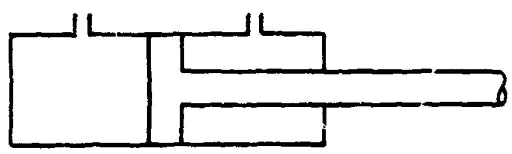
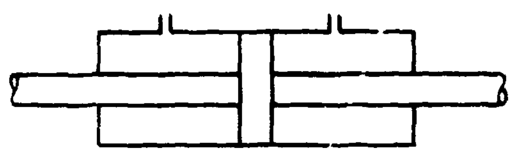
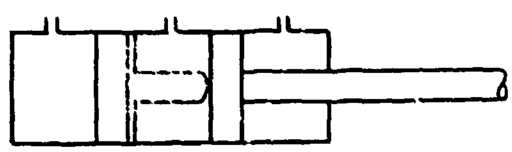


Figure 6.9.3.1c. Piston-Rack Rotary Piston-Cylinder Actuator
(From "Machine Design," C. R. Johnson, Copyright 1962, Penton Publishing Company, Cleveland, Ohio)

Table 6.9.3.1. Piston-Rod Types
(Reference 1-261)

	**Single-Acting** Power stroke in one direction only. This can be either the out stroke or in stroke, return stroke being accomplished by some external means. A double-acting cylinder can be used for this purpose by connecting the actuating fluid line to only one port through a 3-way valve, leaving the other port open. Special single-acting cylinders are designed with piston-sealing devices which seal in one direction only. These cylinders have a port hole in one head and a bleeder hole in the opposite head.
	**Ram** Single-acting cylinder with rod diameter equal to the effective piston diameter, which is equal to the bore of the cylinder. Only one packing, the rod packing, is required.
	**Spring-Return** Single-acting cylinder with return stroke effected by a spring. Length of cylinder, in retracted position, is at least twice actual stroke-length because of spring length. Initial spring force, and increase in spring force due to spring rate during compression, depends on the amount of force required by the spring-actuated stroke.
	**Double-Acting** Cylinder with power stroke in both directions. Actuating fluid line is connected to both heads of the cylinder, usually through a 4-way valve. Most standard catalog cylinders are double acting. Sealing devices operate in both directions.
	**Double-End Rod** Cylinder where rod extends from both sides of piston. Usually double-acting, such a cylinder requires a rod bearing and packing in each cylinder head.
	**Positional** Stroke is split up into two or more portions. Cylinder can be actuated to any one of the positions.

$$L_P + L_B + L_{P\text{-}B} \begin{cases} > 3d_r \\ \geq d_c \end{cases} \quad \text{(Eq 6.9.3.1a)}$$

where
L_P = length of piston land
L_B = length of rod bearing
$L_{P\text{-}B}$ = length of clearance between piston and bearing
d_r = rod diameter
d_c = cylinder internal diameter

The length of clearance between piston and bearing can be made a significant part of the total length by using inserts within the cylinder, preventing the piston from stroking all the way to the cylinder head.

Determining Rod Diameter. To determine the rod diameter for tensile stress only, the rod diameter must be

$$D = \sqrt{\frac{4P}{\pi\sigma}} \quad \text{(Eq 6.9.3.2b)}$$

where
P = load, lb_f
σ = stress, psi
D = diameter, in.

If the stroke is so long that sagging could occur, a tubular rod is often used. If the rod is loaded in compression, the rod is treated as a column which may be subject to compressive stresses. The maximum allowable unit stress is approximated by

(Eq 6.9.3.1c)

$$\sigma_{max} = \frac{\left(\frac{P}{A}\right)}{1 - \left(\frac{P}{A}\right)\left(\frac{L}{k}\right)^2\left(\frac{1}{\pi^2 E}\right)}$$

Equation (6.9.3.1c) is accurate for slenderness ratios from 20 to 200. For L/k values greater than 200, Euler's equation should be used:

$$\sigma_{max} = \frac{\pi^2 E}{(C\ L/k)^2} \quad \text{(Eq 6.9.3.1d)}$$

where
σ_{max} = maximum stress, psi
P = load, lb_f
A = cross-sectional area, in²
E = modulus of elasticity, psi
L/k = slenderness ratio (length of unsupported piston rod/least radius of gyration)
C = fixity factor which has the following values:

both ends fixed — 0.5
one end fixed, one end pinned — 0.7
both ends pinned — 1.0
one end fixed, one end free — 2.0

Determining the Cylinder Pressure Rating. The maximum pressure set for a piston-cylinder actuator is below the burst pressure of the cylinder, and is usually limited by the ability of the seals to continue to seal when clearances have increased due to deformation of the cylinder walls. For stress analysis, the cylinders are considered thick-walled cylinders. The maximum stress occurs at the inside surface of the cylinder and can be resolved into three components:

The *meridional stress*, σ_M, caused by the pressure on the end caps.

$$\sigma_M = \frac{r_1^2 P}{r_2^2 - r_1^2} \quad \text{(Eq 6.9.3.1e)}$$

The *circumferential stress*, σ_C, is given for any point in the cylinder wall by:

(Eq 6.9.3.1f)

$$\sigma_C = \frac{r_1^2 P}{r_2^2 - r_1^2}(1 + r_2^2/r^2)$$

The *radial stress*, σ_r, acting in compression from the center line, is at any point in the cylinder wall

(Eq 6.9.3.1g)

$$\sigma_r = \frac{-r_1^2 P}{r_2^2 - r_1^2}\left(1 - \frac{r_2^2}{r^2}\right)$$

The *radial expansion* resulting from these stresses is

(Eq 6.9.3.1h)

$$\Delta r_1 = \frac{r_1 P}{E}\left\{\frac{1 + r_1^2/r_2^2}{1 - r_1^2/r_2^2} + \nu\right\}$$

where
P = total load, lb
r_1 = inside radius, in.
r_2 = outside radius, in.
r = radius at any point, in.
E = modulus of elasticity, psi
σ = stress, psi
ν = Poisson's ratio

Thickness (t) of Cylinder Ends. The thickness of cylinder ends is determined from the permissible working stress of the material and the shape of the end caps. For flat bottom end caps, an approximation is (Reference 135-9):

$$t = 0.405\, d\sqrt{\frac{P}{\sigma}} \quad \text{(Eq 6.9.3.1i)}$$

For spherical bottom end caps, an approximate equation is (Reference 135-9):

$$t = \frac{Pd}{4\sigma} \quad \text{(Eq 6.9.3.1j)}$$

where t = thickness of end cap, in.
P = internal pressure, psi
d = cylinder diameter, in.
σ = permissible working stress, psi

Rod Packings. There are four basic types of piston rod packings: V-ring, flange, U-cups, and O-ring (Figure 6.9.3.1d). Of these basic designs, many variations can exist. The V-packings are considered the best type of seal for general hydraulic use, and are also excellent for pneumatic service. Flange-type packings are fairly good for pneumatic use; they take little space and are often essential for certain cylinder designs based on space limitations. U-cup packings are similar to the flange type except that there are two sealing lips. One lip seals the OD of the gland, and the other seals against the piston rod. This seal is used for pneumatic systems and low pressure hydraulic systems. With anti-extrusion back up rings, O-ring seals are excellent for positive containment with either pneumatic or hydraulic systems. They are good for very high pressures, but have a relatively short wear life. Some of the commonly used rod packing materials are Buna N, Neoprene, Teflon, and various combinations such as cotton fabric reinforced with Neoprene or Buna N. Buna N is the most common O-ring material for pneumatic and hydraulic actuators. Viton A is common for high temperature applications and for use with the Phosphate-ester base synthetic hydraulic fluids.

Cylinders which must operate in environments where dirt and grit can accumulate on the exposed cylinder rod should employ rod wipers or scrapers with all types of rod packings to keep harmful abrasive foreign materials from being drawn into the packings (Reference 1-261). Techniques of O-ring use in linear reciprocating pistons are presented in References 6-127 and 6-182.

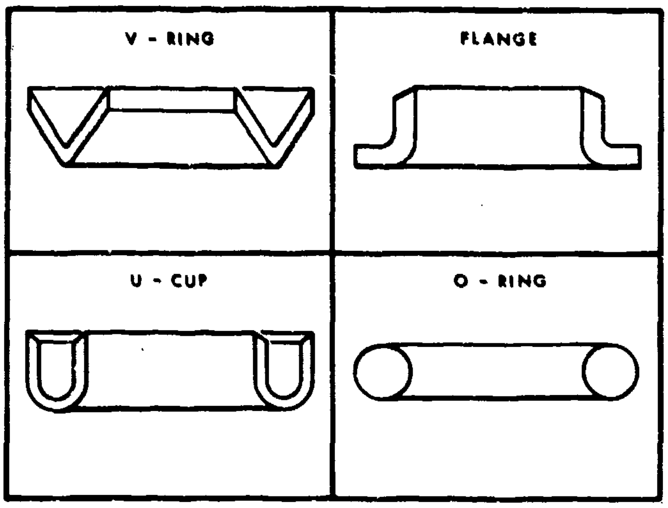


Figure 6.9.3.1d. Types of Piston Packings
(From "Machine Design," T. P. Watson, Copyright 1962, Penton Publishing Company, Cleveland, Ohio)

Cushioning or Snubbing. A cushioning device provides absorption of piston energy at the end of the stroke. In some systems, impact of a piston going against the stop, or impact transmitted to cylinders as shock waves through the fluid itself, can raise internal cylinder pressures to levels of 2 to 3 times normal system pressure and cause cylinder damage. An efficient cushion will remove energy at the end of the stroke and will act over sufficient length of the stroke so that peak force and pressure are reduced sufficiently to prevent damage.

Cushioning is usually done by trapping the exhaust fluid as the piston approaches the end of its stroke. In Figure 6.9.3.1e, as a piston nears the end of a stroke a plunger enters a cushioning chamber and restricts flow of fluid from the exhaust port. The trapped fluid forms a back pressure, thus causing the piston to slow down as it approaches the head. Sometimes the device is designed so that the cushioning action can be regulated by an adjustable orifice. This is particularly true with pneumatic cylinders for speed control. Hydraulic cushion plungers are usually tapered to allow gradual shutoff of the exhausting oil, because of the relative incompressibility of oil. Usually they also have an adjustable orifice. Although non-adjustable hydraulic cushions are quite successful in some applications, some type of check valve is necessary if a free return stroke is desired.

If the cushioning air of pneumatic cylinders is not completely exhausted by the time the maximum stroke is reached, the air will be under higher pressure than actuating air and may cause reverse action, called bounce. Actuating pressure reverses the piston which will bounce back against the cushioning air (Reference 1-261).

Locking. Factors to be considered in selection of a piston and cylinder locking mechanism are weight and envelope size, temperature changes of locked-in fluid, external loads on pin sections, bearing stress on latching balls, the responce for sequencing, and the expense incurred for complicated mechanisms. See Reference 19-36 for a discussion of various types of locking mechanisms used to lock a piston under load to a cylinder.

6.9.3.2 PILOTED PISTON-CYLINDERS. Piston-cylinder valve actuators are commonly piloted, using either an electrical, low pressure, and/or low flow signal to control the pressure differential across the main power piston-cylinder. Four of the most common piloted valve actuators are the pneumatic-pneumatic, electropneumatic, pneumatic-hydraulic, and electrohydraulic. The first term of the actuator title refers to the control or pilot signal, and the second term refers to the main power fluid. Each is explained separately under the following Detailed Topics.

