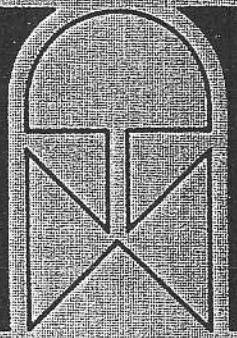
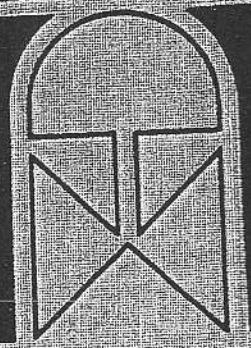


ISA Handbook of Control Valves

2nd Edition

ISA Handbook of Control Valves

2nd Edition ▪ J.W. HUTCHISON



PREFACE

The Instrument Society of America has prepared this handbook as part of an overall program to produce in-depth control valve technical material. The first phase was the production of a 38 minute 16 mm color sound film which gives a quick but broad look at control valve technology. A 15 minute 16 mm color sound film and a workbook have been produced covering maintenance and overhaul procedures. Additionally, a 12-part series of audio cassette study packages is underway, covering valve selection and applications in detail. This handbook complements the films, the audio cassette study packages, and is intended to enable both technicians and engineers to advance their control valve knowledge. The text emphasizes the whys of the variety of valve types, detail design features, manufacturing features, material selection, installation considerations and many other facets of control valve technology, all presented in an integrated text. The material ranges over a spectrum from fundamentals to advanced technical information.

This publication was prepared under the direction of the Final Control Elements Committee of ISA's Process Measurement and Control Division.

Participating authors were selected on the basic of their eminence in the particular topics which they covered. All are recognized authorities in the field of control valve technology and represent manufacturers as well as users. Their expertise has resulted in a creditable compilation of man's present knowledge of control valves. To the individual authors and their companies we express gratitude for the time, effort, and interest they have shown in the preparation of this book.

A cursory review of the technical information presented herein suggests that present knowledge of control valves has been developed to an advanced degree. Much that is now known has been developed in the relatively recent past. Both manufacturers and users have contributed to this store of information by working together. This book, the first to gather so much of this data, presents some material which has heretofore been reported in the literature, but which still seems to be neither well known nor well understood by many people.

What may not be quite so evident is that there is a very extensive amount of investigative work still going on with respect to many different aspects of valve technology. This is notably true in the areas of sizing, noise, flashing, and cavitation. Considerable activity along these lines is presently under way within the ISA Standards and Practices Department Committee on Valve Sizing (SP39 Committee).

Over a period of many years, several specific systems of terminology and nomenclature have evolved with respect to control valves. This is true with valve components and parts and with the terms and symbols used in control valve sizing. This is not uncommon in the earlier stages of any technology, although it may not be a desirable situation. The advantages of standardization are both many and great, not the least of which is that everyone understands exactly what is meant by everyone else. However, getting people to agree to a new universal language is not easy, nor can it be accomplished quickly. This is quite understandable since everyone is reluctant to change from the system he/she knows best and is most familiar with, or in some cases, in which he/she may even have a proprietary interest. It is, therefore, gratifying to be aware that standardization is at present actively being pursued with respect to control valves. The ISA Control Valve Sizing Committee (SP39) has worked long on the project and is still engaged in a great deal of activity in this direction. They have, for example, already published four documents containing terms and symbols which they expect to soon become American National Standards.

The Appendix section of this book is arranged to incorporate a large quantity of frequently used reference material. This compilation of material is as complete as possible with the intent of eliminating the need to search through several different books to find needed data. The Appendix section, however, is not the sole location of tables, charts and graphs. Most of this material appears elsewhere within the framework of the related chapters.

More than two dozen individuals actively participated in the review of the material as it was prepared. Their comments, suggestions, and opinions were of special value in enabling

the authors to revise their drafts for correcness and clarity. Reviewer's comments were of a particularly constructive nature and were well received by the individual authors. Practically every author participated in the review of chapters other than his own. In addition, there were others who were very active in the reviewing process. The names of all reviewers appear in a separate listing. To all who assisted in the review, we express appreciation for the quality and value of their comments and for the interest and cooperation they have shown during the preparation of this book.

It may be noted that some specific material and information appears in more than one location in the book. It is expected that some readers may look for certain information within one chapter while others will expect to find it in another chapter; therefore, to facilitate useage of the book, and to eliminate the need of continual reference to several different parts of the book, some duplication exists.

Relatively minor differences of opinion among the authors may be either observed or construed in certain areas of the book. This is neither a totally unexpected situation to encounter in the still-developing technology of valves, nor is it a cause for undue concern. It merely points up the fact that further investigation and development should be, and in all probability, will be done within these areas. In all cases, the authors have experimental or engineering data to substantiate their statements.

Diagrammatic arrangements of control valve piping and manifolding are presented in Chapter 12 "Installation." These cover a great number and a great variety of situations with corresponding satisfactory solutions. It is felt that many hours of drafting time may be saved in the preparation of engineering documents by simply referring to a page and figure number within this book.

The control valve is often the most costly element in a control loop. Also, since it is in the process stream, it is the element most likely to cause process downtime in the event of malfunction. For these reasons, how it is installed and how it is maintained are of vital concern to everyone who is in any way involved with control valves. Therefore, the information presented in Chapter 11 "Maintenance" and Chapter 12, "Installation" will be found to be of particular interest to all who use this book.

Throughout the book, authors have used illustrations and technical data from their own particular companies because of the availability of this material to them and because of their greater knowledge of it. In many cases, the data presented was developed and established by the author's company through specific research or experience. Therefore, to give the reader verification and credence for the statements he makes, the author refers specifically to his own company's findings. There is no intent on their part, or ours, that this be construed as advertising or sales promotional in nature.

New Material

A large amount of new and useful information has been prepared since the First Edition for inclusion in this Second Edition. A completely new chapter has been prepared: Chapter 13, "Planning, and Selecting Control Valves." Part 1 of Chapter 13 presents a guide which one may follow to very good advantage when *planning* an installation. Planning should be done even before attempting to choose a specific valve. The better the planning, the greater will be the valve's chance of giving good performance. A logical and thorough method of going through the planning procedure is given in the guide, along with a tabulation of the many factors that should be taken into consideration. Part II, dealing with how to *select* a valve, also gives a guide which can be followed in selecting the *best* valve. It relates process conditions to the valve types best suited to satisfactorily meeting these conditions. By studying the "Selection Guide" and the information given in other chapters, anyone confronted with buying a valve from a functional and an economic basis can be relatively sure of coming up with the best choice. A comprehensive guide of this nature has never appeared in valve literature and it is expected to prove a very useful, practical help.

Control valve noise has always been an important, lively, and controversial subject. In this new Second Edition, "Noise" is presented as a separate, independent chapter (Chapter 7). All of the original, First Edition material has been completely updated and substantially expanded. Additional material from two new authors has been included, yielding a book with as broad a viewpoint as possible. All of those who read this chapter will become conscious of the fact that each of the authors is an expert in a very complex subject. Some additional comments in the "Editor's Preface to Chapter 7" may be of further interest to those who are concerned with valve noise. Today, so much work is taking place concerning valve noise, that it is anticipated each new printing of the text will call for an extensive update of the subject.

The method presented for sizing control valves in this edition of the book is based on the latest ISA Standards for Sizing (ISA-S75.01 and ISA-S39.3). One of the most important changes which have taken place in the new Standards is the addition of a number of new valve sizing factors to augment the old familiar valve sizing coefficient, C_V. The addition of these has resulted in the formulas becoming more complex. To anyone who might wish to merely select a proper size valve for a routine application the formulas may appear unnecessarily involved. In this connection it should be mentioned that the new Standards were not intended to be used in in the manner of "cook-book" formulas, but instead were meant to provide basic mathematical expressions for predicting the flow through valves with a degree of precision unattainable with earlier formulas. The ISA-SP39 Sizing Committee anticipated that various "working equations" would be developed and would evolve frrom these Standards for routine applications. This has, in fact, already taken place on several fronts. In this book the author has mathematically developed and listed quite a number of convenient working formulas derived from the Standards. One of the features of the Sizing Standards is their provision for the use of metric units, as well as U.S. units, by means of a series of numerical factors N_1, N_2, etc. The sizing procedure given here is a good reflection of the extent of mankind's progress in understanding the mechanism of flow through control valves, and the progress toward the goal of greater precision in sizing methods.

Beyond the material discussed above, the Second Edition is further characterized by: A greatly expanded amount of information on ball valves and butterfly valves. A descriptive write-up on control valve usage in nuclear power applications to provide the reader with a feel for, and an overall look at, what is involved in this relatively new field. This can be found in the APPENDIX. New ANSI Standards, for example B16.5, 1973, have made their appearance since the First Edition of the handbook. It is anticipated that the earlier Standards (B16.5, 1968), will be in use for some time, therefore both the old and the new Standards are included in Chapter 6. Additional material on actuators (Chapter 8 has been included, from a new author, to supplement the First Edition text.

James W. Hutchison
Editor in Chief
Honeywell, Inc.
(Retired)

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CHAPTER 1

INTRODUCTION

Orval P. Lovett, Jr.

E. I. DuPont de Nemours & Co.
Wilmington, Delaware

The *control valve* plays a very important part in the automatic control of modern plants, which depend on the correct distribution and control of flowing liquids or gases. Such control, whether for the exchange of energy, reduction of pressure, or simply to fill a tank, depends on some form of *final control element* to do the job. Final control elements may be considered the "muscle" of automatic control. They furnish the necessary power amplification between the low energy levels in controllers and the higher energy levels needed to perform their function in controlling flowing fluids.

The control valve is the most widely used type of final control element. Other final control elements include metering pumps, dampers and louvers (a variation of a butterfly valve), variable pitch fan blades, electric current control devices, and electric motors for positioning devices other than valves.

Despite the control valves wide use, there is probably no other element in the control system which receives less attention. In many systems, a control valve is subjected to more severe conditions of temperature, pressure, corrosion and contamination than other components, yet it must perform satisfactorily, with a minimum amount of attention, as it manipulates the flow of process fluid.

A control valve functions as a variable resistance in a pipeline. It provides a pressure drop by changing the turbulence in the process fluid or, in the case of laminar flow, the pressure drop is caused by the changed valve resistance or "drag". This pressure-drop process is often called *throttling.* For gases it approaches adiabatic, isothermal conditions. Deviations depend upon the degree of non-ideality of the gas (Joule-Thompson effect). In the case of liquids the pressure is dissipated by either turbulence or viscous drag. In either case, it converts the pressure energy to heat, resulting in a very small rise in temperature.

The usual control loop consists of three major parts. The first part is a *sensing element,* which usually is a transmitter. This is a device which is able to measure the process variable which is to be controlled, such as pressure, level, or temperature. The transmitter output is sent to a controlling instrument—*the controller,* which defines and measures the error between the set point or desired value, and the actual value of the process variable, and in turn sends a corrective signal to a final control element—the *control valve.* The valve varies the fluid flow to change the process variable to the desired value.

In pneumatic control systems, the pneumatic output signal from the controller may directly actuate the valve by either a spring-and-diaphragm actuator or a piston actuator. The positioning energy in this case is provided by compressed air which should be dry in outdoor installations to prevent freeze-up, and should be clean and filtered.

When a pneumatic valve is used with an electronic controller, either an electronic-to-pneumatic valve positioner or an electronic-to-pneumatic transducer is used. The same considerations for compressed air apply as in all-pneumatic control systems.

In control theory terminology, a control valve has both static and dynamic characteristics contributing to the overall success or failure of the loops controllability. The static or gain term, which is the flow characteristic of the valve, depends upon valve size, configuration of the plug and seat combination, type of actuator, valve positioner, upstream and downstream pressures, and to some extent fluid properties. Chapter V covers this subject in detail.

The dynamic characteristic is determined by the actuator or positioner-actuator combination. For slower processes, such as the control of temperature or liquid level, the

valve dynamics are not usually a limiting factor in controllability. On faster systems, such as liquid flow control, the control valve may be the dominant lag and must be included in loop controllability considerations. Generally, it is only the control systems specialist who needs to be concerned with the valve's dynamics. Normal considerations for positioner applications, as discussed in Chapter IX, will suffice in most control valve installations.

The history of the automatic control valve dates back to the self-actuated regulator which consisted of a globe-type valve with weight-loaded stem. The weights were used to balance plug forces and thus achieve some measure of control. Another early form of self-actuated regulator was the pressure balanced regulator. A process pressure was piped to the diaphragm chamber of a spring and diaphragm regulator. Either a pressure reducing valve, a back pressure regulator, or a differential pressure regulator could be built from variations of this basic valve.

The advent of pneumatic transmitters and controllers naturally led to pneumatic control valves. They were basically pressure reducing valves or back-pressure regulating valves, modified to utilize instrument air instead of process fluids. Many of the present day control valve manufacturers grew from companies which had been in the regulator business. Control valve applications continued to grow in number and complexity and many valve body variations as well as accessories became available to solve a variety of problems.

This handbook is intended to acquaint engineers with the factors of control valve design and application and to assist instrument engineers in the selection of the best valve body, actuator, and accessories for his applications.

CHAPTER 2

CONTROL VALVE COMPONENTS, DEFINITIONS, TERMINOLOGY AND SPECIFICATIONS

M. W. Hott, Jr. *

Jordan Valve Division
Richards Industries, Inc.
Cincinnati, Ohio

PREFACE

This chapter presents the fundamentals that are basic to an understanding of the detailed discussions contained in the following chapters. Part 1 contains labeled illustrations identifying the parts of the control valve components —valve types, cage trim, body assemblies, bonnets and actuators. Part 2 presents the commonly accepted definitions of the components illustrated in Part 1. Part 3 includes the commonly accepted terminology used in discussions concerning control valve applications. Control valve specifications are covered in Part 4.

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* M. W. Hott, Jr. is now with M. W. Hott Co., Cincinnati, Ohio.

Part 1
IDENTIFICATION OF CONTROL VALVE COMPONENTS

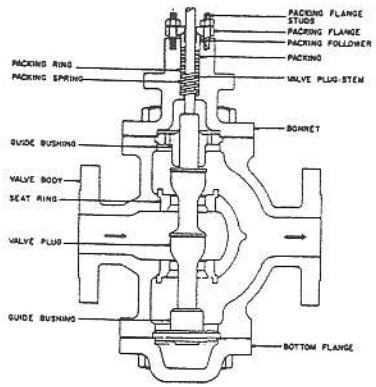


Figure 1. TYPICAL DOUBLE-PORTED GLOBE-BODY CONTROL VALVE.

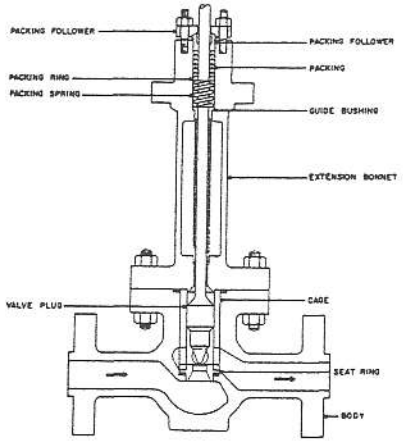


Figure 2. TYPICAL CAGE TRIM BODY.

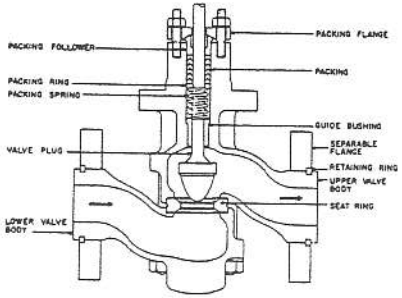


Figure 3. TYPICAL SPLIT VALVE BODY ASSEMBLY.

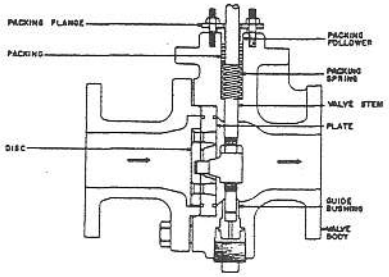
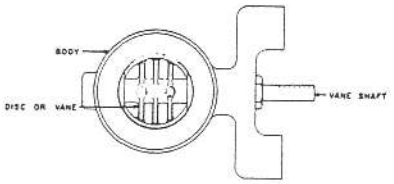


Figure 4. SLIDING GATE VALVE BODY.



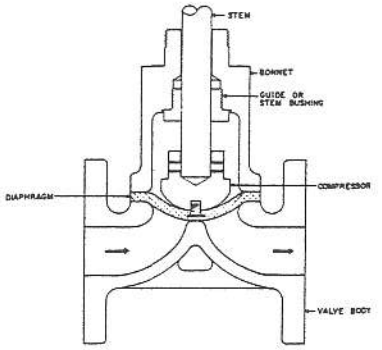


Figure 6. TYPICAL DIAPHRAGM VALVE.

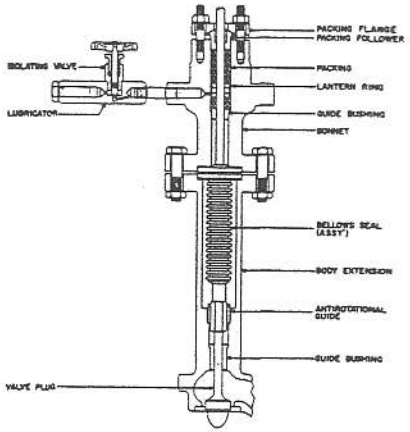


Figure 7. BELLOWS SEAL BONNET.

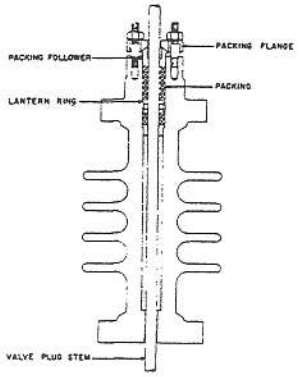


Figure 8. RADIATION FIN BONNET.

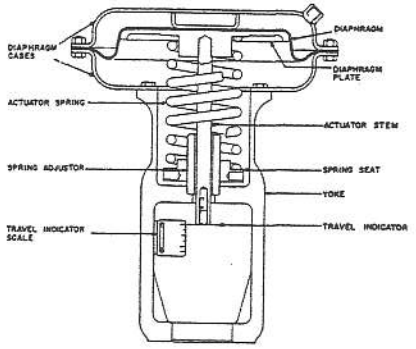


Figure 9. TYPICAL DIRECT ACTING DIAPHRAGM ACTUATOR.

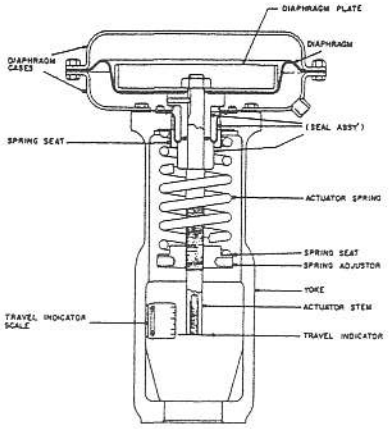
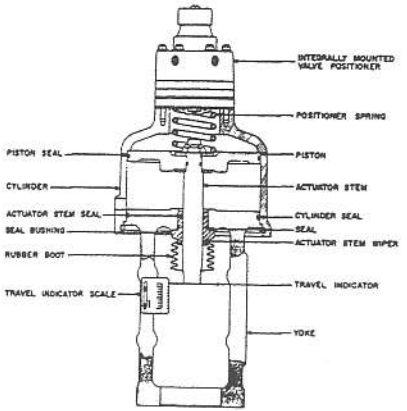


Figure 10. TYPICAL REVERSE ACTING



Part 2
DEFINITION OF CONTROL VALVE COMPONENTS

Actuator Spring: A spring to move the actuator stem in a direction opposite to that created by diaphragm pressure.

Actuator Stem: A rod-like extension of the diaphragm plate to permit convenient external connection.

Ball: The valve closure member in a ball valve.

Ball Valve: A control valve utilizing a rotatable ball shaped valve closure member that is bored, notched, or contoured to obtain various flow characteristics. This member is positioned to close, open, or to control the amount of flow by rotation.

Bellows Seal Bonnet: A bonnet which uses a bellows for sealing against leakage of controlled fluid around the valve plug stem.

Bonnet: The major part of the bonnet assembly, excluding the sealing means.

Bonnet Assembly: An assembly including the part through which a valve plug stem moves and a means for sealing against leakage along the stem. It usually provides a means for mounting the actuator. Sealing against leakage may be accomplished by packing or a bellows. A bonnet assembly may include a packing lubricator assembly with or without isolating valve. Radiation fins or an extension bonnet may be used to maintain a temperature differential between the valve body and sealing means.

Bottom Flange: A part which closes a valve body opening opposite the bonnet assembly. In a three-way valve it may provide an additional flow connection. It may include a guide bushing and, in a three-way valve, may also include a seat.

Butterfly Valve: A control valve that utilizes a rotatable disc or vane as a valve closure member.

Cage: A hollow cylindrical trim element that is a guide to align the movement of a valve plug with a seat ring and also retains the seat ring in the valve body. Often the walls of the cage contains openings which determines the flow characteristics of the control valve.

Center Valve Body: A housing placed between the upper and lower valve bodies to provide another flow connection as in a three-way valve.

Compressor: The part of the stem assembly in a diaphragm valve such as the Saunders patent diaphragm valve which effects flow restriction by forcing the diaphragm toward an internal web.

Control Valve: A valve with a power positioning actuator for moving the valve closure member to any position relative to valve port or ports in response and in proportion to an external signal. The energy for a control valve actuator is provided from an independent source.

Cylinder: The chamber in which the piston moves.

Diaphragm: A flexible pressure responsive element which transmits force to the diaphragm plate and actuator stem.

Diaphragm (As in Diaphragm Valve): The valve closure member in a diaphragm valve such as the Saunders patent diaphragm valve, which provides a variable restriction to flow.

Diaphragm Actuator: An assembly utilizing fluid pressure acting on a diaphragm to develop a force to move the actuator stem. It may, or may not, have a spring for positioning and return of the actuator stem.

Diaphragm Case: A housing, consisting of top and bottom sections, used for supporting a diaphragm and establishing one or two pressure chambers.

Diaphragm Control Valve: A control valve that is actuated by a diaphragm or one having a spring-diaphragm actuator.

Diaphragm Plate: A plate concentric with the diaphragm for transmitting force to the actuator stem.

Diaphragm Valve: A control valve that utilizes a flexible diaphragm as a valve closure member. The diaphragm is forced down toward an internal dividing web to restrict the flow.

Direct Actuator: A diaphragm actuator in which the actuator stem extends with increasing diaphragm pressure.

Disc: The valve closure member in a slide valve which provides a variable restriction in a port.

Disc: The valve closure member in a butterfly valve which provides a variable restriction in a port. Sometimes called vane.

Extension Bonnet: A bonnet with an extension between the packing box assembly and bonnet flange for hot or cold service.

Globe Valve: A basic control valve type which gets its name from the globular shape of its body. It normally uses the basic valve plug as its valve closure member.

Guide Bushing: A bushing in a bonnet, bottom flange, or body to align the movement of a valve plug with a seat ring. Guiding of a valve plug may be accomplished by an internal part of a bonnet or bottom flange or by a seat ring or seat ring extension or by a cage.

Inlet Port: The port which is connected directly to the upstream side of a fluid system.

Isolating Valve: A hand operated valve between the packing lubricator assembly and the packing box assembly to shut-off the fluid pressure from the lubricator assembly.

Lower Valve Body: A half housing for internal valve parts having one flow connection. For example, the half housing of a split body valve.

Lubricator: See *packing lubricator assembly.*

Outlet Port: The port which is connected directly to the downstream side of a fluid system.

Packing Box Assembly: The part of the bonnet assembly used to seal against leakage around the valve plug stem, including various combinations of all or part of the following:

a. Packing
b. Packing Nut
c. Packing follower
d. Lantern ring
e. Packing spring
f. Packing flange
g. Packing Flange Studs or Bolts
h. Packing Flange Nuts
i. Packing Ring
j. Packing Wiper Ring
k. Felt Wiper Ring

Packing Lubricator Assembly: An optional part of the bonnet assembly used to inject lubricant into the packing box.

Piston: A movable, pressure responsive element which transmits force to the actuator stem.

Piston Actuator: A fluid pressure operated piston and cylinder assembly for positioning the actuator stem in relation to the operating fluid pressure or pressures.

Plate: A separate piece inserted in a slide valve body to form a valve body port.

Port Guided: A design on which the valve plug is aligned by the body port or ports only.

Radiation Fin Bonnet: A bonnet with fins to reduce heat transfer between the valve body and packing box assembly.

Retaining Ring: A split ring that is used to retain a separable flange on the upper and lower valve bodies.

Reverse Actuator: A diaphragm actuator in which the actuator stem retracts with increasing diaphragm pressure.

Rubber Boot: A protective device similar to a flexible bellows, which is used to prevent build-up of abrasive deposits on the valve stem.

Seat: That portion of a seat ring or valve body which a valve plug contacts for closure.

Seat Ring: A separate piece inserted in a valve body to form a valve body port.

Separable Flange: A removable flange which fits over a valve body flow connection. It is generally held in place by means of a retaining ring.

Split Valve Body Assembly: An assembly of an upper valve body, lower valve body, and trim elements. The assembly can also include a center valve body to provide another flow connection as in a three-way valve.

Spring Adjustor: A fitting, usually threaded on the actuator stem or into the yoke, to adjust the spring compression.

Spring Seat: A plate to hold the spring in position and also to provide a flat surface for the spring adjustor to contact.

Stem Connector: A clamp, in two pieces, to connect the actuator stem to the valve stem.

Stem Guided: A special case of top guided in which the valve plug is aligned by a guide acting on the valve plug stem.

Top Guided: A design in which the valve plug is aligned by a single guide in the body adjacent to the bonnet or in the bonnet.

Top And Bottom Guided: A design in which the valve plug is aligned by guides in the body or in the bonnet and in the bottom flange.

Top And Port Guided: A design in which valve plug is aligned by a guide in the bonnet or body and the body port.

Travel Indicator: A pointer, attached near the stem connector, to indicate the position of the valve plug.

Travel Indicator Scale: A graduated scale attached to the yoke for indication of valve travel.

Trim: Those parts of a valve body assembly, excluding the valve body, bonnet assembly, and bottom flange, which are exposed and in contact with the line medium usually consisting of but not limited to the seat ring, valve plug stem, valve plug, valve plug guide, guide bushing and cage.

Upper Valve Body: A half housing for a split body type valve. It contains internal valve parts and has one flow connection. It usually includes a means for sealing against leakage along the stem and provides a means for mounting the actuator.

Valve Body: A housing for internal valve parts having inlet and outlet flow connections. Several common body arrangements are employed, as follows:

a. Single ported having one port and one valve plug.
b. Double ported having two ports and one valve plug.
c. Two-way having two flow connections - one inlet and one outlet.
d. Three-way having three flow connections, two of which may be inlets with one outlet (for conveying or mixing flows), or one inlet and two outlets (for diverging or diverting flows).

Valve Body Assembly: An assembly of a body, bonnet assembly, bottom flange and trim elements. The trim includes a valve plug which opens, shuts or partially obstructs one or more ports.

Valve Closure Member: That part of a valve which is positioned to close, open, or to control the amount of flow.

Valve Plug: A movable part which provides a variable restriction in a port.

Valve Plug Guide: That portion of a valve plug which aligns its movement in either a seat ring, bonnet, bottom flange or any two of these.

Valve Plug Stem: A rod extending through the bonnet assembly to permit positioning of the valve plug.

Vane: The valve closure member in a butterfly valve which provides a variable restriction in a port. Sometimes called disc.

Yoke: A structure by which the diaphragm case assembly is supported rigidly on the bonnet assembly.

Part 3
CONTROL VALVE TERMINOLOGY

Actuator Stem Force: The net force from an actuator that is available for actual positioning of the valve plug.

Actuator Travel Characteristic: Relation between per cent rated travel and diaphragm pressure. This may be stated as an inherent or installed characteristic.

Capacity: Rate of flow through a valve under stated conditions.

Clearance Flow: That flow below the minimum controllable flow with the valve plug not seated.

C_v: See *valve flow coefficient.*

Dead Band: The amount the diaphragm pressure can be varied without initiating valve plug motion.

Diaphragm Pressure Span: Difference between the high and low values of the diaphragm pressure range. This may be stated as an inherent or installed characteristic.

Dynamic Unbalance: The net force produced on the valve plug in any stated open position by the fluid forces acting upon it.

Effective Area: In a diaphragm actuator, the effective area is that part of the diaphragm area which is effective in producing stem force.

Equal Percentage Flow Characteristic: An inherent flow characteristic which, for equal increments of rated travel, will ideally give equal percentage changes of the existing flow.

Flow Characteristic: Relation between flow through the valve and per cent rated travel as the latter is varied from 0 to 100 per cent. This is a general term. It should always be designated as either inherent flow characteristic or installed flow characteristic.

Inherent Diaphragm Pressure Range: The high and low values of pressure applied to the diaphragm to produce rated plug travel with atmospheric pressure in the valve body.

Inherent Flow Characteristic: Flow characteristic when constant pressure drop is maintained across valve.

Inherent Rangeability: Ratio of maximum to minimum flow within which the deviation from the specified inherent flow characteristic does not exceed some stated limits.

Installed Diaphragm Pressure Range: The high and low values of pressure applied to the diaphragm to produce rated travel with stated conditions in the valve body.

Installed Flow Characteristic: Flow characteristic when pressure drop across valve varies as dictated by flow and related conditions in system in which valve is installed.

Leakage: Quantity of fluid passing through an assembled valve when valve is in fully closed position under stated closure forces, with pressure differential and temperature as specified. Leakage is commonly measured and may be expressed as a percentage of capacity at rated travel or as a cumulative quantity over a specified time.

Linear Flow Characteristic: An inherent flow characteristic which can be represented ideally by a straight line on a rectangular plot of flow versus per cent rated travel. Therefore, equal increments of travel yield equal increments of flow at a constant pressure drop.

Modified Parabolic Flow Characteristic: An inherent flow characteristic which provides fine throttling action at low valve plug travel and approximately a linear characteristic for upper portions of valve travel.

Normally Closed Control Valve: One which closes when the diaphragm pressure is reduced to atmospheric.

Normally Open Control Valve: One which opens when the diaphragm pressure is reduced to atmospheric.

Quick Opening Flow Characteristic: An inherent flow characteristic in which there is maximum flow with minimum travel.

Rated C_v: The Value of C_v at the rated full open position of the valve plug.

Rated Travel: Linear movement of the valve plug from the closed position to the rated full open position.

Seat Load: The contact force between the valve seat and valve plug.

Spring Rate: Force change per unit change in length.

Static Unbalance: The net force produced on the valve plug in its closed position by the fluid pressure acting upon it.

Stem Unbalance: The net force produced on the valve plug stem in any position by the fluid pressure acting upon it.

Unit Sensitivity: The per cent increase in existing flow for an increase in movement of 1 per cent of rated travel at a stated operating point.

Valve Flow Coefficient (C_v): Number of U.S. gallons per minute of 60° F water that will flow through a valve with one psi pressure drop under stated conditions. Stated conditions include pressure and per cent of rated travel.

Part 4
CONTROL VALVE SPECIFICATIONS

The writing of control valve specifications is an extremely important segment of the complete design, purchase and fabrication of any control system.

When the wrong control valve is specified, due to either incomplete or erroneous

specifications, the replacement of that valve becomes costly to both the user and the control valve manufacturer and often results in undue project delay.

Many control valve users have developed standard control valve specification forms that must accompany all control system project requisitions and they are designed to eliminate as much misunderstanding as possible regarding the valve specifications. These standard control valve specification forms are often based on the standardized specification sheet for control valves that is contained in the Instrument Society of America Standard ISA-S20, "Specification Forms for Instruments, Gages, Thermocouples, Orifice Plates and Flanges, Control Valves, and Pressure Safety Valves". This specification sheet is presently available from the Instrument Society of America as a separate section of ISA Standard S20.

The following paragraphs contain most of the items that should be considered in any control valve specification form. There are undoubtedly some items not covered in this text that are peculiar and necessary to individual situations.

Refer to Parts 1, 2, and 3 of this Chapter for appropriate definitions and terminology; in addition, refer to Fluid Controls Institute's FC 58-2, "Standard Measurement Procedure for Determining Control Valve Flow Capacity," and FCI 62-1, "Standard Formulas for Sizing Control Valves"; and the Instrument Society of America's Standard ISA-S20, "Specification Forms of Instruments, Gages, Thermocouples, Orifice Plates and Flanges, Control Valves, and Pressure Safety Valves".

Specifications

Service: On-off, Proportioning, etc.

BODY

Type of Body: Globe, angle, three-way mixing, three-way diverting, butterfly, Saunder's patented, ball plug cock, gate, etc.

Body Size: Specify nominal pipe size of body in inches.

Port Size: Specify port size in inches.

Guiding: Specify type of plug or inner valve guiding such as top and bottom, trunnion, top, port, cage, etc.

Number of Ports: 1 - Single Port (SP); 2 - Double port (DP); 3 - Three way; etc.

Plug Form: Percentage, linear, parabolic, quick-opening.

Packing Material: Teflon V-ring, Teflon - Asbestos, Durametallic, Buna N "O"-ring, etc.

Lubricator: Indicate only when required.

Bonnet type: Standard, plain, or finned for high temperature, plain extension for low temperature, bellows seal, diaphragm seal, etc.

End Connections: Specify flanged and describe if raised face (R.F.); ring joint (R.T.J.), large male (L.M.); etc.; screwed; socket welded, butt welded, with pipe schedule.

Material, Body & Bonnet: Specify steel, cast iron, 316 stainless steel, etc.

Material, Plug: Specify plug (inner valve, ball or disc) material, such as 316 stainless steel, 416 stainless steel, monel, alloy #6, etc.

Material, Seat: Specify seat ring(s) material.

Material, Posts: Specify post(s) material.

Material, Guides: Specify guide(s) (cage, bushings or bearings) material.

Material, Stem: Specify stem (shaft) material.

Composition Seating: Specify composition seating ("O" rings, composition disc, liners in butterfly valves, diaphragms in Saunders patented) by material such as Teflon, Kel-F, Buna-N, if desired.

ACTUATOR

Type of Actuator: Specify type of actuator such as diaphragm, piston or cylinder, electric-hydraulic, electric motor, solenoid, etc.

Close at and Open at: Specify signal span (psi, volts, amps, etc.) at which valve shall be closed and open. Note: This is signal to valve operator and is not necessarily controller output signal if a positioner and/or transducer is used.

Push down To: Specify if valve is to "Open" or "Close".

Flow Action To: Specify if flow action tends to "Open" or "Close" valve.

Failure Position: Specify position valve is to take on loss of actuating power.

Power Supply: Specify source of power supply such as 115 volt, 60 cycle, 80 psig air, etc. Note: Normal *minimum* air supply should be specified if lock-up and fail safe devices are required. This is especially important when using piston operators.

Size and Travel: Specify size or effective area and/or travel in inches or degrees rotation of actuator.

Handwheel and Location: Specify type of handwheel, if required, such as top mounted, or side mounted on operator, or side mounted on valve body such as butterfly or ball valves.

POSITIONER

Type: Side mounted, Top mounted, etc.

Filter Regulator: Indicate if required.
Gauges: Indicate if required.
By-pass: Indicate if required.

Input Signal: Indicate positioner input signal span from controller.

Output Signal: Indicate corresponding output signal span to actuator.

TRANSMITTER OR CONTROLLER

Input Signal: Indicate input signal span (if controller indicate measured pressure span and material if bourdon tube or bellows. If temperature controller, indicate measured temperature span and class of thermal system.)

Output Signal: Indicate corresponding output signal span.

SERVICE CONDITIONS

Fluid Units: Specify units of fluid flow, such as G.P.M., B/H, #/Hr, SCFH, etc.

Fluid: Name of fluid and state if vapor or liquid if it is not apparent.

Maximum Valve Capacity: Maximum valve capacity when valve is 100% open calculated with sizing pressure drop. Note: Normally this information is supplied by the valve manufacturer and is important for relief valve sizing and/or future capacity.

Valve C_V: Valve coefficient at 100% open - supplied by valve manufacturers.

Quantity Maximum: Maximum quantity of fluid valve is required to pass.

Flow C_V: Calculated flow coefficient at maximum flow using sizing pressure drop.

Quantity Normal: Normal Quantity of fluid valve is required to pass.

Flow C_V: Calculated flow coefficient at normal flow using sizing pressure drop.

Inlet Pressure: Normal inlet pressure with units (psia, psig, inches H_20 press, etc.) Note: Must use *normal* inlet pressure for gas or steam sizing.

Delta P Sizing: Pressure drop (inlet pressure minus outlet pressure with units; psi, inches W.C., etc.). For all flows for sizing of valve port(s).

Maximum shut-off Pressure: The maximum pressure drop in the shut off position to determine proper actuator size [inlet pressure minus outlet pressure with units, psi, inches WC (water column) etc.] when valve is closed.

Temperature, Maximum: Show maximum temperature with units (°F, °C, °K) to establish valve body rating and trim selection.

Temperature, Normal: Normal temperature is required for gas and steam sizing.

Specific Gravity: Specific gravity of flow at normal operating temperature.

Molecular Weight: Molecular weight of fluid.

Viscosity: Show viscosity at flowing temperature with units (c.p.; S.S.U., etc.).

Percent Flash: Percent of maximum flow that will flash to the vapor state due to pressure drop across the valve.

ACCESSORIES

Also include details on any accessories; such as, solenoid valve, limit switches, travel limit stops, air tanks, lock-up in last position relays, etc.

REFERENCES

1. ASME Standard No. 112, 1961, *Diaphragm Actuated Control Valve Terminology*, 14 pp, American Society of Mechanical Engineers, New York, New York.

2. FCI 58-2, 1963, *Standard Measurement Procedure for Determining Control Valve Flow Capacity*, 4 pp, Fluid Controls Institute, Pompano Beach, Florida.

3. FCI 62-1, 1962, *Standard Formulas for Sizing Control Valves*, 8 pp, Fluid Controls Institute, Pompano Beach, Florida.

4. ISA Standard, ISA S20, 1975, *Specification Forms for Instruments, Gages, Thermocouples, Orifice Plates and Flanges, Control Valves, and Pressure Safety Valves*, 48 pp, Instrument Society of America, Pittsburgh, Pennsylvania.

CHAPTER 3

VALVE BODIES

H. D. Baumann, P.E.

Consultant

with portions contributed by

Paul Wing
Masoneilan International, Inc.
Norwood, Massachusetts

and

Rolf Weil *
With Revision
By
William H. Hopf *
Jamesbury Corporation
Worcester, Massachusetts

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* Rolf Weil and William Hopf were employed by Jamesbury Corporation at the time the chapter was written.

INTRODUCTION

A control valve consists of two major sub-assemblies, a valve body sub-assembly and an actuator. The valve body sub-assembly is the portion that actually controls the passing fluid. It consists of a housing, internal trim, bonnet and sometimes a bottom flange. The valve body sub-assembly is a pressure carrying part (a pressure vessel) which must meet all of the applicable pressure, temperature and corrosion requirements in the same manner as a normal pressure vessel.

Body sub-assemblies occur in many shapes and working arrangements depending upon the individual service conditions and piping requirements. Each type has certain advantages and disadvantages for given service requirements and should, therefore, be selected with care.

TYPES OF VALVE BODIES
(Linear Stem Motion)

Globe Styles

The most common control valve body style is in the form of a globe, i.e., it is similar in appearance to a hand valve. Such a control valve body can be either single- or double-seated. *Single-seated* valves, as illustrated by Figure 1, are usually employed when tight shut-off is required, or in sizes of 1 inch and smaller where the partial cancellation of unbalanced forces acting on the valve stem is unimportant as a factor in actuator selection. Tight shut-off in this case usually means that the maximum expected leakage is less than 0.01% of the maximum valve C_V when subjected to an air test with 50 psi upstream and 0 psig downstream pressure.

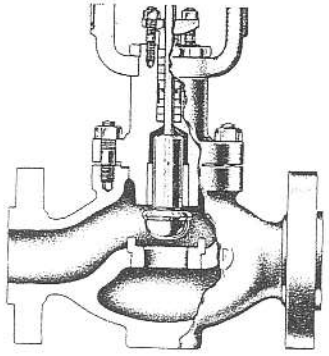


Figure 1. GLOBE TYPE SINGLE SEATED VALVE.
Courtesy Masoneilan International, Inc.

Single-seated valves usually have a top guided construction, i.e., the valve plug is guided within the lower portion of the valve bonnet or top closure. Such top guiding minimizes the effect of the weight of the valve plug and tends to increase resistance to trim vibration. It also allows a somewhat higher flow capacity than top and bottom guided valves for a given orifice size.

A *double-seated valve,* as shown in Figure 2, is generally top and bottom guided. Here the practical leakage figure approaches 0.5% of the rated C_V, because of the fact that it is nearly impossible to close the two ports simultaneously, particularly when thermal expansion provides additional distortion after the valve is installed. However, the advantage of double-seated construction lies in the reduction of required actuator forces, because the hydrostatic effects of the fluid pressure acting on each of the two seats, tend to cancel each other out. Double-seated valves have upper and lower ports of different diameters (to allow withdrawal of the smaller plug through the larger port), which contributes to the unbalanced condition. Additionally, complete cancellation of these forces is not possible, because of the hydrodynamic effects of the fluid that passes the plug contour at high velocity. Fluid passing the lower seat (tending to close the plug) has a tendency to "suck" the plug into the seat, thereby creating a dynamic imbalance between this force and the differential pressure acting across the upper plug area. The dynamic imbalance can reach 40% of the equivalent single seat hydrostatic force. Such a

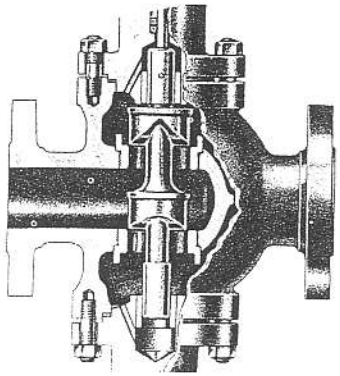


Figure 2. DOUBLE SEATED VALVE - AIR TO CLOSE.
Courtesy Masoneilan International, Inc.

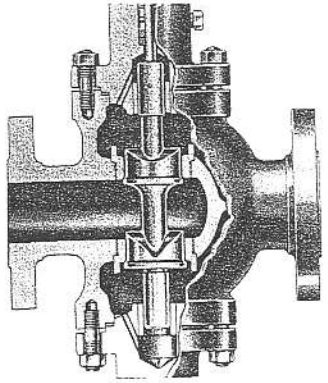


Figure 3. DOUBLE SEATED VALVE - AIR TO OPEN.
Courtesy Masoneilan International, Inc.

double-seated trim is therefore more justly called "semi-balanced".

With the top and bottom guided construction, the stroke direction can be reversed by simply inverting the valve body. This means that when a direct actuator pushes down, as a result of a signal increase, the valve will tend to close as shown in Figure 2. However, with the inverted body (body sub-assembly inverted 180 degrees), air pressure will tend to open the valve, as illustrated in Figure 3.

Removable Bridgewall Globe Valves

The removable bridgewall globe valve is a flangeless design that eliminates the bonnet joint. The removable bridgewall section is sealed to the body wall by the upstream pipeline flange gasket, leaving only the stem stuffing box as a possible leakage path to atmosphere. The plug is assembled through the large open end cavity after removal of the bridgewall section. The flow contour is located in the bridgewall section or what would be the seat ring of a conventional valve.

The removable bridgewall globe valve is shown in Figure 4. It is an economical design, generally well suited for water and steam service at moderate pressures.

Three Way Valves

Three-way valves are a design extension of a typical double-ported globe valve. Here a distinction is made between a valve used for diverting service and one used for mixing or combining service. A typical three-way valve for diverting service

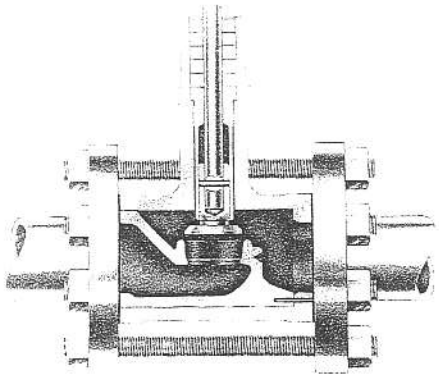


Figure 4. FLANGELESS, BONNETLESS, REMOVABLE BRIDGEWALL GLOBE VALVE.
Courtesy Masoneilan International, Inc.

as shown in Figure 5, has a modified double-ported body with the lower plug seating opposite from the normal shut-off position. Either a direct or a reverse actuator is used, again depending upon the desired fail-safe action. Note that the plug has additional rib guiding in the orifice to compensate for the omission of the lower guide post. The body has a internal bridge to separate the right hand and the lower outlet. A diverting valve might be used for a heat exchanger bypass where the heating medium enters port "C". Part of the fluid leaves port "U" to bypass the exchanger. The remaining portion of the

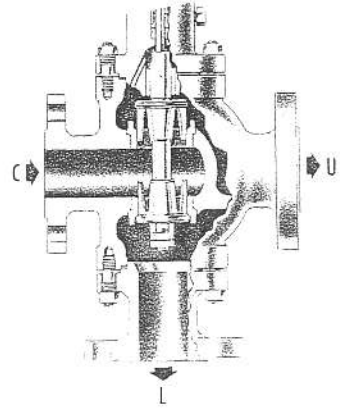


Figure 5. THREE WAY VALVE - DIVERTING SERVICE.
Courtesy Masoneilan International, Inc.

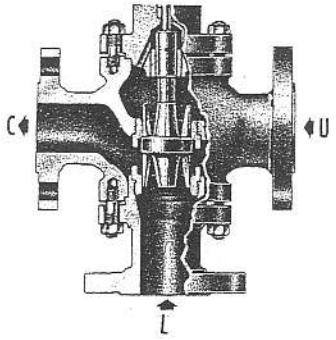


Figure 6. THREE WAY VALVE - COMBINING SERVICE.
Courtesy Masoneilan International, Inc.

fluid then goes to the heat exchanger, through port "L", to heat an independent process stream and, then rejoins the by-pass fluid stream from port "U".

Figure 6 illustrates a three way valve used for combining service. Here, two separate fluids entering port "L" and port "U," respectively, are combined in a desired ratio and leave through the common port "C." The ratio between the amount of fluid coming through either port "U" or "L" is determined by the plug position. An upward movement of the plug decreases the flow passing port "U" and at the same time increases the flow area for port "L." Note again, that skirt guiding is employed to provide additional guiding in line with the upper guide post.

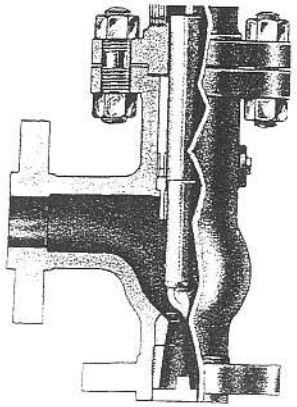


Figure 7. ANGLE VALVE - STREAMLINED BODY.
Courtesy Masoneilan International, Inc.

Casual observation of Figures 5 and 6 might lead to the conclusion that hydrostatic forces acting on either plug are somewhat balanced as they are in valves having a double-seated construction. This conclusion is erroneous, however, since the pressure level in either port (Uand L) could vary considerably and, because the individual plug forces act in opposite directions. It is, therefore, prudent to treat a three-way valve in the same manner as two opposed, single-seated valves for purposes of actuator selection.

Angle Valves

Angle valves are single-seated valves with special body configurations to suit specific piping or flow requirements. A typical angle valve, with streamlined body interior, is shown in Figure 7. This valve was specifically designed for coking hydrocarbons. The streamlined interior passage tends to prevent an accumulation of solids on the body wall. This type of valve has been used, somewhat erroneously in the past, for high pressure-drop service. The underlying thought was to keep the turbulence, that is created by the throttling process, away from the valve's internal parts. However, in doing this, the energy conversion takes place in the downstream pipe, occasionally leading to severe pipe vibration and noise problems. Another disadvantage, in most applications, is the relatively high pressure recovery obtained with the streamlined flow pattern. High pressure recovery means a low cavitation index and severe cavitation under low or moderate pressure drops with liquid media (this is covered in Chapter 6). In addition to the specific application listed above, angle valves may be used in cases where the piping layout does not allow the installation of a globe valve and may be used for handling certain erosive fluids, such as abrasive catalyst material. In the latter, a discharge into the outlet pipe (with flow direction tending to close the plug) prevents erosion of the inside of the valve housing. If cavitation or flashing is inevitable, an angle valve may be arranged to discharge directly into a vessel, or other enlarged fluid volume, thus avoiding damage.

An additional advantage of some angle valves is a self draining design which is of value when handling certain dangerous fluids such as radioactive liquids.

Y-Style Valves

Another valve which is self draining when installed at a certain angle is the

Table 1. RELATIVE FLOW CAPACITIES OF CONTROL VALVES

Valve Type	C_d (1)	$C_d F_p$ (2)	$C_d F_L$ (3)
Double-seat globe	12	11	11
Single-seat top guided globe	11.5	10.8	10
Single-seat split body	12	11.3	10
Sliding gate	6-12	6-11	not available
Single-seat top entry cage	13.5	12.5	11.5
Eccentric rotating plug (Camflex)	14	13	12
60° open butterfly	18	15.5	12
Single-seat Y valve (300 & 600 lb.)	19	16.5	14
Saunders type (unlined)	20	17	not available
Saunders type (lined)	15	13.5	not available
Throttling (characterized) ball	25	20	15
Single-seat streamlined angle (flow-to-close)	26	20	13
90° open butterfly (average)	32	21.5	18

NOTE: This table may serve as a rough guide only since actual flow capacities differ between manufacturer's products and individual valve sizes.

(1) Valve flow coefficient $C_v = C_d \times d^2$ (d = valve dia. in inches)

(2) C_v/d^2 of valve when installed between pipe reducers (pipe dia. = 2 x valve dia.).

(3) C_v/d^2 of valve when undergoing critical (choked) flow conditions.

Y valve. This design is commonly applied in molten metal installations and in cryogenic installations where the cool-down weight has to be kept to a minimum. Figure 8 illustrates the use of such a valve for liquid hydrogen. Here a special insulating jacket is placed around the valve body, to allow the maintenance of a vacuum around the valve housing, to prevent heat transfer. Note that bellows seals are employed to seal the end connection against similar vacuum jacketing on the adjoining pipe. The bellows compensates for thermal distortions and machining tolerances. Y valves have a relatively high flow capacity (C_v) and a low cavitation factor (F_L) as shown by Table 1.

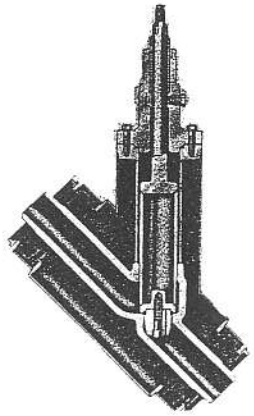


Figure 8. "Y" VALVE - VACUUM JACKETED FOR CRYOGENIC SERVICE.
Courtesy Masoneilan International, Inc.

Split-Body Valves

Certain corrosive services in the chemical industry demand either frequent in-

Such frequent maintenance service is difficult to accomplish with standard globe-type valves, having screwed-in seat rings, for example. Split-body valves were designed to provide easy maintenance. Here the seat ring is clamped between two halves of a valve body, as shown in Figure 9 and can be removed with relative ease. The weight and, therefore, the cost of exotic valve materials is further reduced by the use of slip-on flanges. These flanges, meeting standard ANSI dimensional requirements, can be made of ordinary carbon steel. Each flange locks behind a split-ring and pulls the body end connections against a gasket and the mating line flange.

Note that the yoke connection near the stuffing box, has a similar split-ring connection, that is less affected by corrosion than the more universal threaded retainer, thereby facilitating maintenance.

Stem guiding of the plug, with valves handling slurries or crystalizing fluids, eliminates problems that would be caused by build-up on top of an ordinary plug guide.

One disadvantage of the split-body valves is that welded end connections, a common requirement in power plant service, are not practical because the body halves cannot be separated for removal of the seat ring.

Split body angle valve are available with fluorinated ethylene propylene (FEP) liners for corrosive service in 1 inch and 2 inch sizes rated 150 lb. ANSI class. This valve, as shown in Figure 10, has a temperature range of -20°F to 300°F. The body is self draining. The body valves are bolted with shims so as not to overly stress the liner. Valve plug, seat ring and lantern ring are glass filled TFE (polytetra-fluoro-ethylene) while the stem is a metal that is compatible with the line fluid (usually titanium, tantalum, Hastelloy B® or Hastelloy C®).

Cage Valves

So-called "top entry" or *cage* valves have the advantage of easy trim removal. Figure 11 shows a typical top entry valve with unbalanced, single-seated trim. Valves of this type usually have streamlined body passages to permit increased flow capacity. The inner valve parts, often referred to as "quick change trim," can easily be removed after removing the bonnet, because of the absence of internal threads. Characteristic of all the various cage designs is the fact that a flexible metal gasket, with either asbestos or Teflon filler, is placed between the seat ring and the body bridge. This gasket in turn is

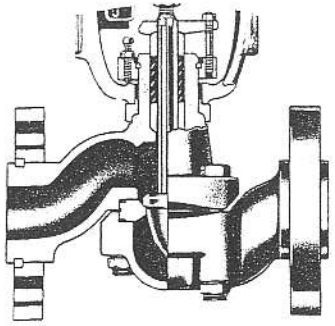


Figure 9. SPLIT BODY VALVE WITH SLIP-ON FLANGES.
Courtesy Masoneilan International, Inc.

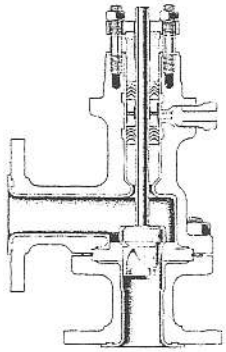


Figure 10. FEP LINED VALVE WITH GLASS FILLED TFE SEAT & PLUG.
Courtesy Fisher Controls Company

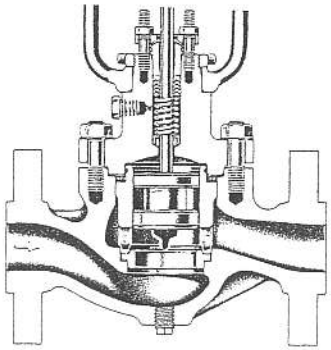


Figure 11. TOP ENTRY VALVE - UNBALANCED
Courtesy Fisher Controls Company

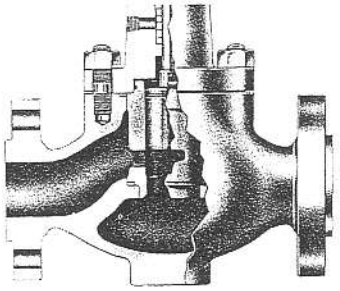


Figure 12. TOP ENTRY VALVE - BALANCED.
Courtesy Masoneilan International, Inc.

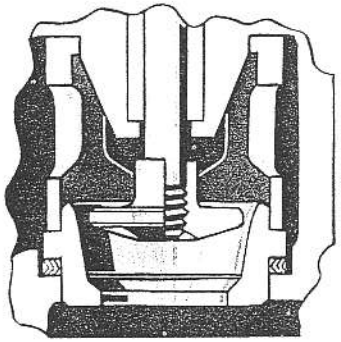


Figure 13. TOP ENTRY VALVE - CONTOURED REVERSIBLE SEAT RING
Courtesy A. W. Cash Company

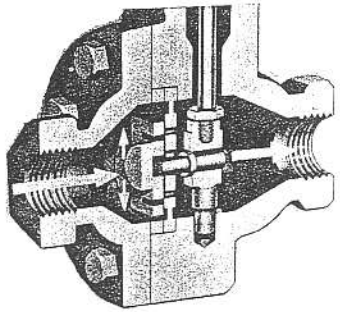


Figure 14. SLIDING GATE VALVE
Courtesy Jordon Valve Division, Richards, Industries, Inc.

compressed by the cage and serves as a spacer between the bonnet and the seat ring.

The cage and seat ring are sometimes designed as one piece. The valve plug then assumes the form of a flat disc, and the flow characteristic is determined by shaped windows, usually cast in the circumference of the cage.

The valve plug can also be made as a piston having a hollow internal passage, that permits the fluid pressure to communicate to both sides of the plug. A sliding seal, located in a groove near the top of the piston, seals the upper plug area against the outlet portion of the valve. This balanced design, as shown in Figure 12, tends to cancel out the hydrostatic forces acting on the plug and leads to a great reduction in the required actuator forces. However, as in double seated valves, a substantial increase in seat leakage must be tolerated.

As mentioned previously, the biggest advantage of the top entry valve is the ease of maintenance. The valve trim can be replaced, if necessary, without removing the valve body from the line. This is an important consideration in this age of trade unions, because maintenance men dealing with pipe flanges usually belong to a different union than the mechanics that service the control valve. This is one of the reasons why this particular valve type has gained wide acceptance, since its quite-recent introduction.

A modified variation of an unbalanced top entry valve is illustrated in Figure 13. Here, the seat ring has an internal contour to provide for an equal percentage flow characteristic in conjunction with the disk type plug and can be inverted 180 degrees to convert from "fail-open" to "fail-close" stroke direction. This valve is stem guided only, which is an advantage for slurry applications.

Sliding Gate Valves

The slide valve approach is another way to reduce actuating forces in control valves. A typical design is shown in Figure 14, which consists of a two-piece housing, clamping a stationary gate that serves as a valve seat. Another "slideable" seal, pressed against the seat by the upstream fluid pressure, serves to adjust the flow rate by gradually uncovering the stationary slots in the seat ring gate. The valve is wide-open when the slots in the sliding member and the stationary gate are completely lined up. Both plates are lapped to light band flatness for tight shut-off. Standard material is chrome-plated,

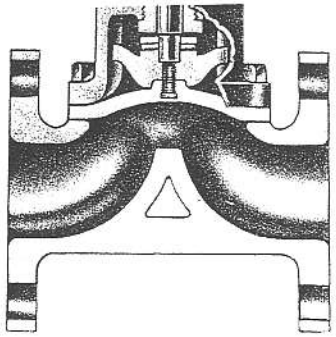


Figure 15. SAUNDERS VALVE
Courtesy Masoneilan International, Inc.

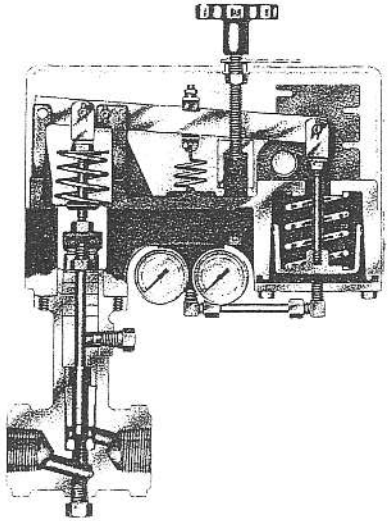


Figure 16. MULTIPLE USE TRIM VALVE.

chrome-nickel type, stainless steel. This type of trim is relatively vibration proof, since the throttling element is pressed firmly against the seat at all lift positions.

Saunders Valves

The Saunders patent valves are used primarily for the control of corrosive fluids, heavy liquids, or liquids having solid particles in suspension. A typical valve, shown in Figure 15, consists of a housing, having a central raised weir and a moveable diaphragm, clamped between the body and the bonnet. This diaphragm is pressed against the weir or central body portion, to provide tight shut-off. The flow characteristic is determined by the distance between the weir and the diaphragm. Advantages are low cost, packless construction, and ease of maintenance. Disadvantages include a bad flow characteristic and high and non-uniform actuator force requirements. In addition, this valve is usually limited to fluid temperatures of below 180° F. If, the valve is to handle corrosive or dangerous fluids, it is prudent to specify a special bonnet with a packing box, to prevent leakage in the event of diaphragm failure.

The valve is available also with glass, plastic, lead or Teflon liners to provide extra corrosion resistance.

Valves Under 1 in. Nominal Size

Accepted practice calls for a minimum body size of ¾ to 1 inch for process control valves. Further reduction in C_V is obtained by using interchangeable trim sets and other specialized design considerations. One example of valves in this class is the 1 inch body size, 1500 lb. ANSI, multiple-use trim valve as shown in Figure 16. It has a built-in rolling diaphragm lever actuator, positioner-manual operator and electric position switches, all housed in a protective case. The cage type trim is compact yet strong.

ROTARY TYPES OF CONTROL VALVES [1]

In recent years there has been a substantial increase in the use of valves broadly classified as "rotary." All types of rotary valves share certain basic advantages and disadvantages. Among the advantages are low weight, simplicity of design, high relative C_V, more reliable and friction-free packing, and generally low initial cost. They are generally not suitable in size, below 1 to 2 inches. Pressure-drop ratings are limited, particularly in the larger sizes, and the operating shaft must be designed to support a fairly heavy side-thrust that is not encountered in reciprocating stem valves. These points

[1] This section contributed by Paul Wing, Masoneilan International, Inc.

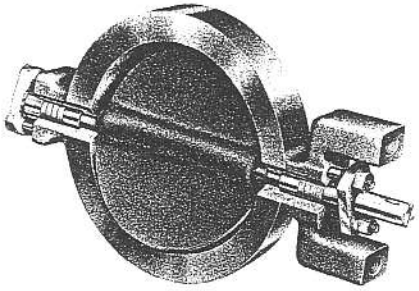


Figure 17. BUTTERFLY VALVE - UNLINED.
Courtesy Masoneilan International, Inc.

are brought out in more detail through the following discussion of some basic designs.

Butterfly Valves

The most common type of rotary valve used for control is the butterfly valve, as shown in Figure 17. The typical application range is in sizes from 2 inches through 36 inches or larger, for low or moderate pressures, or on unusual applications involving large flows at high static pressures, but with limited pressure-drop.

The typical valve, on throttling service, is limited to 60° opening resulting in a C_V rating of about 18 d^2. In all metal construction, the vane to body clearance in the closed position gives a leakage flow that is generally equivalent to the leakage of double-seated valves (0.5% of the rated C_V). Many design modifications are available which minimize leakage, including angle-seating, and the use of piston rings. Because of the simple body design, elastomer inserts can be easily adapted to give tight shut-off within the temperature limits of the insert material. Typical liner materials are neoprene, Buna-N®, or for high temperature and special chemical services, Viton A®. Actuator force requirements are dictated by a combination of two factors; the friction load imposed by side-loading on the shaft, due to the differential pressure; and the dynamic torque induced by the flow around the vane. The dynamic torque acts in the closing direction and, in the conventional design, reaches a maximum at about 70° open.

In designs recently introduced, special shaped vanes, shown in Figures 18 and 19 are used to reduce the dynamic torque and to permit 90° operation for increased capacity on certain installations. Another design modification involves an eccentrically-mounted vane to permit, for example,

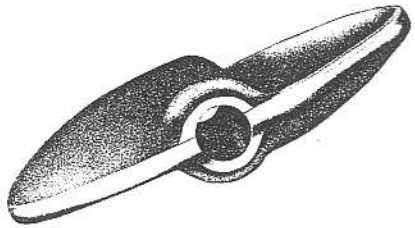


Figure 18. BUTTERFLY VALVE - LOW TORQUE VANE DESIGN.
Courtesy Masoneilan International, Inc.

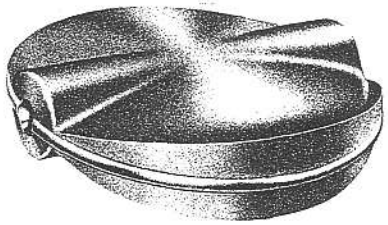


Figure 19. BUTTERFLY VALVE - LOW TORQUE VANE DESIGN
Courtesy Fisher Controls Company

tight closure on cryogenic fluids using a seal of Teflon or Kel-F®, which may be mounted either within the body or on the circumference of the vane.

The most common body design is the flangeless "wafer" type for bolting between line flanges, as shown in Figure 17. ANSI body ratings are used, but the valves are also rated for maximum pressure-drop in the closed position and in the sixty degrees open position.

The rotary shaft and its supporting bearings are an important factor in assuring the success of butterfly valves in control service. Bearing materials range from reinforced Teflon for very low to moderate temperatures and pressures, up to combinations of the hard-facing alloys that permit use at temperatures well above 1000° F. For many years, outboard bearings were specified, almost exclusively. These were located outside the rotary packing box and were not in contact with the process fluid. This simplified the bearing design, but resulted in eccentric loading on the packing. The recent trend has been to inboard bearings, which are now practical, because of the availability of wear and corrosion-resistant materials.

Properly selected, the butterfly valve, generally in sizes 4 inches and above, offers the advantages of simplicity, low

cost, light weight, and space saving, in combination with good flow-control characteristics. For moderate temperatures and pressures, the elastomer-lined valve includes the advantage of tight shut-off. In selecting the valve, consideration must be given to the possibility of cavitation on liquid flow, due to the low pressure recovery coefficient (F_L); the possibility of damage due to water hammer with fast closing on liquid service; and compensation for the effect of pipe reducers in computing the valve capacity, because of the high basic C_V rating.

Eccentric Cylindrical Plug Valves

The valve design shown in Figure 20, is a modification of the plug cock widely used for shut-off service. The eccentrically shaped cylindrical plug cooperates with an off-axis, seating-face within the body, permitting good shutoff, but easy operation, because there is no rubbing contact in the body. An elastomer facing is required for the shut-off feature.

The valve is available in sizes from 1/2 inch to 36 inches, and is available in a wide variety of materials. The temperature range is generally limited, and the pressure ratings are to 250 psi, decreasing as the valve size increases. Relative capacity is high and cost is low. It has been found useful on a wide variety of services, particularly for handling corrosive fluids, highly viscous liquids, or those containing solids in suspension, such as paper stock.

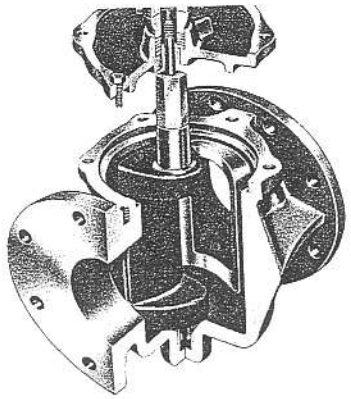


Figure 20. ECCENTRIC CYLINDRICAL PLUG VALVE.
Courtesy Dezurik Corporation

Eccentric-Rotating Plug Valve

A recently introduced valve, of interest, is the eccentrically-rotating plug valve shown in Figure 21. It is available in ratings through 600 lb. ANSI, in sizes from 1 inch to 12 inches. The normal rotary stroke is 50° and an essentially-linear, inherent flow characteristic is obtained with a C_V rating similar to that in the modern, high-capacity, top-entry, cage-guided globe valve. The eccentric motion of the spherical face of the plug reduces the operating torque requirements and permits stable operation when installed for either flow-to-open or flow-to-close. The seating action is positive and the sliding seal-friction associated with ball valves is eliminated. Tight shut-off is obtained with relatively low applied force, because minor misalignment is compensated by flexing of the plug support-arms.

Flangeless construction is used to reduce weight and to allow all ratings of up to 600 lb. ANSI, with a single body machining. A short, plain, bonnet extension, extends the standard temperature range from -320° F to 750° F.

In contrast to other body sub-assemblies, this valve requires a special long-stroke, spring-opposed, rolling-diaphragm actuator. The valve with no modifications, other than an optional stainless steel body, can cover a large percentage of all process control requirements.

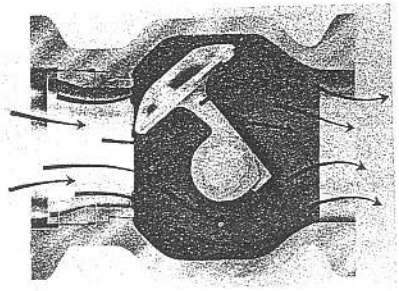


Figure 21. ECCENTRIC ROTATING PLUG VALVE.
Courtesy Masoneilan, Inc.

BALL VALVES[1]

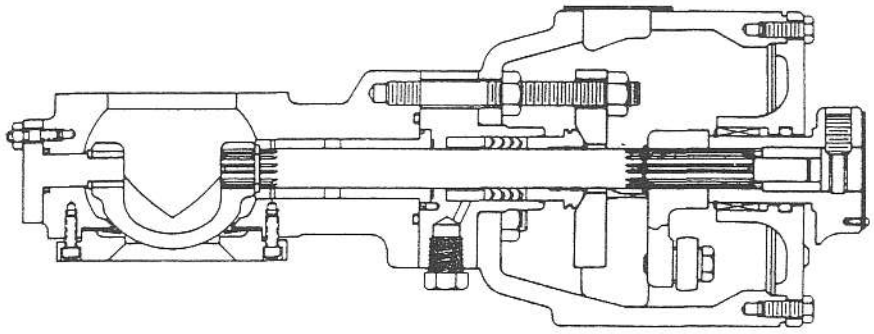
History

Since 1960, the acceptance of a ball valve as a modulating device has grown rapidly. Although the ball valve dates back many years in concept, it was not until the early fifties that an alternative to either metal or rubber seats was generally available. The advent of elastomers and flurocarbons, in particular TFE, allowed the development of resilient seats which give today's ball valve its unique sealing properties. Starting with a single manufacturer in 1950, more than fifty companies now make ball valves, although most of these do not make control valves.

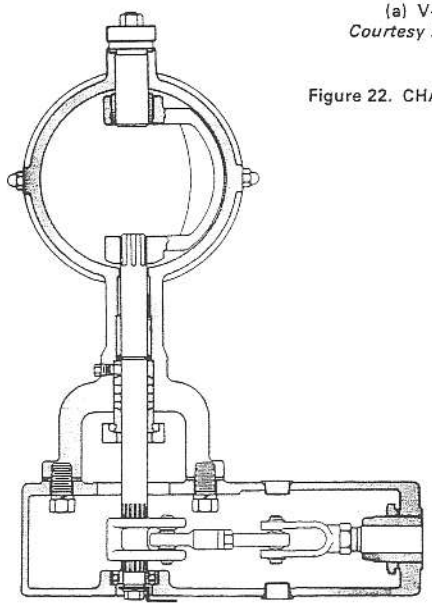
Initially, the ball valve was marketed as a tight shut-off hand valve, but manufacturers soon realized that a quarter turn device could readily be automated either electrically or pneumatically. Early attempts to adapt positioners to the ball valve were relatively crude, but the excellent rangeability and the unique ability of the valve to handle slurries or fibrous materials, without straining or settling, stimulated interest in it. As a result, many improvements were made to the initial ball control valve and a number of modified designs evolved. As the realization came that the ball valve could capture a considerable portion of the control valve market, the major control valve manufacturers began to add ball valves to their line.

This section discusses the various design concepts employed in ball valves and outlines some of their advantages and limitations.

[1]Mr. Rolf B. Weil was the original contributing author. Mr. W. H. Hopf revised the information for the second [illegible]. Both were employed by Jamesbury Corporation when the copy was written.



(a) V-shaped notch.
Courtesy Fisher Controls Company



(b) Parabolic port.
Courtesy Masoneilan, Inc.

Figure 22. CHARACTERIZED BALL VALVES.

Construction

Two distinctly separate concepts developed in the evolution of the ball control valve. One type which evolved from the hand valve, utilizes a complete sphere with a waterway through it. This is referred to as the full ball type. (The term full ball should not be confused with the term "full port" which means that the ball waterway diameter is equal to the pipe I.D.). The ball may be supported entirely by seats or by stub shafts running in bearings.

The second concept which is developed along the lines of the concentric plug valve, utilizes a hollowed out spherical segment or

partial ball which is supported by stub shafts running in bearings. For purposes of clarity, this design will be referred to as a characterized ball valve.

Both designs are quarter-turn, rotary valves, which eliminates alternate wetting and exposure to air of the valve stem. Consequently the valve stem, in the packing box area, does not become as severely corroded as it does in a lift type type valve. This permits the use of packing boxes which are of relatively simple design. The valves are available with threaded, flanged, or flangeless bodies in a large variety of alloys, and may be equipped with either resilient or metal seats.

The characterized ball valve is so named because a simple segment of a sphere which forms a crescent-shaped flow path which produces flows approaching equal percentage for high capacity designs and essentially a linear characteristic for low capacity designs. In order to obtain other characteristics, the leading edge of the segment must be contoured as in the V-notch and parabolic port designs as illustrated in Figure 22a and 22b. The plug consists of a 120° spherical segment which is hollowed out and supported at either side by a stub shaft running in either bushings or bearings. Sealing is effected by a flexible metal or synthetic seal ring, which is pressed against the outside surface of the segment by a retainer ring. Shut-off capability is assisted by line pressure behind the seal ring, while the stub shafts hold the ball segment tightly in place. A cross-section of a characterized ball valve is illustrated in Figure 23.

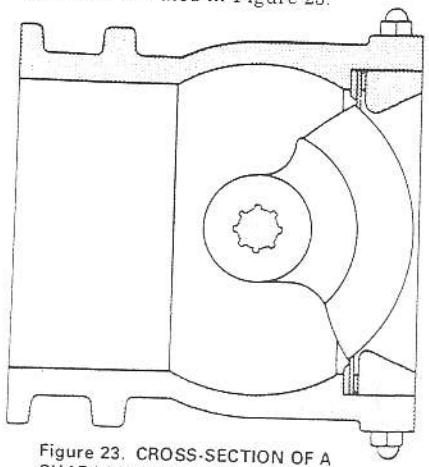


Figure 23. CROSS-SECTION OF A CHARACTERIZED BALL VALVE.
Courtesy Masoneilan, Inc.

The full ball valve as shown in Figure 24 differs from the characterized ball valve in that the closure member is a complete sphere with a waterway.[1] This construction results in two openings or ports, one at each end of the waterway, which causes a "double orifice" effect. The advantages of this concept are:

1) It divides total pressure drop into two approximately equal steps (directly analogous to the classic pressure divided circuit widely used in pneumatic instrumentation). This two step pressure reduction offers the advantage of minimizing waterway velocity with less tendency to strain fluids in slurry applications.
2) It provides an inherent equal percentage characteristic without modification of port shape.

There are two basic types of full ball valves. One is referred to as the "floating ball" where the ball is totally supported by a pair of resilient seat rings. In this design, the stem to ball connection will allow line pressure to assist sealing by moving the closed ball into the downstream seat. Accurate control of angular tolerances between ball and stem, which is a slip fit, must be maintained to prevent hysteresis in the valve. A cross section of a floating full ball valve is shown in Figure 25.

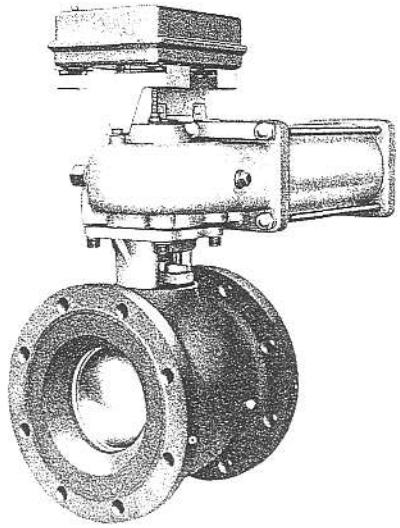


Figure 24. EXAMPLE OF FULL BALL VALVE.
Courtesy Jamesbury Corporation

[1] Editors Note: The term 'waterway' is commonly used to describe the hole in the ball of full ball valves. This differentiates it from the term 'port' which is used for characterized ball valves and is the orifice formed between the ball and the seats.

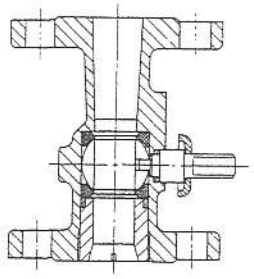


Figure 25. CROSS-SECTION OF A FULL BALL VALVE.
Courtesy Jamesbury Corporation

The other type of full ball valve is called the "trunnion mounted" full ball, where the ball is supported by trunnion bearings or by tightly secured stub shafts running in bearings similar to the characterized ball valve. There may be either two seats or a single seat where sealing is usually assisted by line pressure. There is reduced torque on this valve when it is subjected to higher pressures and the need for controlling angular tolerances between ball and stem are eliminated. A trunnion mounted full ball valve is shown in Figure 26.

A number of ball valve designs have evolved to handle varying types of services. "full-port" valves are available (normally in full ball valve construction) where the diameter of the ball waterway is identical to the bore of the pipe. This type is seldom used as a modulating device except where service conditions require that lines be "pigged" or rodded out. Three-way full ball valves are available in special configurations for diversion or blending applications.

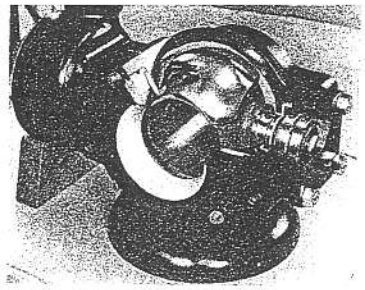


Figure 26. EXAMPLE OF A TRUNNION-MOUNTED FULL BALL VALVE.
Courtesy Fisher Controls Company

Generally, the ball valve consists of a relatively small number of parts, but it must be removed from the line to replace soft or hard trim.

Actuator Requirements

The ball valve is a device in which the imbalance forces are mostly offset by the frictional forces between the ball and the seal ring. When the frictional forces are relatively high, piston-type actuators are normally used to provide the necessary power. These may be either double-acting or spring-return types. The long stroke of a piston type actuator allows the use of a long-moment arm, thus providing efficient conversion of thrust to torque. Piston actuators also allow the use of higher operating air pressures, thus reducing the required diameter of the cylinder. For modulating service, a piston actuator requires a positioner.

The piston actuators that are used vary widely in design, from those which are basically linear and use external conversion to rotary motion, to those which are specifically designed for rotary service and utilize internal conversion of linear to rotary motion by means of a crank arm, rack and pinion, or scotch yoke mechanism. Rotary vane actuators are also used for this service. The various designs include pistons which are rod or skirt guided, and piston seals which may be O-rings, U-cups, or rolling diaphragms.

Where the valve seating frictional loads are low, the use of spring and diaphragm type actuators is becoming more common. If this type of actuator is used with a valve positioner, full output capacity can be realized allowing this device to be used on certain types of valves to 14" nominal size. Use of these devices without a valve positioner limits the size of the spring package which in turn limits the available output torque of the actuator. This tradeoff may result in an actuator size for a particular requirement that is more costly than a smaller spring and diaphragm actuator with valve positioner. Chapter 9 contains more information concerning valve positioners.

Electric actuators have been used for two-position operation, almost from the inception of the ball valve. Because of the low imbalance forces, these electric actuators require only a simple spur gear train and motor brake system, making them relatively inexpensive. Electronic positioners have been developed for use with these actuators to provide total electric modulating ball valves.

Characteristics Of Ball Valves

In discussing characteristics, it is necessary to treat the full ball valve and the

characterized ball valve separately, because the characterized ball valve is a single orifice device, whereas the full ball valve has two orifices in series.

As the full ball valve is moved from open to closed position, the shape of each orifice goes from circular to eliptical. In the full-open position, each orifice is the same size as the waterway in the ball, making the effect similar to that of a single nozzle. As the valve is closed, the orifices become smaller in area than the waterway through the ball, and begin to have the effect of two independent orifices in series, with the waterway acting as a partial recovery area. The resultant inherent characteristic is essentially equal percentage. Changing the shape of the waterway in a full ball valve is not a practical method of changing the valve characteristic; therefore, where a different characteristic is desired, it is normal practice to alter positioner response by using a series of interchangeable cams in the feedback mechanism.

In the characterized ball valve, the leading edge of the ball segment is contoured to achieve the desired characteristic. A segment cut directly from a sphere, without any modification, would provide a convex leading edge to the plug, resulting in a port which is circular in the open position and is crescent shaped in the partially-closed position. The basic characteristic of this shape of plug is linear; however, as an equal percentage characteristic is usually more desirable, some form of characterization is required. The most common approach is to contour the leading edge of the plug, making it concave rather than convex. In Figures 22a and 22b, examples are illustrated with a V-shaped notch and a parabolic shaped contour in the leading edge of the plug.

Capacity Of Ball Valves

Ball valves have the highest flow capacities of any commonly used modulating valves. Their flow coefficient is in the order of two to three times that of an equivalent size globe body. It is common to use a globe valve that is one pipe size smaller than the line in which it is installed; however, a ball valve for the same service could be one-half of the line size. While this offers some obvious economic advantages, somewhat greater care must be given to system design when applying ball valves.

The ball valve will begin to show signs of cavitation and will reach choked flow at considerably lower differentials, relative to the inlet pressure, than will a globe type body. Its high flow capacity and the well defined vena contracta produced by a straight-through valve design are contributing factors. Typically, a wide-open ball valve, in service where vapor pressure is insignificant, may cavitate at a differential roughly equal to three-tenths of the inlet pressure, expressed in psia. As the valve is closed, the critical cavitation index increases, allowing the use of differentials as high as eight or nine-tenths of the inlet pressure. As a consequence, in some systems it may be necessary to install a ball valve of somewhat larger size than that indicated by calculations made for non-cavitating conditions. This allows the use of the higher differentials, because the larger valve would be operating closer to the closed position. Obviously, in this type of system, some of the high capacity advantage is lost. Whereas reduced trim is normally available in globe pattern valves, it is not normal practice to provide this capability in ball valves.

Applications And Limitations Of Ball Valves

The ball valve found its initial application primarily in the pulp & paper industry where its straight-through flow path, tight-shut-off capability and ease of operation were distinct advantages in the handling of fibrous material such as pulp stock. It has remained the most suitable type of valve for handling this and many other types of slurries, because it mimimizes the tendency of these materials to either settle or strain. It has been accepted wherever tight shut-off is a necessity or where the economics of its high flow capacity justify its use. It is available in a wide variety of materials, with either resilient or metal seats.

The limitations of the valve depend largely on the type of seating material used. Resilient seats have essentially no leakage, but limit the temperature capability to that at which the resilient material begins to soften under load. Metal seats increase the temperature capability considerably, but have a higher leakage rate. Current designs are available with allowable differentials in modulating service of up to 1,000 psi.

SPECIAL CONTROL VALVES[1]

In a basic control valve handbook, it is impractical to do more than suggest the enormous variety of body and actuator configurations used on special applications. Two classifications which are of general interest, are discussed in this section; namely, low-flow and low-temperature control valves. A brief treatise on some other valve designs concludes this section.

[1] This section contributed by Paul Wing, Masoneilan International, Inc.

Low Flow Control Valves

While there is not a sharp line of demarcation, the term *low-flow* will be used in this discussion for C_V ratings of 0.05 or less. There are many applications in industrial systems calling for either small, accurately controlled, quantities of material at ordinary pressure levels or somewhat larger quantities having very high pressure-drops, where the calculated C_V is very small. These applications either may be in pilot plants where a process is being conducted in miniature, or they may be in a commercial plant where, for example, a small amount of some reagent is being added to a large amount of process fluid. The theoretical C_V in the literature ranges upward from 0.00001 and as the accompanying Figure 27 shows, covers a wider range than the equivalent for valve sizes 1 inch through 16 inches.

In considering these low flow applications, the selection criteria must be quite different from those used for conventional control valves. For example, there is seldom any friction loss in series with the control valve nor is there an unfavorable pump-head characteristic. There is no real requirement for an equal percentage characteristic. The principal design problem is one of getting down into the desired flow range in a practical manner. C_V ratings become more and more theoretical as they become smaller. They are useful as a capacity index, but are of doubtful accuracy or repeatability, because in many cases, the flow through the principal restriction may be either laminar or a combination of laminar and turbulent and this flow mixture may change radically with valve stroke or with flowing conditions.

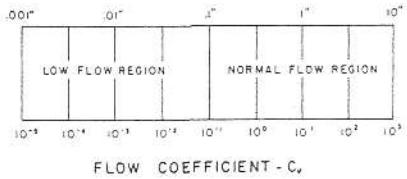


Figure 27. LOGARITHMIC PLOT OF CONTROL VALVE C_V RANGE SHOWS WIDE COVERAGE REQUIRED ON LOW FLOW APPLICATIONS.

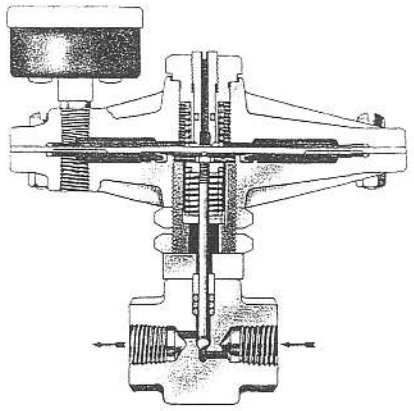


Figure 28. LOW FLOW VALVE WITH BALL PLUG AND ACTUATOR WITH VARIABLE STROKE FOR WIDE C_V RANGE.
Courtesy A. W. Cash Company

From the design standpoint, the physical dimensions of the required flow restriction is very small indeed. An additional problem is imposed by the fact that a great many low-flow applications involve high pressure-drop with the accompanying erosion problems.

In standard valve design, capacity is largely determined by the diameter of the orifice. As the capacity is reduced, a practical minimum diameter of about 1/16 in. is reached. Such an orifice would have a theoretical rated C_V of approximately 0.06 or near the upper limit of the low flow range. All further reductions must be obtained either by radically reducing the plug stroke or by restricting the port opening.

An interesting example of the short stroke approach is shown in Figure 28. In this design, a synthetic sapphire ball modulates the flow through a small hardened metal orifice. A specially designed actuator, with variable pneumatic feedback, permits the stroke to be adjusted so that a 3-15 psi signal will cover a very wide C_V range.

For a short-stroke actuator, which is suitable for smooth throttling action on very high pressure service, another approach has been to rotate a threaded plug stem through part of a revolution, by means of an arm driven by a pneumatic actuator. Two threads of different pitch may be used in such a system to further limit the stroke, which may be reduced in this manner to as little as 0.005

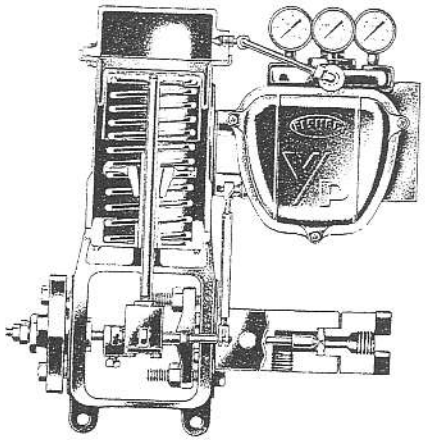


Figure 29. SPECIAL ROTARY ACTUATOR GIVES LOW STROKE BY USING DIFFERENTIAL THREAD DESIGN.
Courtesy Fisher Controls Company

in. An example of this design is shown in Figure 29. Another version using this basic approach, but with a mechanical variable ratio linkage, is shown in Figure 30. In this unit, the maximum stroke of the valve plug is 0.188 in., a 4 to 1 reduction over the 3/4 in-rated actuator stroke. This may be reduced to a minimum of 0.010 in., by adjustment of the linkage. This allows field adjustment for a wide range of capacity, as well as providing a high operating force for high pressure-drop operation. Both of these specialized valves may use a simple cone-type plug. For very low flows, a precision-made, tapered pin is used, operating in a tapered hole.

For less exacting services, tapered needles or pistons with milled slots or notches are generally used in orifices ranging from 1/8 to 1/4 in. in diameter. Valve stroke for this type is in the order of 1/2 in. Such low flow trim may be installed in conventional globe valve bodies for industrial applications, but it is possible to reduce both the body and actuator size substantially as shown in Figure 31. These valves are light in weight, compact, and ideally suited for pilot plant work.

As already mentioned, the C_V rating on low flow trim has less and less meaning as the size decreases. Fortunately, the useable flow range obtained in the valve

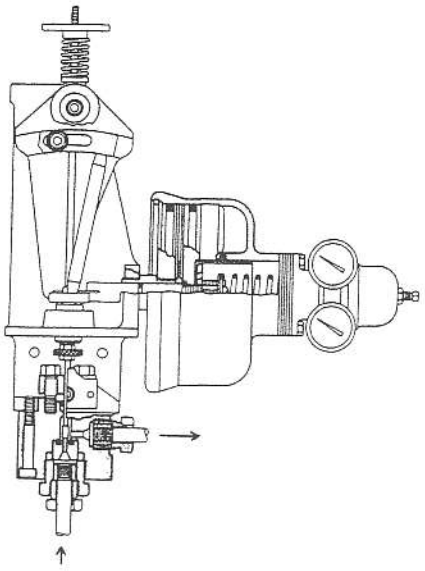


Figure 30. VARIABLE MECHANICAL LINKAGE GIVES MORE THAN 10 to 1 STROKE RANGE FOR APPLICATION FLEXIBILITY.
Courtesy Masoneilan International, Inc.

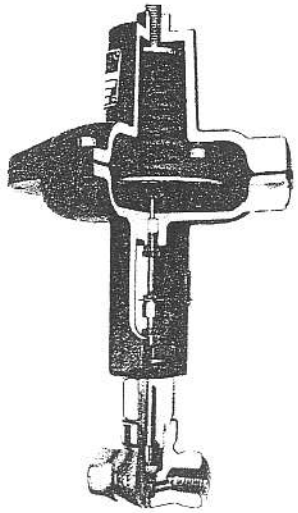


Figure 31. COMPACT VALVE AND ACTUATOR WITH PISTON OR NEEDLE TRIM IS IDEAL FOR PILOT PLANT WORK.
Courtesy Precision Products and Controls, Inc.

orifice is very large. In other than adjustable stroke valves, valve trim can be changed easily and, if a number of similar valves are in service, a few sets of trim can be maintained without excessive cost since all are interchangeable in a given body.

The required C_V is calculated by the conventional formulas and then the next largest-size trim, available in the particular type of valve, is chosen. If the trim is too small in actual operation, a capacity increase may be possible either by simply increasing the valve stroke or by slightly modifying the plug.

Cryogenic and Cold Service Control Valves

When control valves are applied at operating temperatures that are substantially below the freezing point, special precautions must be taken and, in certain cases, special designs are required for satisfactory operation. The principal problem is the proper selection of materials of construction, particularly on moderately cold service (to -150° F). Cryogenics is the term applied to processes operating at temperatures below -150° F, where a number of additional factors must be considered.

In the temperature range from -20° F to -150° F, special impact-resistant, carbon steels are used for the pressure containing parts. The commonly used materials are carbon steel (grade LCB) from -20 to -50° F and 3.5% nickel steel (Grade LC3), to -150° F. These valves are generally equipped with a plain extension bonnet. In this temperature range, the primary objective is to reduce the influx of heat to the system to a reasonably low value, and to prevent the packing box area from getting so cold that frost and ice form on the upper part of the bonnet and on the valve stem above the packing box. The simple extension bonnet is generally installed in the upright position to minimize heat transfer in the operating fluid itself.

At cryogenic temperatures, materials of construction now exclude the carbon steel group and include austenitic stainless steels, bronze, and monel. Valves operating at this temperature range are generally handling extremely cold gases or liquefied gases. Among those commonly encountered are air, nitrogen, oxygen, hydrogen, and helium. Properties of some of these fluids are tabulated in Table 2. For these services, a number of factors must be considered in addition to the material of construction of the valve's pressure containing parts.

Very special attention must now be paid to the design of the extended bonnet. Even though the body material may be bronze, the extension would generally be made of austenitic stainless steel to minimize heat transfer. Care must be taken that liquefied gases cannot become trapped in the bonnet space, where subsequent vaporization on shut-down could generate dangerously high pressures.

In all cases, the valve must be heavily insulated. The process equipment piping and valves in the cold section of the plant

Table 2. PROPERTIES OF CRYOGENIC FLUIDS

Gas	N_2	O_2	CH_4	H_2	*He*	F_2
Density, NTP* lb./cu. ft.	0.0724	0.0827	0.0415	0.0052	0.0103	0.0982
Boiling Point 1 atm. ° F.	-320.4	-297.4	-258.7	-423.2	-452.1	-306.6
Vapor Density at B.P. lb./cu.ft.	0.288	0.296	0.111	0.084	1.06	—
Liquid Density at B.P. lb./cu.ft.	50.41	71.27	26.46	4.37	7.62	94.2
Heat of Vaporization at B.P. BTU/lb.	85.2	91.7	219.2	193	8.8	74.1
Critical Temperature ° F	-232.8	-181.1	-116.5	-400.3	-450.3	-200.2
Critical Pressure, psia	492.3	736.9	673.1	187.7	33.2	808.3

*NTP - Normal Temperature and Pressure, 70° F and 14.696 psia.

are often installed in a "cold box." The valve in this case may have an exceptionally-long, plain, extension bonnet which may be installed in the horizontal position. A special valve design, such as that shown in Figure 32, is often used in this case. A large-diameter bonnet section is fabricated from stainless steel and brought out through the wall of the cold box. This permits removal of the valve trim, including the plug and seat-ring, without disturbing the valve body, which may be welded into the system to minimize the possibility of leakage within the cold box itself. Within the extension at the lower end is a Teflon, chevron seal which acts as a check valve, minimizing leakage of the cryogen into the bonnet section, yet permitting the release of any gas that is generated. In this way, the heat flow into the system is minimized.

A mounting flange is usually provided at the outer end of the extension bonnet, which cooperates through a flexible diaphragm with a larger flanged opening in the cold box itself.

Figure 33 illustrates a very special cryogenic valve that has a vacuum insulating jacket enclosing the entire body and body extension. Stainless steel expansion bellows are provided on the jacket to eliminate the piping strain that is caused by differential expansion. Note that a thin-walled cylinder is used instead of the usual stem extension to provide a long conductive path and to minimize the vapor space between the stem and the body cylinder. Radiation losses become an important factor and are minimized by using an electro-polished valve exterior and an aluminum foil barrier. In this kind of valve, it is important to hold the weight to a minimum, but at the same time, to provide a high relative capacity, because low mass minimizes the boil-off on cooldown. Similar designs are available in a "Y-pattern" configuration to further reduce the head loss, particularly for on-off service.

Rotary valves, also equipped with an extension bonnet, are often specified for cryogenic service where tight shut-off is required. Figure 34 shows a special design, with an offset vane and a spherical seating surface, which gives good shut-off using a special Kel-F® lip-seal insert and yet it minimizes the torque required for breakout. In the larger sizes, the low mass of the valve is an important factor.

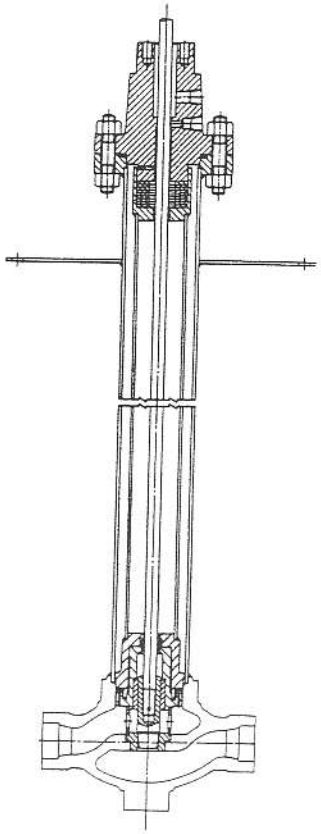


Figure 32. "COLD BOX" VALVE WITH PLUG AND SEAT RING REMOVABLE THROUGH THE BONNET.
Courtesy Masoneilan International, Inc.

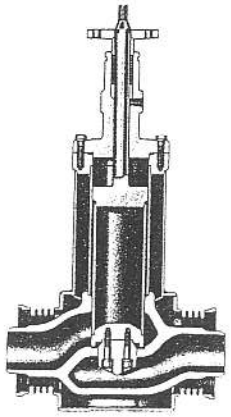


Figure 33. VACUUM JACKETED CRYOGENIC VALVE FOR SERVICE ON LIQUID H_2, He OR OTHER VERY COLD FLUIDS.
Courtesy Masoneilan International, Inc.

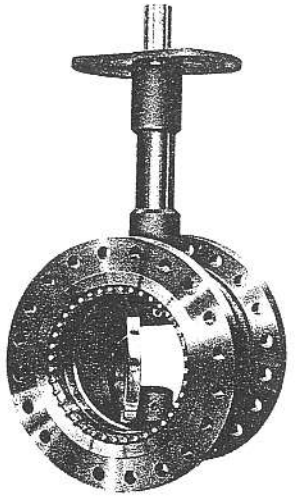


Figure 34. OFFSET VANE CRYOGENIC BUTTERFLY VALVE, WITH ELASTOMER LIP SEAL FOR TIGHT SHUT OFF.
Courtesy Royal Industries

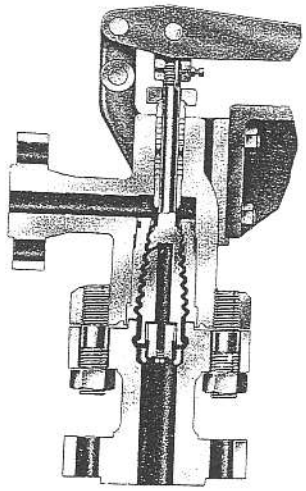


Figure 35. HIGH PRESSURE LOW NOISE, ANGLE VALVE WITH LABYRINTH TYPE SEMI-BALANCED CONE PLUG.
Courtesy Masoneilan, Inc.

Generally, requirements for cleanliness must be more rigid for cryogenic valves, during manufacture and, particularly, during assembly.

Special attention is required in the selection of lubricants, which may be used when assembling parts, to prevent galling. Many lubricants either solidify or become brittle and many may also be completely incompatible with the flowing fluid. A number of compounds have been developed for this service and are readily available, commercially. Very special attention is required, for example, when handling liquid fluorine, which is self igniting on contact with any hydrocarbon. A more common problem exists in the handling of oxygen, either at ambient temperature, or at cryogenic temperatures. In addition to the possibility of self-ignition in contact with certain contaminents, the oxygen supports combustion and a fire, if started, will consume the entire piping system. Most companies, that use cryogenic liquids, have detailed specifications on cleaning and degreasing procedures.

Low Noise Control Valves

Control valves designed for quiet operation are of special and rapidly growing interest. The Walsh-Healey Act, in the United States, defines the permissible noise level for continuous human exposure at 90 dB. Most control valve noise is generated on compressible fluid service as a result of the conversion, into sound, of a portion of the energy dissipated in the throttling process. A general measure for estimating purposes is that, if the product of the C_V and the inlet pressure exceeds 1000, the valve will be excessively noisy under critical pressure drop.

The two principle factors in designing for low noise, are fluid velocity and noise frequency. Noise energy level increases at up to the 8th power of the velocity below sonic, indicating the great importance of this factor. An increase in average frequency at the same noise energy level can also be beneficial, because of greater attenuation of the higher/frequencies through the valve body and the downstream pipe wall.

A low noise design is shown in Figure 35, using a series of steps which force the flowing fluid to change direction continously causing high turbulence and friction energy loss. Compensation for change of fluid density with pressure is provided by increasing the plug and seat diameter toward the downstream section. The resultant cone shaped assembly produces a nearly-constant fluid velocity through the entire throttling process. The maximum velocity attained by this assembly is seldom higher than 300 ft/s, giving a substantial reduction in noise level and improved service life.

To give fail-safe action and, to reduce actuator thrust requirements, partial internal balancing can be provided. This, in combination with short valve stroke and

a conventional spring diaphragm actuator, allows application at pressure drops to 6000 psi. An interesting version of this design using a conventional double-seat, globe body is shown in Figure 36. Valve capacity of approximately 40 percent of the normal rated maximum is attained.

In yet another design, shown in Figure 37, the throttling passages through the valve are produced by stacking a number of annular discs, which are either etched or punched to produce flow passages. The fluid path is tortuous, similar to that in Figure 35, and compensation for expansion on compressible fluids is handled by an increase in the number of flow channels. The valve plug is of simple cylindrical shape and a long valve stroke is required.

Conventional single-seat globe valves, particularly those of ported cage designs, can be modified to increase peak noise frequencies resulting in less audible noise. In all designs, if the valve outlet velocity either reaches or exceeds the speed of sound, objectionable noise may be generated in the down-stream piping. On high-drop applications, therefore, special down-stream treatment is required. This may take the form of one or more special throttling plates, designed to both reduce the velocity in successive stages and to increase frequency. Various commercial silencers may also be used. Acoustical insulation is beneficial to limiting noise in a specific area; however, objectionable noise may still be produced beyond the insulated zone.

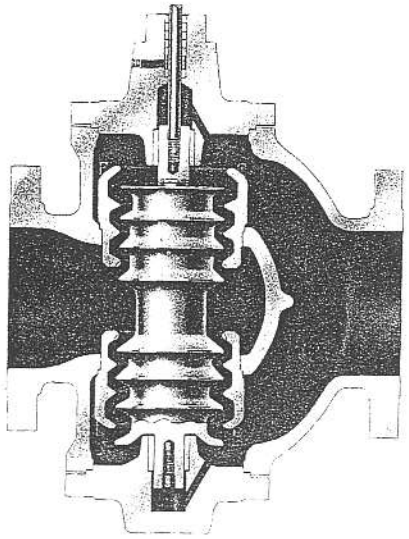


Figure 36. STANDARD DOUBLE-SEAT VALVE WITH MODIFIED PLUG AND SEAT FOR LOW NOISE SERVICE.
Courtesy Masoneilan, Inc.

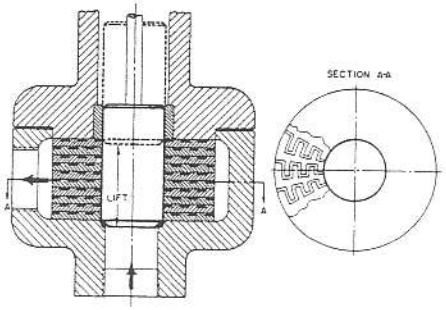


Figure 37. STACKED DISC, LONG-STROKE VALVE DESIGN REDUCES VELOCITY AND NOISE.
Courtesy Control Components, Inc.

Pinch Valves

Pinch valves as shown in Figure 38 are straight-through valves. They are lined with an elastomer or TFE tube which is collapsed by a pinch bar or clamp for throttling and closure. A segmented metal body may surround the tube or the tube may be exposed. In some designs, a lifting mechanism is attached to the top and bottom tube surfaces to insure opening in vacuum service; however, line pressure above 15 psi will readily return the tube to its full port round section.

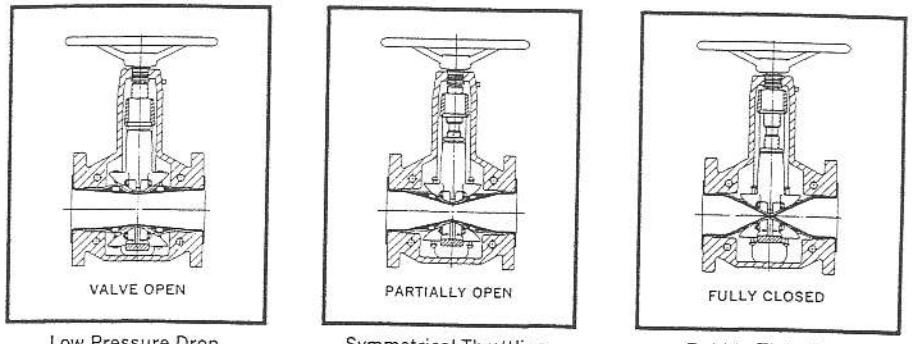
The fluid contacts only the liner since there is no packing box; thus, the valve serves well in corrosive service. TFE liners vary from 3/32 to 5/32 in. wall thickness, whereas rubber liners are heavier and are often reinforced.

These valves are normally rated from full vacuum to 150 psi and TFE liners are suitable to 350°F. Size range is 1 inch to 12 inches. They handle slurries well since any build-up of material in the seat is broken up and washed out on closure. Rubber lined valves close effectively around particles suspended in the line fluid.

Sleeve Valves

The sleeve valve design as shown in Figure 39, has a slotted tube with the slots divided by a bridgewall across the flow axis. Flow must pass outside of the tube, around the bridgewall and into the tube again. A sleeve type diaphragm, the only moving part, seals the slots until upstream line pressure exceeds control pressure in the annular cavity surrounding the outside of the sleeve. The sleeve is lifted and flow travels through the slots and around the bridgewall. Line pressure must also overcome the hoop stress of stretching the sleeve clear of the slotted tube to achieve maximum flow and this is the reason for stating minimum differential pressure ratings. The sleeve valve is very effective in gas flow control and by the addition of a gas pressure regulator to control pressure to the sleeve diaphragm becomes a main line pressure regulator for either reducing or relief service.

Sleeve valves are built in sizes from 1 inch to 12-inch with maximum presure differentials ranging to 1200 psi. Minimum pressure differential for full flow is 45 psi for the 1 inch and 9 psi for the 12 inches size.



Low Pressure Drop — Symmetrical Throttling — Bubble-Tight Seat

Figure 38. PINCH VALVE OPERATION.
Courtesy of Resistoflex Corporation

Digital Valves

With digital valves, the final control element in a closed loop control system may be more readily interfaced with a computer. One design as shown in Figure 40, consists of a series of on-off valve elements installed in a common body. These elements are a computer operated relay system. When open, each element offers a different orifice area, thus, by a binary bit code (on-off) the combination of orifice areas is made to give the desired flow. An eight bit valve (eight valve elements) has a design rangeability of 255:1; i.e., the orifice area of the smallest element is 1/255 of the sum of all the areas.

The construction of the valve shown is coaxial. The incoming fluid enters an annular ring where the valve elements feed it into the downstream bore through a series of orifice cages having plugs that are pressure balanced (except for the stem area that is being acted upon by upstream pressure). It's application is limited to clean fluids and moderate temperatures.

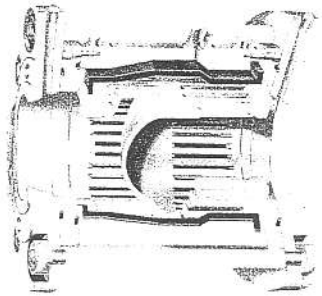


Figure 39. SLEEVE TYPE VALVE USED IN GAS FLOW AND PRESSURE REGULATION.

Other Valve Designs

Special valve designs evolve either through the ingenuity of mechanical designers or in a trial and error process to meet a particular field requirement. Many, such as the pneumohydraulic,

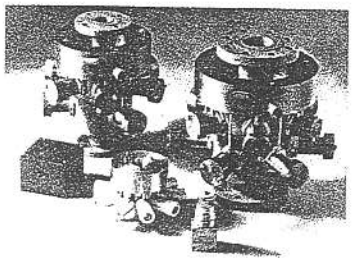


Figure 40. DIGITAL CONTROL VALVE. NOTE ANNULAR AIR SUPPLY RING AND PROJECTING CONTROL VALVE ELEMENTS

axial-flow valve shown in Figure 41 are quite unconventional in the approach to the problem. Others, such as the butterfly valve shown in Figure 42, with a two-piece body and an offset vane shaft to permit the use of an inflatable seal in the body itself for critically tight shutoff, are specialized adaptations of the basic types already shown.

One classification generally labeled *special* includes the valves designed for high-static and high pressure-drop service. These may be divided into two major classes:

1) Valves for the process industries in service such as ammonia letdown, polyethylene letdown, and urea letdown.
2) High pressure-drop, steam and water valves on large power boilers.

For the process industries, the most common valve configuration is shown in Figure 43. For most services such as ammonia letdown and urea letdown, temperatures are moderate and pressures are in the range of 3000 to 10,000 psi. Valve body-size seldom exceeds 3 inches and the orifice diameter 1 to 1.5 in. The design problem is one of pressure integrity and material compatability with the process fluid.

The angle-style body is used either flow-to-open or flow-to-close, depending on the process requirements. In the manufacture of polyethylene, for example, a control

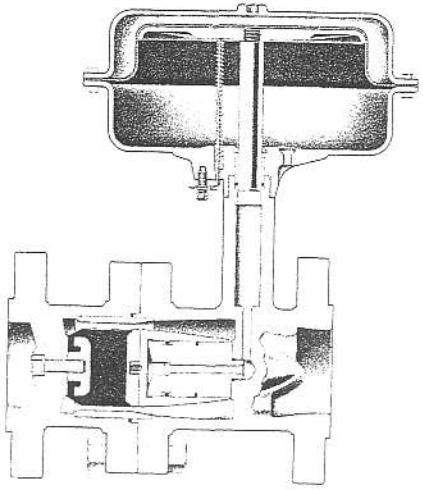


Figure 41. PNEUMOHYDRAULIC VALVE WITH AXIALLY MOUNTED PLUG.
Courtesy American Meter Company

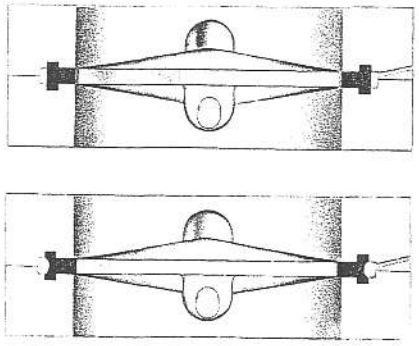


Figure 42. DIAGONALLY SPLIT BUTTERFLY BODY PERMITS USE OF INFLATABLE SEAL FOR CRITICALLY TIGHT SHUT OFF.
Courtesy Fisher Controls Company

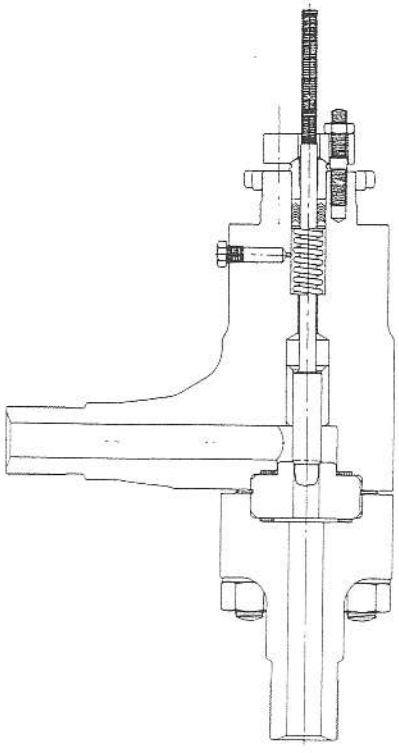


Figure 43. FORGED ANGLE VALVE FOR MODERATELY HIGH PRESSURE PETROCHEMICAL SERVICE.
Courtesy Fisher Controls Company

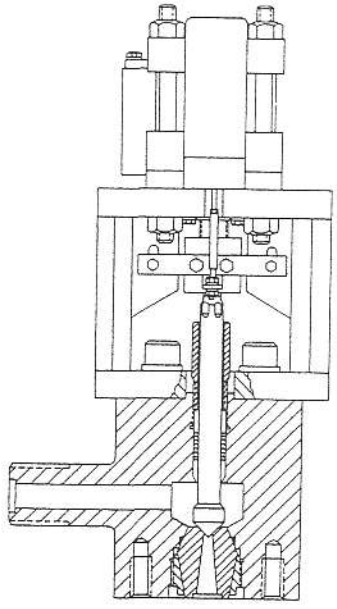


Figure 44. HIGH PRESURE FORGED ANGLE VALVE WITH PRESSURES UP TO 50,000 PSI.
Courtesy Masoneilan International, Inc.