6.9.3.3 PNEUMATIC-PNEUMATIC ACTUATORS. The pneumatic-pneumatic actuator is a piloted pneumatic actuator which receives a pneumatic control signal and utilizes pneumatic power across a differential piston area (Figure 6.9.3.3). Pneumatic supply pressure is impressed on the small effective area side (top) of the piston. Flow of supply gas to the under side of the actuator is controlled

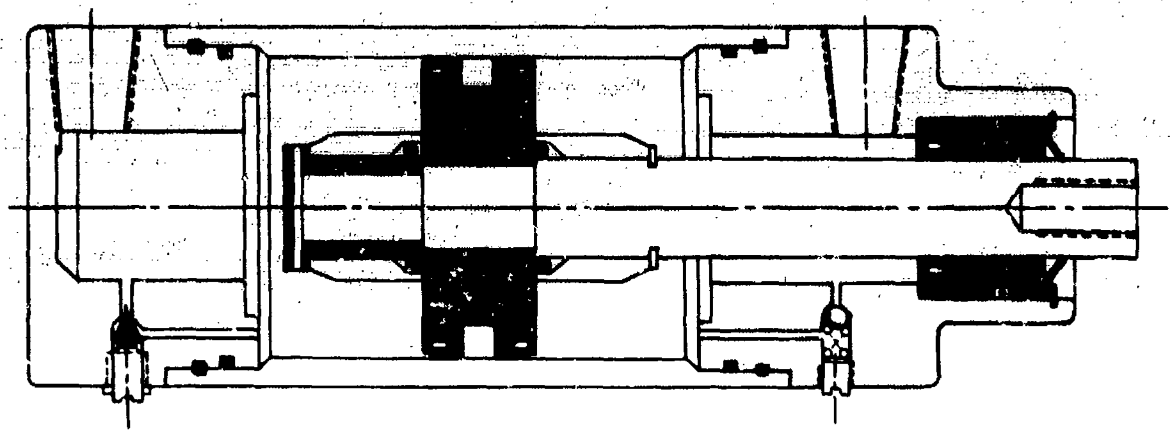


Figure 6.9.3.1e. A Method of Cushioning
(From "Machine Design," T. P. Watson, Copyright 1962, Penton Publishing Company, Cleveland, Ohio)

by the pilot valve. An increase in the pneumatic control signal forces the positioner diaphragm down against the positioner rate spring opening the pilot valve and admitting supply gas to the underside of the actuator piston. The piston moves upward until the force transmitted through the compressed spring equals the force exerted by the signal pressure on the positioner diaphragm.

6.9.3.4 ELECTRO-PNEUMATIC ACTUATORS. The electro-pneumatic actuator, Figure 6.9.3.4, converts a direct current signal to a pneumatic output pressure. Supply pressure is connected to the inlet of the electro-pneumatic transducer and to the pilot valve of the actuator. Changes in electrical signal actuate a lever system which positions a valving element in the transducer. The valving element throttles the supply gas pressure to a corresponding signal pressure. The actuator from this point functions similarly to the pneumatic-pneumatic cylinder, with the exception that the supply pressure is regulated to a preset level thereby maintaining a constant loading pressure on the top of the actuator piston. An orifice diphragm in the regulator provides a port to relieve loading pressure when the piston strokes upward (Reference V-8).

6.9.3.5 PNEUMATIC-HYDRAULIC ACTUATORS. The pneumatic-hydraulic actuator (Figure 6.9.3.5) uses a pneumatic-pneumatic actuator to position a four-way spool valve which controls the flow to a hydraulic piston-cylinder. The pneumatic-pneumatic actuator is described in Detailed Topic 6.9.3.3. On an increase of signal pressure to the pneumatic-pneumatic actuator, the small cylinder rod of the pneumatic cylinder retracts, causing the spring loaded four-way hydraulic spool to move downward. The hydraulic servo valve would then be ported as follows: the lower port of the hydraulic piston would be ported to the return line, and the upper port of the hydraulic servo valve would be connected to the supply side of the hydraulic actuating medium. This extends the main cylinder rod until the motion of the follow-up rod drives the spool of the servo valve to a balanced position. A balanced position can be reached only when the hydraulic cylinder rod is driven to the position which corresponds exactly with the pneumatic signal being applied to the pneumatic-pneumatic actuator (Reference V-8).

6.9.3.6 ELECTROHYDRAULIC ACTUATORS. The electrohydraulic actuator (Figure 6.9.3.6) is a typical actuator used to control ball and butterfly shutoff valves for flight propulsion systems. The three-way solenoid pilot valve is energized by an electrical input signal. Hydraulic supply pressure is thereby provided to the actuator piston, and the hydraulic fluid acting on the piston extends the piston rod compressing the return spring. The return stroke is accomplished by de-energizing the pilot valve, allowing the spring to force hydraulic fluid out the return line.

6.9.3.7 SOLENOIDS. A solenoid is an electromagnetic device consisting of a coil and a magnetic path. The coil is usually wound on a non-magnetic tube, commonly stainless steel, which does not influence the magnetic flux path. The coils are well insulated (silicone being widely used) and sealed against moisture. The magnetic flux path is composed of a stationary core (or stator), a plunger (sometimes called an armature), and the coil housing (or a structure of pole pieces and saddle plates) made from some magnetic material. *Magnetic material* describes materials which will readily form a magnetic flux path and are attracted by a magnetic tractive force, whereas *magnet* or *magnet material* describes material in which a strong permanent magnetic field may be induced, as typified by specific Alnico alloys. The flux path through a typical AC solenoid is shown in Figure 6.9.3.7a.

A solenoid derives its output force from the lines of magnetic flux between the stationary core and plunger. The flux density is generated by the coil and depends on the number of ampere-turns. The amount of flux is normal to the direction of flow of the magnetic circuit, and the density of flux is inversely proportionally to the areas of the respective sections. The flux density is inversely proportional to the air gap between the plunger and stationary core, and is directly proportional to the ampere-turns.

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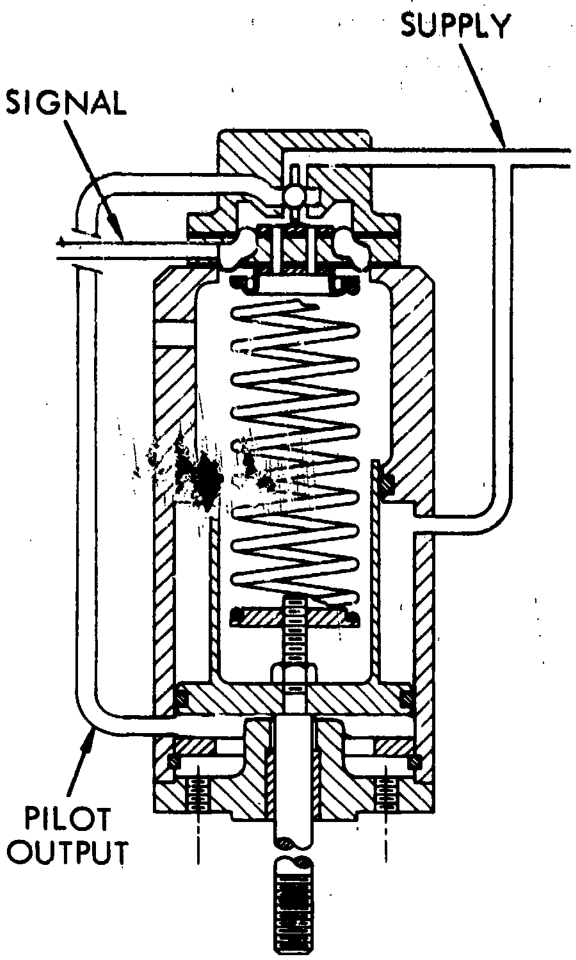

Figure 6.9.3.3. Pneumatic-Pneumatic Actuator
(Courtesy of the Annin Company, Montebello, California)

$$B = 3.2 \frac{NI}{L} \qquad \textbf{(Eq 6.9.3.7a)}$$

where B = flux density, maxwells/in^2
NI = ampere-turns, amperes
L = air gap length, in.

Through the air gap between the plunger and stationary core, the magnetic flux causes a force which varies directly with the square of the flux and inversely with the area of the flux stream; namely,

$$F = \frac{B^2 A}{72} \qquad \textbf{(Eq 6.9.3.7b)}$$

where F = magnetic force, lb_f
B = flux density, kilomaxwells/in^2
A = flux path area, in^2

The above equations are only true when flux density is below the saturation of the magnetic material used. It is necessary to make all areas in the magnetic circuit large enough to prevent saturation defects. Information on the use of a computer to predict solenoid performance can be found in Reference 19-211. The design of a solenoid by computer methods is described in Detailed Topic 8.3.3.1.

The ampere-turns generated by a coil depends on three variables: (1) mean diameter of coil, (2) diameter of coil wires, and (3) resistance per unit length of wire (Reference 19-102). The mean diameter of the coil is usually made as small as possible to reduce weight and still give the desired work output, but increasing the wire diameter increases the ampere-turns value of the coil. Resistivity is inversely proportional to the cross-sectional area of the wire and directly proportional to temperature. Work output decreases as resistivity increases.

External housings not only complete a magnetic path but also act as explosion-proof containers and mechanical protectors (Reference 112-3). In valves, the solenoid plunger may or may not have a packing and packing gland surrounding it. Without a packing (called packless valves) the fluid fills the coil tube and immerses the plunger. Packed valves are sometimes used with corrosive liquids and gases but require larger solenoids to overcome frictional forces exerted by the packing. The terms *wet coil* and *dry coil*, respectively, are commonly used to describe solenoid plungers either immersed in or sealed from the fluid.

Types of Solenoids. Solenoids can be made for both linear or rotary motion. Linear, or push-pull solenoids can be either plunger (Figure 6.9.3.7a) or clapper type (Figure 6.9.3.7b). The plunger solenoid pulls an armature into the center of the coil when the coil is energized. In the clapper solenoid the armature, which is usually hinged, is pulled *against* the core rather than into it. The plunger solenoid is used for longer strokes than the clapper, but still has an excellent short stroke capability. The clapper solenoid compared to the plunger type has a smaller envelope, shorter stroke capability, and twice the force output per given stroke. Force output versus stroke for plunger and clapper type solenoids is illustrated in Figure 6.9.3.7c.

Rotary solenoids can be either pure rotary (motor) solenoids or mechanical (also called helical) solenoids (linear displacement converted mechanically into rotation) (Figure 6.9.3.7d). The motor type solenoid uses a distorted magnetic field to rotate the armature. When the coil is energized, the magnetic field causes the rotor to pivot without axial shift. The mechanical solenoid uses an inclined plane to convert linear motion into rotary motion. One type (Reference V-56) has an armature supported by three precision ball bearings. As the armature is pulled into the coil, the bearings move along an inclined plane, causing a twisting motion. The inclined plane is steep at the start of the stroke but levels off at the end of the stroke. Approximately uniform torque is obtained throughout the stroke. By varying the configuration of the ball races, variable torque outputs can be achieved.

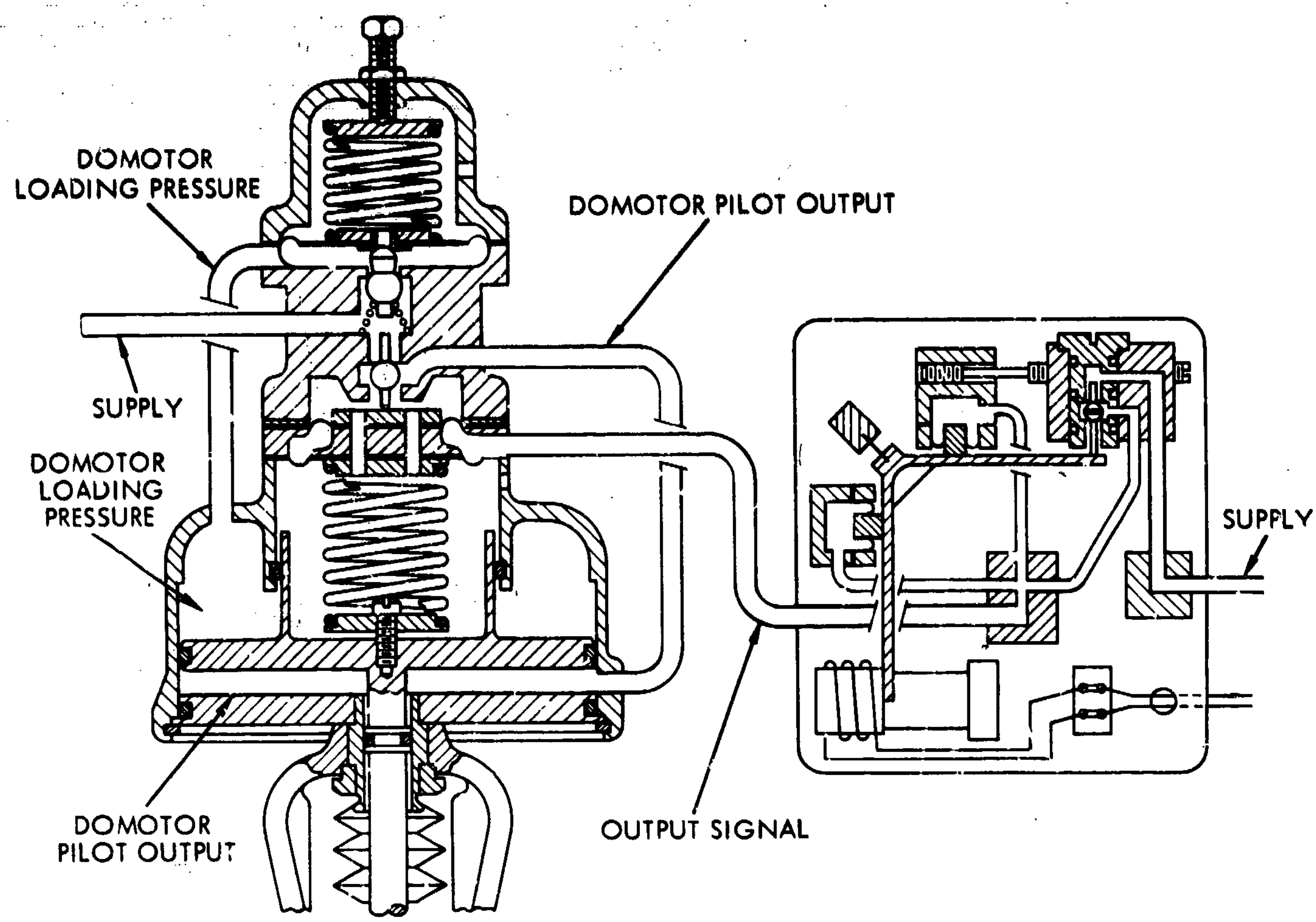

Figure 6.9.3.4. Electro-Pneumatic Actuator
(Courtesy of the Annin Company, Montebello, California)

The motor type solenoid produces no axial motion, while the helical type moves a few thousandths of an inch depending on the solenoid size.

A special combination of the plunger and clapper solenoids, called the pivotal solenoid, produces rotary motion (Figure 6.9.3.7e). The angle of rotation, however, is limited with the design (Reference 19-203).

AC Versus DC Solenoids. Solenoid valves may use either direct or alternating current. AC solenoids are available which use 6, 24, 115, 230, and 440 single phase AC; and DC solenoids are available which use 6, 24, and 115 DC. When direct current is used, the current is the same regardless of the position of the armature. With alternating current the electric current is higher when the air gap is large, but gradually diminishes as the air gap decreases due to increase in coil inductance. This aspect becomes particularly important when the armature or valve stem sticks, allowing a large air gap for a long time duration which may cause coil burn-out.

With direct current solenoids, the core or stator can be a solid or homogeneous piece of magnetic material, but with alternating current, the core will either be laminated or deeply slotted to reduce the circulating eddy currents. Eddy currents cause the core to become excessively hot and could cause coil burnout.

DC solenoids are quiet in operation, whereas AC solenoids may hum or chatter. As alternating current passes through a zero value twice each cycle, the magnetic flux does also. As the current passes through zero, the force holding the plunger or armature becomes zero, causing the plunger to fall away, inducing chattering. To prevent chattering, a shading ring (Figure 6.9.3.7a) is used. It consists of a ring of a low resistance material (e.g., brass, copper, silver)

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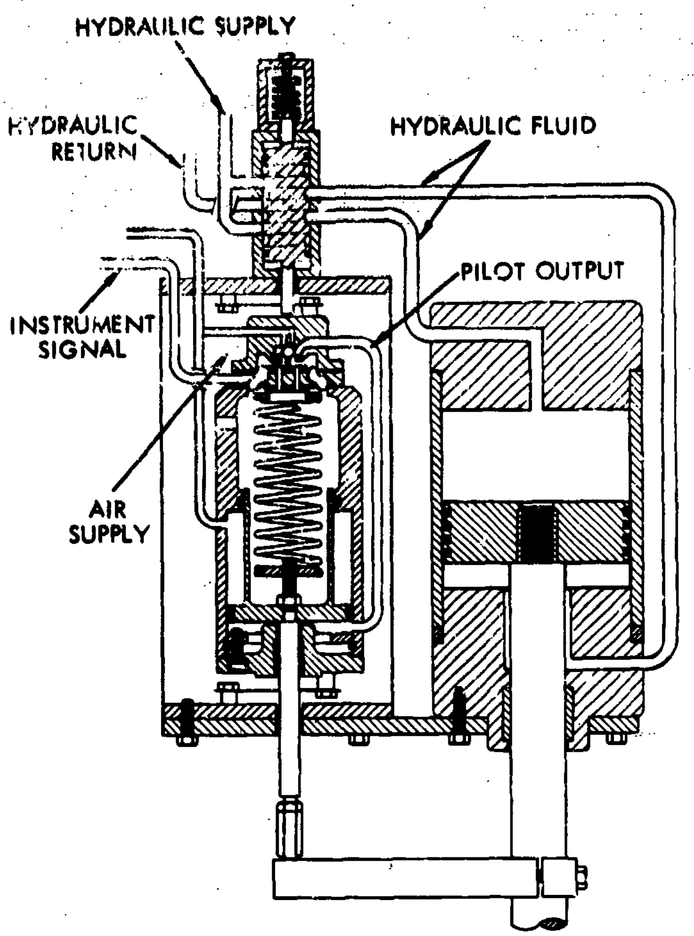

Figure 6.9.3.5. Pneumatic-Hydraulic Actuator
(Courtesy of the Annin Company, Montebello, California)

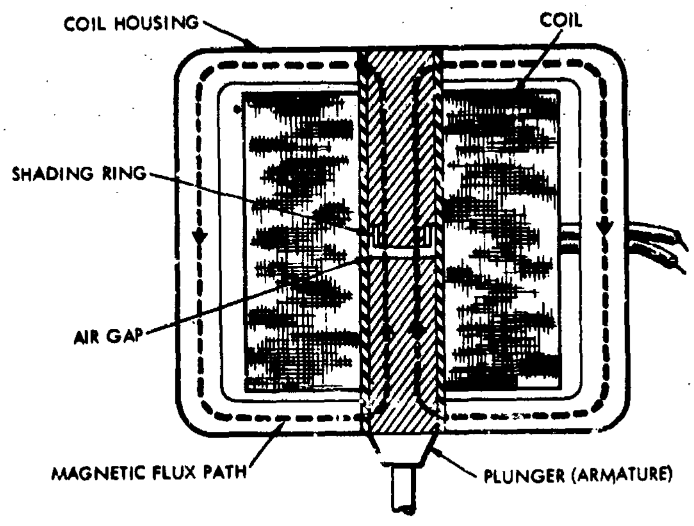

Figure 6.9.3.7a. Flux Path of a Plunger Type Solenoid

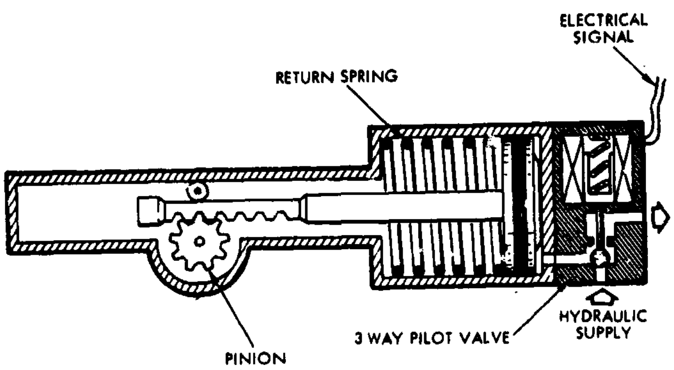

Figure 6.9.3.6. Electrohydraulic Actuator

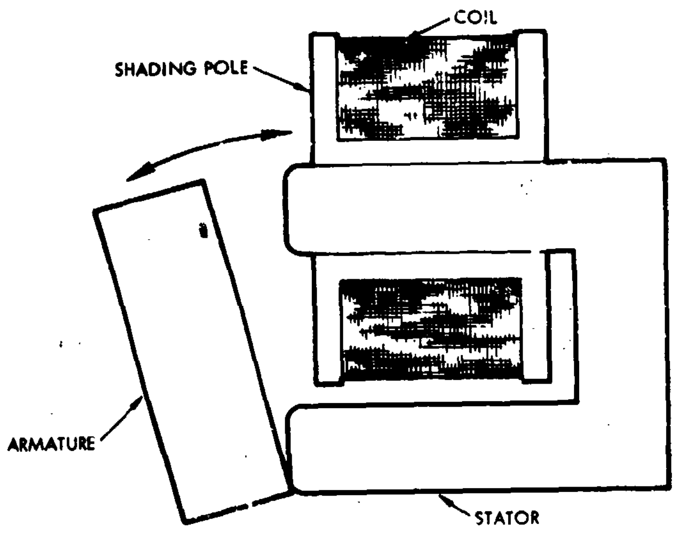

Figure 6.9.3.7b. Clapper Type Solenoid
(Reprinted from "Product Engineering," E. A. Scutari, May 23, 1960, Copyright 1960 McGraw-Hill Publishing Company, Inc.)

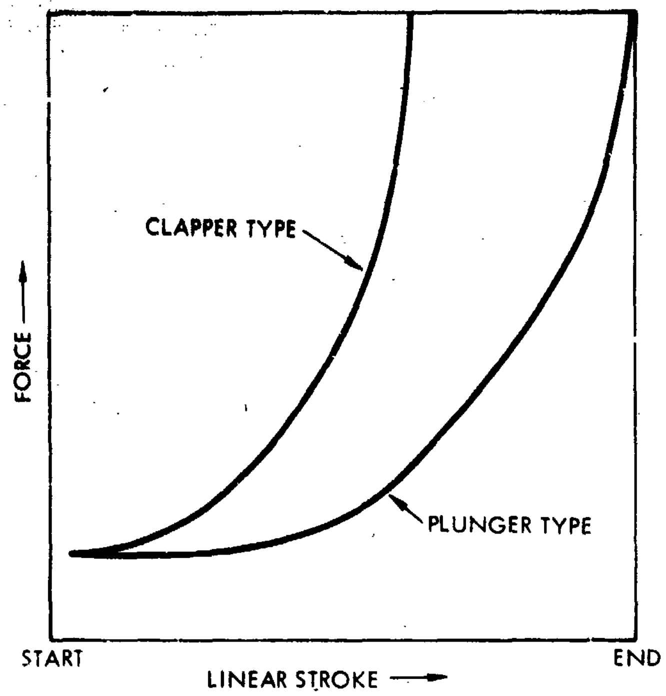

Figure 6.9.3.7c. Force Versus Stroke for Push-Pull Solenoids
(Reprinted from "Product Engineering," E. A. Scutari, May 23, 1960, Copyright 1960 McGraw-Hill Publishing Company, Inc.)

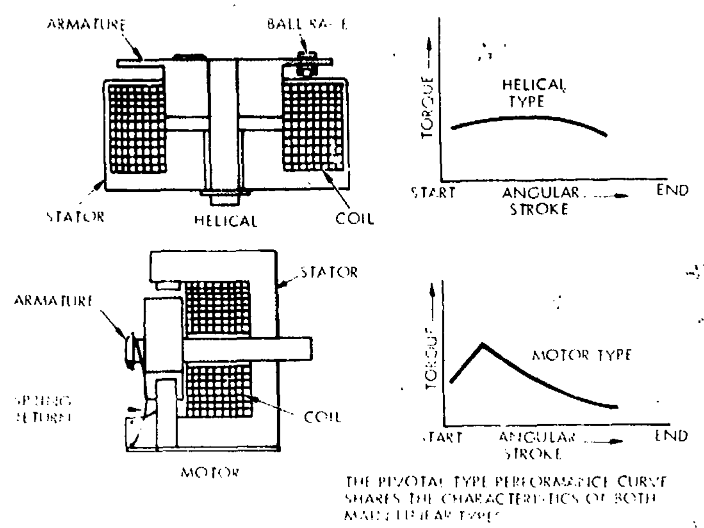

Figure 6.9.3.7d. Rotary Solenoids
(Reprinted from "Product Engineering," E. A. Scutari, May 23, 1960, Copyright 1960, McGraw-Hill Publishing Company, Inc.)

which is mounted to the stationary magnetic core or to the upper end of the armature. It provides an out-of-phase flux which reaches a maximum near the instant the normal flux becomes zero. Most AC solenoids use shading rings. Chattering may also be avoided by using latching solenoids.