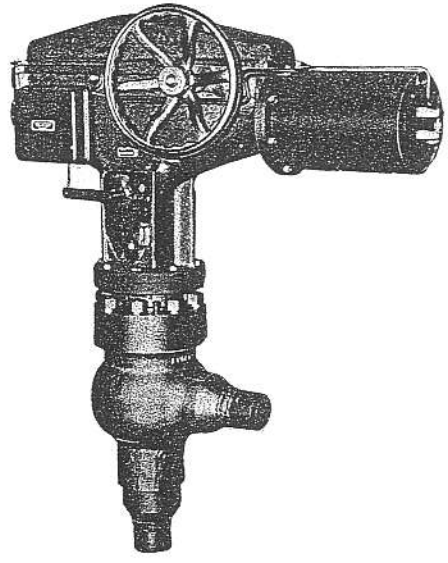


Figure 45. LARGE HIGH PRESSURE SINGLE SEAT VALVE ESPECIALLY DESIGNED FOR POWER PLANT SERVICE.
Courtesy Bailey Meter Company

valve is required, operating at an inlet pressure of 30 to 50,000 psi, and extremely high pressure-drop. A typical design is shown in Figure 44, again using the angle body which permits the use of hammer forgings or barstock. Such valves often incorporate special safety features, such as a back seat in the bonnet section, to minimize the hazard of a packing failure. Body-joint, end-connections are generally established on a custom basis. This is particularly true in operating pressure ranges above 10,000 psi.

On high-pressure power-plant applications, the valves are much larger and the temperatures are higher (up to 1100° F).

On many applications, the valve may be used only during start-up or emergency; therefore, the prevention of seat leakage is an important consideration. The boiler feed-pump recirculation valve, used to maintain a minimum flow through the pump, is generally required to remain shut over long periods of time with differential pressures across the seat of up to 6000 psi. Temperatures are moderate, but the prevention of seat leakage, which rapidly increases due to erosion of even the best hard trim alloys, presents a real design challenge. Such valves are generally high-pressure versions of the conventional cage-trim design shown earlier in this chapter.

As a last example, Figure 45 shows a large, high-pressure single-seat valve designed especially for once-through boiler start-up service. The design exceeds 2500 lb. ANSI rating and has a special breech-block type, body closure that uses internal pressure to aid in the sealing. Special attention is paid to body contouring, to minimize distortion due to thermal cycling. Exceptionally rugged construction is required throughout, because the valve normally sits in the closed position and may be required to fully stroke in as little as 2 seconds, in the event of a serious plant upset.

VALVE BODY END CONNECTIONS

Most flanged control valve bodies conform to the standardized face-to-face dimensions listed in ISA Recommended Practice, RP 4.1 or Fluid Controls Institute Standard, FCI 65-2, with the exception

Table 3. STANDARDIZED FACE TO FACE DIMENSIONS FOR FLANGED CONTROL VALVE BODIES

Pipe Size (Inches)	ANSI RATINGS (LB.) 125 Iron 150 Steel (Inches)	250 Iron 300 Steel (Inches)	600 Steel (Inches)
1/2	-	7-1/2	8
3/4	-	7-5/8	8-1/8
1	7-1/4	7-3/4	8-1/4
1-1/2	8-3/4	9-1/4	9-7/8
2	10	10-1/2	11-1/4
2-1/2	10-7/8	11-1/2	12-1/4
3	11-3/4	12-1/2	13-1/4
4	13-7/8	14-1/2	15-1/2
6	17-3/4	18-5/8	20
8	21-3/8	22-3/8	24
10	26-1/2	27-7/8	29-5/8
12	29	30-1/2	32-1/4
14	35	36-1/2	38-1/4
16	40	41-5/8	43-5/8

of some of the newer flangeless body styles, Saunders types, angle valves, and rotary valves (see Table 3). End connections can be classified as follows: threaded; welded, i.e., either butt-weld or socket weld; and flanged; or flangeless.

Threaded connections (female NPT thread) are standard in valve sizes smaller than 1 inch and are frequently used for control valves up to 2-inches. Steel control valve bodies having threaded ends are generally rated at, and limited to, 600 lb. ANSI. Their use in corrosive service is somewhat limited.

Welded ends are quite common where high pressure, high temperature fluids are encountered, particularly in steam and water service in power plants. Care should be taken to see that the valve body material specified is compatible with the adjoining pipe material. High chrome or carbon alloys should be avoided, because they have a tendency to air-harden and, therefore, require special heat treatment. Butt weld-end connections as illustrated by Figure 46, are standardized in ANSI B16.34. However, socket-welding ends, per

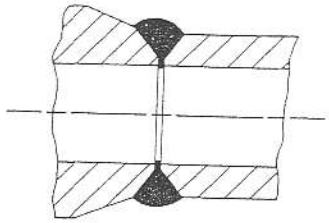


Figure 46. BUTT WELD END.

ANSI B16.11, are usually specified in valve sizes of up to 2 inches.

Steel or steel alloy flange dimensions conform to ANSI B16.5. Similar standards govern flanges for cast iron, bronze or other materials.

Cast iron flanges have no raised face to prevent bending moment around the gasket. Steel flanges below 600 lb. ANSI rating have a 1/16-inch high raised face, and above 600 lb., they have an ANSI 1/4 inch raised faced dimension. This simplifies identification.

The flange rating is determined by the type of the service, required material, presure, and the maximum fluid temperature. In contrast to certain European standards, an ANSI flange rating defines the maximum working pressure at an elevated temperature, i.e., 265 psi at a temperature of 850°F for carbon steel, 300 lb. ANSI rating. Such a flange is rated at 740 psi at room temperature and is tested at 1125 psi hydrostatic pressure.

Flanged valve bodies can be removed quite easily from the pipe; however, they are subject to either thermal distortion or thermal shock. Welded construction should be considered if wide temperature fluctuations are to be encountered.

The flange facing is determined by the type of joint desired. A raised face requiring a flat gasket is the most popular joint for service of up to 600 lb. ANSI. Ring-joint facings are more popular above these ratings. For very high pressure, (10,000 to 50,000 psi) a lens gasket or other special design will be necessary. In a typical design, as shown by Figure 47, extra heavy flanges are threaded onto the adjoining pipe ends to clamp a solid-metal, lens-ring gasket.

Another type of valve body has no flange connection at all and is simply clamped between the adjoining line flanges. This is a method originally used with butterfly valves and other rotary-type valves and is now enjoying increased popularity, because of the decrease in weight and the reduced space requirements.

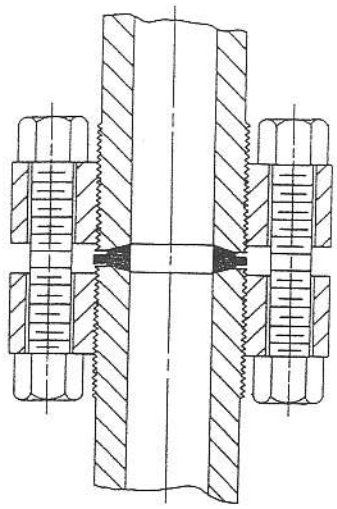


Figure 47. HIGH PRESSURE CONNECTION - LENS RING GASKET.

Such a flangeless valve, as shown in Figure 48, is also somewhat simpler to install, because alignment between the line flange and the body flange is less critical. Long tie rods must be made from a high-strength, bolting material. The imposed strain on these tie rods, when properly installed, is sufficient to compensate for any thermal expansion of the valve body. Good design practice is to specify a flexible gasket with a controlled-compression, metal shoulder, between the line flange and the body end-connection to prevent crushing of the gasket during thermal cycling. Flangeless valve bodies are usually available for flat gaskets only; the body end connection having the same diameter as the adjoining flange, raised-face surface.

VALVE BONNETS

The valve bonnet or top closure is the removable upper portion of the valve body sub-assembly and is normally connected to the body by high strength bolting. It is a pressure-carrying part and is, therefore, subject to the same design requirements as the valve housing. The bonnet normally contains suitable mounting means for the actuator; for the packing, or other similar stem sealing means; and for an upper stem guide. Removal of the valve bonnet generally provides access to the valve trim; however, there are certain valve designs, such as eccentrically-rotat-

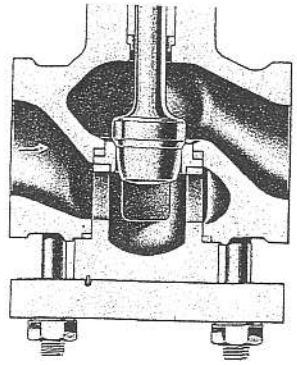


Figure 48. FLANGELESS VALVE - BOTTOM ENTRY.

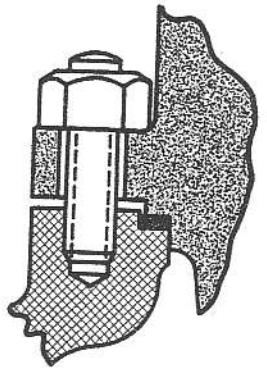


Figure 49. BOLTED BONNET JOINT.

ing plug valves and sliding-gate valves, where the bonnet is a cast-on part of the valve body. In this case, trim removal is possible, either through the inlet or outlet flange connection or through another separable body connection.

The conventional bonnet joint contains a machined recess to provide guiding and containment of a flat gasket, as illustrated by Figure 49. The joint is usually stud bolted; however, through bolting is common on high pressure valves. Both the bolt size and the flange thickness should be calculated in accordance with applicable ASME Code requirements, such as ASME Pressure Vessel Code, Section 8, paragraph UG-34.

Some low-pressure valves, particularly in sizes below 2 inches, have a threaded bonnet connection which is more economical to produce than a flanged joint; however, it may be difficult to remove the bonnet after extended service, particularly when corrosive fluids are involved. The alignment between the stem, plug, and seat ring is less accurate.

Finally, there is the pressure-seal type bonnet connection, which employs a pressure activated metal seal ring and a bridge-lock type of thread engagement, as illustrated in Figure 50. The main advantage of this type of seal, usually employed in high pressure and high temperature service valves, is the absence of a large bonnet flange which can be subjected to high internal stresses with frequent thermal shocks or thermal cycling.

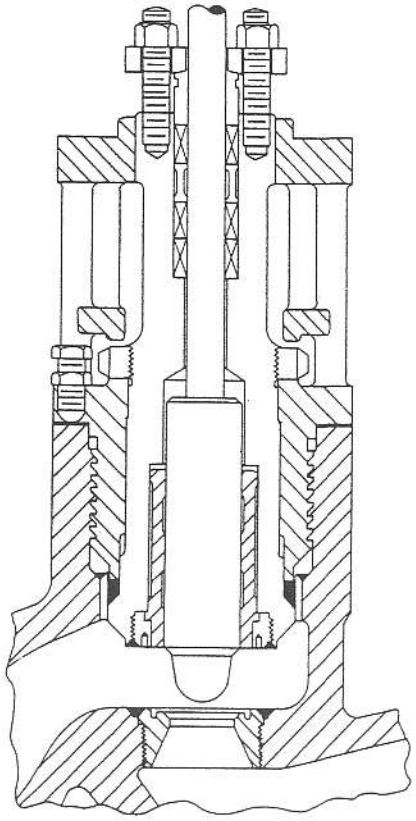


Figure 50. PRESSURE SEALED BONNET.

The plug guide bushing, when employed, is usually made of a hard material such as heat treated ANSI type 440-C stainless steel or 17-4PH stainless; however, special guide materials, such as glass filled Teflon are also available for corrosive or cryogenic applications. These guide bushings are usually either pressed-in or permanently held by a suitable retainer.

The upper portion of the bonnet contains either the valve packing or the stem seal. This packing area is relatively near the bonnet flange connection for fluid temperatures of up to 450°F. A tubular extension separates the packing area from the flange in case cryogenic temperatures, to 450°F, are encountered. This extension bonnet places the packing far enough away from the cold area of the valve to prevent freeze-up of the packing, as illustrated by Figure 51.

For high fluid temperatures (above 450°F) it has been customary to cast either horizontal or vertical cooling fins onto the bonnet extension to provide sufficient radiating heat loss. The representative design shown in Figure 52 is called a "Radiation Fin Bonnet."

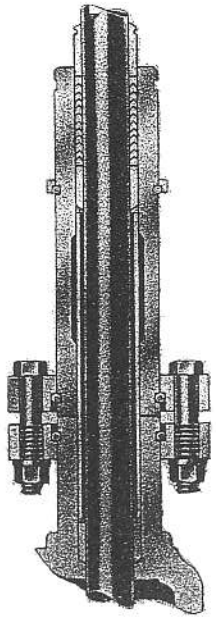


Figure 51. EXTENSION BONNET.

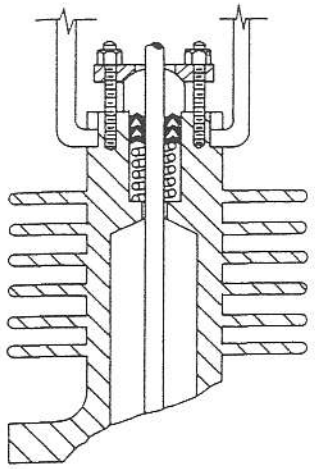


Figure 52. RADIATING FIN BONNET.

Recent designs omit these radiation fins, because they have been proven unnecessary. In other words, a well designed, plain extension bonnet will provide sufficient radiating surface, in combination with a limited cross-section, for heat conduction to keep the packing within acceptable temperature limits. In no case should such a bonnet be covered with thermal insulating material.

For service in which radioactive fluids, highly-toxic fluids, or very rare and costly media are encountered, it is customary to specify a *bellows seal.* A bellows seal usually consists of either a hydroformed metal bellows or a welded leaf-type bellows, surrounding the valve stem and forming a completely enclosed, metal seal for the moving valve stem. Bellows seals are of either internal pressurized or external pressurized design. In the internal pressurized version, as shown in Figure 53, only the inner surface of the bellows is subjected to the fluid pressure. This saves cost in the case of corrosive fluids, because the outer-extension bonnet can be made of carbon steel. On the other hand, the external pressurized bellows can withstand a somewhat higher pressure rating. A well-designed bellows seal has as anti-rotation device preventing bellows damage during assembly and disassembly.

One basic disadvantage of bellows seals is their relatively-low pressure rating. The pressure ratings decrease with an increase in valve size. Typical ratings are 150 psi at 600°F. Higher ratings to about 500 psi can be obtained by the use of special bellows, having increased wall thickness. Unfortunately, this is done at the expense of the number of obtainable life cycles. Another way of increasing the pressure rating is to load the exterior side of the bellows with a pressurized inert gas, but in most cases, this method is too costly and cumbersome to be practical.

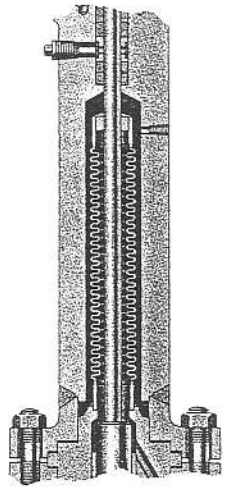


Figure 53. INTERNALLY PRESSURIZED BELLOWS SEAL.
Courtesy Fisher Controls Company

The connection between the bellows and the stem or the lower flange is usually tested with a mass spectrometer that will measure leak rates of less than 1×10^{-6} cc/s of helium, from atmospheric pressure to high-vacuum.

A more economical substitute for the bellows seal is the double packing, described below.

PACKING BOX ASSEMBLY

The purpose of the packing box assembly is to contain an elastic means for preventing the leakage of a process fluid, past the surface of the reciprocating or rotating valve stem. Suitable adjustments should be provided for varying the compression of the packing material against the surface of the stem. The ideal packing material should be elastic and easily deformable, but yet, cause virtually no fric-

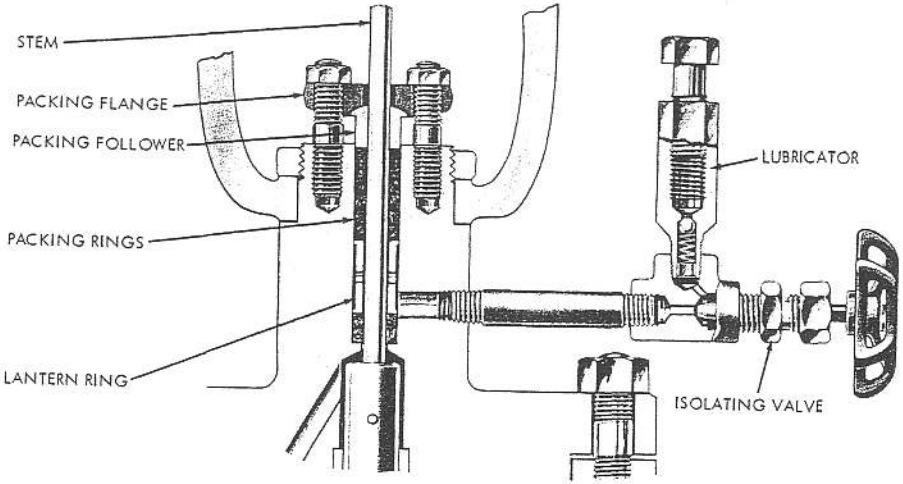


Figure 54. PACKING BOX ASSEMBLY (BOLTED) WITH LUBRICATOR AND ISOLATING VALVE.

tion. In addition, the packing should be as chemically inert as possible, and should be able to withstand high pressures and high temperatures. Existing packing materials will meet all of these requirements; therefore, selection must be made from a variety of materials to suit the specific service conditions.

Figure 54 is a cross-sectional view of the most-popular, bolted, packing box assembly. The assembly consists of a packing flange, packing follower, lantern ring, and a number of equally spaced packing rings. The packing flange contacts the follower, over a ball-shaped surface, to allow for unintentional misalignment of the packing flange, without side-loading on the stem. The lantern ring provides a space for the insertion of lubricating grease through a separate packing lubricator shown here in combination with an isolating valve. The isolating valve effectively seals the stuffing box from the lubricator to prevent the process pressure from blowing the grease out to atmosphere. This is a safety feature, particularly required on high pressure installations. Valve stem lubricants (lubricating greases) are available for different services and for temperatures of up to 550° F. The viscosity varies with different grades, to provide maximum lubricity at given service temperatures. Another variation in the composition of the grease is dictated by the corrosion requirements. The valve manufacturer should be consulted for his recommendation.

Certain low pressure valves for less critical service, such as water or steam, sometime employ screwed packing retainers such as that shown in Figure 55. In this design, a threaded cap compresses the packing rings inside the packing box. This type of construction is not permissible with highly corrosive fluids. Corrosion, either by atmosphere or by leaking process fluid, either could make the packing adjustment unserviceable after a certain time or could lead to the possibility of a packing blowout.

A well designed packing box permits repacking of at least one additional split packing-ring without disassembling the upper valve portion.

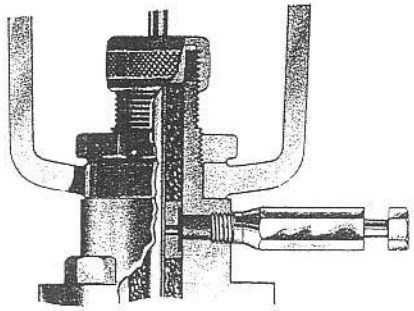


Figure 55. SCREWED PACKING BOX.

A special variation in the packing box design is shown in Figure 56. This is a *double packing,* i.e., an assembly having two stem packings, in series. Such a packing arrangement is popular on atomic reactor valves that handle pressurized water. It can also be used on high vacuum service, but in this case the V rings have to be inverted. Any dangerous fluid, that leaks around the first set of packing rings, can be vented through the side connection to a special waste tank. Another approach is to pressurize the area between the two packings with inert gas at a pressure level, slightly higher than the fluid pressure inside the valve. In this case, the inert gas might leak into the valve, but no fluid is allowed to leak by the stem. This packing arrangement is more reliable and less expensive than a bellows seal.

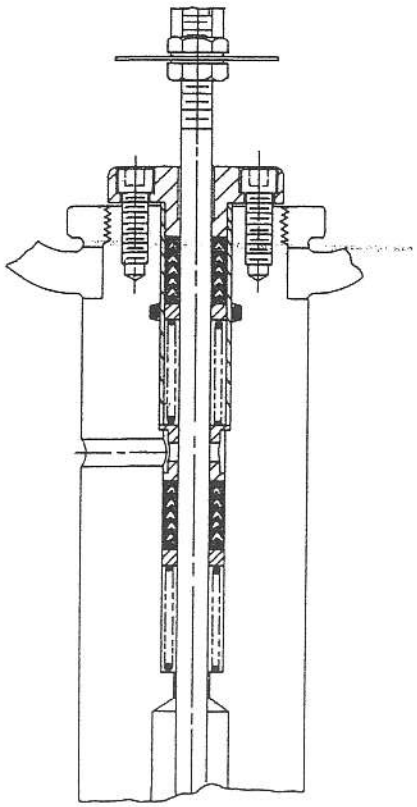


Figure 56. DOUBLE PACKING ARRANGEMENT.

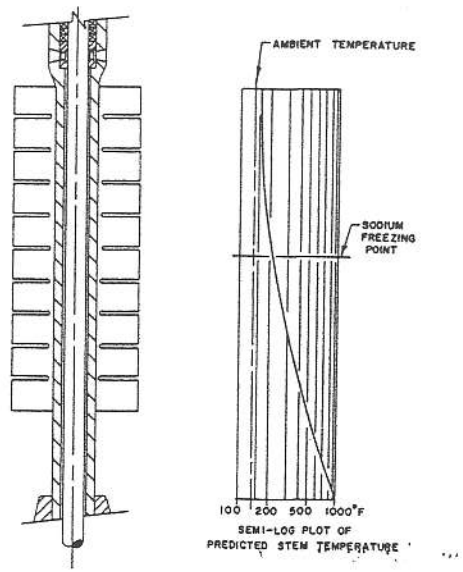


Figure 57. FREEZE SEAL PACKING - MOLTEN METAL SERVICE

Finally, there are special stem seals such as the so-called "freeze-seal," employed with valves for molten metal (see Figure 57). In this design, a fine annular clearance passage is placed between the stem and the bonnet (inside diameter), ahead of the packing, and long enough to allow sufficient heat absorption to freeze liquid sodium. Solid sodium, of course, cannot leak through the packing box. The disadvantage of this type of seal is the high static friction, i.e., the actuator has to overcome the shear force of the frozen sodium everytime the valve moves.

A freeze-seal is more successful with rotary stem design, because it will reduce the amount of sodium oxide formation near the follow-up packing.

TYPES OF VALVE PACKINGS

The most popular valve packing material is Teflon, because of its excellent chemical inertness and its good lubricating properties. Teflon can be used in solid-molded or turned form (chevron rings), as shown in Figure 58, or it can be used as a lubricant for an asbestos packing. Solid chevron rings should be spring-loaded to provide a minimum initial pressure against the stem. Disadvantages of the solid Teflon packing are its high coefficient of thermal expansion, particularly near room temperature, and the require-

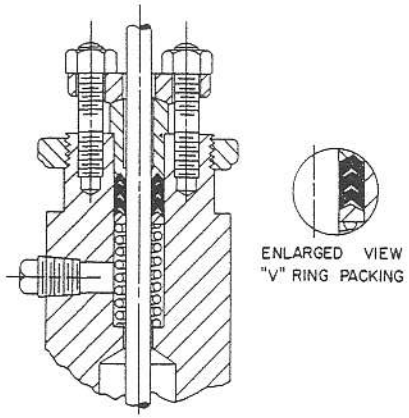


Figure 58. CHEVRON "V" RING PACKING.

ment for extra-fine surface finishes. Surface finishes of 8RMS (roughness in microinches) on the stem surface and 16RMS at the inside of the packing box are commonly specified to prevent undue friction and wear of the chevron rings. Solid Teflon rings are non-replaceable, once the actuator is mounted on the valve.

Braided asbestos is still a popular packing material, because it can be made as split rings, which can be wrapped around the valve stem; therefore, facilitating maintenance when the valve is installed. This type of packing usually employs additives, such as mica or graphite, for lubrication, particularly in high temperature service. The maximum temperature limit of asbestos is approximately 1000° F; however, the use of radiation bonnets lowers the local packing temperature sufficiently, to allow its use in valves handling fluid temperatures higher than 1000° F.

A more popular lubricant for asbestos packing, particularly for temperatures below 450° F is Teflon. Teflon is used either as a suspensoid or as a pure braided cover around an asbestos core. The latter is preferred, because it combines the elasticity and deformability of asbestos with the lubricity of Teflon, i.e., low friction coefficient of Teflon against the sliding stem surface.

A recent addition to the list of available packing materials is Grafoil®. Grafoil is an all-graphite product that is flexible and has direction-dependent properties, similar to pyrolytic graphite. It is essentially, chemically-inert, except when strong oxidizers are handled. The coefficient of friction is low and the packing can be used for quite-high temperature applications (sublimation point is 6,600° F). Care has to be taken in adjusting this packing, because it has a high density and overtightening could lead to a lock-up of the valve shaft.

"O" rings, or chevron rings, made out of elastomers such as Neoprene® or Buna-N® can be used for certain low-pressure valves, handling mild fluids at below 180° F. This type of packing material is found on certain specialized valve applications, such as temperature control valves for air conditioning units.

Table 4 is a selection guide for various packing materials.

Table 4. SELECTION GUIDE FOR VALVE PACKING TEMPERATURE LIMIT

			TEMPERATURE LIMIT	
TYPE	*SERVICE*	*LUBRICATION RECOMMENDED*	*W/STANDARD BONNET*	*W/EXTENSION*
Neoprene or Buna-N O-rings or V-rings	Non-corrosive liquids	No	180° F	—
Braided asbestos with suspensoid Teflon	All except molten alkali and hot hydrofluoric acid	Yes	0-450° F	-450 to 800° F
Braided asbestos with solid Teflon cover	All except molten alkali and hot hydrofluoric acid	No	0-450° F	-450 to 800° F
Solid Teflon V-rings	Limited to fluids that will not attack the 316 SS packing spring	No	0-450° F	-450 to 800° F
Braided asbestos with inconel wire, graphite or mica lubricated	Steam and petroleum	Yes	0-450° F	0 to 1000° F
Grafoil	All except for strong oxidizers	No	0 to >2000° F	

BOTTOM FLANGE CONNECTIONS

As mentioned before, double-seated valves can be made reversible by inversion of the valve body. This necessitates two equally-dimensioned openings in the valve body; the first covered by the valve bonnet and the second covered by a bottom flange. The bottom flange contains the lower stem guide in the case of top- and bottom-guided valves, and sometimes contains a drain connection. Single-seated, top-entry valves do not need the bottom flange, because the lower section of the body is part of the cast housing. However, an interesting variation of the single-seated cage valve is the bottom entry valve in which the cage is contained against the valve bridge by the bottom flange and the bonnet becomes the cast-on portion of the valve housing. A typical design with flangeless configuration was shown in Figure 47. The main advantage of the bottom entry valve is the ability to remove the valve trim, without disconnecting the actuator from the bonnet; however, it may be necessary to provide additional space beneath the valve.

An added benefit of the bottom flange is the ability to remove sediment or other debris from the valve housing. A simple pipe-plug can be substituted for a complete blind flange at the bottom of the valve casting for this purpose, as illustrated by Figure 11.

Another purpose can be served by having the blind flange contain an adjustable-limit stop as illustrated by Figure 59. This limit stop consists of an adjustable screw, locked into postion by a locknut. It prevents the valve plug from completely closing or opening, depending upon on the fail-safe direction. A threaded cap, with a suitable gasket, seals the process fluid. This limit stop is available as an accessory with top- and bottom-guided globe valves. Other valve types have limit stops built into the actuating device.

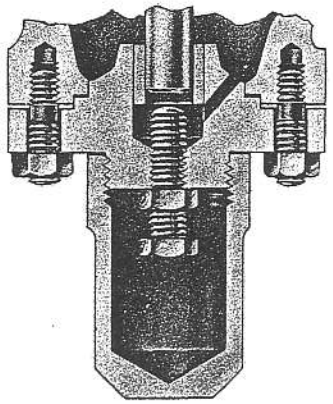


Figure 59. LIMIT STOP IN LOWER BOTTOM FLANGE.

BOTTOM FLANGE AND BONNET GASKETS

For ordinary valve styles, at ratings of 300 lb. ANSI and below, one can use rubber bonded asbestos gaskets, approximately 1/16 inch thick, as bonnet and blindhead gaskets. For higher service pressures and for temperatures above 250° F, metal-clad asbestos gaskets are used. A metal-clad asbestos gasket requires, of course, a higher bolt load; therefore, the bonnet bolting must be designed with this in mind.

Certain top-entry valves, such as the design shown in Figure 11, rely on elastic deformability to make up for the difference in thermal expansion between the cage material and the body. This is provided by spiral-wound stainless steel gaskets with asbestos or Teflon filler. These gaskets are known under brand names such as Spirotallic or Flexitallic. Spiral-type gaskets with Teflon filler are limited to about 450° F, while the ones with asbestos are useable for up to approximately 1000° F. Grafoil or solid metal gaskets can be used on applications exceeding 1000° F.

It is a good design practice to limit the amount of compression of the flexible bonnet and seat-ring gaskets, to a predetermined value, to prevent crushing the gasket. This can be done, for example, by having the bonnet contact the housing material after a certain amount of compression.

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CHAPTER 4

VALVE TRIM

Mathew L. Freeman

AMETEK, Inc.
Power Systems Group
Hatfield, Pennsylvania

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*The author was employed at Rockwell Manufacturing Co. during the time the material on globe type control valves was written for the First Edition. Material on ball, butterfly and other types of control valves was written for the Second Edition while he was employed by Masoneilan International, Inc.

INTRODUCTION

Trim is the heart of the valve with a primary function to proportion the valve orifice area in such a manner that a prescribed relationship exists between flow capacity and valve plug lift. A secondary function may be to shut off tightly. These functions are accomplished in conjunction with body shape, actuator, stem packing, and line fluid forces acting upon the trim. The overall valve design and service application must properly relate all of these parameters to obtain good valve performance.

Valve trim consists of the removable internal parts of the valve contacting the flowing fluid; namely, plug, seat(s), stem plug guide(s), bushings, and cages. Stuffing box components considered as trim are the packing follower, spring, lantern ring, and packing retaining ring. Secondary trim parts are stem-to-plug attachments, seat retaining rings, seat to body seals, spacers and the like. Parts excluded are packing, bonnet, bottom flange and gaskets sealing bonnet and bottom flange to the valve body. (See Figure 1)

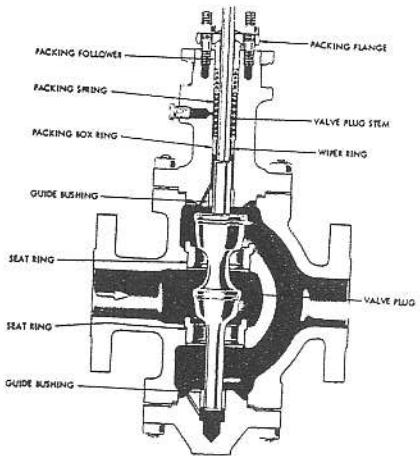


Figure 1. VALVE TRIM COMPONENTS
Courtesy Fisher Controls Company

The plug is the moving component of the valve which throttles flow by positioning itself within the seat orifice and shuts off flow by contacting the seat. The plug is moved against dynamic fluid flow forces by stem force transmitted from the actuator. Throttling may be done by a contoured or V-port plug or shaped opening in a close fitting cage guide.

The plugs (or cage slots) are shaped to throttle flow with a given characteristic such as quick opening, linear, parabolic or equal percentage. Graphs of these flow characteristics are shown in Chapter 5.

Line fluid and flow dynamics of the particular service application may subject trim to an erosive and/or corrosive environment plus static and dynamic forces acting along the plug axis, normal to the axis and rotational to the axis. Vibration is also a factor.

Temperature, pressure, flashing fluids, cavitation, fluid viscosity and solids content all affect trim design. Control of flow requires a pressure drop across the trim and the energy dissipated is absorbed by the fluid and in part by the valve trim and body.

Each valve has a given flow characteristic: i.e., for a given percentage of total lift of the plug off the seat, flow is a given percent of the full open flow. This is true only when the pressure drop across the valve remains constant as in the manufacturer's flow test; it is known as the "inherent flow characteristic."[1]

The installed[2] flow characteristic may differ considerably at much higher pressure drops. In many moderate to low pressure applications the differential pressure across the valve falls off as flow rate increases thus causing a difference between the inherent and installed flow characteristics. This frequently occurs with limited pump capacity or long upstream lines.

Selection of valve trim in regard to flow control is based upon: 1) knowledge of service pressure and flow conditions; 2) the manufacturer's inherent flow characteristic curves for the different trim shapes (these shapes are different profiles on: - plug noses, "V" slots in plug cylinders or cage) and 3) how the inherent characteristic is altered by varying flow conditions. This allows a prediction of the installed flow characteristic for each trim. From these, choose the trim which gives the closest required installed flow characteristic. Beyond this point most controllers will correct discrepancies.

Maximum flow capacity depends upon seat port sizing, body design, and plug

[1] Service applications such as dump valves removing liquid bottoms from vessels do have a fairly constant pressure drop over a wide range of flow. This also occurs in gas reducing service with a large capacity upstream.

[2] Flow characteristics of a specific valve installed in a specific service application.

lift and type, and whether or not the full open flow characteristic must exactly match a straight flow characteristic curve.

Shut off and opening capability are determined by the diameter of the mean seat-to-plug seal contact, pressure differential and available actuator force, provided the stem design is adequate and allowance is made for seating force and packing friction. Not all control valves are required to shut off "drop tight" and manufacturers often state the shut off leakage flow as less than 0.01 to 0.10% of the full, open flow capacity. Upstream block valves are used where necessary.

Double ported valves were introduced many years ago to gain increased flow capacity in a given body size with low lifts. These were necessary for air to close, spring opened diaphragm actuators and also to give a high degree of pressure balance in the closed position thus reducing actuator requirements.[1] Seating tightly on two ports simultaneously is difficult to achieve, except in heavy duty and expensive designs. The balanced cage design valve overcomes much of the double seating leakage, as will be described later.

"Drop tight" or "bubble tight" sealing may be obtained in single seat valves either by using narrow seat joints and high unit compressive seating stress, or by going to soft seats of elastomer materials. Modern actuators possess the power and sensitivity needed for tight shut off combined with good throttling while valve positioners, pressure boosters, and controllers greatly improve performance of simple diaphragm and cylinder actuators to meet tight shut off requirements.

Cage design, incorporating a balance of pressure forces across the closed plug, reduces the differential pressure load by 98% or more, thus allowing a much smaller actuator and greater seat joint compressive load for tight sealing.

Globe valve trim is discussed in detail in this chapter along with a brief description of trim used in non-globe type valves.

TRIM DESIGN FOR MAXIMUM FLOW

Maximum flow capacity in globe valve design is achieved by using the largest possible flow orifice combined with a quick opening plug form. The orifice should not be so large that valve body flow resistance becomes an appreciable part of the total valve pressure drop and changes the flow characteristic. Discharge flow area is equal to the orifice area at lifts of 1/4-seat diameter. (See Figure 2)

Turning fluid past the plug nose causes flow resistance and an additional 10 to 15% lift is required to obtain maximum flow. Thus total lift required is about 35% of the orifice diameter. Y-bodies require lifts of 100% of orifice diameter to clear

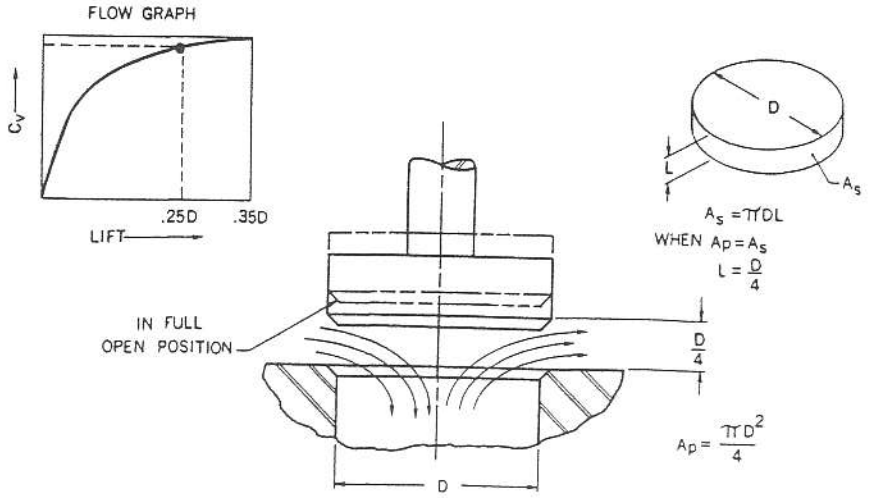


Figure 2. QUICK OPENING PLUG FLOW AREA.

[1] Double ported valves are far from being dynamically balanced in throttling position with flow at moderate pressure drop.

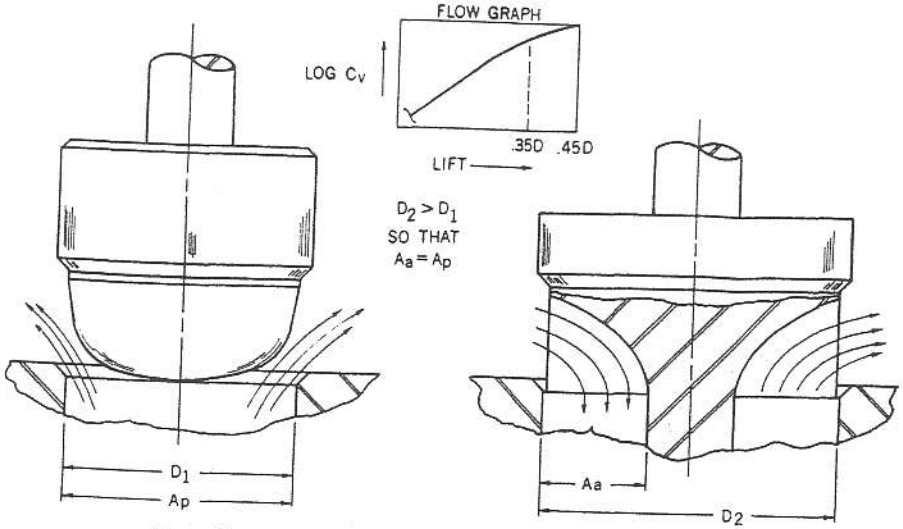


Figure 3. FLOW AREAS OF CONTOURED AND V-PORT PLUGS
Note: A larger port diameter is required for a plug with guide post to equal the port area of a top-guided plug.

the plug out of the flow stream, but have a much higher flow capacity than globe types with an equal seat orifice diameter.

Linear or equal percentage flow characteristic requires a plug nose extending into the seat orifice and this must also be withdrawn to get full flow. In this case the contoured or V-port shape aids in turning flow gradually and the lifts for maximum flow are similar or slightly greater than for quick opening plugs; usually, about 45% of orifice diameter for top guided plugs. (See Figure 3)

PLUG SHAPE EFFECT ON MAXIMUM FLOW CAPACITY

The choice of plug type effects maximum obtainable flow capacity as indicated below for two inch Single seated, full port valves with top and bottom guiding:

Plug Characteristic and Type	C_v *Maximum**	*%Flow*
Quick Opening	59	118%
Linear (contoured)	57	114%
Parabolic (contoured)	56	112%
Equal Percentage (contoured)	52	104%
Equal Percentage (V-Port)	50	100%/ (Base)

*Typical Values

MAXIMUM FLOW CAPACITY OF VALVES HAVING SINGLE AND DOUBLE SEATED PLUGS

Plug type has an important effect on maximum flow capacity in addition to body type. Note how the double seated flow advantage rapidly increases in the large sizes. The cage designs for double seated valves have oversize orifices with flow throttling within the cage. Typical values are given below:

SIZE (inches)	*DOUBLE SEATED* (Parabolic)	*SINGLE SEATED* Top and Bottom Guided (Parabolic)	Cage Type Equal Percentage)	(Linear)
1				
2	20	17	18	21
3	65	55	65	75
4	170	120	130	150
6	220	165	200	235
8	500	375	410	465
	860	590	650	735

Single seated valves without plug guide posts have the highest flow for a given body weight or flow passage area. The post adds turbulence and reduces the orifice flow area. Angle bodies turn fluid one half as much as globes and thus have higher flow capacities for equal orifice diameters. Body flow passages should be streamlined and have no sudden contractions or expansions; this applies only to maximum flow capacity designs used at moderate pressure drops.

The cage trim of single seated valves uses a larger seat orifice than double seated trim, since the principal restriction to flow is the cage flow ports. Some designs have a 2-5/16 in. seat orifice in a 2 in. valve to prevent orifice flow resistance from altering the flow characteristic governed by the contoured cage ports.

"Y" bodies obtain increased flow capacity by streamlining flow, but require much higher lift than globe or angle bodies. This either increases the diaphragm-spring actuator cost or calls for a springless piston type actuator and valve positioner. Stem, plug-guide and packing wear are increased slightly when compared to a vertical valve installation. "Y" bodies are primarily used for on-off high flow capacity service with intermittent throttling.

FACTORS LIMITING MAXIMUM FLOW

The break point in a flow curve, where increasing lift of the plug out of the seat orifice does not increase flow, may be due to either:

1) Plug is completely lifted out and the orifice is the primary flow restriction; or
2) Body flow resistance is becoming an appreciable part of the total valve resistance. (See Figure 4)

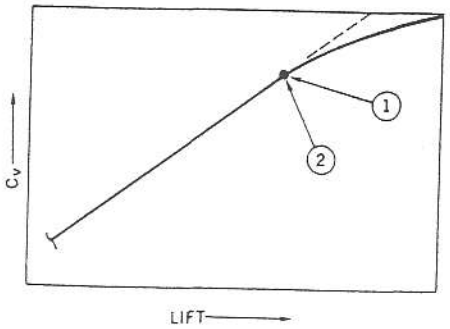


Figure 4. EFFECT OF ORIFICE AND BODY RESISTANCE ON FLOW CHARACTERISTIC.

A break point in the curve, where increasing pressure drop does not increase flow, may be due to either:

1) Critical flow at sonic velocity after pressure drop becomes one half[2] of upstream pressure for gas service; or
2) Flashing flow chokes the body downstream of the seat joint with vapor bubbles in liquid service. In cavitating service no further reduction in pressure can be obtained at the vena contracta to increase flow, since the fluid is vaporizing. (See Figure 5)

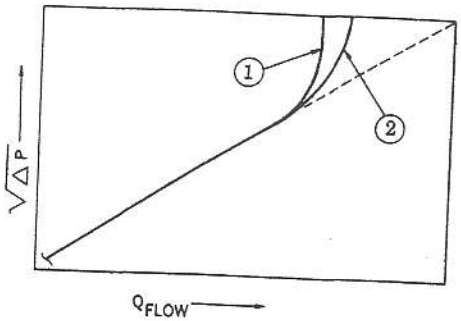


Figure 5. FLOW STAGNATION AND FLASHING EFFECT.

SHAPE OF FLOW CHARACTERIZED PLUGS AND CAGES (See Figure 6)

How closely the manufacturer can get the valve to accurately follow a design flow curve depends upon: 1) valve body type; 2) plug style; 3) available lift in relation to seat diameter; 4) ratio of maximum to minimum flow control as required by the process, called rangeability[1]; and 5) seat joint design. (Refer to Chapter 5)

Maximum flow capacity is usually higher with flow under the contoured plug. Flow-over may be only 80% of flow-under for full port valves and 96% or higher for 40% capacity trim. Flow-under more closely follows the slope or gain of the design flow characteristic resulting in improved control stability. Refer to Chapter 5 for a discussion on gain. V-port plugs show much less difference in flow capacity, over and under.

Positioner Characterizations may be used to attain a special characteristic not available in a standard plug. The positioner cam is shaped to modify the lift for a

[1] No standard of deviation from the specified characteristic has been established; therefore, this must be included in the definition of rangeability.

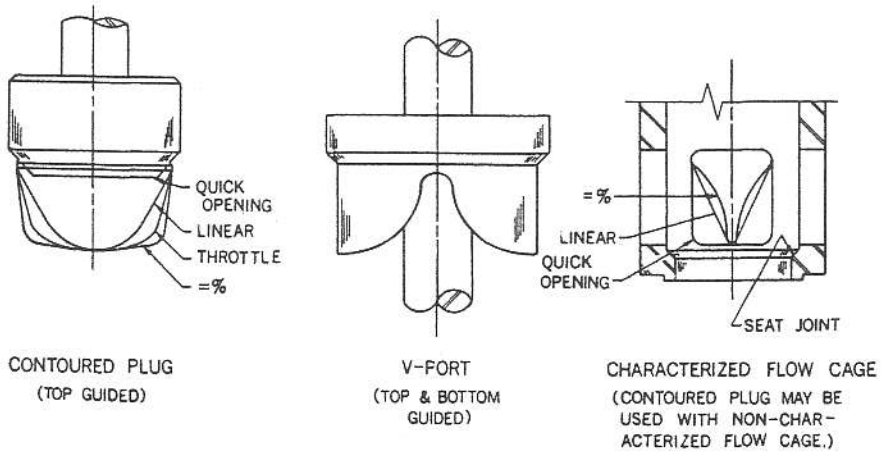


Figure 6. SHAPE OF FLOW CHARACTERIZED PLUGS AND CAGES.

given controller signal to accomplish the special characteristic. There is some limitation on cam shape and follower contact so it is best to start with the nearest plug characteristic, sometimes a parabolic. (See Figure 7)

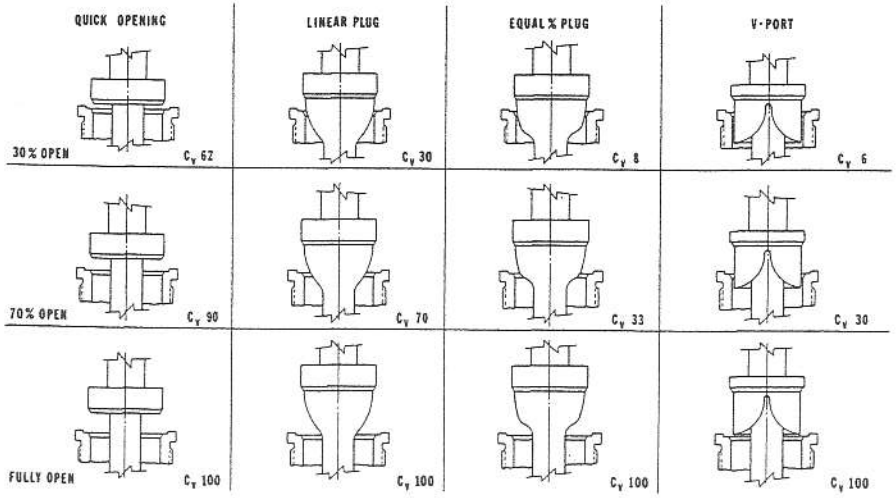
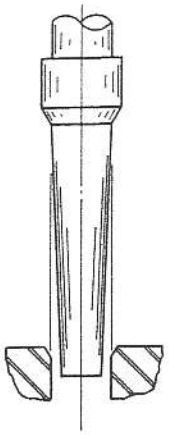


Figure 7. RELATION OF PLUG SHAPE, FLOW CHARACTERISTIC & FLOW CAPACITY AT EQUAL LIFT.

Shown here are the four basic inner valve contours, typified with scalular representation of equal lifts and given flow capacities for each lift. Courtesy WKM Valve Division, ACF Industries

Plug Types

Ultra Low Flow



(a) Metering Pins

Pin sizes 1/16, 1/8 and 3/16 in. (Pin does not lift out of orifice). Long pins ground with a small taper angle. Used in precise flow control with constand upstream pressure. Linear characteristic. Used in valve body sizes 1/4 to 1 in.

(b) Splines

Plug diameters ¼ to ⅝ in. (spline does not lift out of orifice). Top and port guiding. Linear Characteristic. Cv range from 0.00001 to 0.6. Add splines up to four to increase flow capacity in a given size. Used in valve body sizes ¼ to 1 in.

Courtesy of Kieley & Mueller, Inc.

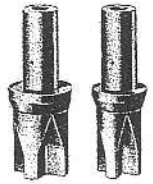
Low Flow

(c) Contoured

Size (in.)	Nominal C_v
3/16	1.0
1/4	1.6
3/8	2.5
1/2	4.0
3/4	6.3
1	12.0

Characteristic: equal percentage. Top guided. Used in ¼ to 1¼ in. body sizes. Plug may lift clear of orifice for max. flow capacity and handling of slurries with less erosion.

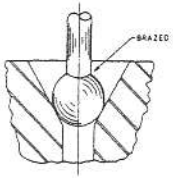
Courtesy of Kieley & Mueller, Inc.



(d) Fluted

Sizes 1/4 to 1-1/4 in. Characteristic - normally equal percentage. Top and port guided. Rangeability, 40:1.

Courtesy of Honeywell, Industrial Division

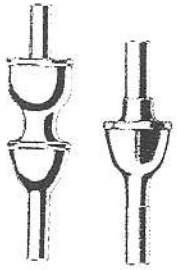


(e) Ball

Tungsten carbide ball brazed to stem, conical seat. For abrasive service. Sizes 1/8, 3/16, 1/4, 3/8 in.

Figure 8. ULTRA LOW FLOW AND LOW FLOW PLUG SHAPES

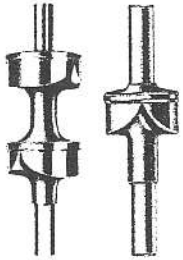
Regular Flow



(a) Contoured,
l. Double Port,
r. Single Port

Single Seated, 1 in. and larger. Double Seated, 1 in. and larger. Characteristics: linear, parabolic, equal percentage and throttle. Top guided, top and bottom guided, cage guided. Better than solid V-port in entrained particle service. For extreme load variations in high pressure service.

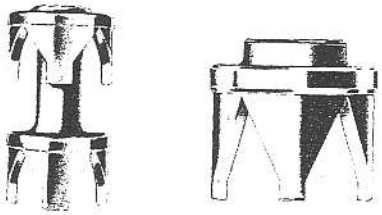
Courtesy Fisher Controls Company



(b) Solid - V[1],
l. Double Port,
r. Single Port

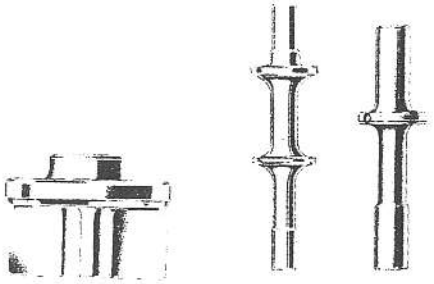
Sizes 1 in. and larger. Characteristic: equal percentage. Skirts of plug are cast solid. Top and bottom guided. Rangeability, 50:1. Greater stability with flow either under or over and less difference between directional C_V than contoured plugs. Best *not* used in entrained particle service. For moderate load variations in high pressure service and extreme load variations to 100 psi ΔP.

Courtesy Fisher Controls Company



(c) V - Port, l. Double Port, r. Single Port
Courtesy Fisher Controls Company

Sizes 1 in. and larger. Characteristics: modified parabolic and equal percentage. Port guided. Skirt is hollow. Moderate pressure drops to 100 psi. Also used in three-way valves. May be obtained with O-Ring seal at plug seat joint. Best *not* used in entrained particle service. May cause high rotary stem torque with large flows.



(d) Flat Disc - Quick Opening
l. Port Guided, c. Top and Bottom Guided Double Port, r. Top and Bottom Guided Single Port
Courtesy Fisher Controls Company

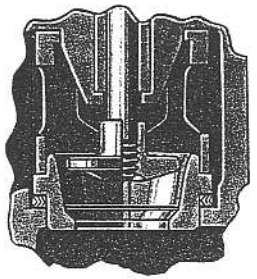
All sizes. Short lift. Characteristic: quick opening. Soft seal on plug available. May be used for moderate load changes with 100 psi ΔP. Add very short nose to prevent slamming on higher ΔP applications.

Figure 9. REGULAR FLOW PLUG SHAPES (continued next page).

[1] Note dissimilar top and bottom "V" shape to characterize flow (Also see Figure 27 in Hard Facing Section).

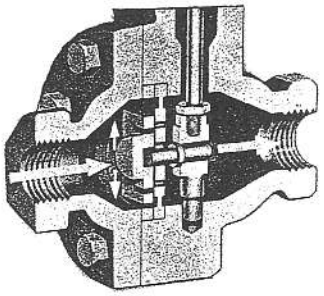
Dissimilar top and bottom shape to reduce dynamic unbalance fluid forces.

(e) Mixed Types (Double Ported)
Courtesy Fisher Controls Company



Sizes to 1.5 in. Plug acts as a sharp edge orifice in contoured seat ring which characterizes flow. Seat ring may be reversed to change direction of flow.

(f) Contoured Seat Ring and Sharp Edge Plug
Courtesy A. W. Cash Company



Sliding gate with multiple orifices. (Refer to section on non-globe valve trim.)

(g) Sliding Gate With Multiple Orifices
Courtesy Jordan Valve

Figure 9. REGULAR FLOW PLUG SHAPES.

MINIMUM CLEARANCE FLOW

Linear, parabolic, and equal percentage plugs are designed with a short cylindrical section just under the seat joint taper as a diameter for plug characterized contour run out and to establish a definite plug operating clearance in the seat orifice. V-port plugs and splined design must also have a clearance fit. (See Figure 10.)

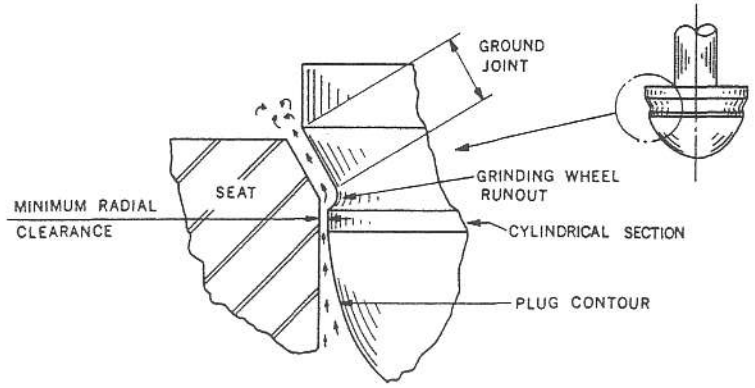


Figure 10. MINIMUM CLEARANCE FLOW OF CONTOURED PLUG.

When the plug is first cracked open, there is a linear characterized flow controlled by the distance across the opening of the joint. Soon the joint opening exceeds the annular flow area of the clearance between the cylindrical plug section and the seat orifice diameter, temporarily causing a choked flow which increases as the plug rise decreases the length of the close clearance flow path. True characterized flow should begin after the plug-contour run-out into the cylindrical section reaches the bottom of the seat-angle run-out in the orifice. Actually this does not occur until the plug makes a slight additional lift to eliminate the affect of the previous resistance.

Minimum clearance flow, as a percent of total flow, decreases as valve size increases, because it varies with the seat orifice diameter, whereas total flow varies with orifice area (diameter squared); therefore, the rangeability of plugs normally increases with size. Double-seated valves have larger clearance flows than single seated valves for the same flow capacity, because the length of the seat circumference of an area divided between two ports is longer, and clearance flow area is greater than for single ports. Guide posts also cause the seat diameter to be larger because they block part of the flow path. Rangeability of top and bottom guided valves, both single and double seated, is thus restricted below 3/4 in. size. Top guided valves should be used in sizes 3/4 in. and smaller.

Minimum flow in cage valves is controlled by the diametral clearance between the cage and plug cylinder, and by the length of the annular flow passage before it reaches the "V" at the bottom of the cage port. The plug-to-seat joint, flow area is similar to that shown in Figure 11.

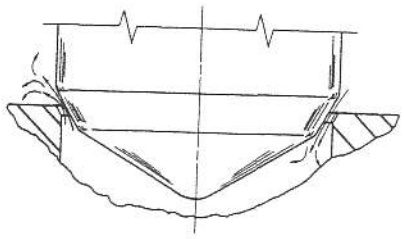


Figure 11. LOW RANGE FLOW LINEAR PLUG DESIGN.

RANGEABILITY

The parameters previously discussed, control the maximum low flow rangeability of linear, parabolic and equal percentage plugs. Quick opening plugs with flat joints have much less of a problem. Eliminating most of the minimum clearance flow is feasible in short-stroke, *linear* plugs, resulting in increased rangeability. The design, illustrated by Figure 11, functions without a cylindrical contour, run-out diameter. Rangeability of 800:1 is readily obtained by this design; however, rangeabilities of over several hundred are meaningless, because no process approaches this variation and, because plug positioning accuracy to obtain them is beyond the precision limits of most actuators.

Plug Trim vs Nominal Rangeability[1]

Characteristic	*Plug Shape*	*Rangeability*
Quick Opening	Flat Disc	100 and up (Higher in larger sizes with lower seat angles)
Linear	Contoured	30:1 to several hundred
Parabolic	Contoured V-Port Skirt	50:1
Equal Percentage	Contoured	30:1 to 60:1
	Solid V-Port	50:1

So much of the equal-percentage plug's lift is taken up by low flow, that in a contoured plug, the nose-to-control low flow often projects into the conical clearance needed at higher lifts for larger flows. As a result, the maximum flow of the equal percentage plug is less than a linear plug; therefore, with the same minimum clearance flow the equal percentage rangeability is lower.

DESIGNING A PLUG TO A SELECTED FLOW CHARACTERISTIC

The theoretical C_V of a perfect orifice having one square inch area is 38.1. This means that an orifice with a flow efficiency of 100% would pass 38.1 gpm (gallons per minute) with a pressure drop of 1 psi. No orifice has such an efficiency, of course, and the addition of a valve plug restriction further reduces the flow efficiency.

When designing a flow characteristic for a plug, the designer must attain a specific flow, corresponding to a specific lift. To do this, he needs to know the efficiency of the annular seat-to-plug orifice area (not the horizontal annular area, but the truncated conical area, normal to

[1]Rangeabilities of 50:1 are for valves with 2 in. port diameters, 25:1 if the valve is smaller and 100:1 if larger than 2 inches.

the plug contour at that lift). The efficiency varies with the flow rate and the length of the restricted flow path; the latter being predominant at low lifts. Also, the body-flow resistance is a factor which must be accounted for at high lifts and each type of body has a different affect on the resistance.

One method of plug design is to make a preliminary flow test using a rough-shaped plug with each new body design to determine the flow efficiency curve from C_V minimum to C_V maximum for that body. The plug is designed to a given flow characteristic using this data, then it is flow-tested and, if necessary, reshaped slightly to follow the design flow curve more closely. (See Figure 12.)

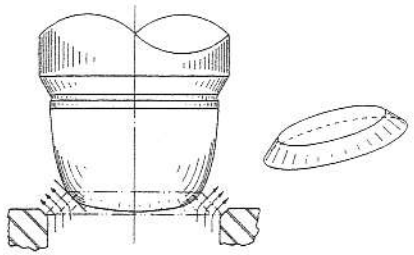


Figure 12. TRUNCATED CONICAL FLOW AREA.

REDUCED CAPACITY TRIM

Reduced capacity trim is used for the following reasons:

1) To achieve precise control at low flow in a process which may soon be expanded in capacity to a much higher flow rate. Valve body change-outs are eliminated. Much of the full capacity stroke length is retained in designing the reduced capacity plug, thus increasing the rate of change of lift to flow, therefore, increasing the precision of flow control.
2) To absorb vibrations and thermal energy in the relatively larger, heavier walled body, having a smaller flow port and much heavier guiding. This occurs in flashing liquid service, high velocity liquid flow, high pressure drop and/or cavitating service. The larger body downstream of the seat orifice in cavitating service, allows the seat wall to be well away from the body wall so that bubbles collapse more in mid stream than against the body wall. By keeping the same guide diameter, the plug is a much stiffer design than the same flow capacity in a smaller orifice valve. Thus, more metal surface is available to dissipate energy for high pressure drop service.
3) To provide ample feeding and gradual turning of fluid into the throttling zone resulting in reduced velocities in order to reduce abrasive erosion by slurry. The body size is relatively larger than the trim.
4) To reduce gas outlet velocity to a subsonic level in the valve body, downstream of the seat orifice.
5) To avoid using pipe line-reducers as a cost savings or to avoid using them where piping strength requires a full size body.
6) To correct initial oversizing errors or to handle flow when the process capacity is reduced.

Most manufacturers offer *standard reduced capacity trim* at 40% of full flow trim capacity for top and bottom guided valves; however, other sizes are available in top guided designs. For example, some 2 in. valves may be fitted with seven sets of trim having seat port sizes of 1/4, 3/8, 1/2, 3/4, 1, 1.5, and 2 in. Top and bottom guided valves and double-seated valves are restricted to fewer sizes by the bottom guide post.

The seven sets of trim are not truly *reduced capacity trim* for high energy dissipation, unless guide post diameters are equal to the full capacity trim posts. If this not the case, they are referred to as *restricted trim.* Reduced capacity trim is illustrated graphically in Figure 13.

In the following example, trim capacity is determined for a 2 in. valve.

Full capacity port of 2 in. diameter has a 3.14 in. flow area (with top guiding).

Reduced capacity port of 1-1/4 in. diameter has a 1.23 in. flow area (with top guiding); therefore,

$$\frac{1.23}{3.14} \times 100 = 39\% \text{ of area}$$

With a lower flow, valve body resistance is less; therefore, reduced trim flow capacity would be approximately 40%.

Plug lift for some manufacturers' 2 in. full port valves is 1-1/4 in. for linear or equal percentage plugs; therefore,

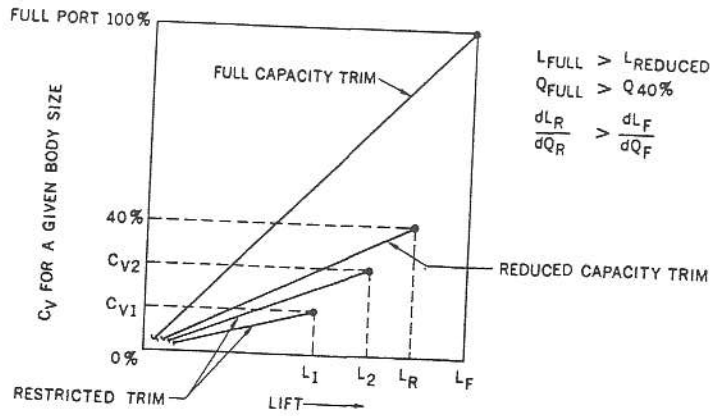


Figure 13. GRAPHICAL REPRESENTATION OF REDUCED CAPACITY TRIM.

the lift: diameter (L/D)ratio is:

$$L/D = \frac{1.25}{2.00} = 0.62$$

Plug lift for some manufacturers' 2 in. reduced port valves is 1 in. for linear or equal percentage plugs; therefore:

$$L/D = \frac{1.00}{1.25} = 0.80$$

As a result, the reduced port plug lift is:

$$100 \times \frac{0.80}{0.62}$$

or relatively 30% longer; therefore, dL/dQ (Q is flow rate) is greater and the flow control precision increases. Reduced capacity may also be obtained by short stroking a linear plug with some reduction in rangeability since there is no change in clearance flow area.

Changes Required By Installing Reduced Capacity Trim

In standard valves, plug and seat ring changes occur. The seat orifice is smaller and the plug has a smaller outside diameter, but the same guide post diameter. The plug lift-to-diameter ratio may be increased causing increased sensitivity by a more gradual profile. Body, guide post bearings and seat ring thread in the bridge wall remain the same.

In cage guided valves only the cage is changed as follows:

1) The new cage has smaller ports, yet seats with the old plug, or the cage may have fewer ports. The same size actuator is used since the seat port area does not change.
2) Non-characterized cage designs, which have contoured plugs, require the replacement of both plug and cage. A smaller actuator may be used with a smaller seat port area.

Some manufacturers use additional supporting parts for undersized trim so it will fit into a larger body. They are usually transfer pieces, fitting the standard, *restricted trim* seat to the larger seat ring thread in the body.

Top guided, single seated valves give a greater precision of low-flow control with reduced capacity trim (greater low flow rangeability) than do top and bottom guided, or double ported valves, because the clearance flow of the latter two is higher.

Reducing the number of cage ports in high pressure service may cause side loads and plug vibration; therefore, a reduction in port size is preferred. A long undercut on the plug just above the seat joint helps to eliminate 60-80% of the side thrust in a non-uniform, flow cage.

SELECTION OF TRIM MATERIALS

Over the years, valve trim has resolved into a limited number of materials which have the necessary balance of physical and chemical properties to give good performance in a wide range of applications. The material selection is coupled with a practical means of fabrication.

Property Considerations

Physical	*Thermal*	*Environment*	*Corrosion*	*Fabrication*
tensile	creep	non-galling couples	chemical resistance	castability
compressive	rupture	erosion resistance	electrolytic potential with body	machinability
torsional				
shear	scaling (oxidation)			finishing surfaces
yield	hot hardness			suitable for hard facing
ductility	cold impact			method of hard facing
Modulus of elasticity				
hardness				
density				

Relative Cost Of Trim

(Relative to a 2 in. single seated valve, including plug, stem and seat)

bronze	0.5X
Type 316 SS	1.0X (base)
Type 440-C SS, hardened	1.2X
Type 17-4 PH SS, hardened	1.3X
Stellite hard faced joint on both plug and seat (316 SS base material)	1.2X

Application Of Common Trim Materials

Application	*Materials*
mild service	bronze
general service	Type 316 SS
severe service	17-4 PH SS Type 410 SS Stellite Colmonoy
very erosive service	Type 440-C SS hardened tool steel tungsten carbide
very corrosive service	Nickel Monel® Inconel® Hastelloy-B® Hastelloy-C® Durimet® titanium

TRIM EROSION

Control valves are active components of flow control systems, constantly throttling and frequently shutting-off flow; therefore, in severe service, trim wear is to be expected. Because of the wear, ease of trim replacement is an important consideration in valve design.

Wearing away of plug and seat material causes leaky valves, alters flow characteristics and, in cases of severe erosion, may create a new flow path and may actually penetrate the valve body wall.

Particle impingement erosion is related to the kinetic energy of the abrasive particles rather than to particle momentum. High viscosity, carrying fluids reduce erosion by creating a "cushioning effect" between the particles and trim. Trim hardness is not the sole criterion in evaluating erosion resistance to particle impingement. Small abrasive particles cause less erosion than larger particles and *erosion damage increases roughly as the square of the flow velocity.*

There are four principal types of erosion: 1) abrasive particle; 2) cavitation; 3) erosive-corrosive action; and *4) high liquid velocity impingement.* Methods of elimination, reduction, or transferring of erosion to a non-critical area differ with the type of erosion.

Abrasive Particle Erosion

Billions of tiny particles, harder than the trim surface, are carried at high velocity in the fluid stream and impinge upon and scour away the trim metal. Slurries have such a high concentration of abrasive particles, that the plug and seat joint literally grind together on each closure. In the latter case the seat join is often propped open by crushed particles and wire drawing occurs.

Abrasive erosion may be reduced by:

1) Selecting trim material or trim with a facing which is harder than the abrasive particles; however, it must not be so hard as to be brittle to impact. This will extend the wear period and restrict abrasion to a few critical zones.
2) Streamlining flow to prevent direct particle impingement upon the trim. To do this, fluid flow should be parallel to the critical seat joint and plug contour surfaces. Flow must be turned gradually and flow attachment and detachment principles must be used to protect plug-to-seat joint areas. These factors are handled by valve designers, but the user and manufacturers' application engineers play important roles in the selection of trim types and body style to aid in achieving these principles.

Valve designers use several techniques to develop and improve flow patterns to reduce erosion. Half- and full-model flow studies using plastic see-through bodies are often used. Flow patterns are shown in air by smoke or attached filaments, while in water either an aluminum flock or a colored fluid injection is used. Trim and body shapes are altered to reduce direct impingement, to remove eddies, and to reduce pressure drops. Once this is done, a final erosion locating method may consist of circulating a high-sand-content fluid through a valve in which the trim has been coated with machinists' layout-bluing. The erosion patterns are visible in a few minutes and further changes may be needed to create a uniform wear rate and to reduce the wear at the seat joint. Once this is accomplished, several days of testing are required to erode the metal to prove the design changes. High pressure drop blow-down dynamic throttling tests may include a range of abrasive particle sizes including pea gravel. This really proves impact resistance.

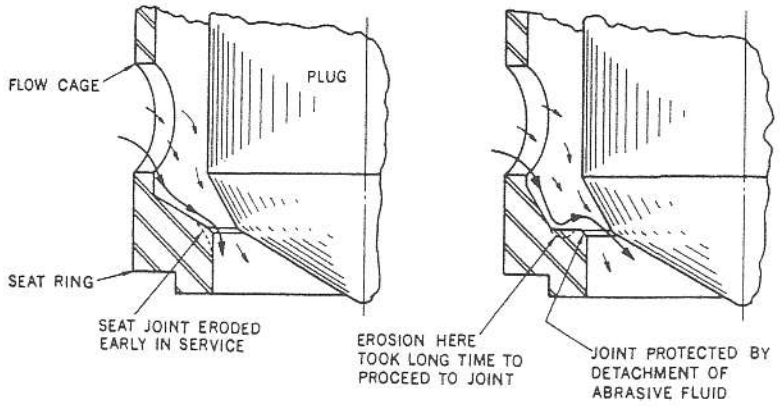


Figure 14. FLOW DETACHMENT PROTECTS SEAT JOINT AGAINST EROSION.

1) In Figure 14, flow attachment and detachment are utilized to greatly reduce the rate of seat joint wear.
2) Figure 15 shows the effect of the removal of an eroding flow eddy in the discharge of a three-way valve. Eddy was necking stem in an area where it passed through the stem seal.

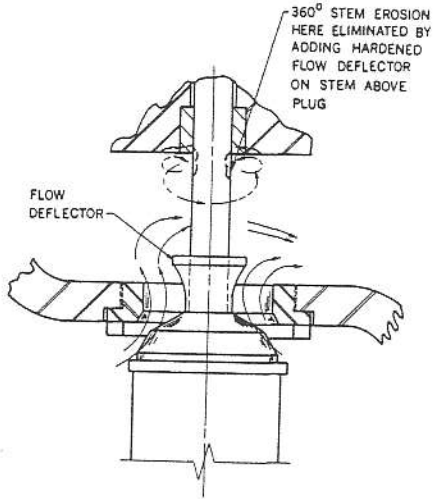


Figure 15. DEFLECTOR ON STEM OF 3-WAY VALVE STOPS EROSION.

3) In Figure 16, the flow cage more uniformly distributes flow around the seat to give longer life and more uniform wear.

Abrasive erosion may be further reduced by using narrow seat joints to limit the particle hang up on the valve closure and to allow sufficient seating force to crush large particles which do hang up. For example in a seat port diameter of 1/4 to 2 in., use a joint width of 0.015 to 0.030 in.

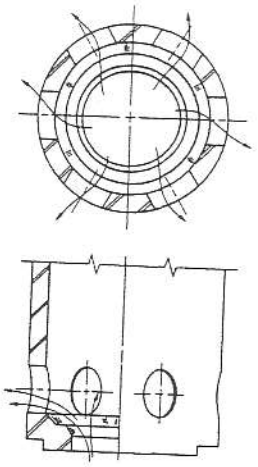


Figure 16. FLOW CAGE DISTRIBUTES EROSIVE FLUID AROUND SEAT JOINT INCREASING SEAT LIFE.

Recommended Pressure Drop Limits For Trim Handling Slurries (Figure 17.)

Material	*Throttling Pressure Drop Limitations*
17-4 PH	125 psi
No. 6 Stellite	125 psi
No. 6 Colmonoy	150 psi
Type 440-C SS	200 psi
tungsten carbide	Above 200 psi

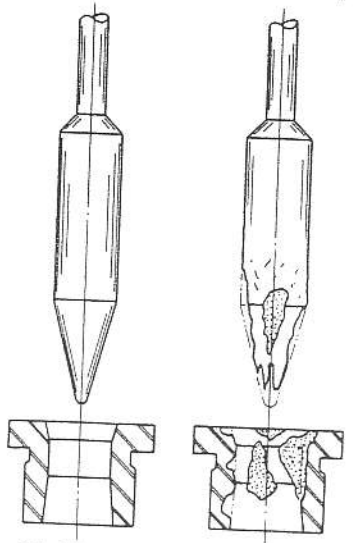


Figure 17. FLUID ABRASIVE AND IMPINGEMENT EROSION.

Cavitation Erosion of Trim

Cavitation may occur any time the pressure of a high velocity liquid flowing in restricted passages of a valve becomes less than the vapor pressure of the liquid at that temperature.

When this happens, vapor bubbles form downstream of the restriction. The enlarging flow passage downstream of the restriction reduces flow velocity and the resultant pressure recovery collapses the bubbles. The impact of bubble implosions causes severe damage to trim. This is covered in greater detail in Chapter 6.

No known material can withstand continuous, severe cavitating conditions without failure, because the impact forces approach or exceed the yield strength of the hardened trim or the binding alloys used for extremely hard tungsten carbides. Ease of trim replacement is, therefore, essential in service under cavitating conditions. Even mild cavitation, if it is continuous, will eventually damage regular trim. Hardened trim is required to withstand a higher intensity of cavitation; whereas, severe cavitation will require a scheduled replacement of even the hardest trim.

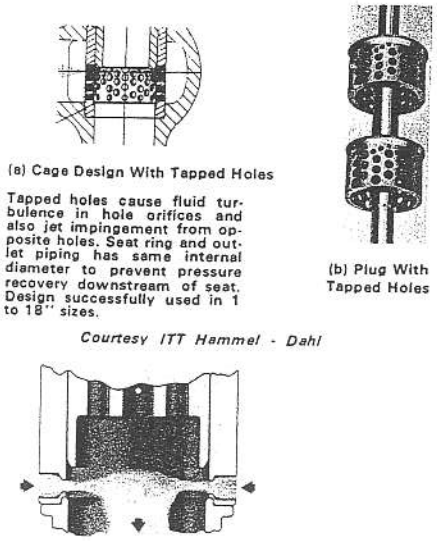
The following trim materials are used for cavitating service, listed in the order of increasing resistance to cavitation effects, R_c (hardness).

Material	*Hardness at 70° F*
No. 6 Stellite	40 / 45 Rc
Type 440/C	50 / 60 Rc
No. 6 Colmonoy	56 / 61 Rc
hardened tool-steel	60 / 63 Rc
tungsten carbide (Sintered and held in nickel binder)	70 Rc[1]

Cavitation erosion of the selected trim may be reduced by:

1) Selecting the hardest trim feasible, without the chance of cracking from the impact of repeated valve closure and thermal shock.
2) Maintaining sufficient downstream pressure to exceed the fluid vapor pressure; therefore, preventing bubble formation in the high velocity low pressure regions. Often 50 to 100 psi downstream pressure, on pressure drops of 3000 psi for water at 120° F, will greatly reduce cavitation damage. This may be done by installing downstream of the valve, either a back pressure regulator or a very hard choke orifice that is smaller than the valve seat.
3) Using a sharp edge seat port orifice to keep the discharge away from the body wall in high pressure angle valves with flow over the disc. Collapse of bubbles within the fluid stream is less damaging than collapse against a rapidly diverging seat port or body wall. If this is not feasible, a gradually diverging seat orifice may be used.
4) Limiting the valve pressure drop by using two valves in series and distributing the drop between them, with a greater drop in the upstream valve.
5) Using special valves that incorporate a multiple series of restrictions and turns in the flow path, thus limiting the pressure drop across any one restriction.

Special trim designs are available for cavitating service. Some valve manufacturers resort to unique designs which greatly reduce or eliminate cavitation in very high-energy dissipation service. These designs cause the fluid to impinge upon itself, creating a high degree of turbulence within the plug-to-seat passage. The upstream potential energy, represented in any throttling valve by inlet pressure, is converted to heat energy in fluid friction (massive turbulence). Pressure reduction is accomplished with essentially no pressure recovery, thus reducing low-pressure, bubble-forming areas. Other designs have large plenums downstream of the seat orifice to allow bubble collapse in the fluid stream rather than against the body wall. Five designs are illustrated in Figures 18, 19, 20, 21 and 34c.



(a) Cage Design With Tapped Holes

Tapped holes cause fluid turbulence in hole orifices and also jet impingement from opposite holes. Seat ring and outlet piping has same internal diameter to prevent pressure recovery downstream of seat. Design successfully used in 1 to 18" sizes.

(b) Plug With Tapped Holes

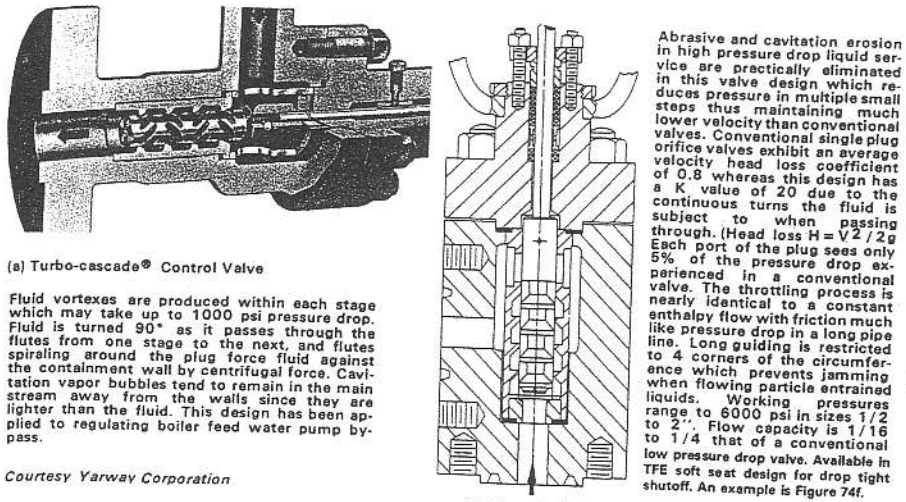
Courtesy ITT Hammel - Dahl

(c) Stepped Holes In Cage

Courtesy Fisher Controls Co.

Figure 18. CAGE HOLE DESIGNS REDUCE CAVITATION.

[1]Some tungsten carbides are harder, but are brittle and do not have sufficient impact strength for repeated valve closure, particularly when cold. Tungsten carbides of 82 R_c may be used in throttle rifices, below seat joints, as shown in Figure 23. Boron carbide also falls in this category.



(a) Turbo-cascade® Control Valve

Fluid vortexes are produced within each stage which may take up to 1000 psi pressure drop. Fluid is turned 90° as it passes through the flutes from one stage to the next, and flutes spiraling around the plug force fluid against the containment wall by centrifugal force. Cavitation vapor bubbles tend to remain in the main stream away from the walls since they are lighter than the fluid. This design has been applied to regulating boiler feed water pump bypass.

Courtesy Yarway Corporation

(b) Cascade Flow Restricting Orifices

Abrasive and cavitation erosion in high pressure drop liquid service are practically eliminated in this valve design which reduces pressure in multiple small steps thus maintaining much lower velocity than conventional valves. Conventional single plug orifice valves exhibit an average velocity head loss coefficient of 0.8 whereas this design has a K value of 20 due to the continuous turns the fluid is subject to when passing through. (Head loss $H = V^2/2g$ Each port of the plug sees only 5% of the pressure drop experienced in a conventional valve. The throttling process is nearly identical to a constant enthalpy flow with friction much like pressure drop in a long pipe line. Long guiding is restricted to 4 corners of the circumference which prevents jamming when flowing particle entrained liquids. Working pressures range to 6000 psi in sizes 1/2 to 2". Flow capacity is 1/16 to 1/4 that of a conventional low pressure drop valve. Available in TFE soft seat design for drop tight shutoff. An example is Figure 74f.

Courtesy Masoneilan International, Inc.

Figure 19. TWO DESIGNS FOR CONTROLLING FLOW BY CASCADING.

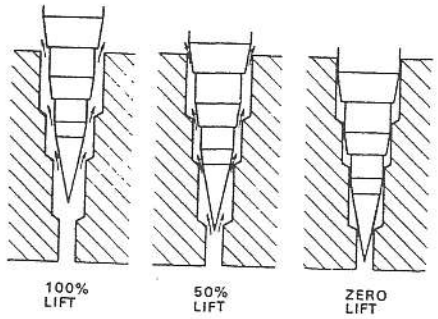
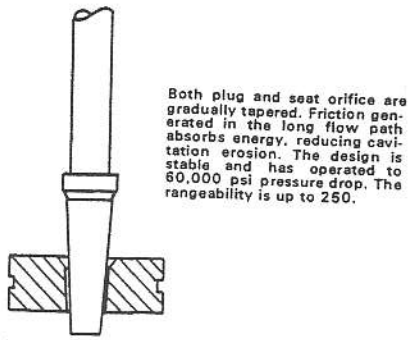


Figure 20. MULTIPLE VELOCITY HEADLOSS TRIM.

Courtesy A. W. Cash Company



Both plug and seat orifice are gradually tapered. Friction generated in the long flow path absorbs energy, reducing cavitation erosion. The design is stable and has operated to 60,000 psi pressure drop. The rangeability is up to 250.

Figure 21. REDUCED FLOW VELOCITY TRIM FOR HIGH PRESSURE DROP.

Courtesy Masoneilan International, Inc.

Erosion-Corrosion

Many metals do not corrode, because they form a protective surface film. In erosive valve service this film is rapidly worn away by abrasive particles, fluid impingement, or cavitation erosion, thereby exposing a fresh surface to the corrosive fluid. Under these conditions erosion-corrosion may proceed at a very rapid rate. Keep in mind that corrosion is a very complex phenomenon and simple guide rules are no substitute for actual valve usage history in the corrosive media.

Erosion-corrosion may be reduced by using more noble metals for trim such as: Monel®; nickel; titanium; Hastelloy®; or Durimet®. Refer to the trim material table given in the appendix for general application and for corrosion resistance to process fluids.

TFE (teflon) Trim in FEP Lined Bodies is the most completely corrosion resistant valve available in the 2 in., ANSI 150 lb. class for -20 to 300°F service (FEP is fluorinated ethylene propylene). The FEP lining is usually locked into grooves cut into a carbon steel body. The plug and seat are composed of TFE which may be glass-filled to increase stability. Alloy valve stems, compatible with the process fluid, are furnished by the manufacturer.

The following list is the relative galvanic corrosion resistance of trim materials, presented in the order of most corroded (anodic) to least corroded (cathodic). Copper and Platinum are shown for reference.

carbon steel	body material
cast iron	body material
Ni-Resist	body material
Type 440-C SS	trim
17 - 4 PH SS	trim
Type 316 SS	trim
Stellite and Colmonoy	trim
nickel	trim
Inconel	trim
copper	reference
bronze	trim
Monel	trim
platinum	reference

The electrolytic potential between trim and body material is higher in proportion to the greater their separation in the above table. The rate of corrosion is, therefore, higher from galvanic action. The larger area of the body, available to corrode, prevents this from being a serious problem because the body is the sacrificial component. Specification codes call for an allowance in body wall thickness for this corrosion.

High Velocity Fluid Impingement Erosion

Extremely high velocity fluid jets sometimes occur in the flow path, causing the fluid to turn abruptly, bouncing off one surface to impinge and erode the adjacent part. Impingement erosion may be a form of erosion-corrosion, whereby, the high velocity fluid jet, blasts away the protective surface coating as rapidly as it forms. Average fluid velocity through the seat-to-plug flow area must exceed several hundred feet per second for impingement erosion to occur.

Boiler feedwater pump bypass valves are a typical example of high velocity erosion. The high pressure drops, to 4600 psi, taken in this service, often cavitates the seat joint and fluid impingement may either wash away the contoured plug nose or cut peculiar flow patterns in its surface. Cage trim helps, but may not completely eliminate, impingement erosion.

Liquid droplets entrained in a vapor stream also cause impingement erosion; however, the distribution is spread over a much wider area. Dry gases and superheated steam cause little wear.

Impingement erosion as illustrated in Figure 22, may be reduced by: 1) streamlining flow paths; 2) using hard trim; and by 3) separating entrained droplets from the fluid stream before it enters the valve.

The following is the relative erosion resistance of various plug types and body combinations, in order from the "best" to the "poorest:"

1) *Single-seated, contoured plug, angle body, flow over plug.* Ample all around fluid feed to seat inlet and gradual turning of the fluid into the throttling zone reduces the inlet velocity and abrasive erosion. The discharge pattern results in the least cavitation damage. Long sweep angle body styles may be used to advantage.
2) *Single-seated, contoured plug, globe body.* Pressure recovery within the body, aggravates cavitation damage.
3) *Single-seated, V-port plug, globe body.* Jet action of the fluid at low lift may erode the body wall.
4) *Double-seated, contoured plug, globe body.* Plug guide post is subject to severe erosion.
5) *Double-seated, V-port plug, globe body.* Same results as items 3 and 4 above.
6) *Splined plugs* concentrate seat wear in a small segment and are unsatisfactory.

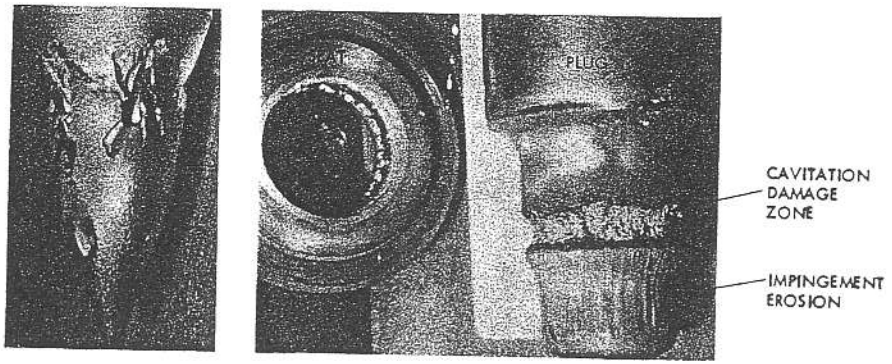


Figure 22. IMPINGEMENT EROSION DAMAGE.

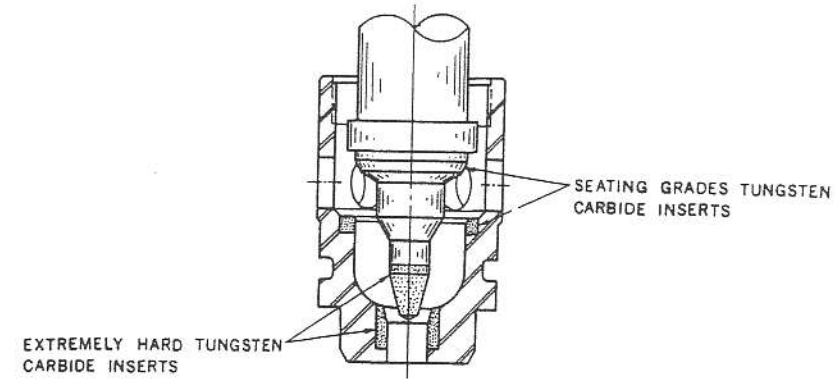
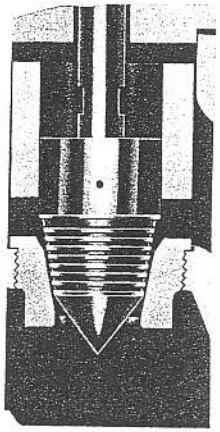


Figure 23. CAGE TRIM WITH SEPARATE THROTTLING AND SEATING ZONES.

Elastomer lined bodies and plugs give excellent results at moderate pressure drops. Ceramic trim is also very resistant, and ceramic discs with orifice ports are used to advantage in sliding gate type control valves.

Separation of the throttling and seating areas leads to a multifold increase in trim life and is often used in severe services. By means of heavy guiding, the plug nose cylinder makes a very close fit in the seat orifice without contacting the orifice side wall. The seat-to-plug joint is located above the orifice and has a larger diameter. It is made of "tough" materials while the plug nose and orifice which do not impact on closure may be of much harder and more brittle materials. These parts take the throttling wear. The cylindrical section of the plug extends well into the orifice, past its entrance taper. On opening, the plug-to-seat joint is cracked and causes little flow restriction. In a 3/8 in. valve orifice, the seat joint offers little flow restriction after lifting 0.001 in. Flow at this point is through the annular clearance of the plug cylinder and orifice, and the valve does not begin to throttle above the minimum flow until the lift reaches 0.030 in. To limit seat abrasion and cavitation, the first portion of the lift is accomplished very rapidly, and then normal lift speed is resumed for throttling. On cracking the first 0.001 in., pressure drop is divided between the seat joint and the plug annular clearance in the throttle orifice, and the major portion of drop rapidly shifts to the orifice. In effect, two valves are momentarily in series, thus reducing cavitation as illustrated in Figure 23.

The cascade trim illustrated and explained in Figure 24, is another means of reducing cavitation.



Provides a series of pressure drops prior to seating surface, thus reducing the velocity across the seat joint. The gap between plunger and cage in the labrinth groove section is held to 0.002 in. for proper cascade effect. A differential seating angle of 20° plug mating a 22° seat, allows seating approaching 100,000 psi, causing a slight deformation of the 440-C SS trim hardened to 53-56 R_C. Used in liquid service above 1500 psi pressure drop and steam service above 2500 psi Δ P. One year of service, throttling 300°F water at 4000 psi pressure drops and 0.050" lift did not damage the seating surfaces.

Figure 24. CASCADE TRIM.

Courtesy Copes-Vulcan, Inc.

In a full port valve, the cylindrical section of the plug above the contour run-out, and below the seat joint, may be lengthened so that the seat is removed from the joint before large flow occurs.

Hard Facing

Trim is hard faced to increase erosion and/or corrosion resistance, and also to reduce galling at high temperatures. Stellite®, Colmonoy®, and tungsten carbide are the principal materials, although the latter has limited corrosion resistance and is subject to thermal shock fracture. The corrosion resistance considered here is a mild-pitting type, which might occur in the plug-to-seat joint contact areas. Hard facing is also used on the plug-to-seat joint to maintain a tight seal. The hard-faced areas are illustrated in Figure 25.

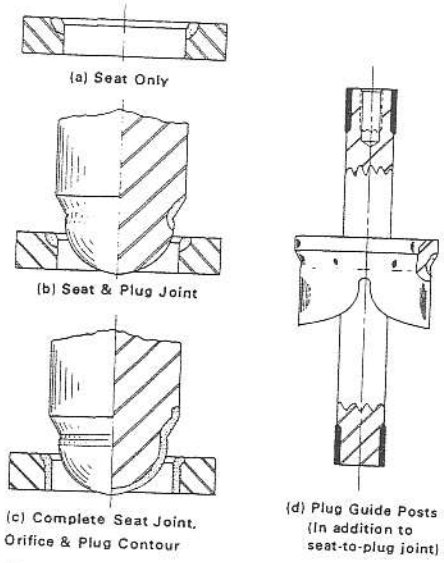


Figure 25. HARD FACED AREAS.

The following is a summary of hard-faced areas for location versus service:

Component	*Service*
seat only (Figure 25a)	Normally open. Non throttling service, closing against high pressure drop.
seat and plug joint (Figure 25b)	Throttling service. Moderate pressure drop and abrasion.
	Normally Closed / opening to high pressure drop.
Complete Seat Joint, Orifice and Plug Contour (Figure 25c)	Tough throttling service. High pressure drop and severe erosion.
Plug Guide Posts, In addition to seat-to-plug joint (Figure 25d)	High temperature service.

Stellites® are patented alloys of hard tungsten and chromium particles in a softer cobalt matrix, which make up a group of materials for valve trim. Number 6 Stellite is used on valve seats, while the slightly harder, but more brittle Number 12 may be used on plugs. The erosion resistance of Stellites is higher than indicated by their surface hardness, which is a measure of the matrix rather than micro particle hardness.

Stellite parts of small valves of 1/4 to 2 in., may be cast solid, cast as rings and projection welded, or applied as hard facing by welding to critical erosive areas of base trim material. Projection welding is a form of electric resistance welding where sharp ridges of the two parts are fused together. The degree of hard face coverage required, as well as economics, determine the method selected. Complete coverage of plugs above 4 in. size is not practical, due to cost and welding difficulties; however, there is no limit to hard facing seat joints of both plug and seat. Twenty four inch control valves with Stellite facings, have been made and used successfully. Figure 26 shows the automatic welding of a Stellite face to a large, quick-opening plug.

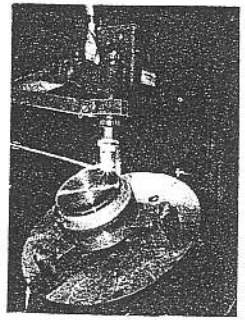


Figure 26. STELLITE - FACING A QUICK OPENING PLUG, USING AUTOMATIC EQUIPMENT.

Courtesy of Rockwell Manufacturing Company

Colmonoy® hard facing (for erosive service) is an alloy of very hard chromium boride crystals in a nickel base, which is also corrosion resistant (similar to inconel). It is applied to all trim shapes by the spray welding process and is then fused to give a non-porous surface. It has a high, hot-hardness and holds this hardness with thermal cycling. A plug with Colmonoy hard facing is shown in Figure 27.

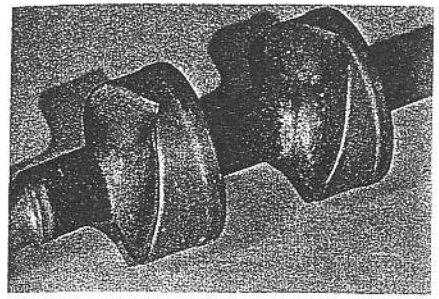


Figure 27. COLMONOY® COATED AND FUSED V-PORT DOUBLE SEATED PLUG.

Ready for grinding seat joint, skirt, and guide post diameters.

Courtesy Wall Colmonoy Corporation.

Tungsten carbide trim is generally made by pressing a powder mixture, sintering, and grinding to size. Various alloy binders of cobalt, chromium and nickel are used to give the desired properties of impact resistance, compressive strength and ability to withstand some degree of thermal shock; however, these vary the apparent hardness.

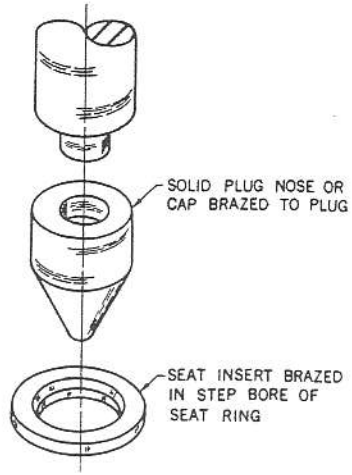


Figure 28. TUNGSTEN CARBIDE INSERTS FOR HARD FACING VALVE TRIM.

Finish ground after attachment by brazing or epoxy bonding of cylinderical fits.

Tungsten carbide is attached to valve parts by brazing, epoxy cement bonding and interference fits. Brazes have been developed for all temperatures, while the bonded trim thermal limit is about 200° F with a bond tensile strength of approximately 8000 psi as compared to 70,000 psi for brazing. Service temperature and thermal shock must be given careful consideration in its application. Valve seat loading must be even and impact of closure should be low, to prevent cracking this material. Cost is generally three to four times that of Type 316 stainless steel trim. (See Figure 28.)

HIGH TEMPERATURE TRIM MATERIAL AND DESIGN

(For Service Above 450° F)

The material properties, considered for establishing maximum operation and peak temperature limits are: tensile; yield; creep and rupture (above 800° F); hot hardness; impact strength; and aging. Other factors to consider are: scaling resistance; heat treating temperature; and galling. The design considerations include: clearances of moving parts and fitted parts (loosening) as related to the differential rate of thermal expansion of their respective materials; and galling of material combinations related to temperature, loading and degree of contact such as repeated impacts from closing or vibration. Also to be considered are the thermal cycling effects on: valve sealing; seat gasket sealing; and loosening of guide bushings.

Materials

Yield, tensile and compressive strengths are lowered by high temperatures, and above 800° F, creep and rupture become factors. In high temperature service, trim undergoes an initial elastic deformation and then continues to deform or "creep" with time, under load. Plastic deformation occurs at stresses lower than the yield strength for a given temperature. This applies to stem-to-plug attachment loading and the compressive seat, stem, and plug loads. Also, the compressive load, holding the cage trim in place, is reduced.

Parts, designed with such low stress levels that creep does not occur, are heavy and uneconomical; therefore, the designer, knowing the creep rate of a material, must select a stress, such that the total creep over the normal service life will not extend into the rupture region or allow deformation to cause interference of moving parts.

Hot hardness is necessary to prevent damaged seating surfaces and to prevent galling. Aging and the accompanying changes in physical properties such as toughness, grain size, etc., must be considered.

Scaling resistance is the ability of a material to withstand oxidation on thermal cycling without repeated scaling or flaking of the surface, exposing new metal.

Maximum temperature limits must be chosen, with a buffer zone to prevent a change of metal hardness, when service conditions approach or exceed the heat treating temperature which would anneal the plug, seat joint and erosion-resistant surfaces.

Differential Thermal Expansion of components is carefully considered in high temperature valve design. When hot fluid enters a cold valve, it surrounds the plug and rapidly brings it to line temperature with plug heat-loss limited by conduction through the, much smaller, cross-section stem and/or plug guide posts. The seat ring, due to its low mass, heats as rapidly as the plug but the heat is initially conducted through the body bridge wall. Since the linear coefficient of thermal expansion of the body is often less than the seat, it acts as a restraining member, limiting the seat expansion in a radial direction. Plug post and guide bushings are similarly affected. These conditions act to reduce the working clearances designed into the standard valve; therefore, the room temperature clearances of standard valves are increased when they are manufactured for hot service. Thus, at maximum working temperature, clearances are nearly normal. The plug post makes a tighter fit in the guide bushing than the plug cylinder-to-seat orifice clearance, in order to prevent galling and drag. As the guide clearances are increased, the plug seat orifice clearance must also be increased. For cold service, or flow testing, this has the effect of reducing the rangeability or control of low flows.

Double seated valves have problems, because the linear expansion of the plug between the two seat joints may exceed the body expansion which supports these seat rings. This problem is avoided in balanced cage valves, which use two separated metal seats, by making the plug and cage of the same material.

The following example is given for a 2 in., double seated valve:

Full port seats separated by a 3 in. elevation.

Plug material, 316SS, 10.3×10^{-6} in/in/° F coefficient of linear expansion.

Body material, Carbon Steel, 6.3×10^{-6} in/in/° F coefficient of linear expansion.

The difference is 4×10^{-6} in/in/° F coefficient linear expansion.

Plugs lapped to fit both seats simultaneously at 70° F.

At 270° F, $\Delta T = 200°$ F

(200° F)x(4 x 10-6 in/in/° F)x(3 in.) = Differential Expansion of 0.0025 in.

Thus with the lower plug seated at 270 F, the top plug would be 0.0025 in. off its seat, and a high leak rate would result.

To overcome the above problem, the plug of high quality valves is often made of the same alloy steel as the body, and the seat joint is hard-faced.

Thermal Cycling causes seat rings and guide bushings to loosen, so they must be either seal-welded or tack welded, respectively. For very high temperature, in large valves it is better to use a hard facing applied directly to the bridge-wall seat opening; this is known as an integral seat. Since seat gaskets must be loaded beyond yield to seal, severe thermal cycling causes them to creep and leak; therefore, they are best avoided for high pressure thermal cycling service.

Trim Design Versus Service Temperature

The following design changes must be considered for the corresponding temperature ranges.

Above 450° F	- The bonnet extension requires a longer stem to keep the packing cool.
Above 600° F	- Clearances must be increased. The plug and seat sealing areas must be hard-faced.
Above 750° F	- All threaded seat rings must be seal-welded to prevent loosening which will cause leakage and undercutting.
Above 900° F	- All guide bushings, plug guides and posts must be hard-faced. All guide bushings must be tack-welded.
Above 1050° F	- Hard-faced, integral seat joints must be used.

An additional consideration is that the fitted parts are subject to increased compressive stresses and creep due to differential thermal expansion, adding strain to the bonnet bolting. This involves the cage trim held by the body-bridge wall and the bonnet.

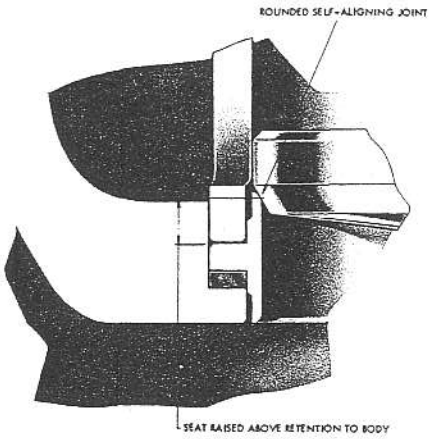


Figure 29. FLEXIBLE SEAT FOR THERMAL CYCLIC SERVICE.

Courtesy Fisher Controls Company

Seat Joint Designs for Thermal Cycling Service

A flexible seat-lip design, as shown in Figure 29, will allow for variations in the expansion rates of parts, for strains causing out-of-roundness and for seat wear caused by many closings. This design has been proven satisfactory in: gas dehydration at 60 to 600° F, 8 times/day for 40 psi pressure drops; and liquid sodium service at 300 to 1200° F, for 50 psi pressure drop.

Figure 30 is a comparison of the hot-hardness of the hard-facing materials, Stellite® and Colmony®. The 400 series, stainless steel-hardened trim materials are shown for reference. Note the rapid softening of the 400 series, above 700° F.

The following are the affects high temperature has on the physical properties of Type-316 stainless steel trim base material:

Physical Property	*70° F*	*1200° F*
ultimate tensile	85 ksi [1]	56 ksi
yield	38.5	20 ksi
Creep (1% in 100,000 hrs. or 11 yrs.)	(see [2])	6 ksi
Rupture (in 100,000 hrs.)	(see [2])	12 to 16 ksi

Galling

Galling is related to temperature, material pairs, surface finish, hardness, and loading; plus the line-fluid effect. One metal surface will score, weld-to, and/or ball-up material from another to: 1) jam the valve stroke; 2) ruin the seat joint; or 3) increase the operating force, causing jerky plug positioning. High temperature will either anneal or soften the metals, in-

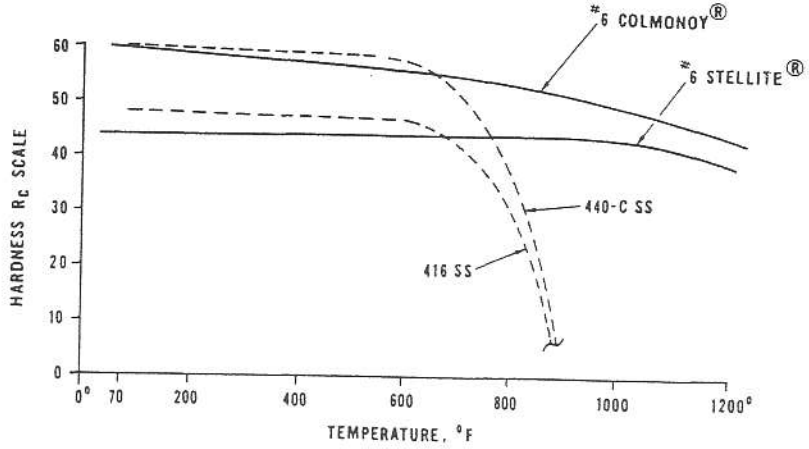


Figure 30. HOT-HARDNESS OF HARD FACING MATERIALS.

[1] Ksi = units of 1000 psi

[2] Creep and rupture below 800° F are insignificant.

creasing their galling potential.

Large abrasive particles in the line fluid will sometimes rough-up the soft trim, initiating galling. Vibrational impact may create a "fretting" condition, leading to guide-bearing surface breakdown and eventual galling.

Liquid metals, such as sodium and potassium, clean the protective oxide films from metal surfaces so effectively, that galling is a serious problem at and above service temperatures of 400° F. Materials must be carefully selected and loading must be kept at low levels for this kind of service (for example, heat exchange in nuclear power applications).

Galling may be prevented by the following:

1) Using hard materials for one, or preferably both, parts;
2) Selecting pairs of material with low galling potential. (See the list given below).
3) Using different materials for both parts rather than the same material.
4) Assuring a 5- to 10-Rc difference in the hardness of the materials.
5) Specifying either a required surface finish and surface hardness or special coating.
6) Designing a reasonable loading.
7) Designing adequate operating clearances.
8) Breaking in the valve at low loads and with a lubricant.

The following metal pairs have low galling potentials:

Plug		*Guide Bushing*	*Rating* [1]
312 SS [2]	vs	17-4PH SS	Good
		400 - Series SS	Good
		Stellites	Good
		Colmonoys	Good
		Inconel	Good
304 - L SS	vs	Inconel	Good
440 - C SS	vs	440-C SS	Good
		17-4 PH SS	Good
Tool Steel	vs	17-4 PH SS	Good
Monel	vs	K-Monel	Good
K - Monel	vs	400 Series SS	Good
Stellites	vs	Stellites	Excellent
		Inconel	Excellent
Colmonoys	vs	Colomony	Excellent
Durimet 20	vs	Carpenter 20	Good
Hastelloy "B"®	vs	Hastelloy "B"®	Good

[1] When used within recommended temperature limit and proper hardness and loading.

[2] 300-Series stainless steel readily galls with itself when in sliding contact, but the plug-to-seat sealing does not usually involve this type of contact.

LOW TEMPERATURE TRIM MATERIALS AND DESIGN

The trim design for low temperature and cryogenic service is based on the following requirements: 1) bubble-tight sealing at low temperature; 2) minimum heat leak; 3) minimum cool-down mass; 4) quick change design; 5) simple design; 6) adequate impact strength of materials at low temperature; and 7) differential thermal contraction of materials. These factors have been resolved in valve design and material selection which have performed very satsifactorily.

Bubble-tight sealing, a must in cryogenic service, is obtained by using TFE or Kel-F on the plug seal. These materials become much harder at low temperature, but still retain enough yield to seal tightly. Kel-F has sealed 225 psi, bubble-tight, at 10° F above absolute zero. Both materials may be used for the guide bushings and the seat joint gaskets.

The stems must pass through extended bonnets to prevent the freezing of atmosphere moisture, resulting in binding of the stem at the packing exit, and to clear the thick insulation of cold boxes.

Hollow plugs are frequently used to prevent rapid heat transfer, yet allow a large-diameter plug for guiding and throttling stability. The void in the plug may be evacuated and/or filled with insulating material or reflectors to reduce radiant or convective heat transfer. Stainless steel is an excellent material due to its poor thermal conductivity; however, cross-sectional metal area is kept to a minimum. The clearance between plug extension and guide is close enough to prevent convection currents and it enlarges, above the guide, to a larger annular area to prevent pumping of liquid out of the body into this space.

Valves are normally installed either vertically, or up to 45° from vertical, to maintain a low-convective vapor-lock in the extended bonnet. The design shown in Figure 31 is applicable to valves which must be installed horizontally. The seal "seals" on downstroke and bypasses on upstroke.

The weight of the parts should be kept to a minimum, to reduce cool-down of the mass and boil-off loss of the liquified gases that are entering the piping system. To reduce weight, the large plugs are hollow and the seats may be integral with the body.

Quick-change trim, with removal through the bonnet, is a must, because

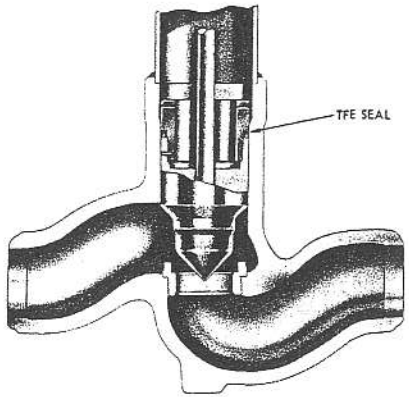


Figure 31. SEAL TO PREVENT PUMPING OF CRYOGENIC FLUID INTO VALVE BONNET WITH VALVE STROKING.