Figure 6.9.3.7f shows the current flow of a DC solenoid actuator. Inductance limits the flow of current during the initial energizing stage. As the current increases, the mag-

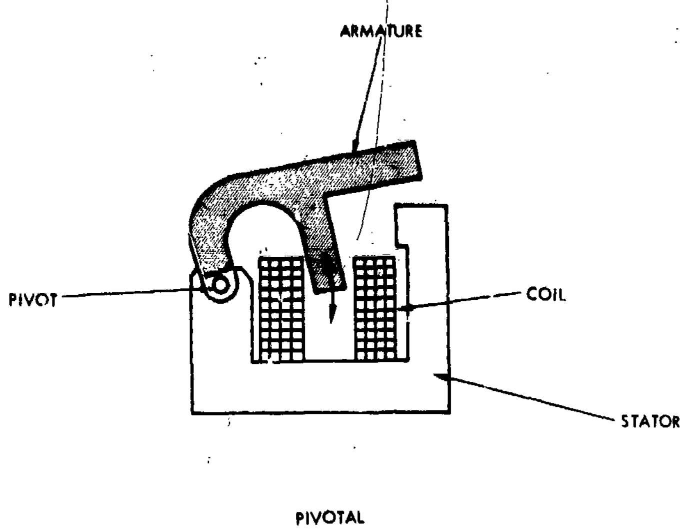

Figure 6.9.3.7e. Pivotal Solenoid
(Reprinted from "Product Engineering," E. A. Scutari, May 23, 1960, Copyright 1960, McGraw-Hill Publishing Company, Inc.)

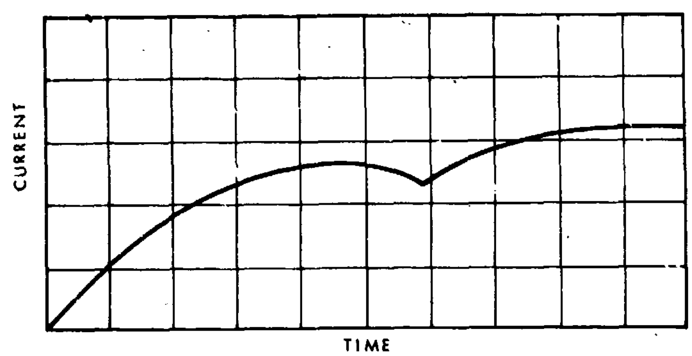

Figure 6.9.3.7f. Flow of Current in a Solenoid Coil
(Courtesy of The IMC Magnetics Corporation, Western Division, Maywood, California)

netic force becomes large enough to initiate movement of the armature. As the armature moves toward its full stroke, a counter emf is created which results in a decrease in current. With the armature seated, the current increases to a steady-state value determined by Ohm's Law.

Response Time. Solenoid response time is the elapsed time interval between the time that voltage is first applied to, or removed from, the solenoid coil and the time that the plunger completes its stroke (either opening or closing). In solenoid valves, valving element response time, during which the valving element (plunger) is actually in motion, is usually a relatively small portion of total solenoid valve response time. This must be taken into consideration when fluid system dynamics such as water hammer are being investigated. Typical solenoid valve response times vary from less than 5 milliseconds to greater than 100 milliseconds, depending on size, stroke, mass, and solenoid force.

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The buildup of current as an implicit function of current and air gap is

$$i = \frac{E - N\left[\frac{\partial\phi}{\partial i}\frac{di}{dt} + \frac{\partial\phi}{\partial x}\frac{dx}{dt}\right]}{R} \qquad \textbf{(Eq 6.9.3.7c)}$$

where i = current, amps
ϕ = flux — $f(x,i)$
R = resistance, ohms
N = coil turns in flux stream

The plunger velocity is $\frac{dx}{dt}$. The response time can be readily determined by the cusp in Figure 6.9.3.7f, which results from a large plunger velocity causing the current to decline below its previously attained value. When the plunger stops, $\left(\frac{dx}{dt} = 0\right)$, the current continues to build up to a steady-stage value (Reference V-47).

For a solenoid valve, the response time to open is (Reference 185-1)

$$t_o = 1.4\left(\frac{W_o d}{F_o g}\right)^{1/2} \qquad \textbf{(Eq 6.9.3.7d)}$$

and the time to close is (Reference 185-1)

$$t_c = 1.4\left(\frac{W_c d}{F_c g}\right)^{1/2} \qquad \textbf{(Eq 6.9.3.7e)}$$

where W_o = weight of armature + spring preload + force of pressure difference on the valving element, lb_f
W_c = weight of armature, lb_f
d = distance of armature travel, in.
F_o = pull-in force of solenoid, lb_f
F_c = spring force + dynamic force of fluid, lb_f
g = 32.2 ft/sec^2

The above equations have given results accurate within $\pm$ 3 milliseconds, neglecting the dynamic force (Reference 185-1).

Locking. Solenoids will lock up only at the closed-air gap position. Locking requires continuous flow of current through the coil and, if the lockup time is very long, the coil will overheat and eventually burnout. To prevent excessive heating, double coil switching solenoids are used.

The winding of the second solenoid, called the latching solenoid, is activated when the armature reaches its seated position. Only low output power from the second solenoid is required to hold the plunger against the coil core. An important consideration in double coil switching solenoids is that the plunger must be allowed to complete its stroke. If the second coil is prevented from actuating, the pull current of the first solenoid will not be interrupted, which could cause the coil or switch contacts to burn out (Reference V-47).

Latching. Solenoids which incorporate mechanical or magnetic features which permit the solenoid to remain in either the open or closed position for an indefinite period of time without drawing current are known as *latching solenoids* or *pulse-latching solenoids*. Simple solenoids or those with secondary latching solenoid coils are locked up when the air gap is closed and require continuous current flow to remain locked up. Where a long continuous on-time is required, the lockup current requirement may be prohibitive, as in many spacecraft applications. In such instances latching solenoids are specified which require only a brief flow of current (in the order of 20 to 100 milliseconds) for actuation. Latching may be accomplished by either mechanical or magnetic means. Mechanical latching is more common in large solenoids (where permanent magnet weight may become prohibitive) and magnetic latching is more common in small valves (especially in spacecraft where mechanical simplicity is desirable).

Mechanical latching may take a variety of forms, such as latching pins, detent balls, Belleville discs, etc. Figure 6.9.3.7g illustrates a pin-type mechanical latching solenoid

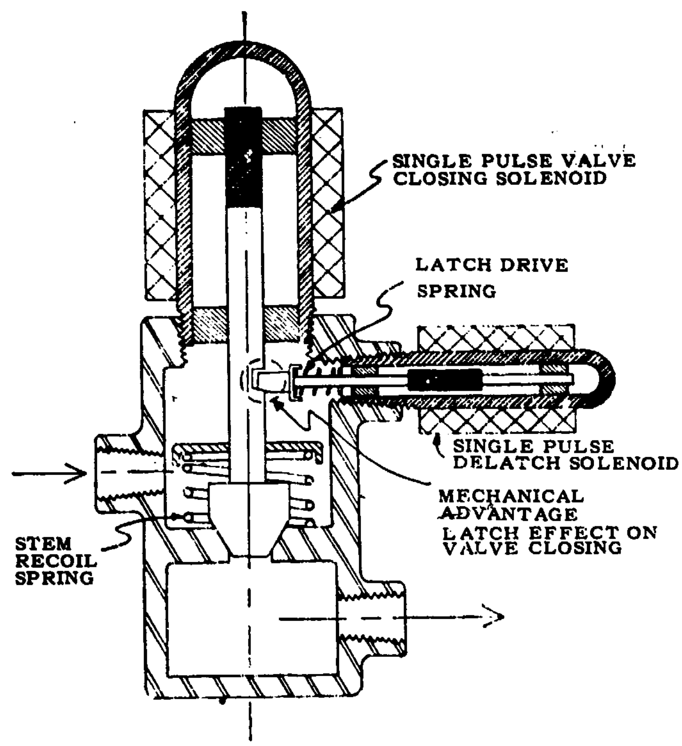


Figure 6.9.3.7g. The Pulse Engagement Method of Latching Solenoids
(From "Instruments and Automation," C. S. Beard, July 1956, Copyright 1956, Instruments Publishing Company)

with a mechanical advantage feature. In this method, voltage applied to a single-pulse solenoid closes the valve, which is latched by a separate spring-loaded plunger made part of an unlatching solenoid. The valve-closing solenoid is automatically disconnected from the power source, eliminating the danger of over-heating. Actuation of the unlatching solenoid disengages the latch and allows the main stem recoil spring to open the valve (References 160-16 and 160-24). A similar arrangement integrated within the solenoid is shown in Figure 6.9.3.7h. The latching solenoid holds the shutoff plunger in the retracted position when coil (1) is energized. As the shutoff plunger retracts, the spring within the latching rod pushes the latching rod to hold balls in the detent groove. When coil (2) is energized, an unlatching plunger moves in the opposite direction to meet the unlatching pin fixed in the latching rod. When the latching rod moves against the spring load, the balls in detent are released and allowed to drop into a groove on the latching rod.

When balls fall out of detent, the shutoff plunger is allowed to extend. The extreme end of the shutoff plunger assembly contacts the position indicator switch to give an electrical signal to a remote location (Reference V-47). Another example of a mechanical latching solenoid which utilizes a Belleville disc for latching is illustrated in Figure 5.5.13a.

Latching of solenoids may also be accomplished without resorting to locking pins, detent balls, etc. by utilizing a permanent magnet. Such solenoids are referred to as *magnetic latching solenoids* to distinguish them from the *mechanical latching solenoids* described above. Magnetic latching is preferred in the smaller sizes where mechanical latching mechanisms become extremely small and delicate, as exemplified by spacecraft valves. Magnetic latching solenoid valves require only a brief electrical pulse (as short as 20 milliseconds) to open or close the valve, thereby minimizing power drain and heat generation.

Magnetic latching solenoid valves, by virtue of the constant magnetic field associated with the permanent magnet, may present problems when considered for certain applications such as a spacecraft carrying magnetometer experiments. The strength of this external field varies greatly with configuration. In some instances it is possible to restrict the magnetic field by means of shielding. Shielding is accomplished with thin layers or "cans" of magnetic material such as ingot iron or mu-metal, which trap much of the magnet's normal lines of force. A critical shielding requirement might require several layers of shielding with varying initial permeability and flux capacity, such as an inner can of ingot iron covered with one of 48 to 50 percent nickel-iron alloy (mu-metal) which in turn is covered with an outer can of 78 to 80 percent nickel-iron alloy. Such shielding tends to reduce the power of the permanent magnet due to the shield's interference with the magnet's normal lines of force. This loss in magnet efficiency must be taken into account when shielding is considered for magnetic latching solenoid valves.