Courtesy ITT Hammel-Dahl

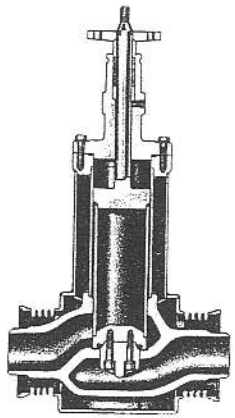


Figure 32. CRYOGENIC VALVE WITH VACUUM JACKET.

Courtesy Masoneilan International, Inc.

it is impractical to remove a vacuum-jacketed valve body from the line, or even to remove the insulation to get to the bonnet. Therefore, the bonnet is always located well above the normal position, requiring a longer plug in order to use a short, stiff stem having a small cross-sectional area necessary to limit heat leakage into the valve.

Stuffing boxes must be deep enough to accommodate a second or partial set of packing, installed back-to-back, to prevent leakage on cool-down if a vacuum should occur in the line.

One type of valve, designed for cryogenic application is shown in Figure 32. Note the vacuum jacket and the soft seat that is retained by the bolted nose plug.

Cold Impact Strength

Trim materials must possess adequate cold impact strength to prevent fracture and complete shattering. Carbon and chromium-alloy steels rapidly lose impact strength below 0° F; therefore, they are limited to -20° F and -50° F service, respectively. 3.5% nickel steel may be used to -150° F, and 9% nickel steel may be used to -320° F. Austenitic 300-Series stainless steels, nickel, Monel, Hastelloys, Durimet, titanium, aluminum alloys and bronze may be used to -460° F. These fall into the group of face-centered, cubic metals. Cooling increases their yield and tensile strength, while gradually reducing ductility. The impact strength of these materials may either remain the same or increase.

The ignition point of austenitic stainless steels in liquid oxygen, is so high that very few applications exist where they are not entirely satisfactory. If they are not satisfactory, Monel or bronze trim should be specified if very high-energy shock impacts are probable.

The following modifications are used for low temperature trim.

Low Temperature Ranges	*Trim Design Requirements*
Cool Valve Service	
0° to -50° F	The extension bonnet requires a longer stem.
	Sometimes the addition of ethylene glycol in the packing follower, annular clearance around the stem will prevent freezing and will permit operation of a standard length bonnet valve to about 0 F.
Cold Valve Service	
-50° to -150° F	The extension bonnet requires a longer valve stem to prevent stem freezing, from the condensation of atmospheric moisture at the exit of the stem from the packing.

Refrigeration Service
(Typical operating temp.)

Freon (-92° F) Ammonia (-84° F) Propane (-106° F)	The guide bushing may be tack-welded to prevent loosening from the differential thermal contraction of the bushing and bonnet materials. The seat rings, of screwed-design, are seal-welded to prevent loosening and leakage.

Cryogenic Service

-150° F to -460° F	A double extension bonnet is used with a length of 12 inches for -320° F nitrogen service, and longer for -450° F He and H_2 Service.

Liquified Gases Service [1]
(Typical operating temp.)

Methane (-259° F) Oxygen (298° F) Nitrogen (-320° F) Hydrogen (-422° F) Helium (-452° F)	The guide bushings may be either welded or lock-rolled in place. The operating clearances for moving parts may be increased by 50%, to prevent binding from differential thermal contraction. A hollow plug extension is used - sealed void, with a low cross-section metal area. Extension in large valves is carried high enabling the use of a short and stiff stem due to the requirement of a long bonnet extension. The seat rings, of screwed-design are seal-welded to prevent loosening and leakage.

The following trim materials are used for plugs and seats, and bushings in cold service.

Plugs and Seats	*Guide Bushings*
Bronze Type-316 SS K-Monel Stellite or Collmonoy hard facing on 316 SS TFE or Kel-F is used as plug-seat joint material and seat gaskets. They are attached to the plug with strain-hardened 300 SS bolting or stem nuts.	Hard Aluminum Bronze K-Monel Hard Chrome Plated Materials 17-4 PH SS and Type 440-c SS are uses for guides only where brittleness is not a factor.

[1] Boiling points at 14.7 psia.
[2] Ksi = units of 1000 psi.

The effect of low temperature on the mechanical properties of type 316 stainless steel trim material is given below:

	-70° F	-423° F
yield strength (0.2% offset)	35 ksi [2]	84 ksi
ultimate tensile strength	84	210
elongation	70	55
impact strength (Izod, "V" notch test)	110	110 (at -320° F)

The following are the cold properties of the soft seat materials (TFE and Kel-F).

Material	*Tensile Strength (psi)*	*Thermal Contraction (from 70° F)*
TFE	1,500 (100° F), 22,000 psi (-460° F)	2.2%
Kel-F	4,000 (100° F), 16,300 psi (-300° F)	1.2%

TRIM EFFECTS ON NOISE

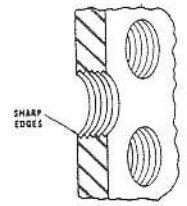
Trim contributes to valve noise by the following: 1) mechanical vibration; 2) natural frequency vibration; 3) throttling instability; 4) flow media - the cavitation of liquids or aerodynamic effects of gas flow; and 5) water hammer-shock on valve closure. These effects are discussed in detail, in Chapter 7.

Mechanical vibration may be reduced by: 1) holding close diametral clearances; 2) using heavy guiding to distribute the impact load and to dampen vibration; 3) selecting materials that will withstand heat and reduce wear, preventing the enlargement of clearances; and 4) using an elastomer dampening ring on the heavy plug guides of cage valves, which may also serve as the seal on pressure balanced cage designs;

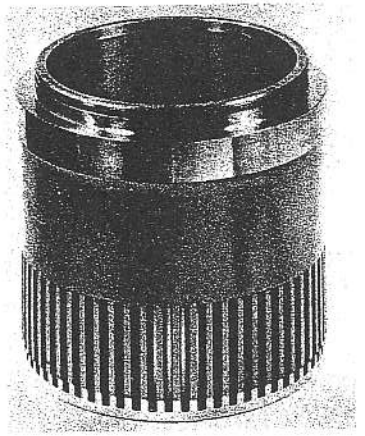
Natural frequency vibration may be eliminated by: 1) breaking up symmetry by using solid-cast plugs and parts rather than using cylindrical skirts welded to posts made from bar stock; 2) changing from a skirted plug to a contoured plug, or vice versa; 3) reversing flow direction; 4) changing stem diameter; and 5) using single-seated valves with heavy plug guides (not guide posts), which are less susceptible to vibration because of greater plug stiffness.

Throttling Instability is the vertical oscillatory movement of the assembly, including plug, stem, and moving actuator components. Both single and double ported non-pressure-balanced valves are unstable when throttling to a high

pressure drop, at low lifts, as explained in the "dynamic flow effects" section. Tremendous up-and-down forces, caused by flow impingement upon the plug, rapidly reverse their direction and/or magnitude, as the plug is moved by the actuator. This effect may be amplified by an actuator with positioner, whose combined frequency of response may fall out of phase with the required control action. This causes pressure fluctuations in the flowing media, generating a rumble with a

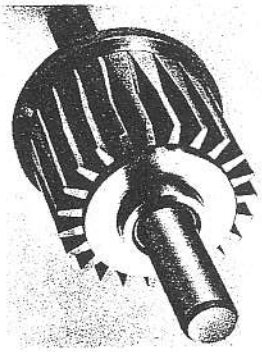


(a) Multiple, high velocity orifices used in flow cages to reduce noise.



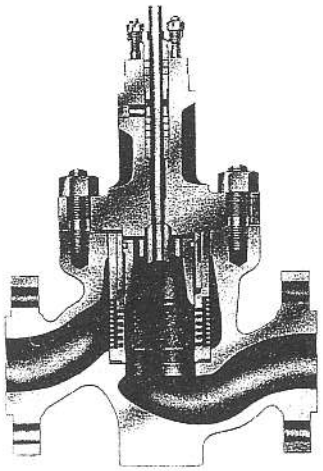
(b) Slotted Cage
Provides multiple orifices which increase frequency and achieve attenuations of 10-15 db.

Courtesy Fisher Controls Company



(c) Multi-V-Port Plug
Creates many Small Orifices (Attenuation - 12db) Used in Gas Pressure Regulators.

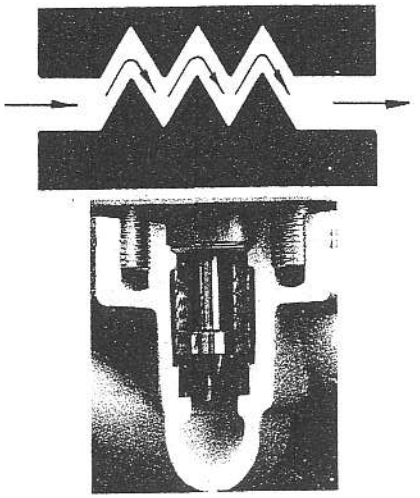
Courtesy Leslie Company



(d) Double Perforated Low-Noise Cage Trim
Dual expansion cage reduces noise substantially while raising frequency of aerodynamic noise allowing higher attenuation by heavy valve body wall. Flow area of outer stage is larger than inner so as to maintain low velocity of the expanding gas.

Patent Pending

Courtesy Masoneilan International, Inc.



(e) Low Noise Trim
Sharp turns in flow path generate multiple velocity head losses attenuating aerodynamic noise and reducing cavitation. Flow is radially outward for gas to give an expanding area constant velocity flow.

Courtesy Valtek, Inc.

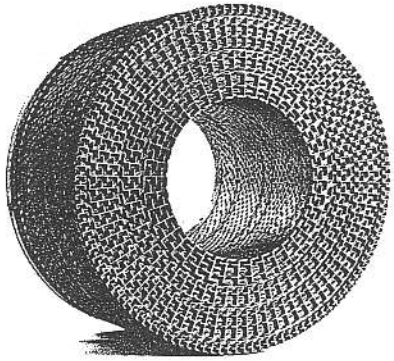
Figure 33. LOW AERODYNAMIC NOISE TRIM DESIGNS.

frequency of around 30 Hz. Vibration depends upon the mass-spring rate relation-ship of the plug-stem-actuator moving parts mass. Seats, plugs and stems are damaged by the vibration and either leakage or broken stems result. In addition, the stem-packing wear rate is increased.

Throttling instability may be reduced by: 1) using a stiff actuator (high spring rate); 2) having a pulsation damper, also known as a "hydraulic snubber," installed on the actuator stem; 3) designing a pressure-balanced cage which reduces the magnitude of the unbalancing forces, therefore, improving stability; and 4) maintaining a fast rate of frequency response for the controller-positioner-actuator combination.

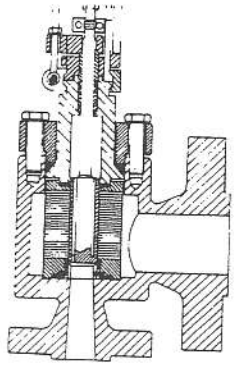
Flow media noise includes: 1) cavitation noise, caused by the impact of collapse of vapor bubbles formed in the passages of the seat-to-plug throttling annular clearance at high pressure drops; and 2) Aerodynamic noise, caused by high-pressure gas entering and exiting at the valve ports, is extremely annoying. Aerodynamic noise is also caused by pressure recovery, accompanied by a reduction in velocity, in the downstream passages, causing sonic shock waves.

(a) 4 in., 2500 lb. Boiler Feedwater Recirculation Valve



(c) Flow Disc Stack for Gas Service

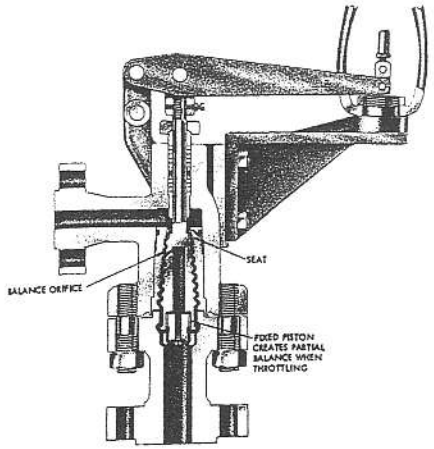
In (c), the multiple, right angle turns in the staggered flow paths of each disc affect numerous velocity head losses. This alters the basic velocity equation from $V=\sqrt{2gh}$ to $V=\sqrt{2gh/N}$, where N represents the number of turns in series in each flow passage. A proved pre-determined number of discs with multiple flow passages etched into their faces is stacked around the plug. The number of fluid passages, open to flow, increases with valve plug lift. Noise and erosion are greatly reduced in high-presure-drop service. Designs are also available for anti-cavitating liquid service. "V" is a selected quiet outlet velocity below sonic, usually 300 ft. per second maximum. "h" is the differential pressure across the valve and "N" becomes the number of right angle turns, each causing one velocity head loss. Solving for "N" determines the required number of 90° bends and overall disc diameter. The flow characteristic with a uniform number of disc orifices in each disc is inherently linear. An equal percentage characteristic may be obtained by increasing the number of orifice openings in each succeeding disc exposed to flow as the plug opens. Some designs have an open cage at the top of the disc stack to give minimum flow restriction, but at velocities well below sonic as differential pressure decreases with valve opening.



(b) Flow Disc Stack in Sectionalized Valve

Figure 34. PRESSURE REDUCTION AT CONSTANT VELOCITY REDUCES NOISE.

Courtesy Control Components, Inc., Patents Issued and Pending



The tapered plug allows gas to expand along a tortuous path with little increase in velocity, since the area of flow-path is expanding; thus, greatly reducing noise. Adiobatic flow, with molecular friction, occurs at a nearly constant level of enthalpy, as pressure drop occurs in steps across a series of orifices. After opening, the plug is partially pressure balanced, but when closed the full pressure drop load is applied to the seat. Partial balance acts to close the valve for safety and for stable throttling. Maximum flow velocity is approximately one-fourth that of a conventional valve; therefore, trim erosion is not excessive, even though particles may impact the throttling surfaces.

Figure 35. SPECIAL LOW NOISE VALVE DESIGN.

Courtesy Masoneilan International, Inc.

Aerodynamic noise may be reduced as illustrated in Figures 33, 34 and 35 by:

1) Using trim that has many, small high-velocity ports which operate at a higher frequency, thus, increasing the attenuation through the body wall.
2) Using trim that has sharp edges and a distorted flow pattern which will reduce gas velocity and vortex size. Streamlining the flow has a negative effect, because it permits further isentropic expansion of the sonic or supersonic jet flow, causing greater shock waves upon pressure recovery.
3) Using in-line valves with expandable elastomer stem plugs rather than standard globe valves, because of the dampening effect of the plug sleeve.
4) Application of acoustical barrier treatment is nearly always necessary for standard trim.

Water hammer-noise and shock are caused by the sudden closure of valves that are stopping a moving column of liquid and by the reflection of shock waves within the liquid and piping system. Intensity varies directly with the density, velocity, and rate of deacceleration of the flowing liquid. In severe cases, flanges may be parted, causing temporary leaks or blown gaskets, line supports may be ripped loose, or piping ruptured in iron pipe systems. Linear and equal percentage plugs usually cause no problem with normal controller-closing speeds; however, with on-off snap acting controller settings, they may give some trouble. Quick opening plugs may require a throttling contour on their last 10 to 15% of travel before closure; this is advisable in the case of high pressure drops, having high flow rates. Hydraulic "snubbers" installed on the actuator stem, may be designed to slow valve closure just before shut off; beyond this, pulsation dampers should be installed in the piping system (these will require "tuning" of their gas charging pressure to match the system characteristics).

Two-phase (liquid and gas) "slug" flow causes some shock when passing through a control valve; however, the gas cushion in the flow line usually limits this to a minor shock.

TYPES OF PLUG GUIDING

Guiding is the means by which the plug is aligned with the seat before the contact for tight seating by the normal stem force. The guiding must resist all side thrust on the plug, from dynamic fluid flow conditions.

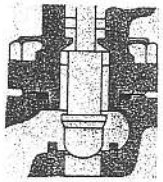
Guiding design is divided into the following four general types:

1) *Top Guiding*, illustrated by Figure 36(a), has a single, extra-long bushing which guides the plug from above. This is used with a contoured, single-seated plug. It is used for high pressure drop in sizes from 1/4 in. to 1 in. Valves above 1 in., use top and bottom guiding for globe bodies.
2) *Top and bottom guiding*, as shown in Figure 36(b) uses contoured and V-port plugs with post extensions that are guided above and below the single or double plug in bodies with bottom covers. This design allows plug reversal to change the closing action and the flow direction. It should be specified for pressure drops greater than 100 psi.
3) *Port guiding* is shown in Figure 36(c). In this design, the plug is guided in the seat ring by a skirt on the plug. It is used in bodies with V-port plugs and is used in some three-way valve designs. Port guiding is also called skirt guiding.

It is used for the moderate pressure drops under 100 psi.

4. *Cage guiding*, as illustrated by Figure 36(d), the cage guides the plug. This design is for moderate to very high pressure drop.

Other types include: (a) combined top and skirt guiding; (b) Stem guiding of large stems, as shown in Figure 36(e); and (c) Bottom guiding of the stem in extension bonnets, as shown in Figure 36(f).

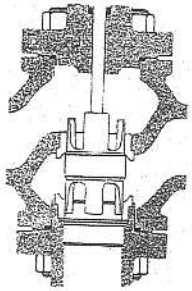


(a) Top Guiding

Courtesy Honeywell Industrial Division

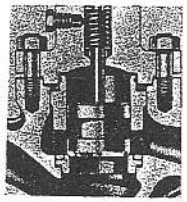
(b) Top & Bottom Guiding

Courtesy Honeywell Industrial Division



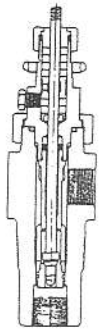
(c) Port Guiding

Courtesy Honeywell Industrial Division



(d) Cage Guiding

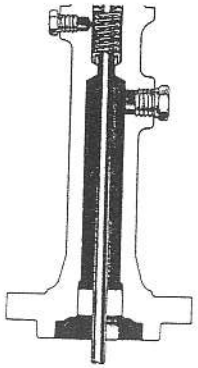
Courtesy Fisher Controls Company



The body design illustrated is used to extend the seat joint into the vessel interior to prevent freezing on high pressure gas, pressure-reduction service.

(e) Stem Guiding

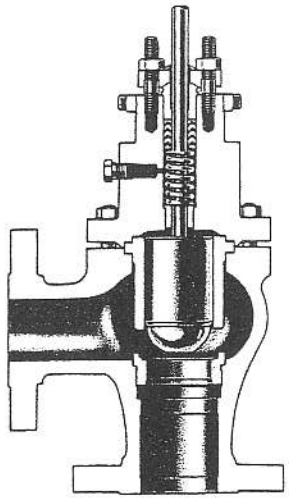
Courtesy Fisher Controls Company



(f) Bottom Guiding of Stem in Extension Bonnet

Courtesy Fisher Controls Company

Figure 36. EXAMPLES OF GUIDING DESIGNS.



37. HEAVY DUTY CYLINDER GUIDE IN ANGLE BODY VALVE FOR HIGH PRESSURE DROP SERVICE.

Courtesy Fisher Controls Company

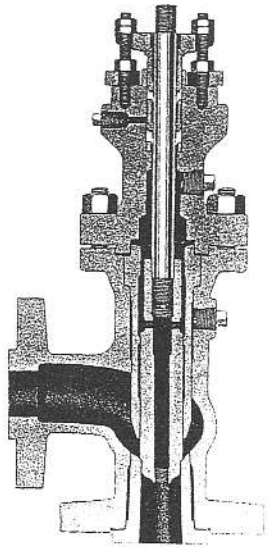


Figure 38. FLUSHING TYPE TOP EXTENSION GUIDING.

Courtesy ITT Hammel-Dahl

Extra heavy guide bushings, shouldering in the body bonnet neck, are used in high pressure drop angle valves of size larger than 1 in. as shown in Figure 37. This design is ideally suited for flow under, thus allowing larger flows with the smaller seat orifices and lower actuator forces that result from the smaller seat. The "stubby" plug deadens high frequency vibrations, giving quiet operation in steam service.

Figure 38 is a valve design that has flushing type, top guiding. This type is used in "dirty fluid" service. The guide may be continuously flushed by supplying a clean fluid into the plugged opening above the bonnet flange. Note the pressure balanced plug (the stem is not balanced).

Redundant guiding should be avoided for accurate plug-to-seat alignment. Cage guiding, followed by top guiding, best meets this requirement. Top and bottom guiding is required for plug side thrust stability in some designs; however, the two guides and one seat must be in perfect axial alignment after assembly. Skirt guiding maintains concentric alignment, but offers little restraint to cocking, because of the low ratio of skirt guide length to seat ring length at relatively large diameters. (The L/D ratio is too small.)

Guide fit clearances, materials, hardness, etc. are discussed throughout the chapter.

The characteristics and requirements of stem post guide bushings include:

1) They are press-fitted for normal service.
2) They are tack welded for thermal cycling at service below -100° F [1] and above 900° F. [2]
3) They are hard-faced for use at temperature above 900° F.
4) They may be screw-fitted with a pilot diameter providing axial alignment and a shoulder that will prevent cocking, thus offering fast maintenance advantage. This design, as illustrated by Figure 39, enables simple field replacement of the plug guide post or the stem guide bushings.

[1] The bushing materials will contract more than the bonnet materials on cool down; thus, the bushing might loosen and fall out if it is not tack-welded or lock-rolled in place.

[2] The bushing material expands more than the bonnet bore and could yield the bonnet, thus loosening the interference fit.

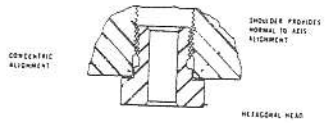


Figure 39. SCREWED GUIDE BUSHING.

The cage guiding technique requires a hardened plug, a Stellite® or Stellite®-faced plug, or a hard chrome-plated plug and guide cage.

DYNAMIC FORCES ACTING ON TRIM AND AFFECTING THROTTLING CONTROL ACTION

Valve plugs, when throttling flow, are subject to forces that are axial along the stem, at 90° to the stem axis and rotational to the plug and stem axis. The side forces cause bending moments and vibration.

Improved flow control can be obtained by valve trim design that will reduce the magnitude of the forces tending to unbalance the plug position and this will reduce the operator force requirements, making a more sensitive valve.

The following considerations will increase valve stability:

1) Select a sensing and controller system that has a high frequency response.
2) Use a relatively stiff actuator, one which has a high spring rate, so that plug force changes must be large to cause a substantial change in plug position.
3) Use flow under the plug when feasible, to eliminate the large "suck downward" force as the plug throttles near the seat. Fluid forces acting on the stem should always be compressive, forcing the plug out of its seat for best stability.
4) Use a small diameter stem to reduce the effects of line pressure change thrusting on the stem area, but keep the stem rigid enough to prevent bowing (packing wear and drag) and to reduce the natural frequency vibration.
5) Select proper trim designs for high energy dissipation to reduce plug drag.
6) Streamline the liquid flow to reduce the variation of flow patterns within the valve, which cause different pressure drops as the flow cycles from one to the other.
7) Use a hydraulic snubber on the actuator stem for severe pulsating service (such as a bypass on duplex or triplex, pump discharge pressure control).
8) The use of cage trim increases stability over regular trim by improving the the flow pattern and reducing side thrust friction and vibration.
9) Use balanced cage trim to counterbalance most of the forces across the plug so they are never transmitted to the actuator stem. Balanced cage trim is most stable of all.
10) Balance the valve stem thrust for extremely high line pressure (10,000 psi and up).
11) "Stubby" plug shapes with heavy guides may be used to help reduce vibration.

Stem Thrust

Stem Thrust varies with the body fluid pressure acting on the unsupported, cross-sectional area of the stem in the packing, thus tending to force the stem out of the bonnet. Figure 40 presents a comparison of stem thrust with various ratios of stem-to-seat diameter. It is usually a small part of the total plug load and may be discounted except for high pressure operation. Conversely, in very small orifice valves, the stem may be equal or larger in size than the port. Thrust also varies with the plug nose position in the orifice (for plugs without guide posts); here the vena contracta, at high flow rates, may cause a low pressure area with pressures below

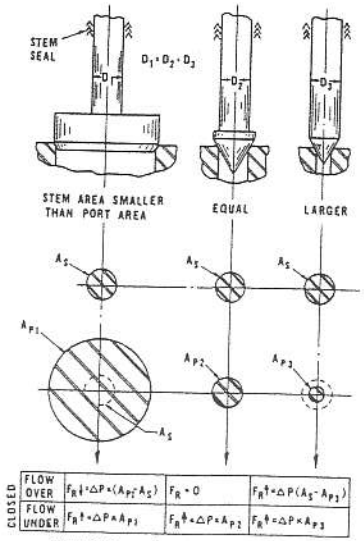


Figure 40. STEM THRUST.

the downstream body pressure for flow over the plug. For flow under the plug, the same conditions apply, plus the velocity head impact of the fluid on the plug nose cross section.

Stem thrust may be minimized by:

1) Using the smallest diameter stems that are practical for strength requirements.
2) Removing stem thrust from the build-up of pressure, when the valve is closed, by flowing over the plug. This applies only where the stem diameter is equal to or less than the orifice diameter and is effectively used in angle valves. This keeps line pressure on the packing at all times.
3) Balancing the stem for use in extremely high line pressure, or by building-up of the shut-in pressure acting on the stem area. This can be accomplished by one of two means:
   a) By using a double extension stem, that goes through both the bonnet and the bottom flange with a second packing box, as shown in Figure 41. This has been very effective for full port valves where a large stem diameter is required to handle pressure drop and seating loads of 20,000 psi WP for 2 in. valves or 15,000 psi WP for 4 in. valves.
   b) By designing a balancing piston on the stem with an annular area equal to stem area, as shown in Figure 42. This has been effectively used on plug-balanced angle valves having small stems (for example, 1 in. orifice - 10,000 and 20,000 psi WP).

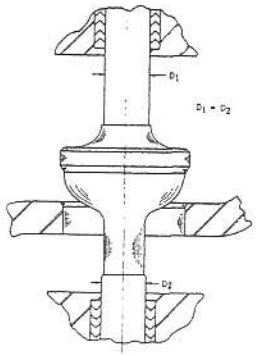


Figure 41. DOUBLE EXTENSION STEM.

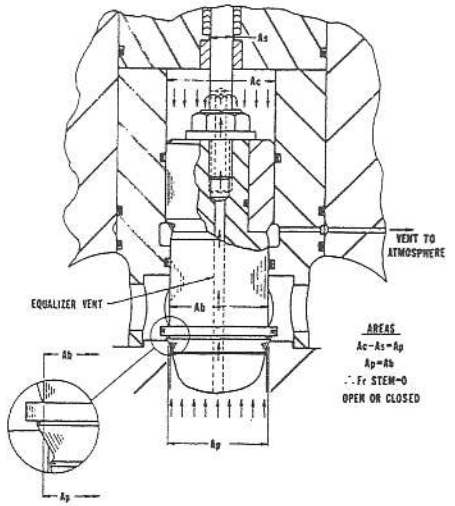


Figure 42. PISTON STEM BALANCE.

Both plug and stem are balanced. A two piece plug is required for assembly. The design is limited to small valves because the plug and balance piston seals always sense the pressure drops between the line and atmospheric pressures. This design has been effectively used on small stem plug balanced angle valves, 1 in. orifice-10,000 and 20,000 psi WP.

Side Thrust

Side Thrust of fluid, forces the plug to one side of the guide bushing(s) increasing the operating friction and the "dead band" for the actuator. This may cause "jerky" plug positioning. Thrust effect depends upon trim type, flow velocity, fluid density, pressure drop and the flow pattern entering the valve body. Plug guide posts add to the side thrust by interrupting the natural flow patterns. Natural frequency vibration of the plug-stem assembly may occur as the plug is moved horizontally in the guide bushing clearances by fluid pulsations and by change of flow pattern. It becomes a problem only with high pressure drop services or with large valves.

The following considerations will reduce side thrust:

1) Select single ported valves with plug guides (not posts).
2) Streamline flow into the valve body and provide adequate inlet and exit passageways around the seat.
3) Use cage trim that has several orifices to uniformly distribute flow around the plug.
4) For double-seat valves, divide flow into the center cavity so that the impact will not be on the stem, midway

between the two plugs. Stem bending, binding and severe guide bushing wear has been greatly reduced in high pressure applications by flowing in this direction.

Y-Body valves inherently have higher side thrust components from fluid flow forces than globe or angle valves which have a larger flow passage around the plug. At low lifts and high pressure drop they impose a high drag on the plug and guide bearing surface, which often causes galling at high temperature. For example, throttling drops of 3500 psi at 1050° F.

This may be eliminated by preventing the fluid flow from getting behind the plug, thus giving a reduced side thrust as illustrated in the design of Figure 43. The plug is closely guided in the body bore by upper and lower, hard-faced guide rings, with an annular leak-off channel between the rings. For high temperature high pressure service, the bore is also hardfaced.

Fluid Swirl Rotational Forces

Fluid Swirl Rotational Forces in the orifice react strongly on large V-port plugs (4 in. and up) and can cause disengagement of the valve and actuator stem connection, unless this joint is secure. Diaphragm twist may also occur in springless actuators. In the later case it is advisable to attach an anti-rotation clamp and guide to the valve stem extension.

Rotational forces may be reduced by:

1) Streamlining flow into the valve body.
2) Using contoured, rather than V-ported plugs.
3) Using cage designs that reduce this force by distributing flow, more uniformly around the plug.

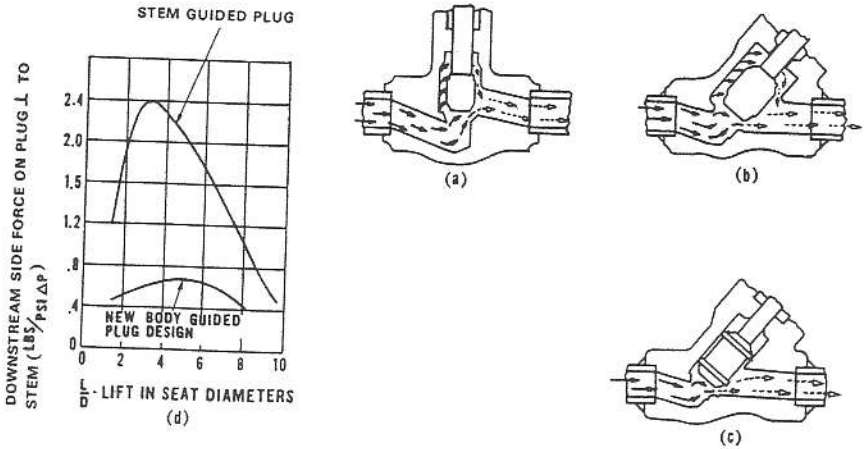


Figure 43. DESIGN FOR REDUCTION OF SIDE THRUST.

Courtesy Rockwell Manufacturing Company

(a) Vertical stem, globe valves are subject to side-thrust under high pressure drop conditions. This illustration shows how upstream pressure can slip past the stem-guided plug and impart a thrust toward the downstream side of the valve. Tests have proven that this thrust causes plug-seat misalignment plus galling and scoring.

(b) Inclined stem Globe Valves of the stem-guided type are also subject to side-thrust under the same conditions. This illustration shows the path of pressure through the valve.

(c) The new design, with Stellite® anti-thrust rings, has a lower guide which eliminates 90% of the flow upward and behind the plug. Both guide rings maintain perfect alignment. This effectively eliminates all side-thrust problems in both globe and Y-body valves.

(d) Illustrates the relationship of side-thrust in a conventional stem-guided globe valve and in a new design with body-guided plug.

Torsional Force Reactions

Torsional Force Reactions on a 4 in. V-Port Dougle Seated Valve Plug are covered in reference 36. Measured stem torque, varying from 8 to 20 ft.-lb., and oscillating at a frequency of about 30 Hertz, has been made for a 4 in. valve, flowing 220 GPM, with a pressure drop of 170 psi. Most of the torque results from an impulse reaction on the bottom skirt caused by the impingement of incoming fluid against the projected area of the edges of the "V" notches. Torque is proportional to the mass velocity of the fluid at the body inlet and impingement area presented by the lower skirt.

The torque may be reduced to ±4 ft.-lb. by changing from four to two "V's" in each skirt and by lifting the bottom skirt completely out of the orifice to achieve maximum flow. No sacrifice need be made in flow characteristic to accomplish this.

Orienting the "V's" in relation to the pipe axis may reduce torque to zero, but this not practical in economic valve design, because the angle of the "V"-to-pipe axis varies for different bodies.

Fluid Forces Acting on Plugs

When the valve is closed, differential pressure acting on the mean-seat joint area tends to open the valve for flow under or to close it for flow over. If the pressure drop is high, these forces build up quickly as the plug approaches its seat and may cause slamming and bounce-back for the flow-over direction. If throttling near the seat is continued, chattering results, which quickly damages the seat joint. This slamming action creates fluid pulsations which cause the plug-stem-actuator spring-mass to oscillate between 8 to 30 Hertz.

Flow capacity of globe bodies is generally 5 to 10 per cent higher with flow under and this goes hand-in-hand with improved throttling stability. Angle valves also have a higher flow capacity with flow under; however, they are usually installed with flow over for abrasion and cavitation reasons.

Flow under removes stem packing as a source of leakage from the build up of a high shut-in pressure.

The main advantage of flow over, is that the seating force rises with the increased pressure drop build up of the shut-in pressure. Sealing is tighter and safety shut-in is assured, even with low actuator thrust. To assure positive safety shut-in from the open position, the actuator must be spring loaded to close the valve against an opening stem thrust from line pressure. A second advantage is that stem area is subtracted from the mean seat joint area in calculating the actuator force required to open against the pressure drop. This applies when stem-to-seat area ratio is equal to, or less than 1.0. Larger stem ratios would create a self-opening action, as in the case of many small splined plugs.

The unity ratio creates a pressure balanced condition for flow over *only* and may be effectively used on small port, high pressure dump valves (3/8 in. port, 3000 psi pressure drop, and up). In this case the valve will always open at the same controller signal, regardless of change in pressure drop, since this change has no effect on actuator stem force in the closed position. This improves level control and *eliminates spring adjustments.* The need for a positioner is eliminated. Stem packing friction increases with pressure in this application.

Vena Contracta Forces

Flow through the contoured plug-to-seat annular area occurs at a much higher velocity, and lower pressure occurs in this vena contracta zone of the flow stream, than upstream or downstream of the orifice zone. This creates a force normal to the plug surface at the vena contracta which tends to draw the plug into the seat. The magnitude of the force acting on the plug, as illustrated in Figure 44, depends upon the following:

a) The downstream pressure recovery distance.
b) The width of the seat joint (length of low pressure zone) and the extent of the vena contracta along plug surface contour.
c) The plug surface angle (to horizontal at this point.
d) The pressure drop across the vena contracta zone.

The horizontal forces are balanced as shown by Figure 44. The magnitude of the vertical force, relative to other forces acting on the stem, is small for moderate size valves, except when used for high pressure drop service. The relative magnitude of the vertical force increases, however, for larger valves throttling low pressure.

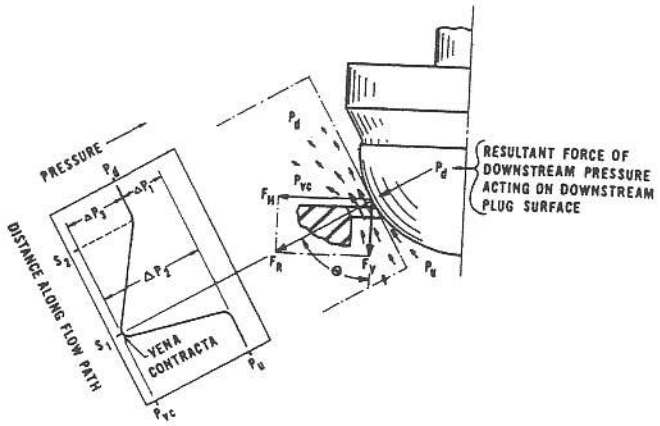
Resultant Force

The resultant of the plug stem forces should always tend to open the valve, or if this is not feasible, the forces should never change direction to any great extent as the plug strokes. Otherwise, the rate of change of force with lift (dF/dL), is so rapid that the actuator-controller system will lose stable control and cycling will occur. This would result in chatter, excessive packing wear, noisy valves and damaged seat joints. The addition of a positioner may aggravate this situation.

$F_v = F_r \cos\theta$

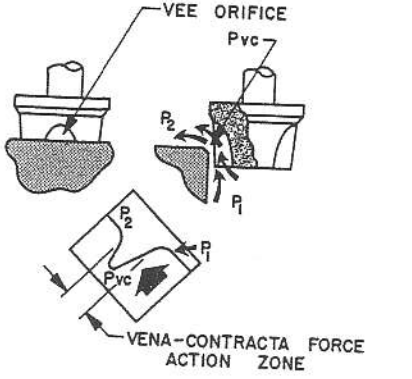
$F_r = \sum_{s_1}^{s_2} (P_d - P_{vc})\, A$

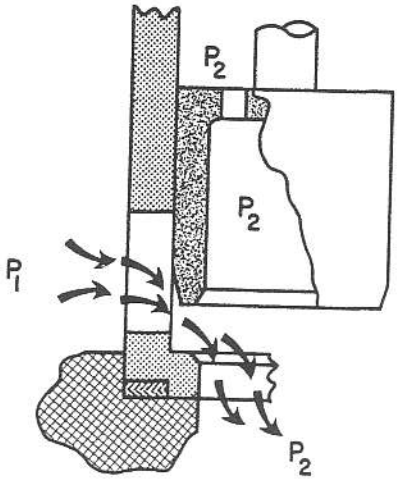
A = Plug surface area sensing a difference in P_d and P_{vc}



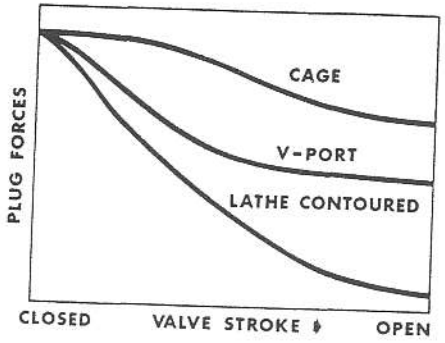
(a) Contoured Plug

(b) V-Port Plug
Please note that it has a sharper edge orifice flow than the parabolic type contoured plug. Also, the vena-contracta zone of forces acting on the plug is much smaller, thus leading to greater stability with flow in either direction.





(c) Cage Design
It has a sharp edge orifice type flow with vena-contracta forces acting over a small portion of the plug.

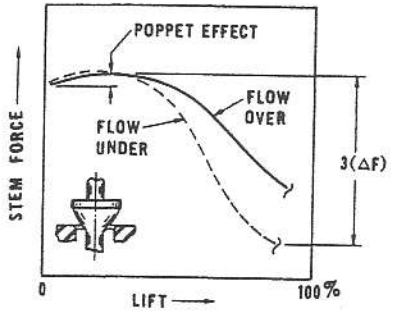


(d) Plug Forces At Constant Pressure drop Versus Valve Stroke
Please notice that as valve stroke increases the plug force decreases.

Figure 44. VENA CONTRACTA FORCES ACTING ON PLUG.

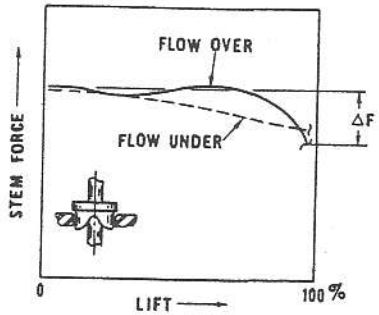
Dynamic Flow Forces

The dynamic fluid flow forces acting on several plug types are illustrated in Figures 45-50.[1] The curves given in the Figures, show forces which act axially on the plug when the pressure drop is held constant, as the valve opens. The magnitude of force change is less for V-Ports than for contoured plug shapes; and is less for double-seated than single-seated types. The reason the magnitude is less for V-ports than for contoured plugs is that the force difference between the downstream and vena contracta zone of the V-port acts on a smaller area of plug surface.



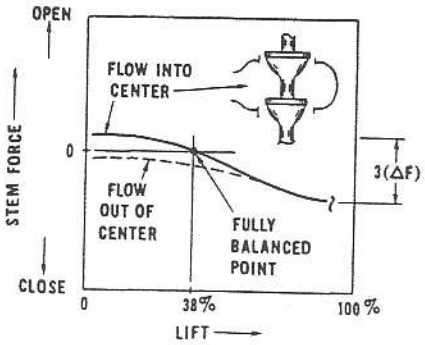
Forces act in same direction except for the poppet effect near the seat, which also occurs with flow-over due to the vena contracta effect moving downstream of the seat joint. Flow-over has the more gradual change (dF/dL); however, chatter will occur near the seat. The poppet effect of the valve on cracking open, exposes more area to the upstream pressure with flow-under, causing a force increase. Narrow seat joints and the least excessive plug diameters reduce this effect.

Figure 45. PARABOLIC.



Flow-under forces are constant in direction and have a more gradual change (dF/dL), giving the plug greater stability. V-port plugs have approximately one-third the force change with lift, compared to parabolic plugs.

Figure 46. V-PORT.



(a)

In (a) complete balance-closed is not obtained, since the upper seat orifice is larger than the lower, to pass the bottom plug. Note the much smaller change of force with lift compared with the single-seated plug of Figure 45. Flow into the center reduces slamming, but high flow rates may bow the plug, causing drag in the stem post guide bearings. Flow out of center, does not change the direction of the stem force with stroking and should give a stable plug.

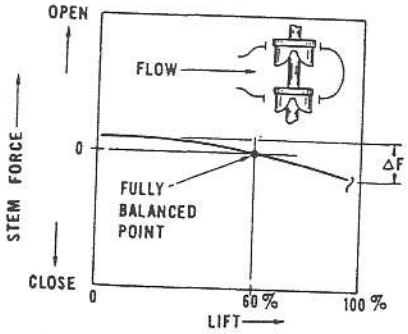
(b)

In (b) the degree of dynamic unbalance is reduced by using a different top and bottom contour form. Actual tests have shown a reduction of force to 90% by this method, applied to 4 in. valves.

Figure 47. DOUBLE SEATED PARABOLIC PLUGS.

Courtesy Fisher Controls Company

[1](Figures 45, 46, 47-a, 48) This information was originally presented in a different form in a paper, "Unbalanced Force Reactions in Control Valve Plugs", R. A. Rockwell, *Instruments*, Dec. 1950. Reference No. 37. Flow tests were conducted



Force change is one-third that of a contoured, double-seated plug (Figure 47(a)), but does change direction; however, (dL/dF) is very gradual.

Figure 48. DOUBLE SEATED V-PORT.

CAGE GUIDED AND CAGE GUIDED, BALANCED TRIM

History

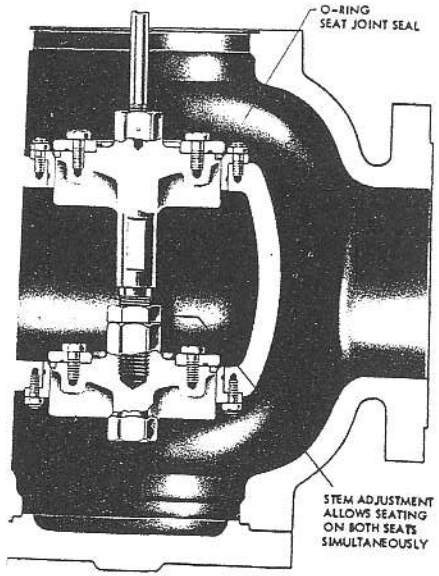
Flow cages have been used for many years as seat hold-down devices and as spacers in a stack of quick-change trim valve parts. Balanced designs appeared in early steam pressure regulators and have been effectively used in valves for high pressure oil and gas production, since the late 1940's. Characterized flow ports in cages have been common to some high-pressure, boiler feed-water, control valves for many years.

Cage trim, as we know it today, made a major market entry into the general control valve field in 1963. Now, nearly all manufacturers offer it, and in 1968 sales reached 40 to 60 percent of the intermediate and best quality control valve lines. This rapid acceptance indicates some definite advantages over regular trim, plus increased demands of process control.

The present massive cage guiding of the plug, combined with pressure balance across the plug and uniform distribution of flow around the plug by the cage ports, results in four principal advantages:

1) Stable throttling at any pressure drop.
2) Reduction in side load and friction.
3) Shut-off of large flows at high tem-sure by much smaller actuators.
4) Longer seat joint life.

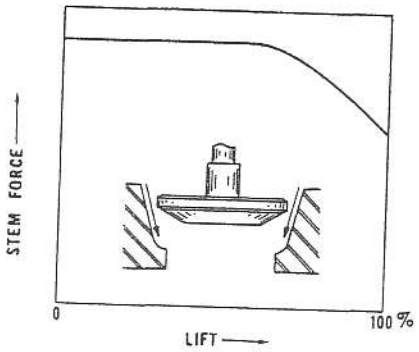
Seat joint life is increased by eliminating chatter at low lifts and by a more uniform distribution of flow through the seat orifice, resulting in less channeling and erosive wear than experienced with standard trim. The characterized cage design reduces erosion by separating the throttling and seating areas, thus removing the seat from the high velocity flow zone.



Some double ported gas regulators have a side entry through the body wall between the seats so that by using a two-piece plug, the ports' areas may be exactly balanced when closed.

Figure 49. STATICALLY BALANCED, DOUBLE SEATED VALVE.

Courtesy Fisher Controls Company



NOTE: CONSTANT STEM FORCE UNTIL 60 % OPEN

This design minimizes fluid separation along the throttling surfaces, which reduces pressure recovery in this region. Cavitation, if any, occurs downstream and valve throttling stability is improved.

Figure 50. SHARP EDGE PLUG IN CONTOURED SEAT RING.

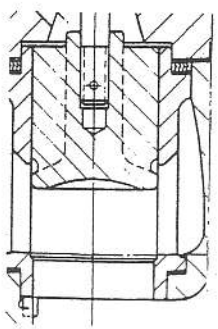
Courtesy W. A. Cash Company

Description and Functioning

The cage is a cylindrical spool piece surrounding the plug. It has a number of flow ports which uniformly distribute flow around the plug and, in most designs, serves as a massive plug guide. It also may carry the seat joint. Cage trim lends itself to a quick-change design. The two most common designs of cage trim are "cage guided" and "cage guided, balanced."

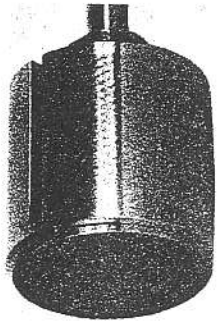
Cage Guided Trim

Cage Guided Trim (without balance), as shown in Figure 51, uses the cage as a massive plug guide, while excellent distribution of flow balances the horizontal side loads on the plug. Contoured cage openings, or a contoured plug establishes the flow characteristics. With a contoured plug the cage flow ports are simply openings which cause little flow restriction. The flow ports have a larger area than the seat orifice where throttling occurs. The plug is grooved along its sides to equalize pressure in the valve body, above the seat, with pressure above the plug, which fits snugly into the cage cylinder. One metal-to-metal seat joint is used for single port valves and this design shuts off drop-tight, if the actuator force is sufficient to overcome the differential pressure load acting on the mean seat joint area. Double-ported valves are also offered with



(a) Plug with Guide Ribs

Courtesy ITT Hammel-Dahl

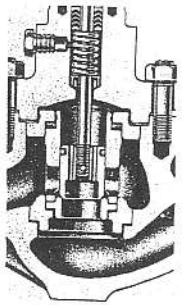


Slots allow pressure above the seat to freely communicate with the enclosed cage cylinder space above the plug.

(b) Solid Plug with Pressure Vent Slot(s)

Courtesy Honeywell, Industrial Division

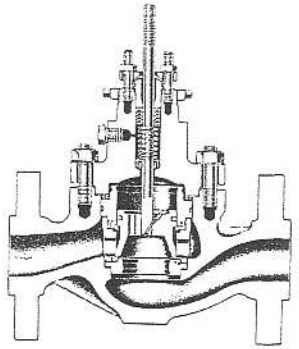
Figure 51. CAGE GUIDED TRIM.



(a) Reduced Port Design

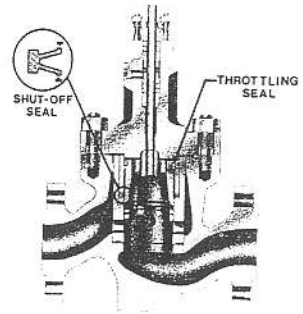
Note pressure equalizing holes in plug to create balance and the seal between plug and cage cylinder for tight shut-off. The reduced port design illustrated uses a smaller cage and plug with transfer pieces to fit in a full port body.

Courtesy Fisher Controls Company



(b) Dual System Sealing
The throttling seal is a piston ring with minor leakage. The shut-off seal is an Elastomer Lip Seal for bubble tight closures.

Courtesy Fisher Controls Company



(c) Metal "K" Seal Balanced Valve Design

The "K" seal is energized when the cage contacts the seat joint in the seat orifice. Throttling seal takes wear. Having the cage supported by the body step allows differential thermal expansion between body and cage with operation from -320° to 1050°F.

Courtesy Masoneilan International, Inc.

Figure 52. CAGE GUIDED BALANCED TRIM.

two cage guides. Leakage is 0.1% C_v maximum for them.

The primary advantage of cage guided over regular trim is the improvement in flow stability. It may be used for high pressure drops when the actuator has sufficient power and is dampened by a hydraulic snubber.

Cage Guided Balanced Trim

Cage Guided, Balanced Trim, as shown in Figure 52, achieves pressure balance across the plug by vent holes passing through it. The plug, depending on trim design, makes either: a piston type seal; a partial sliding seal on the cage wall; or when seated, simultaneously makes a double seal on the regular orifice seat joint and on the upper cage seat joint (used for all-metal construction). In the latter, the plug makes a close sliding fit in the cage when throttling; therefore, the upward force caused by high differential pressures acting under the plug is practically eliminated. For flow over, this force cannot get behind the plug to slam it closed nor act to hold it closed.

The greatly improved throttling stability of cage guided, balanced trim, results in a much smaller actuator size than needed for either standard trim or cage guided trim. Tight shut-off is obtained by using elastomer seal rings on the plug piston; otherwise a slight leakage would occur.

Seating And Sealing

Seating and/or sealing of cage guided, balanced trim occurs at two points:

1) *The seat joint between the plug and the seat.* Regular metal-to-metal joints are used and some manufacturers also offer a soft seat on the plug. Joint widths are made very narrow to effect a better pressure balance design between the orifice seat joint diameter and the cage seal or joint diameter.
2) *The seal between cage guide and plug, which may be of three designs:*
   a) Metal-to-metal seat joints, as shown in Figure 53, are used for temperature service above 450° F and may seal to a leakage of 0.1% C_v maximum (in effect, a double seated valve). The cage and the plug should be made of the same materials for uniform expansion and for retention of sealing at the operating temperature.
   b) Elastomer seal rings are used for bubble-tight shut-off at moderate temperatures, usually -20° F to 350° F.
   c) Loose fitting, piston ring type seals of glass or carbon filled TFE may be used to 450° F. Graphite rings are used above 450° F. All of these rings have leakage rates to 0.5% C_v maximum. See following table of seals, usage and ratings.
   3) Dual System
   Loose fitting piston ring (0.5 C_v max. leakage) seals during throttling while metal "K" seal comes into play when the valve is seated. This bubble tight seal functions with flow over the plug only. See Figure 52(b&c).

The primary advantages of cage-guided, balanced trim are greatly improved throttling stability at low lifts and highest pressure drops, tight shut-off (with elastomer seals), and smaller valve actuator requirements.

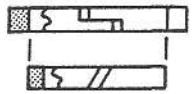
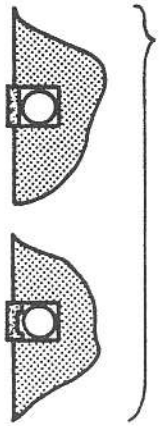
Summary Of Design Features

The following is a summary of the features of the three cage styles used (cage guided and cage guided, balanced as described above, plus erosive service cage trim):

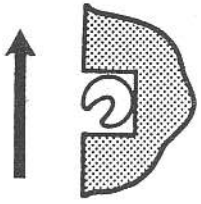
1) *Cage Guided Trim* features both characterized and non characterized (non-restricting) flow ports; massive plug guiding over the entire stroke; and non-pressure balanced.
2) *Cage Guided Balanced Trim* features: characterized flow ports; massive plug guiding over the entire stroke; and pressure balanced across plug.
3) *Erosive Service Cage Trim* features: The cage does not guide plug but allows an annular flow area between the plug and the cage wall. The plug is characterized. The massive guide and plug cylinder sealing spool extends from the top of the cage to the bonnet. Erosive service trim is either non-balanced or balanced across the plug.

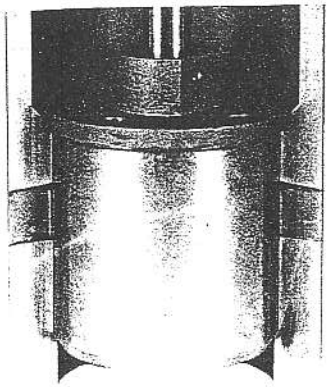
Note: Close fitting cages with flow-under, cause flow jets which will readily penetrate the body wall at low lift, in highly erosive service. On flow-over direction, the plug will be severely eroded in line with the "V" tips of the cage flow ports. At low lifts, flow is concentrated over small areas of the seat joint. Flow must be given room, and guided into turning, to eliminate direct impingement for either flow-over or flow under. This is accomplished in erosive service cage trim design by the annular flow area between the plug and cage and the plug contour, as shown in Figure 54.

PLUG TO CAGE SEALS

DESIGN	MATERIAL	TEMPERATURE RANGE	MINIMUM LEAKAGE
METAL SPLIT RING Lapped Joint 1) Single Spring Ring or 2) Outer Lapped Joint Floating Ring with Inner Spring Ring 	Ni-Resist® Type 3	to 1050° F	0.25% of rated C_v ΔP = Body Rating
METAL "K" SEAL RING 	17-4 PH Silver Plated Inconel Silver Plated	700° F. Above 700° F. Also Cryogenic	.01% of Rated C_v to Drop Right ΔP's to 6000 psi
TFE SLIPPER SEAL Energized by Inner Ring 	Filled TFE Outer Seal Energizing Ring: (1) (Any Elastomer) or (2) 17-4PH Split Spring Ring (Not Shown)	Limited by Choice of Elastomer for Energizing Seal -40° (a) to +450°F (b) a) 17-4PH Limit b) TFE Limit	.01% of Rated C_v 6000 psi ΔP

PLUG TO CAGE SEALS (continued)

DESIGN	MATERIAL	TEMPERATURE RANGE	MINIMUM LEAKAGE
TFE WITH GARTER SPRING Seals Only in One Direction	TFE Spring 300 Series s.s.	-320° to 400° F	Drop Tight 3000 psi △ P
O-RING TFE Back-up Seals Optional	Buna-N 70A Durometer or Other Elastomers	+10° F to 275° F	Bubble Tight 6000 psi △ P
SLOTTED O-RING  Seals Only in One Direction	TFE	-100°F to +450° F but limited to 100°F △T in order to Remain Tight on Start-up	Drop Tight 10,000 psi △ P



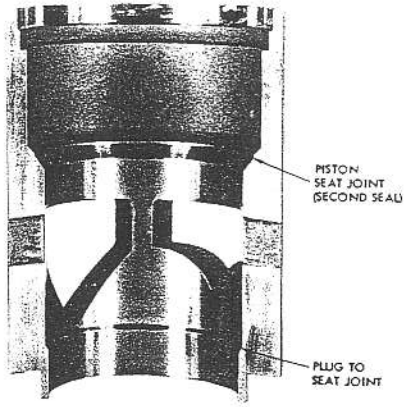


Figure 53. DOUBLE SEAT JOINT CAGE, Metal-to-Metal Sealing.
Courtesy Honeywell, Industrial Division

Note narrow seat joints, characterized cage ports, and degree of plug guiding. Pressure equalizing vent holes are visible in top of plug.

High Temperature Cage Design

Differential thermal expansion between the valve's body and cage must be considered for valve service above 450°F. Even when materials are selected with nearly the same coefficient of expansion, the cage is always at a higher temperature because it is the internal component and is surrounded by flowing fluid on three sides. The body can lose heat through its wall even though it is insulated. Only for a moderate temperature change, ±200° F, may the cage be rigidly compressed by both the bonnet and seat bore step in the body bridgewall. For temperatures as high as 600°F there must be a few thousandths of an inch freedom for expansion and in this case live gasket compressive forces would hold the cage rigidly.

For temperatures above 600°F some designs hang the cage near the top of the body bonnet flange, thus allowing the lower end to be free to expand. Again, a high gasket stress and live action by the gasket maintains the seat anchored in the bridgewall. Other designs restrain the cage with a heavy Belleville spring. See Figure 64. Failure to observe these precautions have resulted in cage fracture, loose seal rings and approaching bonnet joint gasket blow-out.

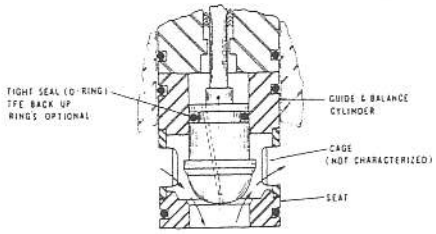


Figure 54. EROSIVE SERVICE CAGE TRIM (Characterized Plug Standard).

Cage Guided, Balanced Trim Simplifies Valve Application

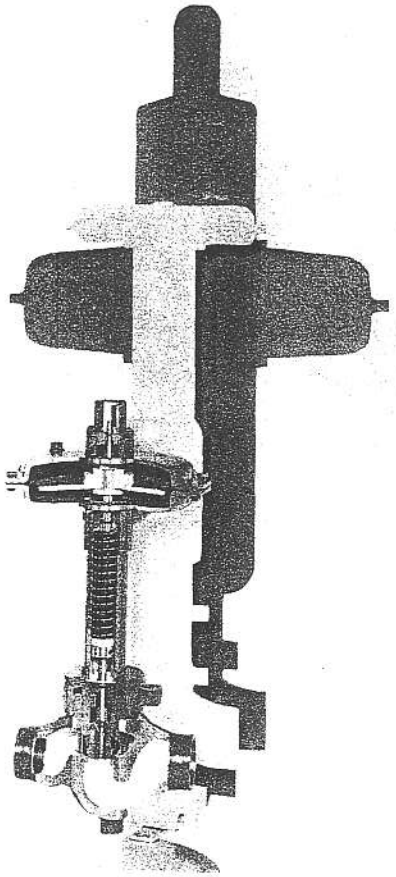
For comparative purposes, a typical non-balanced valve application chart is given below for a 2 in., 3000 psi WOG valve (body rating) with a 1 in. seat port and a *medium* size diaphragm actuator.

Pressure Drop Opening and Closing Limits

Air To Close		Air To Open	
Flow Over	Flow Under	Flow Over	Flow Under
1400 psi	2200 psi	1100 psi	130 psi

The application problem would be apparent if one were required to: 1) enlarge this chart by all port sizes 1/4 in. to 2 in.; 2) make new charts for each actuator size (usually seven); and 3) add the various springs and settings to give all the variations, without exceeding valve stem, load limits.

In contrast, a cage balanced application chart for the same valve with a *small* actuator be as follows: The pressure drop limit equals 3000 psi body rating for all conditions: flow under, flow over, air to open, and air to close. One actuator size and spring covers the entire range. (See Figure 55.)



Typical dimensions and weights of the valves of Figure 55 areas follows:

Sectionalized Valve	erosive service, pressure balanced cage trim (close coupled actuator, oil field design)
	weight 72 lb., height 24 in.
1st silhouette	cage guided, balanced trim (standard open yoke actuator)
	weight 105 lb., height 36 in.
2nd silhouette	standard, non-balanced valve (open yoke with giant actuator)
	weight 250 lb., height 48 in.

Figure 55. RELATIVE SIZE OF BALANCED & NON BALANCED VALVES HAVING EQUAL C_V & ΔP OPERATING PERFORMANCE (2 in., 3000 psi WOG; flow-under; C_V 55 maximum; spring closed; 35 psi actuator air supply.).

Plug Balance

The following is an example of balancing the axial forces acting on a valve plug:

Valve & Service Conditions
2 in. valve, 1000 psi pressure drop service, flow under plug.

Physical Dimensions of Trim
2 in. seat orifice diameter, area - 3.14 in.^{2}.
3/8 in. stem diameter, area - 0.11 in.^{2}.
Seat joint width - 0.030 in./side, normal to stem axis.
Mean seat joint diameter - 2.030 in.
Mean seat joint area - 3.23 in.^{2}.

Unbalanced Design Forces
Pressure drop forces for opening: 1000 psi x 3.23 = 3,230 lbs.
An air-to-close actuator would have to overcome this force plus the thrust for seating, packing friction, and spring compression.

Balanced Design Forces
Upward forces to be balanced - 3,230 lbs.
Set annular area between plug stem and cylinder to counterbalance this force +2% for downward seating thrust as the pressure drop rises:

3230 lb. + 2% of 3230 lb. = 3295 lb.

Area required:

3295 lb. ÷ 1000 lb. in.^{2} = 3.295 in.^{2}

Equivalent Diameter of Balance Cylinder Above Disc

Required Area	3.30 in.^{2}
Stem Area	0.11 in.^{2}
Cylinder Area	3.41 in.^{2}
Cylinder Diameter	2.09 in.

Note: The original unbalanced force of 3230 lbs. has been reduced to a 2% downward force of only 64 lbs.

In the open position the stem thrust from line pressure amounts to 1000 psi x 0.11 in.^{2} = 110 lbs.

It has been shown, therefore, that a force, ranging from 64 lb. (downward) to 110 lb. (upward) gives a close balance to all operating conditions. A design showing this type of plug and stem balance is illustrated in Figure 56.

Cage to Body Seals are essential in designs which shut off drop and bubble-tight. This should be a double acting seal to handle the valve pressure drop in either direction.

Guiding in cage trim valves is ideal from a design standpoint. Massive plug guiding of sufficient axial length and diameter to prevent cocking or shifting from side thrust is carried right down to the seat joint to give a near-perfect alignment. The heavy guide distributes the impact load from vibration over a greater metal area and reduces galling and guide wear. Only two parts are involved, the plug and cage. This is compared with the six parts of conventional valves: seat ring, body, bonnet, stem guide bushing, stem, and plug.

The cylinder guide for erosive service trim does not extend to the seat joint, due to the short cage length, but is located a short distance above it. Guiding is excellent. These designs are more prominent in oil field type valves.

Vent Holes in the plug allow pressure balance and, therefore, must be kept open for good valve performance. These holes should have a flow capacity that is higher than the lengthy annular clearance between the plug and the cage guide to maintain balance in the event of seal leakage, failure, or omission of the seal in field maintenance reassembly.

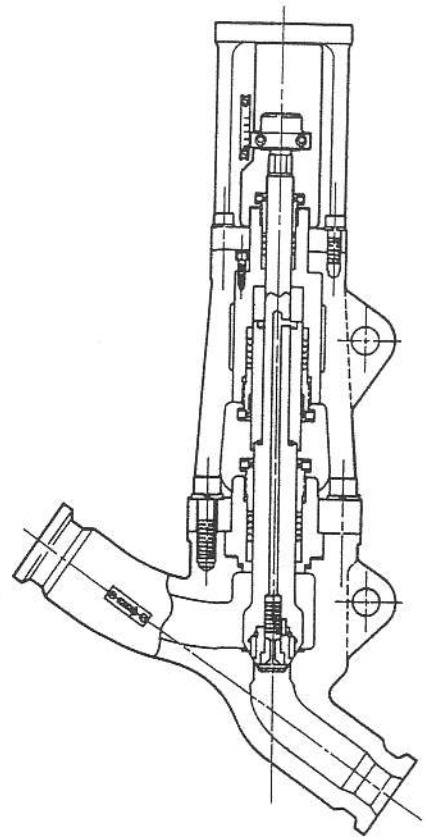


Figure 56. PLUG & STEM BALANCE DESIGN.

Thrust on the underside of the mean plug-to-seat joint area is counter-balanced by the by-piston attached to the yoke. The annular area of this piston equals the seat joint area, thus balancing both the plug and the stem thrust. The stem diameter passing through the bonnet packing equals mean plug-to-seat diameter, thereby balancing the plug to the pressure in the body cavity above the seat. When the valve is opened pressure on the underside of the plug is balanced by the yoke annular cylinder piston. Three sets of packing sense pressure drop in the line relative to atmospheric pressure.

Courtesy AMETEK, Inc., Calmec Division

Loss of Seal will not unbalance the plug. In one proof test a valve was closed by applying air pressure to the actuator until its plug touched the seat, but it had no seating load. A dial micrometer was attached to the stem and a differential pressure applied across the seat, which in an unbalanced design would have been sufficient to cause the plug to lift off the

seat by 1/4 in. In this case, with the balance seal removed, the plug remained closed and only plug cylinder clearance leakage occured. No upward plug motion could be detected.

The cylindrical annular balancing space above the plug is a relatively dead area as far as flow is concerned. Flushing occurs only on valve stroking. The space is self-cleaning when the valve is in a vertical position. For entrained solids service, horizontal mounting may result in jamming, during the last 10 percent of stroke and upside-down mounting is not permitted.

Service for entrained solids are no problem. A two inch valve handling sandy water at the rate of 3600 lb. of sand mixture per hour, while stroking and standing, could not be jammed. It retained full stroke in the vertical position.

Paraffinic oils do not clog vent holes. Valves with long vent holes have been used in oil production service where the flow lines were completely blocked by paraffin and sand, yet the valves remained operable.

Special design for clogging service has included some plugs designed as cylindrical skirts with completely open venting, except for a spider at the top end for stem attachment, as shown in Figure 57.

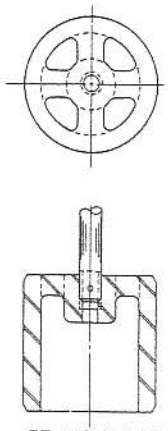


Figure 57. NON-CLOGGING, BALANCED PLUG.

Flow Capacity

Cage trim flow capacity is generally 15 to 20 percent higher than the flow capacity of regular trim. When bodies are designed specifically for cage trim, the capacities are even higher. The uniform distribution of flow around the plug by cage, reduces turbulence and offsets the fluid contraction and expansion pressure losses when passing through the cage ports.

The following example is a comparison showing the increased flow in a body designed especially for cage trim (Typical C_V values are given.).

2 in. Globe Valve (flow under)	*Plug Characteristic* Quick Opening	Linear	Equal Percentage
1) Characterized Cage 2-5/16 in. diameter orifice (non-restricting)	75 C_V	74 C_V	65 C_V
2) Standard plug and seat construction in regular body	60 C_V	55 C_V	55 C_V
Increase in flow capacity	25%	35%	19%

Greater Valve Rangeability is obtained with Cage Guided, Balanced Design. Stable throttling near the seat allows the valve to control at a degree which matches the precision of the actuator. Thus, the reliable low range of flow control is increased.

Reduced Capacity Trim change-out for dual metal seated cage designs is effected by changing only the characterized flow cage of the balanced design. A smaller set of characterized flow ports are used in the replacement cage. The number of cage ports may be reduced instead of altering their size, but at a sacrifice of balancing side thrust forces. Some manufacturers use a smaller plug, cage, seat ring and supporting members. Cage trim with characterized plugs and non-restricting cage flow ports requires a change of both plug and seat as in standard valve construction.

Quick Change Trim is found in many cage designs. The bonnet holds down the cage and the cage may hold down, or be integral with, the seat orifice. Thus, all parts may be removed through the bonnet. This applies to three-way valves as well. Screwed or welded cages are not quick change, but may be used to improve guiding for plugs having very large stems.

Cage design is so effective in stabilizing flow control that cage rings are available to screw into the body seat ring threads of both single and double ported valves.

Double Ported Cage Designs, as shown in Figure 58, are frequently used on high pressure gas and steam reducing service and also on boiler feedwater control. Their flow capacity is about 5% higher than regular trim.

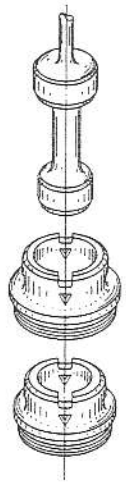


Figure 58. DOUBLE PORTED, CAGE TRIM, Screwed Type.

Courtesy Fisher Controls Company

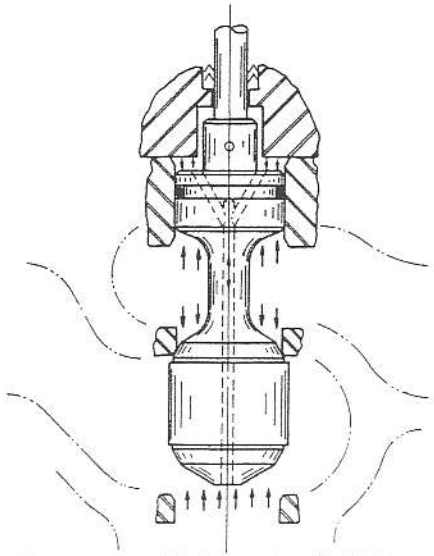


Figure 59. PRESSURE BALANCED THREE-WAY DIVERTING AND MIXING VALVE. (2 in.; 3000 psi WOG; C_V = 55).

Pressure in the upper port is spool-balanced, while forces under the plug are equalized by the central vent hole. The stem is not balanced. Metal-to-metal seating allows tight shut-off to full 3000 psi change in pressure-in across any port.

Three-Way Valve Trim Balance

Three-way diverting and mixing valves are notoriously poor performers in high pressure shut-off or throttling service and their actuators are massive. As shown in Figure 59, balance overcomes these deficiencies for both mixing and diverting service.

By balancing, two inch valves with 1-3/4 in. metal seat joints may shut-off a full 3000 psi pressure drop across either port, with flow in any direction, and may be operated by a 50 in.^{2} diaphragm actuator that has a 35 psi air supply. Shut off with narrow joint, metal seating is drop tight (O-Ring seal used on plug balance cylinder seal).

These principles may be extended to special designs which, in addition to diverting and mixing, shut off both ports at once. These special designs, called "3-way, 3-position" valves are illustrated by Figures 60, 61, 62. Two-inch, 5000 psi, WOG valves have successfully shut off and diverted oil well flow at a well head with shut-in to 4500 psi tubing pressure. Hi-low pressure pilots also allow them to serve as safety valves.

Ordinary three-way diverting valves as shown in Figure 60, simply direct flow from one outlet to the other. In the transition period of changing outlets, fluid flows from both outlets at once. This type of valve has a full "off" position and rapidly passes through "off" in diverting the stream from process to test. One oulet must fully close before the other can be opened. Plugs float on the stem and align themselves independently in their respective seats for tight sealing. The "off" position allows one valve to perform the job formerly requiring two valves, (one for diverting and one for shut off). The stem slides through the top plug (spring seated) as the nut on the bottom of the stem lifts the bottom plug. The stem shoulder, pushing down, opens the top plug while the spring seats the bottom plug.

The valve illustrated in Figure 61 is capable of opening and closing to the differential pressure across any flow port or to a combination of differentials on all ports. Ports P_C and P_T are spool balanced while part P_I is balanced by a floating sleeve, surrounding the spring and sealed to the plugs. The stem senses no pressure. Four dynamic (moving) seals are subject to the difference between the internal and atmospheric pressures; therefore, practical high pressure designs are limited to 2 in., 3000 psi WOG, or 4 in., 1500 psi WOG. Special designs involving dual concentric stems and double diaphragms within a single case may be

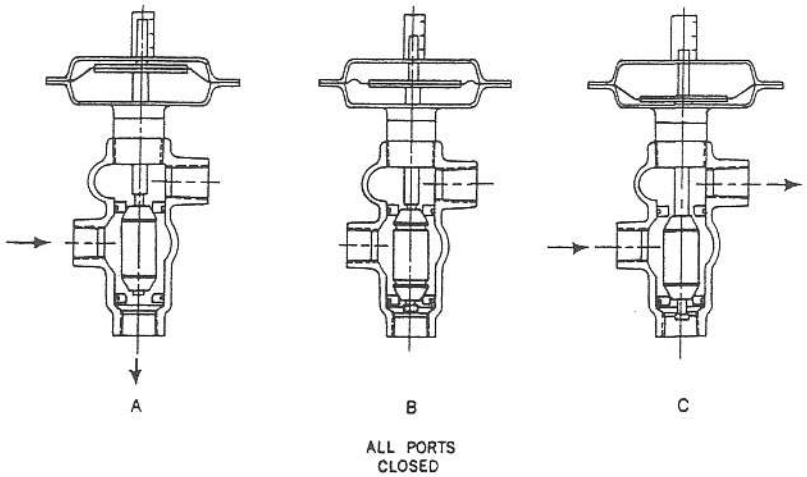


Figure 60. ORDINARY THREE WAY, THREE POSITION VALVES.

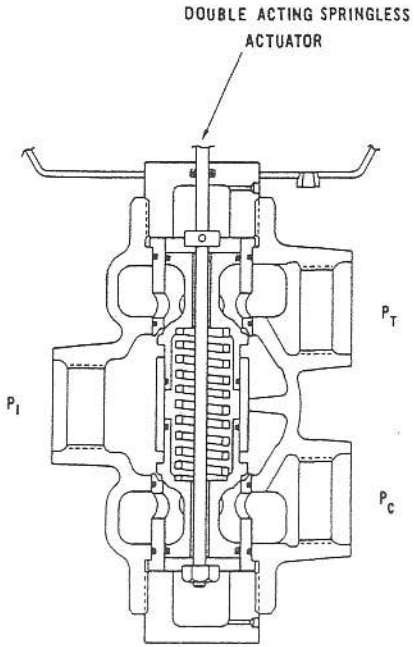


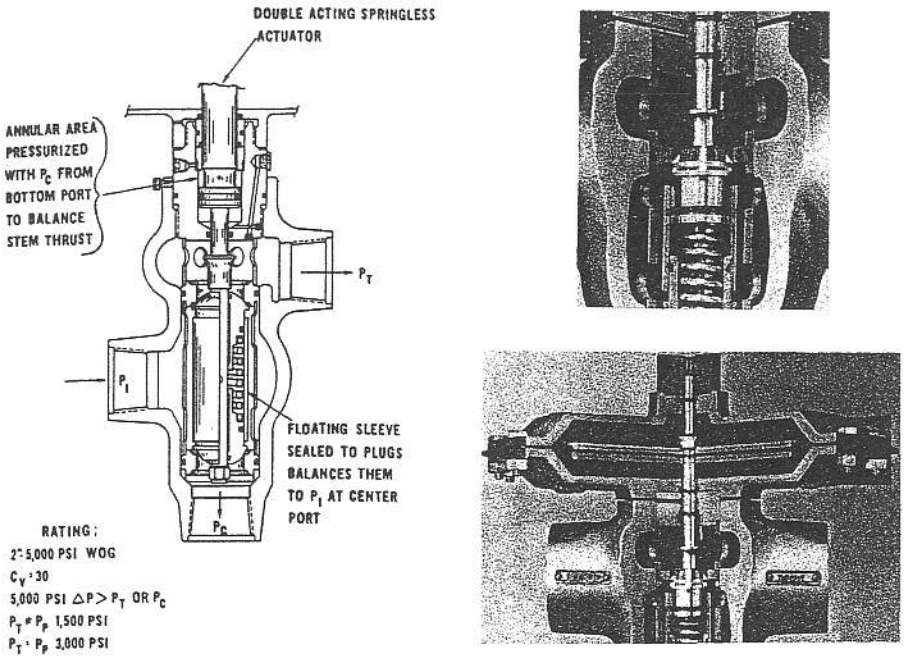
Figure 61. FULLY PRESSURE BALANCED, THREE-WAY, THREE-POSITION VALVE.

U.S. Patent No. 3,063,469.

Courtesy Rockwell Manufacturing Company

air-loaded from two sources to maintain plugs seated on the sleeve with spring compressed. This allows diverting without passing through the off position and eliminates the momentary pressure surge when changing ports. Fail-safe shut-off of all ports occurs upon relief of both loading pressures.

Application of the semi-balanced design, shown in Figure 62(a), is high-pressure, oil-production, battery manifold service at 2 in., 1500 to 5000 psi WOG. Outlets P_T and P_C connect with the test and common separator vessel manifolds, both normally operating at the same pressure. P_I is the incoming well flow line. P_T and P_C are balanced by venting P_C to the sleeve, to get behind, and tend to seat both plugs (this can be reversed to P_T, or in clean service, a shuttle valve in the stem may sense and feed-in the higher pressure). Stem balance is achieved by venting P_C to the annular shoulder on the upper stem. Three small stem seals sense internal pressure relative to atmospheric pressure. P_I will open to shut-in flow line pressures, equal to or exceeding the body pressure rating with normal actuator supply pressure. P_C will remain sealed until $(P_C - P_I) > 3000$ psi (based on spring force and slight degree of unbalance designed into the P_I side of the plug which causes the seating force to increase slightly as the shut-in flow line pressure builds up). P_T will remain closed when: $P_T = P_C$, $(P_T - P_I) > 3000$ psi; or when $P_C = 0$, $(P_T - P_I) > 1500$ psi.



(a). "Semi-Balanced "Three-Way, Three-Position Valve.

(b). "Compact", Three-Way, Three-Position Valve.

U.S. Patent Nos. 3,025,881, 2,969,811, 2,867,239

Figure 62. SEMI-BALANCED AND COMPACT, THREE-WAY, THREE-POSITION VALVES.

Courtesy Rockwell Manufacturing Company

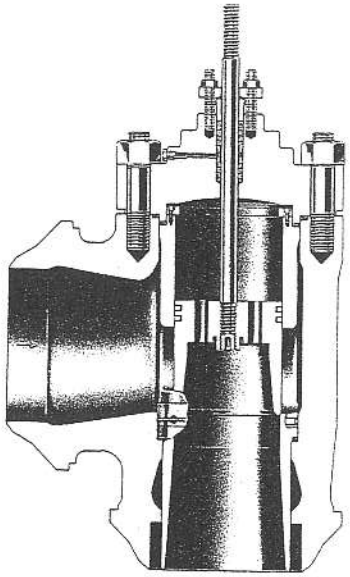
Figure 62(b) shows a compact three-way, three-position valve that is semi-balanced (2 in.; 1500 psi WOG, Cv-20). It has balanced trim as shown in the upper illustration, and a close-coupled, double-acting springless actuator as shown in the lower view.

Large Designs And Applications

Large cage guided, balanced valve designs and applications are shown in Figures 63, 64, 65, 66, and 67. Figure 63 shows the internal and external views of a large balanced valve for boiler feed water control.

Figure 64 is a schematic of an isolation and dump valve for high pressure and large flows. It has a flow-over angle design with a separate pressure balance pilot. With the valve closed, pressure leaks past the plug, piston-to-body sleeve clearance and aids in seating the main valve. The actuator must thrust the stem downward to compress the pilot valve spring and close the pilot. Sealing is tight. To open, the actuator lifts the stem allowing the pilot to open and balance the main plug, since the pilot port has a much greater capacity than the annular plug-to-sleeve clearance area. The pilot plug shoulders against a stop in the main plug and the remaining spring-force causes these parts to remain together for the balance of the opening stroke. Unbalance, created by the plug cylinder being smaller than the seat joint, is downward. Since the main plug and pilot valve stem assembly are free to move except as loaded by the spring, the plug and stem remain rigid during opening and there is no chatter. The opening force is reduced by balancing the main valve.

In the balanced globe valve shown in Figure 65, a cage hole drains condensate out of the upper compartment, above the plug, when the valve is closed with flow-over. Thus, with the main valve and pilot valve closed, upstream pressure enters above the plug and aids seating. Seating



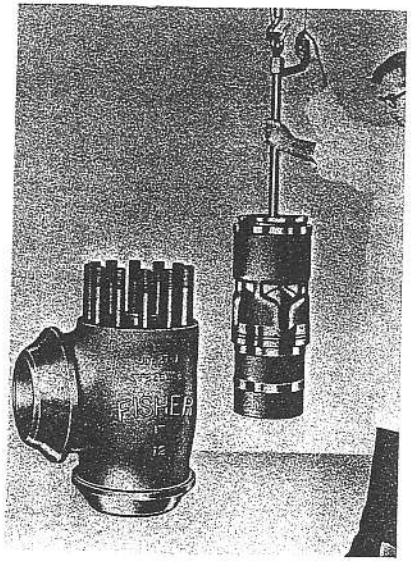


Figure 63. BALANCED VALVE FOR BOILER FEED WATER CONTROL, (12-16 in. USASI 900 Class, 2160 psi CWP).

Courtesy Fisher Controls Company

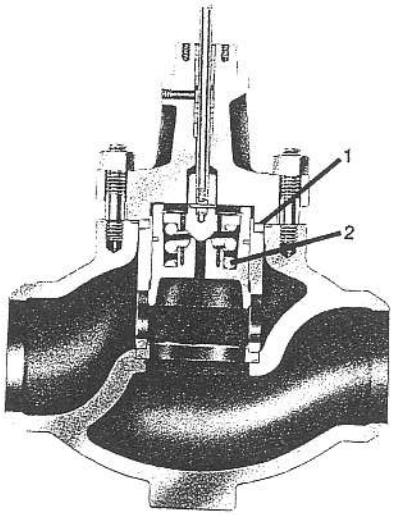


Figure 64. STEAM DUMP AND ISOLATION VALVE.

This is a balanced design with heavy duty pilot spring (2) which makes it insensitive to operating friction. Elastic loading spring (1) maintains cage loaded against seat and seat gasket during thermal cyclic operation.

Courtesy Masoneilan International, Inc.

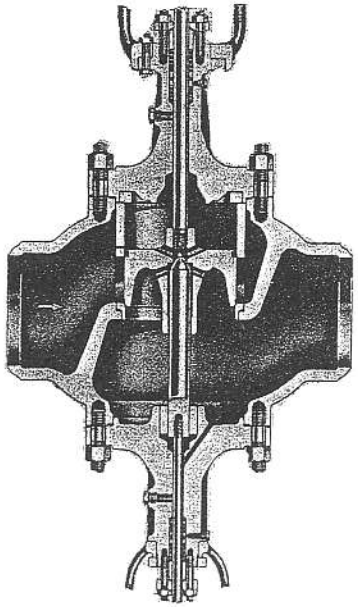


FIGURE 65. BALANCED GLOBE VALVE FOR NUCLEAR STEAM DUMP OR TURBINE BY-PASS.

Courtesy Fisher Controls Company

is tight. To open, the pilot is opened first by its actuator, thereby establishing balance across the main plug, which in turn is opened by its own actuator. Opening speed may be as fast as 350 milliseconds for a 4 in. stroke at 1000 psi pressure drops.

The sixteen-inch, balanced disc valve shown in Figures 66 and 67 is balanced to open with flow-over and, unbalanced when closed, so that the pressure drop adds to the seating force. This is accomplished by an internal pilot valve, operated by the main stem. A double-acting air piston actuator, plus a fail-safe spring closure gives rapid operation, which is regulated by a hydraulic snubber surrounding the stem inside of the nest of springs.

Summary Of Cage Balance Trim Advantages

Flow Control

1) Greatly improved throttling stability at high pressure drop.
2) Ability to close and open against any pressure-to-body rating.
3) Tighter seating.
4) Greater maximum flow capacity, coupled with high pressure shut-off.
5) Increased rangeability for controlling low flows.
6) Fail-safe action.

Physical Improvements

1) No seat damage, since chattering impact is eliminated.
2) Less guide damage and noise from horizontal vibration in the plug guide.
3) Long seat life, because of reduced erosion.
4) Greatly increased packing life, because of reduced stem loads and smooth stem motion. Stems run truer with less bowing and cocking.

Actuator

1) Smaller.
2) Lighter.
3) Less expensive.
4) Faster response and better control, because of:
   a) Less dead air space;
   b) Flatter area vs. stroke curve; and
   c) Lower mass.
5) Higher spring rate, stiffer actuator, more stable control.
6) Much more actuator force may be applied to seating force. More of the actuator force may be applied to compress the spring, since very little is needed to overcome the pressure drop across the plug.
7) It is practical to spring-close large valves to high pressure drops.

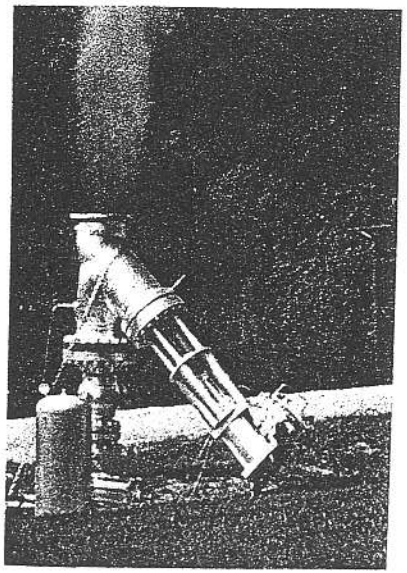


Figure 66. SIXTEEN INCH, BALANCED DISC VALVE, DUMP TEST AT 1500 psi PRESSURE DROP
(Designed for 1250 psi at 575°F Service).

Courtesy Rockwell Manufacturing Company

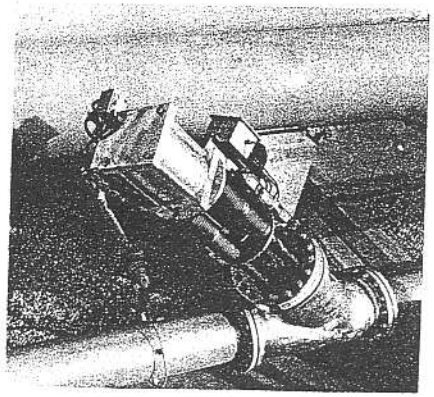


Figure 67. SIXTEEN INCH, BALANCED DISC VALVE IN A LIQUID FLOW TEST LOOP.

Courtesy Rockwell Manufacturing Company

Application

1) Simpler to apply, because of fewer restrictions, such as flow direction, air supply, orifice size, pressure drop, etc.
2) Fewer requirements for positioners.
3) A hydraulic snubber is never required.
4) Quick change trim.

SEATING, SEALING AND LEAKAGE

The three problems discussed in this section include:

Seating - The alignment and contact of the plug with the seat, including joint design and loading.

Sealing - The parameters of metal finish, joint width, and metal yielding which lead to tight sealing.

Leakage - The amount of leakage that may be expected for different sealing parameters and joint designs.

Tight sealing is becoming of greater importance to control valve users, now that improvements in designs of both valve trim and actuators allow tight shut-off. One valve may be used for both stop and throttling service at a cost saving. Diaphragm control, valve actuators with 15 to 35 psi air supplies do not develop the high seating forces used in stop valve designs with manual or automatic operation. If they did, they would be bulky and would be sluggish in their speed of stem movement for throttling action. A small amount of leakage is to be expected, because of the lower seating forces. Manufacturers normally rate their valves for maximum leakage as follows:

Double Seated Valves –
<0.1% C_V maximum[1]

Single Seated Valves –
<0.01% C_V maximum[1]

Development of the springless piston actuator, air loaded on both sides and using a much higher supply pressure (100-150 psi), coupled with a positioner, led to higher seating forces and tighter shut-off. Single seated valves can now be sealed, drop-tight to high pressure drops. Some cage balanced valve designs also shut off drop-tight with small diaphragm actuators.

Seating

To make a seal, a plug must first be perfectly aligned and then must fully contact the seat joint as the seating load is applied by the actuator.

If the plug strikes the seat joint slightly cocked, it will remain cocked until a higher seating load causes it to jump sideways as it slides down into the taper and slams into full joint contact. This deforms the seat, causing leakage. In time, the plug indenture will extend to form a new off center, nearly 360° seat contact. Above a certain plug cocking angle, the plug will not jump into place regardless of loading; therefore, pre-guiding and alignment of the plug before seating, is necessary.

The two principal types of mis-alignment are *Concentric* as shown in Figure 68(a) and *Axial* as shown in Figure 68(b).

Alignment

Alignment of the plug on the seat for a single ported, top and bottom guided valve involves concentric alignment of eight components and axial alignment of eleven, fit combinations; plus consideration of the operating clearances.

One can readily see the precision machining, required of the control valve manufacturer to maintain alignment of these parts. Each part must have an assembly or sliding clearance which allows a minute horizontal axial shift and a very slight cock. The flexibility of the stem is sufficient to allow the plug to move into true seat joint contact with light, initial seat-contact loading.

Seat Joint Design

Flat joints, normal to the plug axis, are used on some low pressure stop valves and soft seated control valves. It is not practical to manufacture them for high pressure service. As discussed later, tighter sealing occurs through a sliding and burnishing action of a tapered joint. Tapered joints turn the fluid gradually and are the best for high pressure drop and for erosive service. The control valve industry uses joints from 15° to 45°. Smaller angles would begin to form sticking tapers and larger angles give too much of a poppet effect, when cracked open at high pressure differentials, which would cause unstable, low-range throttling. In summary:

1) 45° seat angles find their best applications in either normally open or normally closed valves.
2) 30° seat angles are a compromise between high seating forces and streamlined flow, at low lifts, for low erosion service.

[1] Seat leakage flow is laminar rather than turbulent and the C_V formula is not applicable; therefore, this is simply a means of specifying an amount of acceptable leakage.

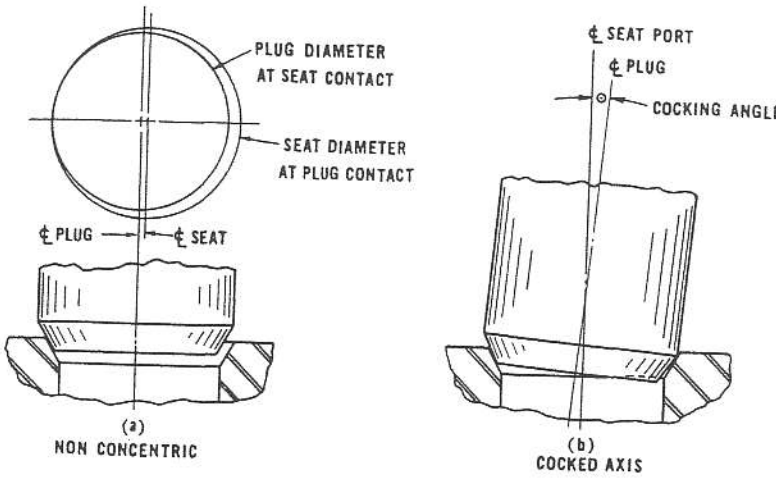


Figure 68. PLUG MISALIGNMENT WITH SEAT.

3) 15° seat angles are best applied to high pressure drop, erosive service.

Seat Joint Width

Seat joint width is a balance of adding length to the flow path to increase flow resistance vs. reduction of the seating force for a given actuator seating load. A certain minimum width is essential to establish a tight seal; however, the joint must have sufficient backup strength to support the compressive seating load and must be wide enough to prevent indention of the plug. Narrow joints are much tighter than wide joints, provided they exceed the minimum width requirements.

Seat Joint Loading

Seat joint loading is usually expressed as pounds of force per linear inch of mean seat joint circumference. Loading may vary from 25-600 lb./inch as given below:

1) 25 lb./in. - Low pressure drop service; leak-tight shut-off is not required; metal-to-metal joint.
2) 50 lb./in. - Moderate pressure drop service; slight leakage expected (0.1% C_V maximum).
3) 100 lb./in. - High pressure drop service; nearly drop-tight service (0.01% C_V maximum); will seal 3000 psi pressure drop on 0.015 in. width, 30° joint of 316 SS[1].
4) 300 lb./in. - Very high pressure drop service; drop-tight (will seal 6000 psi on 0.025 in. width, 20° joint of 440-C SS, hardness 55 R_C).
5) 600 lb./in. - Extremely high pressure service.

The apparent compressive seating stresses on joints described in items 3) and 4) above, are 13,000 psi and 35,000 psi respectively, which is well below the yield point of the given trim materials. These are the stresses normal to the joint. Elastomeric and plastic yielding is occuring at the high points of each surface making a tortuous flow path for leakage restriction.

By contrast, stop-valve seat loading with hard-faced seats in steam service may be 50,000 psi, apparent stress, or four times a nominal control valve, seating load of 100 lb/linear inch for a diaphragm actuator.

To obtain the best circular seat-joint contact at low stem loading use:

1) Cage guided trim with the seat integral with the cage. Horizontal and axial alignment for seating involves

[1] 100 lbs/linear inch vertical seating force is equivalent to a force of 200 lbs/linear inch normal to a 30° joint (100 lb./sin 30°). The compressive seating stress is 200 lb/0.015 in.x 1 in = 13,300 psi.

only two parts in sliding contact. Alignment of regular trim must be transferred from the seat to the body, to the bonnet, to the guide, to the plug, and back to the seat.

2) Cage guided trim with the seat aligned by the cage.
3) A seat that is integral with body; or
4) Extend a thin flexible seat lip above the means of seat retention as shown in Figure 29.

A radial expandable, seat ring design is illustrated by Figure 69. In this design the seating angle creates a large radial component of force which expands the seat ring against the retaining collar. The spring-out action of the non-circular ring allows a near-perfect joint contact giving drop-tight shut-off in severe thermal cycle service. This design has been successfully used on pressure drops to 4,400 psi. The large flow, entrant passage also makes the valve suitable for erosive service.

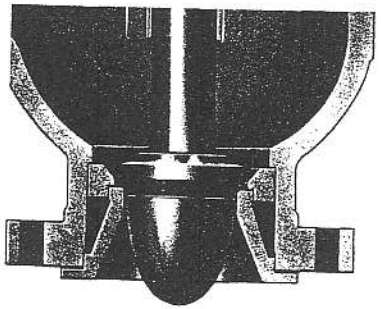


Figure 69. RADIALLY EXPANDABLE SEAT RING DESIGN.

Courtesy Conoflow Corporation

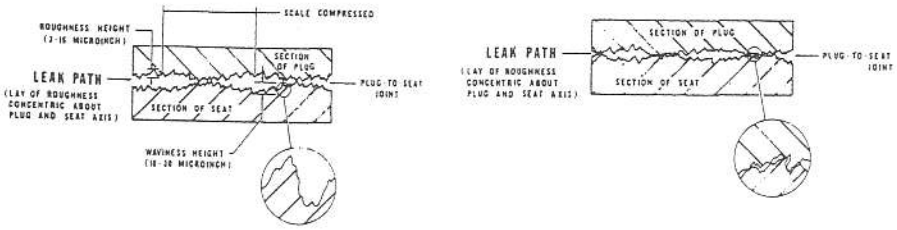
Trim Sealing

Tight sealing requires the yielding of one material into the surface "waviness" and surface roughness of the other, as illustrated by Figure 70, to block direct leakage paths; thus, making these paths long and tortuous. Compressive stresses are far below those that would yield the entire joint; therefore, the contact area is apparent. Actually, only the peaks of each surface are in contact and the concentration of force may then exceed the yield and will plastically deform the high spots on each valve closure. Additional closures require a higher seating load to achieve the same degree of tightness, until the wear particles are formed and conditions tend to stabilize.

Tapered joints provide for a sliding and burnishing action as contact is made and loading occurs. This gives a tighter initial seal than a perpendicular contact and the seal remains tighter with repeated closures.

The minimum width of a joint to seal gas to 1 x 10^{-7} cc/s/linear inch, maximum leakage, is 0.04 inch. Wider joints with the same surface finish and loading will not seal tighter. This width insures sufficient high point contact to form an adequate flow resistance path as shown by the graph of Figure 71.

Extra "super-finishing" of seat joints is unnecessary for tight sealing, because as the joint opens and closes, wear particles ball up on the surface, quickly returning it to a rougher finish. Also, some fluid contaminants tend to remain in the joint on closure and indent the surfaces. Excessive lapping generally either reduces seat tightness by increasing the actual contact area, thus, reducing the unit compressive seating stress provided by a fixed actuator seating force or it destroys the original surface geometry.



(a) Under Light Seating Load

(b) Under Heavy Seating Load

Figure 70. MATING OF SEAT JOINT SURFACES.

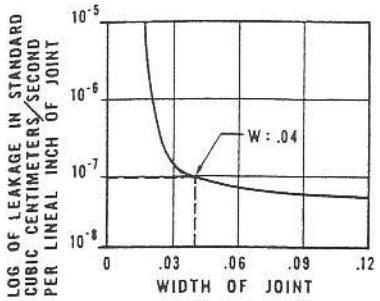


Figure 71. MINIMUM JOINT WIDTH FOR A TIGHT SEAL.

(Leakage rates for 14.7 psi Δ P helium on a flat circular joint.)

Consolidated graph taken from Reference No. 48.

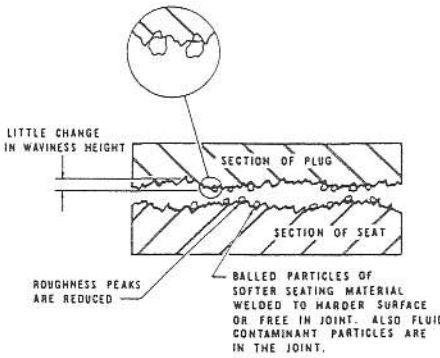


Figure 72. JOINT SURFACE AFTER REPEATED CLOSURES.

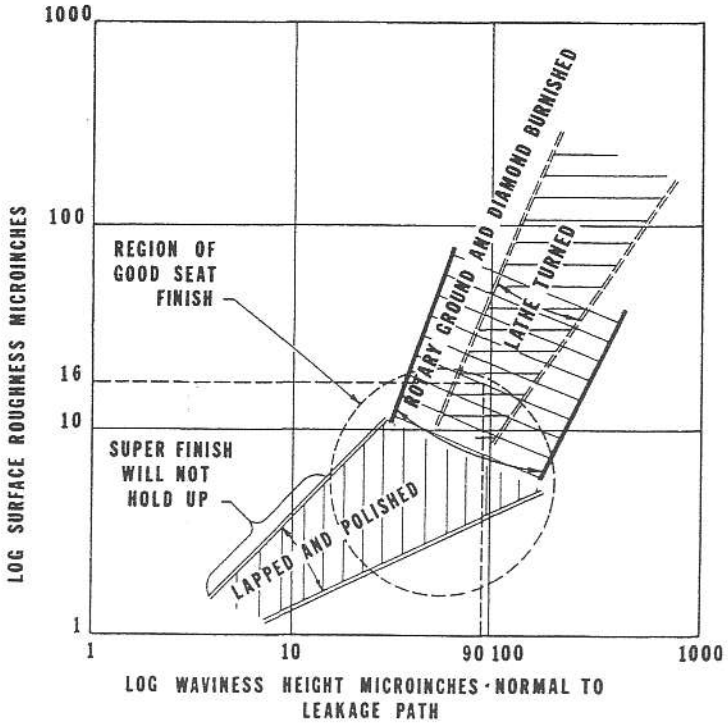


Figure 73. DEGREE OF SEAT JOINT FINISH VS. METHOD OBTAINING FINISH.

Chart presented in "Investigation of Leakage and Sealing Parameters", Paul Bauer, ITT Research Institute, Technical Report AFRPL-TF-65-153, August, 1965.

The following comparison illustrates the relative seating force to obtain same degree of tightness vs. the seat joint finishing method (concentric lay of finishing pattern). Refer to Figures 72 and 73.

Finishing Method	*Relative Seating Force*
diamond burnished (after grinding)	1.0
lathe turned	1.3
ground	2.5
lapped (excessive) (90% of apparent area of) (contact actually mated)	2.9

Trim Leakage

Definitions Relative to Trim Leakage:

Drop tight, to be meaningful, must be specified in terms of the maximum allowed number of drops per unit of time (drops/minute, cc/hour or no visible drop).

Bubble tight, to gas, should be specified as the maximum allowed number of bubbles of a given size per minute (usually 1/8 in. diameter).

Zero leakage is defined as 1×10^{-8} cc/s or about 0.3 cc/year (helium leak rate at standard conditions). Zero leakage is often specified in critical service and requires very careful joint design, material selection, finish control, and sufficient seating force on narrow joints. It is practical only for small valves, at extra cost, and may last for only a few closing cycles.

Stop valve maximum leakage rates are given in the Valve Manufacturers Standardization Society, SP-61 as:

1) Water tests at ΔP = Body CWP rating (10 cc/hour/inch of valve pipe size or about 3/drops/minute).
2) Air tests at $\Delta P = 80$ psi (0.1 SCFM/inch of valve pipe size).

Leakage Specifications, to be meaningful and allow comparison, should include test fluid, temperature, pressure, pressure drop, seating force and duration of test.

Control Valve Leakage

Properly designed control valves can achieve stop valve tightness and maintain it throughout a long service life before trim replacement; particularly with cage guided, balanced trim having elastomer plug-to-cage seals. The control valve, however, is expected to throttle and often shuts off much more frequently than stop valves. For example, some dump valves may have from 4000 to 7000 opening and closing cycles per day, handling high pressure and erosive fluids at 1000 to 4000 psi pressure drop. Few stop valves could match this performance and remain tight.

It is customary to expect a slight degree of leakage and, in general control valve service applications, this has presented few problems. Specified permissible leakage should be based on process considerations (hazards resulting from leakage after emergency shut off, etc.). Tight shut off should be specified only when necessary, because a higher quality more expensive valve is required.

Liquid Leakage is to a great extent affected by surface tension and the particular fluid wetting the joint surface, before the new fluid attempts to enter. For precise leak testing, the joint and body should be free of all traces of oil, which would preferentially wet the metal joint surfaces. Water leakage rates of an oil-free joint are higher. Oil clinging to the roughness, blocks some leakage paths, causing a higher capillary displacement pressure to establish flow of the test fluid.

Seat Leakage Specifications

Single Seated Globe Valves

A maximim allowable leakage of 0.01% of Rated C_v is often based on the nominal or catalog listed actuator size for the given valve tested with 50psi of air across the seat joint.

Double Seated Globe Valves

These valves are specified to have a maximum leakage rate of 0.5% to 0.1% of rated C_v depending upon quality purchased. They are tested with 50 psi of air across the seat joint. See ISA standard 39.1.

Extra Tight Shut Off for Single Seated Globe Valves

A water test is often conducted at the differential pressure rating assigned to the valve by the manufacturer and which is based upon actuator size, air loading, spring force and direction of leakage across the plug (either over or under the plug). Maximum leakage may be specified as 0.0005 cubic centimeters of water per minute per inch of valve seat orifice diameter (not pipe size of valve end) per psi pressure drop. Example: A valve having a 4 inch seat orifice and tested to 2000 psi differential pressure would have a maximum water leakage rate of 4cc/minute. Leakage may be checked with a gas instead of a liquid. The maximum allowable rate is often specified at 6×10^{-7} cubic centimeters per second per inch of seat diameter.

Soft Seats in Globe Valves

These may be leak tested in accordance with SAMA Standard #PMC 23.2c. Under this standard valve actuators are sized to provide a minimum stem force of 150 pounds per inch of seat diameter over and above the plug unbalance force created by the maximum rated differential pressure. The test pressure using air is the maximum rated differential pressure or 300psi, whichever is the *least*, but not to exceed the maximum operating pressure at ambient temperature.

Elastomer Lined Butterfly Valves

These are often tested with 50psi of air as a minimum, or maximum differential operating pressure across the disc. With the downstream side of the disc covered with water, the maximum allowable leakage rate may be specified at one bubble of air in ten seconds, per inch of vane diameter.

Elastomer Sealed Ball and Plug Valves

They are usually bubble tight to their rated differential pressure. Metal seated valves have relatively high leak rates compared to globe style valves. One exception is a rotary cam type plug valve with leakage rates comparable with globe valves.

Theoretical Leakage Formula

The equation relating the fluid properties, flow path geometry, and flow rate is:

$$Q_o = \frac{\pi (p_1^2 - p_2^2) \bar{h}^3}{12 \mu p_o \ln r_o / r_i} \quad \text{(See [1])}$$

where:

$\bar{h}$ = uniform channel clearance [2]
p_o = pressure at standard conditions
p_2 = exit fluid pressure
p_1 = inlet fluid pressure
Q_o = volume rate of flow at standard conditions
r_o, r_i = outside and inside radii of joint sealing area
μ = absolute viscosity

When the terms are rearranged, the uniform channel clearance is:

$$\bar{h} = \sqrt[3]{\frac{12 p_o \mu \ln r_o / r_i \, Q_o}{\pi (p_2^2 - p_1^2)}}$$

for gases the leakage formula becomes

$$Q_o = \left[\frac{\pi h^3 (r_o + r_i)(p_1^2 - p_2^2)}{12 \mu (r_o - r_i) p_o} \right] X \left[1 + \frac{12.76 \, \epsilon p_o \lambda_o)}{(p_1 + p_2) h} \right]$$

λ_o = molecular mean free path at standard conditions.
ϵ = correction factor, 0.9 for a single gas and 0.66 for a mixture.

The problem is that the leakage test gives no indication of whether the leak path is one large scratch or the sum of leakage through millions of tiny tortorous paths. Refinishing the joint may eliminate the first cause, but the condition may already be at the practical limit of seal tightness for the latter.

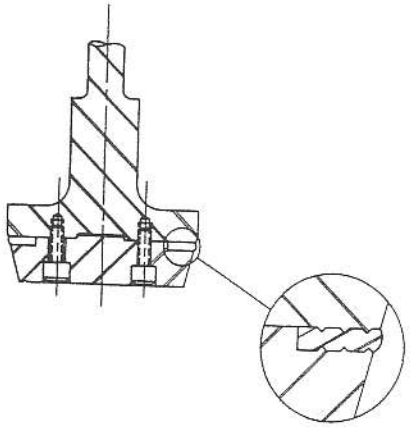
Type Of Gas Flow Through Seat Joint Leakage Pattern

The following is a summary of characteristics for various leak conditions.

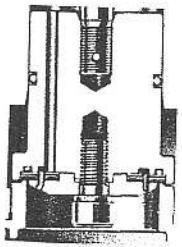
Plug Position	*Type of Flow*	*Minimum Restricting Dimension in Leakage Path*
cracked open	nozzle flow	> 0.005 in.
seating load build up zone	turbulent channel flow	0.0005 to 0.005 in.
valve seated and leakage begins	laminar flow	0.0001 to 0.0005 in.
joint surface waviness, then roughness provides leak paths as deformation begins	transition flow (molecular and laminar)	0.000001 to 0.0001 in.
elastic and plastic yielding of joint has closed large paths	molecular flow	<0.000001 in.

[1] Taken from Reference 50, the formula applies to a flat seat joint.

[2] This is the theoretical separation of two truly plane surfaces to give an equivalent rate of leakage caused by channels, imperfections, etc.

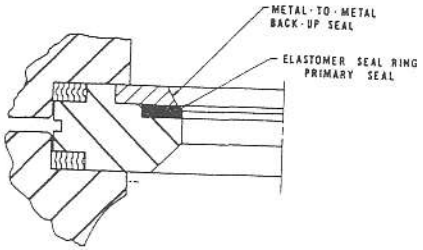


(a) Nylon seal retained in plug.

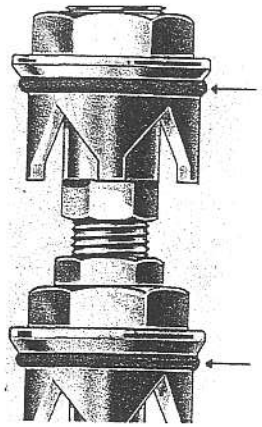


(b) TFE seal in plug of cage guided, balanced valve.

Courtesy Fisher Controls Company

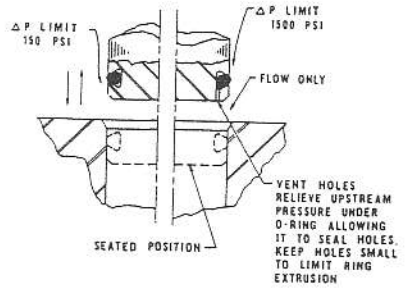


(c) Elastomer seal in seat ring of split body valve.

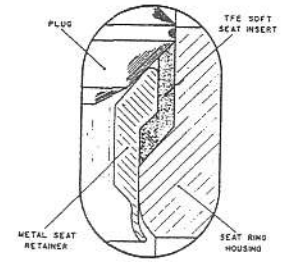


(d) Lightly restrained O-rings for low pressure drop, installed on double V-port plug.

Courtesy Fisher Controls Company



(e) O-ring in dove-tail groove for higher pressure drop. Vent holes relieve pressure under O-ring retaining it in the groove as the seat joint cracks open. Holes are small to prevent seal extrusion.



(f) Soft seat design used in boiler feedwater pump recirculation anticavitation low noise valve.

Handles fluids to 475° F and 6000 psig inlet pressure.

Courtesy of Masoneilan International, Inc.

Figure 74. SOFT SEAT RETENTION.

When sealing to the point of achieving molecular flow, the lighter, smaller and higher velocity molecules, such as helium and hydrogen, will have the highest leakage rates.

Determination of Seat Leakage Rate

Determining the Seat Leakage Rate is not as simple as it may appear. Fluids are subject to thermal expansion and contraction during tests and air may go in or out of solution, causing volume changes in the downstream measuring system.

For accurate testing, maintain the valves and the fluids at an equal temperature and air dissolved in water at equilibrium conditions.

Helium leak detection requires a clean background with large amounts of fresh air for maximum sensitivity.