All magnetic latching valves use a permanent magnet, but a particular design may take a number of forms, including:

a) Spring-loaded, single coil with polarity-reversing circuitry, single-polarity magnet

b) Spring-loaded, dual coil (one for pull-in and one for dropout), single-polarity magnet

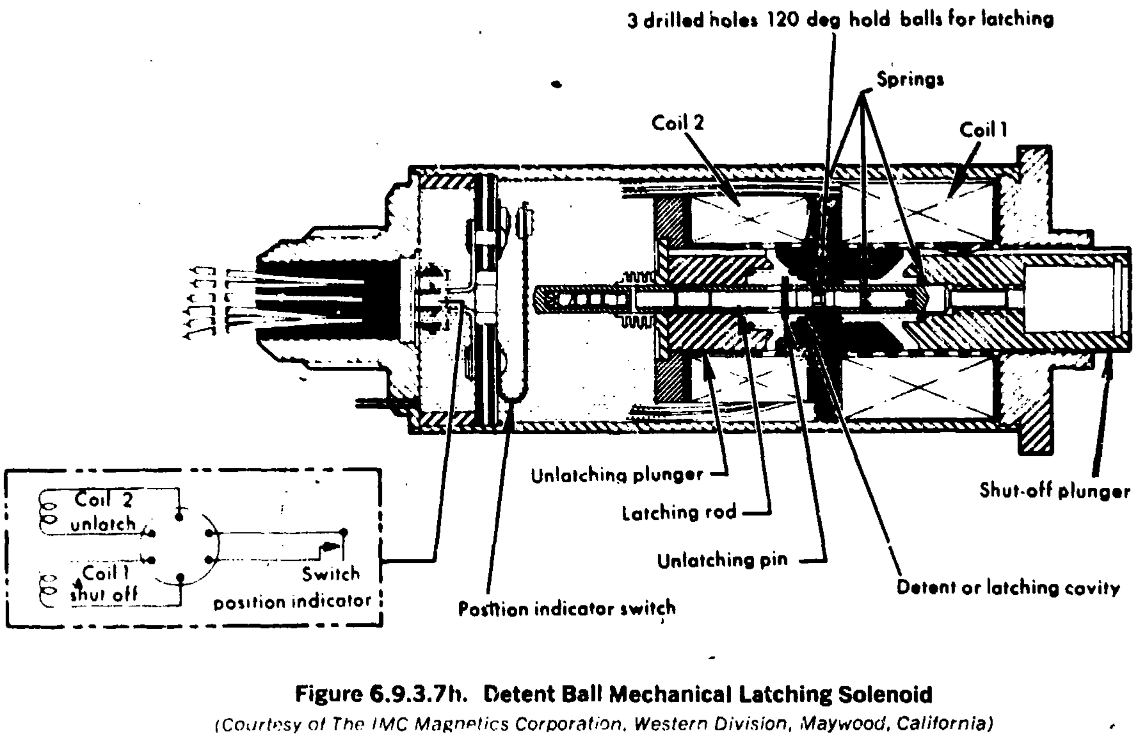


Figure 6.9.3.7h. Detent Ball Mechanical Latching Solenoid
(Courtesy of The IMC Magnetics Corporation, Western Division, Maywood, California)

c) Spring-loaded, dual coil, dual magnet (including one polarity-reversing magnet)

d) Spring-loaded, single coil, with polarity reversing circuitry, "demagnetizable" permanent magnet

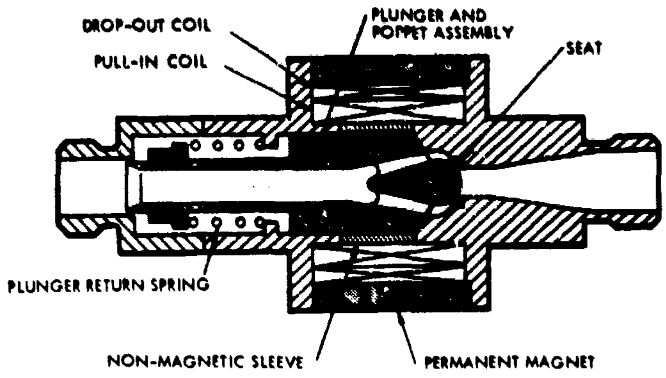
e) No-spring, dual coil with E-magnet for latching in either position.

f) No-spring, single coil with polarity-reversing circuitry and dual single-polarity magnets for latching in either position.

In the magnetic latching solenoid valve shown in Figure 6.9.3.7i, the permanent magnet tends to open the valve, but lacks sufficient force to overcome the spring force holding the poppet against the seat. The valve opening signal energizes the pull-in solenoid coil, whose electromagnetic field supplements that of the permanent magnet and causes the valve to open. Upon completion of the valve opening stroke, the pull-in solenoid is de-energized, but in the full-open position the attraction of the permanent magnet is sufficient to hold the valve open. To close the valve, the dropout solenoid coil is energized, counteracting the field of the permanent magnet and permitting the spring to close the valve. (Alternatively a single coil solenoid would simply send current through the coil in the reverse direction, but through a resistor to avoid degrading the strength of the permanent magnet.) When the valve is seated, the dropout solenoid coil is de-energized, and the permanent magnet field reforms but is again unable to influence the plunger in the closed position. This particular design has two advantages: it isolates the magnet material from the flowing fluid and has a relatively efficient, low pressure drop flow path. It has the disadvantage of not providing a way to avoid accidental latching in the closed position if the return spring force is insufficient to prevent instantaneous closing of the air gap due to axial shock.

Figure 6.9.3.7j illustrates a spring-loaded, dual coil solenoid valve incorporating a reversible polarity permanent magnet (sometimes referred to as a magnetic switch), as well as a fixed polarity permanent magnet. In the closed or unlatched position, the magnetic flux generated by the fixed polarity magnet PM1 is in phase with the magnetic flux generated by the reversible polarity permanent magnet PM2, which is a small disc permanent magnet between two cylindrical pole pieces of magnetic steel. In this condition the magnetic circuit has no effect on the plunger. When a DC pulse of correct polarity is applied to the solenoid coil assembly, it causes PM2 to switch its polarity and repel the flux generated by PM1. This action causes the flux of both magnets to shunt across the plunger gap and open the valve. The valve will remain latched in the open position until the polarity of PM2 is again reversed, causing the flux to revert to the original flux path which will then have less reluctance. This configuration tends to be relatively heavy, but has the advantages that (1) the permanent magnets are isolated from the flowing fluid, and (2) the valve cannot accidentally latch open even if the air gap is instantaneously closed.

The spring-loaded, single coil magnetic solenoid valve illustrated in Figure 6.9.3.7k utilizes a thin disc or toroidal



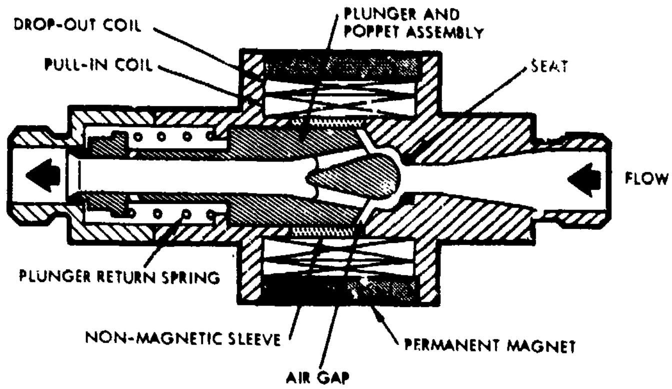


Figure 6.9.3.7i. Dual Coil, Single Magnet Magnetic Latching Solenoid Valve
(Courtesy of National WaterLift Company, El Segundo, Calif.)

permanent magnet. Such a magnet will produce only an extremely weak flux by itself, but can form the heart of a very powerful magnet if placed between two relatively long magnetic pole pieces when subjected to the induced magnetic field (as in the reversible polarity magnet shown in Figure 6.9.3.7j). However, if an air gap is present on either side of the thin permanent magnet, no significant magnetic flux field will exist and the magnet may be considered to be effectively demagnetized. In the case of the valve shown in Figure 6.9.3.7k, energizing the solenoid with a strong pulse of current (a) attracts the magnetic plunger to the permanent magnet force, (b) closes the air gap, and (c) magnetize the magnet. As long as the air gap remains closed, the powerful tractive force of the magnet will persist indefinitely. It is possible to unlatch by applying sufficient force to the plunger to overcome the magnet's attraction. Once the plunger moves away from the magnet the air gap is restored and the magnet becomes demagnetized. The usual method of unlatching is to energize the coil with a pulse of current having reverse polarity. This pulse sets up an electromagnetic field with opposite polarity to that of the magnet, which rapidly diminishes the tractive force holding the plunger in the latched position, permitting the spring to

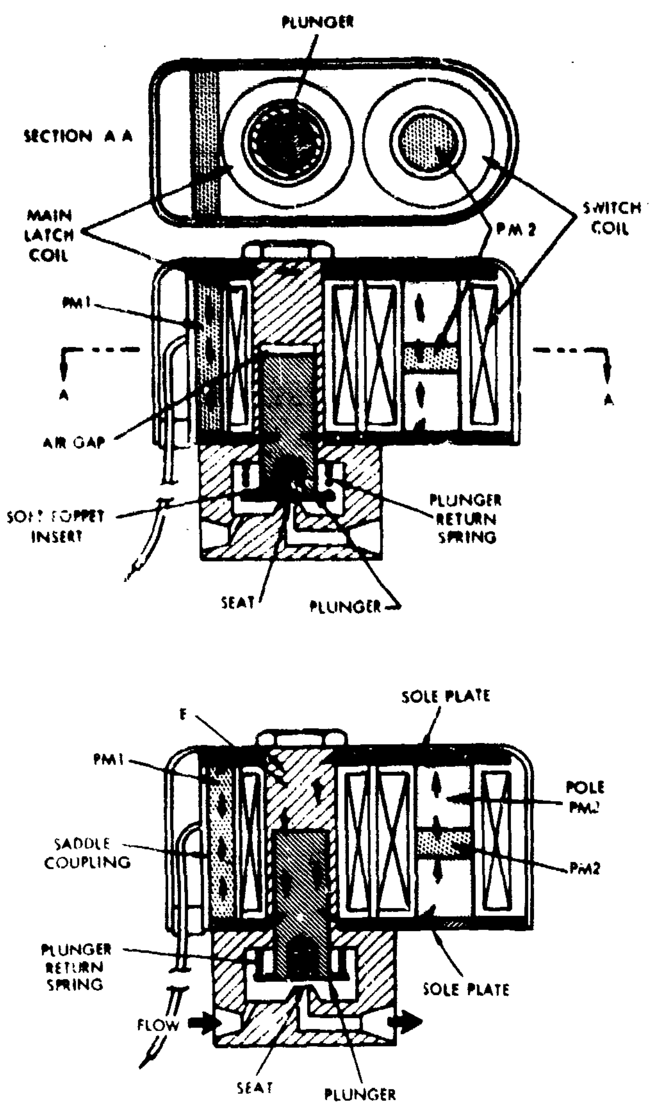


Figure 6.9.3.7j. Dual Coil, Dual Magnet Magnetic Latching Solenoid Valve
(Courtesy of Skinner Electric Valve Div., Skinner Precision Industries, Inc., New Britain, Connecticut)

move the plunger away, re-introducing the air gap, and demagnetizing the magnet. Only a fraction of the original magnetizing current is needed to degrade the magnet's tractive force, and the reduced current applied for unlatching is unable to attract or hold the plunger. It is interesting to note that if full current were applied for unlatching, the poles would transpose very rapidly (as in the design of Figure 6.9.3.7j) and the plunger would either remain latched or would instantly relatch. Time required for unlatching is less than for latching. This configuration has three advantages: (1) accidental latching from shock is impossible even if shock temporarily closes the air gap, (2) the magnet cannot be remagnetized without the electromagnetic field, and (3) the very small magnet results in relatively light weight. The primary disadvantage of this

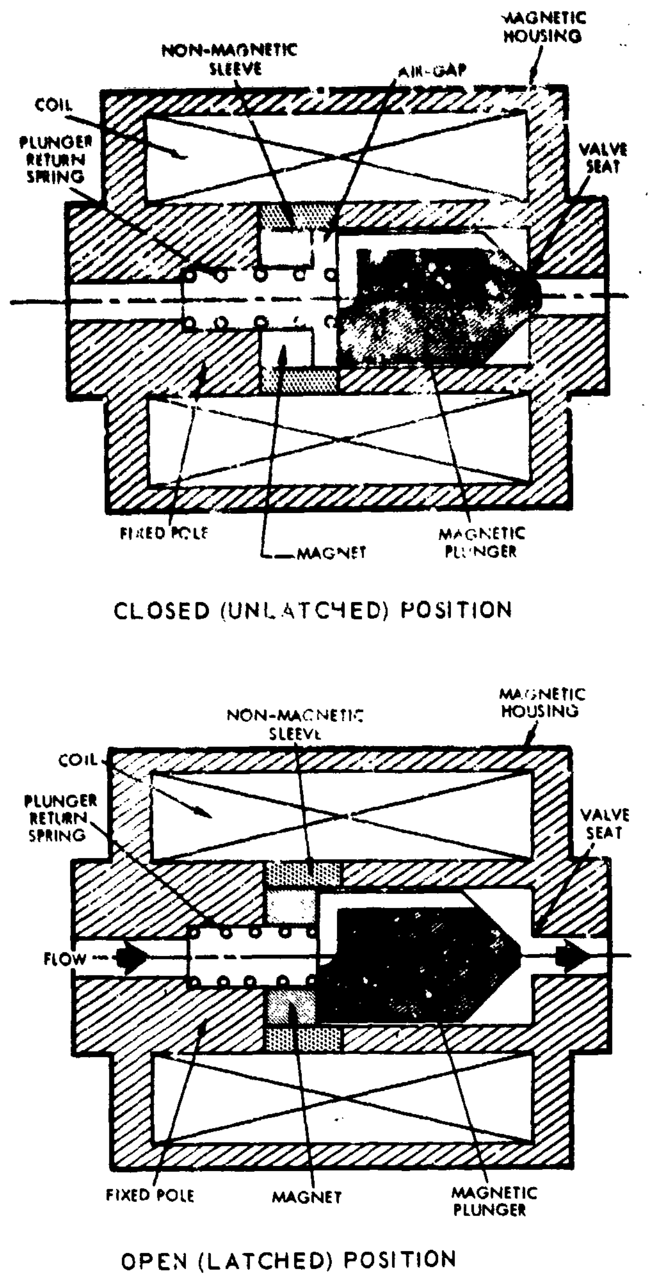
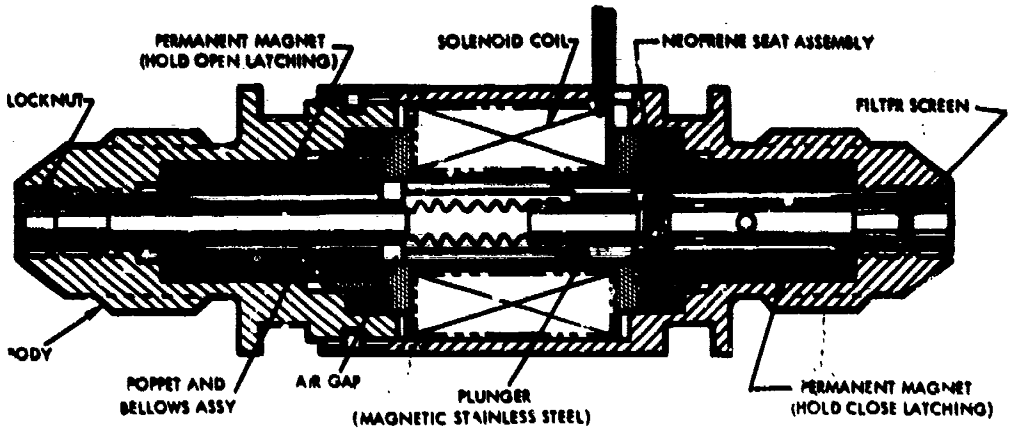


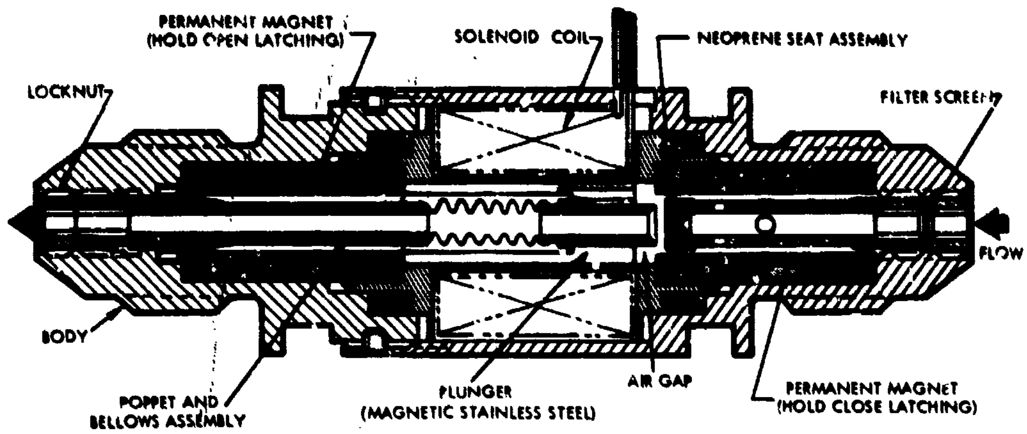
Figure 6.9.3.7k. Single Coil, Magnetic Latching Solenoid Valve with "Demagnetizible" Permanent Magnet
(Courtesy of Eckel Valve Company, San Fernando, California)

design is that the magnet is in contact with the flowing fluid and must be encapsulated if fluid compatibility presents a problem.

Figure 6.9.3.7l shows a single coil, dual magnet configuration which uses no spring and is magnetically latched in either the open or closed position. Energizing the solenoid coil in a given direction counteracts the magnetic field of the permanent magnet with which the plunger is in contact and supplements the magnetic field of the other permanent



CLOSED (LATCHED) POSITION



OPEN (LATCHED) POSITION

Figure 6.9.3.71. Single Coil, Dual Magnet, Magnetic Latching Solenoid Valve
(Courtesy of Carleton Controls Corp., East Aurora, New York)

magnet, causing the plunger to translate from one magnet to the other. Once the air gap is closed between the plunger and the second permanent magnet, the solenoid coil is de-energized and the plunger remains latched in its new position. Reversing solenoid coil polarity and re-energizing reverses the process. This configuration has the distinct advantage of positive magnetic latching in both positions, but shares the disadvantage of magnets exposed to the flowing fluid with the attendant compatibility problems.

Duty Cycle. Duty cycle is an important consideration for solenoid actuators, since an excessive duty cycle results in overheating.

In solenoids, the ON-OFF sequence is the duty cycle which determines the allowable voltage input. The duty cycle is limited by either of two factors:

a) The maximum percentage of time in one duty cycle that the solenoid is energized, disignated "f"

b) The length of time in seconds of the longest single impulse during one cycle, designated "a."

For example, the duty cycle of the solenoid may be specified as f = 65 percent, a = 1, which means that the solenoid is energized 65 percent of each actuation cycle and the longest time of continuous energizing is one second. Since "f" predominates, it becomes the limiting factor to consider for preventing overheating. If the duty cycle were specified as f = 25 percent, a = 500, "f" is submissive to "a" which would become the limiting factor.

Effects of Temperature Change. An important factor in the design and selection of solenoids is the potential temperature rise in the solenoid due to the heat generated in it. As the resistance of the coil increases due to the heat generated, the heated coil will draw less current from a given voltage source and the solenoid will perform less work. A solenoid must be chosen that will be capable of delivering the required force after the energized coil has reached a final stabilized temperature for the specified duty cycle at the specified maximum ambient temperature. Of course, if the solenoid is to be used where the duty will be highly intermittent or where it will be required to operate only once, the only allowance for the temperature should be highest anticipated ambient temperature. Figure 6.9.3.7m shows the effect of coil temperature on operating current versus solenoid force for a high temperature solenoid.

As the ambient temperature decreases, the resistance decreases, increasing the current and power temperature of the solenoid. To avoid large variations in resistance, a coil-compensating network can be employed, using a constant resistor in parallel with the negative temperature compensated (NTC) thermistors. The compensating network is placed in series with the coil. The combined resistance versus temperature is shown in Figure 6.9.3.7n (Reference 71-1).

Plunger Shape The shape of the plunger and face affects the force-stroke relationship of a solenoid. As shown in Figure 6.9.3.7o, flat-faced plungers give high forces for short strokes, while conical-faced plungers give low force with long strokes (Reference V-47).

6.9.3.8 ELECTRIC MOTORS. Electric motors are rotary devices which can be used both as rotary and linear actuators. Linear motion can be achieved through rack and pinion drives, or through the use of a nut moved back and forth by a threaded motor driver shaft. For rotary valve actuation, high torques are usually required to start rotation of the valve stem, but only low torques are needed to complete opening or closing.

An electric motor for valve actuation can be any type which has a high starting torque and can be stilled for prolonged periods of time without having to de-energize to avoid damage. Four of the more common basic types of motors for valves are:

a) DC motors
b) AC induction
c) Polyphase wound rotor
d) Universal.

DC motors are of three types: shunt-wound, series-wound, and compound-wound. The series-wound has the highest starting torque and quickest response time, but will overspeed when the load becomes small. A comparison of speed versus load for DC motors is shown in Figure 6.9.3.8a.

Polyphase AC induction motors, called squirrel cage motors, are adapted to the torque motor application by increasing the rotor resistance, which in turn decreases locked-rotor current and increases the locked-rotor torque. The increase in resistance is accomplished through variations in size and type of material and in rotor electric conductors. The torque-speed curves depend on the magnitude of the applied voltages (Figure 6.9.3.8b). The synchronous speed is the maximum speed and can be expressed as

$$n = \frac{120f}{p} \qquad \text{(Eq 6.9.3.8)}$$

where
n = speed, rpm
f = frequency of power supply, cps
p = number of poles in stator

Many varieties of single phase induction motors exist. For torque motor applications, the split capacitor, repulsion, and repulsion-induction motors are frequently used. In the

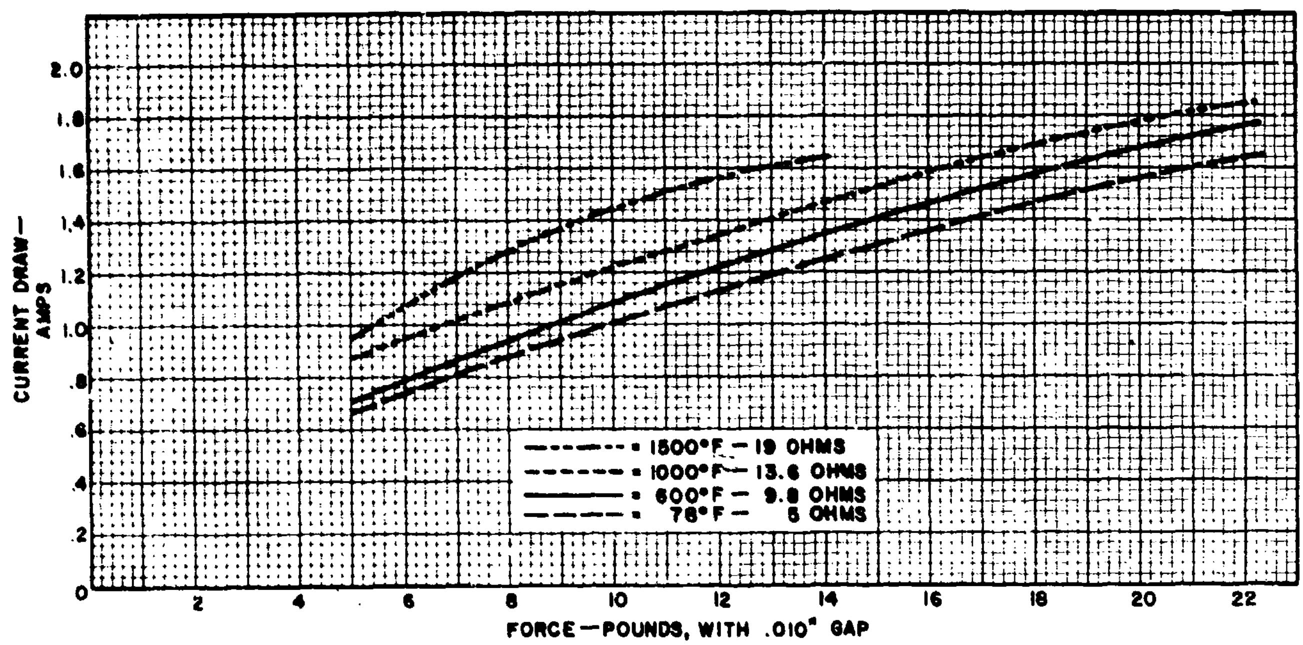


Figure 6.9.3.7m. Effect of Temperature on Solenoid Force and Current Draw
(Courtesy of Kidde Aerospace Division, Belleville, New Jersey)