The following is a summary of the sensitivity of various test methods, given in cc/s at atmospheric conditions.

air bubbles in water (also air and soap bubbles)	1×10^{-3} to 1×10^{-4} cc/s
thermal detectors	1×10^{-4} cc/s
halogen detectors	1×10^{-5} cc/s
mass spectrometer using "sniffer" (helium leak pick up probe)	1×10^{-6} to 1×10^{-8} cc/s

Soft Seating

Resilient composition sealing materials are used to obtain bubble tight sealing with a small actuator force. Compressive seating stresses are such, that the material is elastically deformed into the surface roughness of the mating metal part, to block all leak paths. The permeability of the material, to the fluid, is the source of a very minor leakage.

Materials which are too soft, or that tend to cold-flow (creep) under load, may be stiffened with fillers such as glass. When used in thin sections, and adequately retained, the cold-flow or permanent-set problems may be eliminated.

Seals must be carefully restrained against rupture and blow-out by differential pressure. Several designs of soft-seat retention are illustrated by Figure 74. The bonding of seats to metal parts is an aid, but not a total solution, because bonds are subject to thermal shock cracking and to degradation. Sufficient pressure drop will rupture the bonding material.

A secondary advantage of soft sealing is that the seal, once compressed, is free to re-expand and to follow seat distortion as the pressure drop loading increases. To do this, the sealing material should have a rapid recovery rate upon removal of the load. This action will occur only at resilient temperatures.

Hard materials such as nylon, in thin sections carefully restrained, can handle pressure drops of several thousand psi; whereas, TFE materials are readily abraded.

Soft seals are useful for sealing where contaminants or solid material are trapped in the closed joint. Material as hard as 60D in a raised-seal beadform is capable of sealing 0.01 in. particles, bubble-tight, to 1500 psi pressure drop.

The softer the seal, the better its abrasive resistance, up to the point where it is damaged by pressure drop forces.

Large volume, high pressure blowdown systems are necessary to adequately test elastomer seals and retention means for pressure drop strength. Elastomers, under high loading, tend to act as a fluid and extrusion may occur unless the load is limited or unless a metal-to-metal stop is used. Some joint designs allow soft sealing first, followed by a metal-to-metal closure as a secondary seal in case of soft seal rupture.

The material properties to be considered in the selection of a soft seat are:

1) Fluid compatibility including, swelling, loss of hardness, permeability, degradation;
2) Hardness;
3) Permanent set;
4) Rate of recovery upon removal of load;
5) Tensile and compressive strength;
6) Distortion before rupture;
7) Modulus of elasticity.

Rarely do the physical properties given for sealing materials relate to the actual conditions of loading and strain in valve seals. There is no substitute for thermal and blow-down testing as the means to prove material, seal configuration and joint design.

SEAT ATTACHMENT AND SEALING TO BODY

Seat attachment and sealing to the body is a major consideration of valve sealing, equal in importance to joint seal, bonnet seal, and stem seal. Lack of seat-to-body sealing gives a continuous leak, often blamed on the seat-to-plug joint. In high pressure and/or steam service, leakage behind the seal will actually erode through

the seat or the body bridge wall, creating an enlarging leakage path.

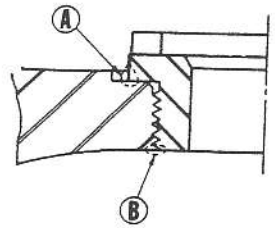
The plug seating load should never be applied directly to a seat-to-body compressible type gasket without a compression stop, or repeated closing will eventually loosen the seal. This also applies to the split body in which the body-joining bolt load should never be applied in this manner. A good design is for the gasket to be shouldered in a metal step, limiting the compressive loading.

Spiral wound gaskets of metal with asbestos and TFE fillers used in a metal step design require a high compression force which increases the bonnet bolting load. This is a particularly important factor where more than one gasket is required in a trim stack. The asbestos filler loses its water of crystallization above 1000°F, turning to powder; therefore, for higher temperatures, ceramic fillers may be used.

Flat metallic gaskets should not be reused if the material is work hardening. This is particularly true for stainless steel.

Split body and cage designs allow the use of a simple, non-threaded seat ring.

Various means of seat attachment are illustrated and explained in Figure 75.



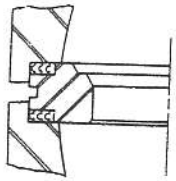
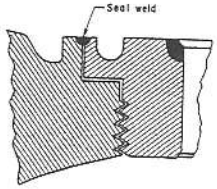
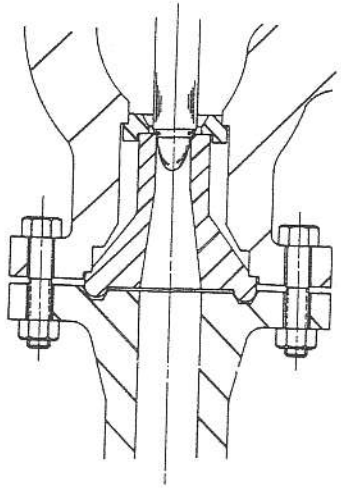
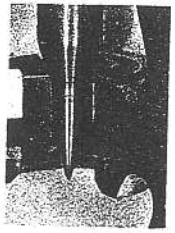
	Type and Applications	Seal Means
a -	Screwed. Standard design up to 12 in., -150° to 450°F (seal-welded above 400° and below -50°F). Limited pressure in large sizes.	Flat metal-to-metal joint (May be seal welded at (A) or (B) for either low or high temperature service).
b -	Screwed. 1/4 to 2 in., -50° to 400°F. Much lower seating torque required.	Metal-to-Metal, differential angle.
c -	Screwed. 1/4 to 2 in., -150° to 850°F.	Soft metal, flat gasket.
d -	Bolted. Used in large size gas regulators to speed seat replacement, -20° to 160°F.	Flat fiber or elastomer gasket.

Figure 75. MEANS OF SEAT ATTACHMENT TO BODY.

	Type and Applications	Seal Means
e -	Integral.[1] All sizes and temperatures. General and slightly abrasive service.	Hard faced joint.
f	Integral welded ring.[1] Used for small restricted ports by some manufacturers.	Hard faced joint.
g	Integral. 1/4 to 2 in., all temperatures.	Projection welded. Stellite seat insert.
h	Brazed.[1] 1/8 to 1/2 in., all temperatures.	Brazed in seat bore step in body.
i	Gasketed.[1] To 1100°F.	Spiral wound gasket, metal O-ring, or flat gasket.
j	Radial.[1] Limited temperatures, -20° to 350°F.	O-Ring, elastomer.

Figure 75. MEANS OF SEAT ATTACHMENT TO BODY (continued).

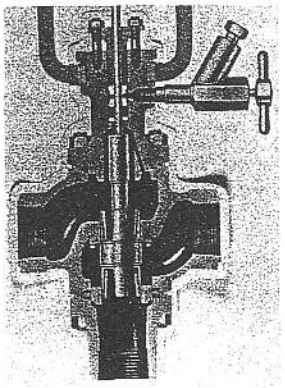
[1] NOTE: Seat to body sealing load is separate from plug seating load; improves seat-to-body seal life.

		Type and Applications	Seal Means
k		Split Body.[1] All sizes. See body rating temperatures.	Spiral wound, or flat gasket.
L		Screwed and Seal Welded. All size valves.	Seal welded with undercut prevents welding strains from cracking or distorting seat joint.
m		Bolted in flange.	Half-oval gasket. Shape of tail piece mates with a straight-side tapered groove of the ring-joint style, pipe flange.
n		Held down by threaded ring.[1]	Any gasket (usually spiral wound metallic).

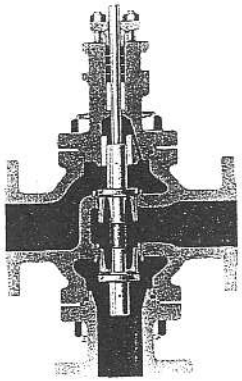
Courtesy Copes-Vulcan, Inc.

Figure 75. MEANS OF SEAT ATTACHMENT TO BODY (continued).

[1] NOTE: Seat to body sealing load is separate from plug seating load; improves seat-to-body seal life.



(a) Mixing service plug type.



(b) Diverting service plug type.

Figure 76. THREE WAY VALVE TRIM.

Courtesy Honeywell, Industrial Division

THREE-WAY VALVES

Diverting or Mixing Trim

Three-way valves with diverting or mixing trim have two plugs fixed to the stem and positioned so that, when one seat is closed, the other is full open, with throttling of flows through both seat orifices for in-between positions.

For mixing service, the plugs are set back-to-back in order for the flow direction of each stream to be under its plug. In diverting service, the position is reversed so flow remains under. V-port plugs are generally furnished to reduce the amount of unbalance forces, both in regular and reversed flow directions. (See Figure 76.)

The plug characteristics are linear and the controller adjusts the valve until the proper ratio of flow is obtained. As flow rate changes in each leg of the process, so does flow resistance in that leg. Thus, it is not possible to maintain a given ratio of resistance between legs and further plug characterizaton is of no value.

Three-way valve trim is subject to thermal stress in temperature control applications. In natural gas dehydration applications, when switching dessicant towers, one flow may be 60°F while the other is 600°F. When removing LPG products from natural gas, the range may be -40° to 120°F.

Conventional three-way valves are limited in their pressure drop performance, because a high seating force is required, in opposite directions, to shut off both ports.

Cage balance trim design overcomes these difficulties and allows shut-off of either port, with flow in either direction, to pressure drops equal to body rating. (Refer to three-way valves in the cage guided, balanced trim section.)

Special three-way plug designs are shown in Figure 77 and 78. The valve of Figure 78 maintains a constant total flow (the sum of flow from both parts), over a broad range.

Three-Way, Three-Position Trim

Three-way, three-position trim as illustrated by Figure 79, consists of two plugs sliding on, and sealed to, the stem with a spring between them that forces the plugs

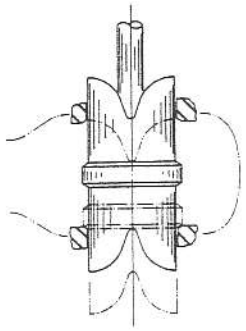
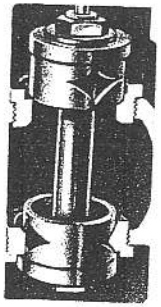
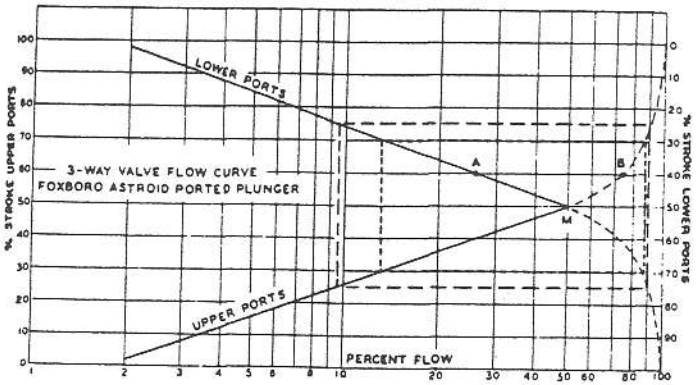


Figure 77. THREE WAY PLUG DESIGN WHICH RESTRICTS FLOW BETWEEN PORTS WHEN STROKING TO DIVERT FLUID.



(a) Astroid plug.



(b) Flow curves.

Figure 78. THREE WAY VALVE WITH ASTROID PORTED PLUG.

Courtesy The Foxboro Company

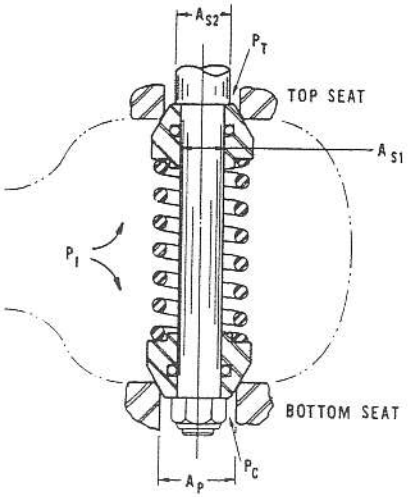


Figure 79. THREE WAY, THREE POSITION TRIM, Non-Balanced Design.

Forces Tending to Seat Plugs: $P_I\ (A_P \cdot A_{S1})$ + spring

Forces Tending to Open Plugs:

bottom plug, $(P_C \times A_P) + P_T(A_{S2} - A_{S1})$

top plug, $P_T\ (A_P - A_{S1})$.

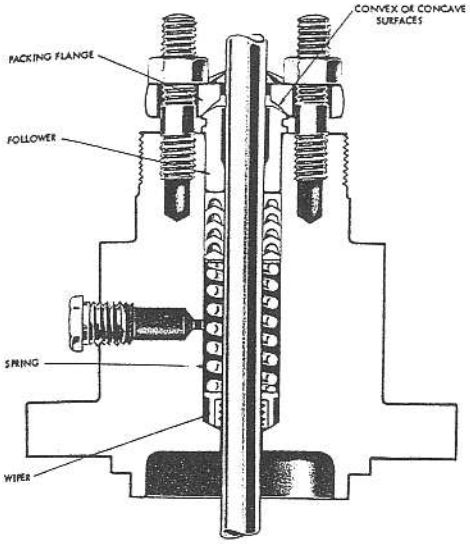
to simultaneously seat on both seats, without interfering with each other. A shoulder on the stem pushes down on the top plug to open as the stem slides through the seated bottom plug, and vice versa, on opening the bottom plug while seating the top plug. Thus, one seat is closed before the other is open.

Three-way, three-position trim finds many applications in manifold service. In oil tank battery service, it either shuts-off the incoming well flow line or directs it to the production or test separator manifold. Cage balanced designs for high pressure service to 5000 psi are shown in the section on cage guided, balanced trim.

PACKING BOX TRIM COMPONENTS
(Metal Parts Wetted by Line Fluid)

Above the top guide is the packing box (see Figure 80) in the valve bonnet neck which the stem enters through a clearance hole in the bonnet. The first part to contact the stem is the wiper ring, when spring-loaded pressure-energized chevron-type packing is used. In most cases a metal housing, with TFE and felt wiper lips, cleans the stem of dirt and abrasive particles before it slides through the packing.

If TFE V-ring seals are used, the next item above the packing box is the helical compression spring which is centered and prevented from contacting the stem by end-guiding or by the inside diameter of

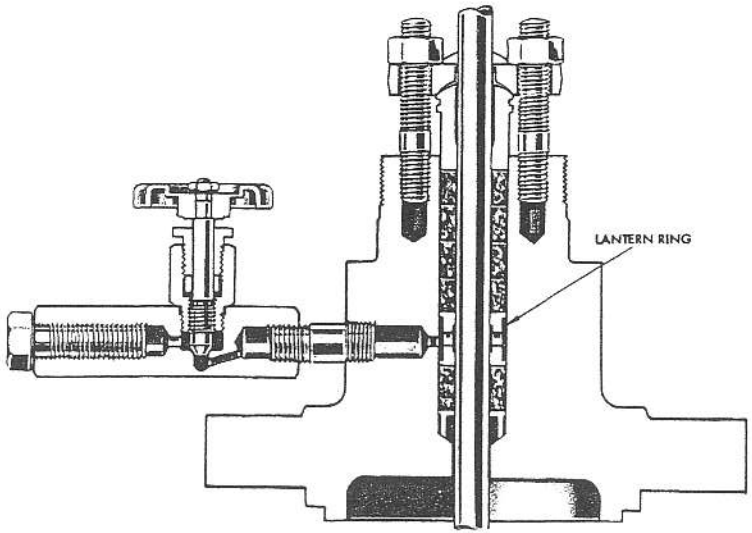


(a) Spring-Loaded, Pressure Energized Packing.

the packing chamber. The spring is eliminated for square packing, because it is energized by compression of the follower, rather than having an initial lip-seal as in the V-ring type.

If square compression packing is used, the wiper ring is replaced with a packing retainer ring to prevent extrusion of the packing downward into the bonnet hole clearance. This ring makes a close fit on the stem, serving as a scraper, and makes a loose fit in the packing box inside diameter. This ring is also called a "junk ring." It is usually a hardened or Stellited part so it won't gall with the stem. In some cases, a metal lantern ring may be installed between the rings of square packing in order to feed lubricant into the packing.

The packing follower, the uppermost part, is the top guide for the stem and has a narrow, close-fitting guide bearing near the bottom. The fit must be tight enough to prevent packing extrusion and "striction" or jamming of stem in the guide bearing. The follower makes a freer fit in the bonnet, letting the main stem post or cage guide center the plug in



(b) Square Ring, Compression Packing Design.

Figure 80. PACKING BOXES.

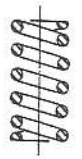
Courtesy Fisher Controls Company

Body material or steel, cadmium plated.

(a) Packing flange.

Type 316 SS
R-Monel
Hastelloy-B
Carpenter-20(w/Durimet trim)

(b) Packing follower.



Type 316 SS
Inconel

(c) Spring.

17-4 PH SS
K-Monel
Hastelloy-B
Carpenter-20

(d) Lantern ring.

17-4 PH SS
K-Monel
Hastelloy-B
Carpenter-20

(e) Packing box ring (wiper).

or

17-4 PH SS
Stellite

(f) Junk ring.

Figure 81. PACKING BOX MATERIALS OF CONSTRUCTION.

proper seat alignment. Packing intrusion does not occur in this larger clearance, because there is no relative motion.

Note, in Figure 80(b), that the follower is finished spherical on its top end so that any cocking of the packing flange, when tightening the square ring packing, cannot cock the follower guide bearing and bind the stem. The follower makes a deep engagement into the bonnet also for this reason.

Figure 81 gives the packing box materials of construction, which follow the principal trim material selection.

Some low pressure, liquid-level control valves (dump valves) use a rotary stem motion, placed 90° to the plug axis. Stuffing boxes seal this rotary motion with O-rings or chevron packing.

Bellows stem seals are used when no stem leakage can be tolerated or where the line fluid cannot be contained by any packing. This may be the case when the process fluid is flammable, toxic, explosive, expensive, or it rapidly destroys packing. They are also used to prevent leakage in vacuum service.

Bellows are fragile and expensive and should be avoided, whenever possible, for continuous stroking (throttling service). They are thin wall convolutions, single or multiple ply, and are subject to fatigue, cracking, and leakage if the manufacturer's rated cycle life is exceeded.

The length of the bellows usually requires an extension bonnet and a conventional packing box is added as a back-up seal. A lantern ring and lubricator valve may be added as a means of adjusting packing tightness (be sure to balance the pressure across the bellows for this test); or for providing a tell-tale leak gauge and bleed-off.

Torsional stress on the bellows from the fluid flow forces tending to rotate the plug are prevented by either:

1) a hexagonal section of the stem below the bellows which slides through a hex bushing; or by
2) a pin, extending horizontally from the stem, which slides in a vertical guide slot.

A good bellows material is A-286 stainless steel which has performed well in

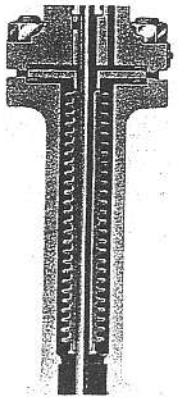
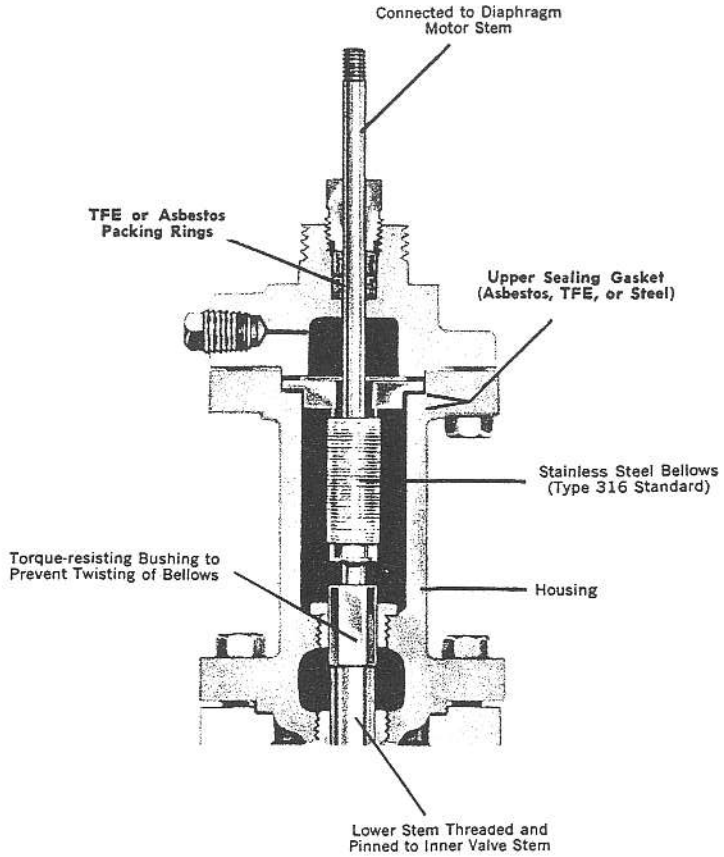


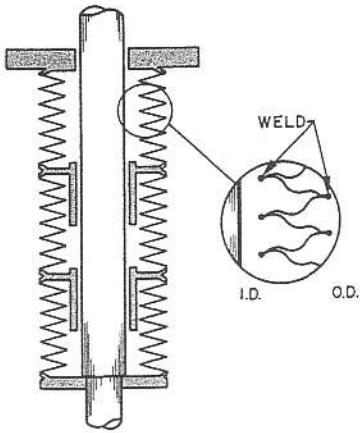
Figure 82. BELLOWS STEM SEAL.

Courtesy Honeywell, Industrial Division



(a) Anti-Rotation

Courtesy Kieley & Mueller, Inc.



(b) High Pressure With Long Stroke Bellows

Courtesy Belfab, a Unit of Bailey Meter Company

Figure 83. BELLOWS STEAM SEALS.

liquid sodium service at 50 psi and 1200°F for as many as 10,000 cycles. Monel is often specified and brass may be used for non-corrosive low pressure service. Regularly supplied bellows for control valve service are usually rated to 600 psi, at 750°F.

Process fluid always surrounds the bellows tending to collapse it. Higher operating pressure may be had by counter balancing; that is, by using an inert gas pressure inside.

Three designs of bellows stem seals are shown in Figures 82, 83a and 83b.

Figure 83(b) is a drawing of a bellows stem seal formed from stamped annular discs which are convoluted and then welded on both inside and outside diameters. The bellows

sections are attached to the spool pieces, which guide the stem in order to prevent distortion or squirm at high pressure. Designs of this type have been used in nuclear service to 650°F. operating, and 800° F. peak temperatures. Designs are available which have a 1.5 inch stroke and only 15 inch overall length. The material used in nuclear service is Inconel 718 which minimizes radiation damage. The life expectancy of a bellows of this type is 2000 full stroke cycles that is, complete opening and closing of the valve or 10,000 throttling cycles traveling over half the stroke length.

VALVE STEMS

The stem carries the load from the actuator to the plug, so in effect, it is a column, loaded at both ends, with horizontal restraint at each end; therefore, it will bow if overloaded. The designer's choice of stem diameter is a compromise between stiffness to prevent bowing, which causes "striction" (drag in packing follower) and results in shortened packing life, and faster bearing wear vs. the effect of line pressure thrusting on the cross sectional area of the stem. In addition, the area of the stem, contacted by the packing, is directly related to the amount of packing friction. Usually stiffness prevails and the latter considerations are counteracted by an additional actuator force requirement.

Some close-coupled, oil field designs have short stems which run very true and increase packing life. In this situation, the open-yoke, replaceable packing feature is traded for a shortened, closed barrel, which protects the stem in an oxygen environment, from corrosive elements such as salt water and acidic gas around the valve.

Cage guided, balanced trim reduces stem thrust sufficiently in high pressure service to give a noticeable increase in packing life, because of the reduction of even very minor bowing and vibration. The stem size is also reduced by using this trim.

Attachment to Plug

Stems are usually threaded and then pinned to the plug to prevent loosening. The stem may bottom in the imperfect thread runout in the plug to make a rigid connection and the pin is usually an "upset" design to anchor tightly. This design presents a maintenance problem in replacing the plug; therefore, plug-stem combinations are often replaced as a unit.

Other designs use an elastic nut to hold the plug against a shoulder on the stem. Some designs are pinned with spring pins which allows repeated plug replacement. The stem shoulders in the plug to take the underseat load, and satisfactory life may be obtained by using Monel pins. Stainless steel spring pins may break from stress corrosion cracking in tough service applications.

Pins are seal-welded in high and low temperature service.

Surface Finish and Hardness

Surface finish and hardness are important considerations for tight sealing and long packing life, concurrent with a low rate of stem wear from packing, fluid particles, and stem guide bearings. Some manufacturers get good results by finishing the portion of the stem that slides in the packing to two or three microinches. Other prefer a finish of 8 to 10 micro inches, based on the theory that the slight surface roughness will provide a small amount of fluid that will lubricate the stem-to-packing interface and they also get good results.

Although stem wipers or scraper rings are used under the lower packing ring, some line fluid abrasive particles eventually enter the stem-to-packing interface. The stem should have a hardened surface to prevent scoring and rapid packing wear and leakage. Either malcomizing[1] or hard chrome plating are often used for highly abrasive service.

The following is a summary of stem materials and their applications (stem pin material should be the same as the stem material).

Material	*Application*
Type 316 SS	- non-abrasive service
R and K Monel	- slightly abrasive service
hard chrome plate or Malcomized[1] 316 ss.	- abrasive service
Hastelloy-B®	- with Hastelloy-B® bodies
Carpenter-20®	- with Durimet® Bodies

QUICK-CHANGE TRIM

Quick-change trim is defined as trim which may be removed and replaced by simply opening the bonnet joint; the valve body remains in the pipe line and the unscrewing of seat rings with a special clamp on lug wrench is eliminated. The seat is sealed to the body bridge wall by either a gasket or an O-ring, and a "flow cage", which holds down the seat, is in turn loaded by the bonnet. The design temperature is limited by the gasket. Two designs of quick-change trim are shown in Figures 84 and 85.

Single and double seated valves as well as three-way valves may be of quick change design. Another design enables removal of the seat supporting cage by opening the bottom cover, then the seat, plug, and stem follow. The actuator is not disturbed in this design.

The advantage of quick-change trim is the reduction in field maintenance time required to replace leaky trim or to change the trim flow capacity. Sometimes, regular trim valves must be removed from the line and taken to the maintenance shop to remove the screwed seat rings.

Some oil field designs, of one and two-inch sizes, use a threaded hammer-lug, union nut to retain the bonnet which is sealed by an O-ring; thus reducing downtime to a minimum.

[1] A surface hardening method for 300 series stainless steel, which is similar to the nitriding of alloy steels.

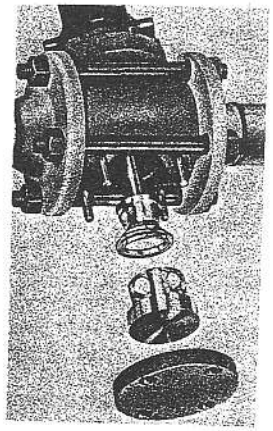


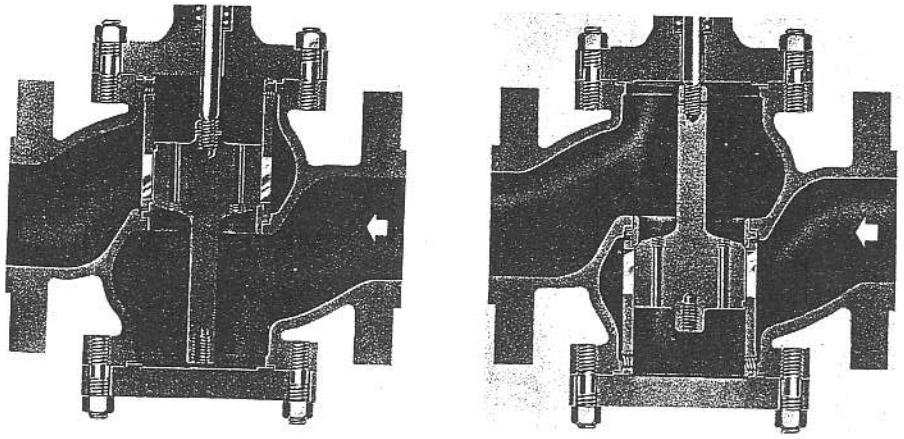
Figure 84. QUICK CHANGE CAGE TRIM.

Courtesy Fisher Controls Company

BUTTERFLY VALVE TRIM

Introduction

Butterfly valves are prominent in control applications, because of their compactness, and their moderate torque requirements. A flat disc, pivoted in the flow stream, seats on an elastomer body liner for tight shut off. Many designs seal tightly to 175 psi WP and special seals are used for higher pressure. The liner may be either replaceable or bonded to the body wall. The valve stem axis may be either normal or canted with respect to the pipe axis. The seats may also be canted. Canted design presents less of a problem in obtaining a tight seal where the stem enters the liner, but requires a more difficult actuator mounting for the canted stem design. Stems are usually



(a) Flow under plug assembly.

(b) Flow over plug assembly.

Figure 85. REVERSIBLE QUICK CHANGE TRIM.

Courtesy Kieley & Mueller, Inc.

pinned to the disc and the pins must be sealed. Offset discs eliminate this problem and will turn free, once they clear the seating zone. Disc stem pins are kept high on the shaft, near the operator, to reduce torsional strain and dead band in valve operation. For the same reason dual pins must be close together so the lower pin will take its share of the load. Disc design may be modified to reduce the peak dynamic operating torque that occurs when approximately 30° to 40° open in a flow control installation. At constant pressure drop, this peak occurs between 70° to 80° (70° for a thin disc). Some designs seal at higher pressures by pressure energizing the liner against the closed disc using a separate pressure source.

Many non-metallic materials are used for liners, including the TFE's which are often energized by compressed rubber rings on their back sides. Some designs have a sealing element in the disc, thus, requiring a canted stem. Special means of seal retention allow these designs to work at much higher pressure. For high temperature service, a clearance may be allowed or a metal expansion-spring type piston ring may be placed in the disc, to give a better seal.

Butterfly valves are manufactured in three grades: standard, medium and heavy duty. The heavy duty valve has a stronger shaft and vane construction then the standard valve. If the differential operating pressure is low, then a standard valve is used. A standard valve has a greater flow capacity than a heavy duty one. Heavy duty valves have pressure ratings up to 1500 lb. and can withstand a pressure differential of 1200 psi.

VALVE SIZE	PRESSURE CLASSIFICATION (ANSI CLASS)
1 in. to 72 inches	25-300
1 in. to 24 inches	300-1500
1 in. to 12 inches	1500-2500

The fully open position of the valve can be in the range of 60 to 90 degrees rotation relative to the closed position. Rangeability varies from 33:1 for the 60 degree opening design to 100:1 for the low torque design, which has a larger opening. Their inherent flow characteristic is equal percentage but cams can be used in a positioner to generate other characteristics. The valves can operate between -460 to 2100 °F.

Butterfly Valve Advantages

Of the valve types, butterfly valves contain the least amount of body metal. This can be an important money saving fact if the valves must be constructed from stainless steel, as in the paper and chemical industries. Elastomer liners provide drop-tight shutoff which is much harder to obtain in metal seated valves. The valve body occupies the minimum amount of flow line space and flangeless designs make up with relatively short studs compared with other flangeless valves. Butterfly valve pressure shutoff capabilities require modest actuator power compared to other types of valves since the butterfly vane is balanced in the closed position. Dynamic flow torque does not peak until the valve is in the range of 30 to 75 degrees open. This is an advantage in that the peak dynamic flow torque does not occur when more torque is required to unseat the valve from the elastomer liner. Another advantage is the broad range of liner and vane materials available to withstand corrosive conditions.

Unlined Body Construction

This type of valve is used where tight shutoff is not required and/or where an elastomer liner is not compatible with the line fluid or operating temperaure. It is the least expensive design. The minimum leakage is 0.5% of rated C_V. A disassembled valve shown in Figure 86, consists of a body, shaft, bearings and disc.

Figure 87 shows the complete valve body assembly. The disc pins are located high on the shaft in order to reduce torsinal displacement of the disc and actuator driving arm.

Figure 88 illustrates how the disc in an unlined valve must clear the body at all angles of rotation. To maintain this clearance, the shaft is positioned by the outboard bearing. The location of the outboard bearing can be found in Figure 87.

The bonnet packing box contains the stem seal of a unlined valve body. The extended bonnet shown in Figure 89 allows operation to 650° F. with the TFE asbestos packing temperature maintained below 450°F.

Leakage Rates of Unlined Valves

The leakage rates given are based on the best fit obtainable within normal manufacturing tolerances while allowing clearance to operate the valve at design temperatures without galling. The factors may vary somewhat from one manufacturer to another and from one type of design to another by the same manufacturer

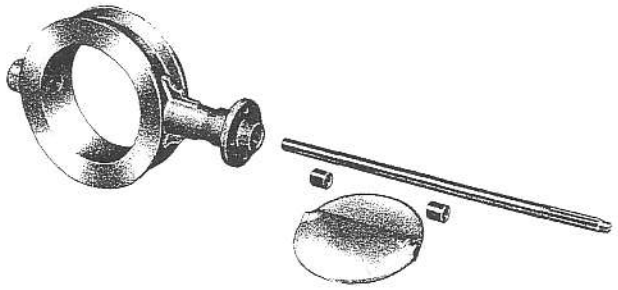


Figure 86. DISASSEMBLED UNLINED VALVE.

Courtesy Masoneilan International, Inc.

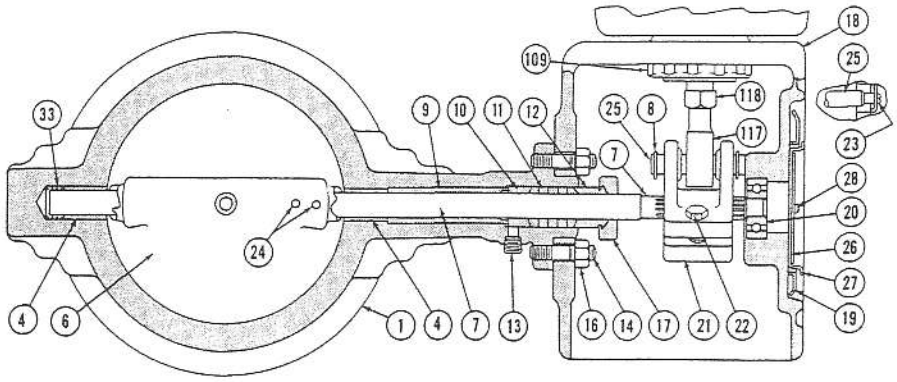


Figure 87. UNLINED VALVE BODY ASSEMBLY.

Courtesy Masoneilan International, Inc.

4. Lower Bearing
24. Disc Pins
9. Spacer Tube
10. Packing Box Ring
20. Outbound Bearing

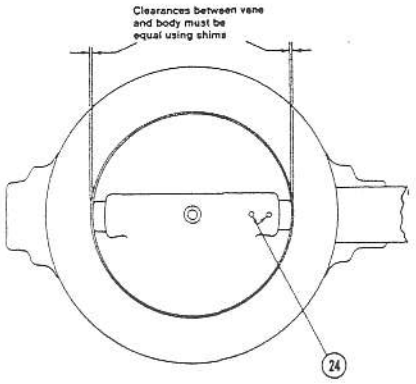


Figure 88. POSITIONING DISC IN UNLINED BODY.

Courtesy Masoneilan International, Inc.

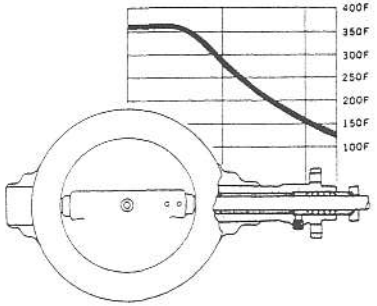


Figure 89. TYPICAL TEMPERATURE GRADIENT ACROSS BONNET.

Courtesy Masoneilan International, Inc.

TYPICAL UNLINED VALVE LEAKAGE RATES

The subject leak rates can be calculated by dividing the tabulated factors by the square root of the static pressure drop. These factors may differ somewhat from one manufacturer to another and from one type of design to another by the same manufacturer.

Valve Size	Factor (g.p.m. / 1 psi drop)
2	.685
3	1.028
4	1.370
5	1.713
6	2.055
8	2.740
10	3.425
12	4.110
14	4.795
16	5.480
18	6.165
20	6.850
24	8.220

Courtesy Kieley Mueller, Inc.

For tighter shutoff in high temperature service, a Ni-resist split ring seal may be inserted in a groove in the outer edge of the disc which will reduce leakage to less than 0.1%. An offset stem design is required to have the leakage less than 0.5% of rated C_V for the conventional unlined valves. Unlined valves have disc to body radial clearances ranging from .005 in. for the 1-20 inch valve sizes to .015 in. for the 72 in. size. The metal ring reduces ambient leakage to 1/3, but at 1000° F. leakage may be 1/20th that of a conventional design. Rings are also available in 316 stainless steel 400 Series stainless steel and Haynes Stellite # 25 for operating temperatures ranging to 2100° F. Figure 90 is an example of a metal seat type valve suitable for -250° to 400°F service. Figure 91 illustrates a high temperature design without a piston seal. This type of special design is available for services up to 1500°F. The body is stainless steel with an internal liner which is attached at one end and free on the other to permit expansion without creating a thermal stress.

Stepped seat designs are also utilized and these have a leakage rate 1/3 that of conventional unlined construction in that the raised seat rib in the body is interrupted in two places to pass the stem. Pressure drops are limited to 10 psi due to erosion of the 90° turnings of the flow path at the cracked open position.

Valves with outboard bearing designs are often specified for high temperature service, however, the long overhang of the shaft limits the pressure differential rating of the valve.

Refractory lined valves are often used at temperatures above 1000° F. to allow use of carbon steel piping. Only the disc, shaft and bearings need be of high alloy materials. These valves often handle hot flue gases that sometimes reach 2100°F.

Weld end bodies are available for extreme hot or cold service at high pressure. Figure 92 illustrates such a design.

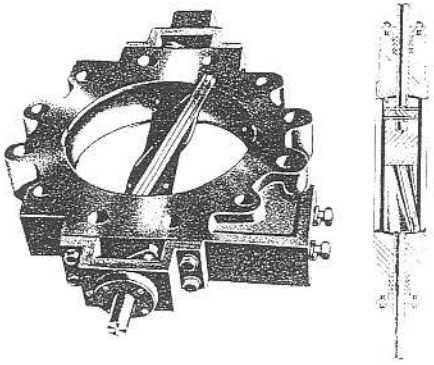


Figure 90. MEDIUM DUTY WITH A METAL SEAT.
It has an offset Disc with a Piston Ring Assembly.
Courtesy Fisher Controls Company, Continental Division

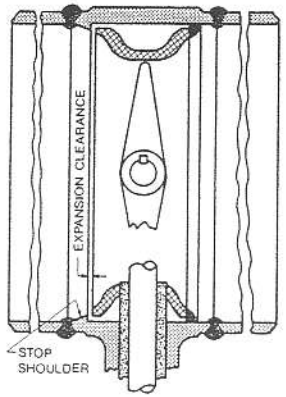


Figure 91. VALVE DESIGN FOR 1500°F SERVICE.
Courtesy Masoneilan International, Inc.

Elastomer Lined Construction (Bubble Tight Closure)

Formerly all butterfly valves were built with bonded liners or wraparound liners. The wraparound or snap-in liner extends around the flange face of the valve. Sometimes the valve body has a raised rim on its outer edge to prevent excessive extrusion of the liner by the compressive stress generated by flange bolt-up. The newer type push-in liners have a metal insert stiffener that is bonded to the liner material which allows higher flow velocities without liner flutter. This can improve valve performance under vacuum conditions without the liner collapsing. Newer wraparound designs have a center support which can resist axial distortion.

Wrap-around type liners must be bonded to the body for high velocity or vacuum service unless they have internal metal supports.

Special undercut discs are available for operation up to 50 psi presure differential. They have less interference with the liner and

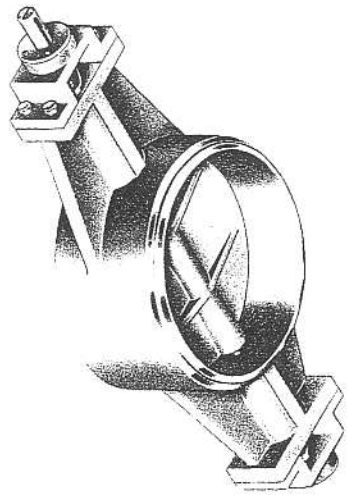


Figure 92. HEAVY DUTY WELD-END TYPE BUTTERFLY VALVE.
Courtesy Fisher Controls Company, Continental Division

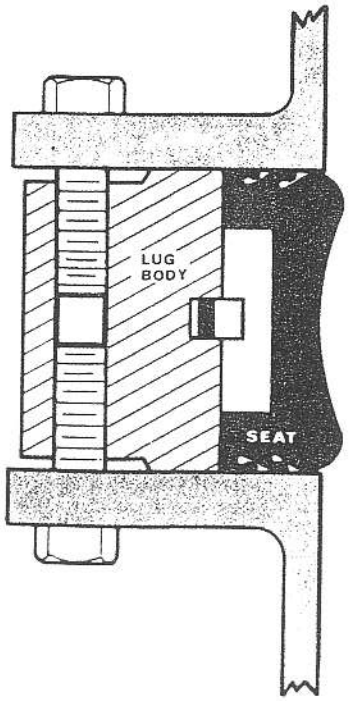


Figure 93. LINER ANCHORED IN BODY.
Courtesy DeZurik Corporation

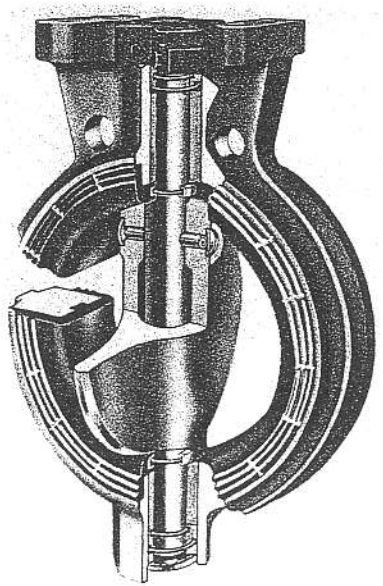


Figure 94. LINER PROJECTING INTO BODY GROOVE AND WRAPPING AROUND FLANGE FACE TO FORM A GASKET.
Courtesy Crane Company

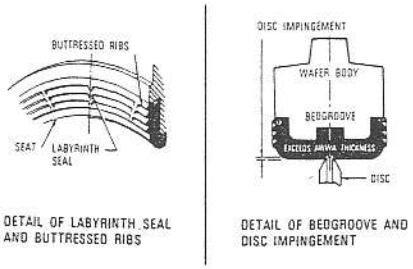


Figure 95. WRAP AROUND LINER.

Courtesy Crane Company

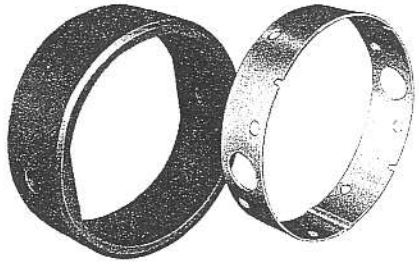


Figure 96. PUSH IN LINER.

Courtesy Masoneilan International, Inc.

lower opening and closing torques.

Sealing of the liner to the mating flanges is accomplished by having the liner extend a small amount past the valve body face on either end. This gives a controlled compression for adequate sealing and prevents overcompression of the liner which could then interfere with the vane. This is important since the opening torque of the valve actuator of a lined valve must always exceed the closing torque of the valve to insure valve opening. The ratio of these torques must not be upset. Some liners have concentric serrated faces to improve sealability.

Elastomer Liner Design (Bubble Tight Closure)

Six liner design types are utilized.

1) Liner Bonded to Body

This type of liner is not replaceable in the field. It is generally used in small size valves and on some large valves only when the application is critical (not illustrated).

2) Liner Anchored to Body

This type is anchored by a metal or rubber retaining ring that projects into a groove in the valve body. Figures 93 and 94 are examples of this valve type.

3) Liner Wrapped Around Faces

The liner is wrapped around the flange faces of the body in order to anchor it in place and to form a gasket. Figure 95 is a drawing that shows the pressure energized lip seals. Controlled compression is obtained by the extension of the rubber liner past the body flange face (usually .030 in.). The grooves allow displacement of the rubber liner on makeup of the valve in the piping without causing distortion and disc binding. This design securely anchors the liner, only being surpassed by the bonded type.

4) Push-In Liner

This type of liner is made from an elastomer ring that is bonded to a metal insert. This type of construction will provide stiffness for vacuum service and will prevent extrusion by differential pressure across the disc. This liner is readily replaceable in the field. A push-in-liner is pictured in Figure 96. Figure 97 shows how the push-in liner is anchored in the valve body by the faces of the standard bore line flanges.

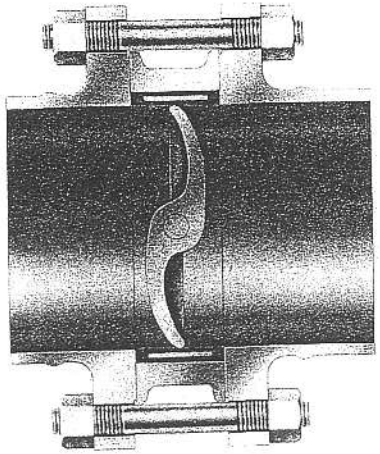
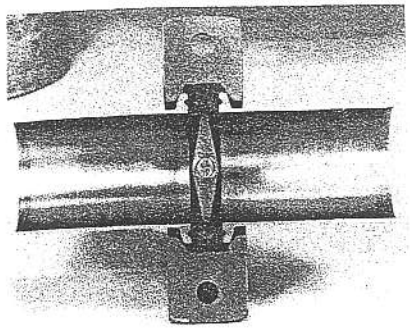


Figure 97. LINE FLANGES ANCHOR PUSH-IN LINER.

Courtesy Masoneilan International, Inc.



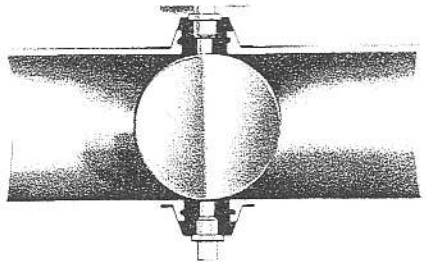


Figure 98. SPLIT CAMP HOLDS BOTH LINER & PIPE FLANGE.

Courtesy Ladish Company, Tri Clover Division

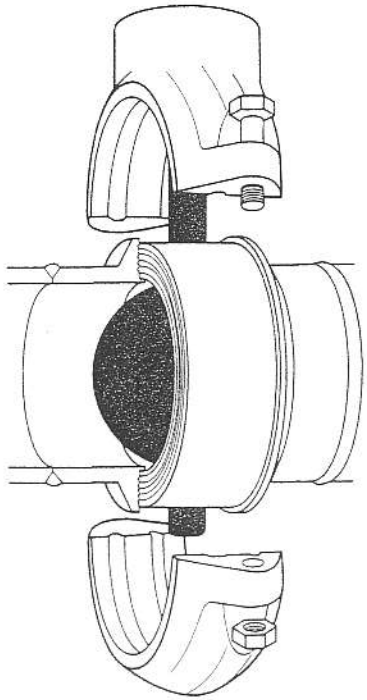


Figure 99. ASSEMBLING SPLIT BODY VALVE.

Courtesy Worcester Valve Company, Inc.

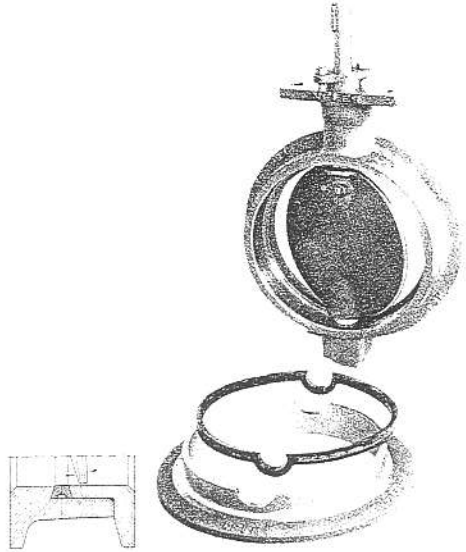


Figure 100. THIN SECTION CLAMPED LINER.

Courtesy Kamyr, Inc.

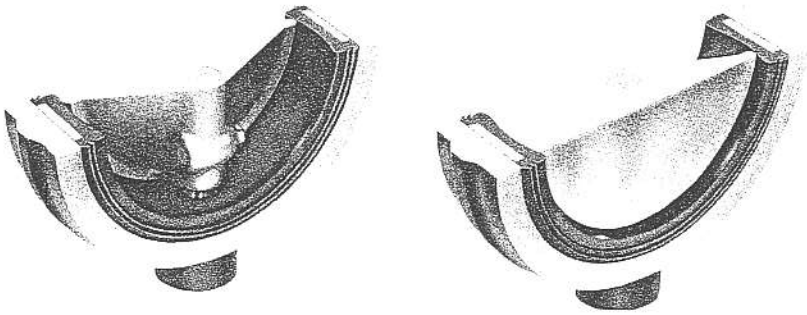


Figure 101. REVERSIBLE SEATS IN OFFSET DESIGN.
Courtesy DeZurik Corporation

5) Clamped Liner

For this type of liner a split clamp holds both liner and pipe ends together. Seat tightness may be adjusted by varying the split clamp bolting torque. Piping must be moved aside to install and remove the liner. This type of liner, as shown in Figure 98, is used by the food industry.

Figure 99 illustrates the assembly of the split body clamp over the integral trunnion-vane. Special pipe hubs are welded to the pipeline which are serrated to fit raised concentric ends of the thicker liner, thus giving positive axial alignment of valve and piping. The thick liner allows deeper penetration of the vane. A lower torque force is needed to seat the valve than with the conventional liners.

Another type of design involves clamping a thin section liner by means of an extension flange as shown in Figure 100. The relief on the outside of the liner allows for swelling. This design has operated satisfactorily on line, with a 3.5% ground wood slurry with an 85 psi shut-off pressure differential at 70°F. for 482,000 cycles. At the end of this test,leakage was only 0.1 gpm. The liner may be replaced without removing the stem and disc.

6) Special Liners

The offset stem design of this valve allows two seating surfaces which are interchangeable when the valve stem is centered on the pipe axis. To change the seating surface, simply disconnect the disc from the actuator and rotate it 180°. Figure 101 illustrates the reversible valve seats.

Lined valves must be used with proper pipe schedules and/or flange bores to adequately support the liner as illustrated in Figure 95. This is somewhat more critical with the field replaceable push in type liners. Heavy wall pipe in some cases will interfere with the vane.

Stem Seals In Lined Valves

Sealing is usually accomplished by having an O-ring inserted in a groove formed in the liner as shown in Figure 102.

Figure 103 illustrates how larger valves sometimes have the O-ring groove in a housing bonded to the liner. The housing can be a laminated phenolic thermosetting plastic. This is necessary to prevent groove distortion in the larger size liners.

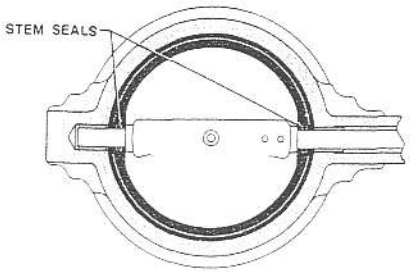


Figure 102. O-RING STEM SEAL IN LINER GROOVE.
Courtesy Masoneilan International, Inc.

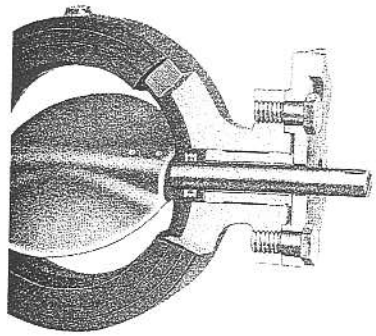


Figure 103. O-RING HOUSING BONDED IN LINER.
Courtesy Hammel-Dahl / Conoflow Division of ITT Corp.

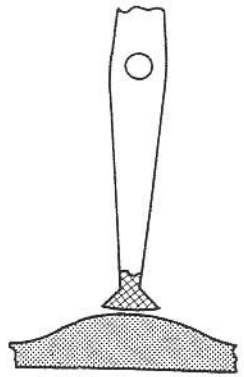


Figure 104. KNIFE EDGE DISK.

Erosive Service

Rubber lined valves in combination with hard disc and bearing materials are used in situations that experience moderate pressure drops. When scale or sludge accumulation is extensive around the seat area, a hard faced "knife edge flow" disc may offer a satisfactory solution to the problem. Figure 104 illustrates how the valve seat is kept clean.

Temperature Ratings For Unlined Valves

Body Material	Useful Temperature Range
Iron	-20° to +400° F.
Steel*	-20° to 650° F.
Alloy Steel**	-50° to 850° F.
Stainless Steel	-460° to 1500° F.

*Carbon Steel, WCB -20°F, LCB -50°F.
**Alloy Steel, LC1 -75°F, LC2 -100°, LC3 -150°F

Temperature Ratings For Lined Valves

ELASTOMER	TEMPERATURE LIMITATIONS °F LOW LIMIT	HIGH LIMIT	SERVICE
Natural Rubber	-20	160	Air, Water
Butyl	-10	300	Acids, Alkalis, Steam, Alcohols
Silicone	-75	200	High Temperature Air
Neoprene	0	175	Air, Water, Freon 12, 114, 22
Hypalon	+15	225	Acids, Alkalis
EPT Rubber (Nordel)	-30	250	Acids, Alkalis
Polyurethane, Adiprene	+10	200	Air, Water to 150°F.
Fluorocarbon (Viton, Fluorel)	+20	400	Aromatic, Alipmatic Hydrocarbons, Degreaser, High Temperature Air and Gases.
Nitrile (Buna N, Hycar)	+10	180	Freon 11, 12, 13, Air, Water, Alcohols, Aromatic Hydrocarbons
White Hycar	+20	200	Foodstuffs
White Neoprene	-20	175	Foodstuffs
White Butyl	-10	300	Foodstuffs

Performance Characteristics Of Elastomer Liners

Material	Temperature Limits	Resistance to				
		Abrasion	Gas Permeability	Lubricating* Oils	Water*	Tearing
Buna N	10 to 180°F.	Good	Fair	Excellent	Excellent	Fair
Nordel	-30 to 250°F.	Excellent	Fair to Good	Poor	Very Good	Good
Viton	-10 to 400°F.	Good	Poor	Very Good	Very Good	Fair
Neoprene	0 to 175°F.	Excellent	Good	Good	Fair to Good	Fair to Good

*Indicates ability of liners to withstand swelling.

The useful life of elastomer liners is dependent on the type of service it must perform. On a particular wet non-erosive and non-corrosive service line, a valve cycled 30,000 times before leakage occured. This same valve with a new liner, on a dry gas service line operated 3,000 cycles before leakage occurred.

Stop valve liners are often softer in the range of 50 to 60 Durometers than those used in a control valve. Thus there is less resistance (torque) when opening the stop valve after it has remained closed for a long period of time. In that control valves cycle more frequently, a harder liner is desirable to handle the high pressure differential when it is throttling near the closed position.

Materials Of Construction

Valve bodies and discs are available in many different materials to meet cost, corrosion and thermal requirements. Shafts are usually limited to high-strength non-corrosive materials. The disc sealing edge may be inlaid with stainless steel or "hard-faced" for erosive service and service at higher temperatures.

MATERIALS OF CONSTRUCTION

Temperature Range	Body	Disc	Shaft	Shaft Bearings[1]
Standard Construction				
-20° to +400°F.	Cast Iron	Cast Iron Meehanite Carbon Steel	316 s.s. or 17-4 PH	TFE Loaded 440C (hardened) Stellite #6 Roller Bearings
-20° to +650° F.	Carbon Steel	Cast Iron Carbon Steel 316 S.S.	316 s.s. or 17-4 PH	TFE Loaded 440C (hardened) Stellite #6 Roller Bearings
-460° to +1500° F.	316 s.s.	316 s.s.	316 s.s. or 17-4 PH Hastelloy-X® (above 900°F)	TFE Loaded 440C (hardened) Stellite #6 Roller Bearings
Special Construction		Bronze 416 s.s. w/Nickel Edge Aluminum		
Corrosive Service				
-130° to +400° F.	Ductile Iron (TFE Lined) t = 3/32 in.	Monel® Nickel Durcomet 100® Durimet 20® Chloromet 283® Titanium		

[1]Internal bearings must be compatible with line fluid and temperature.
Large bodies may be metal lined with non-corroding material.
TFE liners are spark tested to 10,000 volts to insure there is no permeability. Bodies are shell tested to 225 psi and the seat is given a 150 psi differential pressure air leakage test.

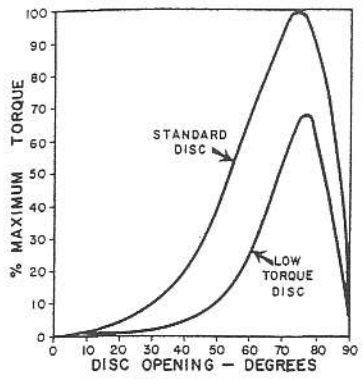


Figure 105. UNLINED VALVE DISC TORQUE WHEN SUBJECTED TO CONSTANT PRESSURE DROP AS VALVE OPENS.

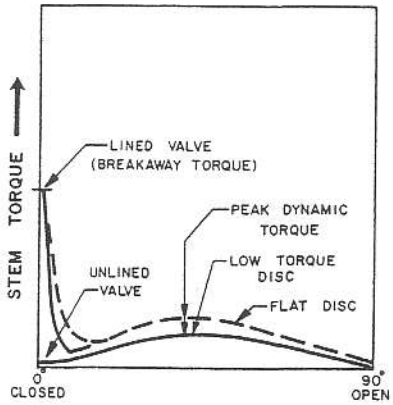


Figure 106. LINED AND UNLINED VALVE TORQUE WITH FALLING DIFFERENTIAL PRESSURE AS VALVE OPENS.

Disc Shapes

Standard discs have a flat shape and when subjected to a constant pressure drop as the valve is opened generate a torque curve similar to the upper curve of Figure 105.

Low torque discs are illustrated in Figures 18 and 19 of Chapter 3. Their torque curves are similar to the lower curve found in Figure 105.

In practice, the differential pressure often decreases with valve opening and thus the breakaway torque of a lined valve is then the predominant factor. Figure 106 helps to explain this graphically.

Valve actuators are chosen to match the needed breakaway torque. The torque value increases with time, as the elastomer deforms in the area where the edge of the disc touches it. Breakaway torque for rubber liners may be 3 times that of immediate reopening torque after a valve stands closed for several days. Newly installed valves should be opened manually if necessary, before going on automatic process control.

Flow Direction

1) Standard Flat Vane
The flow may be in either direction.

2) Cambered Vane
The flow may be in either direction but the valve must operate in the proper quadrant. Figure 107 shows a typical set-up.

3) Vane with Tail
The flow can only be in one direction. Figure 19 located in Chapter 3 is an example of such a vane while Figure 109 of this chapter illustrates proper flow direction.

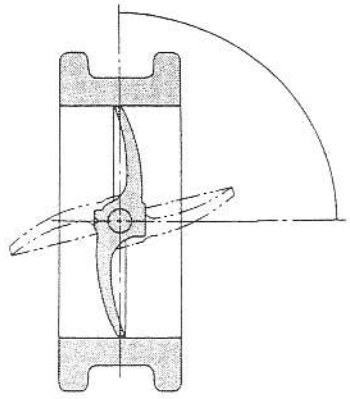


Figure 107. CAMBERED VANE OPERATING QUADRANT.

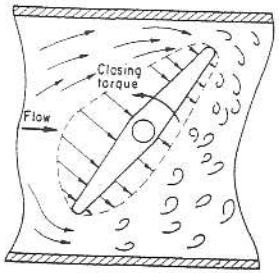


Figure 108. AERODYNAMIC FORCES ACTING ON PARTIALLY OPEN FLAT VANE.

Figure 109. VANE SHAPES.

Aerodynamic Principles Leading To Shaped Vanes And Lower Torque

In both the completely closed and open positions pressure forces are balanced on each half of the vane and therefore torsional forces are non existant. In the open position the vane acts like an air foil in which flow passes over it as if it were the wing of an airplane. The velocity of the flow over the leading edge is less than the velocity over the trailing edge which creates an uneven pressure distribution across the upstream side of the vane. Pressure forces normal to the vane acting on the leading half are greater than the summation of forces acting on the trailing half. This creates a torsional force moment tending to close the valve, and it is this moment which limits the differential pressure operating capability of the valve at different degrees of opening in that the actuator can only offset a torque equal to its rated capacity. Similar negative pressures exist in the downstream side of the vane. These forces increase with the velocity of the flowing fluid. Figure 108 helps to explain the above information.

The torsional closing force increases rapidly as the valve vane approaches 70 degrees open, then falls off rapidly when turned past 75 degrees open. A valve throttling in the 75 degree open region would be unstable due to a great change of dynamic torque for a small change in controller set point. The magnitude of this torque may be reduced by two thirds by placing a "fence" on the trailing edge of the vane. The vane may then be operated through a full 90 degree range with the advantage of greatly increased flow capacity. The fence also extends the equal percentage flow characteristic of the valve into the 70 to 90 degree region. Cambered vanes also reduce dynamic closing torque and this effect is achieved for both forward and reverse flow. Orientation of the valve in the piping is therefore no problem. Vanes with fences can only function in one flow direction and therefore must be properly oriented in the flow line. Figure 109 illustrates the shapes of the vanes described.

The principal advantage of these airfoil vanes is that their use allows the valve to operate at higher differential pressures with smaller actuator power requirements. Figures 105 and 106 help illustrate this concept.

Cavitation

Butterfly valves have a high pressure recovery rate and readily cavitate at high pressure drops. Low torque discs can be used to reduce this effect. See Figure 110 for more details.

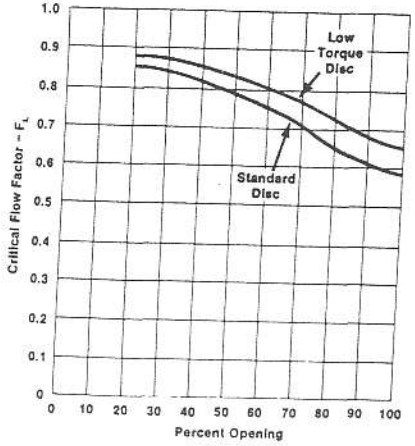


Figure 110. REDUCED CAVITATION SENSITIVITY FOR CAMBERED VANE.
Courtesy Masoneilan International, Inc.

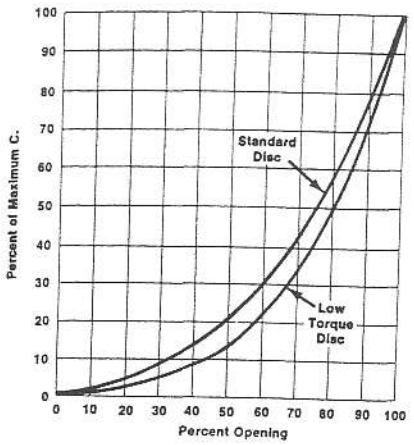


Figure 111. BUTTERFLY FLOW CHARACTERISTIC. It is Equal Percentage.
Courtesy of Masoneilan International, Inc.

Flow Characteristics And Rangeability

The butterfly valve generates an equal percentage curve (as shown in Figure 111) when the disc is symetrical, while offset discs are quick opening the first 10-15°. Rangeability is from 33:1 for a 60° maximum opening valve (usually flat vane) to 100:1 for one with a special shaped vane for a low torque application which may open from 75° to 90°. Larger unlined valves have higher rangeability values than smaller valves. The fact that vane-to-body metal clearance is a smaller percentage of the full open flow area in large valves then in small valves accounts for this.

Flow Capacity And Degree Of Opening

Butterfly valves are designed to exhibit a given C_V value in relation to their pressure differential ratings. The ratings are taken with the disc in the closed position. Proper shaft diameter is essential to handle pressure differential shear stress and valve operating torque. The vane wall surrounding the shaft must be thick enough to support disc pinning. These two factors reduce flow area and pressure differential, therefore, the C_V value must be balanced with the other factors in order to obtain a useful design. Reduced port valves are also available when it is not possible to reduce the piping diameter. Valves opening a full 90° have double the flow capacity of those opening 60°.[1]

Additional capacity may be obtained in the 75° to 90°[1] opening region by using a double stem suppported disc design having a thin center. This is known as an open center vane. Fifty percent more fluid flow capacity is available at 90° over the standard shafted vane. Figure 112 is a drawing of an open center valve vane.

Flat vane type valves are usually limited to 60 degrees opening in throttlng service. This is because the rapid change in actuator thrust output required when the vane is operating near the peak torque position could easily create throttling instability. For on-off service flat vanes may be opened to 90 degrees for maximum flow capacity. Shaped vanes have flatter torque curves and lower torque requirements and usually throttle to 75 degrees open. The fence type of shaped vane may be throttled to 90 degrees opening.

Typical 6 Inch Valve Data

Degrees Of Opening	C_V Valve
60°	600
75°	1000
90°	1400

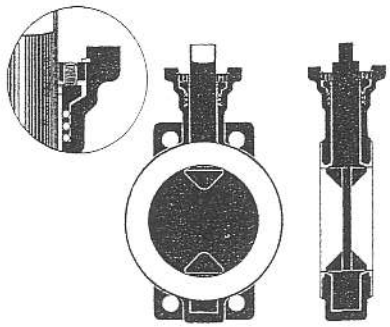


Figure 112. OPEN CENTER VANE WITH HIGH FLOW CAPACITY. Note the stem seal construction.
Courtesy of Xomox, Inc.

[1]Depends upon vane aspect ratio, (diameter-thickness), which limits flow increase as vane approaches 90° full open.

Shut-Off Capability

Butterfly valves are divided into three classes depending upon their ability to close with a specific differential pressure range. This classification is not necessarily related to the valve body pressure ratings and is very general.

Light Duty -	less than 50 psi differential pressure.
Medium Duty -	50 to 150 psi differential pressure.
Heavy Duty -	greater than 150 psi differential pressure.

From a design standpoint differential pressure rating is limited by:

1) Double shear strength of the shaft of given diameter.

2) Shaft bending moment related to distance bearings are displaced from the edge of the vane.

3) Torsional strength of the shaft at operating temperature.

4) Vane thickness.

5) Seating and unseating torque with elastomer lined valves as related to torsional strength of shaft.

6) Desired flow capacity as limited by shaft diameter and vane thickness.

The manufacturer balances the above parameters to manufacture a valve having a reasonable flow capacity coupled with adequate differential pressure shut-off. Also important is adequate dynamic throttling capability to various differential pressures at different degrees of opening because valve operating torque varies with vane rotation.

One piece integral vane and trunnion designs allow a thin vane and heavy trunnion thus yielding higher flow capacity and higher differential pressure capability.

Another factor which must be taken into account for lined tight shut-off valves is that the actuator force to unseat the vane must be greater than the torque to move it into the seating position. This provides the needed safety factor to assure opening of the valve. "Break open" torque after the valve has been closed for several days may be three times higher than immediate seating and reopening torque. This is because the elastomer "cold flows" into the minute surface roughness of the disc edge.

Differential pressure forces also try to move the liner around the vane and thereby creating a greater downstream interference and higher opening torque.

Many valves operate to 150 psi maximum differential pressure, a few types reach 720 psi, and special designs go as high as 1500 psi.

Valve Designs That Can Effect A Large Differential Pressure

1) Externally Pressurized Elastomer Lined

You will note that the differential pressure values range up to 600 psi as found in Figure 42, Chapter 3.

TYPICAL MAXIMUM DIFFERENTIAL PRESSURE DATA FOR A MEDIUM DUTY VALVE DESIGN

Based on shaft strength and disc thickness when valve is operated by a normal spring diaphragm actuator.

Valve Size	C_V at 90° Open, $\frac{\text{Line Size}}{\text{Valve Size}} = 1.0$	MAXIMUM DIFFERENTIAL PRESSURE PSI	
		Unlined Closed / Open	Elastomer Lined Closed / Open
2"	90	300 / 300	150 / 150
6"	1150	200 / 21	150 / 21
12"	4600	130 / 8	100 / 8

Note:
(1) Higher differential pressure values may be obtained by heavier designs having a lower C_V value.
(2) Low differential pressure values in the open position are due to the high dynamic flow torque that is developed.

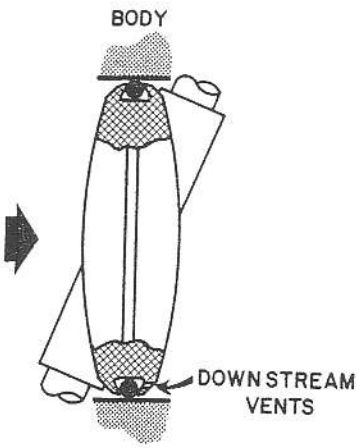


Figure 113. O-RING SEALED BUTTERFLY CANTED SHAFT DESIGN.

2) Line Pressure Energized Seal

The normal value is 150 psi differential pressure but if the design has a vented groove it may be 600 psi. The elastomer O-ring seal is located in the rim of the disc. Figure 113 shows the location of the vents.

3) Bonded Pressure Energized Seal

Figure 114 illustrates the canted disc design in which the upstream pressure seeps past the radial seal compression contact into the pressure equalizing hole in the seal. This forces the downstream portion of the seal in tighter contact with the seat ring. Valves with polyurethane seals of 95A Durometer degree of hardness have been used on 36 inch valve designs for hydraulic water service. Differential pressures were experienced up to 600 psi.

4) Uninterrupted Seal Canted Disc Valve

This type has a field replaceable seal that is held in a groove by a snap ring that is anchored to the body of the valve by a single lock nut. The disc is held in place by pins which do not project through the disc face and therefore preclude a leakage problem. Figure 115 is a design in which the seal is adjustable.

Large Butterfly Valves For Water Service

Large butterfly valves for water service are built to specifications of the American Water Works Association. This type of valve often has an offset disc but the seats are fully adjustable which can lead to improved sealability at any point around the disc circumference. Some designs utilize a narrow disc having a stainless steel edge to reduce the amount of buildup of scale deposits which can seriously tear a butterfly liner. Both the offset and canted disc valve designs eliminate the need of stem sealing in the liner. Liners used in this type of valve have a narrow cross-section. The higher seating pressure applied by a narrow edge disc, bearing on the rubber liner, yields a thorough wiping action to keep

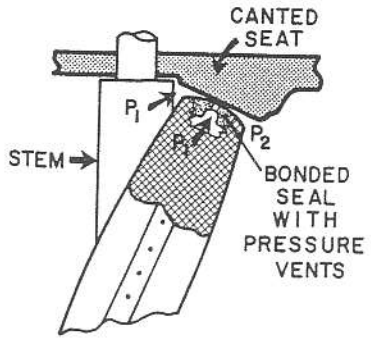


Figure 114. BONDED PRESSURE ENERGIZED SEAL WITH A CANTED DISC DESIGN.

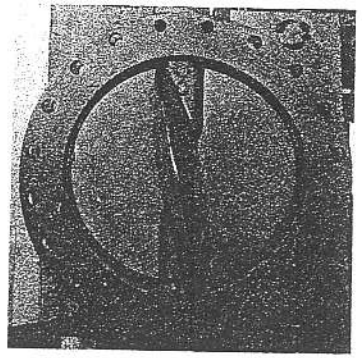


Figure 115. 360° ADJUSTABLE SEAL IN CANTED DISC DESIGN.
Courtesy of Mosser Industries, Inc.

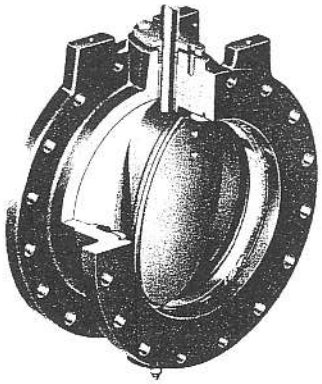


Figure 116. WATER VALVE WITH OFFSET DISC AND 360° ADJUSTABLE NARROW SEAT.
Courtesy of BIF, a unit of General Signal Corp.

the liner free from debris. The narrow seat in the valve illustrated as Figure 116 is replaceable without dismantling the disc and shaft assembly. Note that the seal adjustment screws are around the circumference. Flanged butterfly valve bodies are used in sizes of 24 inches and up, due to higher bending moments of the larger size piping.

Figure 117 is a drawing which illustrates how an adjustment can be made on the seal in a off-set disc valve.

The set screws force segments of a divided ring against the elastomer seal ring to raise its height above the body bore. The seal contacts the vane edge with greater compression along that particular arc segment of the divided ring.

A lighter weight, high-strength construction as shown in Figure 118, provides one sixth more free flow area than a solid disc of equal differential pressure rating. The valve has a narrow seat which is ribbed on its surface making a good seal to the valve body. It is internally filled with a specially compounded epoxy which is injected into a channel behind the seat. This retains the seat in the body and also uniformly provides disc-to-seat interference. Additional epoxy may be injected through the seal in the field with a portable injector kit. Teflon shaft bearings with non-metallic back-up liners plus the rubber seat effectively electrically insulate the disc, thus eliminating galvanic corrosion problems. This type of valve is manufactured in the 30-72 inch range.

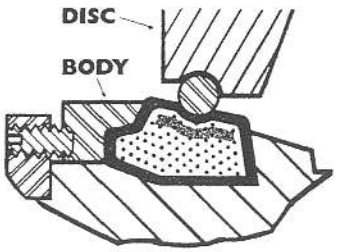


Figure 117. MEANS OF 360 DEGREE ADJUSTMENT OF SEAL WITH AN OFF-SET DISC.
Courtesy of BIF, a unit of General Signal Corp.

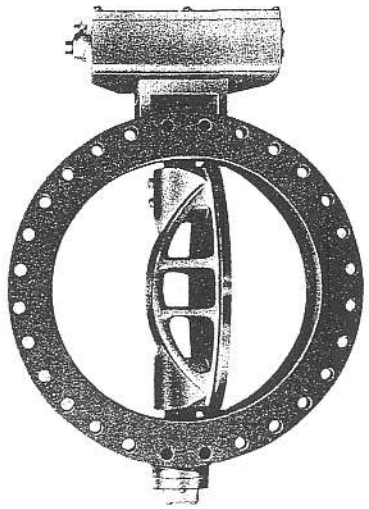


Figure 118. OFFSET RIBBED DISC FOR GREATER STRENGTH.
Courtesy Henry Pratt Company

Off-Set Vanes

Operating torque curves for valves which have offset discs differ from torque curves of valves having their shafts centered in the vane. See Figure 119. Note that when the shaft is downstream of the vane, the torque increases in a uniform manner from when the valve is closed to completely open, without peaking and falling off in the region of 70° as would be expected from a valve vane which is centered on the shaft. This provides the advantages of more stable control and higher flow capacities. However, these can be negated by the fact that the disc being offset from the shaft creates a greater obstruction in the flow-way. These valves seal equally well in either direction but for cold service it is best to place the shaft upstream due to the fact that the seat becomes harder at temperatures approaching -150° F. Rangeability is less for this type than for the valve with the disc centered on the shaft. (about 20:1) Figure 119 is a graph of Dynamic torque versus Disc rotation.

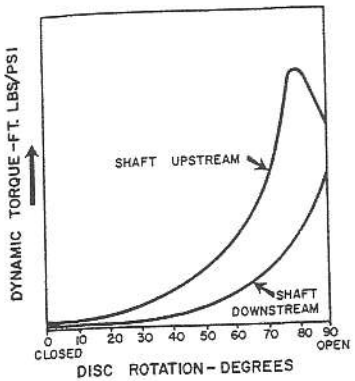


Figure 119. OFFSET DISC TORQUE.
Courtesy of BIF, a unit of General Signal Corporation

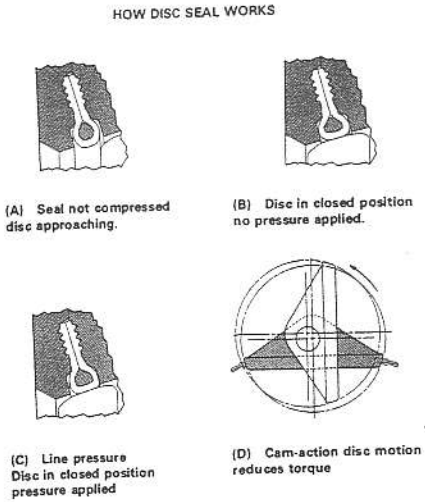


Figure 120. TFE SEAT RING IN AN OFFSET DISC BODY.
Courtesy of Hills McCanna Company

Operating Torque

Factors affecting torque are:

- Valve Diameter
- Shaft Diameter
- Coefficient of Bearing Friction
- Type of Liner Material (Seat)
- Differential Pressure on Shut-Off
- Fluid Velocity
- Type of Disc
- System Head Characteristics
- Piping Arrangement

These factors combine to make up the following classifications of torque:

- Seating & Unseating Torque, T_s (1)
- Bearing Friction Torque, T_b
- Dynamic Fluid Flow Torque, T_d
- Hydrostatic Torque, T_h (2)

Torque is related to so many factors, including the physical sizes relation of valve parts, that it is normal practice for each manufacturer to rate his own valves.

(1) Unseating torque for a dry gas valve increases when standing closed several days as compared with immediate reopening unseating torque. Torque with wet liners is about 30% lower than dry liners and wear life is much longer when wet due to reduced friction of the metal vane against the surface of a rubber liner wet by the line fluid or moisture droplets contained in a saturated gas.

(2) Caused by difference in static head of water on valve disc above and below the shaft in larger valves when the valve is in a horizontal pipe line.

TO CALCULATE THE REQUIRED TORQUE FOR BUTTERFLY VALVE ACTUATORS Using the AMERICAN WATER WORKS ASSOCIATION METHOD[1]

Use formulas (1) to (4) listed below to calculate each of the four classifications of torque previously listed. Use positive values of T_d when calculating opening torque, and negative values for closing torque. C_t and C_f vary with disc angle, disc shape, and method of shaft mounting. For these consult your manufacturer. Use formula (5) to calculate velocity. Then use the two formulas (6-a) and (6-b). The required actuator torque T_o will be the one having the higher value of the two equations.

1) $Ts = Cs\ D^3$

Cs = Coefficient of Seating Torque
D = Valve Diameter, ft

2) $Tb = 4.71\ D^2\ d\ f\ P$

d = Diameter of shaft, inches
f = Coefficient of Bearing Friction
P = Differential Pressure Across Valve, psi

3) $Td = Ct\ D^3 \Delta P$

Ct = Coefficient of Dynamic Torque

4) $Th = 3.06\ D^4$

5) $V = Cf\ \sqrt{\Delta P} = \frac{Q}{0.785\ D^2}$

V = Velocity, ft/sec
Q = Flow Rate, Cu ft/sec
Cf = Coefficient of Flow

6) Determine Valve Operator Torque from the Higher of these Two Formulas:

a) $To = (Tb + Ts + Th)$ at maximum ΔP in valve closed position

b) $To = (1.2\ Tb \pm Td)$ solve successively for several values of velocity using highest To determined

To = Operator Torque Requirement, ft/lbs

Dynamic Fluid Forces

Conservative pipeline velocities from 15 to 25 ft/second (maximum) for liquids and 15,000 to 25,000 ft/minute (maximum) for gases are safe limits for moderate temperature non-flashing fluids, straight flow and without reducers. Exceeding these limits may cause noise, vibration, cavitation, and poor control. Consult your manufacturer when in doubt.

Liquids:
$V = .0022 \frac{Q}{A}$

V = Velocity, ft/sec.

Q = Flow, gpm

A = Area of upstream pipe, sq. ft.

Gases:
$V = \frac{Qf}{A}$

V = Velocity of gas, ft/min.

Qf = Flow rate in cubic ft/minute (at flowing pressure)

Water hammer will occur on rapid closing in liquid service and pressure buildup could exceed the rated ΔP of the disc-shaft assembly and could cause damage. The minimum recommended closing time is:

$t = \frac{0.14\ Q}{A \times \Delta P}$

t = Minimum closing time, seconds

Q = Flow Rate, gpm

A = Area of upstream pipe, sq. ft.

ΔP = Maximum allowable pressure drop when valve is closed.

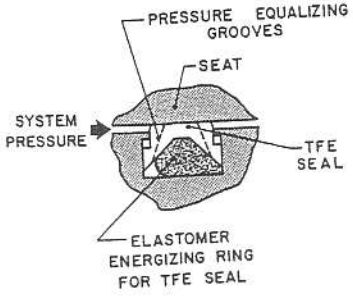


Figure 121. SLIPPER SEAL DESIGN.
Courtesy of Posi-Seal International, Inc.

[1]Reprinted from AWWA Standard C504-74 by permission of the Association. Copyrighted 1974 by the American Water Works Association, Inc., 6666 West Quincy Avenue, Denver, Colorado 80235

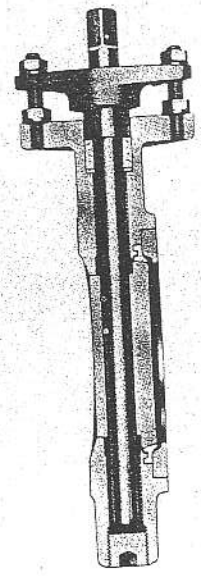


Figure 122. OFF-SET DISC WITH FLEXIBLE LIP SEAT DESIGN.
Courtesy of Jamesbury Corp.

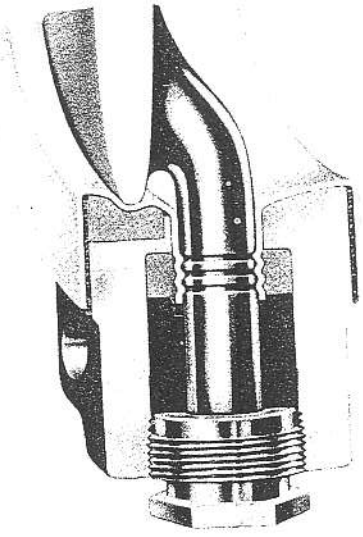


Figure 123. OFFSET DESIGN TFE LINED VALVE WITH LABYRINTH STEM SEAL.
Courtesy of Duriron Company, Inc.

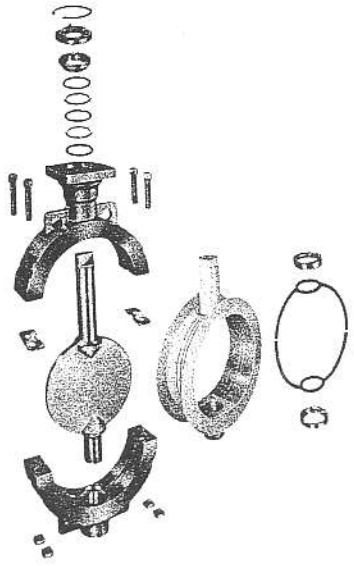


Figure 124. TFE LINED VALVE.
Courtesy of Tufline Division, Xomox Corp.

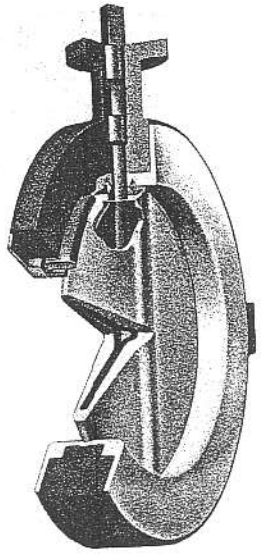


Figure 125. ALIGNED STEM AND DISC TFE LINED VALVE.
Courtesy of Garlock, Inc.

TFE Seated Valves

TFE seated valves are suited for moderately corrosive service. An offset stem design is required to give an uninterrupted seal. In Figure 120 the TFE is wrapped around an elastomer rubber ring and locked into the split body by grooves. The rubber core produces the resilience needed to offset the tendency of the TFE to be slow on recovery after opening, so as to shut off tightly on the next closure. This design has sealed to 720 psi differential pressure after 100,000 opening cycles.

The rubber O-ring is completely isolated from contact with the line fluid. The seal is self compensating for wear and thermal changes. An offset disc with cam action reduces torque and seal wear,

Another design consists of a TFE "Slipper Seal" installed over a resilient rubber O-ring. This design has operated successfully to 1500 psi differential pressure. Figure 121 illustrates the design.

A seat clamped between the valve body and ring is retained by the pipeline flange. The combination of lip seal action with the glass filled TFE seat permits thousands of bubble tight closures. Operating temperature ranges from -320 to 500° F and differential pressures reach 720 psi for the smaller size valves using this material. The special dry lubricated metal seats can operate to 700° F. Eccentric disc and offset shaft designs allow the disc to lift away and clear the seat within the first few degrees of rotation, thus eliminating uneven loading and therefore increasing seat life. The size of this type valve (Figure 122) ranges to 30 inches.

TFE Lined Valves

TFE lined valves are used for extremely corrosive service. The TFE completely covers the body and prevents contact with line fluid. A rubber element located under the TFE seat ring surface provides the resilience for repeated tight shutoff. Figure 123 illustrates a layrinth stem seal.

This type of valve can take a considerable amount of wear due to the offset disc clearing the seat very quickly (in a few degrees of opening) and does not continually contact parts of the liner as would a valve whose shaft is centered in the vane.

Another type of design uses a split body while shafts and disc are integral, allowing a heavy shaft for high differential pressure without blocking the flow area in the center of the disc. A triple continuous O-ring energizes the TFE liner which is heat stretched over the disc-shaft unit and heat shrunk. The body is split and is assembled with shims yielding proper liner interference with the disc. See Figure 124.

Fully TFE Lined Body And TFE Encapsulated Disc

(All wetted parts are TFE)

Figure 125 illustrates an aligned stem and disc valve design with the stem seals furnished by octagonal shaped TFE seal rings. This type of construction is available in sizes 2 thru 14 inches and operates to 150 psi in a temperature of -130° to 400° F. Silicone rubber is used to energize the TFE liner seal.

Three-Way Butterflies

By installing two discs in a tee body and connecting the discs with linkage to a common actuator, a three-way valve results. One disc opens as the other closes.

BALL VALVE TRIM

Introduction

Ball valves offer the advantages of highest flow capacity, low operating force for high pressure operation and tight shut off. The dynamic effect of high velocity flow in the solid ball design is to rotate the ball to a closed position; therefore, the actuators must counteract this effect. Floating ball designs seal tightly on the downstream seat but operating torque is from two to three times higher than for a trunnion-supported ball which seals upstream. This is due to the much smaller annular area of loading of the upstream seat. Seats in trunnion supported solid ball designs are spring-loaded and pressure-energized and account for more than one half of the operating torque; trunnion bearings plus the stem thrust bearing and the packing friction account for the remainder. This seating load is encountered in use for throttle flow and the actuators generally require positioners. Very large size valves may have pressure retractable seats to reduce the seat-wear and operating torque when throttling. The ball port may be specially shaped to improve the low-flow characteristic. Some balls are only segments of a spherical shell with a flow characterized "vee cut." They are effectively used to prevent dewatering of paper pulp mixtures. Other designs employ an eccentric-mounted, spherical surface to remain free of seat friction torque until the moment of closure.

Balls may be hard-chrome plated and polished or may be either steel or iron-coated wih a few ten thousands of an inch of TFE which remains in the valleys of the surface roughness for thousands of cycles at full pressure drop service. Breakaway time-set torque is greatly reduced by the use of both designs, because the non-metallic seal is prevented from entering and binding in the rough surface. Seat seals are usually TFE, nylon, rubber, or other elastomers and, in large valves, are encased in metal seat rings, which in turn are sealed to the body by a separate elastomer seal. Metal-to-metal seating is used in some high temperature valves. At temperature service below 600° F a viscous sealant and lubricant may be injected into the seat joint to give bubble-tight sealing. Bearings may be of either special low friction sleeve material or cylindrical roller bearings.

Trunnion-supported ball design allows "double block and bleed," that is, shut-off of both the upstream and the downstream seats simultaneously. The body cavity may be bled to check seat tightness, replace stem packing, or to prevent the mixing of fluids in a manifold if seat leakage occurs.

Characterized balls are built in varying sizes up to 16 inches which must operate at more moderate pressure drops than do full balls, which have been built as large as 72 inches for 1000 psi differential pressure throttling service in giant water and hydroelectric systems.

Characterized Ball Valves

The ball is a partial spherical shell supported by trunnions as illustrated in Figure 126.

The ball can be splined or keyed to the driving trunnion and is free to slide along the trunnion's axis to align with the seat ring. The ball surface is a 120° segment of a spherical surface so that it fully opens in 90°. The seat ring is either a thin metal or hard plastic flat ring which is forced slightly against the ball by the seat retainer. Flow must be in the correct direction so that differential pressure always forces the seat in contact with the ball to effect a pressure energized seat. Reverse flow at sufficient differential pressure will permanently damage the ring by bending it away from the ball. A back-

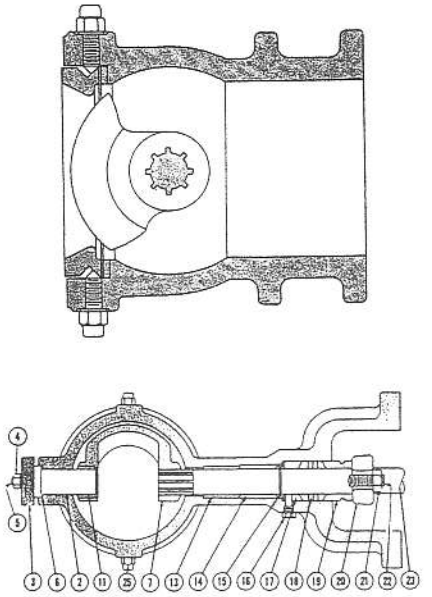


Figure 126. CHARACTERIZED BALL VALVE.
Courtesy of Masoneilan International, Inc.

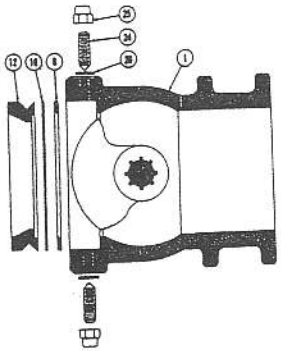


Figure 127. SEAT RETENTION.
Items are: 12-Orifice Ring, 10-Seat Ring, 8-Back-up Ring.
Courtesy of Masoneilan International, Inc.

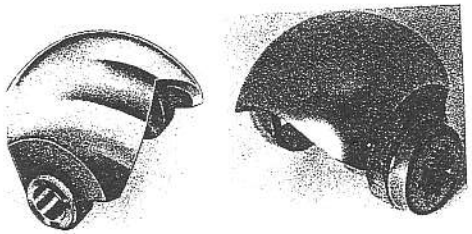


Figure 128. BALL EDGE SHAPES.
Left: *Courtesy of Fisher Controls Company.*
Right: *Courtesy of Masoneilan International, Inc.*

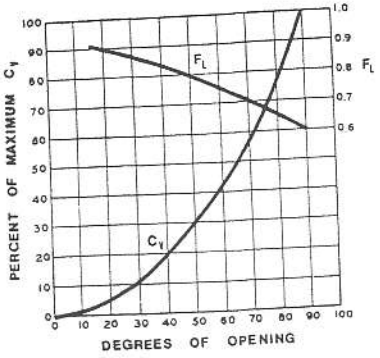


Figure 129. FLOW CHARACTERISTIC FOR PARABOLIC BALL VALVE.
Courtesy of Masoneilan International, Inc.

up ring under the seat on the ball side, is used for higher pressure valves to prevent buckling of the thin flat seal ring. The seal ring is held in place by a tapered flow orifice ring which may be bolted to the flange or held in by cone-pointed set screws passing through the body wall to engage the ring on its circumferential "V" groove. Care must be taken during the installation to insure that the mating pipe flange uniformly contacts the seat orifice retaining ring. Figure 127 illustrates how the seat is retained. All internal parts are readily field replaceable.

The relatively sharp edge of the ball creates a shearing action as the line of seat contact progressively crosses the ball face on closure, thus a self-cleaning action is assured making the valve an excellent performer for handling paper pulp solutions.

This type of valve ranges from 2 to 16 inches in size in pressure classes 125, 150 and 300 lb. ANSI. Small sizes are also built in 600 lb. ratings. With proper selection of body and seat ring, the temperature range is -250 to 750° F. Differential pressure limits range from 720 psi for the 2 inch to 275 psi for the 16 inch valve.

The ball may have a parabolic shaped edge or have a "V" notch as shown in Figure 128. The V-ball may be reversed to present a "U" ball with higher flow capacity.

The flow characteristic is a modified equal percentage curve having good low flow control as shown in Figure 129. The ball surface is highly polished hard chrome plate on a 316 stainless steel base metal. This plating is harder than sand and ball erosion is thereby reduced.

Seat Ring

The flat metal seat is hardened for resistance to deflection. The edge coming into contact with the ball is slightly coined to present a smooth line contact. Figure 130 is a drawing of the seat to ball contact area. Sealing is generally in the flow direction shown, but polyurethane seals may be used for bi-directional sealing but with lower differential pressures (40 to 50%) when the flow is opposite to normal.

Erosive Service

Erosive service is accommodated by having a close fitting hard-faced orifice ring and eliminating the seal ring. Rangeability is reduced to 100:1 and shut-off leakage is higher. For severe service an internal liner may be installed which protects the ball and body surfaces. See Figure 131 for positioning of the internal liner.

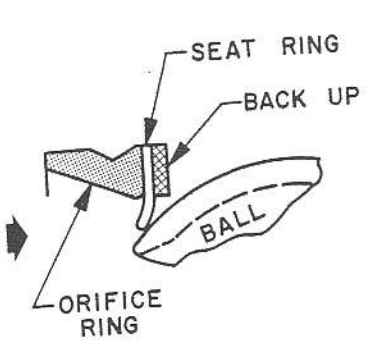


Figure 130. SEAT TO BALL CONTACT.
Courtesy of Masoneilan International, Inc.

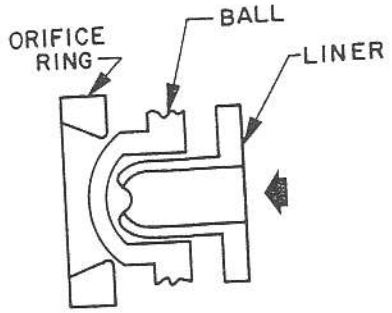


Figure 131. INTERNAL LINER FOR EROSIVE SERVICE.
Courtesy of Fisher Controls Co.

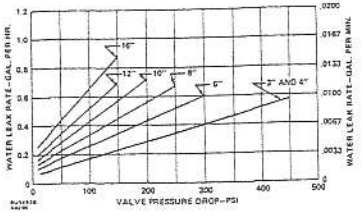
Materials of Construction

Temperature	Body	Ball	Trunnion	Bearing
-20 to 400° F.	Cast Iron	316 & 317 Stainless Steel Hand Chrome Plated	17-4 PH	440-C Stainless Steel (hardened) TFE Loaded Material
-20 to 400°F.	Ni-Resist (1)	"	"	"
-20 to 600°F.	Carbon Steel	"	"	"
-460 to 700°F.	316 Stainless Steel	"	"	Stellite #6

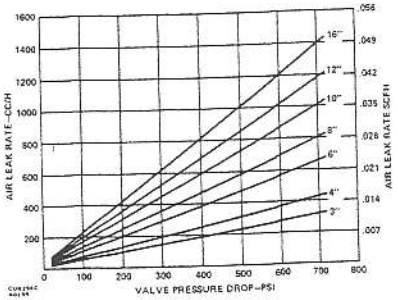
Seat Components

Temperature	Service	Seat Ring (2)	Back-up Ring	Orifice Ring
-460 to 600°F.	General	316 Stainless Steel	316 Stainless Steel	Same as body
-250 to 150°F.	"	Delrin®		
-250 to 500°F.	"	Vespel®	"	"
0 to 150°F.	Erosive & Bi-Directional Seal	Polyurethane (3)	"	"
750°F. (Max.)	Very erosive	None	"	"
140°F. (Max.)	Food	Polycarbonate Lexan®(4)	"	"

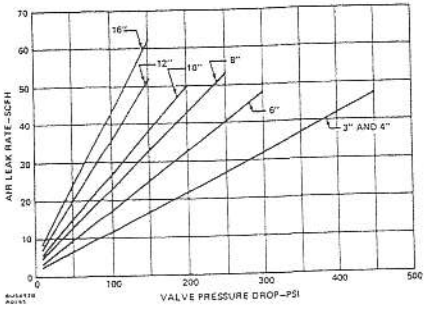
(1) Trade Name of International Nickel Company.
(2) Inconel is often used for general service to provide a tougher seat material. The thin ring is easily damaged so it must be handled carefully when servicing the valve.
(3) Do not use in hot water or saturated steam service.
(4) Withstands steam cleaning to 250°F. for short periods.



LIQUID LEAK RATE
Also used for Saturated Steam



AIR LEAK RATE
Plastic Seal Ring



AIR LEAK RATES
Metal Seal Ring

Figure 132. BALL VALVE LEAKAGE RATES WITH DISC SEALS.

Courtesy of Fisher Controls Co.

Sealing

Metal seat ring leakage is .01% of rated Cv. Plastic seats may shut off "drop tight" to liquids but leak slightly to gases. Maximum leak rates are shown in the following graphs, Figures 132.

Flow Capacity

Flow capacity is limited by the 80% of line area in the orifice. Reduced capacity orifices and seat rings are available from a few manufacturers, but reduction is limited due to axial space in the body.

Cavitation

The critical flow factor varies from 0.9 at low flow to 0.6 at the full open position. The valve is subject to cavitation and therefore pressure drops must be limited when the valve is open all the way. Since the valve offers low flow resistance it can pass a substantial flow before this occurs.

Rangeability

The rangeability is 350:1 for the standard seats and 100:1 for an erosive service orifice without a seal ring.

Dynamic Fluid Forces

Dynamic fluid forces always tend to close the ball valve, however seat and bearing torque tend to exceed the dynamic torque valve until the ball is about 75 degrees open.

FULL BALL VALVES

Full ball valves have a solid ball and round flow port, either line diameter or reduced. Two designs are available; one has a floating ball that is forced against the downstream seat by differential pressure to shut off the flow. The other has a trunnion-supported ball. Two trunnion ball valve designs are available. One has a single seat upstream which is pressure energized. The other has two seats, and again, the upstream seat is pressure energized; thus it is a bi-directional flow valve. The floating ball design gives the most reliable tight shutoff but has the highest torque, sometimes 2 to 3 times the value for the trunnion design. Floating balls are made in sizes ranging up to 36 inches with differential operating pressures to 1440 psi. Trunnion balls can be as large as 72 inches with 1000 psi WOG[1] differential pressure for the 72 inch size. Pipeline valves operate to 1440 psi differential pressure. Temperature ratings are often limited to 400°F. by the elastomer seat ring seals; but spring energized seats are available for higher temperatures.

(1) Refers to water, oil, or gas pressure rating at 100°F.

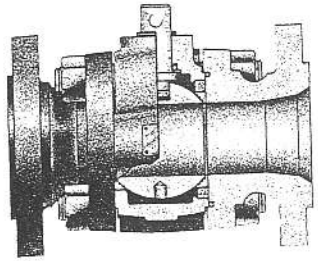


Figure 133. FLOATING BALL DESIGN.

Courtesy of Demco, Inc.

Floating Ball Valves

Floating ball valves are quite popular in the smaller sizes of ¼ in. to 4 inches with pressure ratings of 150 to 2500 pounds. They are also manufactured in sizes up to 36 inches with 600 pound ratings for pipeline and compressor station service. The ball has a key slot or splined fit on the drive shaft. Figure 133 illustrates a floating ball design.

Body styles vary from a three-piece bolted construction to the two-piece design for large sizes as illustrated in Figures 134, 135 and 136.

Some styles have a top entry by means of a bonnet flange. Others have the body split diagonally. Figure 137 is an example of a top entry style.

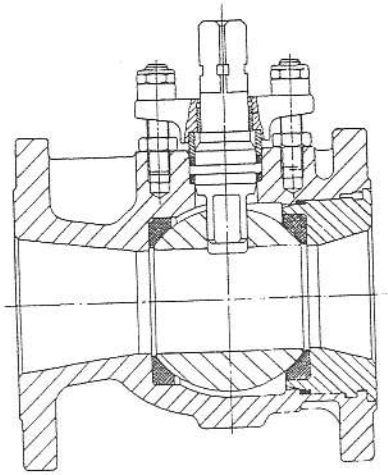


Figure 134. TWO PIECE BODY.

Courtesy of Jamesbury Co.

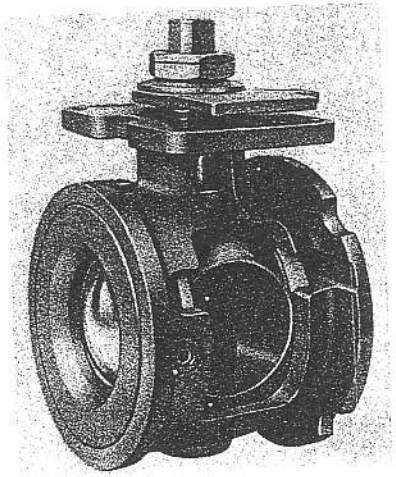


Figure 135. FLANGELESS BALL VALVE.

Courtesy Worcester Valve Company

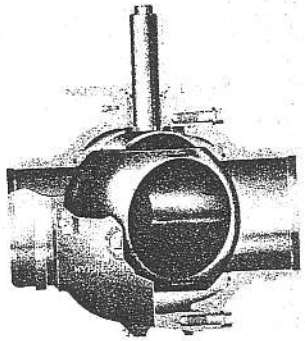


Figure 136. TWO-PIECE BODY FLOATING BALL DESIGN.

Used in Larger Valves.
Courtesy of Rockwell Manufacturing Co.

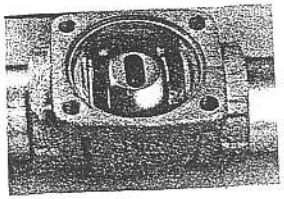


Figure 137. TOP ENTRY STYLE.

Courtesy of Hills McCanna Company

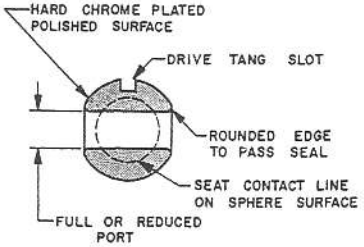


Figure 138. FLOATING-FULL BALL VALVE DESIGN.

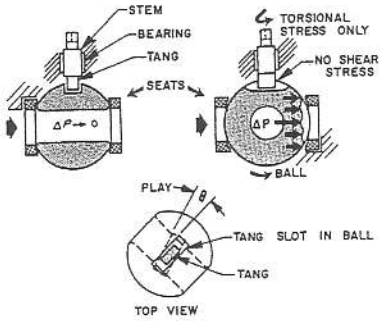


Figure 139. TYPICAL FLOATING BALL STEM.

Floating Ball Design

The floating ball design consists of a solid ball having a drive tang slot on the top side. Figure 138 is an example of a floating ball design.

The Drive Stem

The drive stem is subject to torsional stress only from frictional load of the ball against the downstream seat ring. This load is created by differential pressure on the ball-to-seat joint area. The ball is free to move along the tang slot while in the closed position and thus does not transmit load to the stem when the ball loads and deforms the downstream seat ring as differential pressure builds up on closure. Fine throttling may be prevented by the freedom of fit of the tang in the ball slot. Once the actuator has moved the ball to a given opening, the dynamic fluid forces tending to close the ball may actually move it towards the closed position a few degrees. Deadband is wider when actuator motion is reversed because of the clearance of mating parts. Figure 139 shows a typical floating ball stem valve.

Seat Loading

Seat loading takes on a fairly high value due to the differential pressure acting on the area of the line of seat contact. This will load the ball on whichever seat happens to be downstream. Some seat distortion can occur from this heavy type of loading, but it is restricted to the surrounding body bore. The downstream seat is always in wiping contact with the ball and by this means maintains a

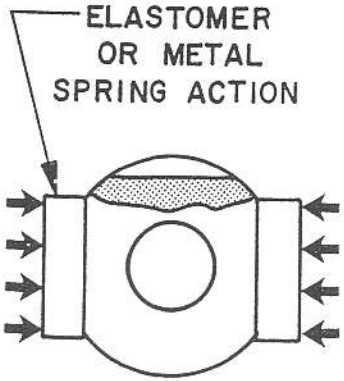


Figure 140. SEATS CONTACTING BALL.

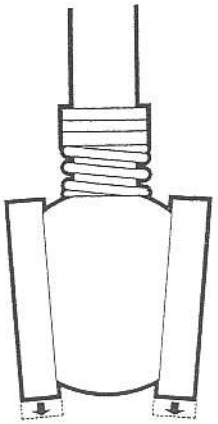


Figure 141. BALL SPRING-LOADED INTO SEATS.
Courtesy of Hills McCanna Company

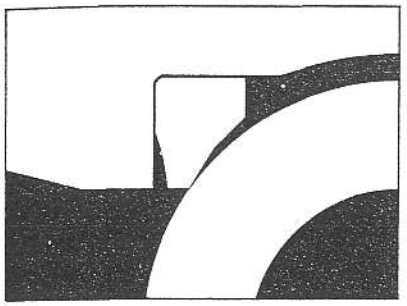


Figure 142. LIP LOADING SEAT.
Courtesy of Jamesbury Corp.

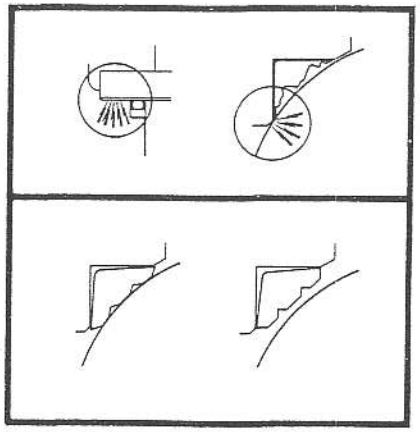


Figure 143. SERRATED SEAT RING.
Courtesy of Demco, Inc.

seating mostly free of foreign material from the line fluid. Figure 140 illustrates how this is accomplished.

Some top entry valve types are constructed in such a manner as to spring load the ball into both seats as shown in Figure 141.

Elastomer Seats

Elastomer seats are manufactured in two principal patented designs shown in Figures 142 and 143. The ball is supported by the seat lips which effect a tight shutoff at low differential pressures. As differential pressure increases, the lips can bend which allows the entire seat to support the ball load created by the differential pressure acting on the seat-to-ball joint circumscribed area. The serrations of the second design allow foreign paricles to collect in the gaps rather than become embedded in the seat.

For high pressure service, seats made from TFE are often encapsulated in stainless steel rings, to prevent cold flow, as shown in Figure 144.

This design is fire safe, in that, on burn-out of the TFE material the metal housing ring contacts the ball face and provides a reasonably tight seal. Valve fire proof testing is often required for LPG fuel service.

Non resilient seats are used when the temperature limits of TFE will be exceeded. Carbon - Graphite seats are used if the temperature doesn't exceed 1000° F. and work best in clean liquids with some lubricting properties. They are not bubble-tight in air or gas service and are not recommended for steam service. They should not be used with strong oxidizing agents, and if used in an oxidizing service, the temperature should be limited to a maximum of 700° F.

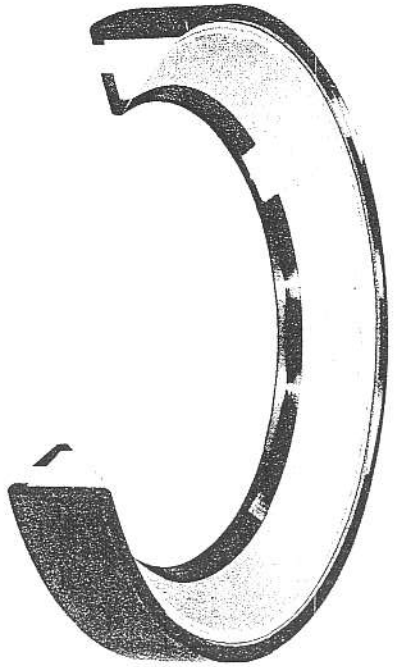


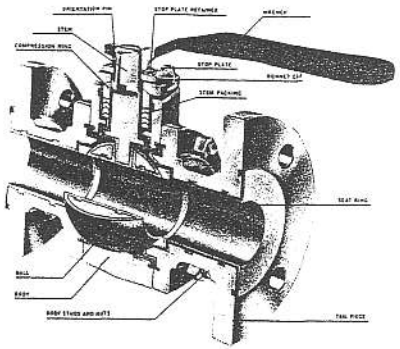
Figure 144. ENCAPSULATED IN METAL RING.
Courtesy of Hills McCanna Co.

Flow Capacity

The flow capacity of a full ball valve is nearly that of a straight pipe section for the full port design, since both seats are in contact with the ball and there is no flow into the body cavity when the valve is in its maximum open position. Pressure drop should be limited to prevent cavitation as with the characterized ball type.

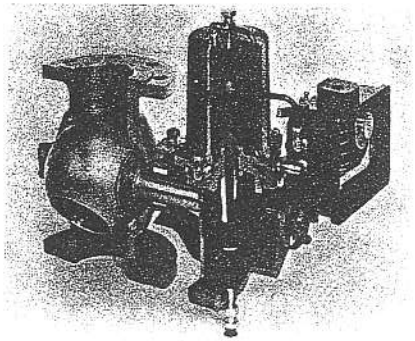
Valves For Corrosive And Abrasive Service

Valves to be used for corrosive service may be elastomer or plastic lined, including the ball and stem. Rubber and some types of plastics may be bonded to the metal parts for high pressure service. Other designs use a completely molded plastic (polyvinyl chloride) valve. Ceramic and glass balls and bodies are often used in varying sizes up to 4 inches. The TFE liners must be locked in place in dovetail grooves as illustrated in Figure 145a. These valves are available in sizes from 1 inch to 10 inches.



(a) TFE LINED BALL VALVE USED FOR CORROSIVE SERVICE.

Courtesy of W-K-M Division of ACF.



(b) VALVE WITH A STELLITE SEAT RING CONSTRUCTION USED FOR ABRASIVE SERVICE.

Courtesy of Kamyr, Inc.

Figure 145. VALVES FOR CORROSIVE AND ABRASIVE SERVICE.

Valves designed specifically for severely abrasive service (as shown in Figure 145b) are constructed with a hardened ball surface, integral ball-stem, and solid stellite seat rings which allow operation to 1100°F with reasonable life in the following services:

Iron Ore Dust
Fiberglass, Rockwool, etc.
Crushed Limestone
Sand and Gravel
Catalytic Cracker Bottoms
Fly Ash Slurry
Mining Tailings
Wood Chips

The ball-stems are of integral construction to prevent erosion and loss of strength at their junction. The greater cross sectional area of the joint over that of a drive tang or pin attachment allows for higher torque operation. Although the ball-stem is integral, the ball is not supported by the stem type trunnion. The ball floats downstream under differential pressure loading in the closed position, loading the seat ring against the seat bore recess in the valve body to achieve closure. This motion is only a few thousandths of an inch in length while the stem moves slightly in the packing gland to allow this ball movement. However, the horizontal stem motion is not sufficient to create a leaky gland. The integral construction eliminates the slight deadband in control resulting from loose joint design.

The body is split perpendicular to the pipe axis and in the plane of the stem diameter in order to allow assembly with the integral ball-stem construction. These surfaces are gasketed and are made up against a positive stop with shims to achieve proper adjustment of the ball for free turning and seating.

For service to 450°F, PTFE seat inserts may be used. Fire safety is provided by having the PTFE insert soften and allow loading of the ball against a narrow raised portion of the metal seat ring. For operation to 1100°F, solid stellite seat rings are used. Service above 1100°F approaching 2000°F may be obtained by using Iconel® and other exotic alloys for both body and ball.

The valve is self-cleaning on trashy service, provided the stem is installed in the horizontal position. The actuator must be attached so that upon opening, the ball rotates such that the topside of the ball on the upstream side opens first to the inlet flow. The flow thus scours trash out of the pocket on the downstream side between the ball and body wall while the turning ball helps to lift large debris. Valves of this design have successfully flushed out stones, scrap material, and sand.

Plastic Valves

Polyvinyl Chloride (PVC) and Fiber Glass valves are available for chemical and corrosive services. They are priced about one third less than stainless steel and a fraction of high alloy valve prices. They are compatible with PVC and fiber glass piping. Valve sizes range from 2 to 10 inches having 150 pound ANSI pressure ratings and a temperature range of -40 to 200°F. TFE seat rings and stem seals are standard. Some designs incorporate continuously wound glass fibers for added strength. TFE and Molybdenium may be compounded with the ball surface material for increased lubricity and lower break away torque. These valves should not be subjected to severe mechanical impact. Consult each manufacturer for specific material resistance to chemicals. The Filament wound fiber glass ball valve is shown in Figure 146.

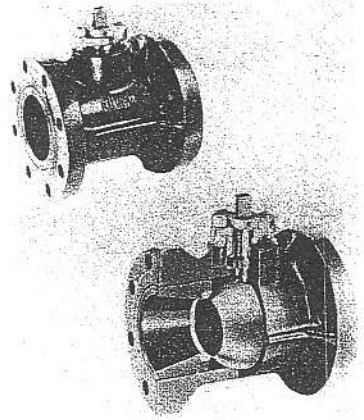


Figure 146. FILAMENT WOUND FIBER GLASS BALL VALVE.

Courtesy Rheem Fluid Systems, Inc.

Materials Of Construction

Materials of construction exist in such a wide variety that only a few commonly used valves are listed.

MATERIALS

Body	Ball	Stem
Bronze	Stainless Steel	Stainless Steel
Aluminum	Aluminum	Aluminum
Iron	Stainless Steel	Stainless Steel
Steel	Stainless Steel Chrome Plated Steel TFE Coating on Steel	Steel or Stainless Steel
Stainless Steel	Stainless Steel	Stainless Steel
PVC*	PVC	PVC

*Polyvinylchloride

SEALS

Material	Max. Temp. Range
TFE	400°F.
Kel-F	200°F.
Nylon	200°F.
Polyurethane	150°F.
Rubbers	300°F.
Carbon (Graphite)	1000°F.

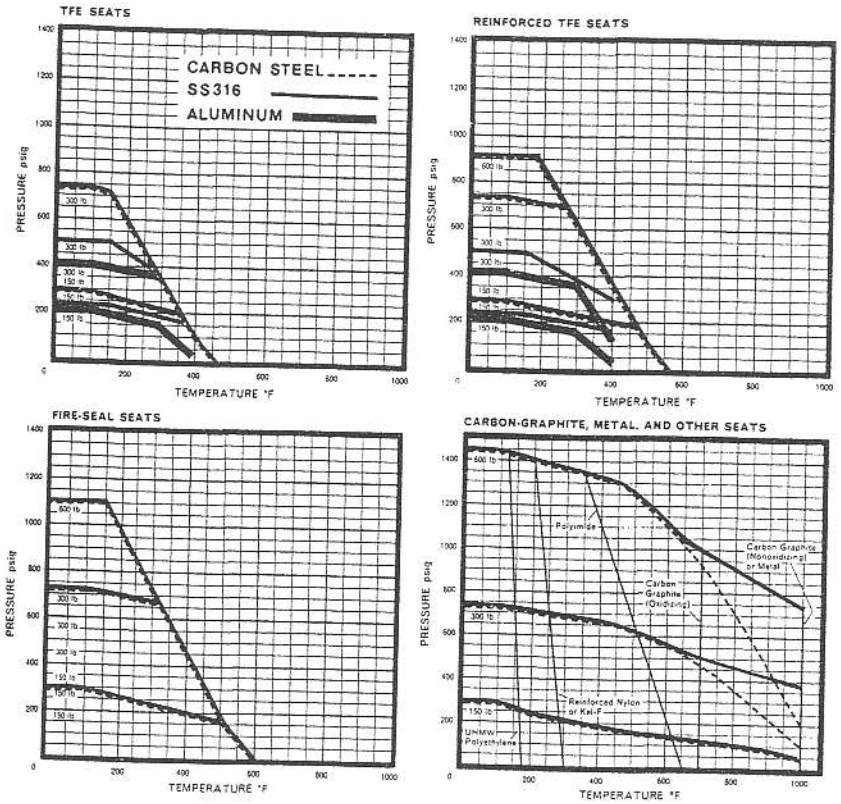


Figure 147. TYPICAL PRESSURE VERSUS TEMPERATURE GRAPHS FOR DIFFERENT SEAT CONSTRUCTIONS AND MATERIALS.

Courtesy of Hills McCanna Co.

Pressure Ratings

Pressure ratings are dependent upon temperature and selection of seal materials as illustrated in Figure 147.

Valve Torque Factors

Valve torque factors depend upon:

1) The ball load which is created by differential pressure acting on the area circumscribed by the seat-to-ball sealing diameter, the angle of seat contact with the ball, and the varying radial torque arm distances of each point on the seat from the valve stem center. These are explained in more detail in the Trunnion Ball section later in this chapter.

2) The vertical stem thrust from the line pressure acting on the stem thrust bearing.

3) The stem packing friction.

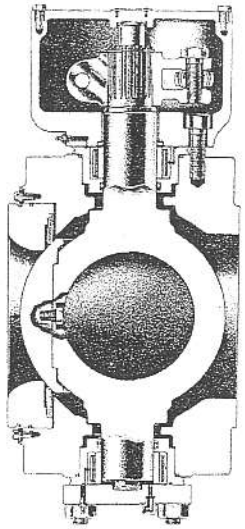


Figure 148. INTEGRAL TRUNNION - SPLIT BODY VALVE DESIGN.

Courtesy of Fisher Controls Co.

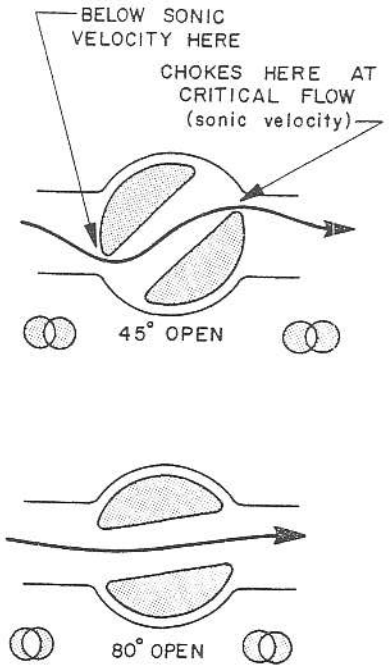


Figure 149. BALL FLOW PATHS.

Single Seal Trunnion Supported Ball Valves

Single seat trunnion supported ball valves for upstream sealing may have the trunnions integral with the ball which requires a split body design to assemble around the ball-trunnion component as shown in Figure 148. This type of design provides a rigid drive to position the ball for fine throttling and in this respect gives better performance than the floating ball which has a loose fitting tang driving the ball. Balls may be faced with replaceable caps which cover the surface which comes into contact with the seat ring. This permits replacement of the ball wear surfaces, which is important if the valve is exposed to an errosive substance.

Flow Capacity

Full ball valves present two orifices in series as illustrated in Figure 149. The limiting factor for gas flow occurs when the gas reaches sonic velocity in the vena contracta area. For globe style valves this is usually when $\frac{P^1}{P^2} = 0.5$, but in ball valves this occurs around $\frac{P^1}{P^2} = 0.8$. The difference in ratios is due to the greater pressure recovery downstream for the ball style valve.

Because of the different flow patterns in a ball valve, conventional methods used in sizing globe valves cannot be used. This type of valve can closely approach the characteristics of a globe valve in the throttling range and has the advantage of high capacity and low pressure drop. Figures 150, 151 and 152 help to explain the relationship between ball rotation and percent of flow.

Past 10° open, the flow characteristic is equal percentage and flow is controlled beginning about 5° open, but this varies with seal overlap on the ball from one manufacturer's valve to another.

Although the flow area changes almost uniformly with the valve opening, the flow does not increase uniformly. This is caused by the flow not having to turn as much as the valve approaches the full open position.

Dynamic Fluid Forces

The flowing fluid always has a tendency to close the valve. If the fluid attains a high velocity, the actuator must have a safety fea-

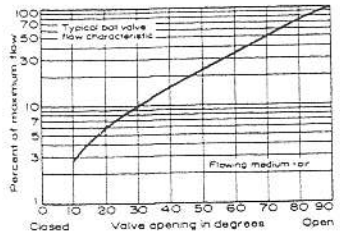


Figure 150. BALL ROTATION VERSUS LOG PERCENT MAXIMUM FLOW.

Below choked flow conditions.

Courtesy of Fisher Controls Co.

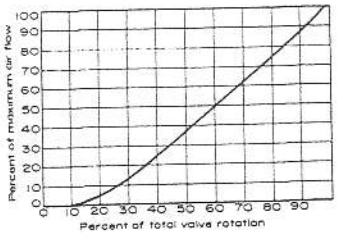


Figure 151. BALL ROTATION VERSUS PERCENT OF FLOW.

Courtesy of Fisher Controls Company

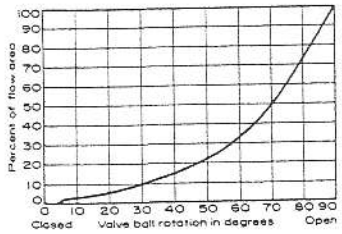


Figure 152. BALL ROTATION VERSUS PERCENT OF MAXIMUM AREA.

Courtesy of Fisher Controls Company

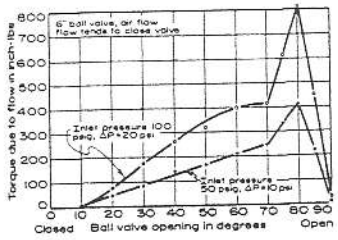


Figure 153. DYNAMIC CLOSING TORQUE VERSUS DEGREE OF OPENING.

For a Single Seated Full Ball with Trunnions.

Courtesy of Fisher Controls Co.

ture so as not to let the valve slam shut and create a severe water hammer. A fully open valve exerts no torque. Figure 153 is a graph relating dynamic closing torque with degree of opening of the valve.

Double Seated Trunnion Ball Valves

These types of valves are primarily used for pipe line, compressor stations and gas processing plant service; also on oil and petroleum products lines. They frequently throttle gas and liquids and a tight shutoff is required for block valve service. They are known as through conduit type, in that when the valve is in the open position, the body is isolated from the line to prevent particles from accumulating in the body cavity and jamming the valve operation. The flow port is full size and round in shape to accommodate the passage of line cleaning devices and to protect the seats from damage. In either the completely open or closed position, the body cavity may be bled of pressure as a means of checking seat tightness. For manifolds that carry several fluids this can provide a positive means of fluid isolation to prevent mixing should a valve leak occur. Since the body may be bled to atmospheric pressure while in service, stem packing replacement is an important feature. A viscous sealant backup system is often provided to give bubble tight sealing at high differential pressures and also to lubricate the seat-ball contact for lower torque operation.

Valve sizes range from 8 to 42 inches with pressure ratings of ANSI 300, 400 and 600 pounds. They are required to operate from cold start-up arctic temperatures of -90°F. to hot compressor station gas discharges of nearly 300°F. They are designed to operate to differential pressure values equal to their pressure ratings.

Valves can be placed under severe conditions. One major gas transmission company accepts new valve types only if they seal after blasting sand, pea gravel and welding slag through them on high pressure differential

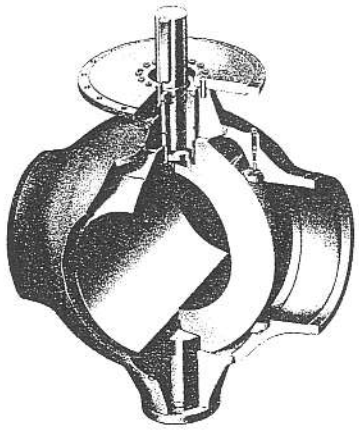


Figure 154. ALL WELDED BALL VALVE CONSTRUCTION.

Courtesy of Rockwell Manufacturing Co.

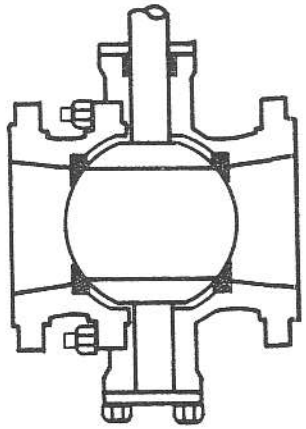


Figure 155. SPLIT BODY FLANGE TYPE VALVE.

opening and closing tests. Elastomer seat seals must be securely retained against blowout and the metal seat ring housings strong enough to cross the ball port unsupported at high differential pressure loadings without bending. The seat joint temperature will drop immediately to -100° F. when initial opening of a 600 pound valve to compressed air at 1500° F. is performed. In that this kind of severe thermal shock will usually destroy any elastomer bond to the metal ring some type of mechanical retention is required.

Body Styles

Body styles are: all welded, center flange, and three piece bolted, as illustrated in Figure 154, 155 and 156 respectively.

Ball Designs

Some balls have integral trunnions in an all welded body construction; however most designs have loose trunnions. To withstand the high differential pressure forces (equal to full pressure ratings) the ball must be of a well constructed type. It may be solid or ribbed, and some have the sealing area raised to reduce the amount of seat contact, seal wear and torque. The ball surface may be hard chrome plated and highly polished or it may have a TFE coating only a few ten thousandths of an inch thick. The TFE layer greatly reduces friction even though most of it is blasted away on the first few high pressure openings. The TFE will remain in the surface

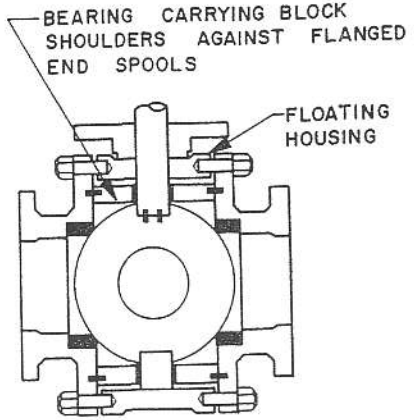


Figure 156. THREE PIECE BOLTED BODY VALVE.

Courtesy of Grove Valve & Regulator Co.

roughness of the ball for thousands of full differential pressure cycles maintaining its initial low torque. (1500 psi differential pressure and 5000 cycles before significant torque increase). TFE is also transferred to the seat ring and that thus prevents the elastomer seat from deforming into the ball roughness and yielding a high time-set torque. Salt bath

corrosion and valve operating tests have shown that the corrosion of the partially TFE covered surface is soft and the wiping action of the seat against the ball easily shears away corrosive products.

Bearings

Bearings are often TFE compounded metals quite capable of withstanding and operating with unit loads to 30,000 psi. Design loads are not usually this high but due to trunnion deflections under heavy differential pressure loading, the bearing contact "spot-loads" do reach these amounts. The trunnions, subject to shear and torsion, are massive in size.

Seat Rings

Upstream seating is accomplished by means of both a spring and a pressure energized flexible metal seat ring housing. Under high differential pressure loading, the ball becomes egg-shaped and its surface becomes distorted. A flexible ring can be provided to follow this ball distortion and maintain the seal. Since the metal ring cannot completely follow the ball, relaxation of an elastomer seal, still under a high compressive stress, is utilized. Belleville springs are used to help keep the seat in contact with the ball for initial sealing at low differential pressure. The ring is also sealed to the body, often with an O-ring. Seat designs are shown in Figure 157 and 158.

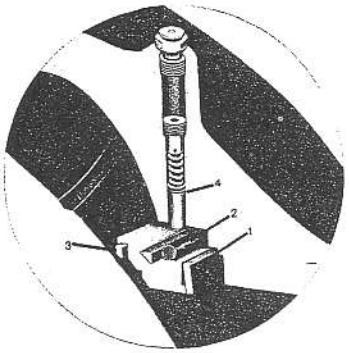


Figure 157. TYPICAL UPSTREAM SEATING.

Note:
1) Belleville Spring for Initial Seal Contact
2) O-Ring Sealing Seat Ring to Body
3) Hard Elastomer Seal Ring Contacting Ball
3) Sealant Back-up Injection System

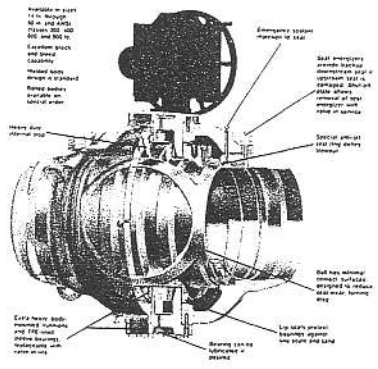
Courtesy of Rockwell Manufacturing Company

Upstream Seating

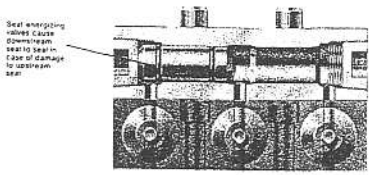
The metal seat ring of Figure 157 is individually lapped to its own ball providing a wide seating surface. The primary seal is a pressurized nylon seal clamped into the metal seat ring; it acts independently of the metal seat.

Double Seat Downstream Valve

A second seat downstream valve design is illustrated in Figure 158. It simply brings a back-up seat into use if the upstream seat leaks. The buildup of pressure in the ball cavity causes the spool pilot valve to shift and places upstream pressure against the back side of the seat to force it against the ball. Annular seat pressure loading areas are carefully selected to allow this action to take place.



(a) Sectionalized valve with pilot valve feature.



(b) Pilot valve.

Figure 158. BALL VALVE WITH DOWNSTREAM SEATING FEATURE.

Note: Ball Surface is Recessed to Reduce Seat Torque Component as Ball Opens.

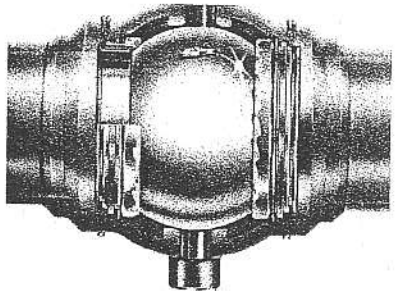
Courtesy of W-K-M Divison of ACF

Rotary Seats

One manufacturer offers valve seats which are rotated a few degrees with each valve opening. Ratchet steps on the side of the seat facing the ball are moved one ratchet each time the ball approaches the full closed position by a pawl on the ball. Ball valve seats suffer their most severe wear at the point where the seat is cracked open and by changing this position wear is distributed around the entire seating surface giving it longer life. This feature works very well on moderately clean fluid but for extremely dirty fluids, seats often become jammed and the pawl may be destroyed. Figure 159 illustrates how this system works.

Replaceable O-Ring Seats

Replacable O-ring seats are illustrated in Figure 160. A series of helical springs effects the initial seat to ball contact force. The ball-to-O-ring seal is retained in a "dove-tail" groove which is vented downstream, thus the seal is retained even at high differential pressure opening values equal to body pressure ratings.



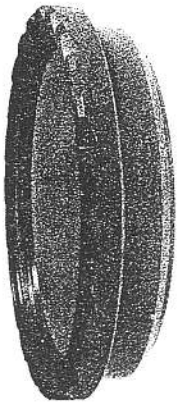


Figure 159. ROTATING SEAT INCREASES WEAR LIFE OF VALVE.

Courtesy of Cameron Iron Works, Inc.

Fire Safe Design

In the event of a fire, if the valve is closed, the burn-out of the seat-to-ball seal lets the metal seat contact the ball for a reasonably tight closure. The seat ring pressure loading on the annular arc of the seat holds it against the ball. Liquified petroleum gas (LPG) service valves must often pass a fire test.

Valve Body Cavity Pressure Relief

Valve body cavity pressure relief is necessary in liquid service to prevent the thermal expansion of liquid in a closed valve from causing damage or torque that is so high the valve cannot be opened until the pressure rise is bled off. The seat areas acted upon by upstream pressure are designed to relieve the pressure automatically to the side of lowest line pressure. Lockup of valves in liquid service has not been uncommon in the past before this type of design was produced.

Seat Retraction

Valve seat retraction is sometimes used to lower the torque and reduce seat wear in the throttling service of very large valves. The valve operating mechanism triggers a pilot valve to vent the pressure downstream which energizes the upstream seat and allows upstream pressure to actually cause a reversal of the thrust on the seat. In actual tests, 30 inch valves have been opened at 1000 psi differential pressure with almost no increase in torque over that at zero differential pressure.

Torque at zero differential pressure is that caused by the spring loading of both seats against the ball. The seat thrust reversal with upstream pressure removes this force, and

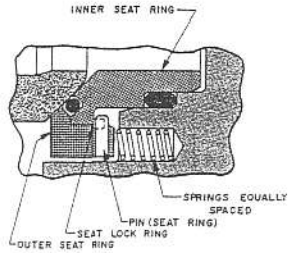


Figure 160. REPLACEABLE O-RING SEATS.

Courtesy of Grove Valve & Regulator Co.

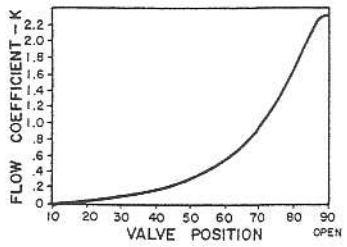


Figure 161. TYPICAL VALVE FLOW COEFFICIENT VERSUS VALVE POSITION.

For 12 in. Full Ported Valve in 12 in. Piping.

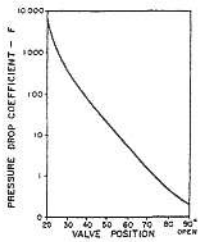


Figure 162. TYPICAL PRESSURE DROP COEFFICIENT VERSUS VALVE POSITION.

For 12 in. Full Ported Valve in 12 in. Piping.

the differential pressure loading on the pressure energized annular area of the upstream seat.

Sealant Back Up

A sealant is sometimes used to hydraulically jack the seat against the ball for tighter sealing as well as providing a viscous sealant at the ball-to-seat joint. Injection pressures can vary from 2000 to 15,000 psi which is a highly efficient means of improving sealing in services which are not contaminated by the slight amount of sealant in the flow stream. Figure 157 illustrates the sealant back-up injection system.

Large Pipeline Ball Valves

Large pipeline ball valves used in liquid throttling service are often made without having the seat rings contact the ball for tight shut-off. For example, the seat rings of a 24 inch valve may clear the ball by .06 inch. This allows the valve to throttle with reduced operating torque in that the seat ring load generated by differential pressure and the backing spring is removed. The remaining load consists of dynamic fluid flow torque and friction from trunnion bearings and stem packing. Valves normally operate near the full open position to reduce pumping energy requirements and some cavitation may be experienced when flow is greatly restricted for short periods.

Flow Coefficient

The flow coefficient is a function of the number of degrees the valve is open. Figure 161 is a graph of this relationship. The following equation will help in calculating a definite number.

$$w = 189\ K\ d^2\sqrt{\Delta P \rho}$$

w = Flow rate, lbs /hr.

K = Coefficient of flow resistance

d = Port diameter, inches

ΔP = Pressure drop, psi

ρ = Density, lbs /cu ft.

The relationship of pressure drop coefficient to valve position is illustrated in Figure 162 where:

$$F = \frac{1}{K^2}$$

Flow Characteristic For Pipeline Pressure Control

When the valve is in the almost entirely open position, very little of the total system pressure drop exists across the ball which it needs to effectively control pressure. The use of a conventional actuator whose output gives a constant rate of ball rotation provides very sluggish control of the valve. Special actuators are available which have a variable rate of ball rotation. Fast rotation of the ball near full open and slower near closed position will result in better throttling results. Figures 163, 164 and 165 illustrate the valve, actuator leakage, and flow characteristics.

Torque

Torque formulas for the double seated ball valve design involve many factors as shown in Figure 166. The seat spring force is almost constant but other forces vary with differential pressure. Ball valves exhibit a peaking of torque at the "crack open" point, then rapidly returning to dynamic operating torque. Figure 166 also illustrates a typical torque chart used to size ball valve actuators at 1.5 times the maximum valve torque for the rated differential pressure which usually is equal to the body pressure rating.

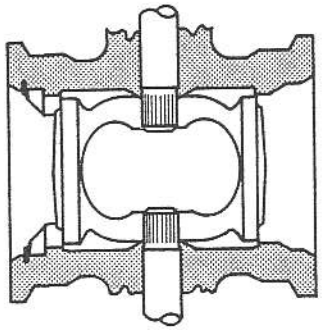


Figure 163. THROTTLING PIPELINE BALL VALVE.

Cylindrical plug inserts into single piece body through the end. Flange or flangeless construction. Actuator with special linkage generates an ideal flow characteristic. 600 lb. rating in Sizes 10 x 8 x 10 inches to 24 x 19 x 24 inches.

Courtesy of Masoneilan International, Inc.

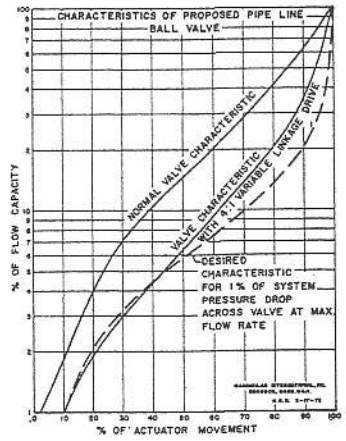


Figure 165. SPECIAL FLOW CHARACTERISTIC.

Gives Maximum Control with Minimum Pressure Loss Near Full Open Position.

Courtesy of Masoneilan International, Inc.

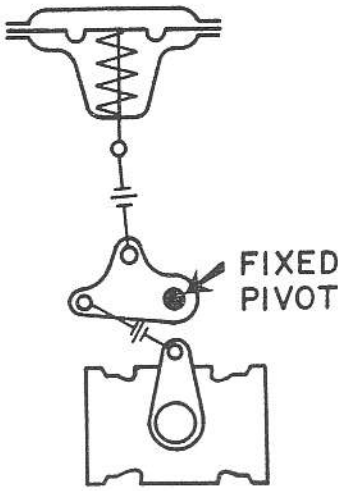


Figure 164. ACTUATOR LINKAGE USED TO GENERATE SPECIAL FLOW CHARACTERISTIC.

Courtesy of Masoneilan International, Inc.

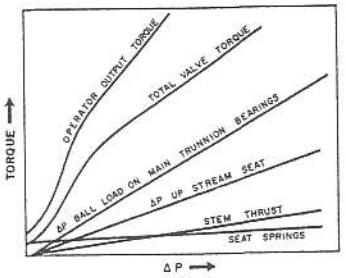


Figure 166. DOUBLE SEATED TRUNNION MOUNTED BALL VALVE TORQUE.

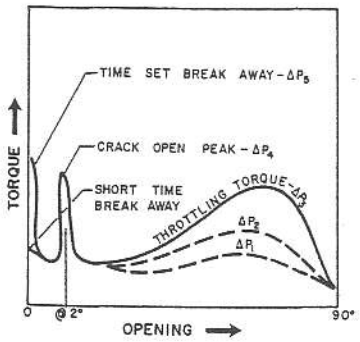


Figure 167. TYPICAL OPERATING TORQUE.

Note: Time Set Break Away is Higher Than All Dynamic Operating Torques.

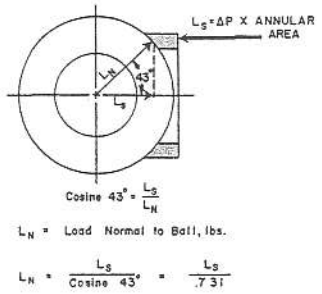


Figure 168. SEAT THRUST LOAD NORMAL TO BALL AT POINT OF CONTACT.

Typical operating torque is shown in Figure 167. Note that "time set breakaway torque" exceeds all other torque conditions. The needed torque increases up to a maximum over a period of several days when a valve remains in closed position with differential pressure loading. The elastomer seal material cold flows into the minute surface roughness of the ball surface which requires a high torque or shearing force to initiate ball movement.

Figure 168 illustrates the seat load that is normal to the ball. The angle shown, 43° becomes smaller as valve size decreases. A 45° angle would give a maximum flow port diameter but would leave no ball surface for the seat contact in the closed position.

ROTARY ECCENTRIC PLUG VALVES

Operating Sequence

The valve plug rotates through a 50° angle from the full open position and cams into the seat. The leading edge of the plug passes the top of the seat by a narrow margin and then the trailing edge contacts the upper seat section. The leading edge is now opposite the lower seat but standing off a few thousandths of an inch. Further rotation and application of stem torque by the actuator flexes the plug arms, forcing the leading edge of the disc into contact with the lower seat while camming the plug tighter into the upper seat. The seat ring is adjusted vertically during assembly to be in a position to contact the valve plug as described above. The plug is free to move axially along its shaft to align with the seat ring in a horizontal plane that is through the center of the seat axis. Figure 169 is an illustration of a rotary eccentric plug valve.

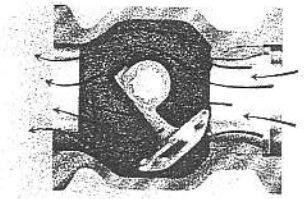


Figure 169. ROTARY ECCENTRIC PLUG VALVE.

Courtesy of Masoneilan International, Inc.

Flow

The dynamic torque for this valve has a fairly constant slope which yields stable throttling. Figure 170 is a graph of dynamic force versus stroke. The inherent flow characteristic is linear and the critical flow factor (a measure of resistance to cavitation) is higher than either the butterfly or ball valves between 10 to 100% of valve stroke. Below 10% the valves are equal.

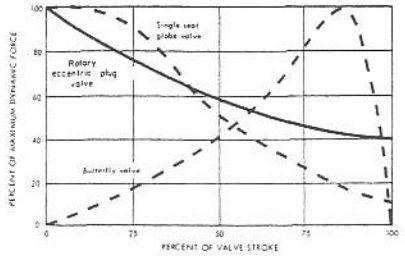


Figure 170. DYNAMIC FORCE VS. STROKE FOR A ROTARY ECCENTRIC PLUG VALVE.

Courtesy of Masoneilan International, Inc.

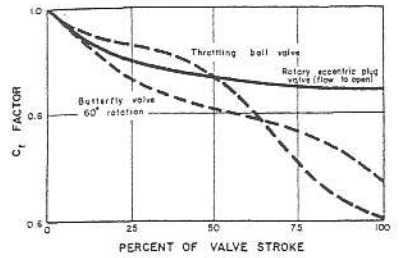


Figure 171. CRITICAL FLOW FACTOR VERSUS STROKE FOR A ROTARY ECCENTRIC PLUG VALVE.

Courtesy of Masoneilan International, Inc.

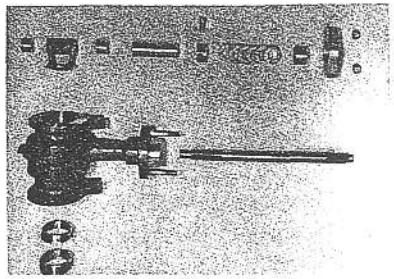


Figure 172. PARTS OF A ROTARY ECCENTRIC PLUG VALVE.

Courtesy of Masoneilan International, Inc.

Figure 171 relates valve stroke and critical flow factor.

The valve has a slightly reduced port but has a good flow capacity in that flow resistance is less for this type than for a globe style valve. It is particularly well suited to services which tend to clog globe and ball valves due to pockets in their flow stream.

Materials

The valve plugs are manufactured from solid Stellite® in sizes up to 4 inches and from 316 Stainless Steel with Colmonoy® hard facing for larger sizes. Seat rings are 316 Stainless Steel and may be hard faced. For extremely erosive service such as lime mud in paper mills, the seat and flow orifice of the retainer may be made from Stellite, and this design has given much longer service than many other types of valves. Bearings at each end of the shaft may be 440-C Stainless Steel or Stellite depending upon the type of service. This valve design (Figure 172) functions from -320°F. to 750°F. without modification or additions. The leakage from the rotary packing is less than for a reciprocating stem because there is no wiping action in liquid service. For additional information refer to Chapter 3.

ROTARY BLADE VALVE

The one piece combination blade and shaft of this design rotates in ball type bearings then cams into the seat giving bubble tight shut off. The large diameter stem seal is a single TFE "U" ring energized by a garter spring which yields tight sealing with low friction. When the valve is in the full 90 degree open position, the eccentrically rotated blade is clear of 95% of the flow-way cross-sectional area which results in exceptionally high maximum flow capacity. Flow capacity for this valve is double that of a throttling type ball valve havng slightly reduced ports and 50% greater than a flat blade butterfly valve of the same size. The rotating spherical blade serves as a flow control surface which cams into a high molecular weight polyethylene seat ring which can withstand temperatures up to 350°F. A carbon face seal, 440-C Stainless Steel bearings and a netal seat allow operation to 600°F. The cam action on closing eliminates rubbing wear on the seat surface and greatly reduces sticking when standing closed for long periods. Backlash is eliminated by the one piece blade and shaft construction. Figure 173 illustrates the rotary blade valve. It is available in sizes 4 thru 12 inches in ANSI 150 and 300 lbs. pressure classes.

Dynamic flow operating torque for the rotary blade valve is one eighth that of a flat vane butterfly valve, and torque changes gradually with the 90 degrees of opening. There is a strong flow force to open the valve when throttling near the seat which gives the valve stability. This force continues but to a lessor extent throughout the full opening. This is only true if with flow is into the face of the blade which is the preferred direction to prevent build up of solids within the body cavity behind the blade. The flow characteristic is approximately equal percentage but may be altered by means of a cam if a positioner is used on the actuator. Cavitation

characteristics and noise levels are about the same as for a ball valve.

Trim change for this type of valve in the flow line is relatively fast since all parts are removable through the bonnet.

This type of valve is applicable to any high flow and or low pressure drop requirement system. It is capable of handling slurries. Seating has remained nearly bubble tight in excess of 500,000 closures with clean fluids.

PLUG COCK STYLE VALVES WITH CAGE TRIM

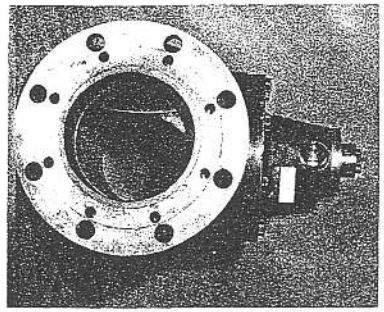
This type of valve has a hollow cylindrical plug with a rectangular slot in it. The plug slips down around the cage, and is the rotating member. The cage is concentric to the plug while being internal, and having its position fixed. A TFE insert sleeve is locked into grooves in the valve body and this serves as the plug seal. Metal-to-metal throttling is therefore possible while the metal-to-TFE seating area is protected. Figure 174 is an exploded view of plug cock style cage trim.

In its full-open position this valve has a high capacity because of it's straight through flow pattern. This is illustrated in Figure 175. It also has a high rangeability.

There is a TFE bearing around the internal bore of the plug, below the port opening, which allows plug throttling rotation to take place with a minimum of friction. The upstream cage orifice is identical to the port in the plug, but the downstream orifice of the cage is characterized to give the flow characteristics shown in Figure 176. At this point the metal surfaces may be hardened since this is where the pressure drop occurs.

Double sealing action is provided by the liner in both the upstream and downstream direction in that the plug has a slight taper (2°) and is spring loaded into the body liner by the rotary stem and bonnet seal. Relief area in the valve body allows for thermal expansion of the sleeve. Figure 177 illustrates the stem and bonnet seal.

Valve sizes range from 1 to 6 inches in 150 and 300 lb. ratings with allowable pressure drops to 150 psi. Bubble tight sealing is obtained when the temperature is down as low as 0°F. but the maximum temperature fluctuation is limited to 200°F. Maximum temperatures can approach 400°F. Flow capacity is about 70% that of a conventional plug valve.



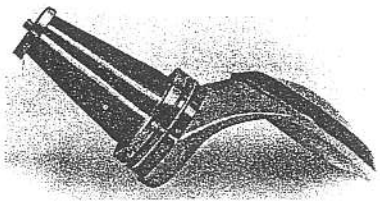


Figure 173. ROTARY BLADE VALVE.

Courtesy of Keilay & Mueller, Inc.

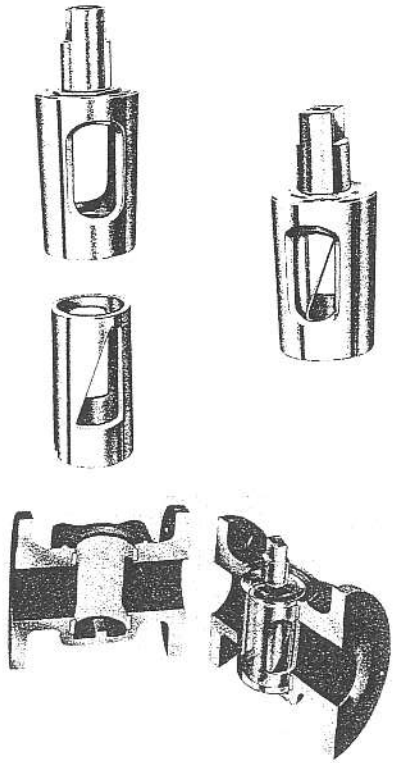


Figure 174. PLUG COCK STYLE CAGE TRIM.

Courtesy of Tufline Division of Xomox Corp.

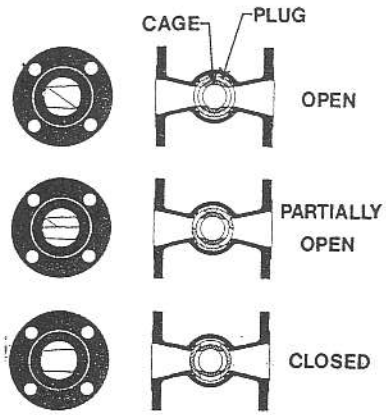


Figure 175. FLOW PATH OF PLUG COCK CAGE TRIM.

Courtesy of Tufline Divisin oi Xomox Corp.

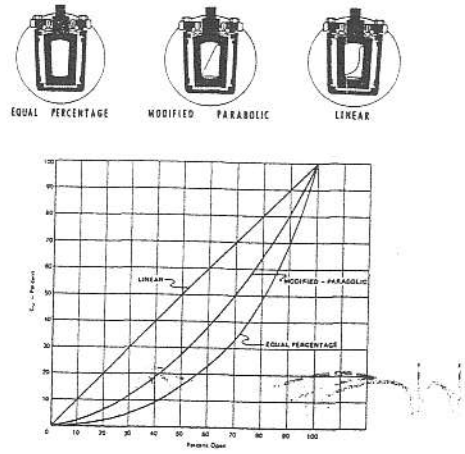


Figure 176. FLOW CHARACTERISTICS OF PLUG COCK CAGE TRIM.

Courtesy of Tufline Division of Xomox Corp.

Rotary plug valves characteristically have very high flow capacities and high rangeability. Flow capacities are often more than double those of globe pattern valves. Rangeability varies with valve size from about 100 to one for a one-inch valve to more than 300 to one for larger sizes. Plug valves are inherently high recovery devices when in the full open position, but the amount of pressure recovery diminishes as the valve approaches the closed position. Typical values of the pressure recovery coefficient, F_L, range from .65 in the full open position to .95 at 10 percent open. The flow characteristic is inherently equal percentage. Other characteristics are available from either characterized trim or characterized position feed back. Restricted capacity trim is also available with the common flow characteristics.

V-PORT ROTARY PLUG VALVE

A V-port hollow plug cams into a rectangular seat to give an inherent equal percentage characteristic at low flow and a linear characteristic at high flow. It is widely used for control of paper stock without dewatering because sharp corners and cavities have been eliminated. See Figures 178, 179 and 180 for the valve, throttling action and flow characteristic. The valves are available in sizes ranging from 2 thru 8 inches in the 150 lb. class with a temperature range of -40 to 140°F.

SPLIT BODY ANGLE CONTROL VALVES

This type of valve has an acid resistant tantalum liner (.015 to .030 inch) which offers

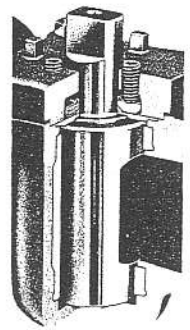


Figure 177. STEM AND BONNET SEAL.

Courtesy of Tufline Division of Xomox Corp.

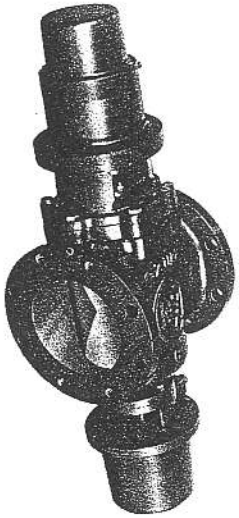


Figure 178. ROTARY V-PORT THROTTLING VALVE.

Shown with Electric Actuator (Top) and Position Switches (Bottom).

Courtesy of DeZurik Corporation.

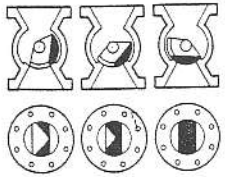


Figure 179. V-PORT PLUG THROTTLING ACTION.

Courtesy of DeZurik Corporation

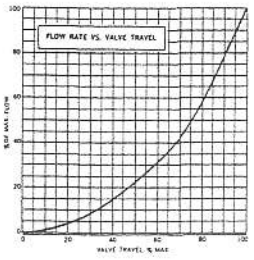


Figure 180. INHERENT FLOW CHARACTERISTIC FOR V-PORT ROTARY PLUG VALVE.

Courtesy of DeZurik Corporation.

considerable savings over a solid tantalum valve. Other liner materials may also be used. It has operated successfully in nitric acid at 600°F. and 800 psig. The contoured plug is also clad. This type of design covers 1 to 10 inch sizes in ANSI 150 and 300 lb. classes.The liner is replaceable. Figure 181 illustrates a metal lined angle valve which is suitable for corrosive service.

V-ORIFICE GATE VALVES

This valve incorporates the wafer body design. In operation, the thin gate slides against a raised face seat, while a relief groove around the seat allows the gate to push solid particles in the fluid aside to prevent packing them in the seat area. The relief groove is flushed on each opening. The seating materials may be metal to metal or resilient with an elastomer seal ring in the gate. Figures 182 and 183 illustrate the V-orifice gate valve and an elastomer seat arrangement. The valve flow characteristic is modified equal percentage at first, then changes to linear. Valves are available in a range of sizes from 2 to 36 inches, constructed from stainless steel with pressure ratings of 125 and 150 lbs. ANSI. This type of valve is frequently used in paper stock service.

DIAPHRAGM VALVES

Diaphragm valves offer literally hundreds of combinations of body material, linings and elastomer materials to handle both water and corrosive fluids. Drop tight shut-off is possible with a 50 psi differential pressure for large valves and higher valve differential pressures for smaller valves up to a maximum of 200 psi. The valve size ranges from 1/4 inch to 20 inches. This valve design is both abrasive resistant and non-clogging. The bonnet is completely isolated from line fluid and top entry allows in-line maintenance. Positive support of the diaphragm against high line pressure is

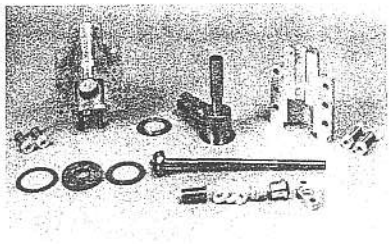


Figure 181. METAL LINED ANGLE VALVE FOR CORROSIVE SERVICE.

Courtesy of Fansteel

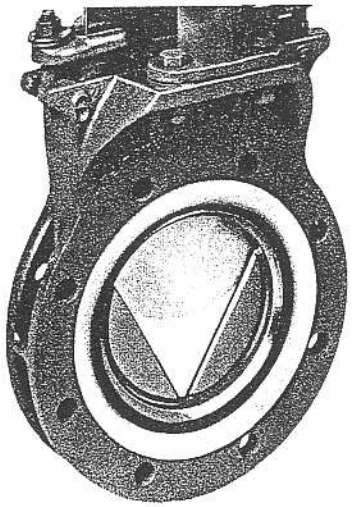


Figure 182. V-ORIFICE GATE VALVE.

Courtesy of DeZurik Corporation

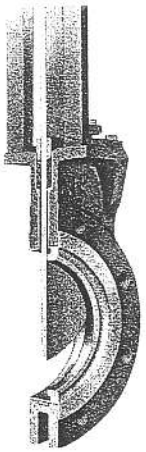


Figure 183. ELASTOMER SEAT IN V-ORIFICE GATE VALVE.

Courtesy of DeZurik Corporation

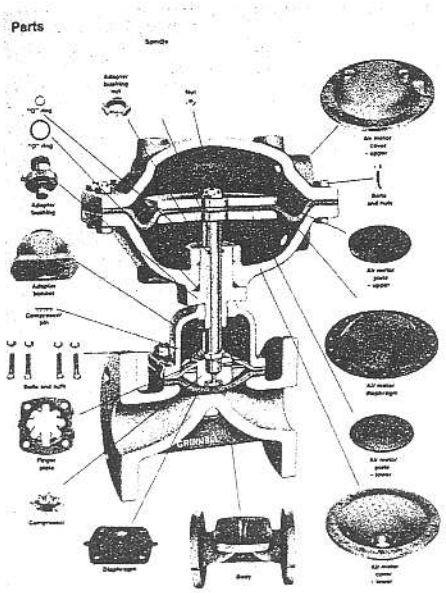


Figure 184. DIAPHRAGM VALVE.

Courtesy of ITT Grinnell Co., Inc.

provided by the fingers of the compressor finger plate as illustrated in Figure 184.

Since there is no packing, leakage from this source is eliminated. The metal bonnet offers a pressure retaining backup member in case of diaphragm rupture. The seat closure is bubble tight and the valves find frequent use in water treatment service as well as chemcal and abrasive services. Diaphragm valves may be completely serviced while still being in line. Operating pressures are limited by the large area of the diaphragm and the effect of line pressure on this area versus available actuator force. Diaphragm valves should not be used on dead end service since on closure there is a pumping action as the diaphragm displaces liquid back into the pipeline.

Two diaphragm valve designs are available. The Weir type with a raised bridge wall, and the straightway type where the diaphragm projects into approximately the full pipe bore. The latter design has a higher flow capacity and is better suited for fibrous slurry service but has half the differential pressure rating of the Weir type. The Weir type is also built in what is known as full bore design so as to pass line-cleaning balls used in food and brewery service.

The following chart and Figure 185 will help in the selection of the best diaphragm material for the type of service the valve will be subjected to.

Valve	Grade	Material	Typical Services	Temp. °F. Min.	Max.
Weir type elastomers	A	Subnatural rubber	Abrasives	0	180
	B	Black hi-temp butyl	Chemicals, gases, stronger acids	-20	250
	C	Hypalon	Oxidizing fluids	0	225
	M	Ethylene propylene	Chemicals, acids, hi-temp abrasives	-30	300
	P	Hycar (nitrile)	Hydrocarbons, gases	10	180
	S	Natural rubber	Water	-30	180
	T	Neoprene	Weak chemicals, air	-20	200
	U	Urethane	Abrasives, solvents, hydrocarbons	-40	225
	V	Viton	Specific solvents & chemicals	-20	325
	WB	White hi-temp butyl	Foods, beverages	0	225
	WF	Pure gum rubber	Beer, beverage, foods	-30	160
	Y	Black Tygon (plasticized pvc)	Chemicals	0	150
	YH	Clear Tygon (½" & 1" only)	Hydrogen peroxide	0	150
Weir type plastics*	E	Polyethylene	Specific acids	10	135
	R-2	Tetrafluoroethylene (TFE)	Severe chemicals, solvents	-30	350
	Z-2	Tetrafluoroethylene (TFE)	Severe chemicals, solvents	-30	280

*With ethylene propylene backing cushion.
Diaphragms at maximum temperatures cannot be used satisfactorily at maximum pressures. Pressure/temperature chart may be used as guide to the maximum combinations.

Courtesy of ITT Grinnell Co., Inc.

Diaphragm valves have been successfully used in refrigerant (freon), chemical, lime slurry, sewage sludge, water demineralizaton, and nuclear service.

Flow Characteristic

The flow characteristic for the diaphragm valve is linear but changes to quick opening above 60% open. A dual range valve design gives more of an equal percentage type curve. Figures 186 and 187 illustrate the dual range flow characteristics and plug functioning.

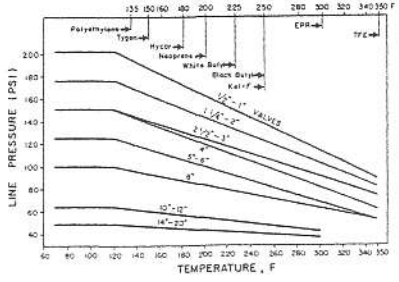


Figure 185. PRESSURE VERSUS TEMPERATURE RATINGS FOR DIAPHRAGM MATERIALS.

Courtesy of ITT Grinnell Co., Inc.

FOOD INDUSTRY VALVES

This type of valve is manufactured in sizes ranging from 1½ to 4 inches. The valves are built in angle, offset, Y and three-way body patterns having internally polished, smooth, and full draining features. The body materials are thin wall stainless steel and other noncorroding alloys. The trim parts are all polished. Plugs are of hard rubber, bonded to hard chrome plated stems with O-ring stem seals. A three-way design is shown in Figure 188.

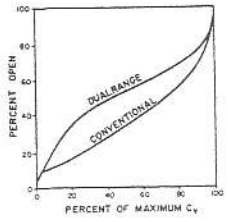


Figure 186. DUAL RANGE FLOW CHARACTERISTIC.

Courtesy of ITT Grinnell Co., Inc.

TRIM IN OTHER TYPES OF CONTROL VALVES

Many types of valves are available and some offer certain advantages over other styles for particular applications. A full discussion of trim for these valves is not included here, but a brief discussion of the most common types is given in the following.

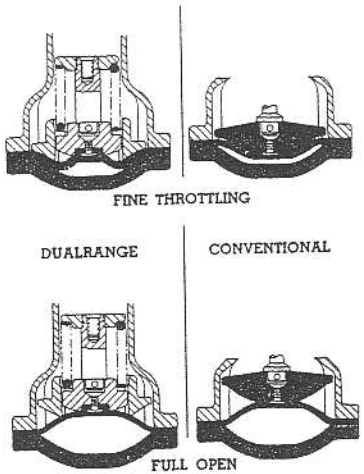


Figure 187. DUALRANGE PLUG FUNCTIONING.

Courtesy of ITT Grinnell Co., Inc.

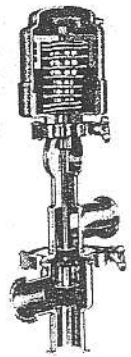


Figure 188. THREE-WAY FOOD PROCESSING VALVE.

Courtesy Ladish Company, Tri-Clover Division.

Digital Signal Operation

Digital signal operation of conventional valves may be had by using a device which changes the signal to either an air pressure or a mechanical loading. A unique valve design now accepts the parallel binary information directly to open or close a number of flow control orifices. Eight flow orifices, geometrically progressing in area, may be controlled by an eight-bit binary word for a 255 to 1 rangeability of flow control. Extremely fast response is assured by pilot solenoid-operated, poppet valves which are pressure balanced (20 milliseconds fully open to closed). These are located at the circumference of a flow tube. Orifices are graduated in size and are in multiples for precise flow control.

Sliding Gate Diaphragm Valves

Sliding gate diaphragm control valves employ the principle of a fixed plate with orifices and a spring-loaded sliding gate with orifices. When both sets of orifices coincide, flow is at maximum; when they are offset, the valve is closed. This provides a short-stroke, highly responsive valve. The gate and plate are self-cleaning and provide a self-lapping action with stroking. The flow characteristic is linear and much of the turbulence is reduced by the straight through flow; noise and wear on internal parts are also minimized. Thermal change does not cause leakage, because both the gate and the plate are free to expand, yet remain in close contact. Overlap of the closed seat-joint provides a long sealing path and the lapped surfaces seat tightly. The opening force is the product of the coefficient of friction and the pressure drop loading on the orifice area; thus, the operating force requirements are lower than for a non-balanced globe design of similar orifice area. The surface of the stationary orifice plate, which is in sliding contact, is hard chrome-plated, which effectively resists fluid erosion and gate wear. See Figure 9(g).

In-Line Valves

In line valves are compact designs in which the flow passes through an annular space surrounding the plug. Some designs anchor the plug to the body wall with struts, and shut-off and throttling

is done with a sliding seat, moved by an annular piston actuator. This presents a nearly pressure balanced design. A second version of this design incorporates a rubber plug which is compressed axially, forcing a radial expansion which throttles or seals the annular flow-way. A hydraulic cylinder in line with the plug serves as an actuator. It, in turn, is actuated by an air-operated diaphragm actuator, driving a free floating hydraulic piston to force oil into the cylinder attached to the compressible plug. These valves are frequently used in high pressure gas reduction (1000 psi down to 40 psi in one stage) and are much quieter than conventional globe style regulators. A third type of design uses a traveling, slotted tube, attached to a plug with a seat fixed to the body wall.

Pinch Valves

Pinch valves squeeze a heavy-wall, reinforced elastomer tubing section between the smooth bars of a pinch clamp connected to the actuator. Shut-off is tight and the valve cannot be jammed by a build up of solid material, because precipitates are broken away with each closing, by the flexure of the elastomer tube. It is an excellent valve for slurries. See Chapter III.

Plug Valves

Plug valves are noted for tight shut-off and high flow capacity. Their ports may be either round, rectangular, or shaped to characterize the flow. The plugs may be solid cylindrical, split, or tapered. To seal a tapered plug, the plug must be mechanically loaded into the taper, usually a 9° included angle, in metal-to-metal joints. Smaller taper angles bind, whereas with larger angles, the plug may pop out of the taper upon standing, under a pressure drop load. Metal jointed valves usually require either some type of anti-friction plug coating or a viscous lubricant-sealant to seal tightly and maintain a reasonable operating torque. Many plug valves for corrosive service use TFE body liners or seat ring type seals, located within the plug or body wall. In cylindrical plug design, the seat rings are both spring and pressure energized, permitting an upstream seal.

Sleeve Valves

Sleeve valve types generally consist of a slotted section of tubing with a barrier plate between two rows of axial slots. An elastomer sleeve, overlying the slots, is actuated by line fluid and seals the sleeve against the tube. Flow is throttled by the clearance between the sleeve and the tube as the sleeve is forced outward from the tube wall by upstream pressure. Many applications for sleeve valves are found in gas pressure-reducing service. See Chapter III.

ACKNOWLEDGMENT

This chapter is a summation of information on valve trim covered in the technical writings of the many authors listed in the bibliography and also the un-named writers of valve catalogs and bulletins. To this the author has added from his personal experience in development, production design, field testing and application of conventional control valves as well as stop, ball, plug, and gate valve types. The section on Cage Guided Balanced Trim is primarily his own effort, but much of the information in the other sections is knowledge accumulated over the years through the untiring efforts of many engineers and technicians who have worked with control valves. The author expresses his appreciation to the Rockwell Manufacturing Company for their generous assistance and permission to publish this information. Particular credit for art work is due Messers Dwight P. Bey, Melvin A. Roberts and David J. Diemert. Secretarial and stenographic credit goes to Miss Sherry Donovan and Mrs. Sandra Pumilia. Thanks is also expressed to my wife, Lucille, who expressed patience and encouragement throughout preparation of the chapter.

Credits for extensive additions for this second edition go to Mrs. G. Nearing for her splendid secretarial and stenographic support and Paul Perrotta of Masoneilan's Norwood Drafting Section for many of the figures.

REGISTERED TRADE NAMES USED IN CHAPTER 4

17-4 PH (stainless steel)	ARMCO Steel Corporation
Stellite and Hastelloy	Haynes Stellite Cabot Corporation
Colmonoy	Walls Colmonoy Corp.
Nickel K/Monel S/Monel Inconel/X Ni Resist	International Nickel Company
Durimet	Duriron, Inc.
Malcomizing	Chapman Valve Co.

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APPENDIX FOR CHAPTER FOUR

PROPERTIES AND APPLICATIONS OF COMMON TRIM MATERIALS

Material & Composition	Temperature Range °F	Hardness as Used	Nominal Size Range	General Applications & Limitations
MILD SERVICE				*Non-erosive, non-corrosive low pressure drops, 50 psi Maximum.*
Bronze	-460° to 450°F	60 BHN	1/4 to 4 in.	Water, air, gas saturated steam.
GENERAL SERVICE				*Non-erosive, corrosive* moderate pressure drops.*
Type 316 Stainless Steel: 18% Cr, 8% Ni, 2% Mo	-450° to 600° F[1]	(14) R_C Max.	All	Steam, water, oil, and gas to 200 psi pressure drop. Cold gases to 300 psi drop.
(Most corrosion resistant of 300 series)				*For severe H_2S service specify special materials.
TOUGH SERVICE				*Higher pressure drop, erosive, slightly corrosive.*
17-4 PH Stainless Steel	-40° to 750°F	40-45R_C	All	Erosive and corrosive service (corrodes in pure water).
Type 410 Stainless Steel (plug only), 12% Cr.	-150° to 750°F[1]	37-42R_C	All	erosive service or non-corrosive service.
No. 6 Stellite® 30% Cr. 4.5% W, 3% Ni, 3% Si, 3% Fe, 2% mn, 1.5% Mo, Balance Cobalt	-450° to 1200°F	40-45R_C at 70°F 38R_C at 1200°F		
Cast Solid			1/4 to 2 in.	Erosive and corrosive service.
Hard Facing			1 in. and up	Slightly erosive and slightly corrosive service.

[1] Refers to seat joint. May be used at higher temperature for base trim material.

Material & Composition	Temperature Range °F	Hardness as Used	Nominal Size Range	General Applications & Limitations
No. 5 Colmonoy® Hard Facing	-150° to 1200° F	45-50R_C at 70°F	1 in. and up	Seats for erosive and corrosion resistant service.
Spray welded and fused 12% Cr, 3% B, Fe-Si-C 9% maximum, balance Ni.		38R_C at 1200°F		Good impact resistance. Corrosion resistance similar to Inconel.
No. 6 Colmonoy® Hard Facing	-150° to 1200°	56-61 R_C at 70°F	1 in. and up	Plugs for erosive and corrosive service. Corrosion resistance similar to Inconel.
Spray welded & Fused 14% Cr, 3% B, 4% Si, 5% Fe, Balance Ni (Chromium Boride crystals in nickel base)		(44R_C at 1200° F)		
VERY EROSIVE SERVICE				*Abrasion, cavitation and impingement erosion.*
Type 440-C Stainless Steel 17% Cr high carbon steel	-50° to 750° F	50-60R_C	1/4 to 4 in.	Non-corrosive service.
Hardened Tool Steel	-50° F to 450° F	60-63R_C	1/4 to 2 in.	Non-corrosive service.
Tungsten Carbide sintered WC with cobalt binder	0° to 250° F	67-72R_C	1/4 to 1-1/2 in.	Use limited by thermal shock. Brittle at low temperature. Non-corrosive service.

Material & Composition	Temperature Range °F	Hardness as Used	Nominal Size Range	General Applications & Limitations
VERY CORROSIVE SERVICE				*Very corrosive fluids, erosive-corrosive service*[1].
Nickel	-325° to 600°	16 R_c		Reducing solutions (hot concentrated caustic). Neutral sats.
K-Monel® 29% Cu, 1% Fe, 3% Al, Balance Ni	-400° to 900°	27-35R_c		Alkalies, salts, food products, air-free acids, reducing agents.
S-Monel® 30% Cu, 4% Mn 2% Si, Balance Ni	-400° to 900°	30-38R_c (35$_R$ at 1200° F)		Alkalies, salts, food products, air-free acids, reducing agents.
Inconel - X®	-400° to 1200° Good high temperature strength.	35-37R_c (at 1200° F)		Used for springs.
Hastelloy - B® 28% Mo, 5% Fe, 2.5% Cu, Balance Ni	700°	14R_c		Corrosive mineral acids, sulfuric, phosphoric wet HCl gas, oxidizing agents.
Hastelloy-C® 17% Mo, 16% Cr, 4% Fe, 2.5% Co, 4% W, Balance Ni	1000°	23R_c		Free chlorine acid solutions of ferric or cupric salts, strong oxidizing agents.
Durimet-20® 29% Cr, 20% Ni, balance Fe	-50° to 600° -425° to 800°	Cast Wrought		Oxidizing conditions, sulfuric acid at all concentrations.
Titanium	-320° * to 600° F	28-35R_c (at 70° F)		Oxidizing acids, chlorides, and brines at ambient temperature.* *Do not use with liquid oxygen (may detonate).*

[1] Corrosion is a very complex process, often affected by many minor variables; therefore, specification of trim materials must be related to actual user experience in a given application. This information represents general usage areas.

CHAPTER 5

FLOW CHARACTERISTICS OF VALVES

Ralph L. Moore

E. I. DuPont de Nemours & Co.
Wilmington, Delaware

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INTRODUCTION

The valve plugs described in Chapter IV have been shown to be available in a variety of geometrical shapes, each with a characteristic relationship between fractional valve lift and the relative flow through the valve. This relationship is denoted as the *flow characteristic*. The proper selection of the flow characteristic is an important part of control valve application.

Flow characteristics are shown by plotting the flow through the valve at various openings versus valve stem position. Quite obviously, the flow through the valve is affected by the pressure drop across the valve or, more specifically, by the variation in pressure drop across the valve. Thus, pressure drop must be specified as a part of the flow characteristic. Valve catalog flow characteristics are shown at a constant pressure drop throughout the opening of the valve, because the manufacturer has no way of knowing the variations of pressure in systems where the valve might be applied. The characteristic at constant pressure drop is called the "inherent flow characteristic." Some of the more common flow characteristics are shown in Figure 1.

The five basic valve characteristics are shown in Figure 1. Only three of the characteristics, quick opening, linear, and equal percentage, are commercially available.

TYPES OF INHERENT FLOW CHARACTERISTICS

If the valve gain, or sensitivity, is defined as:

$$K_c = \frac{\text{change in flow}}{\text{change in lift}}$$

Then it is evident that the slope of the flow characteristic curve is the *valve gain*. Referring to Figure 1, the quick opening characteristic provides large changes in flow for small changes in lift, therefore it usually has too high a valve gain for use in modulating control. It is limited to on-off service such as sequential operation in either batch or semi-continuous processes. For this reason the quick opening characteristic is sometimes called an "on-off" characteristic (1).

The bulk of control applications use valves with linear, equal percentage, or modified flow characteristics. The modified characteristics generally fall between the linear and equal percentage characteristics shown on Figure 1. The linear inherent flow characteristic provides a change in flow which is linear with valve lift and, thus, with the signal to the valve. The linear characteristic would seem intuitively to be the most desirable characteristic for control, since it provides constant valve gain throughout the stroke at constant pressure drop. However, inclusion of the valve into a system, with its associated piping, equipment, and control loop, leads to considerations which generally make equal percentage the most widely applied characteristic. The equal percentage characteristic produces a change in flow, with a change in lift, that is a constant percentage of the flow before the change was made.

All valves, quite obviously, have flow characteristics and a few of the inherent characteristics(2) are shown by Figure 2.

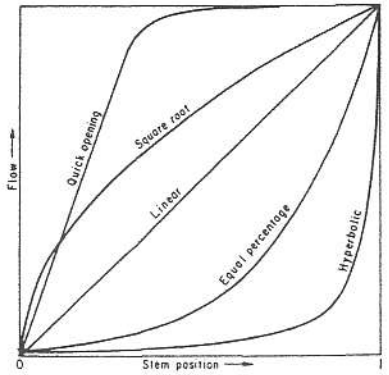


Figure 1. INHERENT VALVE CHARACTERISTICS.

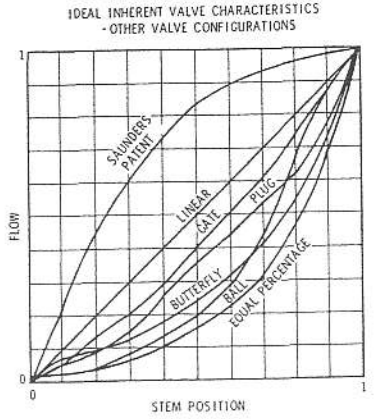


Figure 2. PERCENT OF MAXIMUM FLOW VERSUS STEM TRAVEL FOR TYPICAL VALVE TYPES.

Contrary to Murphy's Law, this figure indicates that many valve configurations have characteristics between the linear and the equal percentage types; therefore, they are conceptually useful in control application. The butterfly valve characteristic is shown to fall within this range. The Saunder's patent valve, which has a characteristic between linear and quick opening, is also useful in control.(3)

INSTALLED FLOW CHARACTERISTICS

The inherent flow characteristic is valuable in describing the valve as supplied by the manufacturer; however, it is immediately evident that few valves actually operate at a constant pressure drop. Thus, the flow characteristic of the valve installed in the pipeline provides the basis for the proper application of the valve. This, appropriately enough, is called the "installed flow characteristic" and is unique for each valve installation; however, some generalities can be considered. Conison(6) has described the typical installation in the petrochemical industries as a centrifugal pump supplying the pump head, piping, and any other equipment in series with the valve which represents a dynamic head loss. This configuration is shown in Figure 3 where the pressure drop across the valve varies with the throughput.

The centrifugal pump provides decreasing pump head pressure with increasing flow demand, as shown in Figure 3a. The pressure drop across the piping and equipment in the system is made up of a static (elevation) component, which is unchanging, and a dynamic loss component which varies directly with the flow rate squared. The plot of dynamic loss versus flow rate is sometimes called the system resistance curve. The pump head and the dynamic loss flow rates are quite close together at the normal design flow shown in Figure 3a, and thus the system would be expected to be unsatisfactory because a small increase in flow demand would leave no control valve pressure drop whatsoever and the system would be uncontrollable. Figure 3b shows the relationship between pump head pressure and dynamic flow rate loss. At this design flow rate, a reserve valve pressure drop is available that would accommodate a larger flow if required.

Lovett(7) has investigated the relationship between the installed valve characteristic and the system pressure distribution by studying the system shown in Figure 4. Parameters relating pump head and dynamic loss are shown in Figure 5. The parameters are then combined into a valve pressure drop ratio P_R, which is the ratio of the valve pressure drop to the total system dynamic loss.

$$P_R = \frac{P_V}{P_V + P_{L^1} + P_{L^2}}$$

Figures 6 and 7 show the distortion of the inherent linear and equal percentage valve characteristics as they relate to the installed characteristics with a varying valve pressure drop ratio. Distortion does increase with a decreasing ratio, and Lovett postulates that a ratio of 0.05 be the limit for acceptable performance.

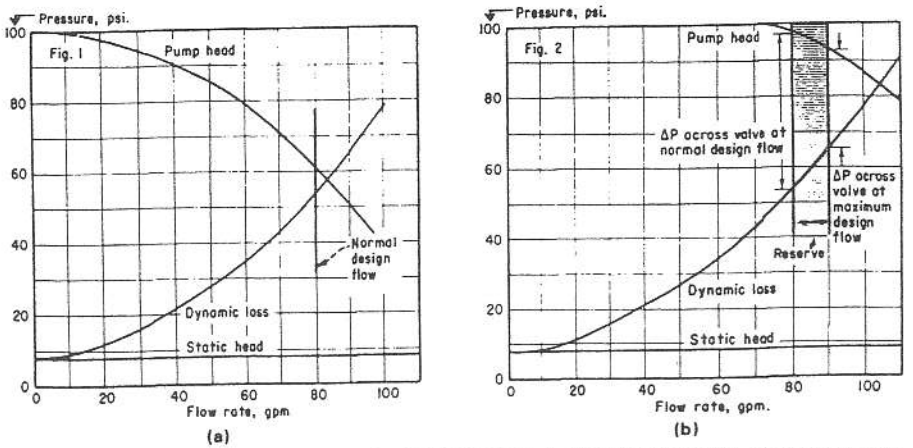


Figure 3. PLOTS OF FLOW RATE VS. PRESSURE IN A PETROCHEMICAL INSTALLATION, WHERE: a) NO RESERVE ABOVE THE NORMAL DESIGN FLOW GIVES WEAK SYSTEM CONTROL; AND b) ADDED FLOW PROVIDES SUFFICIENT CONTROL VALVE PRESSURE DROP.(6)

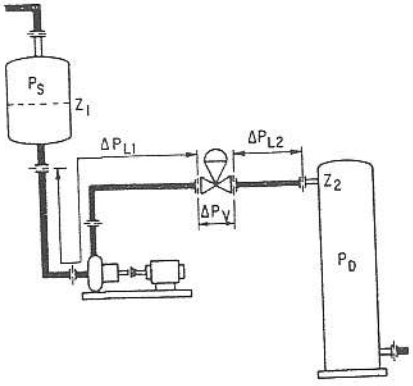


Figure 4. PUMPED-LIQUID FLOW SYSTEM. (7)

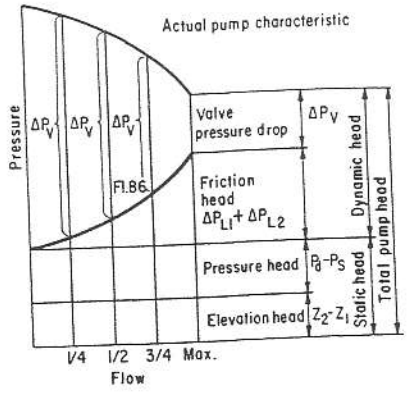


Figure 5. SYSTEM PRESSURE DISTRIBUTION. (7)

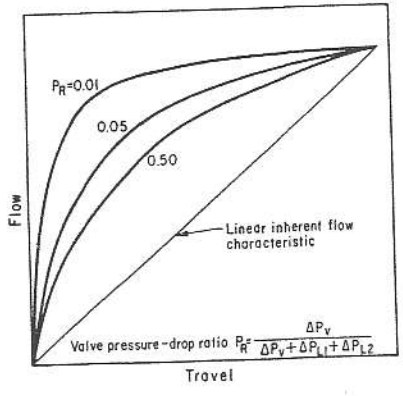


Figure 6. INSTALLED FLOW CHARACTERISTIC WITH LINEAR INHERENT FLOW CHARACTERISTIC. (7)

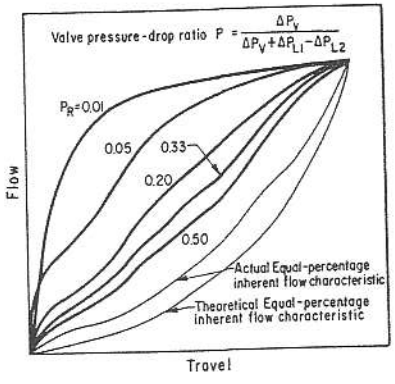


Figure 7. INSTALLED FLOW CHARACTERISTIC WITH EQUAL-PERCENTAGE ACTUAL INHERENT FLOW CHARACTERISTIC. (7)

Moore(15) has developed guidelines for the allocation of pressure drop across the control valve, acknowledging that "the choice of pressure drop is a complex problem which cannot be defined by a set of numerical rules." The guidelines should thus be used more as benchmarks than design criteria. They are as follows:

1.) In a pumped circuit, the pressure drop allocated to the control valve should be equal to 33% of the dynamic loss in the system at the rated flow, or 15 psi, whichever is greater.

2.) The pressure drop allocated to a control valve in the suction or discharge line of a centrifugal compressor should be 5% of the suction absolute pressure, or 50% of the dynamic losses of the system, whichever is greater.

3.) In a system where static pressure moves liquid from one pressure vessel to another, the pressure drop allocated to the valve should be 10% of the lower terminal vessel pressure, or 50% of the system dynamic losses, whichever is greater.

4.) Valves in steam lines to turbines, reboilers, and process vessels should be allocated 10% of the design absolute pressure for the steam system or 5 psi, whichever is greater.

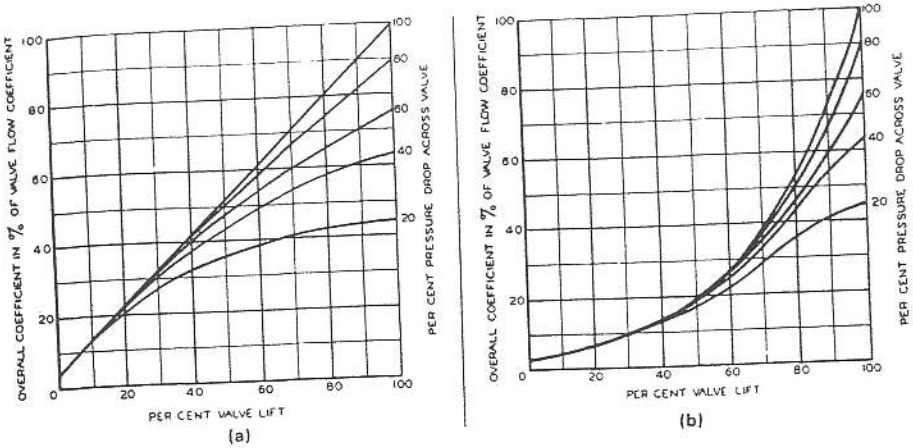


Figure 8. THE OVER-ALL FLOW COEFFICIENT IN PERCENT OF VALVE FLOW COEFFICIENT VS. PERCENT OF VALVE LIFT FOR A VALVE WITH 30 TO 1 RANGEABILITY AND a) LINEAR CHARACTERISTIC; b) EQUAL PERCENTAGE CHARACTERISTIC.(4)

Kneisel(4) developed a formula for the installed flow characteristic using the simplification assumption that the pump head is constant:

$$C_o = \frac{nC_v}{\sqrt{\left(\frac{n}{m}\right)^2 + 1}} \qquad (1)$$

where, for a linear inherent characteristic:

$$m = \frac{C}{C_v} = 0.96X + 0.03 \qquad (2)$$

and for equal percentage:

$$m = \frac{C}{C_v} = (30)^{X-1} \qquad (3)$$

where

C_v = valve flow coefficient at full lift
C = valve flow coefficient at lift X.
X = valve lift, fraction of full open.
C_o = combined flow coefficient for valve and piping

$$n = \sqrt{\frac{\text{pressure drop across valve}}{\text{pressure drop across piping}}}$$

$$m = \frac{C}{C_v}$$

Equations 2 and 3 are based on a valve rangeability of 30. Successive substitutions of these equations into equation 1 result in the graphs of installed valve characteristics shown in Figure 8.(4) The graphs of Figures 6,7,8 show that the linear inherent characteristic takes the form of a quick opening characteristic as smaller proportions of the total-system pressure drop are taken across the control valve; therefore, the linear characteristic becomes less desirable for control under these conditions. The equal percentage inherent characteristic, on the other hand, becomes more linear with a decreasing proportion of pressure drop across the valve; therefore, it is more desirable for control. These graphs, and other similar developments in the literature (5,6,7) provide the simplest rule for flow characteristic selection: *the equal percentage characteristic is recommended if either (a) the major portion of the control system pressure drop is not available through the valve; or, (b) pressure drop is high at low flows, and vice versa.*

Boger(5) has illustrated the effect of parallel flow paths on the inherent characteristic, which is of interest in assessing controllability under a condition such as operation with a by-pass valve open.

The parallel resistance configuration is shown in Figures 9a and 9b with a constant system differential, thus by-pass flow is constant. The by-pass will alter minimum flow and reduce controllability by reducing the rangeability of the control valve. However,

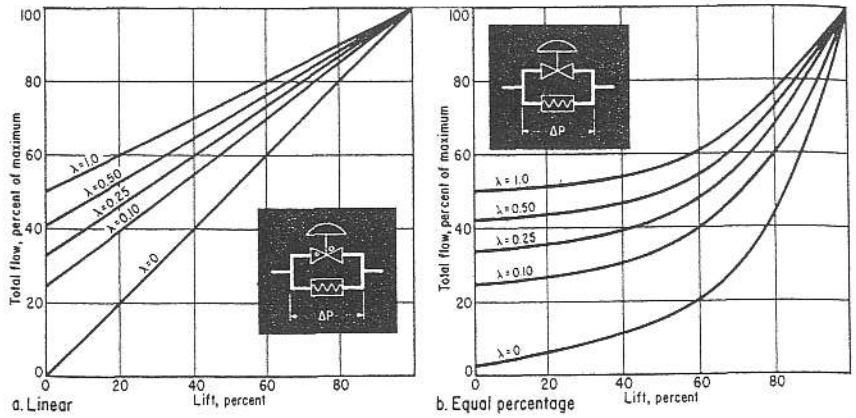


Figure 9. SYSTEM DIFFERENTIAL PRESSURE IS CONSTANT, PARALLEL CONNECTION. [5]

Control over total flow diminishes as the bypass resistance decreases. ($\lambda = P_R$)

the valve flow still exhibits its inherent characteristic form.

The effect of streamlining valve trim components is to produce an erratic flow characteristic. Wing[18] has pointed out that while streamlining reduces wear and tear, it provides a pressure recovery which adds complexity to the prediction of the flow characteristic. The effect of pressure recovery is strikingly illustrated by the flow characteristics of the angle valve with an annular divergent recovery core below the seat, as shown in Figure 10. The characteristic in the flow to close direction, which takes advantage of the recovery, is quite different from the characteristic with flow to open, where the angle configuration provides no recovery.

As a result of this basic investigation, refinements in the techniques for sizing high recovery (e.g., butterfly, ball) valves have been developed. Basically, the new developments include a universal valve sizing equation which accommodates the recovery factor[19], and dual coefficients [20]. While these continuing efforts have resulted in more reliable sizing techniques, little information has been published on the effect of high recovery on the flow characteristic.

Consideration of the installed characteristics also emphasizes the distinction between valve rangeability and turndown. Rangeability has been described as

$$\text{rangeability} = \frac{\text{maximum controllable flow}}{\text{minimum controllable flow}}$$

while turndown is [8]

$$\text{turndown} = \frac{\text{normal maximum flow}}{\text{minimum controllable flow}}$$

Thus, turndown also decreases as the proportion of system pressure drop across the valve decreases, because less pressure drop dictates a larger valve size and a larger minimum controllable flow.

CHARACTERISTIC DETERMINATION

The valve flow coefficient Cv referred to in equations 1, 2 and 3 is determined by physical testing, as described in ISA Standard

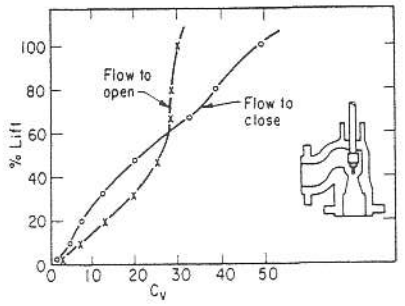


Figure 10. NOTE THE REMARKABLE DIFFERENCE IN Cv BETWEEN FLOW-TO-OPEN AND FLOW-TO-CLOSE IN THIS ANGLE VALVE. (18)

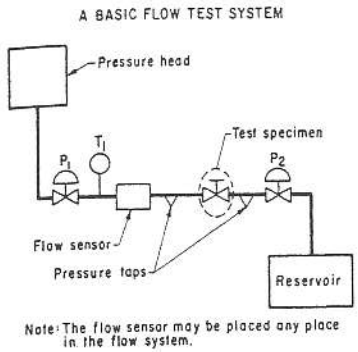


Figure 11. A BASIC FLOW TEST SYSTEM.

Practice SP39.2. It is defined as the volume of water (in gpm) which flows through a valve at rated opening under a 1 psi pressure differential.

A typical valve flow test loops is shown in Figure 11[16]. The test fluid is usually water at temperature between 40°F and 120° F[17]. It is suggested that measurements be taken at each 10% of rated capacity throughout the entire stem travel. Test runs are made at three pressure drops between 5 and 20 psi at each valve opening. The valve coefficient can then be calculated by the following formula for each test, and the results averaged:

$$C_v = \text{gpm}\sqrt{\frac{\text{Specific Gravity of Fluid Flowing}}{\Delta p}}$$

A turbine meter is often used as a flow measuring device because of its accuracy.

Flow Characteristic In The Control Loop

The optimum installed flow characteristic would cause the control system to be able to meet desired objectives of stability, speed of response, and accuracy while being subjected to different disturbances. Valstar [10] and Ziegler and Nichols [13] point out that different disturbances change the process dynamics and static properties in different ways, and thus each disturbance might call for a different valve characteristic.

Ziegler and Nichols demonstrate the differing requirements of the process by the automatic temperature control of the water in a

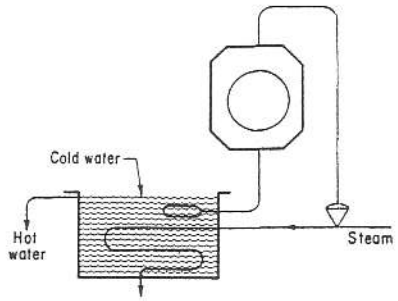


Figure 12. TEMPERATURE CONTROL PROCESS. [13]

vessel which is heated by steam flowing through a coil in the vessel, as shown in Figure 12.

The process can be described simply by the heat balance on the vessel:

$$Mc \quad \frac{dT}{dt} = qc\,(T_0 - T) + q_s \quad \lambda \qquad (4)$$

where

M = weight of water in the vessel, lb.

c = specific heat of water, Btu/lb °F.

T = temperature of water in vessel, °F.

T_0 = temperature of incoming water, °F.

q = water flow, lb/min.

q_s = steam flow, lb/min.

λ = latent heat of steam, Btu/lb.

The left hand side of equation (4) represents dynamics, while the right hand side is a relationship among the variables which dictate vessel temperature (T). Considering the right hand side, then, a change in inlet water temperature (T_0) from low load (warm) to high load (cold) would require a linear change in steam flow (q_s), and a linear flow characteristic would be required. But, suppose steam supply pressure changed. The relationship for critical pressure drop through the valve is of the form:

$$q_s = K \quad C \quad p \qquad (5)$$

Where
K = constant

C = valve flow coefficient at lift X

p = steam supply pressure, psi

In order to maintain constant steam flow (q_s) in the heat balance, it would be necessary that (Cp) remain constant, which is not a linear relationship. The authors [13] point out that an equal percentage characteristic is required for this application. Still another relationship could be required for the non-critical presure drop flow condition.

Brown[12] has developed a "working" pressure flow characteristic for liquid flow control which is defined as, "the relationship between valve fractional opening and the quantity to be controlled for a constant condition of the process." A linear working characteristic thus provides a constant open loop gain. Computer calculated working characteristics[12] illustrate that, (a) different installed flow characteristics are required to maintain a constant open-loop gain in the same control loop with different disturbances, and (b) deterioration in control occurs with undersized and oversized valves. The requirement for different flow characteristics in the same loop, to compensate for different disturbances, means that no "best" flow characteristic can be selected, but that the chosen flow characteristic must be a compromise that is adequate for all anticipated conditions. The foregoing discussion illustrates that an optimum valve characteristic is in general, unattainable. The basic pragmatic criterion for flow characteristic selection is to provide constant relative stability for the control loop over the range of its operation [21]. Many processes and measuring devices exhibit characteristics (gains and time constants) which vary with throughput or some related variable. The heat balance (equation 5) when rewritten into a form describing the dynamic response of temperature, illustrates the varying characteristics:

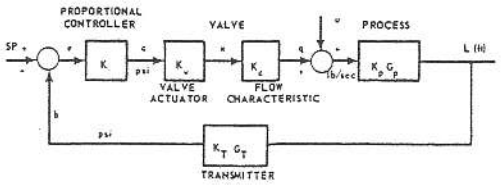
$$\frac{M}{q}\frac{dT}{dt} + T = T_o + \frac{\lambda}{qc} q_s \qquad (6)$$

The time constant $\frac{M}{q}$ is the coefficient of the derivative, and changes with flow. Likewise, the coefficient (gain) of the steam flow (q_s) changes with throughput flow. Both of these changes with throughput will change the relative stability of the temperature control system. Ideally, a flow characteristic could be chosen that would compensate for these changes. Under these ideal conditions, it would be possible to make controller settings at any load condition and have these provide adequate dynamic response under all operating conditions.

The value of a linear installed flow characteristic in providing constant valve gain in the control loop has been discussed. The constant relative loop stability criterion requires in many cases that the installed characteristic be non-linear in a specified fashion. It has been suggested that the choice of an installed characteristic which provides constant open loop gain is desirable [9, 10].

Open-loop gain is the product of all the component gains in the control loop illustrated in Figure 13. A constant, open-loop gain results in constant attenuation of disturbances at all load conditions. Further, Buckley [11] points out that for processes with negligible process dynamics, i.e. $G_p = G_T = 1$ in Figure 13, a constant open loop gain will hold the relative stability of a system constant.

Buckley [11] discusses relative stability from the viewpoint of constant magnitude ratio (M), where M is the ratio of the controlled variable to the referenced variable used in the frequency response technique. The M-ratio is a measure of relative stability. Process dynamics are quite fast in a liquid flow con-



WHERE

K = CONTROLLER GAIN = $\frac{psi}{psi}$

K_v = VALVE ACTUATOR GAIN = $\frac{\text{FRACTIONAL LIFT}}{psi}$

K_c = FLOW CHARACTERISTIC GAIN = $\frac{\text{FLOW (cu.ft./sec.)}}{\text{FRACTIONAL LIFT}}$

K_p = PROCESS GAIN = $\frac{\text{LEVEL (ft.)}}{\text{FLOW (cu.ft./sec.)}}$

G_p = PROCESS DYNAMICS

K_T = TRANSMITTER GAIN = $\frac{psi}{\text{LEVEL (ft.)}}$

G_T = TRANSMITTER DYNAMICS

TRANSFER FUNCTION OF REGULATOR

$$L = \frac{(K_p G_p)\, d}{1 + K K_v K_c K_p G_p K_T G_T}$$

Figure 13. A CONTROL LOOP TO ILLUSTRATE OPEN-LOOP GAIN.

trol system in comparison to the measurement and control equipment. This means that changes in process dynamics have little effect on the open loop gain, and a constant open loop gain results in a constant M value and constant relative stability. In terms of Figure 13:

$$K\,K_V\,K_C\,K_P\,K_T = K_L = \text{Constant}$$

where K_L = open loop gain

K_P = piping gain (for the liquid flow process)

K_c = inherent valve characteristic gain $\frac{\partial Q}{\partial X}$

where the partial derivative is a more general representation of the valve gain, or sensitivity, formerly defined. The installed valve characteristic then becomes:

$$K_P \frac{\partial Q}{\partial X} = \frac{K_L}{K\,K_V\,K_T} \qquad (7)$$

Substituting for K_T the square law relationship for a primary element such as an orifice, results in the following installed characteristic:

$$\frac{X}{X_{max}} = \frac{Q^2}{Q^2_{max}} \qquad (8)$$

A square root installed flow characteristic is thus required to provide constant relative stability as production rate varies.

The inherent characteristic is then obtained by tabulating X, Q, Δp, and the corresponding C. A plot of X versus Q at constant Δp is the inherent characteristic.

Subsequently, Buckley[22] has shown numerically the variation in relative stability provided by the traditional linear and equal percentage flow characteristics when used in the foregoing flow control system, and thus has illustrated strikingly the advantage of the square root characteristic indicated by equation 8.

Table I shows the controller gains calculated for a liquid flow control system with an orifice primary flow element. The gains are calculated for a constant relative stability (Mp = 2db). The left hand column expresses flow as a fraction of maximum flowmeter reading.

Constant relative stability therefore would require a significant change in controller proportional band for each throughput. Pemberton[23] points out the digital computer, when a part of the process control system, represents a way of applying more effective methods of molding the valve - process - measurement characteristic to provide satisfactory control performance. If the installed flow characteristic is known with reasonable certainty, precisely formulated compensation can be introduced to properly match the valve with the process. Equation 1 can be transposed into the following installed characteristic formula for a linear valve:

$$Q = \frac{XQ_{max}}{\left[a+(1-a)\,X^2\right]^{1/2}} \qquad (9)$$

TABLE I
VARIATION IN FLOW CONTROLLER
PROPORTIONAL BAND TO MAINTAIN CONSTANT
RELATIVE STABILITY.[23]

Flow / Max. Flow	Linear		Equal Percentage	
	Gain	Prop. Band	Gain	Prop. Band
0.2	3.85	26	6.4	15.6
0.4	1.93	52	1.03	97
0.6	1.28	78	0.71	141
0.8	0.96	104	0.40	250
1.0	0.77	130	0.26	391

The valve gain is then determined by taking the derivative:

$$K_V = \frac{\partial Q}{\partial X} = \frac{a\,Q_{max}}{[a+(1-a)\,X^2]^{3/2}} \qquad (10)$$

where

$$a = \frac{\text{valve pressure drop when fully open}}{\text{valve pressure drop when fully closed}}$$

Using the constant open loop gain criterion, the desired compensation is:

$$K_{compensation} = \frac{K_L}{K\,K_V\,K_P\,K_T} \qquad (11)$$

and the gain of the compensator is:

$$K_{compensation} = \frac{\partial X}{\partial c}$$

where c = controller output, from Figure 13

If the control valve is linear and the process gain is inversely proportional to flow rate (as in equation 6), then

$$\frac{\partial X}{\partial c} = \frac{K_L\,X\,[a+(1-a)\,X^2]^{3/2}}{K\;K_T\;a} \quad \text{is true.}$$

Integration results in the relationship for the gain compensator

$$X = \left[\frac{ae^{\,2Gc}}{1-(1-a)\,e^{\,2Gc}}\right]^{1/2} \qquad (12)$$

where $G = \dfrac{K_L}{K\,K_T}$

The function of equation 12, when inserted into the control loop between controller and valve, will provide approximate constant open loop gain and thus constant relative stability for the process considered[23]. It should be remembered that the dynamics of the processes being considered are extremely rapid, and thus have a negligible effect on stability.

Boger [24] presents a relationship for the installed flow characteristic of an equal percentage valve which is similar to equation 9:

$$Q = \frac{R^{X-1}\,Q_{max.}}{\left[a+(1-a)\,R^{2(X-1)}\right]^{1/2}} \qquad (13)$$

and the derivative for the gain:

$$K_V = \frac{\partial Q}{\partial X} = \frac{R^{X-1}\,a\;\ln R}{\left[a+(1-a)\,R^{2(X-2)}\right]^{3/2}} \qquad (14)$$

This relationship can then be used to determine the gain compensation required to meet the constant open loop gain criterion.

Schuder[21] comments on the relative stability of a liquid pressure process. This process also has negligible dynamics. A valve with a linear installed characteristic used in this process is shown to provide a narrow range of loads over which acceptable control stability would be obtained without retuning the controller. An equal percentage characteristic, on the other hand, would be expected to perform well over a wide range of loads. This results from consideration of the gain of the equal percentage valve, which is found by taking the derivative of equation 3:

$$K_c = C \log R$$

Where R is the valve rangeability, and is given the value of 30 in equation 3. It is evident from this equation that the gain is proportional to the flow coefficient, or the flow through the valve. Process gain is inversely proportional to flow in the liquid pressure process[21]. Thus the product of these gains produces an open loop gain which is constant, or which is independent of load.

Valstar[10] and Buckley [11] suggest the hyperbolic and square root inherent flow characteristics as required to provide an approximately constant open-loop gain in some control loops. These characteristics are not commercially available as contoured valve plugs but can be obtained by characterized cam positioners. Care must be taken not to significantly alter the positioner dynamics with a characterized cam[21].

Returning to the M-ratio definition of system response, Buckley[11] points out that this technique can also be used to determine the valve flow characteristic for constant relative

TABLE II. OPTIMUM CHARACTERISTICS BASED ON CONSTANT PRESSURE DROPS AND ON SIGNIFICANT LINE OR PUMP DROP.

	Application	*Installed Valve Characteristic*	*Inherent Valve Characteristic* Constant ΔP_v	*Significant Line Drop and-or Significant Pump Drop*
1.	Liquid or Gas Flow Control (negligible Process Lags)			
1.1	Orifice Measurement	$\frac{X_v}{(X_v)max} = \frac{Q^2}{(Q_v)^2max}$	Square Root	Linear - Equal Percentage
1.2	Linear Measurement	$\frac{X_v}{(X_v)max} = \frac{Q}{(Q_v)max}$	Linear	Equal Percentage - Hyperbolic
2.	Ratio Flow Control			
2.1	Open-Loop	See 1.1 and 1.2 above	See 1.1 and 1.2 above	See 1.1 and 1.2 above
2.2	Closed-Loop (Orifice or Linear Measurement)	$\frac{X_v}{(X_v)max} = 1 + \frac{\text{Ln } Q/(Q_v)max}{\text{Ln } \frac{(Q_v)max}{(Q_v)min}}$	Equal Percentage	Hyperbolic
3.	Averaging Level or Pressure Control			
3.1	Simple	$\frac{X_v}{(X_v)max} = \frac{Q}{(Q_v)max}$	Linear	Equal Percentage - Hyperbolic
3.2	Cascaded to Flow Control	See 1.1 and 1.2 above	See 1.1 and 1.2 above	See 1.1 and 1.2 above
4.	Mixing and Blending Control			
4.1	Measurement Exit First Tank	$\frac{X_v}{(X_v)max} = \frac{Q}{(Q_v)max}$	Linear	Equal Percentage/ Hyperbolic
4.2	Measurment Exit Second Tank	$\frac{X_v}{(X_v)max} = 1 + \frac{\text{Ln } Q/(Q_v)max}{\text{Ln } \frac{(Q_v)max}{(Q_v)min}}$	Equal Percentage	Hyperbolic

stability with dynamics in the control system. Ultimately, dynamics must be considered, since a large proportion of control systems do have significant dynamics, and the simpler criterion of of constant open loop gain does not provide constant relative stability.

Consider a control system that has a process time constant (τ_p) and a measurement time constant (τ_T) which is the largest instrument time constant in the loop. Then, if $\tau_p \gg \tau_T$, the open loop gain (K_L) can be shown[11] to be approximately represented by:

$$K_L = \frac{\tau_p}{\tau_T}$$

An analytical expression can be developed for τ_p. Consider, for example, equation 6 in which

$$\tau_p = \frac{M}{Q}$$

and where

$$K_p = \frac{\lambda}{Qc}$$

and equation 7 becomes

$$\frac{\partial Q}{\partial X} = \frac{\tau_p}{\tau_1 K K_V K_p K_T}$$

and with substitution

$$= \frac{Mc}{\tau_1 K K_V K_T \lambda}$$

which can be integrated.

The results of analyses such as this one are summarized in Table II, which lists the form of valve flow characteristics to provide constant relative stability for various processes.

The most desirable valve flow characteristic will provide constant relative stability throughout the operating range of the process being controlled. Constant relative stability is provided by constant open loop gain where the dynamics are negligible. Approximations can be made for dynamics where certain relationships exist between time constants. In the general case, however, the characteristic to provide constant relative stability must be calculated by an analysis of the specific control application. Consider the control system in Figure 14. The block diagam for this process is shown as Figure 13.

The process is governed by a mass balance:

$$A\frac{dh}{dt} = u - q \qquad (15)$$

where

A = cross sectional area of tank, ft^2
h = height of level above drain line, ft
q = modulated flow, ft^3/sec.
u = disturbance flow, ft^3/sec.

The valve in the exit line is analogous to a variable resistance:

$$q = CX\sqrt{h} \quad \text{(specific gravity = 1)} \qquad (16)$$

Letting lower case letters represent small deviations of the defined variables, and capital letters represent average values of the same variables, the linearization of equation 16 results in:

$$q = \frac{C^2X^2}{2Q}\,h + \frac{\partial Q}{\partial X}\,x \qquad (17)$$

The flow coefficient, C, appears in the gain in equation 17 in order to eliminate the head, h, which varies with throughput with proportional control. Substituting equation 17 into equation 15.

$$A\frac{dh}{dt} = u - q = u - \frac{C^2X^2}{2Q}\,h - \frac{\partial Q}{\partial X}\,x$$

where by comparison with Figure 13:

$$K_c = \frac{\partial Q}{\partial X} \approx \frac{dQ}{dX} \text{(in differential form)}$$

then,

$$\left(\frac{2QA}{C^2X^2}\,s + 1\right) h = \frac{2Q}{C^2X^2}(u - K_c x)$$

Also by Figure 13:

$$K_p = \frac{2Q}{C^2X^2}$$

$$G_p = \frac{1}{\frac{2QA}{C^2X^2}\,s + 1}$$

and let

$$G_T = \frac{1}{0.5s+1}$$

The closed loop response to a small change in the disturbance, u:

$$h = \frac{K_pG_p\,u}{1 + \frac{K K_v K_c K_p K_T}{\left(\frac{2QA}{C^2X^2}\,s+1\right)(0.5s+1)}} \qquad (18)$$

where $K K_v K_c K_p K_T$ = open loop gain and $\frac{d}{dt} = s$ (the Laplacean operator).

The denominator of equation 18 is the characteristic equation which determines stability. Clearing this of fractions:

$$\frac{QA}{C^2X^2}s^2 + \left(0.5 + \frac{2QA}{C^2X^2}\right)s + 1 + K K_v K_c K_p K_T$$

This second-order response equation is written in the following general form [14] to determine system stability:

$$\tau^2 s^2 + 2\tau\zeta s + 1$$

Comparing coefficients of like-order terms:

$$\tau = \sqrt{\frac{\frac{QA}{C^2X^2}}{1 + K K_v K_c K_p K_T}}$$

$$2\tau\zeta = \frac{0.5 + \frac{2QA}{C^2X^2}}{1 + K K_v K_c K_p K_T}$$

Substituting the characteristic time, τ, into the second equation, and letting $\zeta = 1$, for constant stability:

$$2\sqrt{\frac{\frac{QA}{C^2X^2}}{1 + K K_v K_c K_p K_T}} = \frac{0.5 + \frac{2QA}{C^2X^2}}{1 + K K_v K_c K_p K_T}$$

Substituting the foregoing value for K_p, and solving this for K_c; the required installed flow characteristic for constant system stability:

$$K K_v K_T K_c = \frac{C^4X^4}{32AQ^2} - \frac{C^2X^2}{2Q} + \frac{A}{2} \qquad (19)$$

$$\frac{dQ}{dX} = \frac{1}{K K_v K_T}\left[\frac{C^4X^4}{32AQ^2} - \frac{C^2X^2}{2Q} + \frac{A}{2}\right] \qquad (20)$$

Integration of equation 20 on a computer results in the required installed characteristic of the valve. The gain, $K K_v K_T$ (which must remain constant to eliminate controller retuning), is estimated from equation 19, and values assigned to the area, A, and the coefficient, C. Values representing widely varying tank parameters and thus widely varying tank dynamics have been assigned in the following table. The results of the integration of equation 20 (values of stem lift and throughput flow) are plotted as shown on Figure 15, which is similar to Figure 1 in also showing the standard flow characteristics. Note that it would be necessary to resort to equation 1 if significant differences between the inherent and installed flow characteristics exist.

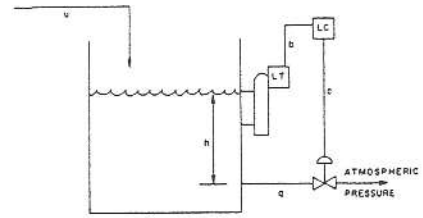


Figure 14. A LIQUID LEVEL CONTROL SYSTEM, REPRESENTED BY THE BLOCK DIAGRAM OF FIGURE 13.

TABLE III. TABULATED SETS OF TANK LEVEL SYSTEM PARAMETERS.

Parameter	*Characteristic Curve (on Figure 15)* A	B	C
h	2 ft	20 ft	2 ft
A	50 sq ft	5 sq ft	5 sq ft
q	26.8 cu ft/sec	39.9 cu ft/sec	7.9 cu ft/sec
τ_p	7.4 sec	5 sec	2.5 sec
K_p	0.22	1.6	1.0
$K K_v K_T$	0.9 ft^{-1}	0.06 ft^{-1}	0.3 ft^{-1}

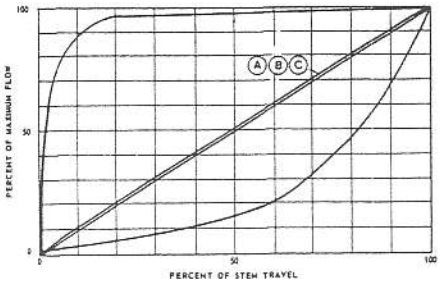


Figure 15. THE PLOTTED RESULTS OF THE INTEGRATION OF EQUATION 20, AS SHOWN IN TABLE III.

Figure 15 indicates only minor differences in the preferred valve flow characteristic for the three tabulated sets of tank level system parameters given in Table III. Further, the preferred installed characteristics are so close to being linear, that a valve with a linear characteristic would be chosen. This result conforms to the recommendations found in the literature (10,11).

The foregoing, simplified example illustrates the computational complexity in the quantitative selection of an installed valve characteristic. The complexity increases rapidly as higher order systems (such as reset in the controller) and the many possible disturbing variables are considered. Valstar (10) has dealt with this complexity by classifying the disturbances and the process dynamics, and tabulating the preferred valve flow characteristics based on these two arguments.

Quite obviously, economics dictate that most valve flow characteristics be chosen without resorting to the foregoing complex procedure, and indeed an overwhelming proportion of valve characteristics are satisfactorily chosen on the basis of the very simple rules previously listed. An even simpler rule has been suggested: (1) that is, all valves be equal percentage on the basis of (a) greater rangeability, and (b) the assumption that almost all valves have dynamic head losses in series with them which tend to convert an inherent equal percentage characteristic into a linear installed characteristic. Such a rule must be viewed as an oversimplification, however, and each application must unquestionably be given at least a cursory analysis. Ziegler and Nichols (13) show by a series of examples that linear characteristics are superior in many common applications; therefore, they take vigorous exception to the universal use of characterized valves.

Finally, Lovett (7) has differentiated between the "theoretical" and the "actual" valve flow characteristics. The "theoretical" characteristic is described by mathematical equations and graphs and is discussed in the foregoing pages. The actual inherent characteristic is determined by physical testing, in accordance with ISA Standard SP39.2, as formerly discussed.

Figure 16 shows actual flow characteristics developed experimentally by Lovett (7) and indicates the deviation of the actual characteristic from the theoretical formulation. Lovett reports that this deviation can be as much as 40%. Thus, the precise mathematical formulation of the optimum characteristic is not only prohibitive in terms of complexity, but is also futile unless the actual characteristic of a given manufacturer's valve is known.

CONCLUSION

With the foregoing difficulties in the quantitative definition of valve flow characteristics, the characteristic is typically chosen with respect to the following guidelines:

1. The valve characteristic will have little effect if (a) controller proportional band is expected to be very narrow; or if (b) the required valve rangeability is 2 to 1 or less.
2. The equal percentage flow characteristic should be used if (a) the major portion of the process system pressure drop is not available through the valve; (b) valve pressure drop is high at low flows, low at high flows; or if (c) oversizing is desirable because of limited data.
3. The linear flow characteristic should be used in flow control systems if (a) a variable head flowmeter is used; and if (b) the major portion of the process system pressure drop is available through the valve.

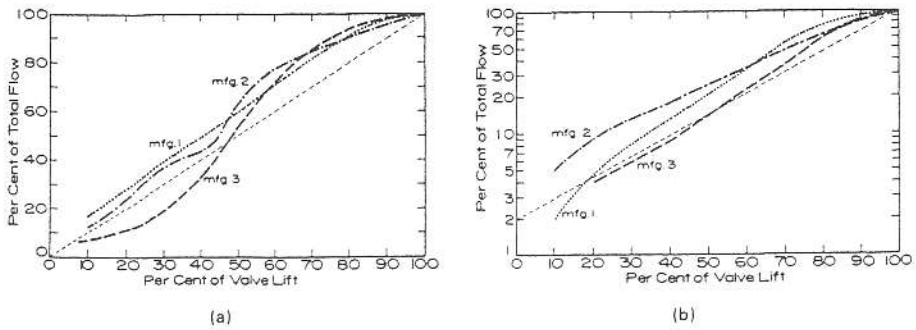


Figure 16. ACTUAL FLOW CHARACTERISTICS: a) LINEAR; b) EQUAL PERCENTAGE. (7)

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CHAPTER 6

SIZING CONTROL VALVES

PART 1
Sizing Theory and Applications
L. R. Driskell, Chemical Plants Division, Dravo Corporation, Pittsburgh, Pennsylvania

PART 2
Cavitation and Flashing Considerations
G. F. Stiles, Fisher Controls Company, Marshalltown, Iowa

with Supplementary Notes by
E. A. Bake, Rockwell International, Pittsburgh, Pennsylvania

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PART 1 SIZING THEORY AND APPLICATIONS*

(L. R. Driskell)

INTRODUCTION

The problem of sizing a control valve may be broken into several more or less distinct parts. The person sizing a valve should carefully consider the effect of these factors on valve size.

1. Flow Application Data:

   Flow rate, maximum and minimum.

   Pressure upstream and downstream (at both maximum and minimum flows).

   Temperature of the stream.

2. Fluid Data:

   Name of fluid or its generic identity.

   Fluid phase—liquid, gas, slurry, etc.

   Density (specific gravity, specific weight, molecular weight, etc.)

   Viscosity (liquids).

   Vapor pressure (liquids).

3. Piping Influence:

   Presence of reducers or other disturbances at the valve which will change the rated capacity.

4. System Influence:

   Control dynamics (is oversizing unimportant?).

   Economic factors (is downstream relief valve size affected?).

   Safety.

5. Style of Valve—Selection based on application:

   Capacity, order of magnitude of size.

   Rangeability.

   Corrosion or erosion resistance.

   Special requirements (tight shut-off, low noise, etc.)

6. Sizing Calculations:

   Manufacturer's sizing coefficients.

   Sizing formulas, slide rules, nomographs.

7. Judgment:

   Based on discriminating analysis of past successes and failures.

The first four items can be properly evaluated only by one who is knowledgeable about the whole process; therefore, this is usually the purchaser.

The principal responsibility of the valve manufacturer, in valve sizing, is to publish certified valve sizing coefficients. These values simply state the performance characteristics of his valves, based on standard test data and quality-control of production. In some instances a valve manufacturer may assume additional responsibilities. This usually involves sizing the valve based on data provided by the purchaser and may include the selection of valve type and materials as well. This additional engineering on the part of the vendor involves added work and liability on his part and involves some degree of risk on the part of the purchaser.

If there is divided responsibility, the application must fly without the last important ingredient—*good judgment.*

THEORY

To be a good aircraft pilot it is necessary to have the seat-of-the-pants feel of the ship. It is also important to know *why* the ship responds the way it does. For the same reasons, the art of valve sizing goes hand-in-hand with the science of fluid mechanics. It is helpful to know why valves work and what their limitations are, to properly apply the cookbook formulas.

Incompressible Fluids

A fluid flowing through control valves follows the same laws of conservation of mass and energy as expressed in the equations of fluid mechanics. First consider the flow of liquids, which essentially are incompressible fluids. When any fluid flowing inside a pipe, passes through a narrower passage or restriction, it must accelerate. The energy for this acceleration must be taken from the pressure of the fluid, or the *static head.* After passing the restriction, the fluid slows down again and part of this head, or pressure, is recovered. The unrecovered part of the pressure has been converted into internal energy by friction. Figure 1 shows the pressure gradient around a valve or an orifice.