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SUPERSEDES: MAY 1964

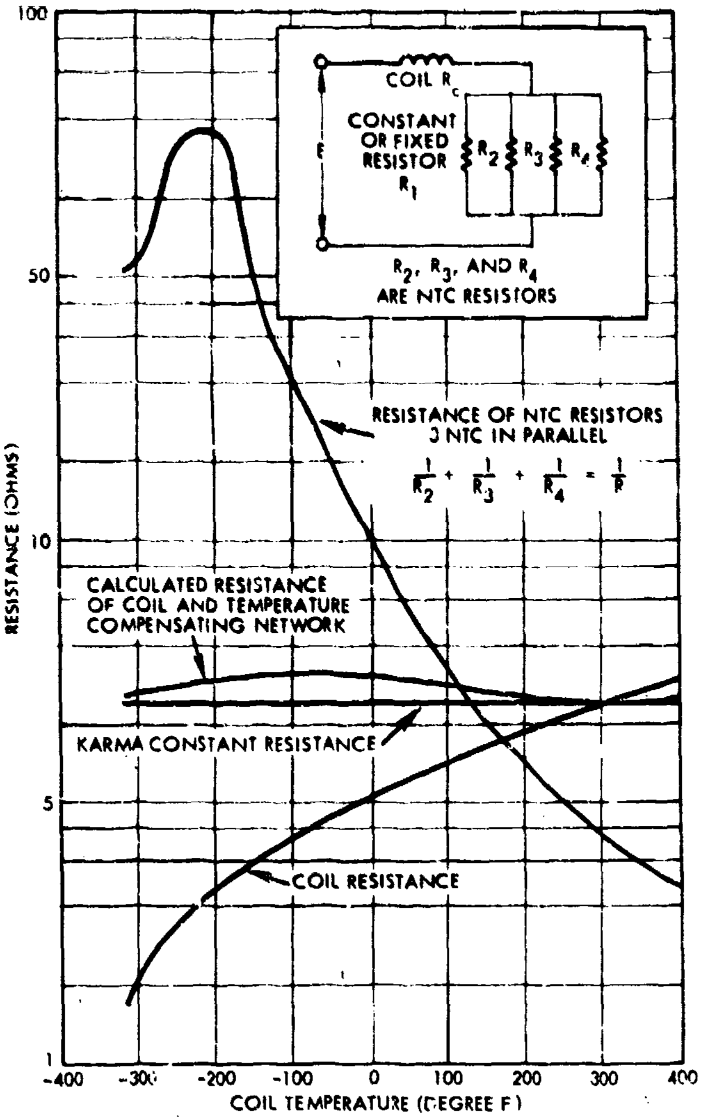


Figure 6.9.3.7n. Effect of Temperature on a Temperature Compensating Resistance Network

(From AFBMD-TR-60-.79, prepared by AiResearch Manufacturing Company, a Division of The Garrett Corporation, Phoenix, Arizona)

split capacitor motor, the phase shift caused by a capacitor in series with the starting winding causes the motor to start. The repulsion motors are commutator type devices, similar to DC motors. The wound rotor is short-circuited by brushes placed on the commutator such that the magnetic axis of the rotor winding is offset from the magnetic axis of the stator winding. High starting torque and variable speeds are achieved through high short-circuit currents. In the repulsion induction motors, the rotor has two windings. One winding is connected to the commutator while the other winding is a conventional squirrel cage winding. Start-up is achieved through short-circuiting the brushes on the winding connected to the commutator. At full speed the squirrel cage winding is used (Reference 1-271).

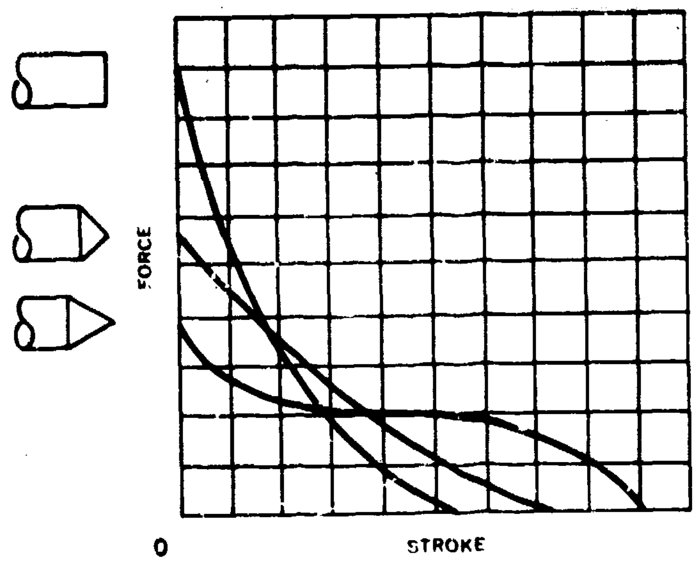


Figure 6.9.3.7o. Effect of Solenoid Plunger Face on Output Force

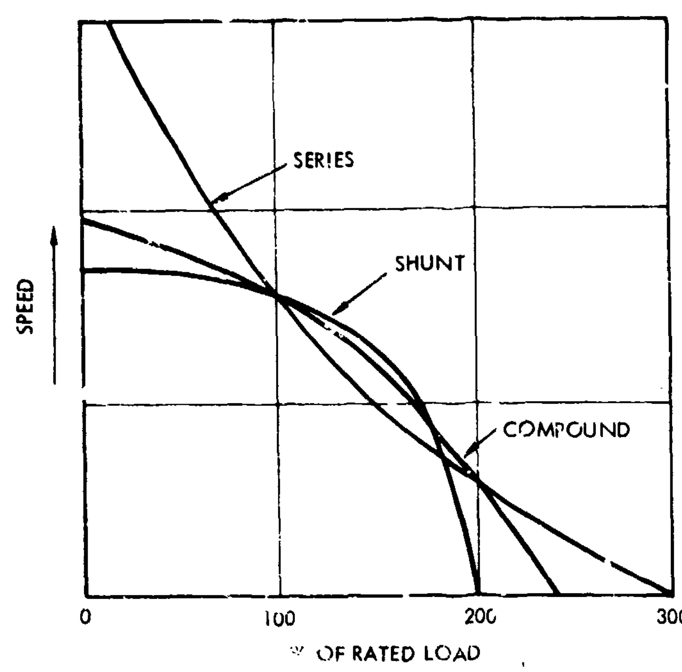


Figure 6.9.3.8a. Comparison of Speed Versus Percent Rated Load for D.C. Motors

(Reprinted from "Product Engineering," R. W. Matthews, March 21, 1961. Copyright 1961, McGraw-Hill Publishing Company, Inc., New York)

A universal motor is a series motor which can operate with either DC or 60 cycle AC with approximately the same performance. See Reference 1-271 for a comprehensive discussion of universal motors.

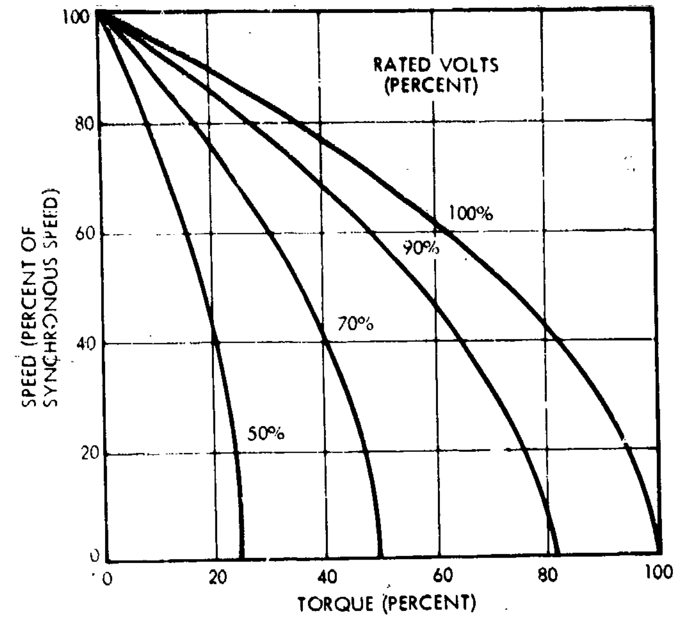


Figure 6.9.3.8b. Torque-Speed Curves for 3-Phase Induction Motor with Various Applied Voltages

(From "Machine Design," E. A. Roller and R. C. Hughes, Copyright 1961, Penton Publishing Company, Cleveland, Ohio)

Dynamic Braking. A motor used for valve control must be capable of dynamic braking, which is the ability of a motor to come to an abrupt stop when properly energized. DC motors will brake when the armature is shorted and the field is energized. Single phase AC motors will stop when energized by direct current. Some permanent split capacitor motors will brake when both the starting and main windings are energized from the same line. This is termed capacitor-shorting (Reference 19-201).

Reversibility. The direction of rotation of DC motors can be reversed by reversing either the armature or field windings with respect to each other. Small DC motors can be reversed while running. All series DC motors can be reversed if first allowed to come to a stop. Single phase AC induction motors that use a starting mechanism are not normally reversible while running unless a special switching circuit is used. They will reverse if connected to rotate in either direction and allowed to come to a stop. The small, permanent-split capacitor motor can be built with balanced windings to give reversibility while running. The polyphase AC motors can always be reversed if allowed to stop, and many small sizes are reversible while running.

Thermal Protection. In the stalled condition, the amount of heat generated is in direct proportion to the stalled time. The amount of heat which can be dissipated depends upon the available ventilation, the motor frame size, and type of insulation. To prevent damage to the winding due to overheating, a thermal protector is usually factory installed in the winding. The most common protector is a bimetallic, heat sensing element. At temperature, the protector contacts will open, causing the motor to de-energize. The contacts reclose when the motor has cooled (Reference 1-271).

Another method is the installation of a time delay, current-sensitive relay which is set at the allowable locked-rotor time and installed in series with the windings. If the current through the windings remains at the locked rotor value too long, the relay will open, de-energizing the motor. A third alternative is to use a blower to circulate the air (Reference 1-271).

6.9.3.9 TORQUE MOTORS. A torque motor differs from a typical electrical motor in that its displacement is small and it has a differential control winding composed of two separate coils. Torque motors are used extensively as servo valve actuators, and although they are designed to use either AC or DC power, DC torque motors are used almost exclusively because they are economical to operate even though for some servo valve applications they require push-pull amplifiers and demodulation circuits.

A torque motor is controlled through the interaction of two flux components which exist in the air gaps between poles. One component is a polarizing flux furnished by a permanent magnet, and the other component is a varying flux furnished by the coil. The polarizing flux establishes a balanced force condition by balancing the air gap torques. The control flux upsets the force equilibrium by changing the net flux in certain gaps relative to the other gaps. An electromagnetic torque induces armature movement until it is balanced by the strain energy stored in a spring.

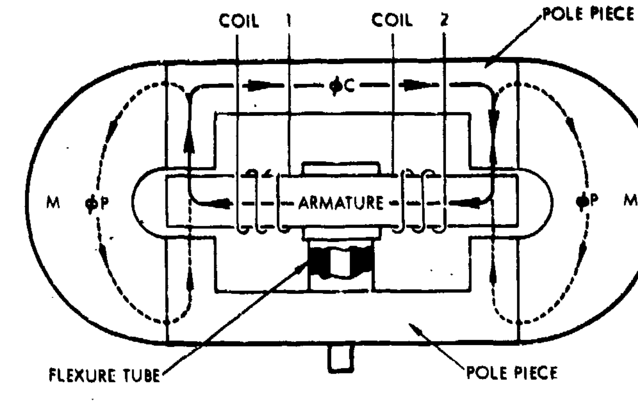
The armature is a flat plate which has a flexure attached to the center. The tube is rigidly fixed at its extremity and acts as a spring. The axis of the tube is parallel to the rotational axis of the armature. Surrounding the armature is a coil assembly which is mounted between magnets attached to poles which end a short distance from the armature face. When a differential current exists in the coils, a resulting net flux in the armature supports the flux flowing between the magnets on one end of the armature gap and a gap on the other end of the armature. This flux produces a torque which moves the armature. Figure 6.9.3.9a shows torque motor construction and flux paths.

The duty cycle rating for torque motors is limited by the maximum temperature capacity of the components, which in turn affects the following two factors:

a) The percent of the duty cycle that the motor may be stalled

b) The maximum time of continuous stall.

Thus, for example, a 30 percent, 8-minute duty cycle means that for a one-hour total duty cycle, the motor is stalled for 18 minutes with 8 minutes being the longest period of continuous stall.



M = PERMANENT MAGNET
ϕP = POLARIZING FLUX
ϕC = CONTROL FLUX

Figure 6.9.3.9a. Torque Motor Schematic
(Courtesy of Midwestern Instruments, Inc.)

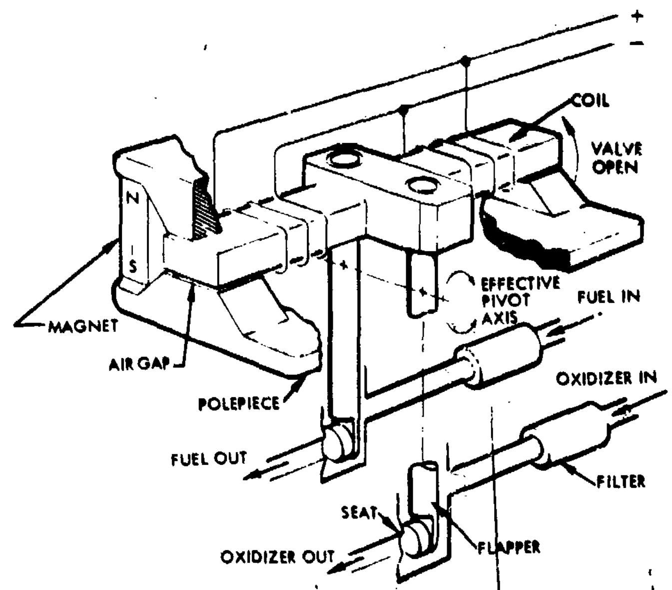


Figure 6.9.3.9b. Torque Motor Actuated Bipropellant Valve Schematic
(Courtesy of Moog Servocontrols, Inc., East Aurora, New York)

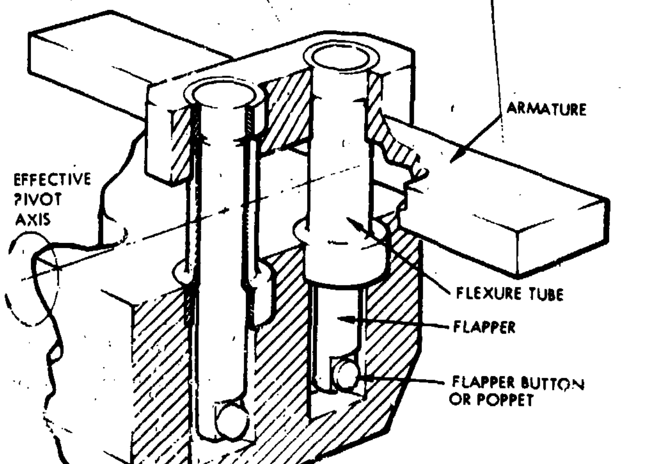


Figure 6.9.3.9c. Flexure Tube Mounting of Bipropellant Valve Torque Motor
(Courtesy of Moog Servocontrols, Inc., East Aurora, New York)

Although torque motors have been used in servo valve applications for many years, it is only recently that this actuator has been adapted to small propellant shutoff valve applications for pulse mode rocket engines. These small rocket engines (or thrustors) usually of from 25- to 300-pound thrust, are operated for pulses as short as 20 milliseconds in applications such as spacecraft attitude control systems. For bipropellant engine systems, a single DC torque motor usually actuates both the fuel and oxidizer poppets or flappers, as shown schematically in Figures 6.9.3.9b and 6.9.3.9c. For cold gas or mono-propellant systems a similar configuration utilizing only one flapper and valving unit is used. Both soft seat and all-metal hard seat valving units have been used with this type of actuation. Primary advantages of the torque motor actuator are the complete lack of sliding parts and the elimination of dynamic seals. There is complete isolation of the coils and permanent magnets from the fluid. The armature is supported by flexure tubes which act as flexible fluid barriers between the electromagnetic (torque motor) and fluid (wetted) sections of the valve. An additional reliability advantage stems from the potential coil redundancy resulting from the use of two coils in parallel, as shown in Figure 6.9.3.9b. The valve can be designed to operate with a single open coil or with a partially shorted coil. Some valve applications employ a third coil for operation from a separate or redundant power source (Reference 145-3). Torque motors perform well where very rapid response is required, as in pulse mode rocket engines. Reference 145-3 states that in addition to the fact that a torque motor requires less power than a comparable solenoid, from consideration of the transient current buildup in a resistive/inductive circuit the time required to reach a current level that will move the armature will be less for a torque motor than for a comparable solenoid valve. Both opening and closing response times of less than 5 milliseconds have been obtained with torque motor actuated propellant valves. From the standpoint of system dynamics it is significant to note that only a small portion of this time, often less than 1 millisecond, is valving element response time (the period during which the poppet or flapper is actually in motion). This is because the major portion of the total response time is that required to alter the magnetic field sufficiently to permit flapper motion. The stroke of torque motor actuated valves is characteristically short, usually from 0.005 to 0.030 inch for bipropellant valves.

The force-position relationships of a typical torque motor actuated propellant valve are illustrated in Figure 6.9.3.9d. The permanent magnets generate sufficient flux to overcome the spring (flexure tube) force and keep the valve closed. When current is passed through the coils, the magnetic flux of the permanent magnets is reversed, and the armature will move to the open position. Note the bias of the spring null position in Figure 6.9.3.9d. In a servo application, the spring null of a torque motor will usually coincide with the magnetic null and with no current applied both spring force and magnetic force will be zero in the centered position. With a propellant valve, however, it is desired to utilize a positive spring force to hold the poppet or flapper against the seat in the absence of fluid pressure. Inspection of Figure 6.9.3.9d shows that although below the spring null position the spring tends to open the valve, the magnetic force (with no current) always exceeds the spring force and will hold the poppet against the seat in the closed position.

It may also be seen from Figure 6.9.3.9d that although the valve may be either opened or closed with or without upstream fluid pressure applied, should the upstream pressure exceed the design value sufficiently to overcome the combined spring force and magnetic force (with current) in the closed position, the valve will not open. In Figure 6.9.3.9d the image of spring force is plotted to facilitate visualizing the net forces, exclusive of pressure and fluid dynamic forces, which act during opening (the vertical distance between the *magnetic force with current* and *image of spring force* lines) and closing (the vertical distance between the *magnetic force with no current* and *image of spring force* lines). It may be seen that when current is removed from the coil, the magnetic force is reduced to a level below the spring force and the valve will start to close. Below the spring null position all closing force is magnetic, supplemented by fluid pressure force.

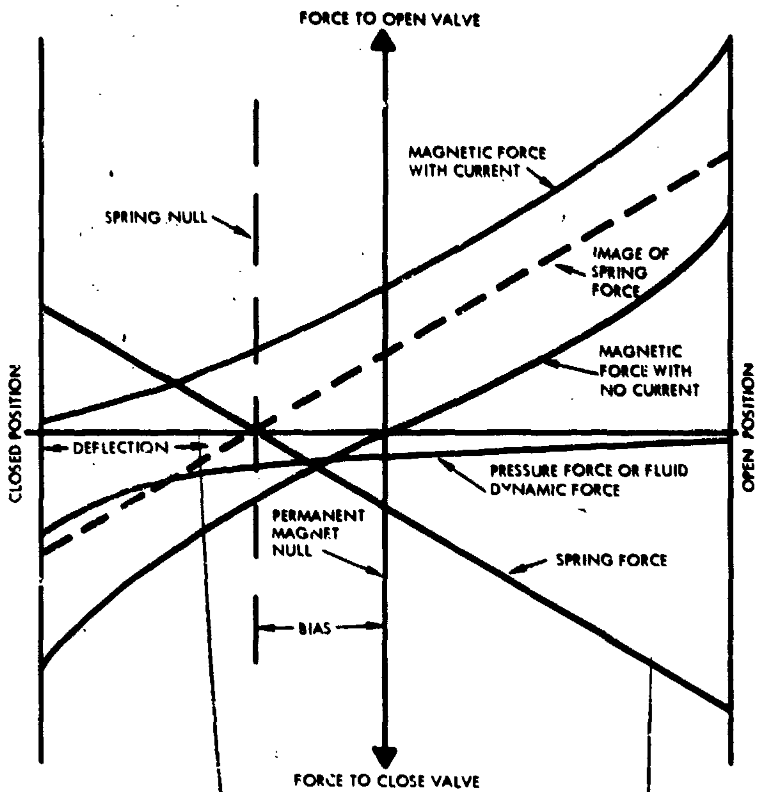


Figure 6.9.3.9d. Torque Motor Force-Deflection Relationships

REFERENCES

*References added March 1967

1-21	19-142	160-5
1-67	19-201	160-10
1-81	19-203	160-15
1-261	19-211	160-24
1-271	27-23	185-1
6-127	51-4	193-3
6-182	61-3	410-1
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V-167*	V-45*	V-264

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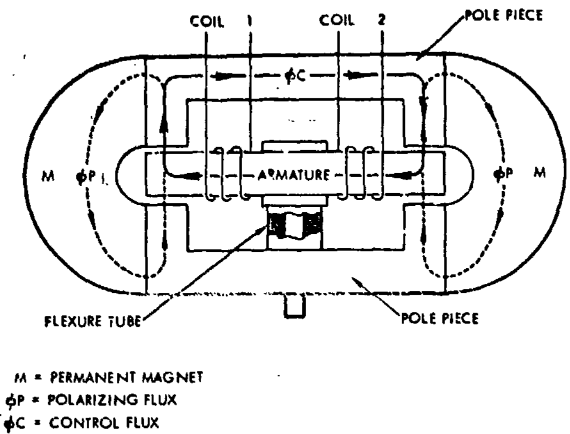

Figure 6.9.3.9a. Torque Motor Schematic
(Courtesy of Midwestern Instruments, Inc.)

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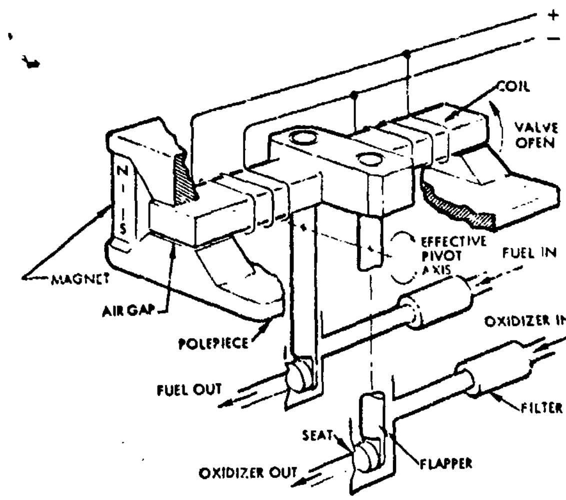

Figure 6.9.3.9b. Torque Motor Actuated Bipropellant Valve Schematic
(Courtesy of Moog Servocontrols, Inc., East Aurora, New York)

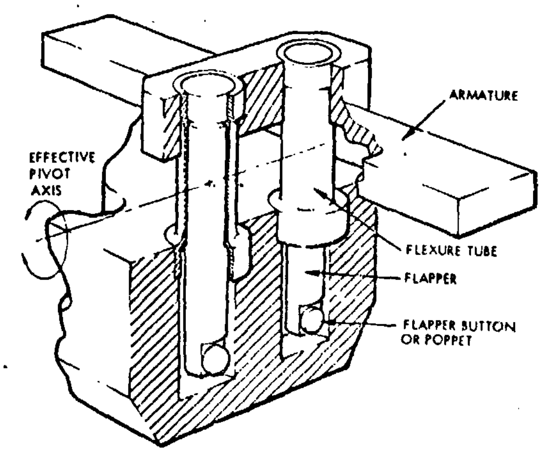

Figure 6.9.3.9c. Flexure Tube Mounting of Bipropellant Valve Torque Motor
(Courtesy of Moog Servocontrols, Inc., East Aurora, New York)

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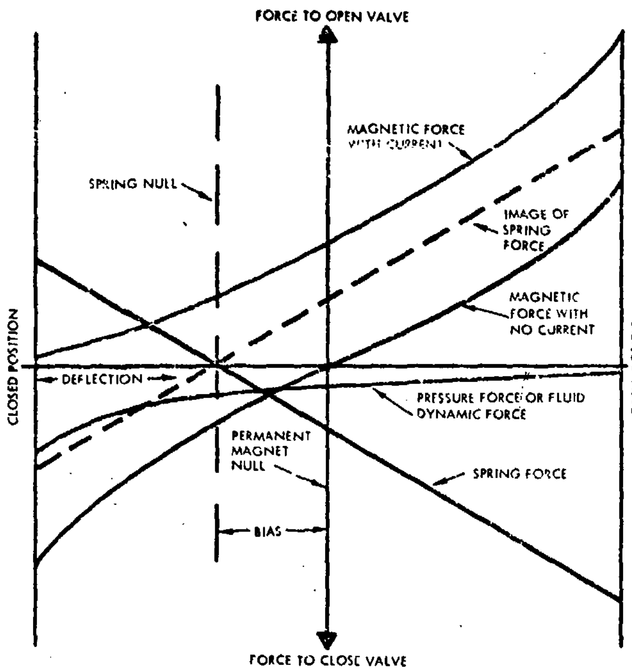

Figure 5.9.3.9d. Torque Motor Force-Deflection Relationships

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