Neglecting friction and other non-ideal influences for a moment, Bernoulli's theo-

* The sizing information and data contained in this chapter is based on the latest ISA Standards S-39.1 and

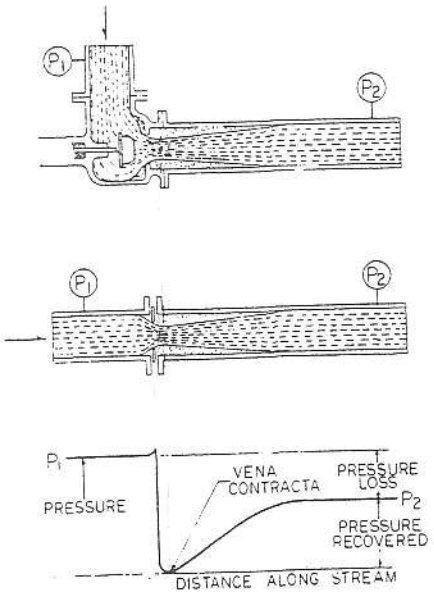

Figure 1. CONTROL VALVE AND ORIFICE PLATE INSTALLATIONS SHOWING REDUCED AREA AT VENA CONTRACTA. THE CURVE DESCRIBES THE PRESSURE GRADIENT AROUND EITHER DEVICE.

rem tells us that

$$U_2^2 - U_1^2 = 2gh \qquad (1)$$

where U_1 and U_2 are the mean axial speeds of the fluid across areas a_1 and a_2 respectively, g is the acceleration of gravity, and h is difference in head (pressure) measured in feet of the fluid.[1]

If area a_1 is the area of the inlet pipe, and area a_2 is the area of the stream vena contracta, as shown in Figure 1, Equation (1) can be converted to:

$$q_t = a_2\sqrt{2g\,\frac{p_1-p_2}{\gamma}}\;\;\frac{1}{\sqrt{1-\left(\frac{a_2}{a_1}\right)^2}} \qquad (2)$$

where q_t is the theoretical flow in cubic feet per second.

Since the actual flow through a restriction is always less than the indicated theoretical rate, and since the area of the vena contracta is less than the orifice area, an additional factor "C," called the discharge coefficient, must be included in the equation. This factor accounts for the contraction of the stream as well as the pressure lost to friction. The fundamental hydraulic equation then becomes:

$$q = a_2\overline{K}\sqrt{2g\,\frac{p_1-p_2}{\gamma}} \qquad (3)$$

where $\overline{K} = \dfrac{C}{\sqrt{1-(a_2/a_1)^2}}$

and a_2 is the orifice area.

It is well to be reminded at this point that the foregoing theoretical development is based on the assumption that p_2 is being measured at the vena contracta, or point of minimum stream area. If p_2 is being measured at a point farther downstream, as is customary in control valve work, another correction factor must be included.

$$F_L = \sqrt{\frac{p_1-p_2}{p_1-p_{vc}}} \qquad (4)$$

The relationship expressed by Equation (4) is correct only as long as the fraction of Δp_{vc} recovered remains constant. When flashing or cavitation takes place, for example, this relationship does not hold, and formulas based on p_2 must be modified.

If the fundamental hydraulic Equation (3) is converted to customary engineering units, and p_2 is being measured downstream at a point of full pressure recovery, it then becomes

$$q = \left(\frac{38.0\,a\overline{K}}{F_L}\right)\sqrt{\frac{\Delta p}{G}} \qquad (5)$$

In control valve work the value of the term in parentheses is replaced by an experimentally determined coefficient called C_v; therefore, Equation (5) becomes:

$$q = C_v\sqrt{\frac{\Delta p}{G}} \qquad (6)$$

Compressible Fluids

Liquids are nearly incompressible and, as they pass through a valve, their specific weight remains constant. Gases and vapors, on the other hand, expand as the pressure drops; therefore, their

[1]In this part of the discussion of theory all units are in standard absolute units, such as pressures in pounds per square foot, areas in square feet, etc. The working equations discussed later use customary engineering units of pounds per square inch, square inches, etc. When metric units are applicable, it will be noted.

TABLE I. NUMERICAL CONSTANTS

N	w	q	p	γ	T	d,D	ν OR μ
N_1		m³/h gpm	kPa psia			mm in	
0.0865 1.00							
N_2						mm in	centistokes centistokes
0.00214 890							
N_3		m³/h gpm				mm in	
645 1.00							
N_4				kg/m³ lb/ft³	K R		
76,000 17,300			kPa psia				
N_5		m³/h scfh			K R		
0.00241 1,000			kPa psia				
N_6	kg/h lb/hr				K R		
2.73 63.3		m³/h gpm	kPa psia				
N_7							
4.17 1,360		m³/h scfh	kPa psia		K R		
N_8	kg/h lb/hr						
0.948 19.3							
N_9		m³/h scfh	kPa psia		K R		
22.4 7320							
N_{10}		m³/h gpm	kPa psia			mm in	centipoise centipoise
1.72 52.3							

*The standard cubic foot is taken at 14.73 psia and 60°F, and the standard cubic meter at 1.013×10^5 N/m² and 15° C.

specific weight decreases as they pass from the valve inlet to the vena contracta within the valve body. The effective specific weight, as far as flow is concerned, is approximately mid-way between the value upstream at p_1 and the value at the vena contracta.

The most convenient way of accounting for the change of specific weight in sizing formulas for valves, as well as for orifice meters, flow nozzles, and venturis, is by applying an expansion factor "Y" to the formula for liquids. If the basic equation for liquids is converted to weight units it becomes:

$$w = N_6 \ C_v \sqrt{\Delta p \ \gamma_1} \qquad (7)$$

(See Table 1, page 183.)

where N_6 is the constant for units. If the fluid is compressible, Y must be included; therefore,

$$w = N_6 \ C_v Y \sqrt{\Delta p \gamma_1} \qquad (8)$$

The relationship of Y to x for any particular valve fortunately turns out to be an essentially linear curve. The slope of this curve is determined experimentally by methods prescribed by ISA Standard S39.4. With air as the test fluid, the slope of the curve is defined by the pressure drop ratio factor x_T, which is the value of x when Y reaches its lower limit of 0.667. Mathematically, when air is the fluid Y has the value: $1 - 0.33 (x/x_T)$.

The relationships of Y *vs.* x for several widely differing styles of valves are shown in Figure 2. It will be noticed that all of the plots are straight lines and that they terminate at a value of Y = 0.667. Although the true plots may deviate somewhat from a straight line, the test data to date indicate that the linear representation is within the tolerance of experimental error (about 2%). Termination at Y = 0.667 results from the fact that the ultimate flow through a restriction at any selected p_1 must occur when the quantity $Y\sqrt{x}$ reaches a maximum. With the linear relationship of Y to x, it can be demonstrated by differential calculus that $Y\sqrt{x}$ reaches a maximum at Y = 0.667.

Nitrogen, oxygen, hydrogen and other diatomic gases behave like air when passing through a valve, but some gases and vapors behave differently and require an adjustment to the slope of the Y curve. The adjustment is brought about by multiplying x_T by the correction factor F_k, which is called the ratio of specific heats factor. Table VI lists the values

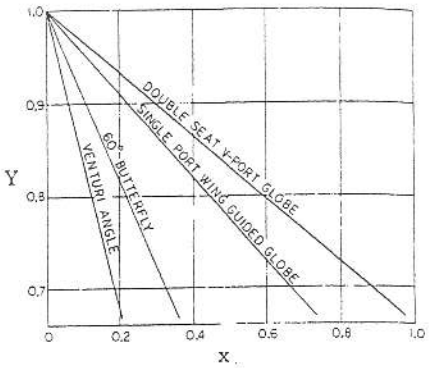

Figure 2. EXPANSION FACTOR, Y, AS A FUNCTION OF PRESSURE DROP RATIO, X, FOUR WIDELY DIFFERING STYLES OF CONTROL VALVES.

of F_k for a number of common gases, and the value for any gas can be found from Equation (19) if the ratio of specific heats for the gas is known. Taking this effect into account, the resulting formula for Y for any gas or vapor becomes:

$$Y = 1 - 0.33 (x/F_k x_T) \qquad (9)$$

and since the maximum flow is reached when Y attains it minimum value of 0.667, then the maximum useful value of x is $F_k x_T$.

Considering the foregoing, we can now arrange Equation (8) to facilitate the handling of this limitation of x:

$$w = N_6 C_v Y \sqrt{x p_1 \gamma_1} \qquad (10)$$

In this form, if x exceeds $F_k x_T$, the value of $F_k x_T$ is used in place of x, and Y becomes 0.667. If x is less than $F_k x_T$, then Y is determined by Equation (9).

If U.S. customary units are used, (units based upon the foot and the pound commonly used in the United States of America and defined by the National Bureau of Standards.) Equation (10) expresses the weight flow in pounds per hour based on the specific weight in pounds per cubic foot. If SI units are used, the mass flow rate in kilograms per hour and density in kilograms per cubic meter are the units employed. Weight units are customarily used for steam and other vapors.

For a gas, it is common to use volumetric units for flow and specific gravity relative to air instead of density or specific weight. Equation (10) may be restated as:

$$q_g = N_7 C_v p_1 Y \sqrt{\frac{x}{G_g T_1 Z}} \qquad (11)$$

Here you will notice the appearance of Z, the compressibility factor, which is now needed because the density term has become a calculated value based on the inlet pressure and temperature, and the specific gravity at base pressure and temperature. If the fluid behaves as a perfect gas, Z will be 1.0, but for real gases Z may vary widely. Compressibility charts[1] show Z ranging from 0.23 to about 4.0 which would change the predicted flow by a factor of 2 in either direction. For most real gases Z usually falls between 0.5 and 1.5, resulting in a flow 40% greater or 20% less than would be predicted if the compressibility factor were omitted. In all cases, if the actual stream density is known, the weight formula will give more accurate results.

WORKING EQUATIONS

Liquid Formulas

$$C_v = \frac{q_f}{N_1 F_p F_R} \sqrt{\frac{G_f}{\Delta p}} \quad \text{or}$$

$$q_f = N_1 F_p F_R C_v \sqrt{\frac{\Delta p}{G_f}} \qquad (12)$$

$$C_v = \frac{w}{N_6 F_p F_R \sqrt{\Delta p \gamma}} \quad \text{or}$$

$$w = N_6 F_p F_R C_v \sqrt{\Delta p \gamma} \qquad (13)$$

At this point it is appropriate to explain why certain factors have been added to the basic equation. The numerical constants N_1, N_6, etc. have been introduced to accommodate the transition to metric units. These numerical constants will, of course, have a different value depending upon whether U.S. customary units or SI units are used in the equations. (SI is the official abbreviation for The International System of Units.). The factor Fp is introduced to correct for piping geometry differing from that of the standard test manifold. Factor F_R is applied to correct for a non-turbulent flow condition, which can occur with viscous fluids or with low pressure drop across the valve. These new factors will be described in detail in the following pages. The inclusion of all the new factors certainly adds to the complexity of control valve sizing, but they are necessary if reasonable accuracy is desired.

Valve capacity is also affected by vaporization of the flowing liquid at the valve orifice. The conditions where this occurs, called cavitation or flashing, are discussed in Part 2 of this chapter.

The formulas presented in ISA valve sizing standards cannot be used blindly to come up with a properly sized valve. They are intended to be basic mathematical expressions for flow through a valve. The ISA SP39 Committee recognized that various working equations based on the standards would be developed which may be either more convenient for routine valve sizing, or perhaps more easily handled by digital computer techniques.

Example 1 *Liquid Flow*

Given:	Fluid	Brine
	Specific gravity at F.T.	1.2
	Flow rate maximum	250 gpm at F.T.
	Δp at maximum flow	10 psi
	Flow rate minimum	40 gpm at F.T.
	Δp at minimum flow	25 psi

$$C_v = q_f \sqrt{\frac{G_f}{\Delta p}} = 250 \sqrt{\frac{1.2}{10}} = 87 \quad \text{(Equation 6)}$$

$$C_v \text{ (minimum)} = 40 \sqrt{\frac{1.2}{25}} = 8.7$$

(Rangeability 10:1)

Gas and Vapor Formulas

$$C_v = \frac{w}{N_6 F_p Y \sqrt{x p_1 \gamma_1}} \quad \text{or}$$

$$w = N_6 F_p C_v Y \sqrt{x p_1 \gamma_1} \qquad (14)$$

[1] See Nelson-Obert Charts in Appendix.

$$C_v = \frac{q}{N_7 F_p p_1 Y}\sqrt{\frac{GT_1Z}{x}} \quad \text{or}$$

$$q = N_7 F_p C_v p_1 Y \sqrt{\frac{x}{GT_1Z}} \qquad (15)^1$$

$$C_v = \frac{w}{N_8 F_p p_1 Y}\sqrt{\frac{T_1Z}{xM}} \quad \text{or}$$

$$w = N_8 F_p C_v p_1 Y \sqrt{\frac{xM}{T_1Z}} \qquad (16)^1$$

$$C_v = \frac{q}{N_9 F_p p_1 Y}\sqrt{\frac{MT_1Z}{x}} \quad \text{or}$$

$$q = N_9 F_p C_v p_1 Y \sqrt{\frac{x}{MT_1Z}} \qquad (17)^1$$

$$Y = 1 - \frac{x}{3F_k x_T} \qquad (18)$$

$$F_k = k/1.40 \qquad (19)$$

For dry saturated steam (U.S. Customary Units):

$$C_v = \frac{w}{p_1\left(3-\frac{x}{x_T}\right)\sqrt{x}} \quad \text{or}$$

$$w = C_v p_1\left(3-\frac{x}{x_T}\right)\sqrt{x} \qquad (20)$$

In all the above formulas the value used for x may not exceed ($F_k x_T$), regardless of the actual pressure drop ratio.

Limitations of Equations

Equation (20) is a simplified form for steam. Results will not deviate by more than 6% from the more precise Equation (14) over the range of steam pressures from 0 to 1600 psig.

Mixed phase— For gas-liquid mixtures see the section on "Mixed Phase."

Near critical pressure—Error can become significant when the pressure drop ratio is high and the vapor is near its critical pressure. Steam begins to deviate above 1400 psia.

High Δp—Although the equations are valid for all pressure drops, extremely high outlet velocities may cause excessive noise. Pipe increasers which are normally installed at the valve outlet can cause supersonic shock waves. To reduce the possibility of excessive noise from this source, it is recommended that the valve outlet velocity be limited to one Mach, preferably as low as one-third Mach. On this basis the valve outlet (increaser inlet) size for one Mach may be determined for a gas by

$$\text{Diam.} = 0.0024\sqrt{\frac{q}{p_2}\left[\frac{T_1M}{k}\right]^{1/2}} \qquad (21)$$

or

$$\text{Diam.} = 0.047\sqrt{\frac{w}{p_2}\left[\frac{T_1}{Mk}\right]^{1/2}} \qquad (22)$$

For saturated steam -

$$\text{Diam.} = 0.012\sqrt{\frac{w}{p_2}} \qquad (23)$$

For one-third Mach multiply the diameter found above by 1.7.

Example 2 Gas or Vapor Flow

Given:

Fluid	Refrigerant R-12
Rate, w	20,000 lb/hr.
Pressure, p_1	114.5 psia
Pressure, p_2	54.5 psia
Sp. Vol, v_1	0.355 ft.³/lb.
Line size	2 inch

If the valve selected is, for example, a single-seat cage guided type, the value of x_T is about 0.75 and the C_v is about 11d².

$$F_p C_v = \frac{W}{N_6 Y\sqrt{xp_1\gamma_1}}$$

$x = 60/114.5 = 0.524$ (by definition)

$F_k x_T = (0.75)(1.14)/1.40 = 0.610$

$Y = 1-0.524/(3)(0.610) = 0.714$ (Eq. 18)

$$F_p C_v = \frac{20{,}000}{(63.3)(0.714)}\sqrt{\frac{0.355}{(0.524)(114.5)}} = 34$$

If a 2-inch valve of the style selected has a C_v of 44, it would pass the required amount at 75 to 80% opening. Reducers are not required and F_p is equal to 1.0.

[1]Equations (15), (16) and (17) are variations of Equation (14) using conversion factors for customary units.

Other Gas Equations

Other gas equations in current use are listed below. All are based on U.S. Customary units, and usually neglect the compressibility factor Z.

$$q = 963\, C_v\sqrt{\frac{(p_1-p_2)(p_1+p_2)}{GT_1}} \qquad (24)$$

with $p_2 \geq 0.5p_1$

This equation comes from the Fluid Controls Institute standards.[3] Flow predicted is almost always more than the actual flow. If the error (in addition to the compressibility error) is to be no more than 10%, the pressure drop ratio must be less than the limiting values that follow:

RATED VALVE x_T	MAXIMUM x	RATED VALVE x_T	MAXIMUM x
0.80	1.00	0.50	0.21
0.75	0.44	0.40	0.15
0.70	0.38	0.30	0.10
0.60	0.28	0.20	0.065

One manufacturer of control valves[1] uses the following formula:

$$q = C_v C_1 p_1\sqrt{\frac{520}{GT_1}}\ \sin\left[\frac{3417}{C_1}\sqrt{\frac{\Delta p}{p_1}}\right]_{\text{Deg.}} \qquad (25)$$

The bracketed expression is limited to 90°, which represents the terminal flow condition. The factor C_1 is derived by physical testing with air. It is approximately equal to $40\sqrt{x_T}$. This equation agrees with equation (15) within the limits of measurement tolerance if the compressibility factor Z is included.

Another manufacturer[2] uses this formula:

$$q = 834\ \frac{C_v C f p_1}{GT_1}\ (y-0.148y^3) \qquad (26)$$

where $y = \frac{1.63}{C_f}\sqrt{\frac{\Delta p}{p_1}} \leq 1.50$

This equation is based on the proposition that $C_f = F_L = \sqrt{x_T/0.84}$ (within testing tolerance). When C_f is determined by air testing and the factor Z is included for other gases, the accuracy of Equation (26) should compare with Equation (15).

Slide Rules and Nomographs

Many control valve manufacturers provide slide rules and nomographs to be used for valve sizing. When properly designed and used, the results of these tools are as accurate as direct calculations. They, of course, have their advantages and disadvantages. Generally these methods are faster, because all constants in the equations are incorporated into the scales and require no additional settings. The tools must be used frequently in order to avoid re-reading the procedural instructions. The units of measurement used must conform to those employed by the slide rule or nomograph. The chances of error for the various methods depends upon the calculator. Usually no record is preserved when these devices are used; therefore, checking is difficult.

LOW REYNOLDS NUMBER

The nature of flow in pipes depends upon the combination of four variables (diameter, viscosity, density, velocity) arranged to produce a dimensionless number called the Reynolds number. The value of this number determines whether the flow is laminar or turbulent, and it also affects the coefficient of discharge of an orifice.

The flow coefficient, C_v, is determined at a reasonably high Reynolds number with the fluid in turbulent flow. The C_v changes very little as the Reynolds number is increased. However, as the Reynolds number is decreased, the effective C_v becomes smaller and a correction factor must be applied. At extremely low Reynolds numbers the flow becomes laminar rather than turbulent, and the flow rate is proportional to Δp rather than $\sqrt{\Delta p}$. This change in flow mechanism is provided for by the correction factor, F_R.

The Reynolds criterion is valid only when there is precise mechanical similarity between the devices. For example, in the determination of the friction factor for pipes the "length" dimension used to calculate Reynolds number is the pipe internal diameter, but the relative roughness of the pipe wall must also be the same as that of the test specimen to validate the method. For meter-

TABLE II

F_R - REYNOLDS NUMBER FACTOR FOR TRANSITIONAL FLOW

F_R	(1) For C_v	(2) For q	(3) For Δp	F_R	(1) For C_v	(2) For q	(3) For Δp
0.46	0.46	0.47	0.46	0.74	1.35	1.70	1.16
0.48	0.49	0.50	0.48	0.76	1.50	2.10	1.27
0.50	0.52	0.53	0.51	0.78	1.67	2.40	1.40
0.52	0.56	0.57	0.54	0.80	1.90	2.90	1.55
0.54	0.59	0.62	0.57	0.82	2.20	3.40	1.72
0.56	0.63	0.67	0.61	0.84	2.50	4.3	1.95
0.58	0.68	0.73	0.65	0.86	3.00	5.6	2.30
0.60	0.74	0.81	0.69	0.88	3.75	7.9	2.70
0.62	0.80	0.89	0.74	0.90	4.9	12.	3.25
0.64	0.86	0.99	0.79	0.92	6.5	18.	4.1
0.66	0.93	1.10	0.85	0.94	9.5	31.	5.2
0.68	1.01	1.23	0.92	0.96	13.	50.	7.
0.70	1.10	1.38	0.99	0.98	20.	100.	10.
0.72	1.22	1.55	1.07				

(1) Ratio: C_v assuming turbulent flow divided by C_v assuming laminar flow.

(2) Ratio: q assuming laminar flow divided by q assuming turbulent flow.

(3) Ratio: $\sqrt{\Delta p}$ assuming turbulent flow divided by $\sqrt{\Delta p}$ assuming laminar flow.

To use table II, calculate the unknown value (C_v, q or Δp) using both the turbulent flow equation and the laminar flow equation. With the ratio of these two values as described in (1), (2) or (3) above, enter the appropriate column. The value of the number factor F_R will be found on the corresponding line. Interpolation is not warranted. For example, if C_v turbulent divided by C_v laminar is 1.0, then F_R = 0.68.

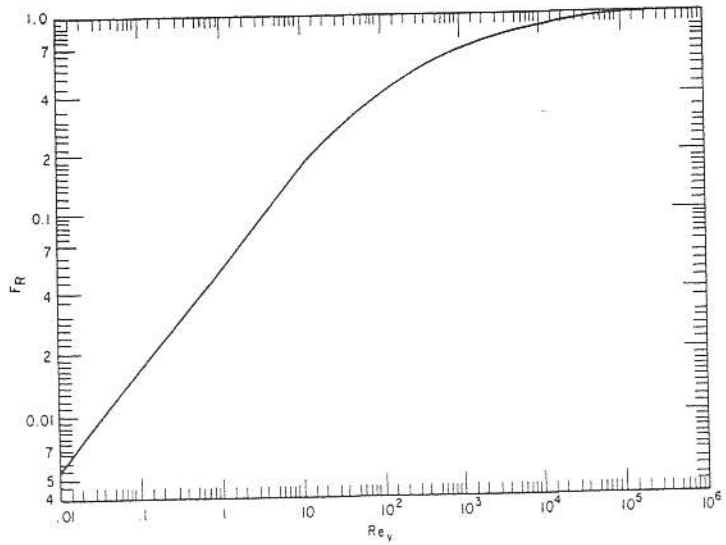

Figure 3. REYNOLDS NUMBER FACTOR FOR LIQUID SIZING.

ing orifices, the pipe internal diameter is again used as the reference for Reynolds number, but the orifice diameter ratio must also be identical to that of the test specimen if the orifice coefficients are to be the same. In addition, to ensure mechanical similarity, parameters such as pipe wall roughness, orifice plate thickness and sharpness of orifice edge must be kept within specified tolerances.

In view of the above, the use of Reynolds criterion for predicting valve flow coefficients introduces obvious problems. Mechanical similarity seems to be impossible except perhaps for valves having very simple flow paths. Correlation from size to size and style to stye presents a formidable challenge.

One method for solving the problem was used by Liptak.[4] It is applicable to special styles of valves which, when fully open, are essentially smooth circular channels matching the pipe diameter. The method rests on a good theoretical basis, but has practical limitations when applied to conventional control valves.

Correction Factor F_R

The correction for high viscosity adopted by the ISA Standards Committee in Standard S39.1 rests on the work of Stiles.[5] The linear dimension used for calculating Reynolds number is the diameter of an imaginary circle which is equal in area to the area of the stream vena contracta. The following equation is used to calculate a valve Reynolds number:

$$Re_v = \frac{N_4 F_d q}{\upsilon (F_p F_L C_v)^{1/2}} \left[\frac{(F_p F_L N_3 C_v)^2}{N_2 D^4} + 1 \right]^{1/4} \quad (27)$$

The correction factor F_R may be taken from a curve which has been experimentally determined as shown in Figure 3. The valve style modifier F_d is designed to correlate F_R versus Re_v for various valve styles. At present only two values of F_d are provided by the standard; for single-ported globe and V-notch ball valves F_d = 1.0; for double-ported globe valves, butterfly valves, and other valves with two parallel flow paths, F_d may be taken as 0.7.

The use of equation (27) to size a valve is somewhat cumbersome. A trial valve with known C_v, F_L and F_d must be chosen. From this F_p can be found and Re_v calculated. Then F_R is determined from the curve. By use of Equation (12) the maximum valve capacity q may be calculated. If this quantity is not compatible with the design flow rate, a new trial selection must be made.

Examination of Figure 3 will show that at Reynolds numbers greater than 33,000 the flow is *turbulent* and the correction factor F_R equals 1.0. At Reynolds numbers below 100 the flow is *laminar* and follows the Poiseulle-Hagen relationship, where flow is proportional to pressure drop rather than to the square root of pressure drop. A Reynolds number that is between these two values indicates the flow is *transitional*.

A formula has been derived from the ISA Standard which permits a direct calculation for C_v. In the fully laminar region:

$$q = N_{10} \frac{\Delta p}{\mu} (F_s F_p C_v)^{3/2} \quad \text{or}$$

$$F_s F_p C_v = \left[\frac{q \cdot \mu}{N_{10} \Delta p} \right]^{2/3} \quad (28)$$

The factor F_s is composed of all the sizing coefficients which are intrinsic to the valve and which affect its capacity in the laminar region. This factor is defined as

$$F_s = \left[\frac{F_d^2}{F_L} \right]^{1/3} \left[\frac{(F_{LP} C_d)^2}{N_2} + 1 \right]^{1/6} \quad (29)$$

A representative list of values for F_s is given in Table IV for various valve styles.

In the transitional zone, which is between the turbulent and laminar regions, it is possible to determine the correction factor F_R and find the required valve C_v directly. To do this, first calculate C_v for turbulent flow from equation (12), taking F_R = 1, then calculate C_v assuming laminar flow, using Equation (28). From the ratio of these two numbers we can establish the flow region and determine the factor F_R. If the C_v (turbulent)/C_v (laminar) < 0.46, the stream is fully laminar and equation (28) can be used directly. If the C_v (turbulent) / C_v (laminar) > 20, the stream is fully turbulent and equation (12) can be used directly, taking F_R = 1.0. If the C_v ratio lies between these limits, the stream is transitional and F_R can be taken from Table II and used directly in equation (12); where $q = N_1 F_R F_P C_V \sqrt{\frac{\Delta p}{G}}$

Limitation of Equations

Non-Newtonian Liquids— All gases and most simple, low-viscosity liquids are Newtonian; that is, the shear rate is directly proportional to the shear stress. Unfortunately, most high viscosity liquids such as polymers, slurries, sols, gels, colloids, etc., are non-Newtonian. For these fluids the *effective* viscosity must be used in the equations.

Tolerance - The values of F_d used here and in the ISA Standard are based on limited test data which have not yet been corroborated by independent laboratories. It is quite likely there are valve styles which do not comply with the Reynolds number curve in Figure 3.

Example 3A *High Viscosity Liquids*

Given:		
	Fluid	Newtonian Liquid
	Specific Gravity, G_f	0.90
	Pressure, p_1	85 psia
	Pressure, p_2	65 psia
	Flow rate, q	50 gpm
	Viscosity, μ	20,000 cp
	Line Size, D	3-inch Sch. 40

Assume turbulent flow (F_R=1.0):

$$F_pC_v = \frac{q_f}{N_1F_R}\sqrt{\frac{G_f}{\Delta p}} = \frac{50}{1x1}\sqrt{\frac{0.90}{20}} = 10.6 \qquad \text{(Equation 12)}$$

Assume laminar flow:

$$F_pC_v = \frac{1}{F_s}\left[\frac{q\mu}{N_{10}\Delta\rho}\right]^{2/3} \qquad \text{(Equation 28)}$$

$$= \frac{1}{F_s}\left[\frac{50 \times 20{,}000}{52 \times 20}\right]^{2/3} = \frac{97}{F_s}$$

If we try a 60 Deg. Butterfly Valve, F_s = 0.92 (Table IV)

So F_pC_v = 97/0.92 = 105

Then the C_v ratio is 10.6/105 = 0.10

Since 0.10 is less than the 0.46 limit shown in Table II, the flow is laminar and Equation 28 applies (required $F_p C_v$ = 105). The rated C_v of a 3-inch valve is $N_3C_d d^2$ = 1x17xq = 153 (Table IV). With a line size valve F_p = 1.0.

Example 3B *Transitional Flow*

Fluid	Newtonian Liquid
G	0.84
P_1	76 psia
P_2	60 psia
q	1070 gpm
μ	5900 cp
D	6-inch Sch. 40

Turbulent: $F_pC_v = \frac{q}{F_R\sqrt{\frac{G}{\Delta P}}} = 1070\sqrt{\frac{0.84}{16}} = 245$ (Equation 12)

Laminar: $F_pC_v = \frac{1}{F_s}\left[\frac{q\mu}{52\Delta P}\right]^{2/3} = \frac{1}{F_s}\left[\frac{1070 \times 5900}{52 \times 16}\right]^{2/3} = \frac{386}{F_s}$ (Eq. 28)

If we select a 60° butterfly valve, F_s = 0.92 (Table IV)

Then C_v laminar is 386/0.92 = 420
The ratio is 245/420 = 0.58
From Table II F_R = 0.53
From Eq.(12) $F_pC_v = 245/F_R = 462$

Valve selected: d = 6-inch, C_v = 612, F_p = 1.0

MIXED PHASE FLUIDS

Liquid / Gas

In this portion of the "Mixed Phase" section, the sizing procedure described applies only to homogeneous mixtures of liquids and non-condensible gases. The equations are based on the premise that the pressures within the valve body are always high enough to prevent cavitation or flashing of the liquid. If vaporization occurs, choked flow may result, and the allowable pressure drop will determine the capacity limit. (Refer to Part 2 of this chapter on cavitation and flashing.) There are no known experimental data for flashing, mixed-phase fluids.

When a liquid/gas mixture passes through a valve orifice, the liquid's specific weight remains constant. The gas, on the other hand, expands; therefore, its effective specific weight is reduced. The factor that expresses this effect is the expansion factor, Y, which has been described previously.

It will be seen from equation (14) that the weight rate of flow of a gas is proportional to $Y\sqrt{\frac{x}{v_1}} = \sqrt{\frac{x}{v_e}}$, where v_e is the effective specific volume. It follows that $v_e = v_1/Y^2$. In other words, the weight rate of flow of a compressible fluid with a specific volume of v_1 is the same as the weight rate of flow of an incompressible fluid with a specific volume of v_e when other conditons are the same.

To calculate valve C_v for liquid/gas mixtures it is only necessary to use Equation (13), replacing γ_1 with the effective value of $1/v_e$. Using U.S. Customary units, and taking F_R equal to unity:

$$F_pC_v = \frac{w}{63.3}\sqrt{\frac{v_e}{\Delta p}} \qquad (30)$$

$$v_e = \frac{f_g v_g}{Y^2} + f_f v_f \qquad (31)$$

Limitations Of The Above Formulas

1. These formulas agree with experimental data gathered on air and water mixtures in turbulent flow through small size conventional globe valves, in horizontal lines. For similar situations the error should not exceed approximately 10%.
2. The performance of valves on mixed phase flow in vertical lines has not been evaulated experimentally. If the fluid is homogeneous the formulas are theoretically valid.
3. Stratified streams, where the gas and liquid are not thoroughly mixed, are not predictable.
4. Vaporization will occur within the valve body if the pressure falls below the vapor pressure of the liquid. This can result in choked flow at some limiting value of the pressure drop ratio, x. Experimental data is unavailable; however, it is known that as the gas volume fraction of the stream approaches zero, the limit of x approaches $F_L^2(1-F_F p_v/p_1)$; where F_L is the recovery coefficient, F_F is the flashing coefficient, and p_v is the liquid vapor pressure. (Refer to Part 2 of this chapter on cavitation and flashing.) As the gas volume fraction approaches one, the limit of x approaches $F_k x_T$.

Liquid / Vapor

This part of the "Mixed Phase" section is devoted to a discussion of streams composed of a homogeneous mixture of a liquid and its own vapor. At the present state of the technology it is impossible to predict valve capacity accurately under these conditions.

Mixtures of a liquid and its own vapor present quite a different problem than liquid-gas mixtures. It is much more difficult to deal with because there is a transfer of both mass and energy between the two phases. The fact that the flow process is nearly isentropic from the valve inlet to the vena contracta, yet it is isenthalpic over-all is particularly significant to liquid-vapor flow. Depending upon the upstream conditions, the fluid may tend to evaporate or condense in the first stage, but always evaporates in the second stage.

As with liquid-gas flow, there is good reason to assume that choked flow is controlled by different factors depending upon whether liquid or vapor is in the continuous phase. The most bothersome feature of all, however, is the difficulty of assessing the inlet quality (vapor weight fraction). Not only is the quality continuously changing as the fluid moves along the pipe, but there is strong evidence to support the proposition that the fluid may exist in a metastable state. That is, the liquid and gas may not always be in thermodynamic equilibrium; either phase may be superheated temporarily. These facts are important in valve sizing, in that a small change in quality may cause a large change in density of the stream which in turn governs the

mass flow rate.

Under laboratory conditions some experimenters have found the flow of low quality steam/water mixtures through restrictions to be about ten times the amount that would be predicted by calculations based on thermodynamic equilibrium, and about twice the flow to be expected if there were no change of state between the inlet and the vena contracta. In fact in some instances the flow rate was almost twice the flow which would be predicted if the fluid were taken as completely incompressible.

The above information applies specifically to low quality fluids, that is where there is less than three percent vapor by weight. Calculations yield better results with high quality fluids where the liquid is a small fraction of a the total stream. For example, with steam and water at 100 psia pressure, there is a 71 percent difference in specific volume for a change from 1 to 2 percent quality, but only a one percent difference from 98 to 99 percent quality. Therefore error in the prediction of quality is not nearly as important with the high vapor fractions.

Considering all of these points, the use of a control valve in mixed phase liquid-vapor service is to be avoided if at all possible, particularly if the vapor fraction is small. If the situation cannot be designed out of the process this method is offered without experimental substantiation. For high quality vapor or mist, use Equations (30) and (31). For low quality fluids, use the following equation and, considering the probable error, take the specific volume as the actual specific volume of the mixture at the inlet conditions:

$$w = N_6 F_{LP} C_v \sqrt{\gamma_1 P_1 (1-F_F)} \qquad (32)$$

Example 4 *Mixed Phase Liquid-Gas*

Given:

Air Flow, w_g	460 lb./hr.
Water Flow, w_f	20,000 lb./hr.
Pressure, p_1	100 psia
Δp	36 psi
Temperature, T_1	80 + 460 = 540 R
Line Size, D	3-Inch Sch. 40

Assumed: Try a globe style valve with $x_T = 0.75$.

Calculations: $Y = 1 - \frac{x}{3F_k x_T} = 1 - \frac{0.36}{3(0.75)} = 0.84$ (Equation 9)

$$f_g = \frac{460}{20{,}460} = 0.0225,\ f_f = 0.9775$$

From the gas law: $v_g = \frac{10.72 T_1}{M p_1}$

$$V_g = \frac{10.72(540)}{29(100)} = 2.00$$

$$v_e = \frac{f_g v_g}{Y^2} + f_f v_f \qquad \text{(Equation 31)}$$

V_f is the specific volume of water at 80° F. or 0.01607 ft.³/lb.

$$v_e = \frac{0.0225(2.00)}{(0.84)^2} + 0.9775(0.01607) = 0.0795$$

$$F_p C_v = \frac{w}{63.3} \sqrt{\frac{v_e}{\Delta p}} \qquad \text{(Equation 30)}$$

$$F_p C_v = \frac{20{,}460}{63.3} \sqrt{\frac{0.0795}{36}} = 15.2 \qquad \text{(Equation 30)}$$

Select: One 2-inch restricted trim valve has a rated C_v of 20, making F_p nearly 1.0. The rated x_T of 0.72 compared with the assumed value of 0.75 is negligible.

Example 5 *Mixed Phase Liquid-Vapor*

Given:

Water, w_f	10,000 lb./hr.
Steam, w_g	5,000 lb./hr.
Pressure, p_1	100 psia
Pressure, p_2	50 psia
Line Size, D	3-Inch Inlet

Assumed: Calculations indicate the vapor is only 33% of the mass, while it is over 99% of the stream volume. It may be assumed that the stream will behave essentially as a vapor. As more water vaporizes downstream, the increased volume can be accommodated by increasing the pipe size. Try a restricted angle valve, flow to close, $x_T = 0.55$.

Calculation: For 100 psia steam, $F_k = 0.93$ (Table VI)

$F_k x_T = (0.93)(0.55) = 0.51$

Note that the maximum useful $\Delta p = 0.51 p_1 = 51$ psi

$$Y = 1 - \frac{x}{3F_k x_T} = 1 - \frac{0.50}{3(0.51)} = 0.67$$

From the steam tables, the specific volumes are as follows:

Vapor, $v_g = 4.433$ ft.³/lb. Liquid, $v_f = 0.01774$ ft.³/lb.

$$f_g = \frac{5{,}000}{15{,}000} = 0.333,\ f_f = 0.667$$

$$v_e = \frac{f_g v_g}{Y^2} + f_f v_f \qquad \text{(Equation 31)}$$

$$= \frac{0.333(4.433)}{(0.67)^2} + 0.667(0.01774) = 3.30$$

$$F_p C_v = \frac{w}{63.3} \sqrt{\frac{v_e}{\Delta p}} \qquad \text{(Equation 30)}$$

$$= \frac{15{,}000}{63.3} \sqrt{\frac{3.30}{50}} = 61$$

Select: A 3-inch valve body with 2-inch trim with $C_v = 94$ and $F_p = 1.0$.

PIPING EFFECTS

The sizing coefficients for control valves are determined from tests with the valve body mounted in a straight run of pipe which is the same diameter as the valve body. If the piping configuration is different from that used in evaluating the coefficients, the valve capacity will be different. The magnitude of this effect depends principally on the type of pipe fittings involved, how near they are to the valve and the ratio of the valve orifice area to the pipe area. This situation can be compared to the effect of piping on a metering orifice, where the same factors affect the relationship of flow to differential pressure.

Piping effects are most pronounced with "high capacity" valves such as butterfly, Saunders patent, Y-pattern, ball, or plug types, all of which have high area ratios, m.

Block valves, if they are full area gate or ball type, do not disturb the flow significantly, but if they are preceded by either a reducer or an elbow, the effect of these fittings will be transmitted. Full-area valves whose openings are not round, such as rectangular port plug cocks, may have an additional effect and cause a reduction in valve capacity greater than indicated by the calculated pressure loss. This is especially true when the cocks are preceded by reducers.

An elbow, when located close to a valve inlet, can cause significant effects in certain valves. If a reducer precedes the elbow the flow may be affected in an unpredictable manner. Even with valves of moderate capacity ($C_d < 15$) the flow may be reduced by 15 percent.

Concentric reducers adjacent to a valve, although they do not skew the velocity profile, cause a reduction in over-all flow capacity. The primary reason for this is the change in area ratio, m. Since line pressure will be measured in a pipe that has a larger cross sectional area than that of the valve body, the velocity of approach, F, has been changed, and

Example 6 Valve with Reducers—Liquid

Given:		
	Fluid	Water
	Specific Gravity, G_f	1.0
	Flow Rate, q_f	360 gpm
	Line Size, D	6 in.
	Δp	9 psi

$$F_p C_v = \frac{q_f}{N_1 F_R}\sqrt{\frac{G_f}{\Delta p}} \qquad \text{[Equation (12)]}$$

$$F_p C_v = \frac{360}{3} = 120$$

A preselected style has a $C_d = 18$, hence a 3-inch valve with a $C_v = 162$ seems reasonable. The 3 in. x 6 in. reducers being used are assumed to have resistance coefficients such that $\Sigma K = 1.5(1-d^2/D^2)^2 = 0.84$ [Abrupt transition]

$$F_p = \frac{1}{\sqrt{1+\frac{\Sigma K}{N_2}\left(C_d\right)^2}} = \frac{1}{\sqrt{1+\frac{(0.84)(18)^2}{890}}} \qquad \text{[Equation (33)]}$$

$$F_p = 0.87$$

$$F_p C_v = (0.87)(162) = 141$$

At the required flow, the valve will be operating at 85% of capacity, which is satisfactory.

F is a component of the valve flow coefficient C_v. In addition, there are losses due to wall friction and losses due to the fluid friction caused by the contraction and expansion at the reducers.

Several methods have been proposed to correct for the effect of reducers. One method is straightforward; it simply depends upon actual valve testing to produce new coefficients for the combination of valve plus reducers. A second method, based on theory, relies on calculation to produce correction factors. The advantage of the second method is that it can be applied generally to all types of valves, for reducers with any specified taper, and even for sets of reducers of different sizes.

The formulas for arriving at the correction factors to be applied when valves are installed with reducers are listed below. In the case of liquids, two factors are affected: the valve capacity is modified by the factor F_p and the pressure recovery is altered, requiring a corrected F_L factor.

$$F_p = \frac{1}{\sqrt{1+\frac{\Sigma K}{N_2}\left(C_d\right)^2}} \qquad (33)$$

Where $\Sigma K = K_1 + K_{B_1} + K_2 - K_{B_2}$ (the coefficients of the inlet and outlet reducers respectively). The Bernoulli coefficients K_{B_1} and K_{B_2} may be omitted if the inlet and outlet reducers are the same size, since one term is positive and the other negative. If the reducers are of different size, or if there is only one reducer, they must be included. As a sim-

Example 7 Valve with Reducers—Gas

Given:		
	Fluid	Sulfur Dioxide Gas
	Molecular weight, M	64
	Ratio of sp. heats, k	1.25
	Critical pressure, p_c	1142 psia
	Critical temperature, T_c	315°F. = 775 R
	Pressure, p_1	15 psig = 29.7 psia
	Pressure, p_2	5 psig = 19.7 psia
	Flow rate, w	185,000 lb./hr.
	Temperature, T_1	150°F. = 610 R
	Line Size, D	18-inch, Sch. 40

Basis: Since both the factors Y and F_p vary with the size and type of valve selected, it is well to proceed with the calculation to a point where we can better choose.

Calculation:

$$w = N_8 F_p C_v P_1 Y\sqrt{\frac{xM}{T_1 Z}} \qquad \text{(Equation 16)}$$

$N_8 = 19.3$ (Table 1)

$x = \Delta p/p_1 = 10/29.7 = 0.34$

For Z: $p_r = 29.7/1142 = 0.026$

$T_r = 610/775 = 0.787$

$Z = 0.98$ (Appendix)

Substituting

$$YF_pC_v = \frac{185{,}000}{19.3(29.7)}\sqrt{\frac{610(0.98)}{0.34(64)}} = 1690$$

If we select a 60-deg. butterfly valve $x_T = 0.38$ (Table IV)

Then Y will be close to 0.67 and $F_pC_v = 2500$

Try a 14-inch size, $C_v = 3330$

with $d/D = 14/18 = 0.78$

$F_p = 0.96$, and $x_{Tp} = 0.38$ (Table III)

$F_k = k/1.40 = 1.25/1.40 = 0.89$ (Equation 19)

$F_k x_{Tp} = 0.34$ (Note: This is the limiting value of x)

$Y = 0.67$

From the above, required $C_v = \frac{1690}{YF_p} = \frac{1690}{0.67(0.96)} = 2630$

plifying assumption, reducers may be taken conservatively as abrupt transitions, in which case

$$K_1 = 0.5\left[1 - \frac{d^2}{D^2}\right]^2 \text{ and } K2 = 1.0\left[1 - \frac{d^2}{D^2}\right]^2$$

The factor F_L is described as the "pressure recovery" factor. It is related to the pressure which is recovered downstream of the vena contracta in this manner: if $(p_1 - p_{vc})$ is the maximum pressure drop across the orifice, the fraction of this pressure drop which is recovered is $(1-F_L^2)$. The unrecovered fraction, or permanent loss, is F_L^2. If cavitation or flashing exists, this relationship no longer exists since pressure recovery may be diminished or reduced to zero.

When a valve is installed adjacent to an upstream reducer or other fitting, the new value of F_L for the valve-fitting combination may be calculated using the method described in ISA Standard S39.1:

$$F_L = \frac{F_L'}{F_p}\left[\frac{(F_L')^2 K_i}{N_2}\left(C_d\right)^2 + 1\right]^{-\frac{1}{2}} \quad (34)$$

Where F_L' is the rated value for the valve alone and K_i is the head loss coefficient $(K_1 + K_{B1})$ of the inlet fitting. What would seem to be a simpler and more direct procedure is to combine the F_L and F_p factors for the valve-fitting combination into a single factor F_{LP} where

$$F_{LP} = \left[\frac{1}{(F_L)^2} + \frac{K_i}{N_2}\left(C_d\right)^2\right]^{-\frac{1}{2}} \quad (35)$$

Where F_L is the rated value for the valve.

In the case of compressible fluids, such as gas or vapor, the expansion factor Y and its determiner x_T are affected by reducers. The corrected value of x_T becomes:

$$X_{TP} = \frac{x_T}{F_p^2}\left[1 + \frac{x_T K_i}{N_5}\left(C_d\right)^2\right]^{-1} \quad (36)$$

ee Table III for calculated values.

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TABLE III
CALCULATED VALUES OF F_P AND x_{TP} FOR VALVES INSTALLED BETWEEN SHORT PIPE REDUCERS ASSUMING TWO REDUCERS OF THE SAME SIZE WITH ABRUPT CHANGE IN AREA.

Cd	10						15						20						25						30					
x_T	.40	.50	.60	.70	.80		.40	.50	.60	.70	.80		.40	.50	.60	.70			.20	.30	.40	.50			.15	.20	.25			
d/D	x_{TP}					F_P	x_{TP}					F_P	x_{TP}					F_P	x_{TP}					F_P	x_{TP}					F_P
.80	.40	.49	.59	.69	.78	.99	.40	.49	.58	.67	.75	.98	.39	.48	.56	.64		.96	.21	.30	.39	.47		.94	.17	.21	.26			.91
.75	.40	.50	.59	.69	.78	.98	.40	.49	.58	.67	.75	.97	.40	.49	.57	.65		.94	.22	.31	.40	.48		.91	.18	.23	.27			.88
.67	.40	.50	.60	.69	.78	.98	.41	.50	.59	.68	.76	.95	.42	.51	.59	.67		.91	.24	.33	.43	.51		.87	.19	.25	.30			.83
.60	.41	.51	.60	.70	.79	.97	.42	.52	.61	.69	.78	.93	.43	.53	.61	.69		.89	.25	.36	.45	.54		.84	.21	.27	.32			.79
.50	.41	.52	.61	.70	.80	.96	.44	.53	.63	.71	.79	.91	.46	.55	.64	.72		.85	.28	.39	.49	.58		.79	.24	.30	.36			.73
.40	.42	.52	.62	.71	.80	.95	.44	.55	.65	.74	.82	.89	.49	.58	.67	.75		.82	.30	.42	.53	.62		.76	.26	.33	.40			.70
.33	.43	.53	.62	.72	.81	.94	.46	.56	.66	.75	.83	.88	.50	.60	.69	.78		.81	.31	.44	.55	.64		.74	.27	.34	.40			.69
.25	.44	.53	.63	.73	.83	.93	.48	.58	.67	.76	.85	.87	.52	.62	.71	.79		.79	.33	.46	.57	.67		.72	.27	.37	.44			.65

EXAMPLE: A 2-inch valve is rated at Cv = 80 and $x_T = 0.65$. Find F_p and x_{TP} if the valve is installed in a 3-inch pipe line with short reducers. $C_d = Cv/d^2$ or 20 and d/D = 2/3 or 0.67. Under the heading Cd = 20 on line d/D = 0.67 find $F_p = 0.91$. For x_{TP} interpolate the values given in the columns headed $x_T = 0.60$ and $x_T = 0.70$ (0.59 and 0.67 respectively). The answer is $x_{TP} = 0.63$.

TABLE IV
VALVE SIZING DATA
REPRESENTATIVE VALVE FACTORS

BODY & TRIM TYPE	FLOW DIRECTION	LINE SIZE BODY (D=d)						HALF SIZE (D=2d)			
		C_d	F_L	X_T	F_d[1]	F_s	K_c	$\frac{N_3 C_v}{D^2}$	F_{LP}	x_{TP}	F_s
Single Seat Globe											
Wing Guided	Either	11	.90	.75	1.0	1.05	c	2.8	.85	.75	1.04
V-Skirt	Either	9	.90	.75	1.5	1.38	c	2.3	.86	.75	1.36
Contoured	Open	11	.90	.72	1.0	1.05	.65	2.8	.85	.73	1.04
Contoured	Close	11	.80	.55	1.0	1.09	.58	2.8	.76	.57	1.08
V-Plug	Either	9.5	.90	.75	1.0	1.05	.80	2.4	.86	.75	1.04
Cage	Open	14	.90	.75	1.0	1.06	.65	3.5	.82	.75	1.04
Cage	Close	16	.80	.70	1.0	1.11	c	4.0	.72	.71	1.08
Double Seat Globe											
Wing Guided	--	14	.90	.75	.71	0.84	c	3.5	.82	.75	0.83
V-Skirt	--	13	.90	.75	.71	0.84	c	3.3	.83	.75	0.83
Contoured	--	13	.85	.70	.71	0.85	.70	3.3	.79	.71	0.84
V-Plug	--	12.5	.90	.75	.71	0.84	.80	3.1	.83	.75	0.84
Angle											
Full Port Contour	Close	20	.80	.65	1.0	1.12	.53	5.0	.69	.68	1.08
Full Port Contour	Open	17	.90	.72	1.0	1.08	.64	4.3	.78	.73	1.04
Restricted Contour	Close	≥6	.70	.55	1.0	1.13	c	1.5	.69	.56	1.13
Restricted Contour	Open	≥5.5	.95	.80	1.0	1.02	c	1.3	.93	.80	1.02
2:1 Tapered Orif.	Close	12	.45	.15	1.0	1.31	c	3.0	.44	.17	1.31
Cage	Open	12[a]	.85	.65	1.0	1.08	c	3.0	.80	.66	1.06
Cage	Close	12[a]	.80	.60	1.0	1.10	c	3.0	.75	.62	1.08
Venturi	Close	22	.50	.20	1.0	1.29	.17	5.5	.46	.26	1.26
Ball											
Std. Bore[b]	--	30	.55	.15	1.0	1.28	.25	7.5	.47	.24	1.22
Characterized	--	25	.57	.25	1.0	1.25	.22	6.3	.50	.33	1.21
Butterfly											
60-Deg. Open	--	17	.68	.38	.71	0.92	.3	4.3	.63	.43	0.91
90-Deg. Open	--	>30	.55	.20	.71	1.01	c	>7.5	.45	.33	0.97

[1]The values of F_d are based on limited test data which have not been corroborated by independent laboratories. F_s is computed from F_d. Key: a = Variable, b = Orif. = 0.8d, c = Unavailable.

TABLE V
SUMMARY OF FORMULAS ARRANGED FOR CONVENIENCE

REMARKS	EQUATIONS	VALUE OF N[1] U.S.	VALUE OF N[1] SI
LIQUID Turbulent and Non-Cavitating	$q_f = N_1 F_p C_v \sqrt{\frac{\Delta p}{G_f}}$	1.00	0.0865
	$w_f = N_6 F_p C_v \sqrt{\Delta p \gamma}$	63.3	2.73
	$q_f = N_1 F_{LP} C_v \sqrt{\frac{p_1 - p_{vc}}{G_f}}$	1.00	0.0865
	$w_f = N_6 F_{LP} C_v \sqrt{\frac{p_1 - p_{vc}}{G_f}}$	63.3	2.73
Choked	$p_{vc} = F_F p_v$ $F_F \simeq 0.96 - 0.28 \sqrt{\frac{p_v}{p_c}}$		
	$F_{LP} = \left[\frac{1}{F_L^2} + \frac{K_i}{N_2} \left(C_d \right)^2 \right]^{-1/2}$	890	0.00214
	K_i = (See Piping Geometry Factor)		
	$q_f = N_{10} \frac{\Delta p}{\mu} \left(F_s F_p C_v \right)^{3/2}$	52	173
Laminar	$F_s = \left(\frac{F_p F_d^2}{F_{LP}} \right)^{1/3} \left[\frac{(F_{LP} C_v)^4}{N_2 D^4} + 1 \right]^{1/6}$	890	0.00214
Transitional	$q_f = N_1 F_R F_P C_v \sqrt{\frac{\Delta p}{G_f}}$	1.00	0.0865

[1]U.S. units are: pounds per hour, gallons per minute, pounds per square inch absolute, pounds per cubic foot, °R, and inches.

SI units are: kilograms per hour, cubic meters per hour, kPa, kilograms per cubic meter, °K, and millimeters.

TABLE V -(CONT'D.)

REMARKS	EQUATIONS	VALUE OF N U.S.	SI
Gas or Vapor - (All Equations: $x \leq F_k x_T$)			
	$w_g = N_6 F_p C_v Y \sqrt{x p_1 \gamma_1}$	63.3	2.73
	$q_g = N_7 F_p C_v p_1 Y \sqrt{\frac{x}{G_g T_1 Z}}$	1360	4.17
Variations for Selected units.	$w_g = N_8 F_p C_v p_1 Y \sqrt{\frac{xM}{T_1 Z}}$	19.3	0.948
	$q_g = N_9 F_p C_v p_1 Y \sqrt{\frac{x}{MT_1 Z}}$	7320	22.4
Expansion factor lower limit = 0.667	$Y = 1 - \frac{x}{3F_k x_T}$		
Sp. ht. ratio factor	$F_k = k/1.40$		
Mfr's. Factors	$x_T = \frac{C_1^2}{1600} = 0.84 C_f^2$		
x_T with reducers	$x_{TP} = \frac{x_T}{F_p^2}\left[\frac{x_T K_i}{N_5}\left(C_d\right)^2 + 1\right]^{-1}$	1000	0.0024
	K_i = (See Piping Geometry Factor)		
Steam (Dry and Saturated)			
For $x < x_{Tp}$	$w = NF_p C_v p_1 \left(3 - \frac{x}{x_{TP}}\right)\left(\sqrt{x}\right)$	1.0	0.152
For $x \geq x_{Tp}$ (Choked Flow)	$w = NF_p C_v p_1 \sqrt{x_{TP}}$	2.0	0.304

TABLE V - (CONT'D.)

REMARKS	EQUATIONS	VALUE OF N U.S.	SI
Piping Geometry Factor For F_{LP} see "Liquid Choked Flow"	$F_p = \left[\frac{\sum K}{N_2}\left(C_d\right)^2 + 1\right]^{-1/2}$	890	0.00214
Sum of velocity head coefficients	$\sum K = K_1 + K_2 + K_{B1} - K_{B2}$		
Bernoulli coefficient	$K_{B1} = K_{B2} = 1 - \left(\frac{d}{D}\right)^4$		
Resistance coefficients for abrupt transitions	$K_1 = 0.5\left[1 - \left(\frac{d}{D}\right)^2\right]^2$		
Inlet fitting coefficient for F_{LP} and x_{TP}	$K_2 = 1.0\left[1 - \left(\frac{d}{D}\right)^2\right]^2$		
	$K_i = K_1 + K_{B1}$		
Line Velocity	**Feet/Second**	**Meters/Second**	**Range (Ft./Sec.)**
Liquid	$U = \frac{q}{2.45D^2}$	$U = 354\frac{q}{D^2}$	5-10 Norm. 40-50 Max.
Gas	$U = \frac{qT}{695pD^2}$	$U = 1.24\frac{qT}{pD^2}$	250-400
Vapor	$U = \frac{w}{19.6\gamma D^2}$	$U = 354\frac{W}{\gamma D^2}$	70 Wet 300 superheated
Steam	$U = \frac{23w}{pD^2}$	$U = 685\frac{W}{pD^2}$	
Acoustic Velocity (Mach 1.0)			
Gas	$U_a = 223\sqrt{\frac{kT}{M}}$	$U_a = 91\sqrt{\frac{kT}{M}}$	< 0.3 Mach
Air	$U_a = 49\sqrt{T}$	$U_a = 20\sqrt{T}$	
Steam, Superheated	$U_a = 60\sqrt{T}$	$U_a = 24.5\sqrt{T}$	<0.15 Mach
Steam, Dry Saturated	$U_a = 1650$	$U_a = 500$	<0.10 Mach
Vapor	$U_a = 68.1\sqrt{kpv}$	$U_a = 1038\sqrt{kpv}$	

TABLE VI
REFERENCE DATA FOR STEAM AND GASES

	P_{l1}	SP. GRAVITY G	SP. HEATS RATIO k	FACTOR F_k
Acetylene		0.897	1.28	0.914
Air		1.000	1.40	1.00
Ammonia		0.587	1.29	0.921
Argon		1.377	1.67	1.19
Carbon Dioxide		1.516	1.28	0.914
Carbon Monoxide		0.965	1.41	1.01
Ethylene		0.967	1.22	0.871
Helium		0.138	1.66	1.19
Hydrogen Chloride		1.256	1.40	1.00
Hydrogen		0.0695	1.40	1.00
Methane		0.553	1.26	0.900
Methyl Chloride		1.738	1.20	0.857
Nitrogen		0.966	1.40	1.00
Nitric Oxide		1.034	1.40	1.00
Nitrous Oxide		1.518	1.26	0.900
Oxygen		1.103	1.40	1.00
Sulphur Dioxide		2.208	1.25	0.893
Steam (dry saturated)				
	0-80		1.32	0.94
	80-245		1.30	0.93
	245-475		1.29	0.92
	475-800		1.27	0.91
	800-1050		1.26	0.90
	1050-1250		1.25	0.89
	1250-1400		1.23	0.88

NOMENCLATURE

a	Area of orifice or valve opening, $in.^2$
C	Coefficient of discharge, dimensionless. Includes effect of jet contraction and Reynolds number, mach number (gas at high velocities), turbulence.
C_d	Relative capacity (at rated C_v) $C_d = N_3 C_v/d^2$.
c_p	Specific heat at constant pressure.
c_v	Specific heat at constant volume.
C_v	Valve coefficient, $38\, a\overline{K}/F_L$.
d	Valve inlet diameter, inches or mm.
D	Pipe diameter, inches or mm.
F	Velocity of approach factor $= \frac{1}{\sqrt{1-m^2}}$.
F_d	Experimentally determined factor relating valve C_v to an equivalent diameter for Reynolds number. (See Table IV)
f	Weight fraction
F_F	Liquid critical pressure ratio factor, $F_F = p_{vc}/p_v$
F_k	Ratio of specific heats factor.
F_L	Pressure recovery factor. When the valve is not choked: $F_L = \sqrt{(p_1-p_2)\,/\,(p_1-p_{vc})}$
F_{LP}	Combined pressure loss and piping geometry factors for valve/fitting assembly.
F_P	Correction factor for piping around valve (e.g. reducers) $F_P C_v$ = effective C_v for valve/fitting assembly.
F_R	Correction factor for Reynolds number, where $F_R C_v$ = effective C_v.
F_s	Laminar, or streamline, flow factor.
g	Acceleration due to gravity.
G_f	Specific gravity of liquids at flowing temperature relative to water at 60°F or 15°C.
G_g	Specific gravity of gas relative to air with both at standard temperature and pressure.
h	Effective differential head, height of fluid.
$\overline{K}$	Flow coefficient = CF, dimensionless.
ΣK	Sum total of effective velocity head coefficients where $K(U^2/2g) = h$.

K_c	Cavitation index. Actually the ratio $\Delta p/(p_1-p_v)$ at which cavitation measurably affects the value of C_v.
K_i	Inlet velocity head coefficient, $K_1 + K_{B1}$
K_1	Resistance coefficient for inlet fitting.
K_2	Resistance coefficient for outlet fitting.
k	Ratio of specific heats of a gas = c_p/c_v, dimensionless.
M	Molecular weight.
m	Ratio of areas.
N	Numerical constant (See Table I)
p	Absolute static pressure.
p_c	Thermodynamic critical pressure
p_r	Reduced pressure, p/p_c
P_v	Vapor pressure of liquid at inlet.
q	Volume rate of flow.
Re_v	Reynolds number for a valve.
T	Absolute temperature.
T_c	Thermodynamic critical temperature
T_r	Reduced temperature, T/T_c
U	Average velocity.
v	Specific volume ($1/\gamma$).
w	Weight rate of flow.
x	Ratio of differential pressure to absolute inlet static pressure, $x = (p_1 - p_2) / p_1$
x_T	Terminal or ultimate value of x, used to establish expansion factor, Y.
x_{TP}	Value of x_T for valve/fitting assembly.
Y	Expansion factor. Ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number (includes radial as well as longitudinal expansion effects).
Z	Compressibility factor. (See Nelson-Obert Compressibility Charts in Appendix.)
γ (gamma)	Specific weight (1/v).
Δ (delta)	Difference, (eg., $\Delta p = p_1 - p_2$).

μ (mu)	Viscosity, centipoise.
υ (nu)	Kinematic viscosity, centistokes
Σ (sigma)	Summation

SUBSCRIPTS

1	Upstream
2	Downstream
e	Effective value
f	Liquid
g	Gas
t	Theoretical
T	Terminal or ultimate value
vc	Vena contracta (point of minimum jet stream area)

PART 2 CAVITATION AND FLASHING CONSIDERATIONS

(G. F. Stiles)

INTRODUCTION

Two phenomena often encountered in liquid flow, which are completely absent in gas or vapor flow, are cavitation and flashing. These phenomena are of significant interest in any comprehensive discussion of control valves since their occurrence will affect the valve sizing procedures, may introduce noise and vibration, and also may limit the life expectancy of the valve components and the immediate downstream piping. While there are definite similarities between cavitation and flashing there are also important differences so that it will be necessary to distinguish between them at the outset of this discussion.

CAVITATION AND FLASHING DEFINED

Cavitation is a two-stage phenomenon, the first stage of which is the formation of voids or cavities within the liquid system. The second stage is the collapse or implosion of these cavities back into an all-liquid state. Some have defined cavitation as merely the formation of cavities but this seems to be an impractical definition from a control valve standpoint, because most of the effects and evidences of cavitation are related directly to the collapse, rather than the formation, of the cavities.

Investigators have theoretically identified and defined two types of cavitation; gaseous and vaporous. Although a rigorous analysis of the difference between these is beyond the scope of this treatment of the subject, it is now understood that both types require the presence of some nucleating agent for their inception. These very tiny nuclei, which will contain either vapor or dissolved gas, or perhaps both, will enlarge into finite cavities within the liquid. Even though it is agreed that nuclei are required, the explanation of what they are or how they work is not completely understood. In one popular theory, dust particles of microscopic dimension are believed to act as hosts for the gas or vapor. Most industrial liquids contain various solid contaminants so that the nuclei necessary for cavitation inception are almost universally present.

Based on their physical structures, ideal liquids would have tensile or fracture strengths of at least the order of tens of thousands of pounds per square inch. By contrast, the first stage of cavitation is observed in most industrial liquids when the local pressure drops to the vapor pressure of the liquid. Several investigators have shown that the inception of cavitation may be delayed significantly in test liquids which have been carefully degassed or held under high pressures for sufficient time for gas and vapor to be completely dissolved.

If the cavitation process were to be halted before the completion of the second stage, so that vapor persists downstream of the region where bubble collapse normally takes place, the process is known as flashing. Obviously, since flashing is directly related to the first stage of cavitation the statements regarding the inception of cavitation also apply to flashing.

CONTROL VALVE ANALOGY

In order to visualize how these phenomena occur in control valves, consider a liquid flowing in a piping system in which a restriction such as a concentric orifice has been placed. In this case the orifice may be considered analogous to a control valve at some fixed opening. The upper part of Figure 4 illustrates the system and, in the lower part, the pressures and velocities are plotted for points along the flow stream.

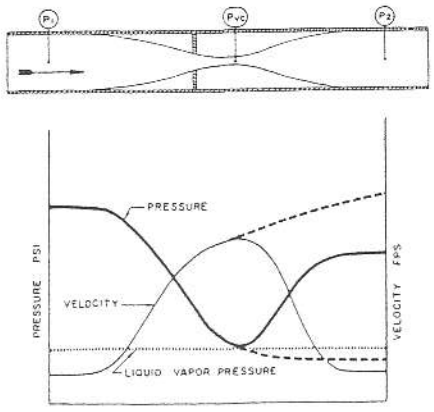

Figure 4. PRESSURE AND VELOCITY CHANGES CAUSED BY A RESTRICTION IN A LINE.

At point P_1, upstream of the orifice, assume the liquid stream to have some static pressure above its vapor pressure. Also assume some flow rate so that the stream velocity may be determined. The total energy level of the stream (ignoring any elevation factor) may then be obtained by summing the contributions of the pressure and velocity written in some dimensionally homogeneous form, feet of head, for example.

The *First Law of Thermodynamics* states that, if no energy is lost through the boundaries of the system in the form of work or heat, the total energy level will remain constant at every point along a given datum plane in the system.

As the fluid stream approaches the restriction in the line its cross-sectional area must decrease in order to pass through the orifice. The velocity is inversely proportional to the stream area and, therefore, must increase. Since the sum of the pressure and velocity heads will remain approximately equal, an energy interchange must take place with the pressure head losing what the velocity head gains.

Immediately downstream of the orifice the stream will reach its minimum cross section and thus its maximum velocity and minimum pressure. This point is called the vena contracta (P_{vc}). If the velocity is increased sufficiently, the pressure will fall to the vapor pressure, thus permitting the formation of voids in the stream, which is the first stage of cavitation.

Downstream from the vena contracta, fluid friction causes the stream to decelerate with resultant increases in both stream cross section and pressure. This reversal of the energy interchange between the velocity and pressure heads is called "pressure recovery" and, as we shall see later, plays an important role in valve sizing. Vapor bubbles, formed as a result of reducing the pressure at the vena contracta to the vapor pressure, cannot exist at the increased pressure downstream and are forced to collapse or implode back into the liquid state. When this has occurred, the cavitation process is complete.

If the pressure in the downstream piping system were being maintained at a level equal to or less than the inlet vapor pressure, the fluid proceeding downstream would have an increasing percentage of vapor, the velocity of the stream would continue to increase and the end result would be flashing rather than cavitation.

REQUIREMENTS FOR OCCURRENCE

As cavitation and flashing have been defined, the following restrictive statements may be made regarding their occurrence in control valves.

Cavitation

1. The fluid at both the inlet and outlet must be in an all-liquid condition. That is, there can be no vapor present in the piping upstream or immediately downstream of the valve.
2. The liquid must be in a sub-cooled state at the inlet. Obviously if the fluid were all liquid at the inlet, but in a saturated state, any pressure drop across the valve would result in residual vapor present downstream.
3. The valve outlet pressure must be either at or above the vapor pressure of the liquid. Conceivably, it would be possible for cavitation to exist if a saturated, but all-liquid, condition were present at the downstream location.

Flashing

1. The fluid at the inlet must be in an all-liquid condition while some vapor must be present at the valve outlet. Obviously, if vapor were present at the inlet, any pressure differential taken across the valve would result in the formation of additional vapor. This restriction, requiring absence of vapor at the inlet, is imposed primarily because the valve sizing procedures are considerably more complicated when the inlet stream contains vapor. Valve sizing procedures, which will be discussed later, will be limited to the all-liquid inlet condition.
2. The fluid at the inlet may be in either a saturated or a subcooled condition.
3. The valve outlet pressure must be either at or below the vapor pressure of the liquid.

CAVITATION EVIDENCES

The occurrence of cavitation in a control valve is always accompanied by one or more of several characteristic evidences which include noise, vibration, physical damage and a decrease in efficiency. The numbers manifested, and their magnitudes, are dependent upon the proximity of the instantaneous operating point to the point of incipient cavitation. All of the effects of cavitation can be traced to the implosion of the vapor bubbles, with the exception of the decrease in efficiency which is traceable to their formation.

Noise

In a control valve the evidence of incipient cavitation is usually a hissing sound, sometimes intermittent, emanating

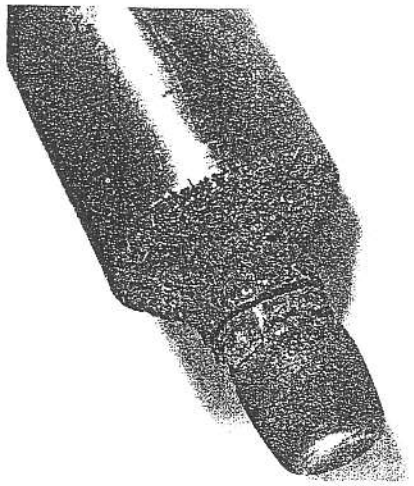

Figure 5. EXAMPLE OF CAVITATION DAMAGE TO A VALVE PLUG.
Courtesy Fisher Controls Company

from the downstream side of the valve body. In its very early stages, however, this sound may be masked by the normal flow noises. As the intensity of the cavitation increases, due to increasing pressure differentials, the sound level also increases. Fully developed cavitation in a control valve produces a rattling sound which gives the listener the impression that gravel is passing through it.

Vibration

The physical process which produces noise also results in vibration. How serious a problem this will be, will depend upon several factors, including the mass of the system, how well the system components are anchored, and whether valve-mounted instruments are vibration-sensitive. Field reports on a variety of control valve types indicate that, in severe cavitating service, these vibrations can reach serious proportions.

Material Damage

Probably the most dramatic effect of cavitation is the material damage which it inflicts. Under severe cavitating conditions, extremely hard control valve components have failed in a matter of a few hours time. An example of this is shown in Figure 5 which pictures a valve plug for a conventional globe style body that had been subjected to severe cavitating conditions on water service in a power generating station. In this example, the plug tip material was solid Stellite No. 6, a material which has been shown to be relatively high in resistance to cavitation damage, while the beveled seating area was 18-8 stainless steel overlaid with Stellite No. 6.

Figure 6 pictures two identical valve plugs machined from Alloy 1100-F (commercially pure) aluminum. The plug on the left is in the untested condition whereas the plug on the right has been subjected to cavitating cold water for a ten-minute period with the valve inlet pressure maintained at 2000 psig. The pressure drop across the valve was 2000 psi. In this case the very soft material was selected to illustrate that substantial metal loss can occur from the effects of cavitation even though exposure times are very short.

Numerous examples could be shown where cavitation has eroded holes completely through valve body or downstream piping walls, has destroyed valve plug seating surfaces and flow contours, and has ruined the guiding surfaces of post guided plugs.

In general it may be stated that resistance to cavitation damage increases with increasing hardness. Numerous investigators have studied this problem and have made attempts to correlate damage resistance on the basis of various material properties including tensile strength, modulus of elasticity and strain energy in addition to hardness. Up to the present

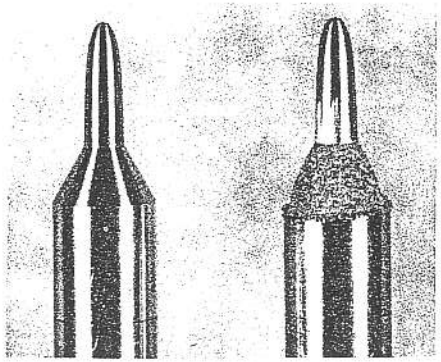

Figure 6. BEFORE AND AFTER APPEARANCE OF A SMALL VALVE USED IN CAVITATION DAMAGE TESTS.
Courtesy Fisher Controls Company

time no single property has been found which will yield good correlations for all experimental data. Other considerations, especially corrosion resistance, often preclude the use of the hardest available material in a particular application.

In the relatively hard materials from which control valve components are normally fabricated, cavitation damage is uniquely characterized by a very rough cinder-like appearance which makes it easily distinguishable from most other types of flow damage that are usually quite smooth in appearance.

Materials have been evaluated by a variety of methods for their resistances to cavitation damage. One method which has been used quite extensively employs a magnetostrictive* device to induce rapid oscillations of a test specimen supported in some test fluid. The oscillating motion of the specimen produces alternate high and low pressures in the fluid, forcing it to vaporize and reliquify at the driving frequency. Cavitation thus produced, damages the test specimen. Various test materials may be evaluated by comparing the relative weight losses for a given test time.

Another method that has been employed to duplicate actual control valve operating conditions, involves machining small valve plugs, as illustrated in Figure 6, from the desired test materials and subjecting them to identical flowing cavitating conditions. The following table of relative resistances of several common valve body and trim materials was developed from tests of this type. A sample of Type 316 stainless steel, a popular body and trim material, was tested until it had received a certain amount of damage. The time required for this damage was noted. Subsequently, samples of other materials were tested until they had achieved approximately the same magnitude of metal loss. Values of an "Index of Resistance to Cavitation Damage" were determined by dividing the times required to produce damage in the various materials by the time required to produce equivalent damage in the Type 316 stainless steel sample (See Table III).

TABLE III
INDEX OF RESISTANCE TO CAVITATION DAMAGE

Material	*Hours Tested*	*Index*
Stellite No. 6 over Type 316	120	20*
Die Steel (Latrobe BR-4FM)	120	20*
17-4 PHSS Hardened to 45 RH-C	12	2*
Type 316 SS Electrolized	8	1.33
Type 316 SS	6	1.00
Cast Iron (A126 Class C)	4.5	0.75
Chrome Moly Steel (C5)	4.0	0.67
Nodular Iron (A395)	2.5	0.42
Carbon Steel (WCB)	2.25	0.38
Carbon Steel (AISI C1213)	1.0	0.17
Brass (ASTM B16)	0.5	0.08
Aluminum (110-F)	(2-min.)	0.006

*Indicates that the damage was less than that of Type 316 stainless steel when the the test was discontinued.

CAVITATION DAMAGE MECHANISM

Several theories have been advanced relative to the mechanism by which cavitation damage occurs. One fact that is known, however, is that damage always takes place in close proximity to the imploding bubbles.

Earlier investigators have reported pressures as high as 100,000 psi in collapsing cavities. One theory suggests that a high pressure shock wave emanates from each and when these shock waves originate near a solid boundary, a highly concentrated, but minutely small blow is struck. Repeated strikes on any given increment of surface tend to fatigue it until eventually the endurance limit is reached and a tiny bit of metal is removed. Bubbles that implode at a sufficient distance from a solid surface are believed to be incapable of physical damage, because their energy is absorbed by the flowing liquid.

Credibility is given to the above theory by the fact that the harder materials can be subjected to known cavitating conditions for appreciable periods of time without exhibiting visible damage. Conversely, very soft materials such as commercially pure aluminum, from which the valve plugs in Figure 6 were machined, will exhibit microscopically visible individual indentations after exposure for a few seconds. In the case of a hard material, the stress imposed by the shock wave is

*Editor's Note: Magnetostriction is a term which literally implies magnetic contraction, but is generally understood to also include a number of closely allied phenomena relating to certain types of material under magnetic influence.

less than the yield stress; therefore, some finite period of time is required to fatigue the surface metal. For the soft material, the yield stress is exceeded by the implosion stress and the result is immediate damage.

Another theory, while subscribing to the high pressure shock wave as the basic damage mechanism, holds that some portion should be attributed to a chemical attack. This theory suggests that when an increment of surface is struck it undergoes a localized rise in temperature due to absorption of energy. Such localized rise in temperature may be sufficient to permit a chemical reaction between the flowing fluid and the surface material, forming a film which is broken up and washed away by subsequent exposure.

Although there may be some disagreement regarding the actual damage mechanism, one fact is clear—*there is no engineering material, available today, which can indefinitely resist the stresses imposed by severe cavitating conditions.*

FLASHING EFFECTS

When a fluid is flashing in a control valve, there are two major problems which must be considered: physical damage and decreased efficiency. It will be recalled that these were also two of the problems associated with cavitation.

Material Damage

Physical damage associated with the flashing process has a very smooth appearance in contrast to the cinder-like resemblance of cavitation damage. The appearance is often compared to a fine sandblasted surface, although in some material the surface may be even smoother. Actually, the damage mechanism probably closely resembles sandblasting. In sandblasting the grit is borne along by some compressible media, usually steam or air, and achieves a velocity approaching that of the carrier. These high velocity particles impact on the surface causing deformation and ultimately removal of the surface material.

In a flashing liquid, the volume of vapor is frequently greater than the volume of liquid so that the liquid droplets tend to achieve the high velocity of the vapor. Their impact upon a surface will have the same tendency to deform and remove material as will solid particles, although perhaps not to the same extent.

Physical damage to control valves passing flashing liquids is usually confined to the downstream section and it is not uncommon for the piping immediately downstream to be eroded. If the liquid is near saturation at the valve inlet, so that flashing begins in the upstream section of the body, the valve plug's seating surface and flow contours may also be affected.

Selection of proper materials will usually solve most flashing damage problems. The practice of specifying chromium-bearing steel alloys such as WC6, WC9 and C5 for valve body materials has significantly reduced the incidence of the flashing damage often referred to as "body washout". A good example of this is the now common specification of C5 body material for control valves on heater drain service in power generating stations.

Decreased Efficiency - Cavitation and Flashing

In one respect the cavitation and flashing phenomena produce the same type of result. The occurrence of either cavitation or flashing within a control valve will result in a decreased ability of the valve to convert pressure drop across it into mass flow rate. This problem has long been recognized for flashing, but was often completely overlooked for cavitation until recent years.

For non-vaporizing flow, the relationship between flow rate, pressure drop, and liquid specific gravity at any given valve opening is usually expressed in a very simple formula which is often referred to as the "Basic Liquid Sizing Equation." This equation, which can be derived easily using the Bernoulli Theorem and the Continuity Equation in conjunction with some simplifying assumptions is:

$$q = C_v \sqrt{\Delta p / G_f} \qquad (37)$$

where

q = liquid flow rate in gpm
Δp = pressure differential across valve in psi
G_f = liquid specific gravity (water at 60° F = 1.0)
C_v = liquid flow coefficient

From this it may be observed that the flow rate is proportional to the square root of the pressure drop and that the constant of this proportionality is the liquid flow coefficient, C_v.

Most control valve manufacturers determine C_v in accordance with a standard procedure which, if adhered to, permits coefficients produced in the test facilities of one manufacturer to be compared directly with those of another manufacturer. Among other stipulations, the procedure requires test valve pressure drops to be within a 5-10 psi range and the downstream tap pressure to have a minimum value of 2 psig. Using water with a specific gravity of 1.0, flow rates are measured at each of three pressure drops within the above range. These flow rates are then divided by the square root of the appropriate pressure drop to yield C_v values. These experimental values are subsequently averaged to produce a result that is indicative of the capacity of the test valve at the test opening. Under such conditions the observed C_v for any type of control valve should be within the expected limits of experimental accuracy.

Increasing the pressure drop beyond the above range, while holding a constant inlet pressure, will ultimately result in a point where the calculated C_v values will begin to decrease. This apparent decrease in the flow coefficient has been shown to be an indication of cavitation within the valve, with the point of initial departure from the proportional relationship between q and $\sqrt{\Delta p}$ indicating incipient cavitation of the main flow stream. It is entirely possible that some extraneous cavitation due to localized high velocity regions may occur at slightly lower pressure drops.

CAVITATION INDEX

A dimensionless ratio, experimentally determined from a plot of q versus $\sqrt{\Delta p}$ at fixed values of inlet pressure and valve opening, is used to describe the point of initial departure from a proportional relationship. This ratio is called the *Cavitation Index*, K_c, and is defined as follows:

$$K_c = \frac{p_1 - p_2}{p_1 - p_v} = \frac{\Delta p}{p_1 - p_v} \qquad (38)$$

where

p_1 = valve inlet pressure in psia
p_2 = valve outlet pressure in psia
p_v = vapor pressure of liquid at valve inlet in psia (will also be the pressure at the vena contracta)
Δp = pressure differential across body at point of deviation in psi

A typical plot of q versus $\sqrt{\Delta p}$, showing the K_c point, is shown as Figure 7.

Inspection of the K_c relationship indicates that it represents the fraction of the total pressure drop from the inlet pressure to vapor pressure which may be taken as the pressure drop across the valve before cavitation begins within the main fluid stream. Infrequent field service reports verify that cavitation, as evidenced by cavitation damage, may occasionally occur at pressure drops slightly lower than those predicted using the Cavitation Index, K_c. This evidence, when it has been observed, has almost always involved valves which exhibit high pressure recovery (pressure recovery in valves is discussed in subsequent sections). Such devices tend to have well-defined flow streams at their vena contractas, and it is believed that cavitation occurs at, and because of, the highly turbulent interface between the main flow stream and the relatively stagnant surrounding fluid. Since cavitation at this location will not affect the area of the vena contracta, it cannot affect the flow rate through the valve. For a variety of control valves, K_c values have been observed in the approximate range of 0.15 to 0.80. The reasons for this variation will become apparent in a later discussion.

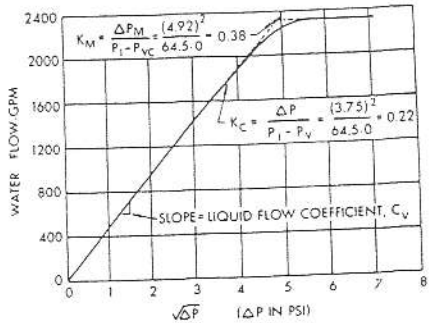

Figure 7. TYPICAL TEST CURVE SHOWING K_c AND K_m POINTS.

The vapor bubbles formed at the vena contracta have a much lower specific weight than the liquid at the same pressure and temperature. Therefore, as bubbles begin to form, the average specific weight at the vena contracta starts to decrease and the flow begins to be choked.

After cavitation has begun, further decreases in the valve outlet pressure (i.e., increased pressure drops) result in increased vaporization, increased cavitation intensity and further decreases in the apparent liquid flow coefficient. Referring again to Figure 7 it is noted that, with sufficient pressure drop, the flow becomes fully choked, so that increasing the pressure drop results in no increase in flow rate. Increasing the pressure drop after choked flow has been reached will result in increased amounts of cavitation damage until the valve outlet pressure is decreased to the value that will permit flashing.

Whether choked flow is the result of the specific volume increasing faster than the velocity at the vena contracta, or is the result of reaching the sonic velocity of the liquid-vapor mixture, is not known. It is obvious, however, that the results are in agreement with sonic flow criteria where pressure impulses cannot move upstream

against the sonic barrier and thus further changes in downstream pressure cannot affect the upstream flow pattern.

VALVE RECOVERY COEFFICIENT

An additional experimental coefficient is determined by flow test to approximate the point above which no increase in flow rate is achieved for an increase in pressure drop. This coefficient is called the *Valve Recovery Coefficient* because its value is a measure of the amount of pressure that is recovered between the vena contracta and the valve outlet. It is designated by the symbol, K_m, and is defined as follows:

$$K_m = \frac{\Delta p_m}{\Delta p_{vc}} = \frac{\Delta p_m}{p_1 - p_{vc}} \qquad (39)$$

where:

Δp_m = pressure drop across the valve body required to produce choked flow in psi
Δp_{vc} = pressure drop between valve inlet and vena contracta at choked flow in psi
p_1 = valve inlet pressure in psia
p_{vc} = vena contracta pressure at choked flow in psia

On an experimental plot of q versus $\sqrt{\Delta p}$, such as that shown in Figure 7, K_m is determined at the intersection of the straight lines: 1) where q is proportional to $\sqrt{\Delta p}$ (i.e., where C_v is constant); and 2) where q is constant. It will be obvious from an inspection of the curve in Figure 7 that the point at which K_m is calculated, while adequately describing the ultimate choked flow rate, does introduce a slight error in the pressure drop. Analyses of a great many plots of this type have not yielded a mathematical expression to describe the shape of the curve between the K_c point and the point of initial choked flow. Values of K_m ranging from 0.20 to 0.95 have been observed in a laboratory using cold water as the test fluid. In laboratory determinations of K_m values, the inlet pressures will usually be 50 psig or greater. Since the vapor pressure of cold water is approximately 0.2 to 0.4 psia and since the vena contracta pressure will be shown to be less than vapor pressure, the contribution of the vena contracta pressure in the computation of K_m values is negligible.

From the definition, it is clear that K_m represents the fraction of the difference between inlet pressure and the choked flow vena contracta pressure that may be taken as pressure drop across the valve, before fully choked flow occurs. An understanding of the physical significance of the valve recovery coefficient and reasons for the

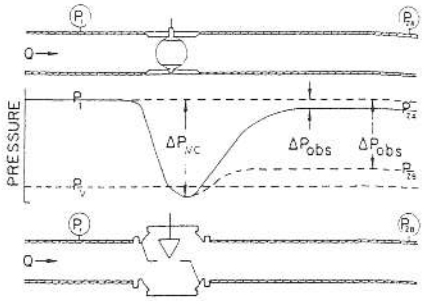

Figure 8. PRESSURE RECOVERY IN TWO STYLES OF CONTROL VALVES.

wide range of observed values may be gained by consulting Figure 8, in which the pressure profiles through two valves with significantly different internal geometries have been plotted. A ball valve in the full-open position is shown at the top of the illustration with a conventional globe style valve at the bottom.

The ball valve is representative of a class of devices which have come to be known as "high recovery valves" whereas the globe valve represents devices at the opposite end of the scale or "low recovery valves". Assume for both of the valves that:

1. inlet pressures are equal;
2. the flowing fluid is the same, and is at the same termperature, so that the vapor pressures are equal; and
3. overall pressure drops have been imposed which are just sufficient to produce choked flow.

It will be noted from the pressure plot of Figure 8, that the pressure at the vena contracta of each valve has fallen to the same value. This appears logical in view of the above assumptions. Note also that the vena contracta pressure is shown to be less than the vapor pressure. It will be recalled that the vena contracta cavitation began when the pressure equaled that of the vapor pressure. Since more flow is realized at the choked flow point than at the incipient cavitation point, it is logical that the pressure at the vena contracta must be lower than the vapor pressure.

Pressure recovery begins after the fluid has passed the vena contracta. In the case of the ball valve, which is represented by the solid line, the pressure recovery is substantial so that only a small overall drop across the valve is observed ($\Delta p_{obs} = p_1 - p_{2a}$). On the other hand the pressure recovery for the globe valve is minimal, resulting in a significantly greater overall pressure drop ($\Delta p_{obs} = p_1 - p_{2b}$). If the C_v's of the two valves were equal, then equal flow rates would have been observed even though the overall pressure drops, as measured from valve inlet to outlet, were substantially different. Stated differently, the pressure drop that is effective in producing flow through any valve is the drop that exists from the inlet to the vena contracta. This is true whether the overall drop is less than, equal to, or greater than the drop required to produce choked flow.

Why do some valves exhibit very high pressure recovery while the recovery in others is very low? The answer to this question is found in the valve's internal geometry. It will be noted that the flow passage through the ball valve is relatively free of obstructions that would introduce turbulence into the main flow stream. Compare this with the tortuous path that the fluid must follow in traversing the globe valve. It is axiomatic to state that anything which introduces turbulence into the flow stream will reduce pressure recovery. Pressure recovery in a well-designed venturi tube may be as much as 90%.

In addition to the conventional ball valve, other examples of high recovery control valves include butterfly valves, segmental ball valves with "V" and "U" shaped openings, and certain flow-closed angle valves. Most globe valves and flow-opened angle valves fall into the low recovery classification. Obviously, examples can be found spanning the complete range of pressure recovery. Generally, control valves that exhibit high recovery in the full-open position lose at least some of their recovery efficiency at decreased openings, while valves with low recovery at full opening, tend to remain that way throughout their travel. Table IV lists representative full-open K_m values for a number of control valve types; however, for specific sizes and types it is recommended that the literature published by the appropriate manufacturer be consulted.

TABLE IV.
REPRESENTATIVE FULL-OPEN K_M VALUES

Body Type	K_m
Globe: single port, flow opens	0.70 - 0.80
Globe: double port	0.70 - 0.80
Angle: flow closes	
Venturi outlet liner	0.20 - 0.25
Standard seat ring	0.50 - 0.60
Angle: flow opens	
Maximum orifice	0.70
Minimum orifice	0.90
Ball Valve:	
V-Notch	0.40
Conventional	0.30
Butterfly valve:	
60° open	0.55
90° open	0.30

Referring again to the pressure plot in Figure 8 it may be seen that the ratio of the overall drop, Δp_{obs}, to the drop from the inlet to the vena contracta, Δp_{vc}, is a measure of pressure recovery. This observation agrees with the definition of the valve recovery coefficient, K_m, as previously stated. Note that the drop, Δp_{obs}, is what would be obtained as the difference between readings on pressure gauges installed upstream and downstream of the body. The pressure drops used for Figure 7, where the method for experimentally determining K_m values is shown, were obtained in just such a manner.

By rearranging the defining K_m equation, an expression may be obtained for the maximum effective overall pressure drop across the valve body:

$$\Delta p_m = K_m (p_1 - p_{vc}) \qquad (39a)$$

This equation, however, still cannot be solved for those cases where p_{vc} is something other than approximately zero, even though K_m values can be obtained from the manufacturers' literature.

THEORETICAL CRITICAL PRESSURE RATIO

The problem, therefore, is to find a way to predict, for any fluid at any vapor pressure, what the vena contracta pressure will be at the lowest overall pressure drop across the valve, which will produce choked flow. In order to do this on a theoretical basis, it is necessary to make an assumption concerning the thermodynamic process that the fluid undergoes between the valve inlet and the vena contracta. The end points of the process from valve inlet to valve outlet will have equal enthalpies, providing that the fluid is all liquid and that the flow areas are equal at both locations. In order for choked flow to exist, however, some vapor must be present at the vena contracta. Also, the fluid velocity is obviously much greater there than at the inlet; therefore, the assumption of an isenthalpic process is impractical.

A constant entropy, or isentropic, process, requires that no fluid friction exist; however, in a flowing fluid stream this is impossible. Thermodynamic texts will agree, however, that most of the losses in flowing fluid streams which undergo contraction and subsequent expansion occur during the expansion. The frictional losses, therefore, will be much

lower during the contraction to the vena contracta and this fact is used to justify the assumption of a constant entropy process to the vena contracta.

In addition to the constant entropy assumption, assume that the fluid at the valve inlet is in an all-liquid, but saturated condition. If increasing flow is induced through the valve by successive reductions in the pressure at the vena contracta, the quality of the resulting mixture may be computed for each value of vena contracta pressure by:

$$X = \frac{S_{vc} - S_f}{S_{fg}} \qquad (40)$$

where

X = mixture quality
S_{vc} = S_1 = entropy at inlet and vena contracta in BTU/lb-°F
S_f = saturated liquid entropy at vena contracta pressure in BTU/lb-°F
S_{fg} = difference between saturated vapor and liquid entropies at vena contracta pressure in BTU/lb-°F

With the quality of the mixture known, rearrangement of the above relationship with substitution of specific volume for entropy, may be used to compute the vena contracta specific volume at each value of assumed vena contracta pressure.

$$v_{vc} = v_f + X\, v_{fg} \qquad (41)$$

where

v_{vc} = vena contracta specific volume in ft^3/lb
v_f = saturated liquid specific volume at vena contracta pressure in ft^3/lb
v_{fg} = difference between saturated vapor and liquid specific volumes at vena contracta pressure in ft^3/lb

A curve may be plotted from this data, defining the relationship between the pressure and specific volume of the fluid at the vena contracta for the assumed saturated inlet pressure to the valve. A typical curve for such a plot is shown by Figure 9.

An expression may be derived for the theoretical mass rate of flow through restrictions in control devices. One such expression is:

$$W/A = 1/v \sqrt{2g \int v_{vc}\, dp_{vc}} \qquad (42)$$

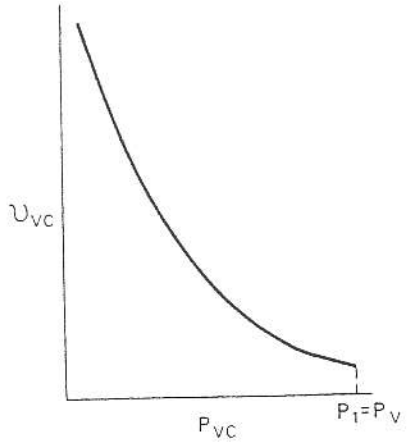

Figure 9. TYPICAL PLOT OF RELATIONSHIP BETWEEN PRESSURE AND SPECIFIC VOLUME AT VENA CONTRACTA.

where

W/A = theoretical mass rate of flow per unit of flow area in $lb/s\text{-}ft^2$

g = acceleration of gravity in ft/s^2

v = vena contracta specific volume at upper limit of integration in ft^3/lb

v_{vc} = vena contracta specific volume in ft^3/lb

dp_{vc} = differential of vena contracta pressure in psfa (lb/ft^2 abs.)

Since the mathematical relationship between vena contracta specific volume and pressure is not known, the integral in the above equation must be obtained graphically. Values obtained for this integral then permit the computation of W/A at a variety of vena contracta pressures for which values may then be plotted. Such plots have a steep slope as the vena contracta pressure is reduced from the saturated liquid condition. The slope of the curve begins to fall off, however, as further reductions are made until eventually a zero slope will be observed indicating a theoretical choked flow condition. It will be recalled that such a condition was also observed experimentally (see Figure 7). Computing mass flow rates at vena contracta pressures lower than that, at which

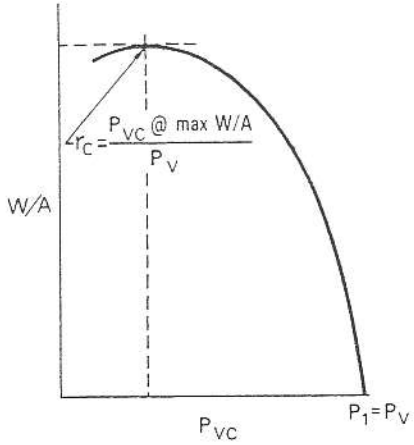

Figure 10. TYPICAL PLOT OF RELATIONSHIP BETWEEN W/A AND VENA CONTRACTA PRESSURE.

choked flow is indicated, show apparent reductions in flow rate. In actual practice, however, the flow rate is observed to reach a choked condition and to maintain this rate despite further reductions in downstream valve pressure. Figure 10 illustrates the typical shape of a plot of W/A versus p_{vc}. The choked flow rates have been experimentally observed to exceed the theoretical values computed using the techniques outlined here.

The ratio of p_{vc} at maximum W/A to the p_v which was applicable to the entire plot in Figure 10 is defined as the *theoretical critical pressure ratio* and is given the symbol, r_c.

$$r_c = \frac{p_{vc}}{p_v} \qquad (43)$$

By rearranging the above equation, it may be seen that the vena contracta pressure required to produce choked flow will be obtained as the product of the theoretical critical pressure ratio and the vapor pressure.

$$p_{vc} = r_c p_v \qquad (43a)$$

The relationship of r_c to p_v over the range of values from zero to the thermodynamic critical pressure, p_c, is shown for water in Figure 11. In order to predict the choked flow vena contracta pressure, the plot is entered at the appropriate p_v, the corresponding r_c read on the ordinate and the product of the entering and exiting values obtained.

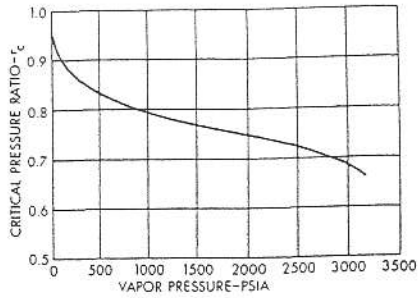

Figure 11. CRITICAL PRESSURE RATIOS FOR WATER.

While compiling such curves for a variety of fluids including two refrigerants, a hydrocarbon and a single-element liquid in addition to water, it was noted that a marked similarity existed in the shape of the curves, as well as in the range of r_c values. With little sacrifice in accuracy, these may be combined into a single curve as shown by Figure 12, where the abscissa is p_v/p_c rather than p_v. Critical pressures, p_c, for several commonly encountered fluids are given in Table V.

TABLE V. CRITICAL PRESSURE OF VARIOUS FLUIDS, PSIA

ammonia	1636
argon	705.6
butane	550.4
carbon dioxide	1071.6
carbon monoxide	507.5
chlorine	1118.7
Dowtherm A	465
ethane	708
ethylene	735
fluorine	808.5
helium	33.2
hydrogen	188.2
hydrogen chloride	1198
isobutane	529.2
isobutylene	580
methane	673.3
nitrogen	492.4
nitrous oxide	1047.6
oxygen	736.5
phosgene	823.2
propane	617.4
propylene	670.3
Refrigerant 11	635
Refrigerant 12	596.9
Refrigerant 22	716
water	3206.2

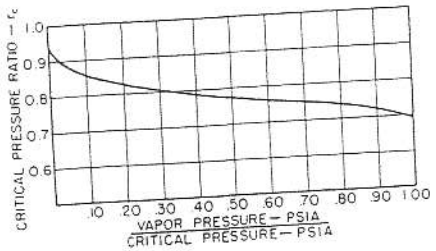

Figure 12. UNIVERSAL CRITICAL PRESSURE RATIO CURVE.

CHOKED FLOW EQUATION

Having shown that the choked flow vena contracta pressure may be approximated by the product of r_c and p_v, this product may be substituted for p_{vc} in Equation 39a so that it will then read:

$$\Delta p_m = K_m \ (p_1 - r_c p_v) \qquad (44)$$

This is referred to as the *Choked Flow Equation* and is intended for use in both cavitation and flashing situations. For the special case where the liquid is saturated at the valve inlet, p_v may be substituted for p_1 and then factored out, leaving the relationship:

$$\Delta p_m = K_m p_v \ (1 - r_c) \qquad (45)$$

The sole purpose of either form of the choked flow equation is to determine the maximum effective overall pressure drop across a control valve. Note from the terms in the equation that, with K_m, it takes into account the type of valve body being used and that, with both p_v and r_c, the thermodynamic properties of the flowing liquid are considered.

In normal valve sizing operations where the pressure conditions for the valves being sized have already been fixed, and where it cannot positively be determined by inspection, that cavitation or flashing will not occur, the choked flow equation should always be solved for Δp_m. The value obtained must then be compared with the actual Δp that the valve will see in service. The smaller of the two values will then be used in the liquid sizing equation to ascertain the required liquid flow coefficient, C_v.

$$C_v = \frac{q}{\sqrt{\Delta p / G_f}} \qquad (37a)$$

It may be shown, referring again to Figure 7, that if the calculated Δp_m is less than the actual service Δp, the flow

other hand, if the calculated Δp_m is greater than the actual Δp, it simply indicates that the valve will be operating in the zone where q is proportional to $\sqrt{\Delta p}$.

Ignoring Δp_m when it is the applicable sizing drop and using, instead, the actual Δp in the sizing equation will result in a calculated C_v that is smaller than what is actually required. Thus, the result may be an undersized valve with all the attendant problems.

When Δp_m is the applicable sizing drop as a result of a cavitating situation, the valve user is thus alerted that severe cavitation damage problems may be encountered so that preventive or protective measures can be taken. The question often arises regarding the severity of cavitation damage at the drop defined by K_m, as compared with the K_c point. In general, it has been found that permitting drops up to the K_m point will not seriously affect the life expectancy of the valve. Obviously there is no magic line of demarcation with serious damage on one side and none on the other. There is growing evidence, however, that in some high recovery valves, significant damage will be experienced at the K_m point.

When process systems are still in the design stage and valve types and locations are being selected, the choked flow equation can be of significant assistance. If, for example, it is determined that a certain valve will exhibit choked flow, a different location, such as at a lower elevation or at the beginning rather than the end of a long line, may permit a higher inlet pressure that would eliminate the problem. It may also be possible to select a type of valve with a higher K_m that would permit more pressure drop.

Experimental Verification Tests

Over a period of several years the choked flow equation has been successfully used to size a wide variety of valves, on a wide variety of fluids and over a wide inlet condition ranging from saturated to substantially subcooled. No reports of incorrect sizing have been received from users; however, until recently no controlled laboratory tests have been conducted specifically for the purpose of substantiating the equation.

Refrigerant-12 (dichlorodifluoromethane-CCl_2F_2) was chosen as the test fluid because it fulfilled several important criteria. The fact that it is most commonly used as a refrigerant was of no importance in its selection. It was desired that the tests be performed with values of vapor pressure covering as much of the range from 0 to p_c as possible. To accomplish this without an excessively expensive test

installation, required that the critical pressure and temperature, p_c and T_c respectively, be moderate. These values for R-12 are 596.9 psia and 233.6 °F respectively. By contrast, comparative values for water, which would otherwise be an ideal fluid, are 3206.2 psia and 705.4 °F respectively. Another requirement was that reasonably complete thermodynamic property data be available, and in this regard, the published information was more than adequate. Since the tests were to be conducted in an indoor laboratory the fluid had to have a very low toxicity and again, R-12 was satisfactory.

Prior to installation in the actual test loop, the entire test section, which included the test valve, upstream and downstream pressure control valves, upstream and downstream pressure taps, and an inlet temperature thermometer well, was installed in a laboratory water test facility. Here the test valve liquid flow coefficient, C_v, and valve recovery coefficient, K_m, were carefully determined in accordance with practices outlined earlier in this chapter.

After installation of the test section into the test loop, the system was evacuated to an absolute pressure of 0.01 in. Hg or less, before charging the system with R-12. The precaution avoided contaminating the R-12 with atmospheric air which would alter the fluid properties. In addition to the test valve section and appropriate flow, pressure, and temperature instrumentation, the closed-loop test system incorporated a water-cooled condenser, a pump, and a steam heater.

Flow versus $\sqrt{\Delta p}$ curves similar to that shown as Figure 7 were acquired at numerous vapor pressure values. For each value of vapor pressure, data runs were made at several values of p_1/p_v. The choked flow for each of the above runs was compared with the flow predicted using the basic liquid sizing equation in which the applicable pressure drop had been calculated from the choked flow equation. The combination of these equations would appear as follows:

$$q_{pred} = C_v \sqrt{\Delta p_m / G} \qquad (37)$$

and (40)

$$= C_v \sqrt{K_m \ (p_1 - r_c \cdot p_v)/G}$$

In Figure 13, the percent errors in predicted flow rates are plotted as a function of the variable, $(p_1 - p_v)/p_v$. The numerator is a measure of the amount of subcooling and this has been divided by the vapor pressure to make the variable dimensionless. It may be seen that at values of the above variable, greater than

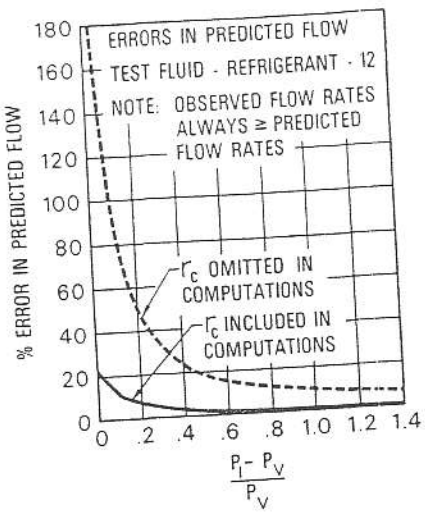

Figure 13. RESULTS FROM R-12 REFRIGERANT TESTS.

approximately 0.8, no errors were noted. However, at decreasing values, increasing errors were observed which approached 20% when the test fluid was saturated. Significantly, the observed choked flow rates were greater than predicted flow rates, so that the use of the choked flow equation to predict the appropriate sizing drop would result in an oversized, rather than an undersized installation. The data shown in Figure 13 is shown only for descriptive purposes and is not intended to be used for actual sizing problems.

To show the necessity for the inclusion of r_c in the choked flow equation, choked flow sizing drops were calculated, omitting r_c from the equation, and flow rates were predicted using these drops. Percent errors in predicted flow were then plotted (Figure 13). Examination of these results will show that, even with substantial subcooling, errors of 7% were observed. Obviously, when p_v is very small in relation to p_1, this error will disappear. Near saturated inlet conditions, the errors become very great and theoretically would be infinite at saturation.

Although use of the choked flow equation does result in some sizing errors near saturated conditions, its use is recommended because it is the most accurate method presently available.

NOMENCLATURE

C_v	Liquid flow coefficient.
G_f	Liquid specific gravity (water at 60° F - 1.00).
g	Acceleration of gravity, in ft/s^2.
K_c	Cavitation Index.
K_m	Valve Recovery Coefficient. ($K_m = F_L^2$, the term used in PART 1)
p_1	Valve inlet pressure, in psia.
p_2	Valve outlet pressure, in psia.
p_c	Thermodynamic critical pressure, in psia.
p_v	Vapor pressure of liquid at valve inlet, in psia.
p_{vc}	Vena contracta pressure, in psia.
dp_{vc}	Differential of vena contracta pressure, in psfa (lb./ft^2 abs).
Δp	Pressure differential, in psia.
Δp_m	Pressure differential across the valve body required to produce choked flow, in psi.
Δp_{vc}	Pressure differential between valve inlet & vena contracta, in psi.
q	Liquid flow rate, in gpm.
r_c	Theoretical Critical Pressure Ratio.
S_f	Saturated liquid entropy, in BTU/lb-°F.
S_1	Entropy at valve inlet, in BTU/lb-°F.
S_{fg}	Difference between saturated vapor & liquid entropies, in BTU/lb-° F.
S_{vc}	Entropy at vena contracta, in BTU/lb-°F.
v_f	Saturated liquid specific volume, in ft^3/lb.
v_{fg}	Difference between saturated vapor & liquid specific volumes, in ft^3/lb.
v_{vc}	Vena contracta specific volume, in ft^3/lb.
W/A	Theoretical mass rate of flow per unit of flow area, in lb/sec-ft^2.
X	Mixture quality.

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SUPPLEMENTARY NOTES ON CAVITATION AND FLASHING IN VALVES*

(E. A. Bake)

Cavitating flow in valves is much more common than may generally be realized. Transient cavitation is commonly heard as a "hiss" or a whistle when an on-off valve is either opened or closed, whether in a high pressure power plant or at the kitchen sink. Liquid control valves may give satisfactory service for months or even years, while operating in a moderately cavitating mode with the characteristic hissing or roaring sounds which accompany the process. Yet, in extreme cases cavitation may destroy valve trim or produce catastrophic damage to adjacent piping in a matter of hours. Between these extremes, there lies a broad gulf of uncertainty which challenges the skill and experience of valve designers and application engineers who must select valves for difficult service.

The key point is that cavitation may vary broadly in intensity. In many applications, the cost or complexity of the steps required to *eliminate* cavitation would be prohibitive; therefore, the objective often is to either limit the intensity of cavitation to an "acceptable" level or to confine the cavitation to a region where its effects can be tolerated.

There can be no absolute definition of an acceptable intensity of cavitation, because the criteria for acceptability are certainly dependent upon valve application requirements. For instance, limitations on environmental noise levels might dictate that cavitation be limited to an intensity much lower than that which would cause rapid damage to valve trim.

Research on cavitation in valves has produced a great amount of data but few confirmable conclusions. Unfortunately, due to variations in the criteria used to evaluate valve tests, results produced by various researchers are difficult to compare. The purpose of this supplement is to illustrate, at least qualitatively, a correlation between some of the important phenomena observed in cavitating flow tests.

Noise and Vibration

Tests have been conducted in a water flow loop using sensitive accelerometers to monitor valve vibration under various conditions of flow, line pressure, and differential pressure. Electronic analysis equipment permitted frequency and amplitude relationships to be identified. Simultaneous notation of audible noises permitted a qualitative correlation of the instrument-recorded data with the observations which might be noted in other valves under field conditions.

Results are best understood if a *system cavitation parameter*, K_{sc}, is introduced as a measure of the tendency of system conditions to cause cavitation in a valve:

$$K_{sc} = \frac{\Delta p}{p_1 - p_v} \qquad (46)$$

where

K_{sc} - System cavitation parameter.
Δp - Pressure drop across valve, psi.
p_1 - Absolute upstream pressure, psia.
p_v - Absolute vapor pressure of fluid flowing, psia.

In typical tests at low values of K_{sc} (low flows and pressure drops), the instrumentation detects no vibrations different from the background levels measured with no flow. In this range, there is no flow-induced noise near the valve. As K_{sc} is increased, a "lower threshold" value is reached where small amplitude, very high frequency vibrations are detected (generally at frequencies above the audible range). With a slight increase in K_{sc} above this threshold value, the instrumentation detects vibrations at somewhat lower frequencies and an audible hissing sound can be heard near an uninsulated valve. It is considered that this range corresponds closely with the "critical cavitation index" as discussed in Reference (1) and numerous other papers.

With further increase in K_{sc}, above the lower threshold value, measurements reveal vibrations of progressively lower frequencies, but these frequencies remain in the acoustic range and amplitudes remain relatively small until an "upper threshold" value of K_{sc} is attained. Between the lower and upper thresholds, audible noise increases in intensity and becomes lower in frequency, producing a roaring sound as cavitation gradually becomes more intense.

The upper threshold value of K_{sc} is characterized by the onset of random vibration patterns in the low frequency range (e.g., well below 100 Hz). Since

*Editor's Note: Continuing investigation and development is at present in progress by several different companies on the phenomena and characteristics of cavitation. Some of the findings of Rockwell International are given here as a supplement to the preceding section by G.F. Stiles.

piping systems often have natural frequencies in this range, measured amplitudes are influenced by the piping configuration; nevertheless, visible mechanical vibration is common at K_{sc} values above the upper threshold, and these may take on dangerous proportions when a valve is poorly supported. The audible noise produced in this range is the characteristic sound of bulk cavitation; often compared to rocks and gravel passing through the valve.

The numerical values of K_{sc} at the lower and upper thresholds vary broadly among valve types and are outside the scope of this supplement. Both thresholds occur at lower K_{sc} values with high-recovery valve types than with low-recovery valves.

Cavitation Effects on Flow

The preceding section on Cavitation and Flashing in Control Valves notes that a reduction of the calculated liquid flow coefficient of a valve, C_v, will be observed if pressure drop is increased beyond a certain limit at a constant upstream pressure. If the pressure drop is increased further, while holding upstream pressure constant, a second point is reached where flow no longer increases with increasing pressure drop. The cavitation index, K_c, defines the first point where reduction of C_v can be observed from experimental data. The recovery coefficient K_m approximates the second point where flow becomes choked.

K_c and K_m can be considered as two values of the system cavitation parameter K_{sc}, as defined in this supplement, characterizing two levels of cavitation intensity in a specific valve. Flow data were logged along with noise and vibration data in the aforementioned tests in the attempt to correlate K_c and K_m with the cavitation thresholds discussed above. The following observations are typical of observations made in testing a number of different types of valves:

1. The lower threshold of cavitation, as determined by vibration measurements or audible noise, was at a lower value of the system cavitation parameter, K_{sc}, than that characterized by the cavitation index, K_c. Practically, this means that there was always detectible cavitation at pressure drops below that where an effect on the valve flow coefficient could be discerned.
2. At conditions where an effect on the valve flow coefficient could just be detected ($K_{sc} = K_c$), the intensity of cavitation, as observed from audible noise and vibration measurements, was quite variable from valve to valve, ranging from mild to moderately severe. Hence, it appeared that the cavitation index, K_c, determined from flow data does not provide a positive indication of the intensity of cavitation which will be produced by a valve in service.
3. The upper threshold of cavitation, as determined from noise and vibration tests, correlated well with the recovery coefficient, K_m, in most tests. Generally, operation at system cavitation parameters exceeding K_m, produced visible vibration and severe "rocks and gravel" noise. An exception was noted in cases of nearly-closed valves (e.g., butterfly valve 10° open) where data revealed choked-flow without excessive noise or low-frequency vibration.

In summary, the recovery coefficient of a valve, K_m, as defined in the preceding section has been found to be a good indicator of conditions producing severe or bulk cavitation, but the cavitation coefficient, K_c, does not correlate well with other experimental methods of establishing the point of inception of cavitation in a valve. Methods based on noise and vibration indicate that cavitation begins at lower pressure drops than would be indicated by K_c; the discrepancy is variable, but quite significant in some cases. While the intensity of the cavitation which is produced at system cavitation parameters below K_c is often mild or moderate, it might, nevertheless, prove to be troublesome in certain applications. Specifically, environmental noise might be unacceptable or damage could be produced in long-term service of valves which do not employ cavitation-resistant trim. Therefore, the cavitation index, K_c, is useful in valve sizing and control analysis work, but caution should be exercised in applying this index to any decision regarding whether or not a valve application will limit cavitation to an "acceptable" level.

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CHAPTER 7

CONTROL VALVE NOISE

PART 1
Control Valve Noise
E. E. Allen, Fisher Controls Company, Marshalltown, Iowa

PART 2
Universal Valve Noise Prediction Method
H. D. Baumann, P. E., Independent Consultant

PART 3
Coping With Control Valve Noise
J. B. Arant, E. I. duPont de Nemours & Co., Wilmington, Delaware

PART 4
Control Valve Noise Rating:
Prediction Versus Reality
W. L. Scull, Leslie Co., Parsippany, New Jersey

EDITOR'S COMMENT

Control valve noise has taken on added interest and importance, since the appearance of the First Edition of this book. In addition, a considerable amount of new investigative and developmental work has made the scene.

For these reasons Control Valve Noise now appears as a new, separate chapter. Presented in it are the viewpoints of four different authors in order that the reader may be presented a balanced picture of this involved subject. Each author is a recognized authority on valve noise, and each is backed up by the resources and support of a well-known, reputable company.

With certain aspects of the subject, differences in opinion may be observed between the authors. The differences are included here for good reason. Each opinion is substantiated by the first-hand knowledge and experience of the author. The fact that one may appear at variance with another does not mean that one is right and the other is wrong. What it does mean is that there are certain areas in the subject of valve noise where further work and investigation remains to be done before a consensus agreement on these points can be reached. This is quite similar to the evolutions that many earlier technologies have gone through.

The intent of this chapter is to present the most up-to-date, most authoritative, and most balanced and comprehensive material now available on the subject of control valve noise.

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PART1
CONTROL VALVE NOISE
(E.E. Allen)

Introduction

Fluid process and transmission systems are a major source of industrial noise. Elements within the system that contribute to the noise are: control valves, abrupt expansions of high velocity flow streams, compressors, and pumps. This chapter deals with control valve noise, a principal source of noise in many fluid systems. Control valve noise is a result of the turbulence introduced into the flow stream in producing the permanent head loss required to fulfill the basic function of the valve. Specific topics covered are: a brief review of noise terminology, identification of source of valve noise, relation of gross flow parameters to noise potential of valve, techniques for prediction of control valve noise, methods of noise control, and noise analysis of existing fluid transmission systems.

Noise Terminology

Noise is commonly defined as unwanted or annoying sound. Sound is the auditory sensation evoked by oscillatory pressure fluctuations propagated in the ambient atmosphere. Socio-political behavior of the average person in response to sound environment is the accepted criterion for determining what is noise. Thus, dependent upon the audience, a specific sound could be considered either acceptable or annoying.

Because of its vague definition, neither an absolute method of measurement nor specific dimensions of noise have been devised. For lack of a better method, noise is frequently described or specified by the physical characteristics of sound. The definitive properties of sound are the magnitude of sound pressure and the frequency of pressure fluctuation, as illustrated by Figure 1.

Loudness of a sound is determined primarily by the amplitude of sound pressure (P_S). Sound pressure measurements are normally root mean square values of sound pressure expressed in microbars. Because the range of sound pressure of interest in noise measurements is about 10^8 to 1, it is customary to deal with sound pressure level instead of sound pressure. Sound pressure level is a logarithmic function of the relative amplitude of sound pressure and is expressed mathematically as:

$$\text{Sound Pressure Level (SPL)} = 20 \log_{10} \frac{P_S}{0.0002\ \mu\ \text{bars}} \quad \text{decibels (dB)}$$

The arbitrarily selected reference sound pressure of 0.0002 microbars is approximately the sound presure required at 1000 Hz to produce the faintest sound that the average young person with normal hearing can detect. The characteristic of the SPL scale is such that each change of 6 decibels in level represents a change in the amplitude of sound pressure by a factor of two.

The apparent loudness of a sound varies not only with amplitude of sound pressure, but also as a function of frequency. The human ear responds to sounds in the frequency range between 20 - 18,000 Hertz. The normal ear is most sensitive to pressure fluctuations in the neighborhood of 3000 Hz; therefore, the degree of annoyance created by a specific sound is a function of both sound pressure and frequency.

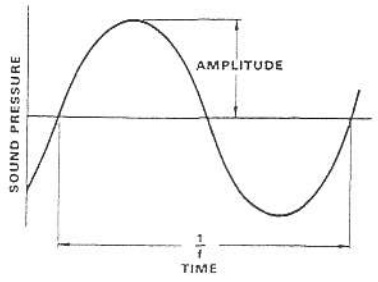


Figure 1. DEFINITIVE PROPERTIES OF SOUND.

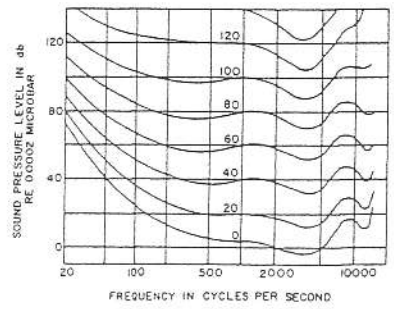


Figure 2. EQUAL LOUDNESS CURVE

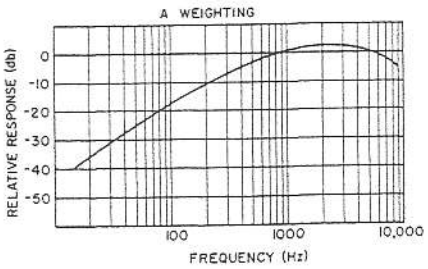


Figure 3. "A-WEIGHTING" CURVE.

Figure 2 shows a plot of sound pressure level versus frequency. On this plot are equal-loudness contours for the average ear. Each contour gives combinations of sound pressure levels and frequencies that seem to be equally loud; that is, have the same loudess level. For example, a sound at 30 Hz and a sound pressure level of 100 dB is identical in loudness level to a sound at 1000 Hz and a sound pressure level of 80 dB.

Sound pressure level measurements are often weighted to adjust the frequency response. Weighting that attenuates the lower frequencies to approximate the equal loudness contours of the human ear is called "A-weighting". Figure 3 shows SPL correction as a function of frequency for A-weighted octave band analysis.

Sound intensity (I) is a measure of acoustic energy and is defined as the acoustic sound power transmitted through a unit area, perpendicular to a specified direction. The common unit of measurement for sound intensity is watts per square centimeter. Sound intensity for a plane wave is given by the relation:

$$I = \frac{Ps^2}{\rho c} \qquad (1)$$

where the product of the density (ρ) and sonic velocity (c) of the transmitting medium represents the characteristic impedance.

For measurement and comparison of sound intensities, it is more convenient to deal with *sound intensity levels,* than with absolute values of sound intensity. Sound intensity level is commonly defined by the following relation:

$$\text{Sound Intensity Level} = 10 \log_{10} \frac{I}{10^{-16}} \text{ decibels} \qquad (2)$$

TABLE I. APPROXIMATE SOUND LEVEL OF FAMILIAR SOUNDS.

Sound Type	Sound Level Decibels
Pneumatic rock drill	130
Jet takeoff (at 200 ft.)	120
Boiler factory	110
Electric furnace (area)	100
Heavy street traffic	90
Tabulation machine room	80
Vacuum cleaner (at 10 ft.)	70
Conversation	60
Quiet residence	50
Electric clock	20

The reference sound intensity is arbitrarily selected as 10^{-16} watts per square centimeter. This is approximately the minimum intensity audible to the average human ear at 1000 cycles per second for standard air. The characteristic of the sound intensity level scale is such that each change of 3 dB in level represents a change in sound intensity by a factor of two.

Sound Measurement

A sound-level meter is the most common means for measuring sounds and noises. Meters are available with built-in weighting networks and contiguous frequency band analyzers. Table I presents the approximate overall sound levels of some familiar sound environments.

The major sources of control valve noise are: mechanical vibration of valve components and fluid noise of which there are two kinds, hydrodynamic noise and aerodynamic noise.

Mechanical Noise

Vibration of valve components is a result of random pressure fluctuations within the valve body and/or fluid impingement upon the movable or flexible part. The most prevalent source of noise resulting from mechanical vibration is the lateral movement of the valve plug relative to the guide surfaces. Sound produced by this type of vibration will normally have a frequency less than 1500 Hertz and is often described as a metallic rattling. The physical damage incurred by the valve plug and/or associated guide surfaces is generally of more concern than the noise emitted.

Early control valves commonly employed valve plugs having skirts into which flow openings were cast or machined. These skirts guided the plug in the valve body ports. Relatively large clearances between the skirts and the body guides made this construction quite

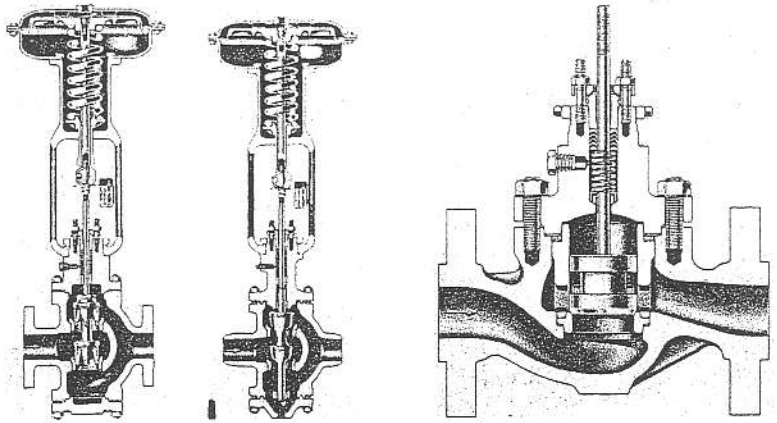


Figure 4. EXAMPLES OF (a) SKIRT GUIDING, (b) POST GUIDING, (c) CAGE GUIDING.

Courtesy Fisher Controls Co.

susceptible to vibration. An improvement was created when the guiding was transferred from skirts to posts incorporated into one or both ends of the valve plug. These posts were guided by bushings rigidly held in the bonnet and blind flanges of the body. For difficult valve applications a further improvement was made by increasing the diameter of the posts and reducing clearances as much as possible. Today's more-or-less standard control valve features cage guiding. In this design a cage member containing the flow openings is rigidly retained in the valve body, and the movable valve plug is closely guided on its inside diameter. These three types of construction are illustrated in Fig. 4. As a result of the evolution in valve trim design, vibration problems from lateral movement of the plug have been minimized.

A second source of mechanical vibration noise is a valve component resonating at its natural frequency. Resonant vibration of valve components produces a single pitched tone, normally having a frequency between 3000 and 7000 Hertz. This type of vibration produces high levels of stress that may ultimately produce fatigue failure of the vibrating part. Valve components susceptible to natural frequency vibration are contoured valve plugs with hollow skirts and flexible members such as the metal seal ring of a ball valve.

In summary, noise that is a by-product of the mechanical vibration of valve components:

1) is not predictable
2) is of secondary concern relative to the physical damage that can occur
3) may even be deemed beneficial in that it warns conditions exist that could produce valve failure
4) for the most part has been eliminated by improved valve design.

Hydrodynamic Noise

Control valves handling liquid flow streams can be substantial sources of noise. The flow noise which is produced is referred to as hydrodynamic noise and may be categorized with respect to the specific flow classification or characteristic from whence it is generated. Liquid flow can be divided into three general classifications:

1) Non-cavitating
2) Cavitating
3) Flashing

Non-cavitating liquid flow generally results in very low ambient noise levels. It is generally accepted that the mechanism from which the noise is generated is a function of the turbulent velocity fluctuations of the fluid stream which are usually referred to as the "Reynolds stresses" or turbulent momentum flux. The high intensity turbulence in control valves occurs as a result of the rapid decelera-

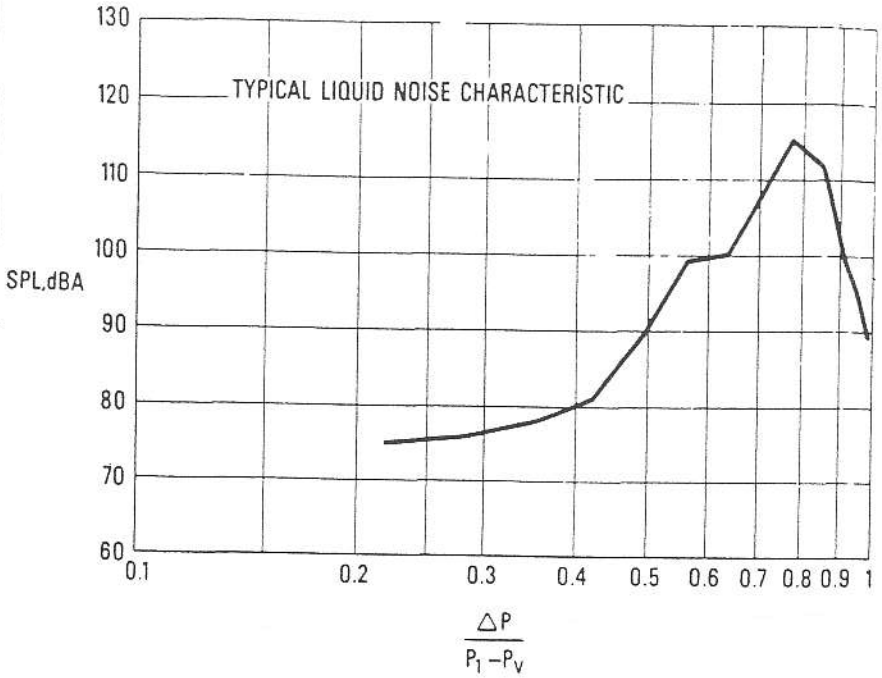


Figure 5. TYPICAL LIQUID NOISE CHARACTERISTIC.

tion of the fluid downstream of the vena contracta due to the abrupt area change.

Test results supported with field experience, indicate that noise levels from non-cavitating liquid applications are quite low and generally would not be considered a noise problem. Figure 5 depicts the typical characteristic of hydrodynamic noise as a function of the ratio of differential pressure across the valve (ΔP) to the pressure at the inlet (P¹ - psia) minus the vapor pressure (Pv - psia).

The major source of hydrodynamic noise is cavitation. This noise is caused by the implosion of vapor bubbles that are formed in the cavitation process. Cavitation occurs in valves controlling liquids whenever the static pressure downstream of the valve is greater than the vapor pressure and when the local static pressure at some point within the valve is less than or equal to the liquid vapor pressure. The low local static pressure may be the result of high velocity and /or intense turbulence.

Figure 6 depicts the pressure profile of a cavitating flow stream as a function of distance along the stream. Vapor bubbles are formed in the region of minimum static pressure and subsequently are collapsed or imploded as they pass downstream into an area of higher static pressure. Noise produced by cavitation has a broad frequency range and is frequently described as a rattling sound similar to that which would be anticipated if gravel were in the fluid stream.

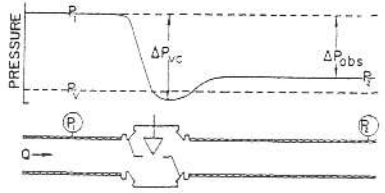


Figure 6. STATIC PRESSURE ALONG A STREAM LINE CAVITATING FLOW.

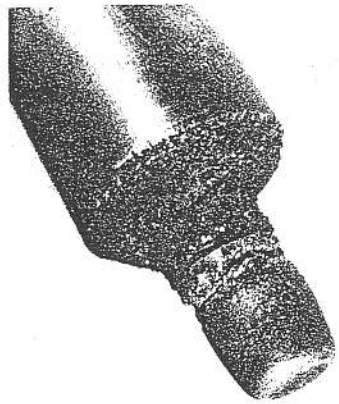


Figure 7. EXAMPLE OF CAVITATION DAMAGE TO A VALVE PLUG.

Courtesy Fisher Controls Co.

Cavitation may produce severe damage to the solid boundary surfaces that confine the cavitating fluid. Generally speaking, noise produced by cavitation is a secondary concern. Physical damage as a result of cavitation erosion is depicted in Fig. 7.

Flashing is a phenomenon that occurs in liquid flow when the differential pressure across a restriction is greater than the differential between the absolute static and vapor pressure at the inlet to the restriction, $\Delta P > P_1 - P_v$. The resulting flow stream is a mixture of the liquid and gas phases of the fluid. Noise resulting from a valve handling a flashing fluid is a result of the deceleration and expansion of the 2-phase flow stream.

Aerodynamic Noise

The major source of valve noise is aerodynamic noise. Aerodynamic noise is noise generated as a by-product of a turbulent gas stream, or noise produced without the inter-action of the fluid with vibrating boundaries or other external energy sources.

Aerodynamic noise is a result of the Reynolds stresses or shear forces created in the flow stream as a result of deceleration, expansion, or impingement. The principal area of noise generation in a control valve is the recovery region immediately downstream of the vena contracta where the flow field, characterized by intense turbulence and mixing, is of chaotic quality with phase being completely random and discontinuous.

VALVE NOISE PREDICTION

Aerodynamic Noise

Most of the theories, to date, on aerodynamic noise have been developed with special reference to the noise of free jets as opposed to a gas stream restrained by solid walls. A standard reference on this topic is Lighthill's paper(1).

Several hypotheses describing the acoustic power generated by gas flow through a control valve have been advanced on the postulates: (a) Lighthill's theory is applicable; and (b) the turbulent level of a flow stream can be accurately defined analytically. The validity of either or both postulates is very questionable, because:

1) Lighthill's theory is based on a free jet and would not appear applicable to the vastly more complex flow field produced by high subsonic and sonic gas streams contained within the solid boundaries of a control valve.
2) Very little, either qualitatively or quantitively, is known concerning the turbulence level of a bounded fluid stream.

Empirical Advantage

Development of an analytical model which accurately describes the precise mechanism of aerodynamic noise generation by control valves is very unlikely, in view of the complexity of the problem and the lack of knowledge concerning absolute values of the relevant parameters. It would seem obvious; therefore, that an accurate technique for predicting valve noise must be based on empiricism. Such an approach requires the noise characteristic for each valve style and adjacent piping configuration to be established by actual test. Obviously, it is neither practical nor feasible to determine noise levels for all valve applications by laboratory tests. Dimensional analysis would be the logical tool for correlation and extrapolation of the test data. It is a very useful and powerful tool in formulating problems which defy analytical solution and must be solved experimentally.

In fluid mechanics, the relevant parameters in any physical situation may be combined into independent, dimensionless groups that characterize the flow. Two flow streams are considered similar, if they are geometrically similar and if the dimensionless groups have equal values for both flows. This technique of expressing a physical relationship between quantities by utilizing dimensionless ratios of relevant parameters is the field of mathematics known as dimensional analysis.

Dimensionless groups (or as they are called; numbers) may be obtained directly either from the differential equations that describe

the flow field or by simply combining the relevant variables into as many independent dimensionless numbers as possible. An advantage of the latter system is that one does not need to know the governing equations or the laws for complex problems; however, one must identify all the variables.

The variables to be considered for a dimensional analysis approach to the problem of predicting aerodynamic noise generated by compressible fluid flow through control valves are P_s, ΔP, V, ρ, D, μ, c, C_p, C_v and λ. The independent dimensionless numbers formed from the above list are:

1) Prandtl Number; ($P_r = C_p \mu/\lambda$),
2) Reynolds Number, ($Re = \rho VD/\mu$),
3) mach number, ($M = V/c$),
4) specific heat ratio, ($k = C_p/C_v$),
5) ratio of sound pressure to differential pressure across valve, ($P_s/\Delta P$)

Relative to the generation of aerodynamic noise, the significance of each of the dimensionless numbers listed above is explained below.

1) The Prandtl Number (P_r) is nearly the same for all gases and can be considered insignificant in the analysis of processes where heat transfer does not play an important role. Generation of aerodynamic noise is considered an adiabatic process and thus P_r can be dropped from consideration.
2) Reynolds Number (Re) is a measure of the ratio of inertia forces to viscous forces. Thus, in the study of compressible fluids, the role of Reynolds Number as an important parameter is limited to areas concerning low velocities or the boundary layer. For control valve noise levels of concern, the principal noise generation mechanism is located in the area of high subsonic or supersonic velocity, thus the Reynolds Number would not be considered a relevant parameter.
3) From the preceding argument it is obvious that the mach number (M) should be considered the most important factor. It can be shown that (M) can be presented as $f(\Delta P/P_1)$.
4) Not enough experimental data has been obtained to accurately define the influence that the specific heat ratio (k) has on noise generation; however, it has a relatively weak influence on the flow stream characteristic.

Based on the dimensional analysis logic presented above, the acoustic power generated by mpressible flow thru a control valve

$$W \propto Cg^2 (\Delta P)^2 f(\Delta P/P_1) \quad (3)$$

where Cg is the critical flow coefficient and is directly proportional to the open area of the flow restriction exposed to the flow stream[2]. The rationale used in development of the above relationship assumed a single port or noise generator. In the following equation an extension is made to the basic relationship to include the effects of multiple ports.

$$W \propto 1/N\, Cg^2 (\Delta P)^2 f(\Delta P/P_1) \quad (4)$$

where: N = number of ports open to the flow stream.

AERODYNAMIC NOISE PREDICTION TECHNIQUE

The graphical solution of the following equation based on relationship (3) and the logic previously presented, provides a very expedient and accurate technique for predicting ambient noise levels resulting from the flow of compressible fluids through control valves.

$$SPL = SPL_{\Delta P} + \Delta SPL_{Cg} + \Delta SPL_{\Delta P/P_1} + \Delta SPL_K \quad (5)$$

where:

SPL = overall noise level in dB at a predetermined point (48 in. downstream of the valve outlet and 29 in. from the pipe surface).

$SPL_{\Delta P}$ = base SPL in dB, determined as a function of ΔP.

ΔSPL_{Cg} = correction in dB for Cg.

$\Delta SPL_{\Delta P/P_1}$ = correction in dB for valve style and pressure ratio.

ΔSPL_K = correction in dB for acoustical treatment; i.e., heavy wall pipe, insulation, inline silencers, etc.

The following example serves as a guide for using the noise prediction technique. Information presented in Figs. 8 through 19 and Table II facilitates the prediction technique.

Table II. ΔSPL_K CORRECTION FOR PIPE WALL ATTENUATION.

NOMINAL PIPE SIZE IN.	SCH 10	SCH 20	SCH 30	SCH 40	SCH 60	SCH 80	SCH 100	SCH 120	SCH 140	SCH 160	STD	XS	XXS
1				0		−4.5				−9.5	0	−4.5	−15.5
1-1/2				0		−4.5				−9.5	0	−4.5	−15.5
2				0		−5				−11.5	0	−5	−15.5
3				0		−4.5				−10	0	−4.5	−15
4				0		−5		−8.5		−11.5	0	−5	−15
6				0		−6		−9.5		−13	0	−6	−16
8		+3.5	+2.0	0	−3	−6	−8.5	−11	−13	−14.5	0	−6	−14
10		+5.0	+2.5	0	−4.5	−6.5	−9.5	−11.5	−14	−15.5	0	−4.5	
12		+5.5	+1.5	−1	−5.5	−8	−11	−13.5	−15	−17.5	0	−4	
14	+5.5	+2.5	0	−2	−6	−9.5	−12.5	−14.5	−16.5	−18.0	0	−4	
16	+5.5	+2.5	0	−4	−7.5	−11	−13.5	−16	−18.5	−20	0	−4	
18	+5.5	+2.5	−2	−5.5	−9.5	−12.5	−15.5	−17.5	−19.5	−21.5	0	−4	
20	+5.5	0	−4	−6	−10.5	−13.5	−16.5	−19.0	−21.0	−22.5	0	−4	
24	+5.5	0	−5.5	−8	−12.5	−16	−19	−21.5	−23	−25	0	−4	
30	+2.5	−4	−7								0	−4	
36	+2.5	−4	−7	−9							0	−4	
42													

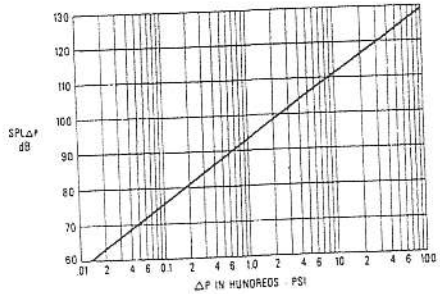


Figure 8. ALL VALVE STYLES, SPL BASE.

Given:

A 4 inch valve, 300 lb (ANSI), Fisher Design ES (a cage Style globe body) installed in an 8inch Sch 40 pipe line.

P1	= 615 psia	
ΔP	= 348 psi	
Cg	= 4000	(calculation required)
$SPL_{\Delta P}$	= 103.5 dB	(Fig. 8)
ΔSPL_{Cg}	= - 2 dB	(Fig. 9)
$\Delta SPL_{\Delta P/P_1}$	= + 9.5 dB	(Fig. 12)
ΔSPL_K	= 0 dB	(Table II)
SPL	= (103.5 + (-2) + 9.5 + 0) dB	
SPL	= 111 dB	

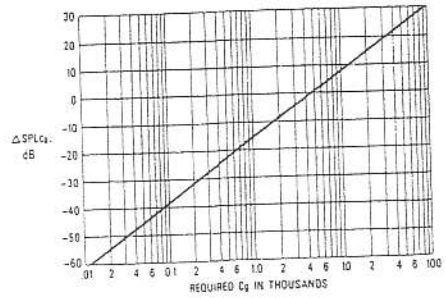


Figure 9. ΔSPL_{Cg} CORRECTION, ALL VALVE STYLES.

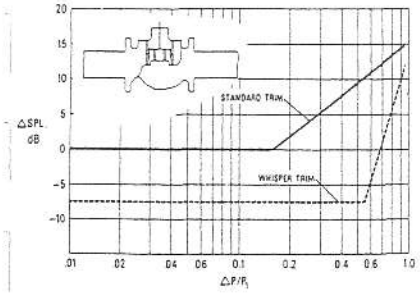


Figure 10. CAGE STYLE GLOBE VALVES.
Line Size Equals Body Size (4-12 inches).

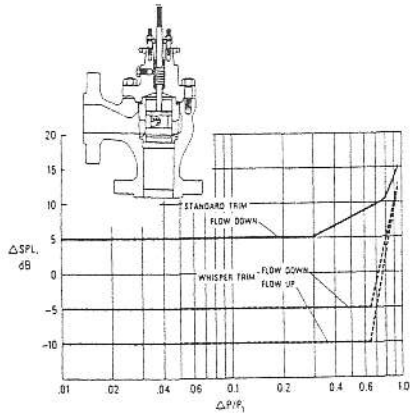


Figure 11. SWAGED 4-INCH ANGLE VALVE.

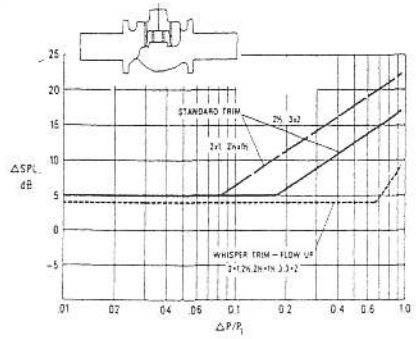


Figure 12. 2-3 INCH CAGE STYLE GLOBE VALVES.
Line Size Equals Body Size.

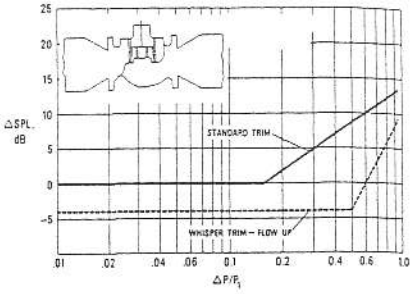


Figure 13. 3-6 INCH SWAGED CAGE STYLE GLOBE VALVES.

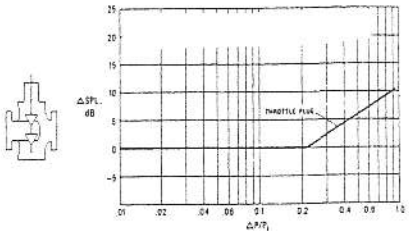


Figure 14. TOP AND BOTTOM GUIDED GLOBE VALVES.
All Line to Body Size Ratios (All Valve Sizes).

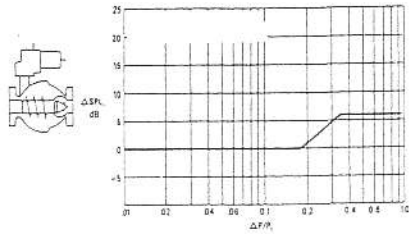


Figure 15. SLIDING SLEEVE REGULATORS.
All Line To Body Size Ratios (All Valve Sizes).

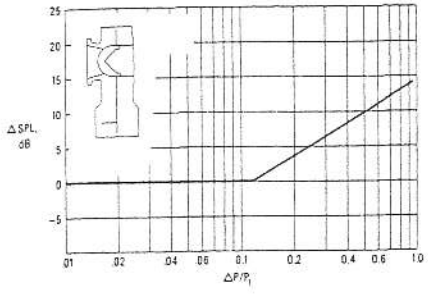


Figure 16. V-NOTCH BALL VALVES.
All Line to Body Size Ratios (All Sizes).

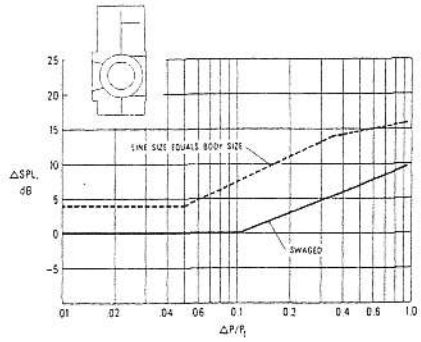


Figure 17. STANDARD BALL VALVES.
All Line to Body Size Ratios (All Valve Sizes).

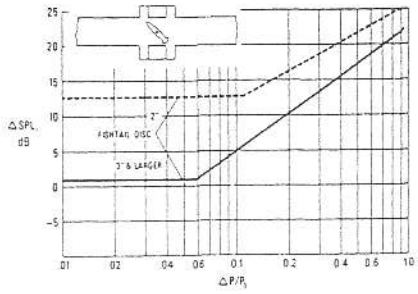


Figure 18. BUTTERFLY VALVES.
Line Size Equals Body Size (All Sizes).

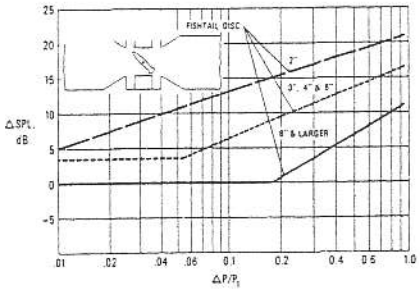


Figure 19. BUTTERFLY VALVES.
Swaged (All Sizes).

As an alternative selection of a valve to minimize noise, the calculations are repeated as in the above problem only using an 8 inch x 4 inch Design EW (a cage style globe body) with Whisper Trim®. Note: the only change in our calculation will be the $\Delta SPL_{\Delta P/P_1}$ value.

$\Delta SPL_{\Delta P/P_1} = 7.5$ (Fig. 13)

$SPL = [\,103.5 + (-2) + (-7.5) + 0\,]\,dB$ (Eq. 5)

$SPL = 94\ dB$

If we were to use Schedule 80 pipe from Table II,

$\Delta SPL_K = -6\ dB$

$SPL = (94 - 6)\ dB$

$SPL = 88\ dB$

Hydrodynamic Noise Prediction

As with aerodynamic noise a pure mathematical approach to prediction of hydrodynamic noise would be questionable due to approximations and assumptions that must be made during the course of derivation. However, by considering a dimensional analysis approach, some insight into the governing parameters can be achieved. The variables considered as possibly pertinent to hydrodynamic noise generated by a control valve are:

P_s, P_1, P_2, P_v, V, ρ, D, E, σ, ν

The above variables can be combined to form independent dimensionless numbers:

1) Reynolds number, $(R_e = VD/\nu)$

2) Cauchy number, $(C_u = \rho V^2/E)$

3) Weber number, $(W_e = \frac{\rho DV^2}{\sigma})$

4) Euler number, $(E_u = \frac{\rho V^2}{2(P_1 - P_v)})$

5) Ratio of the rms pressure fluctuation to the pressure drop $P_s / \Delta P$

Not all of the above dimensionless numbers are pertinent in describing hydrodynamic noise generation and scrutinization of each will result in eliminating those of little importance.

1) The Reynolds number (R_e) is the ratio of inertia forces to viscous forces. For the most part, the viscosity effect on the flow is unimportant for fully developed turbulent flow. Since turbulent flow exists in all control valves of interest from a noise standpoint, Reynolds number will be dropped from consideration.

2) The Cauchy number (C_u) is the ratio of inertia force to the compressibility force of the fluid. The Cauchy number becomes very important for numbers approaching 1.0 and relatively unimportant for small numbers less than 0.1. Since the Cauchy number for most liquid flow applications is less than 0.1, the Cauchy number can be ignored. However, some error may result as flow conditions approach flashing and downstream velocities become quite high.

3) The Weber number (W_e) is equivalent to the ratio of inertia force to the surface tension force. It becomes important when considering small gaseous cavitation. This should play a minor role in influencing the noise generated in control valves since the inertia forces far exceed the surface tension forces.

4) The Euler number (E_u) is the ratio of inertia forces to the pressure forces affecting the flow stream. Since pressure conditions are of major importance describing the occurence of cavitation, the Euler number is probably the most important of the dimensionless numbers discussed so far and will be of major importance.

Based on dimensional analysis logic, the acoustic power generated by incompressible flow thru a control valve obeys the following relationship:

$$W \propto C_v^2 f\ \Delta P^2 \left(\frac{\Delta P}{P_1 - P_v}\right) f\left(\frac{\Delta P}{P_1 - P_v}\right) \quad (6)$$

where C_v is the liquid sizing coefficient needed to pass the required flow rate.

HYDRODYNAMIC NOISE PREDICTION TECHNIQUE

Based on relationship (6) a technique of the same format used for aerodynamic noise is presented for hydrodynamic noise:

$$SPL = SPL_{\Delta P} + \Delta SPL_{C_v} + \Delta SPL_{\Delta P/(P_1 - P_v)} + \Delta SPL_K \quad (7)$$

where:

SPL = overall noise level in dBA at a predetermined point (48 inches downstream of the valve outlet and 29 inches from the pipe surface)

$SPL_{\Delta P}$ = base SPL in dBA determined as a function of ΔP and $\Delta P/(P_1 - P_v)$

$\Delta SPL_{\Delta C_v}$ = correction in dBA for C_v

$\Delta SPL_{\Delta P\ (P_1 - P_v)}$ = correction in dBA for valve style and flow regime

ΔSPL_K = correction in dBA for acoustical path treatment; i.e., heavy wall pipe, insulation, etc.

The noise levels are based on standard weight pipe; however, a correction factor, ΔSPL_K, to be used for other schedules of pipe is provided in Table II.

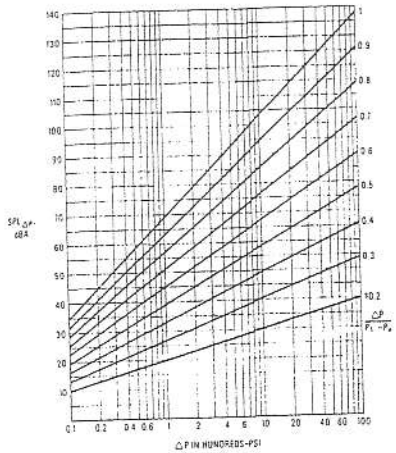


Figure 20. BASE $SPL_{\Delta P}$ FOR ALL VALVE STYLES.

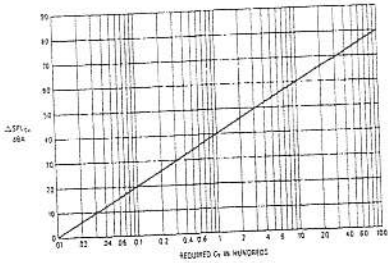


Figure 21. ΔSPL C_v CORRECTION FOR ALL VALVE STYLES.

Figure 20 illustrates SPL as a function of total pressure drop across the valve. Note that the base $SPL_{\Delta P}$ is also dependent upon the pressure drop ratio $\Delta P/(P_1 - P_v)$. The correction for C_v is shown in Fig. 21. Figures 22 through 26 are the corrections for a few valve styles as a function of the flow regime parameter $\Delta P/(P_1 - P_v)$.

An example problem:
Given: A 4 inch valve, 300 lb (ANSI), Fisher Design ED with Cavitrol®Trim, installed in a 6 inch Schedule 80 pipe.

P_1 = 600 psia
Δ = 400 psi
P_v = 100 psia
Required flow coefficient: C_v = 100
$\Delta P/(P_1 - P_v)$ = 0.80
From Fig. 20, $SPL_{\Delta P}$ = 74 dBA
From Fig. 21, ΔSPL_{C_v} = 40 dBA
From Fig. 26, $\Delta SPL_{\Delta P/(P_1 - P_v}$ = -22 dBA
From Table II, ΔSPL_K = -5
Thus SPL = 74 + 40 - 22 - 5 = 87 dBA

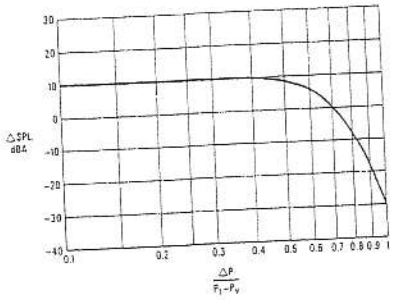


Figure 22. CORRECTION FOR V-NOTCH BALL VALVES IN SWAGED SETUP.

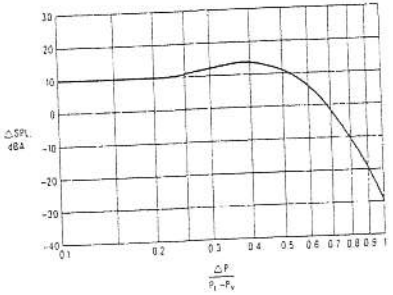


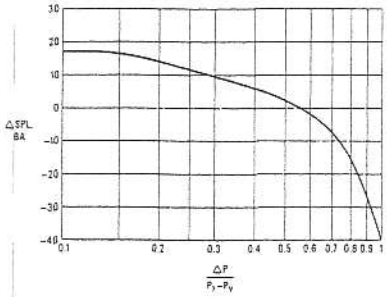
Figure 23. CORRECTION FOR 4 INCH BUTTERFLY VALVE IN A SWAGED SETUP.

NOISE CONTROL

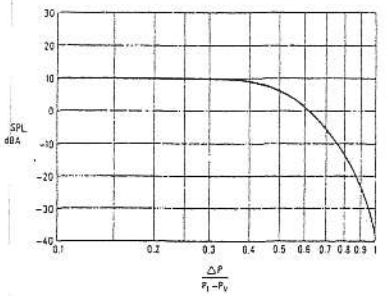
Noise control employs either one or both of the following basic approaches:

1) Source Treatment - Prevention or attenuation of the acoustic power at the source (quiet valves).

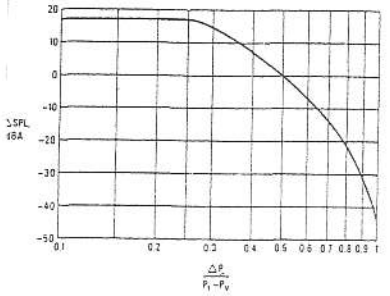
2) Path Treatment - Reduction of noise transmitted from a source to a receiver.



igure 24. CORRECTION FOR GLOBE VALVE
TYLES.



ıre 25. CORRECTION FOR STANDARD CAGE
M IN A GLOBE BODY, FLOW DOWN.



ıre 26. CORRECTION FOR CAVITROL TRIM IN
LOBE VALVE, FLOW DOWN.

QUIET VALVES *

From the preceding, the parameters that determine the level of noise generated by compressible flow thru a control valve for a given application are: the number of ports or restrictions exposed to the flow stream, the total Cg, differential pressure across the valve, and the ratio of differential pressure to absolute inlet pressure.

Figure 27 shows four basic approaches (a,b,c,d) to the design of quiet control valves. Common to all, is the use of a multiplicity of restrictions versus a single or a few large restrictions.

Approach (a) utilizes a small tortuous path (Dia. < 1/32 inch) providing a flow path with a high FL/D ratio designed to maximize the per cent of total pressure drop obtained by viscous stresses induced by the shearing action of the fluid and boundary layer turbulence.

The noise characteristic or noise potential of a valve increases as a function of the square of the differential pressure (ΔP^2) and the ratio of differential pressure to absolute static pressure at the inlet ($\Delta P/P_1$). Thus for high pressure ratio applications ($\Delta P/P1 > 0.7$) an appreciable reduction in noise can be effected by staging the pressure loss thru a series of restrictions to produce the total pressure head loss required. Concept (b) uses multiple series restrictions to limit the $\Delta P/P_1$ ratio across individual restrictions to optimum operating points and provide a favorable velocity distribution in the expansion area. Approaches (a) and (b) can easily be incorporated into cage style trim fabricated from stacks of discs machined to stage the total pressure drop thru a series of circumferential restrictions. It should be pointed out that both (a) and (b) act as strainers and are very susceptible to plugging as a result of either solid particles in the gas stream (dirty gas) or as a result of hydrate formation prior to the last stage.

The third approach shown (c) uses a multiplicity of narrow parallel slots specifically designed to both minimize the turbulence level and provide a favorable velocity distribution in the expansion area. This is an economical approach to quiet valve design and can provide substantial noise reduction (15-20 dB) with little or no decrease in total flow capacity.

nments made in this section are applicable to the design of valves that introduce turbulence into the flow
am in producing the permanent head loss required to fulfill the basic function of the valve. It is conceivable
sign a valve that utilizes viscous losses to produce the permanent head loss required. Such an approach
d require valve trim with a very high FL/D ratio which becomes impractical from the standpoint of both
omics and physical size.

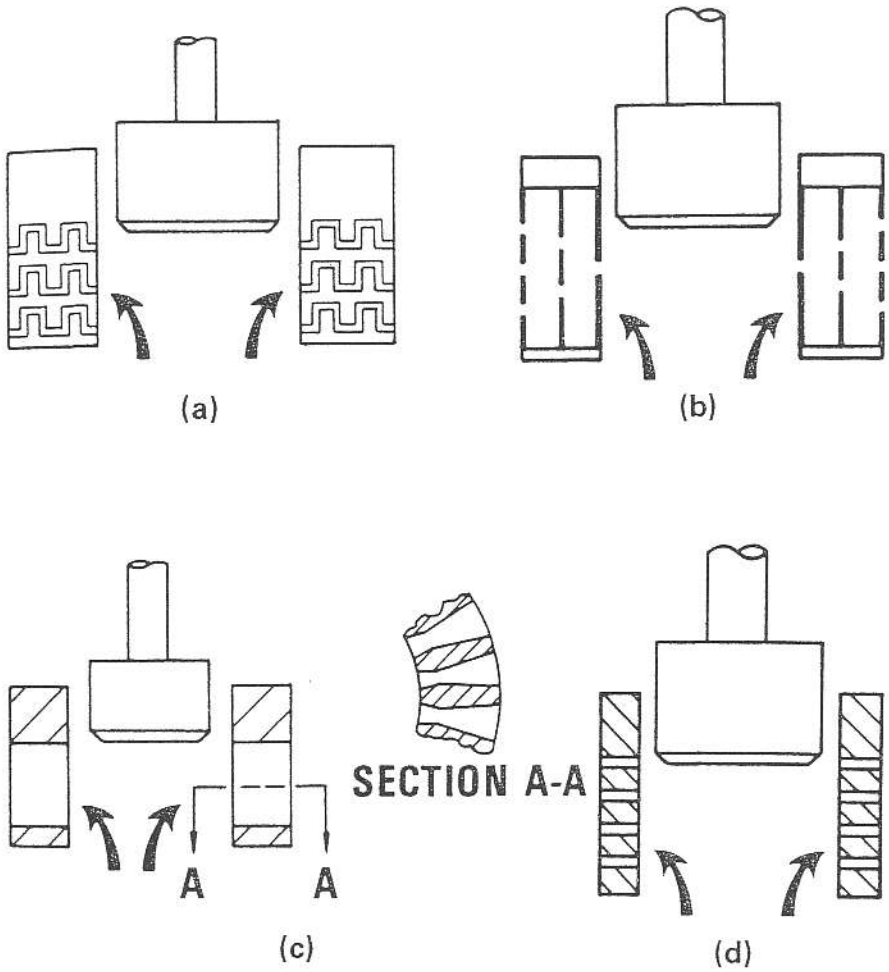


Figure 27. ILLUSTRATIONS OF VALVE TRIM DESIGNED FOR NOISE ATTENUATION.

It can be shown that the acoustic power of a single flow restriction increases as a function of area squared. Changing the area by a factor of 2 results in a corresponding 6 dB change of power level; whereas, the power level is changed only 3 dB when the number of equal noise sources acting independently is changed by a factor of 2. Critical to the total noise reduction that can be derived from utilization of many small restrictions versus a single or few large restrictions is the proper size and spacing of restrictions such that the noise generated by jet interaction is not greater than the summation of the noise generated by the jets individually. It has been found that the optimum size and spacing is very sensitive to pressure ratio, $\Delta P/P_1$. The author's company has recently developed the technology to utilize the above approach for the design of quiet valve trim using single-stage pressure reduction. This approach depicted by (d) provides equivalent or better

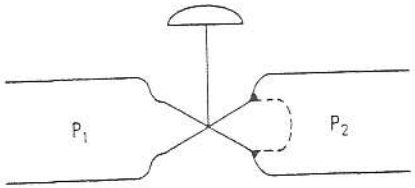


Figure 28. TWO STAGE PRESSURE REDUCTION WITH A DIFFUSER.

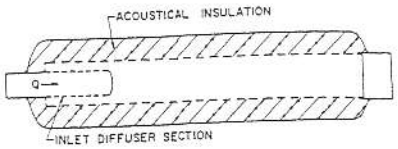


Figure 29. TYPICAL INLINE SILENCER.

noise performance than (a), (b), or (c).

For control valve application operating at high pressure ratios ($\Delta P/P_1 \geq 0.8$) the series restriction approach, splitting the total pressure drop between the control valve and a fixed restriction (diffuser) downstream of the valve can be very effective in minimizing the noise. In order to optimize the effectiveness of a diffuser, it must be designed (special shape and sizing) for each installation so that the noise levels generated by the valve and diffuser are equal. Figure 28 depicts a typical valve-plus-diffuser installation.

Pertaining to the design of quiet valves for liquid application, the problem resolves itself to one of designing a valve to eliminate cavitation. Service conditions that will produce cavitation can readily be calculated. [3] The use of staged or series reductions provides a very viable solution to cavitation and hence hydrodynamic noise.

Path Treatment

A second approach to noise control is that of path treatment. Sound is transmitted via longitudinal waves through the elastic medium or media that separate the source from the receiver. The speed and efficiency of sound transmission is dependent on the properties of the medium through which it is propagated. Path treatment consists of regulating the impedance of the transmission path to reduce the acoustic energy that is communicated to the receiver.

The fluid stream is an excellent noise transmission path. When critical flow exists (fluid velocity at the vena contracta is at least at the sonic level), the vena contracta acts as a barrier to the propagation of sound upstream via the fluid. At sub-critical flow, however, valve noise can be propagated in the upstream direction almost as efficiently as it is downstream. The impedance to the transmission of noise upstream at sub-critical flow is primarily a function of valve geometry. The butterfly valves), offers little resistance to noise propagation. Globe style valves provide approximately 10 dB attentuation. In any path treatment approach to control valve noise abatement, consideration must be given to the amplitude of noise radiated by both the upstream and downstream piping.

Dissipation of acoustic energy by use of acoustical absorbent materials is one of the most effective methods of path treatment. Whenever possible the acoustical material should be located in the flow stream either at or immediately downstream of the noise source. This approach to abatement of aerodynamic noise is accommodated by inline silencers. Inline silencers effectively dissipate the noise within the fluid stream and attenuate the noise level transmitted to the solid boundaries. Where high mass flow rates and/or high pressure ratios across the valve exist, inline silencers are often the most realistic and economical approach to noise control. Use of absorption-type inline silencers can provide almost any degree of attenuation desired. However, economic considerations generally limit the insertion loss to approximately 25 dB. Figure 29 is a cross-sectional view of a typical inline silencer.

Noise that cannot be eliminated within the boundaries of the flow stream must be eliminated by external treatment or isolation. This approach to the abatement of control valve noise includes the use of heavy walled piping, acoustical insulation of the exposed solid boundaries of the fluid stream, and use of insulated boxes, rooms and buildings, to isolate the noise source.

In closed systems (not vented to atmosphere) any noise produced in the process becomes airborne only by transmission thru the solid boundaries that contain the flow stream. The sound field in the contained flow stream forces the solid boundaries to vibrate which in turn cause pressure disturbances in the ambient atmosphere that are propagated as

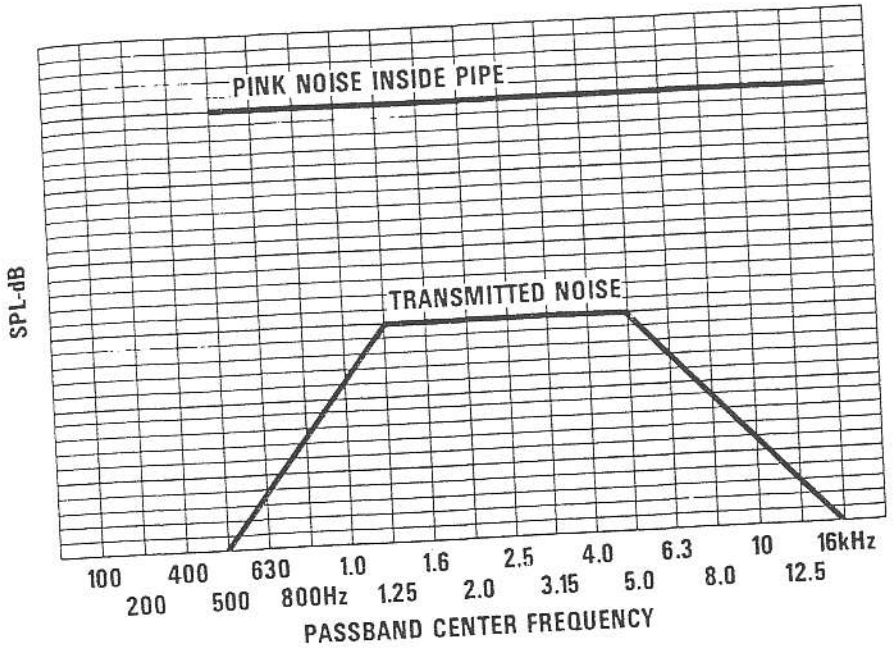


Figure 30. EFFECTS OF TRANSMISSION LOSS CHARACTERISTIC OF PIPE.

of noise radiation to the atmosphere is the piping adjacent to the valve. An understanding of the relative noise transmission loss as a function of the physical properties of the solid boundaries of the flow stream can be beneficial in noise control of fluid transmission systems.

A detailed analysis of noise transmission loss is beyond the scope of this chapter. However, it should be recognized that the spectral density of the noise radiated by the pipe has been shaped by the transmission loss characteristic of the pipe and is not that of the noise field within the confined flow stream.

Figure 30 depicts the general transmission loss characteristic for commercial piping. The vibrational response of pipe to an acoustic field within is characterized by the following frequencies:

1) Coincident frequency (f_c) is defined as that frequency at which the acoustic wavespeed is equal to the phase velocity of a flexural wave in the pipe wall.

$$f_c = \frac{\sqrt{3}(c_i{}^2)}{\pi t c_s} \text{ Hz} \qquad (8)$$

2) Acoustic cutoff frequency (f_{co}) lower limiting frequency where acoustic energy will propagate freely within the pipe, below f_{co} acoustic energy propagates as a plane wave without a radial component to excite the pipe wall.

$$f_{co} = \frac{0.586\ c_i}{d} \text{ Hz} \qquad (9)$$

3) Ring frequency (f_r) is the frequency at which the longitudinal wave in the pipe material equals the pipe circumference.

$$f_r = \frac{c_s}{\pi d} \qquad (10)$$

Above the ring frequency (f_r) the pipe wall tends to behave much like a flat plate and mass law apples. Below the maximum value of the coincident frequency (f_c) or cutoff frequency (f_{co}) the vibration response is stiffness controlled. In the frequency range between

e coincident and ring frequency the resonant modes of pipe vibration are classified acoustically fast and are relatively efficient in e transmission of sound.

The relative noise transmission loss as a nction of pipe size and schedule has been quantified in Table II. For a comprehensive analysis of pipe transmission loss, see refer- ice Number 4.

Acoustical insulation of the exposed solid undaries of the fluid stream is an effective means of noise abatement for localized areas. Test results indicate that ambient noise vels can be attenuated as much as 10 dB per ch of insulation thickness.

Path treatment such as heavy walled pipe or external acoustical insulation can be a very economical and effective technique for calized noise abatement. However, it should e pointed out that noise is propagated for ng distances via the fluid stream and that the effectiveness of the heavy wall pipe or external insulation terminates where the eatment is terminated.

oise Analysis

Control valve noise can be accurately predicted, however, to the best knowledge of the thor, reliable techniques for predicting ise resulting from other sources in the sys- m (i.e., abrupt expansions, pumps, compressors, pipe fittings, etc.) are not available. hus, actual measurement of the sound diated to the atmosphere may be required determine the total contribution made by e piping system to the overall noise level.

A study of sound transmission loss through e walls of commercial piping indicated the asibility of converting pipewall vibrations sound levels. Further study resulted in a alid conversion technique.[5]

coustic Power to adiation Efficiency

Basic to the vibration-to-sound conversion chnique is the relationship between acoustic power and radiation efficiency. Ideally, the essure of an acoustic wave is proportional to e particle velocity of the medium through hich the wave passes, with the constant of oportionality being the acoustic impedance that medium. At the surface of a pipe, particle velocity is assumed equal to the velocity at hich the pipe wall is vibrating. From this, oustic wave pressure at the wall can be reted ideally to wall velocity by:

$$p = \rho_0 c_0 v \qquad (11)$$

here p and v are root-mean-square values. lote: all equation terms are defined in the menclature list.)

It is helpful when discussing the transfer of acoustic energy from one location to another to utilize the parameters acoustic power and acoustic pressure. Acoustic pressures exist as a result of a net acoustic power flow through a finite area. Given a power level, an increase in area through which the power flows results in a decrease in pressure acting on that area.

Acoustic power is related to acoustic pressure by the following general formula:

$$W = \frac{p^2 A}{\rho_0 c_0} \qquad (12)$$

Substituting Equation (11) into Equation (12) yields the ideal acoustic power radiated by the vibrating pipe surface where the area of interest (A) is the surface of the pipe.

Therefore:

$$W_I = \pi D \ell \rho_0 c_0 v^2 \qquad (13)$$

An actual acoustic power can be calculated from Equation (12) based on sound pressure level measurements at a point away from the pipe surface. The area term would be that of a cylinder with a radius equal to the distance (r) from the observer to the centerline of the pipe. The actual power may be written from Equation (12) as:

$$W_A = \frac{2\pi r \ell p^2}{\rho_0 c_0} \qquad (14)$$

A radiation efficiency term (σ) can be defined as the ratio of the actual acoustic power to the ideal acoustic power (W_A/W_I). Using this definition and Equations (13) and (14) the relationship between the velocity of the pipe wall and the acoustic pressure at a point in space is:

$$p^2 = \rho_0^2 c_0^2 v^2 \frac{D}{2r} \sigma \qquad (15)$$

It is now evident that if the radiation efficiency is known, the conversion from wall vibration velocity to acoustic pressure at any point in a free field can be made easily.

The efficiency with which a surface radiates sound is a function of frequency. A coincident frequency (f_c) can be defined as the frequency at which the propagation velocity of a flexural wave in the pipe surface equals the velocity of sound in the acoustic medium. Coincident frequencies for many common steel pipes (air as the acoustic medium) are provided in Table III.

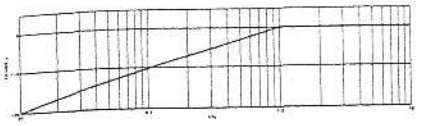


Figure 31. ABOVE COINCIDENT FREQUENCY (f_c), RADIATION EFFICIENCY (σ) EQUALS UNITY; BELOW, IT IS DIRECTLY PROPORTIONAL TO FREQUENCY.

Earlier studies indicate that radiation efficiency is equal to unity above the coincident frequency and is directly proportional to the frequency in the region below coincidence. This is shown in Figure 31. In summary:

$$p^2 = v^2 \rho_0^2 c_0^2 \frac{D}{2r} \frac{f}{f_c} \qquad f < f_c \qquad (16)$$

$$p^2 = v^2 \rho_0^2 c_0^2 \frac{D}{2r} \qquad f \geq f_c \qquad (17)$$

Acceleration measurements are often made rather than velocity measurements. The relationship

$$v^2 = \frac{a^2}{\omega^2} \qquad (18)$$

can be used in Equations (16) and (17) to yield:

$$p^2 = a^2 \frac{\rho_0^2 c_0^2}{4\pi^2} \frac{D}{2r} \frac{1}{f f_c} \qquad f < f_c \qquad (19)$$

$$p^2 = a^2 \frac{\rho_0^2 c_0^2}{4\pi^2} \frac{D}{2r} \frac{1}{f^2} \qquad f \geq f_c \qquad (20)$$

TABLE III. COINCIDENCE FREQUENCIES (f_c in Hz) OF STEEL PIPE IN AIR.

PIPE DIA.	PIPE SCHEDULE 40	80	160
2	3241	2290	1455
4	2106	1481	939
6	1783	1155	695
8	1550	998 †	551
10	1367	998 †	---
12	1331*	998 †	---
16	1331*	998 †	---
24	1331*	998 †	---

*Standard wall
†XS wall

Considerations

Certain limitations must be recognized. As presented, the theory is appropriate for shell modes only. Shell mode reponse is what would be present if a flat vibrating plate were rolled into a cylinder. At low frequencies, however, the shell modes are not present and the response of the pipe is due to the entire length of pipe acting as a beam. These beam modes can vibrate with very high amplitudes; however, the efficiency of their coupling to the acoustic field on the outside of the pipe is extremely low. Recent studies have shown that the radiation efficiency for shell modes is considerably greater than for the beam modes.

TABLE IV. FIRST SHELL MODE FREQUENCIES (HZ).

PIPE DIA.	PIPE SCHEDULE 40	80	160
2	3115	4675	8289
4	1301	1940	3363
6	694	1124	2053
8	466	756	1517
10	338	475 †	---
12	244*	332 †	---
16	153*	208 †	---
24	67*	90 †	---

*Standard wall
†XS wall

This means a vibration measurement may indicate high energy content at low frequencies with very little contribution to the observed sound pressure level. Frequencies associated with the lowest shell mode are tabulated for each standard pipe size in Table IV and should be considered as a low frequency cutoff for the direct application of the theory. This is generally not restrictive in evaluating control valve noise or other broadband high frequency noise.

In order to use the preceding formulation it is necessary to convert the mean-square values to decibels. This can be accomplished using the following definitions:

1) Sound pressure level in dB,

$$SPL = 10 \log \frac{p^2}{p_0^2}$$

2) Wall velocity level in dB,

$$VdB = 10 \log \frac{v^2}{v_0^2}$$

3) Wall acceleration level in dB,

$$AdB^{1} = 10 \log \frac{a^2}{a_o^2}$$

Widely accepted values for the various reference parameters are:

$P_o = .0002$ dynes/cm²

$v_o = 10^{-6}$ cm/sec

$a_o = 10^{-3}$ cm/sec²

Using the above definitions, Equations (16) and (17) can be changed to decibel notation.

$$SPL = 10 \log \frac{v^2}{10^{-12}} + 10 \log \frac{D}{2r} + 10 \log \frac{f}{f_c} - 13.7 \qquad f < f_c \tag{21}$$

$$SPL = 10 \log \frac{v^2}{10^{-12}} + 10 \log \frac{D}{2r} - 13.7 \qquad f \geq f_c \tag{22}$$

If the absolute value of the wall velocity is obtained with a vibration meter for example, then this value may be substituted directly in the first term (v) on the right hand side of the equations. This velocity must be expressed in cm/sec.

When velocity measurements are taken in decibels referenced to 10^{-6} cm/sec, this velocity dB level may be substituted for the entire first term $\left(10 \log \frac{v^2}{10^{-12}}\right)$.

In the same manner, Equations (19) and (20) may be changed to decibel notation for acceleration measurements.

$$SPL = 10 \log \frac{a^2}{10^{-6}} + 10 \log \frac{D}{2r} - 10 \log f f_c + 30.4 \qquad f < f_c \tag{23}$$

$$SPL = 10 \log \frac{a^2}{10^{-6}} + 10 \log \frac{D}{2r} - 20 \log f + 30.4 \qquad f \geq f_c \tag{24}$$

Absolute values of the acceleration in acceleration levels are taken in decibels relative to 10^{-3} cm/sec² then this level may be substituted for the term $10 \log \frac{a^2}{10^{-6}}$.

When vibration levels are taken in decibels relative to a reference value other than presented here, it is then necessary to equate an absolute value using the definitions of velocity-dB or acceleration-dB and substituting for (v) or (a) in Equations 21 thru 24.

The equations presented for velocity (21, 22) and acceleration (23, 24) allow conversion as a function of frequency. Ideally, to gain an overall equivalent sound pressure level from vibration measurements a summation of the corrected levels from each frequency band would be made. This is essential when acceleration is the quantity measured. However, when converting overall wall velocity to overall acoustic pressure, Equation 22 serves as a reasonable approximation as long as the velocity spectrum is not dominated by low frequency components.

Conversion Examples

Examples of converting velocity and acceleration level measurements to equivalent sound pressure levels illustrate the previous analysis.

Velocity

The objective is to determine the equivalent sound pressure level that would be observed at a point 29 inches from the surface of the pipe.

Given:

1) Velocity band levels measured on a 12 inch standard wall pipe are plotted in Figure 32.
2) Equations 21 and 22 apply.
3) From Table III, $f_c = 1331$ Hz.
4) Pipe diameter = 12.75 inches; therefore r = 29 inches + 6.375 inches.

Solving for the correction factor SPL-VdB:

$$SPL\text{-}VdB = 10 \log \frac{D}{2r} + 10 \log \frac{f}{f_c} - 13.7$$

$$= -7.4 + 10 \log \frac{f}{f_c} - 13.7$$

$$= -21.1 + 10 \log \frac{f}{f_c} \quad \text{when } f < f_c$$

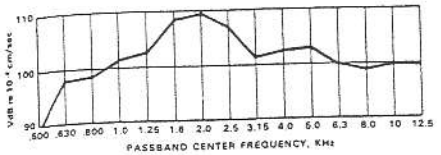


Figure 32. VELOCITY LEVEL IN dB AS A FUNCTION OF FREQUENCY FOR 12 INCH STANDARD WALL PIPE.

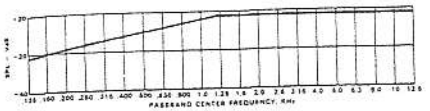


Figure 33. TYPICAL CORRECTION FACTOR CURVE FOR USE WITH VELOCITY LEVEL MEASUREMENTS.
Based on 12 inch Standard Wall Pipe.

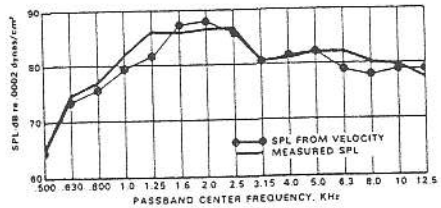


Figure 34. COMPARISON OF ACTUAL SPL TO SPL BASED ON VELOCITY MEASUREMENTS, 12 INCH STANDARD WALL PIPE.

Equivalent sound pressure levels for each 1/3 octave band are found by algebraically adding the SPL-VdB factor to the velocity band level determined from Figure 32. For example at 1331 Hz, the measured VdB equals 106. Adding this to the SPL-VdB value yields an equivalent SPL of 84.9.

When faced with a number of conversions for a given pipe diameter, the task of solving the velocity equations can be lessened by developing a SPL-VdB factor curve as illustrated in Figure 33.

Figure 34 compares actual SPL and equivalent SPL as derived through use of Equations 21 and 22.

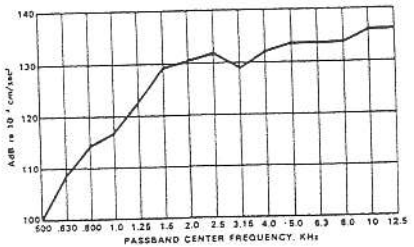


Figure 35. ACCELERATION LEVEL IN dB AS A FUNCTION OF FREQUENCY FOR 12 INCH STANDARD WALL PIPE.

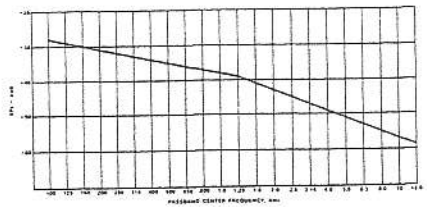


Figure 36. TYPICAL CORRECTION FACTOR CURVE FOR USE WITH ACCELERATION LEVEL MEASUREMENTS.
Based on 12 inch Standard Wall Pipe.

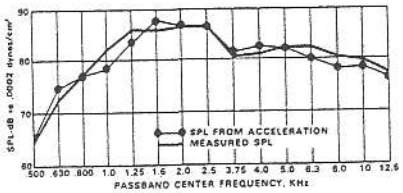


Figure 37. COMPARISON OF ACTUAL SPL TO SPL BASED ON ACCELERATION MEASUREMENTS, 12 INCH STANDARD WALL PIPE.

Acceleration

Determine the equivalent sound pressure level that would be observed at a point 29 inches from the surface of the pipe.

Given:

1) Acceleration band levels measured on a 12 inch standard wall pipe are plotted in Figure 35.
2) Equations 23 and 24 apply.
3) From Table III, $f_c = 1331$ Hz.
4) Pipe diameter = 12.75 inches; therefore r = 29 inches + 6.375 inches.

Solving for the correction factor SPL-AdB:

$$\text{SPL-AdB} = 10 \log \frac{D}{2r} - 10 \log ff_c + 30.4$$

$$= -7.4 - \log ff_c + 30.4$$

$$= 23.0 - 10 \log ff_c \text{ when } f < f_c$$

$$= 10 \log \frac{D}{2r} - 20 \log f + 30.4$$

$$= 23.0 - 20 \log f \text{ when } f \geq f_c$$

Equivalent sound pressure levels for each 1/3 octave band are found by algebraically adding the SPL-AdB factor to the acceleration band level determined from Figure 35. As with velocity measurement conversions, the equation solving task can be eased by developing a SPL-AdB factor curve as shown in Figure 36.

Figure 37 compares actual SPL and equivalent SPL as derived through use of Equations 23 and 24.

Equivalent overall sound pressure levels can be obtained from the corrected vibration measurements by summing the energy in all frequency bands. This yields the following results for the data in the sample problems:

1) Total SPL (measured data) is 94.7 dB.
2) Total SPL (corrected velocity) is 94.8 dB.
3) Total SPL (corrected acceleration) is 94.6 dB.

Measurement Techniques

The accuracy of the conversion method is a function of the accuracy with which the vibration measurements are taken. A typical vibration measuring system consists of an accelerometer to sense the vibrations plus analysis instrumentation for displaying or storing the information.

When making measurements care should be taken that the frequency response of the measuring equipment is compatible with the intended application.

Assuming that all equipment is operating properly, the important variable in making a measurement is the attachment of the accelerometer to the pipe wall. Rigid attachment to the pipe wall is critical to obtain accurate field results.

Ideally, the accelerometer should be rigidly attached to a small metal pad that is welded to the pipe. Also, some device should be used to electrically isolate the accelerometer from the pipe. An insulated stud or washer placed between the surfaces is satisfactory. This attachment method will yield valid information over the entire frequency range for which the particular probe is specified.

An alternative is to attach pads or studs to the pipe wall using an adhesive. As long as a stiff, thin layered adhesive is used this method can be effective over the specified range of the probe. Different adhesives are necessary depending on the temperature of the application.

Magnetic attachments should be of special design to give a firm bond to a cylindrical surface. Even with a good magnetic attachment, the high frequency response is limited. If a magnetic base is used, then the surface should be clean of paint and dirt to ensure maximum contact.

Hand held accelerometers generally are limited to a very low frequency measurements.

This theory cannot be used if the accelerometer is located on a flange, elbow, valve body, or other pipe fitting. Measurements should be taken a minimum of two diameters from the end of a straight run of pipe.

It should be recognized that vibration measurements are not a panecea for all of the problems associated with noise analysis of a fluid transmission system. Sound and/or vibration surveys can quantify the noise levels radiated by a fluid transmission system, however, because the noise generated within the flow stream is both structureborne and fluidborne it is not necessarily obvious what is the primary source of noise generation and hence which sources (if any) are controlling the sound field.

Closure

From the foregoing discussion it should be obvious that a substantial amount of progress in the area of control valve noise has been achieved in the past very few years. Prior to about 1968, the control valve industry as a whole had done little work on this extremely important subject. Perhaps the consensus was that the noise which these devices generated in performing their jobs was only a price which had to be paid. More probably, however, the real reason was simply the absence of a sufficient reason for the industry to spend the time and money required. Although the control valve buying public may have, by their occasional complaints of excessive noise, helped provide the incentive to attack the problem, the major impetus was furnished by the federal government in the form of legislation designed to protect people from noise pollution.

From strictly noise considerations, there are very few control valve installations which can be considered truly standard. They will be unique from the standpoint of installation geometry, service conditions, noise attenuation requirements, or some combination of these. With so many possible installation variables and the numerous pieces of control valve noise abatement equipment, it becomes extremely important that knowledgeable persons are consulted in the application of this equipment. For example, several approaches may be taken to the same problem. One approach might produce the very quietest installation but at a prohibitive cost whereas another approach could meet the required noise specification and at a substantial saving. Without the ability to predict noise levels and without the choices of equipment, optimizing the given installation from the noise and cost standpoints would not be possible.

Where do we go from here? Comparing current noise technology with other important control valve technologies, such as systems analysis and valve sizing, indicates that the noise technology is in an emerging status. If this be the case, then dramatic progress can be expected in the immediate future. Studies are in progress which are intended to increase the understanding of the noise generation mechanisms and identify parameters not presently being considered. These studies and others should generate new and more efficient items of equipment and result in more precise techniques for the prediction of control valve noise.

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2. Buresh, James F. and Schuder, C. B., "Development of a Universal Gas Sizing Equation for Control Valves", TM-15, Fisher Controls Co., Marshalltown, Iowa 50158.
3. Stiles, G. F., "Development of a Valve Sizing Relationship for Flashing and Cavitating Flow", ISA Final Control Elements Symposium, Wilmington, Delaware, 1970.
4. Fagerlund, A. C., "Transmission of Sound Through a Cylindrical Pipe Wall", ASME, November 1973.
5. Fagerlund, A. C., "Conversion of Vibration Measurements to Sound Pressure Levels", TM-33, Fisher Controls Co., Marshalltown, Iowa 50158.

NOMENCLATURE

a	- rms acceleration
a_o	- Reference acceleration
c	- Velocity of sound
c_a	- Velocity of sound in atmosphere
c_i	- Velocity of sound in medium inside pipe
C_g	- Gas sizing coefficient
c_o	- Ambient wavespeed
C_u	- Cauchy number $\frac{\rho v^2}{E}$
C_p	- Specific heat at constant pressure
C_v	- Specific heat at constant volume
d	- Pipe diameter
dBA	- Decibel, A-weighting
D	- Characteristic diameter
E	- Modulus of elasticity
E_u	- Euler number $\frac{\rho v^2}{2(p1-Pv)}$
f	- Frequency
f_c	-Coincident frequency
f_{ca}	- Coincident frequency with atmosphere
f_{co}	- Cutoff frequency
f_r	-Ring frequency
F	- Friction factor
k	- Specific heat ratio, Cp/Cv
ℓ	- Pipe length
L	- Characteristic length
M	- Mach number, V/c
p	-rms acoustic pressure
P_1	- Upstream pressure, psia
Po	- Reference pressure
ΔP	- Pressure differential
Pr	- Prandtl number, Cp μ/λ
Ps	- Sound pressure rms
Pv	- Vapor pressure at upstream conditions
r	- Radial distance from axial centerline of pipe
Re	- Reynolds number, μ VD/λ
SPL	-Sound pressure level
t	- Pipe wall thickness
V	- Velocity
v	- rms wall velocity
vo	- Reference velocity
W	- Acoustic power
W_A	- Actual acoustic power
We	- Weber number $\frac{\rho D v^2}{\sigma}$
W_I	- Ideal acoustic power
ζ	- Radiation efficiency
λ	- Thermal conductivity
μ	- Absolute viscosity
ν	- Kinematic viscosity
ρ	- Density
ρa	- Density of atmosphere
ρ_o	- Density of water at 60°F

PART 2
UNIVERSAL VALVE NOISE PREDICTION METHOD
(FOR NON PROPRIETARY VALVE STYLES)
(H.D. Baumann)

The following equations provide a mathematical approach, not predicted on proprietary designs, for the estimation of control valve noise. The noise level generated when cavitation occurs on high pressure drop liquid applications or the aerodynamic noise created on gas or steam applications can be accurately predicted, usually within plus or minus three dbA. Graphical charts developed from equations are included to aid in calculating the noise level in dbA as measured three feet from the pipe surface and three feet downstream of the valve outlet.

Noise Calculation for Cavitating Valves (Mathematical)

With liquid flow applications the sound pressure level produced by cavitation within valves operating under choked flow conditions[1] may be predicted by the following empirical equation suitable for Schedule 40 steel piping:

$$SPL = 10 \log_{10} (C_v F_L) + 8 \log_{10} (P_{2crit.} - P_2) + 20 \log_{10} (P_2 - P_v) + 33$$

Where:

SPL = sound pressure level at 3 feet, dbA (ref. 10^{-4} microbar).
Cv = actual required valve sizing coefficient.
F_L = liquid pressure recovery factor at given valve stroke.

Typical values are given in Table I.

$P_{2crit.} = P_1 - (F_L)^2 (P_1 - P_v)$, psia[2]

P_1 = static pressure immediately ahead of the valve, psia.
P_2 = static pressure immediately after the valve, psia.
P_v = fluid vapor pressure psia.

Note: The noise is of a very broad frequency band and is not easily treated with external insulation.

Table V. F_L AND K_c VALUES FOR SOME TYPICAL VALVES

Valve type (100% open unless noted)	Flow-to-open		Flow-to-close	
	F_L	K_c	F_L	K_c
Globe valve, parabolic plug	0.90	0.65	0.85	0.58
Globe valve, reduced trim	0.90	0.65	0.80	0.52
Globe valve, V-port plug	0.98	0.80	0.98	0.70
Globe valve, cage trim	--	--	0.90	0.65
Globe valve, needle trim	0.80	0.64	0.50	0.23
Split-body valve	0.75	0.46	0.80	0.51
Angle valve, venturi type	0.90	0.65	0.48	0.17
Angle valve, parabolic plug	0.89	0.64	0.81	0.53
Y valve	0.75	0.46	0.75	0.46
Butterfly valve (60% open)	0.58	0.32	0.58	0.32
Ball valve, throttling type	0.60	0.24	0.60	0.24

[1]When $\Delta \geq (F_L)^2 (P_1 - P_v)$.
[2]Substitute K_c = "Coefficient of Incipient Cavitation" for $(F_L)^2$ if the difference between K_c and $(F_L)^2$ exceeds 10%.

aphical Noise Calculation Solution For avitating Valves

The following graphical solution is an easy rt cut to the previous mathematical equa-:

$$SPL = A + B - C$$

ere:

SPL = Sound pressure level at 3 feet, dbA (Ref. 10^{-4} microbar).

A & B = values determined from figures 39, dbA

C = value determined from figure 40. Omit factor C if K_c exceeds 0.9 $x(F_L)^2$. Factor K_c compensates for part of total mass flow that does not cavitate between incipient cavitation and full cavitation.

mple
ven:

6 inch butterfly valve handling condene at 305°F.

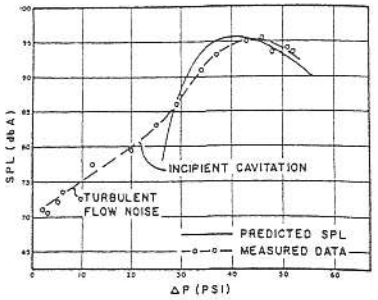

Figure 38. TYPICAL CAVITATING NOISE PROFILE.

P_1	= 200 psia
P_2	= 125 psia
P_v	= 58 psia
C_v	= 328
F_L	= 0.61
K_c	= 0.32

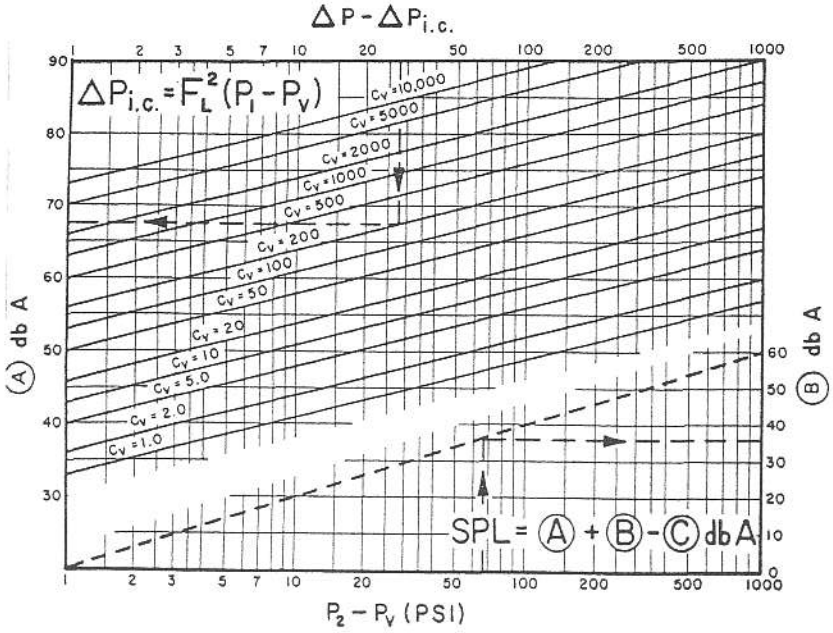

Figure 39. USE THIS GRAPH TO PREDICT VALVE CAVITATION NOISE.

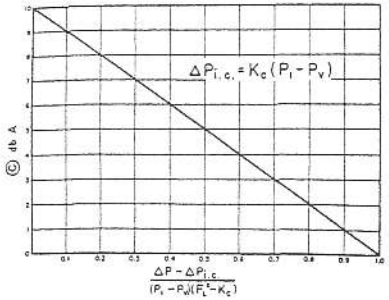

Figure 40. IF THE Kc VALUE IS LESS THAN 0.9 (F_L^2) USE THIS GRAPH TO OBTAIN THE AMOUNT TO SUBTRACT.

From: Figure 39
A = 67.5 because

$$\Delta P_{i.c.} = F_L^2 (P_1 - P_v)$$
$$= .61^2 (200 - 58)$$
$$= 53 \text{ psi}$$

$$\Delta P - \Delta P_{i.c.} = 75 - 53$$
$$= 22 \text{ psi}$$

From: Figure 39
B = 36 because
$P_2 - P_v = 125 - 58 = 67$ psi

From: Figure 40
C = 0 because

$\frac{\Delta P - \Delta P_{i.c.}}{(P_1 - P_v)(F_L^2 - K_c)}$ is greater than 1.

Therefore SPL = 67.5 + 36 - 0 = 103.5 dbA.

Figure 38 illustrates a typical cavitating noise profile.

Aerodynamic Valve Noise Calculation (Mathematical)

The following mathematical method can be used to estimate the sound power and sound pressure level of the aerodynamic noise developed by control valves of any style operating under critical pressure drop[1] conditions on air, gas or steam service.

[1] When ΔP is more than $P_1/2$.

Sound Power Level

$$PWL = 10 \log_{10} (X\eta\ 1.5\ C_v\ F_L\ P_1\ 10^{13}).$$

Where: PWL = Sound Power Level before pipe attentuation, db (Ref. 10^{-13} watts).

Sound Pressure Level

$$SPL = 10 \log_{10} (X\ \eta\ 10^{11}\ C_v\ F_L\ P_1\ P_2) - TL + Sg + 3.$$

Where:

SPL = sound presure level at 3 feet, dbA (ref. 2 x 10^{-4} microbar).

X = fraction of mechanical power conversion

$$= \frac{P_1 - P_2}{0.47\ P_1} \text{ (limited to 1)}$$

η = acoustical efficiency factor (See figure 41).

C_v = actual required valve sizing coefficient.

F_L = Valve pressure recovery factor, dimensionless.

P_1 = static pressure immediately ahead of the valve, psia.

P_2 = static pressure immediately after the valve, psia.

T_L = transmission loss, dbA (omit for valves discharging directly to atmosphere)

$$= 17 \log_{10}\left(\frac{15000\ m\ n}{\sqrt{C_v\ F_L}}\right) - 36$$

m = weight of pipe wall, lb/ft² (See Table VI.).

n = number of apparent frequency producing orifices (See Table VII)

Sg = gas property correction factor, db (See Table VIII.)

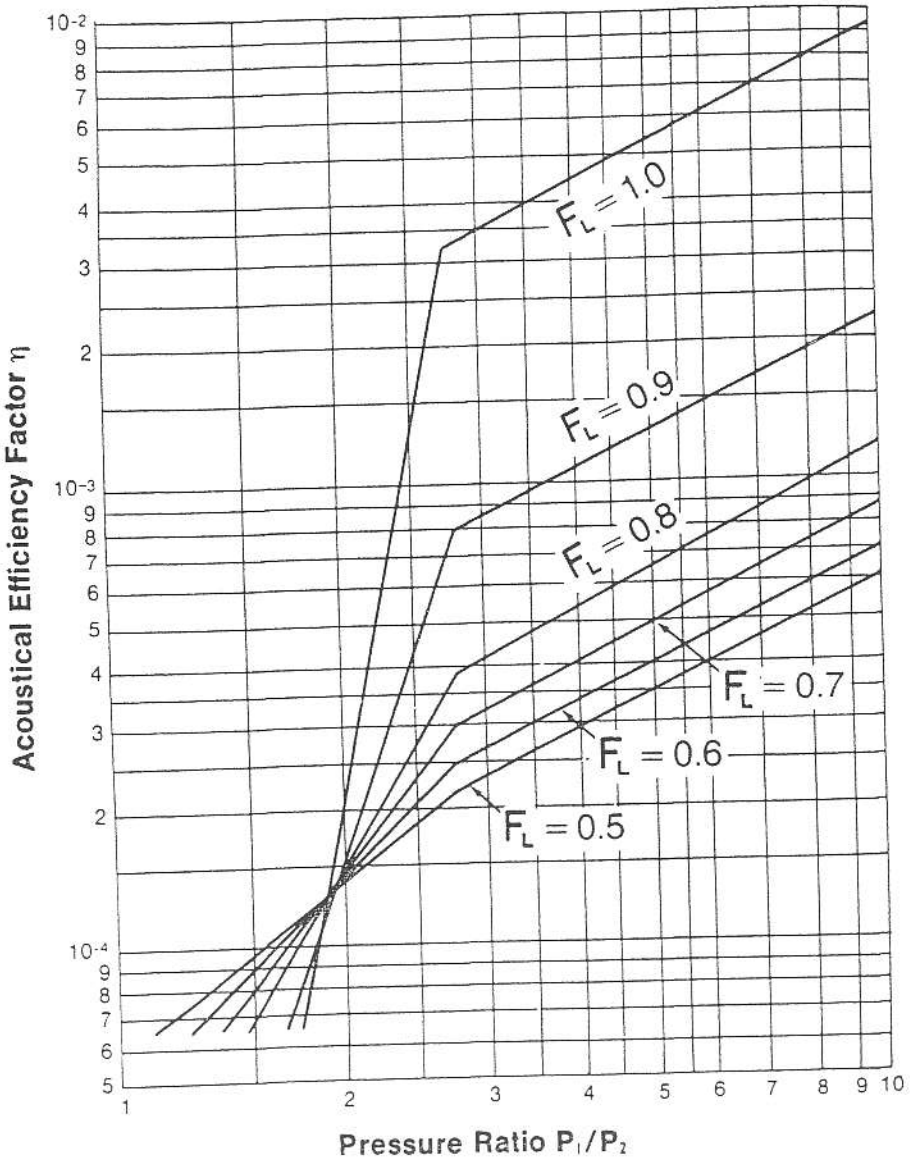

Figure 41. ACOUSTICAL EFFICIENCY FACTOR.

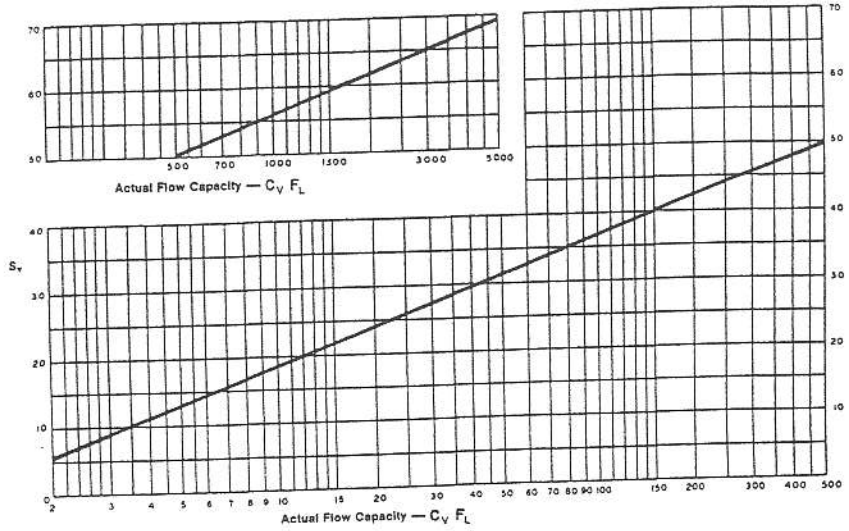

Figure 42. ESTIMATION OF SOUND PRESSURE LEVEL GRAPH.

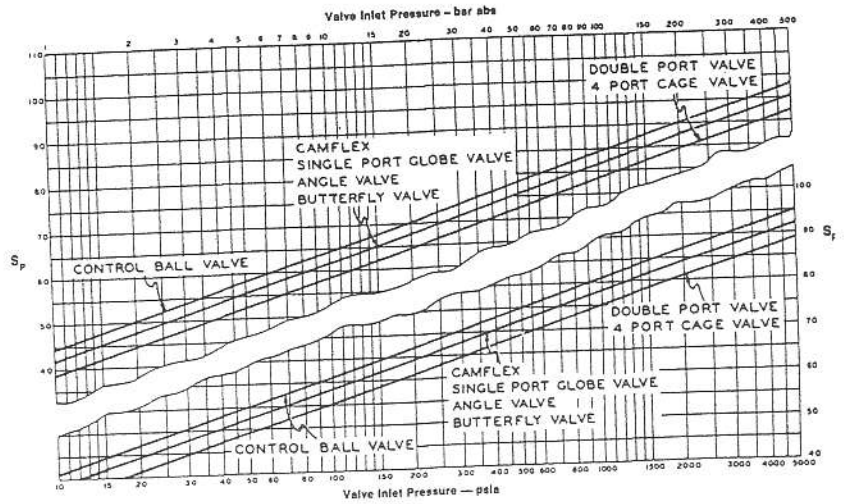

Figure 43. USE THIS GRAPH TO OBTAIN THE S_p VALUE.

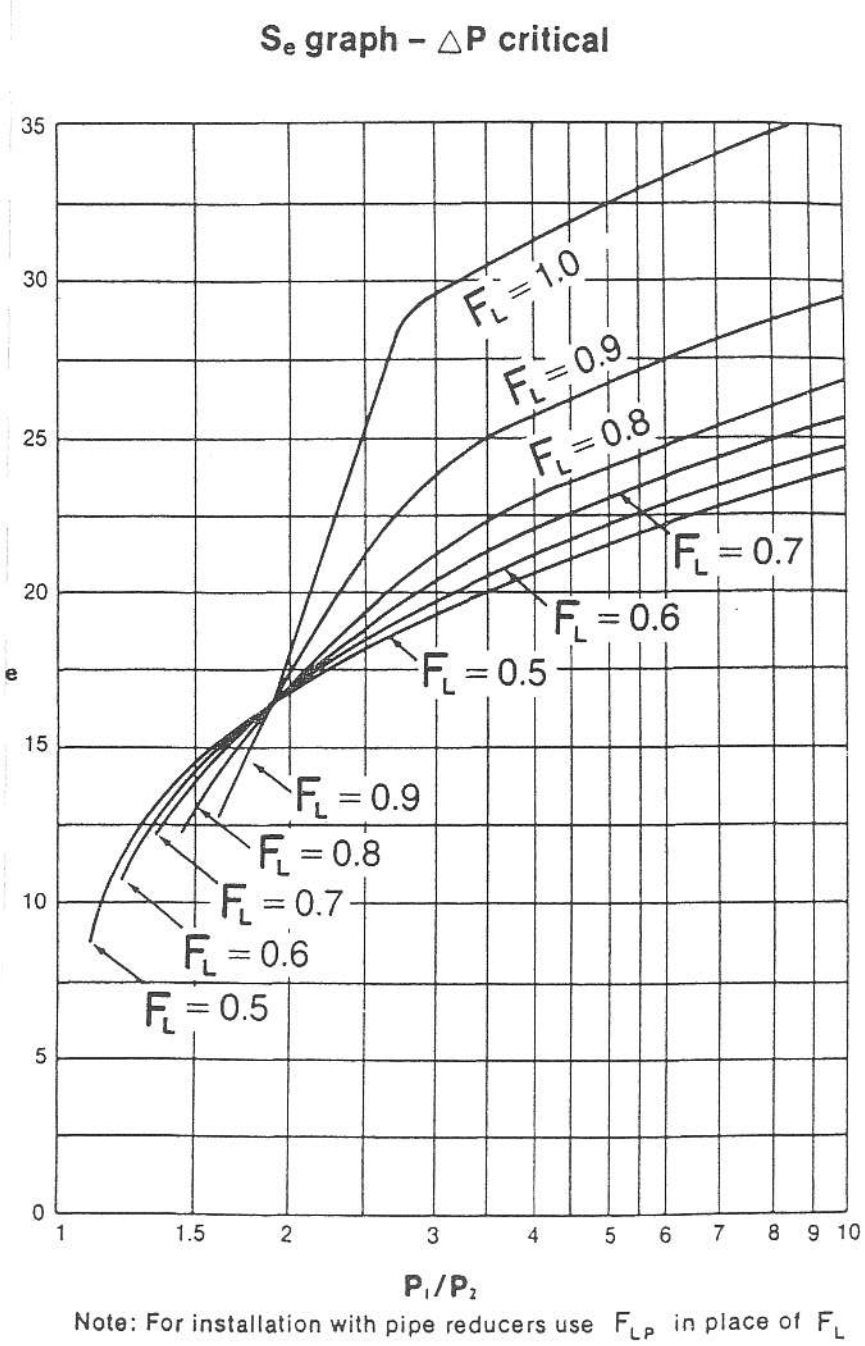

Figure 44. USE THIS GRAPH TO OBTAIN THE S_e VALUE.

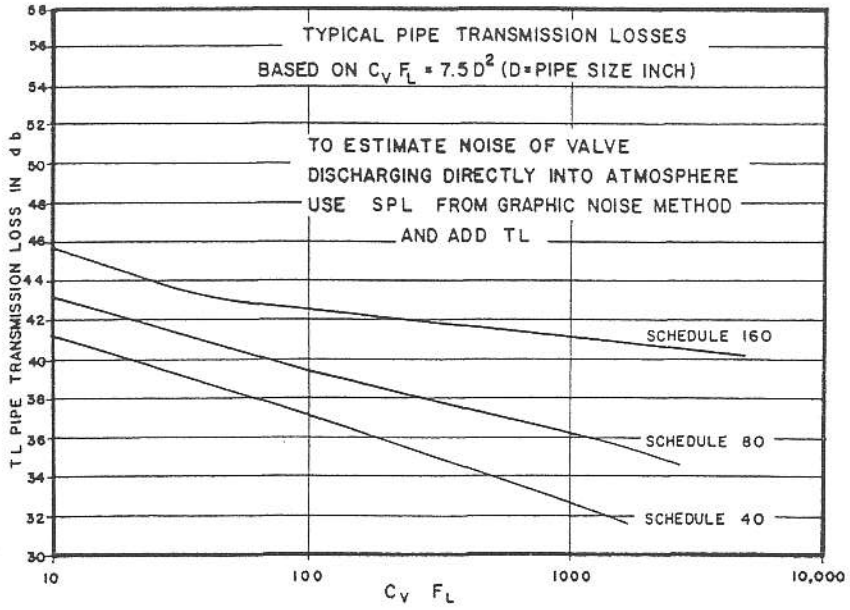

Figure 45. FOR VALVES DISCHARGING DIRECTLY TO ATMOSPHERE ADD THE APPROXIMATE TRANSMISSION LOSS VALUE TO CALCULATED SPL.

TABLE VI. WEIGHT OF PIPE WALL.

Pipe Size (inches)	*"m" Schedule 40* (lb/Ft^2)	*"m" Schedule 80* (lb/Ft^2)	*"m" Schedule 160* (lb/Ft^2)
1	5.4	7.3	10.2
2	6.3	8.9	14
3	8.8	12.2	17.8
4	9.7	13.7	21.6
6	11.4	17.6	29.2
8	13.1	20.4	37.0
10	14.9	24.1	46.0
12	16.5	28.0	53.4

TABLE VII. n FACTOR FOR SEVERAL VALVE TYPES.

Valve Type	n *Factor*
Control ball valve	1.0
Camflex Single port globe valve Angle valve Butterfly valve	1.4
Double port valve 4 port cage valve	2.2

TABLE VIII. GAS PROPERTY FACTOR, Sg.

	(dB)
saturated steam	-2
superheated steam	-3
natural gas	-1
hydrogen	-10
oxygen	+0.5
ammonia	-2
air	0
acetylene	-1
carbon dioxide	+1
carbon monoxide	0
helium	-6.5
methane	-1
nitrogen	0
propane	+1
ethylene	-1
ethane	-1

TABLE IX. S_e TABLE, ΔP subcritical.

P_1/P_2	S_e (F_L=1.0)	S_e (F_L=0.9)	S_e (F_L=0.8)	S_e (F_L=0.7)	S_e (F_L=0.6)
1.15	-4.5	-4.0	-1.0	2.5	7.0
1.20	0.0	0.5	3.0	7.0	11.0
1.30	5.0	5.5	8.5	12.0	
1.40	8.5	9.0	12.0		
1.50	11.0	11.5			
1.60	13.0	13.5			
1.70	14.5				
1.80	16.0				
1.90	16.5				

TABLE X. ATTENUATION FACTOR Sa (dbA).

Not equal to TL

Pipe Size (in.)	Pipe Schedule		
	40	80	160
1	14.5	17	19
1½	15	17.5	20
2	15.5	18	21.5
3	18	20.5	23
4	18.5	21	24.5
6	20	23	27
8	21	24.5	28.5
10	22	25.5	30.3
12	22.5	26.5	31.5
14	23	27	32
16	24	28	33
18	25	29	33.5
20	25	29.5	34.5
24	26.5	31	35.5

Graphical Noise Calculation Solution For Aerodynamic Valves

A convenient solution of the aerodynamic SPL equation is provided by the graphs (Figures 42, 43 and 44) together with Tables VIII, IX and X. This graphical method solves for the SPL values at both critical and subcritical pressure drops.

Briefly stated:.

$SPL = Sv + Sp + Se + Sg - Sa$ [1].

where:

S_v = value from Figure 42, dbA.
S_p = value from Figure 43, dbA.
S_e = value from Figure 44 for critical pressure drop, dbA.
S_e = value from Table IX for subcritical pressure drop, dbA.
S_g = gas property correction factor from Table VIII, dbA.
S_a = attenuation factor from Table X, dbA.

Example:
Given: A 1 inch high pressure angle valve with reduced trim, reducing natural gas. Pipe Schedule is 80.

P_1 = 3460 psia
P_2 = 750 psia
T = 100 F
G = 0.6
q = 330,000 SCFH
C_v = 3.8
F_L = 0.55
C_vF_L = 2.1

[1]For valves discharging directly to atmosphere add the approximate TL (Transmission Loss) from Figure 45 to the SPL calculated by this method. This method is based on a typical

From: Figure 42 S_v = 6.0
Figure 43 S_p = 89
Figure 44 S_e = 21.5
Table VIII S_g = -1
Table X S_a = 17

Therefore SPL = 6.0 + 89 + 21.5 -1 -17 = 98.5 dbA.

Noise Reduction Using Piping Insulation

Figure 46 illustrates noise attentuation which can be obtained (limited to 12 db maximum) using one of three common insulating materials. When insulation is used, it must cover the entire downstream piping system, or at least within the area that should be protected from noise.

Pipe Coincidence Frequency Considerations

Caution should be used when selecting outlet piping. Ideally, the pipe wall thickness should have a coincidence frequency well under the peak valve noise frequency. Figure 47 gives the minimum pipe wall schedule for elimination of a possible error in SPL prediction. On the other hand, higher transmission loss figures can be expected if the peak frequency is *above* the ring frequency of the pipe.

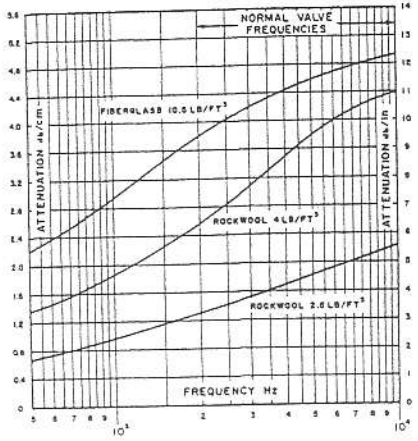

Figure 46. NOISE ATTENUATION WITH COMMON INSULATING MATERIALS.

OSHA Considerations

To meet OSHA requirements the valve *does not* have to meet the 90 dbA SPL (8 hour exposure) level. This applies only to the location of the nearest worker. If a worker is located a given distance *away* from the valve and piping, then you can add the following to the allowable limit (85 or 90 dbA) in order to find the maximum allowable valve noise:

Distance of Valve or Pipe from Worker	Add to OSHA Limit
10 feet	5 dbA
20 feet	8 dbA
50 feet	12 dbA
100 feet	15 dbA

Example: Personnel is permanently located in a control room 50 feet from a pressure reducing valve and its downstream piping. In order to keep the noise level below 85 db in the control room, valve noise should not exceed 85 + 12 = 97dbA. (from table)

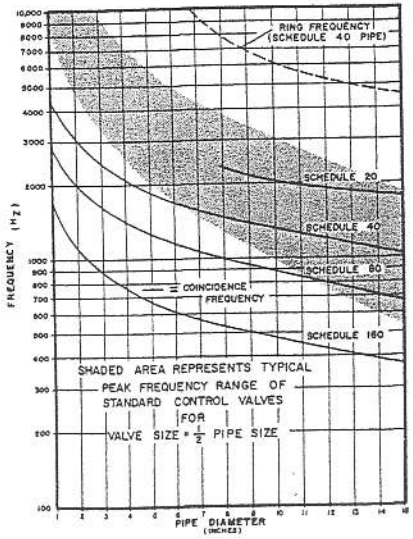

Figure 47. MINIMUM PIPE WALL SCHEDULE FOR ELIMINATION OF A POSSIBLE ERROR IN SPL PREDICTION.

Rules of Thumb

1) Valve noise does not exceed 90 dbA if product P1 x Cv does not exceed 1000.
2) If thermal or acoustical insulation is used subtract about 10 db from calculated valve noise.
3) You lose about 5 db when you double the wall thickness of the pipe.
4) Downstream silencers will reduce noise only 10-15 db unless upstream silencer or insulation is used.
5) Downstream fixed Lo-noise restrictions absorb some of the overall valve pressure drop and can reduce noise up to 20 db.
6) Regardless of worker location do *not* allow valve noise (before insulation, silencers, etc.) to exceed 105 dbA. This number represents a pipe acceleration level of ≈152 db at a typical peak frequency of 4000 Hz and is equivalent to a G level of 40. Major damage in trim and attached instruments can be expected if this level is exceeded.

REFERENCES

1) Baumann, H. D., "Control Valve Noise: Its Cause and Cure" TEXAS A & M SYMPOSIUM PROCEEDINGS, January 1971.
2) Baumann, H. D., AND Villier, R., "Prevention of Cavitation and of Wire Drawing Helps Modern Valves Control H-P Liquids", POWER, May 1973.
3) Baumann, H. D., "On the Prediction of Aerodynamically Created Sound Pressure Level of Control Valves". ASME PUBLICATION WA/FE-28, 1970.

PART 3
COPING WITH CONTROL VALVE NOISE
(J. B. Arant)

Noise Effects

Before getting to the subject of noise with respect to control valves, consider several items concerning its effects upon the human animal. This may serve as an additional stimulus to increase your efforts to correct or prevent noise problems in your place of work and to convince your management that it is a desirable and worthwhile effort—the spear of government legislations in their back notwithstanding. In other words, don't wait until the issue is forced via the OSHA inspector and a violation citation.

Noise is considered by many to be the world's most prevalent pollution. Consider the following items:

Dr. Gerd Jansen of Germany's Max Planck Institute for Industrial Physiology says noise contributes to cardiovascular problems such as heart disease and high blood pressure. Workers in high and low noise exposure areas have been compared. The high noise group had more circulatory problems, cardiac disturbances, neurosensory and motor impairment and even more social conflicts at work and at home.

Dr. Lester Sontag of the Fells Research Institute in Yellow Springs, Ohio has found that noise can influence unborn babies. It can quicken the fetus heart rate and cause its muscles to contract, and prolonged exposure can produce malformations in the nervous system that may affect behavior later in life. Experiments with rats show that unborn fetus subject to auditory stress showed disorganized behavior when grown.

The contribution of noise to hypertension can result in gastrointestinal problems such as peptic ulcers. Russian studies in this area show that workers in high noise areas had twice the incidence of hypertension and up to four times the incidence of peptic ulcers. Also, ten times the incidence of impaired hearing and twelve times the subjective complaints as compared to workers in a low noise environment.

Exposure to noise reduces the depth and quality of sleep and thus may affect overall mental and physical health. A two-year British study found that persons living near London's Heathrow Airport had a significantly higher rate of admission to mental hospitals than persons living in a socially similar but quieter area.

All of these items show that noise is extremely detrimental to humans. Some have called it "a slow agent of death". Of course, the one end result that most often comes to mind is hearing loss and deafness. This type of noise damage is so subtle and insidious that by the time it becomes apparent, it is too late. The damage is irreversible. The World Health Organization estimates that noise costs the United States more than $4 billion annually in accidents, absenteeism, inefficiency and compensation claims. Of course, not all of this is due to industrial noise, but it certainly bears the lion's share.

Noise Levels

What is noise pollution? It can be defined as unwanted sound or as sound without value. Noise fighters fondly point out that the word apparently derives from the Latin "nausea". As engineers who are either designing control systems involving control valves or are involved in noise control work, we should be concerned with how our equipment contributes to the overall pollution problem. Our main contributor to noise pollution is the control valve: One of the few, sometimes the only source of over 100 dbA sound levels found in chemical, petroleum and power plants. For those not completely familiar with noise terminology, the decibel, or db, is the most commonly used unit of measurement. The decibel measures a sound's intensity or, to the ear, its loudness. The term dbA means decibels to the A-weighted scale, which is an approximation to the human ear. Decibels progress logarithmically, that is, for each jump of 10 db there is a tenfold increase. Thus a 100 db sound is 10 times as intense as 90 db and 100 times as intense as 80 db. The human ear perceives each 10 db increase as an approximate doubling of loudness. It is generally agreed that excessive exposure to noise above 85 dbA can damage hearing from most people. Some few are either more susceptible or less susceptible to this exposure level. Therefore, this points out the importance of semi-annual or yearly hearing audiograms and continuing records.

While we are primarily discussing the industrial worker, let us not forget the neighbors in nearby residential areas. Increasing stringent regulation of noise levels by federal, state, and local government authorities relate to its effect on residential areas. Residential areas are becoming more outspoken and demanding as to their rights against noise intrusion into their daily life. If you think you can ignore or fight this—guess who has the

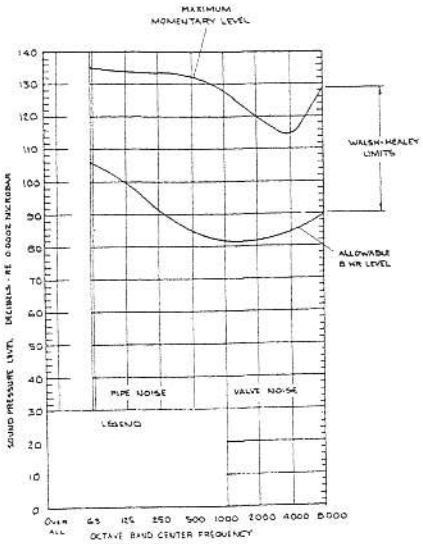

Figure 48. TYPICAL ALLOWABLE SOUND LEVEL CONTOURS.

votes? The original Walsh-Healey Act and the enactment of OSHA are but the beginning. Time is running out on voluntary compliance with noise regulations. You may not be aware that there is considerable effort being made within the Department of Labor to lower this 8-hour exposure from 90 to 85 dbA.

Octave Band Contours

Now let's look at a graph (Figure 48) that shows recommended octave band contours equivalent to the tolerable overall weighted sound levels. An infinite number of octave band SPL (Sound Pressure Level) contours can be drawn for each level, but those shown are typical "white noise" curves. "White noise" is a broad frequency band noise analagous to "white light". Note that in the 1000 to 5000 HZ range, where the human ear is most responsive, the dbA contour is at its lowest level. As mentioned earlier, for hearing protection, an 85 dbA overall level measured 3 feet away from the valve is a good design criteria. Normally, this level will also overall additive effect of other noise sources in the area. This will require, as a rough rule of thumb, that gas or vapor velocity in the valve and piping be limited to less than Mach 0.3

Evaluation Of Installations

Since valve noise is usually the major offender in many plants, understanding of the problem and the possible solutions is a big step towards achieving acceptable noise conditions. As will be discussed later, aerodynamic noise produced in valves is reasonably if not always accurately predictable. Therefore, in the design of new installations, we have a chance to manage the noise pollution problem at the beginning. Existing installations can be similarly evaluated and the proper corrective action taken. Criteria of application are two-fold:

1) Will the selected hardware be effective over the range of operating conditions?
2) What is its cost relative to other noise control systems?

For a roughcut, it may be useful to classify valves into two groups—under 100 dbA and over 100 dbA SPL. Those over 100 dbA are cataloged for source treatment correction, although it may be desirable to take a look at alternate path treatment methods if the noise is close to the 100 dbA level. Valves below 100 dbA have many more treatment options open, including source treatment. Above about 105 dbA, experience dictates that source treatment is the only viable answer. Even so, there are process fluid conditions that may not permit source treatment valves and we must do the best we can with other methods in our bag of tricks.

Sources Of Valve Noise

What are the sources of valve noise? Basically there are three; mechanical vibration, cavitation and aerodynamic. Mechanical vibration and cavitation are usually under 100 dbA with few exceptions. Aerodynamic can be of any level and in most cases is over 100 dbA.

Mechanical vibration is caused by turbulence, pressure oscillations or unsteady flow forces generated by velocity and/or large mass flows. It is normally considered unpredictable. Noise levels are low and are rarely over 90 dbA and in the 50-1500 Hz range. Mechanical vibration with frequencies above this range are due to natural resonant excitation of the valve internals. This type of noise may be in the 90 dbA and up range, with frequencies up to about 7000 Hz. It is characterized by a single-pitch tone and is unusually annoying as compared to broad band or

ginally acceptable as noise in the lower levels, but prudence dictates that it be eliminated because of the risk of valve failure. It often can be eliminated by changing the plug type. The change in mass and spring rate shifts the resonant frequency out of the excitation range. Mechanical restraint such as cage or post guiding will also do the job in many cases. Current work in this area indicates that it will eventually be possible to examine the mechanical design of a valve and predict acoustically driven vibration problems and to devise corrective redesign.

Cavitation noise is a function of the severity level of cavitation. It increases with pressure drop and flow to some maximum level and then decreases to a lower magnitude. This decrease is due to the reduction in downstream pressure recovery to a level near the fluid vapor pressure which results in reduced collapse of bubbles or cavitation. Thus the noise generated is similar to that of a pure flashing liquid. Much work is being done in the area of cavitation noise prediction, although it is still very sketchy and fragmented. Most information available is for water and there is a question as to how well it can be extrapolated to other fluids. Even so, you may get a reasonable lead on your potential noise problem. Maximum cavitation noise levels are generally around 100 dbA. This can be handled by properly selected acoustical insulation since the noise is highly localized. Nevertheless, the associated physical damage cannot be ignored, so reduction or elimination of cavitation is the best solution. There are a number of established ways to do this and a number of valve designs available with varying degrees of effectiveness and applicability. Among these are Hammel-Dahl's "Flash-Flo", Fisher's "Cavitrol I and III", Yarway's "Turbo-Cascade", Cope's and Masoneilan's step trim designs, Valtek's "Dragon Tooth" and Control Components' "Self Drag".

Aerodynamic noise is the commonest and worst source of noise. The mechanisms are complex, but this noise is basically caused by turbulent fluid interacting with laminar fluid in the mixing region or shock waves due to high velocity and/or mass flow of gas. It can be predicted as to occurrence and approximate magnitude. Various manufacturers have developed their own "in-house" methods of prediction, although much additional work is needed and is being done by various groups in and out of industry. This research should lead to better understanding of the driving mechanisms and vastly improved prediction of noise level in any installation configuration.

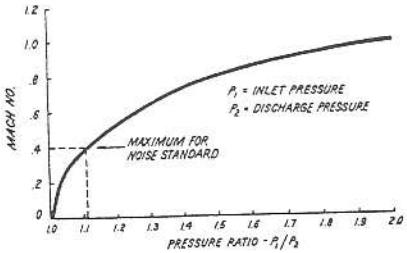

Figure 49. RELATIONSHIP OF PRESSURE RATIO TO MACH NUMBER.

While Mach 1.0 velocity or sonic flow guarantees noise, high noise levels can be generated as low as Mach 0.4 and at relatively low pressure drops where large mass flows are being handled. See Figure 49. It is interesting to compare noise sources. Typical air terminal jet noise is around 117-120dbA. A hard rock music concert is around 110 dbA, although the Rolling Stones on a recent tour hit a high of 126 dbA. Control valves can get up to about 140-150 dbA in some high pressure vent service.

Aerodynamic Noise Considerations

Noise from valve mechanical vibration and cavitation is relatively easy to handle, so let's concentrate on the real problem, aerodynamic noise. Since we can't prescribe the cure until we know the extent of the disease, our first job is to define the magnitude of the problem. As mentioned earlier, several predictive methods have been devised by various valve manufacturers as well as some unpublished industry user methods. All of these reflect "state of the art" and are not always accurate in every situation. At least three companies offer a computer program based upon their in-house method and quite naturally the solutions offered by the computer are based upon their particular product line. While these are helpful, particularly to those users with little or no expertise, doing your own screening has at least two benefits. It will teach you a lot and give you greater insight into noise problems and it will allow you to select from the complete range of solutions regardless of manufacturer.

One quick rule of thumb screening method to start with is the simple equation $P_1 C_v = X$, where P_1 is the absolute upstream pressure

and C_v is the maximum service flow valve capacity coefficient. If X is over 500 but less than 1000, then the noise will probably not exceed about 90 dbA. If X is over 1000, then the noise may be 90 dbA and up. A definite calculation is needed then to define the magnitude of the problem. Again, recall the two group classifications - 90 to 100 dbA and 100 dbA and over. The 90 to 100 dbA group can be evaluated for alternate approaches. Source treatment may be the best engineering solution, but path treatment may offer the best economics for the 10-15 dbA reduction required.

Path Treatment

Path treatment is based upon one or more of the following: distance, transmission loss, dissipation and velocity.

1) Distance means location of the valve far enough away from people, workers or neighbors so that a small amount of noise reaches them. This can be checked by a conservative equation for linear SPL reduction as a function of distance:

$$SPL_{distance} = SPL_{source} - 10 \log \frac{\text{distance in ft.}}{3 \text{ ft.}}$$

This is based upon subsonic valve exit velocity. However, if the outlet goes sonic, then jet shock waves and turbulence mixing due to high velocity becomes the predominant noise source and a source treatment solution is required.

2) Transmission loss can be gained by increasing pipe wall thickness and/or adding acoustical insulation. For example, you can reduce SPL by about 5 dbA in pipe sizes up to 4 inch and about 7-10 dbA from 6 to 24 inches by doubling the pipe wall thickness. Adding one or two inches of acoustical insulation of the proper type can reduce SPL by another 10 dbA. Some tests indicate that there is a large variability in noise loss effectiveness by various commercial acoustic insulations. Surprisingly, it has been found that very little is usually gained beyond a two-inch thickness. Likewise, the frequency band at which these insulations were effective was very definite. You can buy a "pig in a poke" if you don't know what you are doing in this area. It should also be recognized that sound travels down a pipeline. Therefore, increasing the pipe wall thickness and/or applying acoustical insulation to the valve and piping may be a very expensive solution. Typically, acoustic insulation will run about $10-30 per foot installed, depending upon type and size.

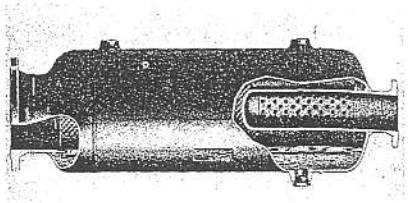

Figure 50. TYPICAL DISSIPATIVE SILENCER.
Courtesy Fisher Controls, Inc.

3) Dissipation is achieved by using a silencer. This can be expensive and may not always be completely effective. Certain criteria must be maintained. Inlet velocity must be subsonic and the silencer cannot serve as a pressure reducer. Likewise, it must be of very heavy wall construction or the silencer can be an excellent sound speaker if the shell resonates. A typical dissipative silencer is shown in Figure 50. Note the inlet diffuser. This is always helpful, since it will break up shock waves or shock cell oscillations that generally characterize the downstream sound field and which can render the silencer ineffective. Experience indicates that there is a maximum effective size for this type of unit, although units of all sizes are offered and can be helpful if properly applied.

4) Velocity must be kept to a minimum to reduce noise. Pipe should be sized so that velocity is no more than 300 feet/sec. Subsonic valves should be sized with body exit diameters to limit the velocity to about Mach 0.3. Otherwise, even if downstream piping is expanded to a larger pipe size, exit turbulence concentrates at the expander and generates noise. It is a fallacy to try to minimize noise by reducing the valve exit velocity via pipe enlargement. What will help are the oversize valve bodies and/or exit diffusers as shown by this typical configuration in Figure 51. As their name implies, diffusers break up the normal exit turbulence. They may also be built to function as pressure drop devices to reduce the control valve pressure drop and fluid velocity. This will reduce noise in a properly designed system. Keep in mind that flow turndown may not allow this to be effective over the entire operating range, although increased valve pressure drop is counterbalanced by lower mass flow noise to some degree.

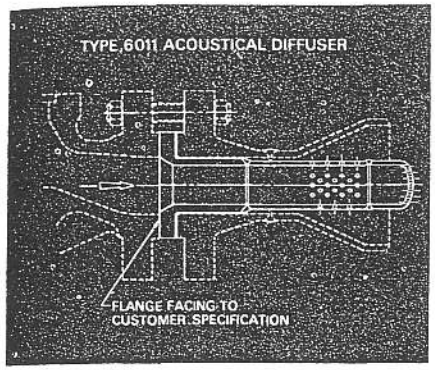

Figure 51. CONTROL VALVE AND EXIT DIFFUSER COMBINATION.
Courtesy Fisher Controls, Inc.

If the predicted noise level is greater than 100 dbA and especially greater than 105 dbA, then the preferred solution is source treatment.

Source Treatment

There are a number of things to consider in treating valve noise at the source, such as:

1) Noise reduction required, dbA.
2) Pressure ratio, P_1/P_2 or $\Delta P/P_1$.
3) Mass flow and turndown requirements.
4) Metallurgy and mechanical design requirements.
5) Other velocity induced problems.
6) Plant operating utility requirements.
7) Economics—the purchase, installation and maintenance costs of the total solutions - not just the valve.

Vendor proposals must be carefully and completely evaluated against these factors. The weight attached to each factor is a matter of judgement and understanding of your particular application and plant needs. Individual component cost is not the primary consideration, as it may be found that the more expensive equipment is in the long run the most economical solution.

Source Treatment Valve Designs

There are four basic source treatment valve designs currently on the market. The first type of design uses multi-path flow geometry in various cage type designs. Typical of these are Fisher Controls' "Whisper Trim I" (Figure 52), ITT Hammel-Dahl's "Flash-Flo" and similar designs from W-K-M, Kieley & Mueller, Serck Controls and Masoneilan. Leslie's "Les-Sonic" and Jordan's "Sliding Gate" regulator designs are also based on the multi-path principle. A properly selected multi-path valve may provide noise attenuation up to 15 dbA if P1/P2 is in range of 2 to 2.5 maximum and the valve exit velocity is about Mach 0.3 maximum. Therefore, these valves can be evaluated for use up to about 100 dbA maximum. A later multi-path design by Fisher Controls called "Whisper Trim III" has been designed for use at higher pressure ratios with a maximum noise attenuation up to about 20 dbA with proper application, although the manufacturer claims that up to 30 dbA has been achieved in some applications. The total solution may require use of a diffuser and/or silencer plus some acoustical insulation.

The second type of design uses a labyrinth step plug with a Christmas tree shape (Figure 53) to provide for volume expansion with

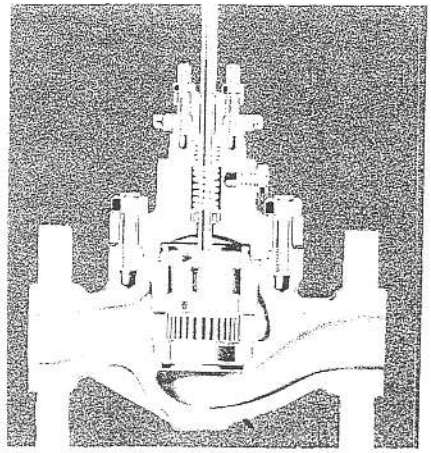

Figure 52. TYPICAL MULTI-PATH NOISE VALVE DESIGN.
Courtesy of Fisher Controls, Inc.

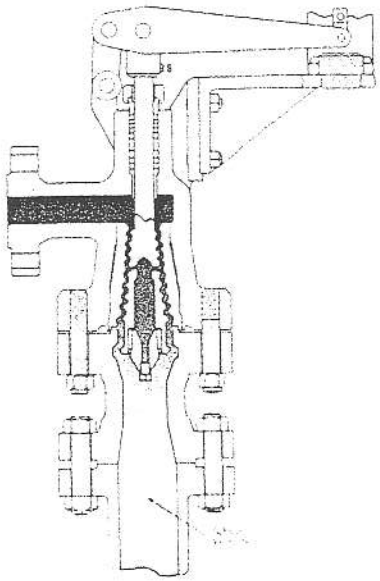

Figure 53. TYPICAL STAGED TRIM NOISE VALVE DESIGN.
Courtesy of Masoneilan International, Inc.

pressure drop. This design comes from Masoneilan and, if properly selected, can provide noise attentuation up to about 20 dbA provided that the P1/P2 pressure ratio is no more than 3 or 4:1. At higher pressure ratios, the exit velocity will exceed Mach 0.3 and will require additonal path treatment such as a line silencer or Masoneilan's "Lo-db Plate" and possibly some acoustic insulation as a total system. The "Lo-db Plate" (Figure 54) deserves a little more discussion since it can be a very useful device. These plates have special orifices which act as a pressure restrictor and noise reducer. They are compact (see Figure 55) and offer economy in installation. Good results can be obtained in properly selected applications with attenuations up to about 10 dbA. They can be used with plug and orifice valves in on/off applications such as venting as well as throttling. In throttling services, they behave similar to diffusers and lose some effectiveness with flow turndown. Again, this is in part compensated by less noise with lower mass flow. Usually it is necessary to install acoustic insulation on the piping between the valve and plate.

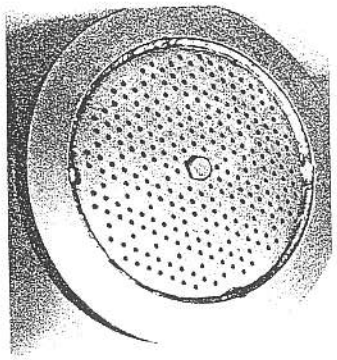

Figure 54. SPECIAL MULTI-PATH NOISE REDUCTION STATIC ELEMENT.
Courtesy of Masoneilan International Inc.

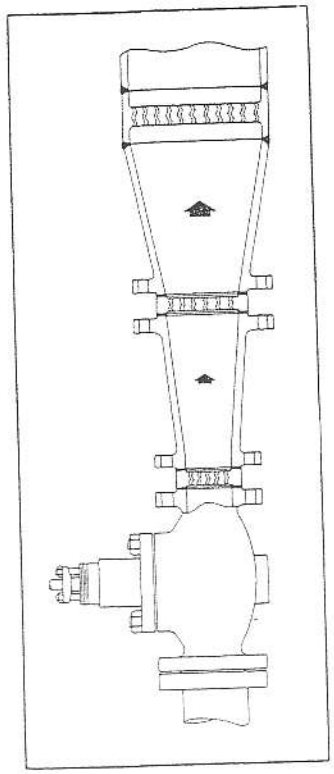

Figure 55. TYPICAL NOISE REDUCTION ELEMENT INSTALLATION.
Courtesy of Masoneilan International Inc.

The third design type (Figure 56) uses multiple short friction passages on a metal plate. These passages have a large length to diameter ratio and maximize the pressure drop by means of velocity head losses with many right angle turns. Path configuration and total number of plates are design variables. We recognize this as the Control Components' "Self Drag" valve as shown in Figure 57. It will provide whatever noise attentuation is needed at any pressure ratio and flow capacity. In this ability it is unique. No other equipment is needed because the trim limits the velocity to about that of the line, normally about Mach 0.3. Care must be taken in applying these valves in dirty services and temporary screens are recommended for initial start-up until line debris is essentially eliminated.

The fourth design type uses multiple concentric grooves on a metal disc. In cross-section these grooves look like V-shaped teeth and intermesh closely with the teeth on the adjacent disc as illustrated in figure 58. The spacing between the grooves across the face and the intermesh can be varied to suit the application. The design is by Valtek and is a variation of their cavitation "Dragon Tooth" trim. This is a variation on a step trim and the short friction passages concept. The discs are stacked in a manner similar to the "Self Drag" valve. A wide range of pressure ratios with noise attenuation in excess of 20 dbA can be obtained with proper application.

Path Treatment versus Source Treatment.

With all this, we can see that a noise problem can be solved by path treatment, source treatment or a combination of the two. Except for low level noise, a general conclusion can be made that in most cases source treatment is the most predictable, effective and economical solution. Especially so in the more difficult situations involving high mass flows, high pressure drops and high temperatures.

The cost of large silencers, support steel, insulation, additional piping and space can be substantial. It is usually at least as much and often more than the cost of special quiet valves. See figure 59. Further, such path treatment is often marginally effective. Evaluations of source treatment versus conventional valves plus silencers almost always show substantial savings with source treatment as illustrated in Figure 60. Even though source treatment valves are costly, savings as much as 25% on the overall system cost can often be obtained. Source treatment has been effective in a wide variety of services. For example: 1500 PSIG steam blow to air, 5000 PSIG process gas blow to air, 300 PSIG process gas blow to air, 1500 PSIG steam reduction to 350

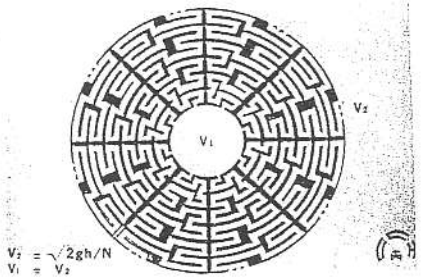

Figure 56. SPECIAL LABYRINTH PATH NOISE REDUCTION DESIGN.
Courtesy of Control Components, Inc.

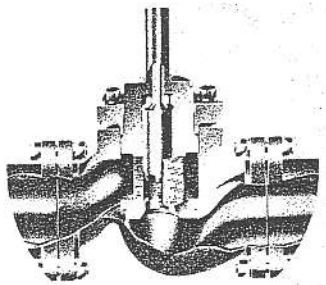

Figure 57. LABYRINTH PATH ELEMENT STACK CONTROL VALVE.
Courtesy of Control Components, Inc.

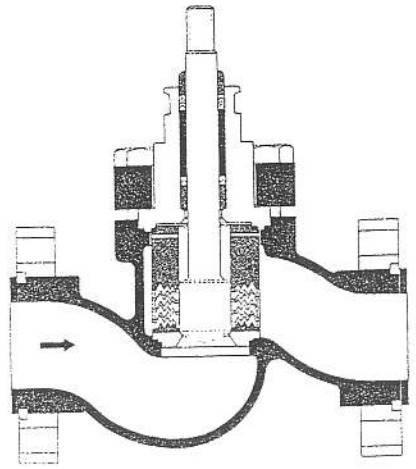

Figure 58. COMBINATION STEP AND LABYRINTH NOISE REDUCTION ELEMENT DESIGN.
Courtesy of Valtek, Inc.

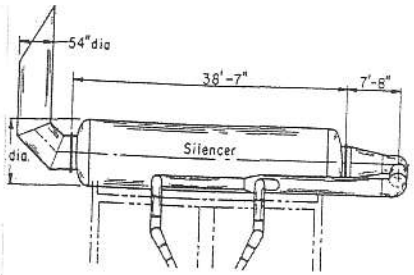

Figure 59. TYPICAL VENT VALVE AND SILENCER INSTALLATION.

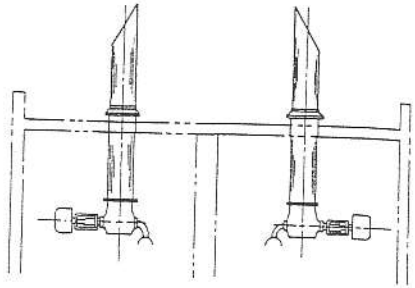

Figure 60. TYPICAL NOISE REDUCTION VENT VALVE INSTALLATION.

SIG. The last mentioned application is so quiet that it is difficult to know if the valve is throttling or shut off except by checking stem position. A typical air vent test shows the type noise reduction that can be obtained with urce treatment. Note figure 61.

In addition, experience with source treatment valves indicates substantial maintenance cost savings can be gained as well as educed plant downtime. Service experience ith parallel valves, one conventional and ne source treatment, showed retrimming osts savings of about $2000 for every 250-300 ours of operation in a large high pressure tdown application. 1500 pound steam vents ave been blown numerous times before reoval was required for maintenance. So an xpensive valve may really end up the most conomical valve on long-range cost. First st isn't everything.

onclusion

Let us examine a basic valve noise relationiip. Figure 62 shows the approximate relaonship between mass flow and sound power vel at various P1/P2 ratios. This gives you a al feel for the relative influence of these ctors. Note that beyond 10:1 pressure ratio, e change in sound power is relatively inor. From 2:1 to 10:1, each increase of level very marked. These curves could be used to lculate the SPL generated outside the pipe. would require calculation of the transmisn loss through the pipe wall, conversion of und power to sound pressure level (SPL) in and converting it to dbA if the peak freency is calculated. This again is a "state of e art" approach. It is expected that continug work in the noise predictive field by manacturers, industry users, and universities ll lead to better understanding of this ghly complex field. With better understandg will come improved predictive methods that are more accurate, cover a broader range of fluid flow conditions and better recognize the influences of the actual installation on the noise generation. Until then, we have several methods available that will hopefully put us in the right ballpark for evaluating solutions.

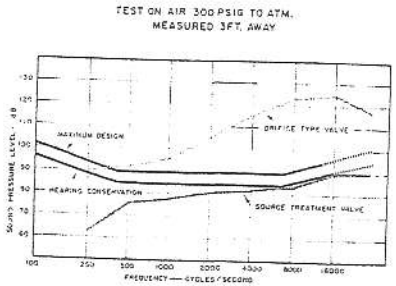

Figure 61. AIR VENT TEST OF STANDARD VALVE VERSUS NOISE REDUCTION VALVE.

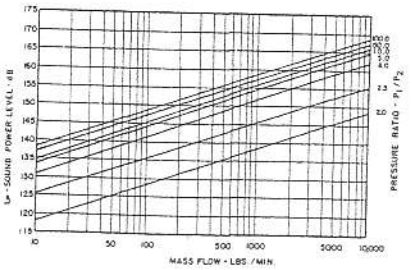

Figure 62. RELATIONSHIP OF MASS FLOW TO SOUND POWER LEVEL AT VARIOUS PRESSURE RATIOS.

PART 4
CONTROL VALVE NOISE RATING: PREDICTION VERSUS REALITY*
(W. L. Scull)

INTRODUCTION

Despite all the methods developed to predict control valve noise over a broad range of operating conditions, expensive miscalculations are still far too common because existing rating methods usually are based upon laboratory measurements of sound pressure levels. A valve rated at a sound pressure level (SPL) of 85 dBA by the manufacturer can produce significantly different noise levels depending on the physical installation.

The old axiom still holds: Specify what is required—no more, no less. If you overspecify by 20 dB, you might double the cost of the control valve that is to be purchased. If you underspecify and the valve generates too high a noise level, you may have to tear up the installation and replace the valve.

There appears to be a general feeling throughout industry that valve noise problems are totally solved and can be handled as routinely and accurately as control valve sizing. This is not true. A lot more work must be done before it will be possible to *accurately predict* installed valve noise, but one can come a lot closer by considering some additional data which can be put to good use.

Noise Rating Methods

Existing noise rating and prediction methods are usually based upon laboratory SPL measurements in a semianechoic test chamber. (ISA Committee SP-59 is now developing standardized methods for making these measurements). Three other factors, however, should be considered in the SPL determination: sound power, directivity, and characteristics of the environment. Directivity is defined as the way in which sound radiates from its source. Acoustical radiation is spherical from a ball or point, and cylindrical from a line source. Figure 63.

Since standard SPL measurements don't take these factors into account, it is important to recognize the limitations of these measurements in terms of predicting noise:

These tests do not provide information on:

- Sound power of the valve.
- Directivity of the noise.
- Propagation of noise in the upstream and downstream piping.

Sound Power versus SPL

The sound power of a device is a constant for any given operating condition—regardless of the environment. For example, a loudspeaker with a given electrical input produces the *same sound power* whether in a small reverberant room or in an open field. But sound pressure level is a function of the acoustic environment, distance and directivity of the noise generating device as well as the operating conditions. For the loudspeaker, the SPL measured in a reverberant room would be sig-

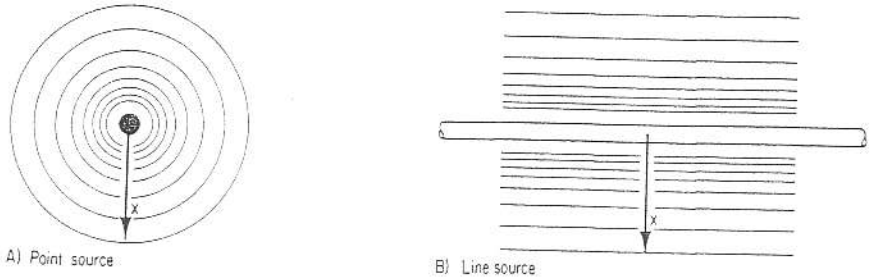

Figure 63. A CONTROL VALVE CAN BE CONSIDERED TO BE A POINT SOURCE (A) OF NOISE; HOWEVER, A VALVE AND ITS PIPING USUALLY BEHAVES MORE LIKE A LINE SOURCE (B).

*This part was originally printed in *Instrumentation Technology*, October, 1974.

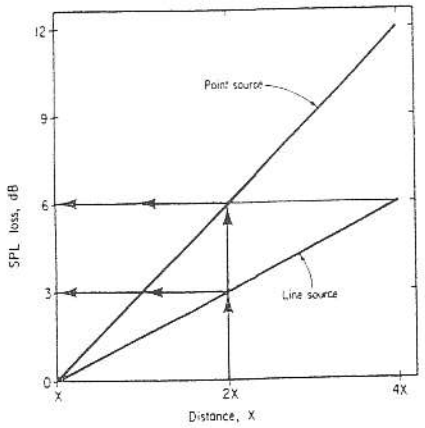

Figure 64. NOISE DECAY FOR A LINE SOURCE IS SIGNIFICANTLY LESS THAN FOR A POINT SOURCE.

nificantly higher than a reading taken at the same distance from the speaker in a free field.

By definition, the sound pressure level created by a device will change if the acoustic environment is modified. Therefore, the SPL measured at an actual valve installation may be significantly different from the valve's laboratory rating.

The only accurate method of predicting the SPL which will be produced in different environments *requires* knowledge of the sound power and directivity of the device, and the characteristics of the environment. Techniques for making these calculations are well established and are discussed in standard texts (e.g. Ref. 1). The availability of sound power and directivity ratings would permit control valve users to more accurately predict and specify *installed noise levels* for any installation.

Problems With Sound Power Ratings

Sound power cannot be measured directly. In the usual procedure, the test engineer places the test specimen in a known acoustic environment and makes measurements of the sound pressure level at several points around the specimen (noise source). Special reverberation chambers can be employed to reduce the number of SPL measurements required. Once the characteristics of the sound field are established, the sound power of the device can be calculated.

Problems in determining valve sound power ratings arise from the definition of the "test specimen." It is obvious that to deter-

entire specimen must be inside the test chamber. This is not a difficult requirement when dealing with loudspeakers, machine tools, or other finite devices.

The definite of "test specimen" for a valve sound power test would theoretically include the entire piping system, since noise travels and radiates for long distances through the pipe wall and fluid. Including an entire piping system in a test chamber is usually impractical, so several alternate schemes have been proposed. These involve breaking of the system into manageable elements and determining the interaction between these elements. For example, if the acoustic and fluid dynamic properties of the fluid exiting the valve were known (in addition to the acoustic characteristics of the piping system), it might be possible to predict the sound power characteristics of the piping at any point in the system without full-scale testing.

Directivity Of Noise

At present, relatively little information has been published on the propagation of noise through fluid piping systems, but the information that is available indicates that pipe and fluid noise transmission cannot be ignored when analyzing valve noise problems. One group of investigators measured attenuation levels over a range of frequencies of only 0.001 to 0.1 dB per foot in a water-filled 2-in. pipe (Ref. 2).

The ability of noise and vibration to travel through the piping and radiate to the air over large distances from the valve has a significant effect on the sound field around the system. Rather than acting as point sources, valve-piping systems can approach a line source of noise in their field characteristics.

The SPL produced by a point source in a free field decays a 6 dB for each doubling of distance, whereas the SPL for a line source decays only 3 dB for each doubling of distance, Figure 64. Even the simplest of calculations (i.e., computing the SPL at some distance other than 3 ft) cannot be accurate without complete information on valve noise directivity.

Field Tests On Valves

An obvious source of noise data on full-scale piping systems is test data on actual field installations. Some of the problems that arise in measuring installed noise levels include the effect of other noise sources in the area, and establishing accurate operating conditions.

Sometimes other noise sources in the measurement area can be shut down, however, the pumps, compressors and other noise sources that are an integral part of the test system cannot be eliminated during the test.

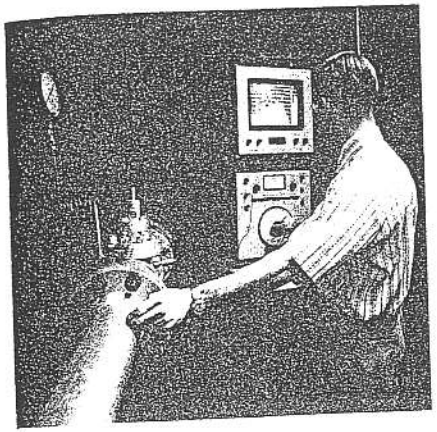

Figure 65. AN ACCELEROMETER IS BEING APPLIED TO A STEAM PIPE IN A SOUND RESEARCH LAB TO INVESTIGATE THE CORRELATION BETWEEN PIPE WALL VIBRATION AND THE AIRBORNE NOISE THAT IT GENERATES.

Courtesy of Leslie Co.

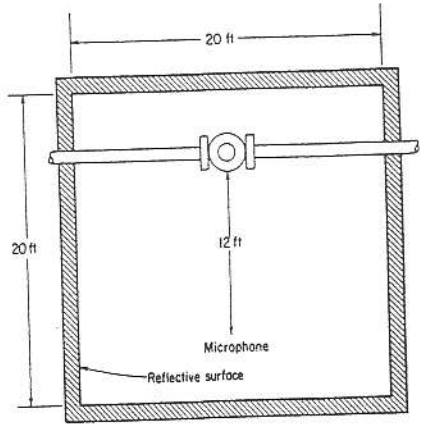

Figure 66. THIS NOISE TEST IS SET UP IN A REVERBERANT INDUSTRIAL ROOM. THE SOUND LEVEL TENDS TO BE UNIFORM THROUGHOUT THE ROOM BECAUSE OF MULTIPLE REFLECTIONS.

One proposed solution to this problem is the use of accelerometers to measure pipe wall vibration. The accelerometer is not as sensitive to the acoustic environment — or extraneous airborne noise — as a microphone. Leslie Co. has investigated this technique monitoring wall vibrations, Figure 65. A method for calculating the airborne noise that can be generated by wall vibration has been developed (Ref. 3).

The problem of establishing the actual operating conditions in field installations can usually be solved by the installation of temporary instrumentation. If upstream and downstream pressures, valve travel, and fluid temperature can be established. Then the flow can usually be calculated to the required accuracy using the valve Cv.

Despite the problems of making noise measurements in the field, actual valve installations are a prime potential source of information.

Improving Noise Predictions

To gauge the need for improved noise prediction methods, one should know the magnitude of errors in the existing methods and assumptions. Ideally, results from a large number of predictions should be compared to actual field data. Since very little data of this type has been published, other methods must be used.

By assuming a range of variables within a conceivable spectrum, one can evaluate the possible range of correct answers against the results of present prediction methods. For example, assume that a valve is rated at 85 dBA when tested in an anechoic (echo free) chamber at a distance of 3 ft with 20 ft of 4-in. piping in the chamber. Now examine the theoretical installed noise levels at the extremes of two variables:

A) Directivity
1) Valve and piping acts as a point noise source.
2) Valve and piping acts as a line noise source.

B) Environment
1) Valve is installed outdoors in free-field conditions.
2) Valve is installed in an industrial room measuring 20x20x10 ft with an absorption coefficient of 0.03 (highly reverberant), Figure 66.

It is apparent that the lowest noise level will be produced by combination A1-B1 and the highest noise level will be produced by combinations A2-B2. Now to calculate the resulting noise levels at some fixed distance (assume 12 ft).

A1 - B1 combination, point source in a free field

Since the original test measurement and the actual installation were made under free-field conditions and the valve is assumed to be a point source, the law of spherical divergence can be used:

$SPL = SPL_R - 20 \log (X/R)$
$SPLx = 85 - 20 \log (12/3) = 73\text{dB}$

where $SPLx$ equals sound pressure level (dB) at the distance X, and $SPLR$ equals sound pressure level (dB) at the distance R. X is the distance (ft) at which the SPL is being calculated, and R is the original measurement distance (ft).

A2-B2 combination, line source in a reverberant room.

To calculate the SPL produced in a reverberant environment, the sound power must be calculated. Assuming symmetrical radiation from a line source, the equation from Reference 2 can be used:

$PWL = SPLR + 10 \log (2\pi RL) - 10.4\ dB$

where the PWL is the sound power level (dB), L is the length of source (ft).

$PWL = 85 + 10 \log (2\pi x3x20) - 10.4\ dB$
$PWL = 100.4\ dB$
From Reference 3:
$S = A\ \alpha$
$SPLc = PWL - 10 \log S + 16.4\ dB$

where S is the total absorption (sabines), A is the room surface area (ft²), α is the absorption coefficient (sabines/ft²), and $SPLc$ is the sound pressure level in a reverberant environment (dB).

$S = [(2x20x20) + (4x20x10)] \times 0.03 = 48$ *sabines*
$SPLc = 100.4 - 10 \log 48 + 16.4\ dB$
$SPLc = 100.0\ dB$

Figure 67 summarizes the calculations. Although the above calculations do not include all the factors that must be considered in a rigorous analysis, they do illustrate the effect of environment and noise directivity on the installed noise level. Information on the sound power and directivity of valve-piping systems would enable the control valve user to calculate actual installed noise levels and accurately specify the required hardware.

Present ability to predict valve sound levels under standardized conditions represents a major advance in the field of valve noise technology. It is important, however, to recognize that present prediction methods *do not provide all* of the information necessary to accurately predict noise levels of actual installations.

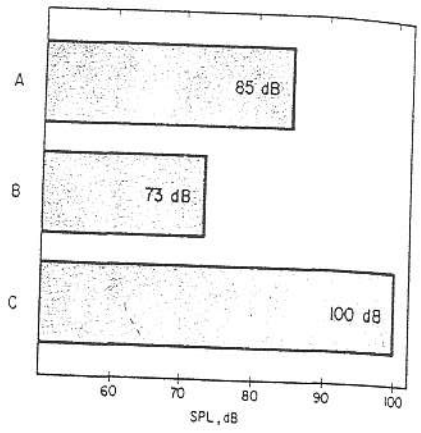

Figure 67. THIS BAR GRAPH PRESENTS A CONTROL VALVE'S SPL RATING AT A DISTAINCE OF (A) 3 FT UNDER LABORATORY CONDITIONS. (B) 12 FT ASSUMING A POINT SOURCE IN A FREE FIELD, AND (C) 12 FT ASSUMING A LINE SOURCE IN A REVERBERANT ROOM.

Sound power and directivity information is needed to predict installed noise levels. Both of these factors are determined by the very complex interactions among the valve, piping, environment and flowing fluid. Developing the required theory and methods is a challenging goal which will require the participation of all sectors of the fluid control field—manufacturers, users and the academic community.

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2. Callaway, D. B., Tyzzer, F. G., and Hardy, H. C., *Journal of the Acoustic Society of America*, Vol. 23, 1951; p. 550 and Vol. 24, 1952, p. 725.
3. Cooper, J. H., Aldridge, L. L., *"Considerations in the Specification of Pipeline Noise," Flow, Its Measurement and Control in Science and Industry*, Vol. 1., pg. 1175, 1971.

CHAPTER 8

CONTROL VALVE ACTUATORS

PART 1
Selection And Types Of Actuators
J. O'Connor, Jr., Kieley & Mueller, Inc.,
Division of Johnson Service Company, Middletown, New York

PART 2
Supplemental Data
J. L. Lyons, Chemetron Fluid Controls,
Division of Chemetron Corporation, St. Louis, Missouri

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PART 1
SELECTION AND TYPES OF ACTUATORS
(J. O'Conner)

INTRODUCTION

There is a wide variety of valve types used in control systems as has been discussed in great detail in prior chapters. Each valve type (i.e., globe valve, butterfly valve, or ball valve) imposes varying problems with respect to actuation.

For example, a standard globe control valve, which is a device to introduce a pressure loss within a system, generates a force that is determined by its ability to induce a pressure loss. This generated force is essentially the pressure drop, multiplied by the port diameter of the control valve. Further, this force is generated in a linear fashion, that is, in the direction the valve stem moves.

On the other hand, a butterfly valve generates angular forces within itself. The forces or torques induced by pressure drop within a butterfly valve are not simply the product of an area times a pressure drop, but are greatly affected by the ability of the butterfly valve to behave as a lifting body.

The control valve body assembly is selected by the control or instrumentation engineer, based on the variety of service conditions under which the control valve must operate and compared with the wide variety of control valve body types available. In the same fashion, the control engineer has at his disposal, many actuators, among which there is a great deal of diversity, with each type having some particular advantage for application in a particular condition.

This chapter discusses the general types of actuators currently in widest use in the operation of control valves. These actuators may be classified into six general types:

1. Pneumatically operated, diaphragm actuators;
2. Piston (cylinder) actuators;
3. Electro-hydraulic actuators;
4. High-performance servo actuator;
5. Electro-mechanical actuators; and
6. Manual handwheel actuators.

In actual practice today, it is quite probable that more than 90% of all control valve actuators in use are the pneumatically operated piston or diaphragm type; therefore, this chapter deals most extensively with them.

SELECTION PARAMETERS

Before considering any actuators, it is well to consider the parameters under which an actuator is to be selected. It is perhaps even more important to consider the features of the actuators to be placed in a control loop.

In any control loop, we have of course, the process which is being controlled and which is capable of being, within the measured variable, in error. To that extent, the first component of a control loop is, therefore, the error sensing device; that is, the instrument which measures the controlled variable and compares it to an acceptable value. If an error between the measured variable and the acceptable value is determined, the instrument automatically delivers an error-correcting signal to the control valve.

This error-correcting signal can take many forms. It can be a low-power electric signal, a hydraulic signal, or a pneumatic signal. These error-correcting signals may be of sufficient magnitude to be used within the control valve as a source of power to control the process, particularly if they are hydraulic or pneumatic signals. The error correcting signal (which may also be power) is then transmitted to the final control element, the device which causes an energy conversion to occur within the system by means of pressure loss within the control valve. It is the actuator that moves the control valve to that position which will change the energy input to the process so that it may return to an acceptable value, as determined by the error measuring device.

The actuator is, therefore, that part of the final control element which, in response to the error correcting signal from the error sensing device, moves the control valve. It moves the control valve either in a linear direction as in a globe valve, or in a rotary direction as in butterfly and ball valves.

Among other things, the actuator has to be compatible with the error-correcting signal. If, for example, the error-correcting signal from the instrument is hydraulic, the actuator must be compatible with the hydraulic fluid.

The actuator must be capable of generating enough force from the error-correcting signal to overcome the maximum forces that may be generated by the pressure drop phenomena within the valve body, whether the forces are the linear pressure times area relationship, as in a globe valve, or are varying forces such as the torque in a butterfly valve.

Lastly, the actuator must be compatible with the environment in which it will operate. This environmental compatibility may, for example, require explosion-proof housings on electric or electro-hydraulic actuators in service in oil refineries or chemical plants where the hazard of electrically caused fires or explosions is high. The environmental compatibility may require preparation by special corrosion-resistant or acid-resistant paint, particularly where the environmental conditions might expose the actuator to acid vapors, or other corrosive agents.

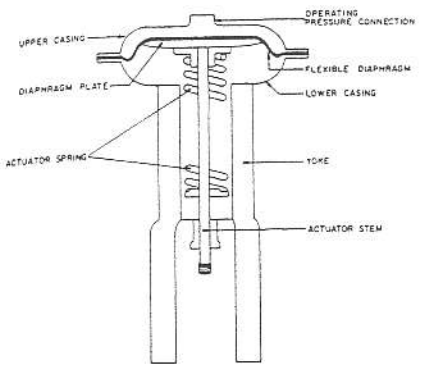


Figure 1. A DIRECT-ACTING DIAPHRAGM ACTUATOR.

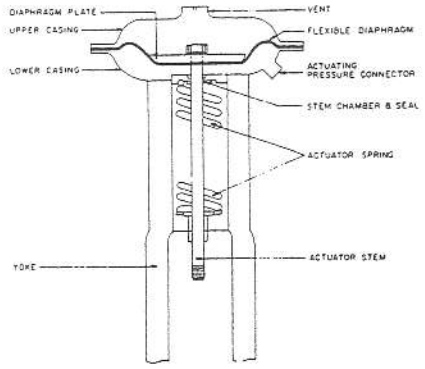


Figure 2. A REVERSE-ACTING DIAPHRAGM ACTUATOR.

TYPES OF ACTUATORS

Diaphragm Actuators

Diaphragm actuators usually take the form of a flexible diaphragm, placed between two stampings or cast "casings," and at least one section of the two resultant chambers is made pressure tight. A diaphragm actuator generally has a "range" spring opposing the force generated within the actuator. The control system air signal is connected to the pressure tight chamber, and an increase or decrease in the air pressure results in a force which is used to overcome the pressure-drop forces within the valve body, the hysteresis forces within a control valve body (stuffing box), and the force of the actuator's range spring.

A diaphragm actuator, in which the pressure-tight chamber is above the diaphragm, and in which an increasing pressure within that chamber will result in a downward motion, is generally described as being a direct-acting diaphragm actuator.

By using a seal around the actuator stem, the lower chamber can be made pressure-tight and the control system air signal may be connected to it. By venting the upper chamber to atmosphere and by changing the spring mounting arrangement so that upward stem motion will compress it, an increase in signal pressure in the lower chamber will result in an upward motion of the stem. This type of diaphragm actuator is ordinarily called a reverse-acting actuator. Cross sections of each type of diaphragm actuator, together with their essential nomenclature are shown in Figures 1 and 2.

Diaphragm actuators, whether direct or reverse acting, are not necessarily directly related to valve size in terms of pipeline inches. Each diaphragm actuator has to be calculated to make sure that it is powerful enough to overcome the forces within the valve body assembly during service, because the sole purpose of the actuator is to move the valve in response to the error-correcting signal. All major manufacturers of control valves catalog a fairly wide range of actuator sizes and generally specify them in relation to the square inch area of the diaphragm; however, it is quite possible that a very large diaphragm actuator may be required for a small valve, if the service conditions were such that a high pressure drop was being induced across a small line-size valve. Conversely, in a large valve with low pressure drop, it is entirely possible that a small actuator can be used.

Consider the sketch of a single seated direct acting valve body assembly, shown schematically in Figure 3. Assuming an inlet pressure (P_1) of 40 psig, an outlet pressure (P_2) of 10 psig, and a port area of 12.56 sq. inches (nominally a 4" valve), the force generated along the axis of the valve stem, tending to open the valve, would be P_1-P_2 x port area (30 psi x 12.56 in^2), or 376.8 pounds.

We would expect that this valve would

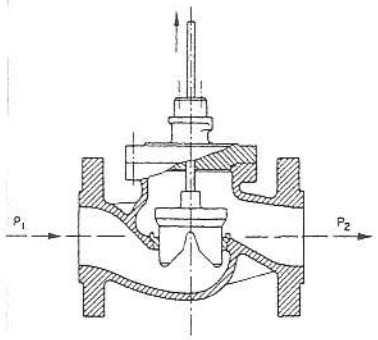
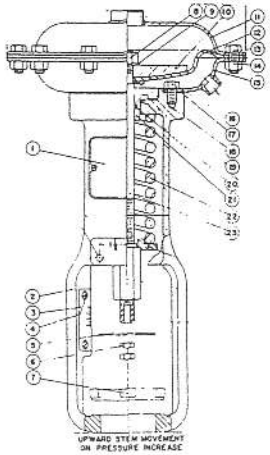


Figure 3. PLUG FORCE DEVELOPMENT.

begin to move downward at the 3 psig signal; assuming that the instrument signal is 3-15 psig, which is a conventional signal from a controlling instrument, and assuming further a direct acting response, (that is, increased pressure on the actuator results in downward motion of the actuator), air-to-close, and proportional service. We could, therefore, assuming a constant stem force upward over the full length travel of the valve, divide the 3 psig instrument signal into the total force of 1876.8 pounds, and determine that an actuator having 125 sq. inches in area would be sufficient to begin to move the valve. However, we would also then have to consider that at the conclusion of the stroke of the valve, we would have 15 psig (instrument pressure) x 125 sq. inches, or 1875 lb. force on the valve stem. Therefore, we have an excess force at the seat, to the extent of 1875 pounds, less 376.8 pounds, or a net surplus of force on the valve seat of 1499 pounds. It will therefore, be necessary for us to absorb this additional or surplus force by the use of the range spring, if we're fortunate enough to find a manufacturer who has a 125 square inch diaphragm. Assuming that the travel of the valve is one inch, we would have to put into the valve, a spring that would have a compression rate of 1499 pounds per inch. The additional pressure that may be required for seating force has not been accounted for here.

It should be noted that the above is really an ideal situation, assuming that a diaphragm of 125 square inches area could be located and that spring rates could be chosen to fit a given situation. In actual practice, this is not usually the case and most often, some compromise has to be made with spring range and diaphragm area in order to come as close as possible to the instrument signal and power output. Most manufacturers of control valves carry a sufficiently wide range of diaphragm actuators and springs so the normal instrument signal and power ranges (3 to 15 psig, 3 to 27 psig, and 6 to 30 psig) can be accommodated by 1/2 psig within the given range.

In concluding the discussion of diaphragm actuators, reference is made to Figure 4 which shows the reverse-acting actuator (a direct acting actuator would leave the same components) of one manufacturer, with all of the parts called out. It will be noted that there are really only two parts that can be greatly affected by ambient conditions. These two parts are



NO	PART NAME
1	NAMEPLATE
2	YOKE
3	SCALE, INDICATOR
4	SCREW, IND. SCALE
5	INDICATOR
6	LOCKNUT, STEM
7	LOCKNUT, YOKE
8	LOCKNUT
9	LOCKWASHER
10	DIAPHRAGM PLATE
11	ASS'Y, UPPER CASING
12	DIAPHRAGM
13	BOLT, CLAMPING
14	NUT, CLAMPING
15	ASS'Y, LOWER CASING
16	SCREW, CAP, MOUNTING
17	GASKET
18	WASHER, DIAPHRAGM
19	O-RING, BUSH/YOKE
20	O-RING, BUSH/STEM
21	BUSHING, O-RING
22	SPRING
23	STEM

Figure 4. DIAPHRAGM ACTUATOR ASSEMBLIES, COMPONENTS, AND NOMENCLATURE.

the diaphragm in either the direct or reverse acting motor and the stem seal in the reverse acting motor. Most valve manufacturers use neoprene diaphragms reinforced with fiberglass, cotton, or nylon for normal ambient conditions. If the ambient conditions exceed 200° F, higher temperature materials such as silicones, Viton®, or polyacrylics with Dacron or fiberglass fabric may be used. Occasionally a hydraulic fluid or other source of pressure may be used in the diaphragm actuator; it may be necessary to either back or overlay a diaphragm with a thin flexible piece of some fluid resistant material such as teflon or Kel-F®. With respect to the material used for the stem seals (which are ordinarily O rings) or the diaphragms, most major control valve manufacturers can offer special solutions to given ambient or corrosion problems that may exist.

Diaphragm actuators are, as we have indicated, the most widely used type in proportional control today. The chief advantages are a) their adaptability to a wide variety of valve sizes and circumstances, because of the wide range of sizes available through most manufacturers; and b) the diaphragm actuator is still the least expensive proportioning actuator available on the market.

Despite its advantages of wide adaptability and low cost, however, there are a number of instances in which diaphragm actuators cannot be applied satisfactorily. One of the inherent disadvantages of a diaphragm actuator is the fact that it must use large diaphragm areas, which in turn require large casings. As these casings become larger, the pressure which they are able to contain effectively, becomes less. One manufacturer, for example, has a 300 square inch actuator, but the maximum pressure that can be withstood within the diaphragm chamber itself is 30 psig. Although this is sufficient to generate a 9,000-pound force, the outside diameter of the diaphragm casing is approximately 27 inches; therefore, where small valves with large unbalanced or generated forces are involved, the entire assembly tends to become top heavy, imposing other stress problems on the valve and operator connection.

Piston Actuators

Another type of pneumatic or hydraulic actuator, commonly available from most valve manufacturers, is the piston actuator (also called cylinder actuator). Although the piston actuator for proportional control generally requires the use of valve positioners, which are described in Chapter 9, these cylinder actuators do offer some advantages that are not available in diaphragm actuators.

The cylinder is usually either cast or wrought iron designed to withstand substantially higher pressures than the cast or stamped casings in the diaphragm actuator. Because of its higher pressure rating; therefore, it is capable of delivering more force for smaller diameters. The cylinder actuator can be a very effective means of coping with high pressure drops, particularly when used in conjunction with a valve positioner which is capable of delivering pressures to the actuator, often up to the limit of the plant air rated pressure.

Cylinder actuators may be spring opposed although, in general, the cylinder actuators are usually used in proportioning or positioning control valves in conjunction with a valve positioner, by use of air pressure alone.

Figure 5 shows a cylinder actuator with a valve positioner, and air-spring mounted. The instrument signal is fed into the valve positioner at the top of the cylinder and through the feedback spring that is placed below the valve positioner. By the pressure on the piston, the positioner correlates the valve position with the instrument signal and begins to either load or unload air to the top of the piston to move the valve to the required position.

Beneath the piston, a constant air pressure (called air spring) is maintained by an air regulator. If the piston is pushed down, the airset relieves the pressure; if it is required that the piston move upward, the airset increases its pressure to maintain a constant pressure below the piston.

Figure 5 also shows the same type of cylinder which is hooked up so that, instead of being direct-acting (that is, an increasing instrument signal results in the downward motion of the piston), an increase in instrument signal will result in an upward motion of the operator stem by use of a reverse acting positioner. This particular hook-up also demonstrates another feature; that it is possible for the valve stem to go down in the event of air failure. If the air fails, a check valve in the supply line closes, the three-way valve opens and permits air pressure from an accumulator to go directly into the cylinder chamber, pushing it down. In the meantime, air pressure that may have remained below the chamber from the air spring regulator, will have been bled out by the downward motion of the piston.

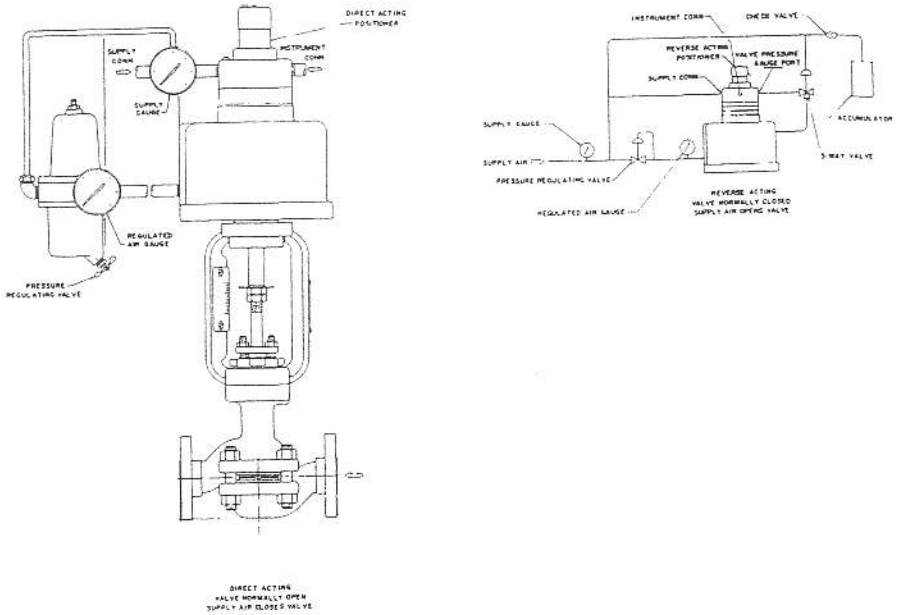
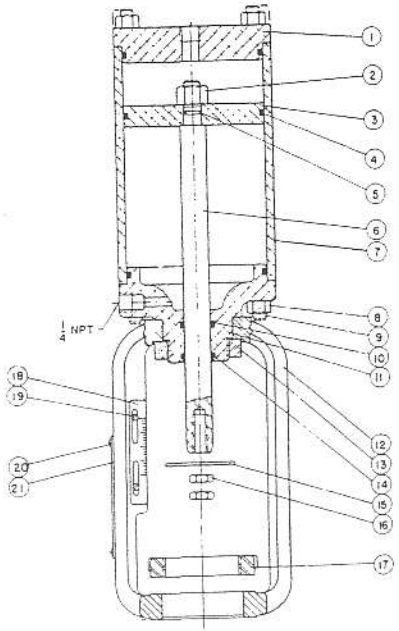


Figure 5. A DOME ACTUATOR CONTROL VALVE.



NO.	PART NAME
1	CLOSURE, BLIND END
2	NUT
3	PISTON
4	"O" RING, CLOSURE/PISTON HSG.
5	"O" RING, ROD TO PISTON
6	ROD, PISTON
7	HOUSING
8	NUT, TIE ROD
9	ROD, TIE
10	"O" RING, CLOSURE TO ROD
11	CLOSURE & INTEGRAL BUSHING
12	YOKE
13	RING, LOCK
14	RING, WIPER
15	INDICATOR, TRAVEL
16	NUT, LOWER STEM JAM
17	NUT, YOKE LOCK
18	SCALE, TRAVEL
19	SCREW, SCALE MTG.
20	SCREW, NAMEPLATE MTG.
21	NAMEPLATE

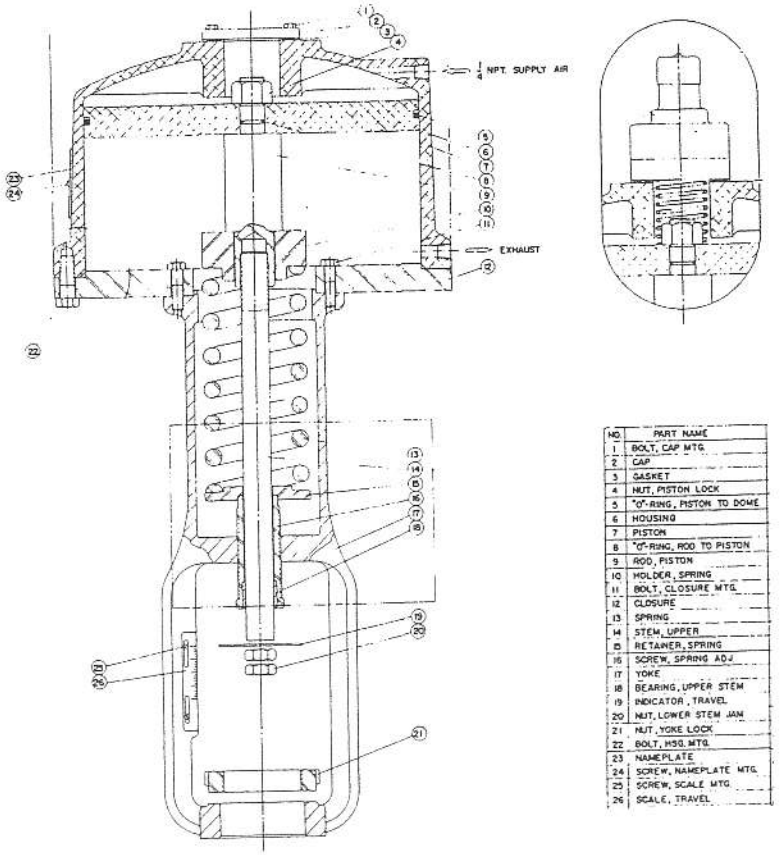
Figure 6. A DOUBLE-ACTING CYLINDER ACTUATOR.

Figure 6 shows a double-acting cylinder, without any accessory instrumentation or control devices. This actuator, which is intended essentially for an on-off service, can supply very swift open-shut action when used with a three-way solenoid air pilot valve and is useful in batching operations or cyclic work. This actuator can be made from either tubing or centrifugal castings and can be used with extremely high operating pressures for high thrust service.

Figure 7 shows a cylinder actuator used on a conventional diaphragm motor yoke with a spring opposing its motion. It can be used in the same manner as a conventional diaphragm motor, but can offer longer strokes of travel for a given yoke and spring combination. It can also be used with a positioner, as shown in the inset, and does not require the addition of the "air spring" and the accompanying piping.

Electro-hydraulic Actuators

There has been an increasing acceptance of electric controlling instruments in the instrumentation and controls industry over the past several years. These instruments, instead of producing a pneumatic error-correcting signal, yield a variety of low-power electric signals which can be either transduced or transposed at a pneumatic control valve in the field, as described in Chapter 9; or when combined with an electro-hydraulic valve actuator, can make a single control loop without any addition-



NO.	PART NAME
1	BOLT, CAP MTG.
2	CAP
3	GASKET
4	NUT, PISTON LOCK
5	"O"-RING, PISTON TO DOME
6	HOUSING
7	PISTON
8	"O"-RING, ROD TO PISTON
9	ROD, PISTON
10	MOLDER, SPRING
11	BOLT, CLOSURE MTG.
12	CLOSURE
13	SPRING
14	STEM, UPPER
15	RETAINER, SPRING
16	SCREW, SPRING ADJ.
17	YOKE
18	BEARING, UPPER STEM
19	INDICATOR, TRAVEL
20	NUT, LOWER STEM JAM
21	NUT, YOKE LOCK
22	BOLT, HSG. MTG.
23	NAMEPLATE
24	SCREW, NAMEPLATE MTG.
25	SCREW, SCALE MTG.
26	SCALE, TRAVEL

Figure 7. A DIRECT ACTING CYLINDER ACTUATOR ASSEMBLY.

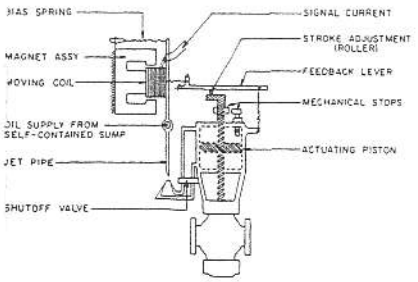


Figure 8. AN ELECTRO-HYDRAULIC VALVE ACTUATOR.

al requirement for a pneumatic power source. They do, however, require a separate electric power source.

These low power electrical signals are generally 24 to 65 volts d.c., and are of the order of 1 to 5, 4 to 20, or 10 to 50 milliamperes.

The current signal in one representative design, as illustrated by Figure 8, is fed into a moving coil. As the signal current varies, the coil moves correspondingly within the field of the permanent magnet. The coil in this particular design is also attached to a pivoted nozzle, through which a high pressure hydraulic oil flows. As the signal varies the position of the moving coil through the pivot, the nozzle injects fluid at a high pressure into one or the other of two cylinder pressure pickups. which either increases or decreases the hydraulic pressure on either side of the piston, causing the valve to move. The forces acting on the nozzle are balanced when the valve reaches a new position; therefore, the nozzle re-centers between the cylinder pressure pickups and there is no further valve movement.

Other electro-hydraulic actuators on the market are similar to the one described here, except that they differ in the way in which the hydraulic pressure is picked up by the cylinders. At least one other manufacturer uses a conventional flapper nozzle design of the coil to change the pressures on either side of the piston.

Electro-hydraulic actuators, in general, offer the advantage that they can be placed remotely from an instrument, where there may be no other auxiliary pressures, such as a pneumatic pressure, to operate a valve. They have not been widely received in the chemical and petroleum processing plants, probably because they continue to be expensive in relation to a diaphragm actuated control valve with a transducer, and because the essence of the design requires a constant source of pressure, which means a constant use of electric power to pump the hydraulic power fluid. Further, their operating speeds are sometimes lower than can be obtained with a diaphragm actuator, and even at this writing, their maximum stem thrusts are somewhat lower than can be obtained with large diaphragm actuators or high pressure cylinder actuators.

High Performance Servo-Actuators

A variation in the electro-hydraulic actuator has been developed by a number of companies, for service where extremely high pressures and pressure drops within a valve are encountered and which may be accompanied by high shut-off or opening speed requirements, extremely swift valve stem movement requirements, or extremely high frequency response characteristics.

Figure 9 shows the essentials of the operation of this high performance type servo-actuator. The valve stem motion, through a potentiometer or linear differential transformer, is compared with the instrument signal within a servo amplifier. A signal is then sent to a high performance servo-valve which either loads or unloads high-pressure hydraulic fluid to one side or the other of a high pressure cylinder actuator that is directly affixed to the valve stem. The excess hydraulic fluid is returned to a reservoir.

Ordinarily, these high performance servo-operators use hydraulic operating pressures that range from 1000-3000 psig. With these pressure and flow rates, extremely high valve stem speed can be obtained, extremely fast response to changing instrument signal conditions can be obtained,

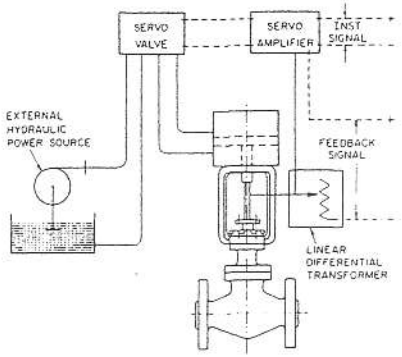


Figure 9. A HIGH PERFORMANCE SERVO-ACTUATOR.

and extremely high pressures or pressure drops within the main valve can be accommodated.

Their chief disadvantage is that they do require a very powerful external hydraulic source. Of course, this source can be used to supply a number of valves; however, their operating costs and installation costs are such that they have found a limited market, chiefly in the aerospace field.

Electro-mechanical Actuator

Another type of actuator which has been developed for proportional control in recent years is the electro-mechanical valve actuator. One of the principle disadvantages of the electro-hydraulic actuator, which is the constantly running motor and pumps, is not present in the electro-mechanical design. An electro-mechanical valve actuator is composed of a motorized gear train and screw assembly which drives the valve stem, an electronic positioner, and an electronic feedback device. A typical example is shown schematically in Figure 10. The varying input signal, whose magnitude corresponds to the required position of the inner valve stem, is fed into

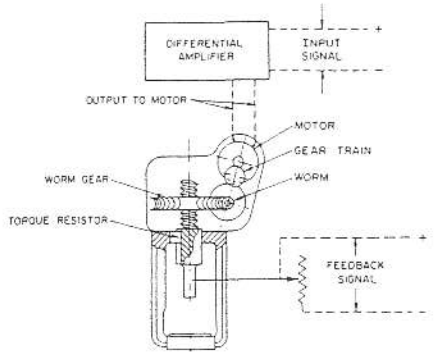


Figure 10. AN ELECTRO-MECHANICAL VALVE ACTUATOR.

the positioner (usually a differential amplifier) and produces a voltage to actuate the motorized gear train and screw. The resultant movement of the stem and the take-off attached to it and to a potentiometer or linear differential transformer, produces a voltage that increases with

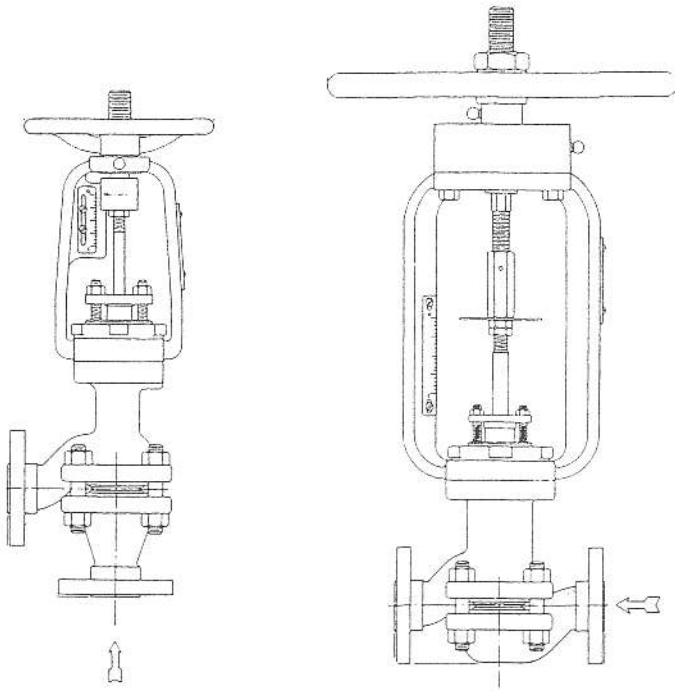


Figure 11. A HANDWHEEL CONTROL VALVE.

stroke, and is sent back into the positioner. When the input signal voltages and feedback voltages are equal, the output voltage to the motor goes to zero and the motor stops with the valve stem at the required position. Conversely, when the voltages are not equal, the motor is run in the direction to make them equal.

The electro-mechanical valve actuator has essentially the same advantages as the electro-hydraulic actuator with respect to field use. It is capable of being used over long distances with only inconsequential signal transmission delays. It is also immune from the pneumatic system problem of freeze-up in extremely cold ambient conditions. Electro-mechanical actuators are still, however, generally more expensive, although more efficient, than electro-hydraulic units.

Manual Actuators

Manual actuators are not, strictly speaking, proportioning actuators; however, they are briefly described in this section because they are used in control systems.

One type, as shown in Figure 11, is a straight manual actuator, with no power supplied. This really makes the control valve body assembly nothing more than a hand valve. There are occasions, however, when flows must be passed through valves precisely, even though not in conjunction with an automatic control system. Control valve body assemblies with manual actuators can serve this purpose where the flow characteristic is known. These valves can have their position read directly from the valve travel indicator, or can be equipped with accurate auxiliary indicators for even more precise work.

Two other types of handwheel operators are the top-mounted handwheel actuator, shown in Figure 12, and the side-mounted handwheel operator shown in Figure 13. Both of these operators are actually designed to override the control signal that is positioning the valve and to give operating personnel the option to override the control system if warranted. They can also be used to facilitate the start-up of a control system, for example, to preposition a valve to a given flow. They can also be used as devices to shut off the valve, eliminating the need in some systems for costly bypass valve arrangements.

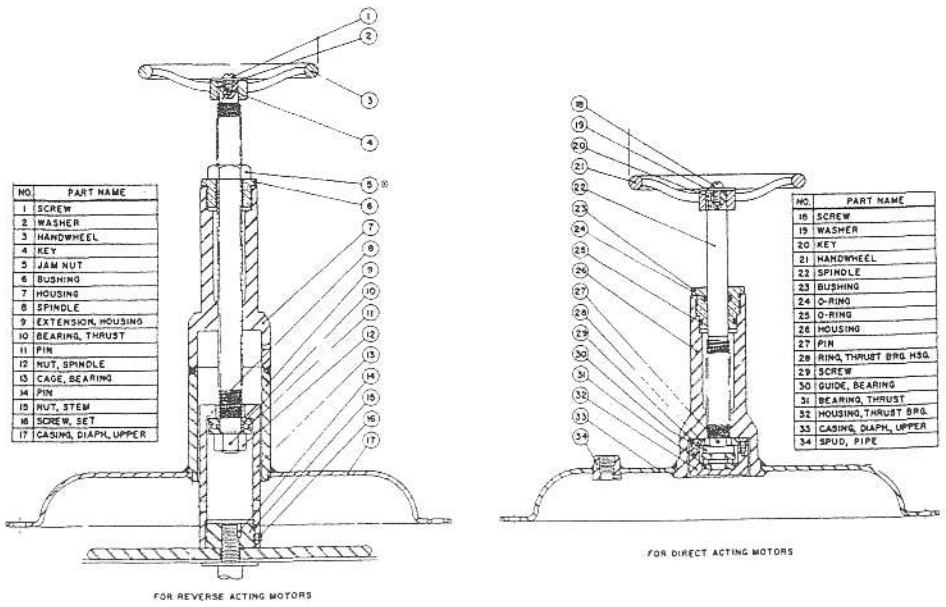


Figure 12. A TOP-MOUNTED HANDWHEEL OPERATOR.

NO.	NAME
1	NUT, HEX.
2	LOCKWASHER
3	ARM, ROCKER
4	TAKE OFF, VALVE STEM
5	SPACER, ARM
6	BOLT, HEX. HD.
7	SCREW, MACH. FL. HD.
8	NUT, HEX.
9	PLATE, YOKE ADAPTOR
10	BOLT, HEX. HD.
11	U-BOLT
12	SCREW, MACH. R. H.
13	WASHER
14	SCALE, TRAVEL
15	CLIP, NEUTRAL POSITION
16	SCREW, SELF TAP.
17	INDICATOR, NEUTRAL POSITION
18	BEARING
19	HOUSING, HANDJACK
20	COLLAR, FORWARD THRUST
21	BLOCK, TRAVEL
22	COVER PLATE
23	SCREW, SET, ALLEN HD.
24	COLLAR, AFTER THRUST
25	SCREW, SELF TAP.
26	SHAFT, HANDWHEEL
27	SCREW, SET, ALLEN HD.
28	BUSHING, BEARING
29	DETENT
30	SPRING
31	HANDWHEEL
32	KEY, WOODRUFF
33	SHAFT, ROCKER ARM
34	RING, SNAP

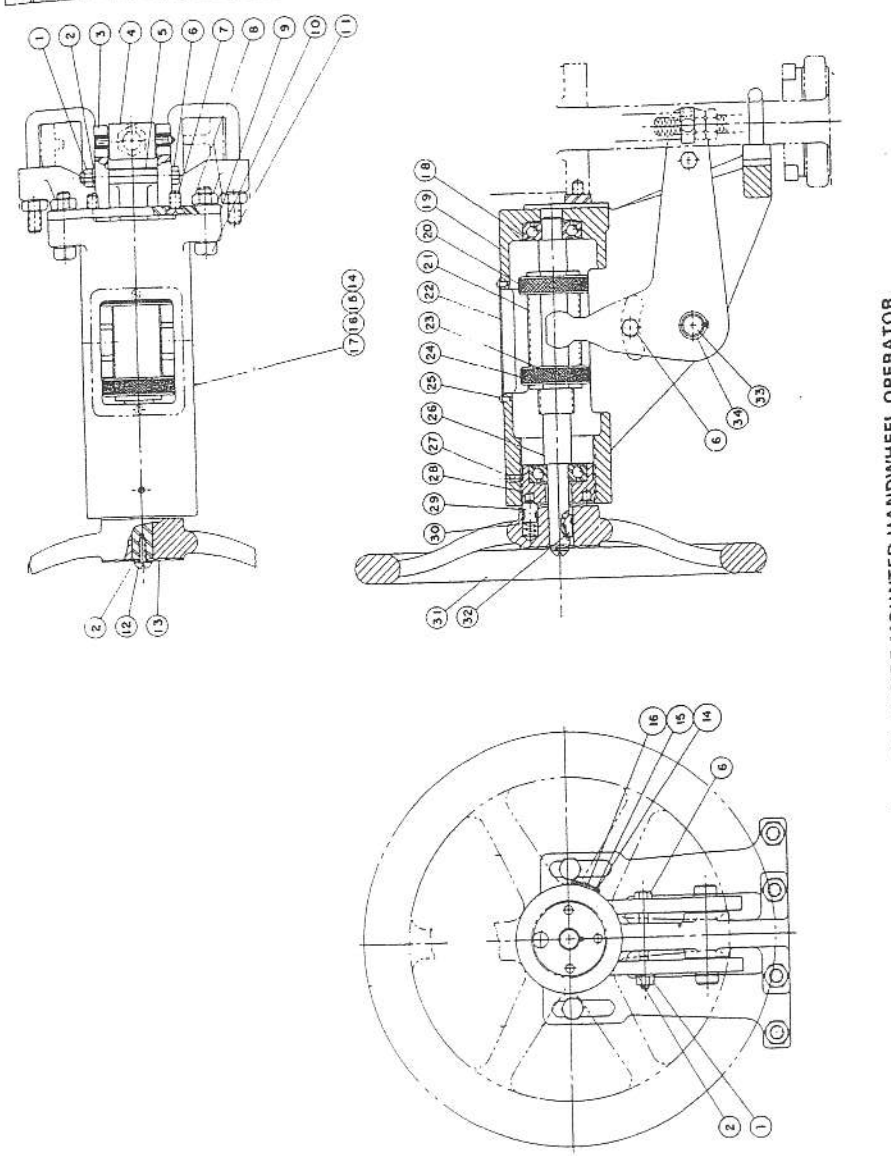


Figure 13. A SIDE-MOUNTED HANDWHEEL OPERATOR.

PART 2
SUPPLEMENTAL DATA
(J. L. Lyons)

FAIL SAFE ACTUATORS

In many types of piping systems it is essential for reasons such as safety, ease of repair, avoidance of fluid loss, reduced down time, etc. that a valve fail in a "safe" mode. Generally, this requires that the valve actuator be fail safe.

For example, in many process systems hazardous chemicals are flowing through the system and must be contained in event of valve or system malfunctions. In many such cases the valve must fail in the closed position. In other cases the fluids must be rerouted to prevent possible dangerous chemical reactions, contamination of the system, etc.

There are many types and styles of fail safe devices most of which fall into two categories, external design and internal design.

External

The external design actuator is assisted by an outside source such as a three way solenoid, servo valve, a manual override, or a combination electropneumatic, electrohydraulic or electromechanical override. As an example, consider a valve which is required to fail in the closed position. One possible solution is shown in Figures 14-A and 14-B where a three way solenoid is attached to the actuator. With zero current flow or a low predetermined current, the solenoid will close off the supply pressure to the actuator's open position, subsequently forcing the control or main valve to the closed position. This same procedure can be applied to double acting diaphragm or piston actuators.

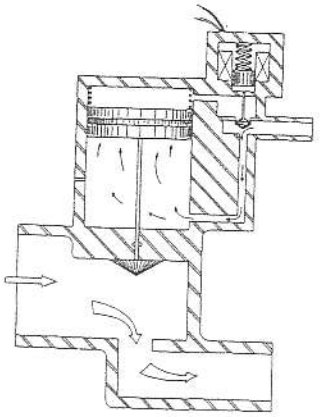
Another type of external design is the self-contained pressure bottle or accumulator type system which provides available pressure even though the main system malfunctions.

Internal

The internal type of fail safe actuator may be inherent in the actuator design or the actuator may be modified internally. There are two main types of internal fail safe actuators, spring type and weight type (see Figures 16 and 22).When the pressure applied to the actuator is insufficient, the spring in a reverse acting actuator will return the valve to the closed position. In cases where the valve must fail in the open position a direct acting actuator may be used. (See Figures 1 and 2 of this chapter.

Why Use A Fail Safe Actuator?

1) A fail open actuator will prevent pressure build-up of a system or component due to a power failure to the actuator.



ure 14A. VALVE OPERATION UNDER NORMAL ONDITIONS WITH ELECTRICAL POWER ON.

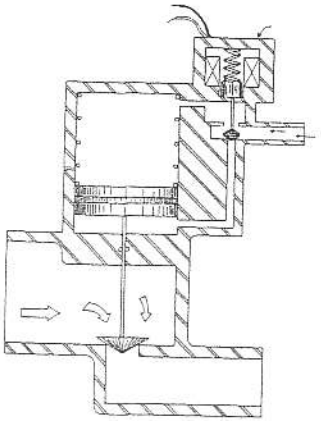


Figure 14B. VALVE OPERATION WITH ELECTRICAL OR PNEUMATIC FAILURE.

2) A fail open actuator will prevent pressure build-up of a system or component due to a power loss of a system pressure using a pilot operated valve or actuator of that system.
3) A fail close actuator will prevent pressure build-up of a system or component due to a power failure to the valve actuator.
4) A fail close actuator will close a valve due to a leak or rupture in the line.
5) A fail close actuator can be used to prevent fueling of a fire, such as in oil companies, chemical plants, etc. when actuator power failure is possible due to that fire.
6) A fail close actuator is used to prevent toxic gases or liquids from escaping because of actuator or system power failure.
7) A fail open actuator will allow toxic gases or liquids to vent in case of power failure.
8) A fail open actuator will allow coolant to circulate in a system or component such as in the heat exchanger of a nuclear reactor or in the event of a cycling or power failure of the actuator power source.

It may be noted that fail-safe actuators may also be used in conjunction with other systems when equipped with other signal or sensing devices for either fail open or close applications. The following is a list of some of these:

a) high or low liquid level relief.
b) run-away pump or motors.
c) over heating or cooling.
d) low or high pressure relief.
e) dangerous liquid or gas handling.
f) life support with liquids or gases such as water or oxygen.

TIME DELAY ACTUATORS

A time delay actuator is normally used with shutoff valves in on-off service. The time delay is usually caused by a variable orifice and is intended to help prevent water hammer, cavitation, or other similar phenomena. The orifice is generally adjustable to suit the system that it is installed in. Since timing relays in many cases are variable orifices, the solenoid usually incorporates a fixed orifice or restrictor such as shown in Figure 14-A. With the quick response of a 90 degree full open-close valve such as taper plug or ball valves, the actuator may need to be regulated for a slower response. In high pressure gas systems shock waves are propagated. Heat or stagnation pressure may cause serious or even catastrophic failures if the system is closed too rapidly.

ROTARY ACTUATORS

A rotary actuator is used to open and close valves with rotary closure members such as ball, plug, and butterfly valves. A linear motion of one element is converted into rotary motion of the valve element.

Rack and Pinion

This type of actuator uses a linear motion of the rack to rotate a pinion connected to the valve stem. The stroke or amount of opening of the valve is limited by the travel of the rack. This type of actuator is available in a wide variety of sizes and configurations such as single acting, double acting and spring return. An example is shown in Figure 15.

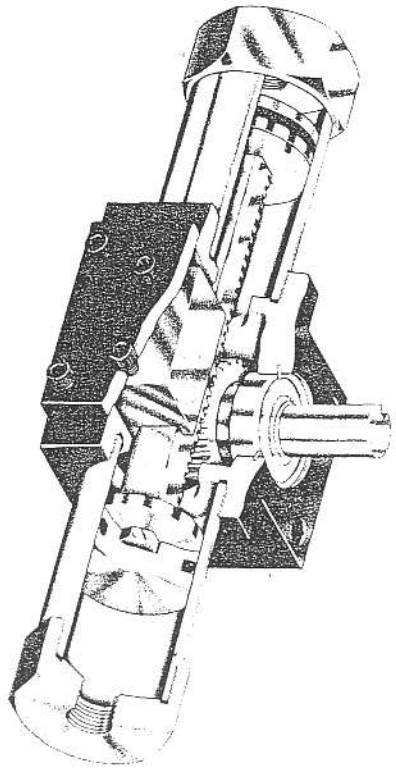


Figure 15. RACK AND PINION ACTUATOR.
Courtesy of Chemetron Corp. Fluid Controls Division

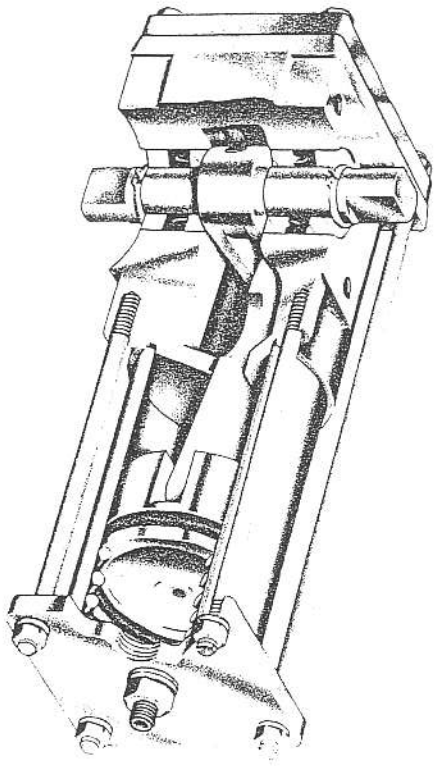


Figure 16. PISTON LEVER ACTUATOR WITH SPRING RETURN FOR FAIL SAFE OPERATION.
Courtesy Of Chemetron Corp. Fluid Controls Division

Piston-Lever Actuator

The movement of a piston (as illustrated in Figure16) is transmitted by means of a lever attached to the valve stem. This actuator is normally used for on-off applications and is usually restricted to 90 degree rotation. This, like the rack and pinion type, may be single or double acting, with spring return, and used with other devices such as time delays, solenoids, etc.

Diaphragm-Lever Actuator

It is similar to the piston-lever actuator but instead of a piston, the force is transmitted by a diaphragm and lever. This type of actuator (pictured in Figure 17) is usually used for control rather than on-off application. Positioners and fail safe devices along with other accessories may also be added.

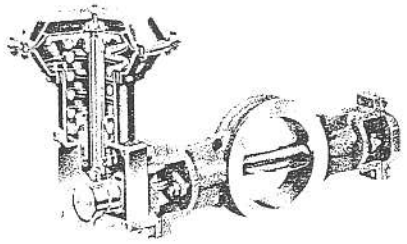


Figure 17. DIAPHRAGM-LEVER ACTUATOR FOR A BUTTERFLY VALVE.
Courtesy Of Keystone Valve Co.

Electro-Mechanical Actuator

This type of rotary valve actuator is used for a number of applications. It is economical and often used where pneumatic power is unavailable. Figures 18 and 19 are examples.

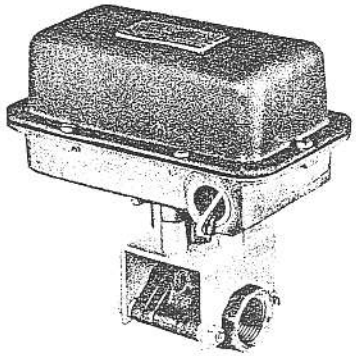


Figure 18. ELECTRO-MECHANICAL ACTUATOR.
Courtesy Of Chemetron Corp. Fluid Controls Division

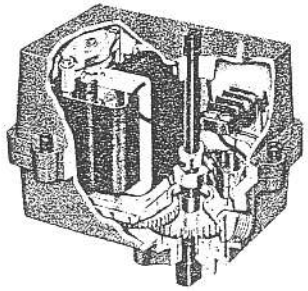


Figure 19. CUT-A-WAY VIEW OF ELECTRO-MECHANICAL ACTUATOR.
Courtesy Of Chemetron Corp. Fluid Controls Division

Another type of rotary valve used for 90 degree flow control which requires a piston or cylinder type actuator is the V-port ball valve as shown in Figure 23. As can be seen, the valve is equipped with a positioner and other devices commonly used in diaphragm actuators.

OTHER TYPES OF ACTUATORS

Wheel, Pulley, Crank and Key

These types of actuators are generally used for on-off service but may be used for flow control when systems require special service or additional regulation. As can be seen in figure 20, one valve type can have many combinations for manual actuation.

Rolling Diaphragm (Deep Convolution)

This type of actuator has become more popular recently (see Figure 21). The reason is that the longer stroke permits the travel to match that of piston types without the diaphragm working area being affected as with most other types of diaphragm actuators. The actuator also requires less pressure and subsequently has lower hysteresis and break-away friction than a piston actuator. The primary disadvantage is its inability to work as a double acting member with use of only one diaphragm. The user of this type actuator should keep in mind the limitation of the diaphragm's operating parameters such as temperature and pressure when operating at the upper limits of the stroke unless special consideration is given to the diaphragm material.

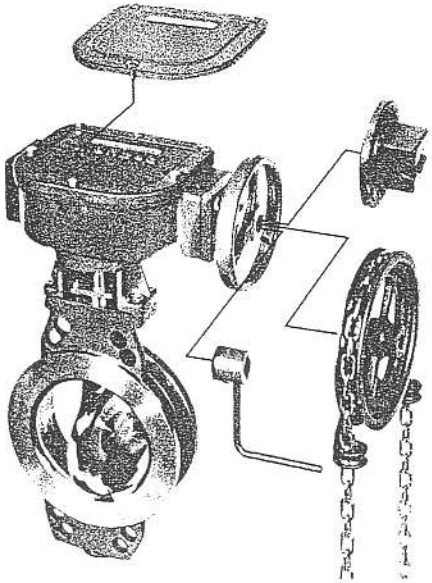


Figure 20. WHEEL, PULLEY, CRANK AND KEY ACTUATOR.
Courtesy Of DeZurik, A Unit Of General Signal.

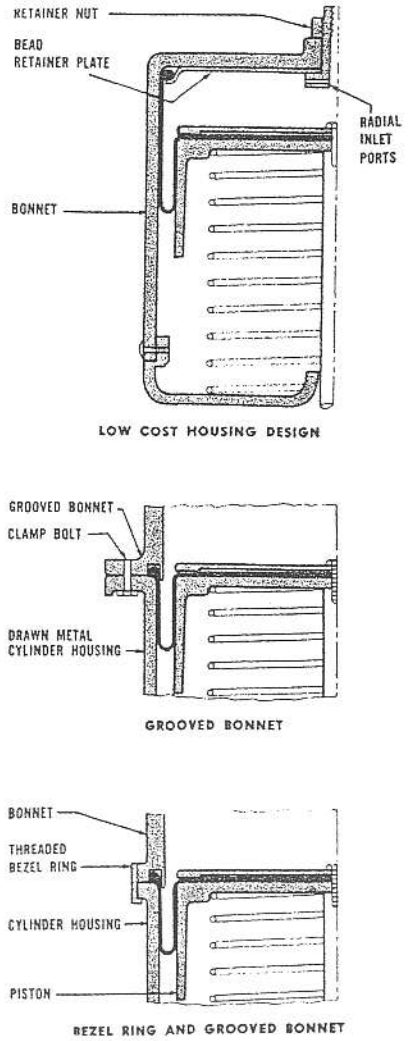


Figure 21. ROLLING DIAPHRAGM ACTUATOR.
Courtesy Of Bellofram Corp.

Weight or Springless Actuator

Many types of weight actuators are in use today. One type is shown in Figure 22. This device permits adjustment of the power output of the actuator by adjustment of the fulcum position on the rocker arm. Other types of weight actuators are designed with the weight attached to the lever arm located outside the valve body. The force is increased by the movement of the weight to the outer limit of the lever arm as shown in Figure 24.

SUMMARY

The actuator is an important member of the valve family. Designing or selecting the correct one can be a major task. In the past, the actuator was frequently poor in response, large in size, and expensive. Today, however, much as been done to improve the situation. Future actuators may be designed with a whole new outlook, utilizing space age technology such as fluidic design positioners, timing and sensing devices. The use of solid state and logic laser optics will enable actuators to perform with better response at lower cost with increased reliability. This is now becoming a reality.

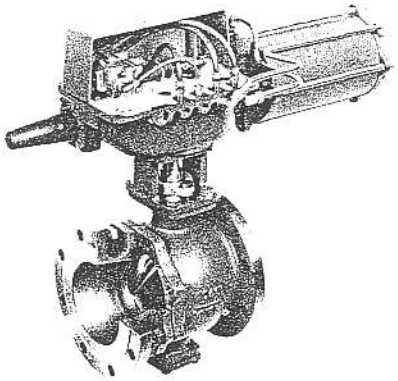


Figure 23. V-PORT BALL VALVE ACTUATOR.
Courtesy Of De Zurik; A Unit Of General Signal.

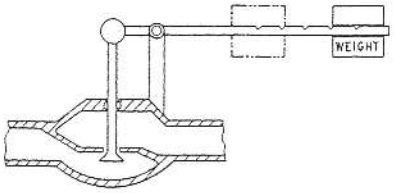
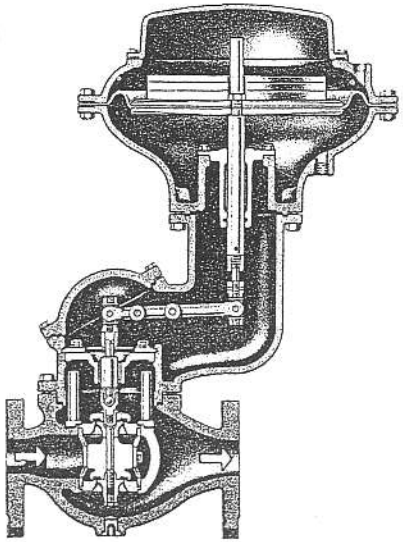


Figure 24. WEIGHT ACTUATOR.



WEIGHT OR SPRINGLESS ACTUATOR.
[illegible] Motor Controls, Inc.

CHAPTER 9

POSITIONERS AND OTHER DIAPHRAGM CONTROL VALVE ACCESSORIES

Edward L. Vollmer, III
Honeywell, Inc.
Fort Washington, Pennsylvania

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POSITIONERS

This section describes some of the most common types of pneumatic control valve accessories. Since the positioner is often considered to be the most important of them, it is covered first.

A pneumatic valve positioner is a device which precisely positions, by the use of air, the moving part (or parts) of a pneumatically operated valve in accordance with a pneumatic signal. A typical positioner is shown in Figure 1.

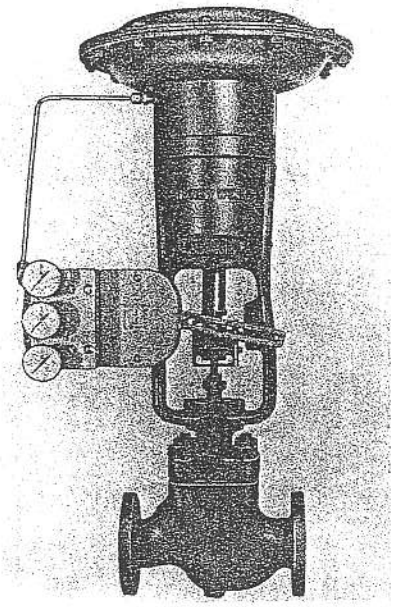

Figure 1. SIDE MOUNTED VALVE POSITIONER ON DIAPHRAGM CONTROL VALVE.

Courtesy Honeywell, Inc.

Principles of Operation

Figure 2 shows schematically a valve that is operated directly by a controller. A typical controller output varies from a minimum of 0 psig to a maximum of 20 psig. The most common valve spring operates from 3-15 psig. Fifteen psig is required to compress the valve spring. The air pressure above 15 psig is used to close the valve against the upstream pressure.

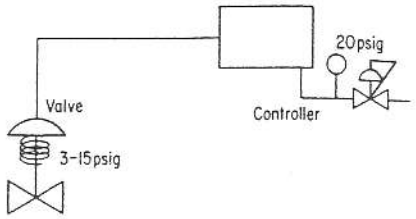

Figure 2. SINGLE-SEATED CONTROL VALVE WITHOUT POSITIONER, AIR-TO-CLOSE.

Figure 3 shows a valve with a positioner, in addition to the controller. The signal from the controller in this case goes to the positioner instead of directly to the control valve. The positioner compares the signal with the valve stem position. If the stem is incorrectly positioned, the positioner either adds or exhausts air from the valve actuator until the correct valve stem position is obtained.

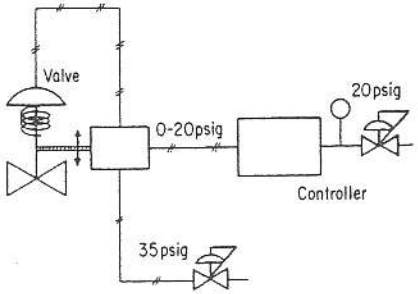

Figure 3. SINGLE SEATED CONTROL VALVE WITH POSITIONER, AIR-TO-CLOSE.

Examples of Construction

A schematic diagram of a pneumatic positioner is presented in Figure 4. The instrument control signal is applied to the signal diaphragm. An increasing signal will drive the diaphragm and pilot stem to the right. The pilot stem will then open the supply flapper admitting supply pressure into the valve chamber. The exhaust flapper remains closed when the pilot stem is deflected to the right. Since the valve chamber is connected to the valve actuator, whatever pressure exists in the valve chamber will also exist in the actuator.

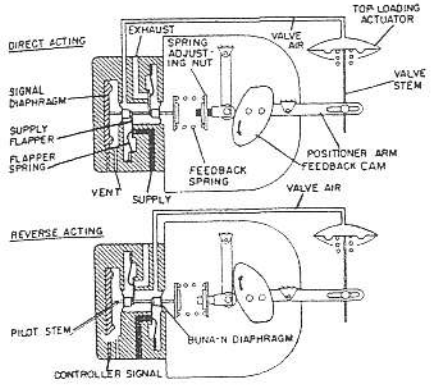

Figure 4. SCHEMATIC DIAGRAM OF PNEUMATIC VALVE POSITIONERS.

In the direct acting type, an increasing instrument signal increases the pressure in the valve chamber and in the valve actuator. This increased pressure on the actuator drives the valve stem downward and rotates the positioner arm clockwise. This rotation compresses the feedback spring through the cam. When the valve stem reaches the position called for by the controller, the compression on the feedback spring will give a force equal to the force caused by the controller signal acting on the signal diaphragm. When this "force balance" exists, the pilot stem moves back to the left and both flappers are closed.

If the controller signal is decreased, the force exerted by the signal diaphragm will also decrease and the force from the feedback spring will push the pilot stem to the left, opening the exhaust flapper. This causes a decrease in valve chamber pressure and allows the valve stem to move upward until a new force balance is established.

The positioner illustrated in Figure 5 employs the principle of force balance operation by spring loading the bellows unit (from the actual motion of the valve stem) to the same force as that exerted by the control-instrument air pressure.

Either a rise or a fall of the control-instrument air pressure will move the bellows and operate the air pilot. Supply air is either admitted to the diaphragm-actuator chamber or exhausted past the upper pilot stem, to the atmosphere. As the valve moves, the balance-spring load is changed. This action brings the bellows and pilot back to their original positions,

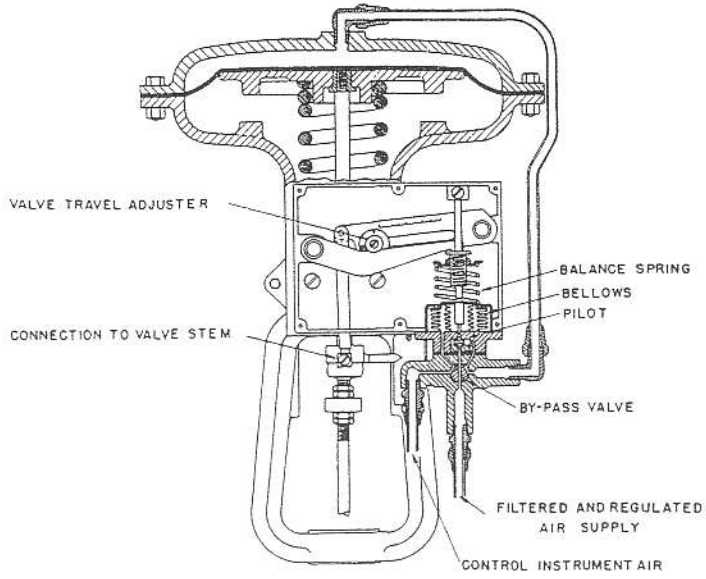

Figure 5. SIDE MOUNTED VALVE POSITIONER.

Courtesy Moore Products Company

thus restoring the balance between the spring loading and the control-instrument pressure.

Bellows characteristics do not affect the calibration, because the bellows remain in virtually the same position at all points of balance. The arrangement of the levers compensates for angularity errors. A linear relationship exists between the control-instrument pressure and the valve stem position.

The bellows and air pilot are a self-contained unit, without elements that would introduce friction and cause errors. Any friction in the lever bearings is on the valve side; therefore, it does not influence the performance of the positioner.

Applications

Some of the common applications of positioners are listed below, along with pertinent discussion. These applications should be tempered by the limitations given under the subheading entitled, "Limitations of Positioners."

To Overcome Friction

A positioning accuracy of 0.001 in. with a friction of 10 per cent to 15 per cent of full diaphragm actuator power is possible with some positioners.

To Increase the Shut-Off Rating of Single-Seated Valves

A common application is the use of a positioner, with a supply pressure of 35 psig, on a single seated valve. If a 3-15 psig spring is used in the valve, the first 15 psig from the positioner output compresses the valve spring. The remaining 20 psig (35 psi minus 15) is available to close the valve against the upstream line pressure. This is considerably more air pressure than is available from the controller; therefore, the close-off rating of the valve against line pressure is increased accordingly.

The supply pressure to the positioner is generally limited by the maximum allowable air pressure to the control valve actuator. A further gain in control valve shut-off rating can be obtained by reducing the initial compression in the control valve spring. For example, a 3 to 15 psig spring range can be adjusted to a 1 to 13 psig range. If 35 psig is the positioner supply pressure, then only 13 psig is needed to compress the valve spring and the remaining 22 psig is available to close the valve against the up-stream pressure. A further increase in valve shut-off rating can be obtained by using a 3-9 psig spring in the control valve. Here, the extra pressure available for closeoff is 26 psig (35 minus 9). As long as the positioner input spring is 3 to 15 psig and sufficient positioner supply pressure is available, the valve will operate precisely over a 3-15 psig range regardless of the control valve spring range.

Figure 6 shows an air-to-open valve with a 6 to 30 psig spring. Here the heavier spring and high initial compression provides greater close off capability than that which could be obtained with a 3-15 psi valve spring. The control valve's shut-off rating is limited by the initial compression in the valve spring, because the positioner can only remove the air pressure to the control valve actuator to close the valve.

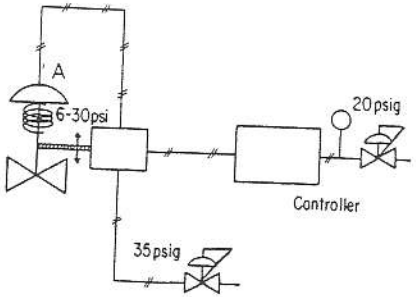

Figure 6. POSITIONER AND AIR-TO-OPEN VALVE WITH 6-30 PSIG SPRING.

To Increase Control Valve Speed of Response

If a control valve is operated directly by a pneumatic controller, the speed of operation of the valve depends upon: 1. the length and size of the connecting tubing; 2. The actuator volume; and 3. the capacity of the pilot relay in the controller.

If a positioner is used, the controller signal goes to the positioner instead of to the control valve actuator. Since the input to the positioner requires approximately one cubic inch or less of air, large quantities of air need not flow between the controller and the positioner. This, in itself, can increase the speed of valve response. Also, the pilot relay in the positioner generally has a larger supply port than the controller; therefore, the air from the positioner to the valve actuator moves faster and in greater quantity to achieve greater speed of valve response.

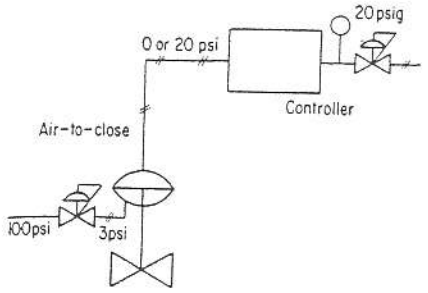

Figure 7. SPRINGLESS ACTUATOR CONTROL VALVE WITHOUT POSITIONER, FOR ON-OFF SERVICE.

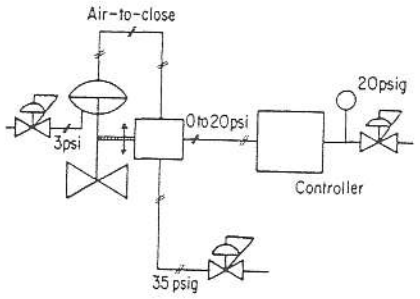

Figure 8. SPRINGLESS ACTUATOR CONTROL VALVE WITH POSITIONER, FOR THROTTLING SERVICE.

To Operate Springless Actuators

Figure 7 shows a controller operating a pneumatic control valve with a springless actuator. The springless actuator is the diaphragm type with the lower chamber supplied with a constant loading pressure of 3 psig air. This configuration is generally used only with two-position control since any air pressure above 3 psig on top of the diaphragm tends to close the valve, and any pressure less than 3 psig above the diaphragm permits it to open. Its chief advantage is that it provides greater shut-off forces to close the valve, since there is no spring to compress.

Figure 8 shows a pneumatic control valve with a springless actuator and a positioner. The controller is of the throttling type, with an output varying from approximately 0 to 20 psig. The positioner will apply the air pressure (up to 35 psig) necessary to correctly position the valve stem in accordance with the controller signal. The chief advantage of this arrangement is a higher control valve close-off rating against the upstream pressure, since there is no spring to compress. The chief disadvantage is a decrease in stability, due to the absence of a valve actuator spring. It also fails to provide reliable fail-safe performance.

Split Range Operation

It is sometimes desirable to operate a pneumatic control valve over only part of the controller output range. This can be accomplished by either changing or adjusting the input spring of the positioner. A common arrangement is to have one valve and positioner operate over 3 to 9 psig of the controller output, while another valve and positioner operates over 9 to 15 psig of the controller output. In the above example, it is only necessary to modify the positioner springs. The control valve springs may be standard 3 to 15 or 6 to 30 psig springs.

To Reverse Action

A positioner whose output air pressure increases as the input signal increases is called a direct-acting positioner. A positioner whose output air pressure decreases as the input signal increases is called a reverse-acting positioner. Positioners are available, whose action can easily be changed in the field.

The use of a reverse-acting positioner is not common and is not necessarily required with a reverse-acting control valve actuator. One application, however, is for heating and cooling service with two normally closed valves having a split-range operation.

To Change Control Valve Flow Characteristics

Most positioners are linear. That is, they change the valve stem position linearly with respect to controller output pressure. However, some positioners are provided with a means (generally a cam) to change this relationship to effect different control valve flow characteristics. Figure 9 shows this situation graphically. The combined results may not offer an improvement

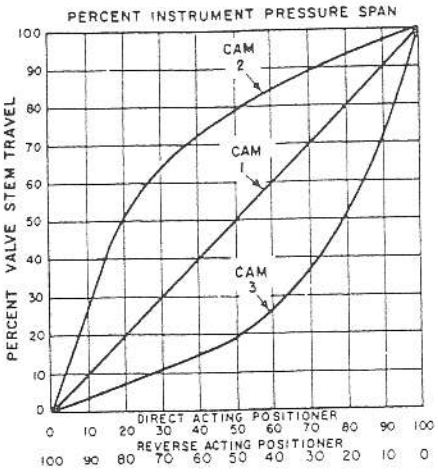

Figure 9. CAM CHARACTERISTICS.

in certain cases, because the valve time constant as well as the gain are affected by the cam shape.

Wide Proportional Band

When the valve must respond to very small changes in air pressure (less than 0.25 psi), a valve positioner is often recommended. This is the situation when the proportional band is wide.

Nature of Flow Medium

If the process fluid tends to gum or cake on valve parts and increase the friction between moving parts, a valve positioner is often helpful to provide the additional force required to overcome the friction.

High Pressure Flow Medium

On high pressure applications, the valve stem packing may tighten considerably resulting in higher friction on the valve stem which increases hysteresis and deadband. A valve positioner can sometimes be used to advantage under these circumstances. (See References 1 and 2)

Accessories For The Positioner

An integral pneumatic switch is sometimes supplied with a positioner and is used to bypass the positioner. The bypass may not be recommended where the valve will not operate without the positioner.

Pressure gages are often supplied to indicate the controller output pressure, positioner output pressure, and supply pressure to the positioner.

Limitations Of The Positioner

The above applications are traditional; however, recent research indicates that the use of a positioner may be detrimental to the quality of control in fast processes such as liquid pressure and flow processes. Where required (for example, higher air pressures to close the valve, or to speed the operation of the valve) pressure or volume boosters and relays may be used instead of positioners.

For slow processes, such as most thermal systems, liquid-level, and some large volume gas pressure processes, the positioner will improve the quality of control even though it appears not to be needed. (For further information see References 1 and 2).

Other Types of Positioners

Top-Mounted

The top-mounted valve positioner as shown in Figure 10, operates on the pneumatic force-balance principle. Control-instrument air pressure exerts an upward force on the larger (center) diaphragm of the positioner, because of the difference in the diaphragm areas. This force, plus the upward force of the range spring, is balanced by the downward force of the suppression spring. At balance, a small

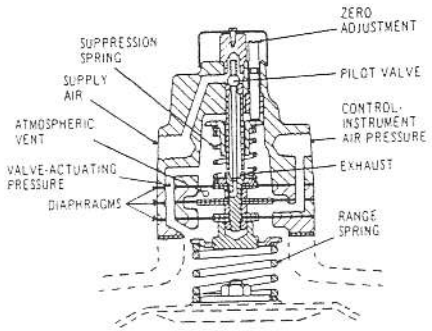

Figure 10. TOP MOUNTED VALVE POSITIONER.

Courtesy Moore Products Company

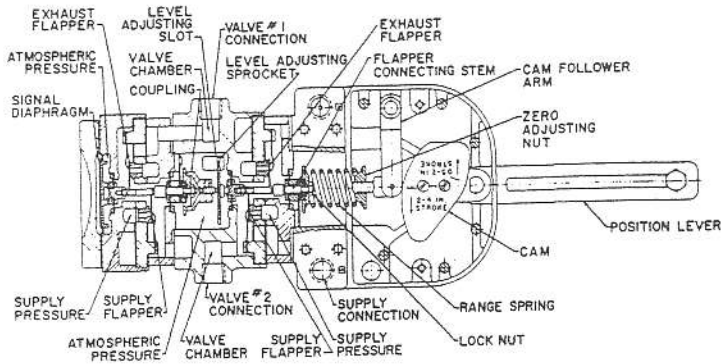

Figure 11. DOUBLE ACTING VALVE POSITIONER.

pilot flow (approximately 0.15 scfm) provides instant response to the slightest change in the control-instrument air pressure.

An increase in control-instrument air pressure temporarily raises the diaphragm assembly, closes the exhaust, and opens the pilot valve. The increase in valve pressure moves the motor diaphragm downward, reducing the force on the range spring. When the valve stem is in the position required by the control-instrument air pressure, the forces are again in balance.

Double-Acting Positioner

The double-acting pneumatic positioner is a force-balance instrument which positions a control valve stem in accordance with a pneumatic signal received from either a controller or a manual loading station. Figure 11 shows a direct-acting positioner set up for use on a piston actuator. With this configuration, an increasing signal will drive the piston stem downward.

In operation, the instrument signal is applied to the signal diaphragm. An increasing signal will drive the diaphragm and flapper connecting stem to the right. The flapper connecting stem then opens the supply flapper on "valve #2," admitting supply pressure into the upper chamber. The exhaust flapper on "valve #2" remains closed when the flapper connecting stem is deflected to the right. On "valve #1" the opposite action occurs, exhausting the lower chamber. An increasing instrument signal increases the pressure in the upper chamber of the piston and decreases pressure in the lower chamber. This pressure differential on the piston drives the valve stem downward and rotates the positioner lever clockwise. Clockwise rotation of the positioner lever compresses the range spring through the cam. When the valve stem reaches the position called for by the controller, the compression on the range spring creates a force equal to the force caused by the controller signal acting on the signal diaphragm. When this *force balance* exists, the flapper connecting stem moves back to the left and all flappers are closed.

If the controller signal is decreased, the force exerted by the signal diaphragm will also decrease and the force from the range spring will push the flapper connecting stem to the left, opening the exhaust flapper on "valve #1" and closing the supply flapper on "valve #2." This causes an upward force on the piston and allows the valve stem to move upward until a new force-balance is established.

Electropneumatic Positioner

The Electropneumatic Positioner, shown in Figure 12, is a force-balance device which positions a control valve in ac-

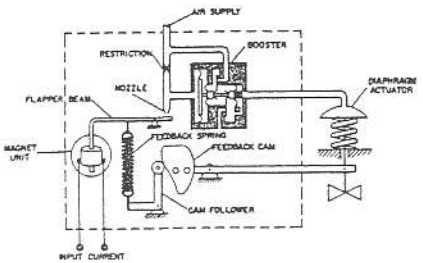

Figure 12. ELECTROPNEUMATIC VALVE POSITIONER.

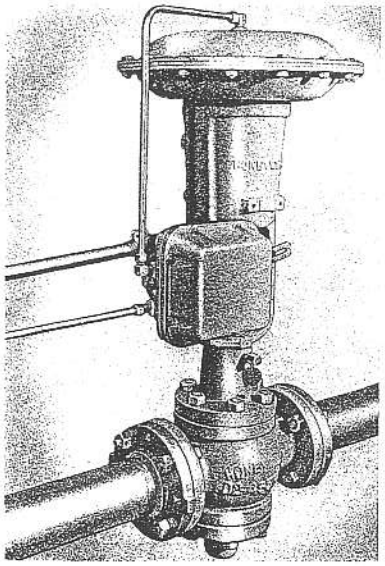

Figure 13. CONTROL VALVE WITH DIGITAL PNEUMATIC POSITIONER.

Courtesy Honeywell, Inc.

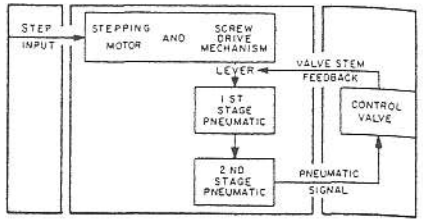

Figure 14. BLOCK DIAGRAM OF DIGITAL PNEUMATIC VALVE POSITIONER CONTROL LOOP.

cordance with the current signal from a controller. As shown in the figure, the positioner supplies air to the control valve until the force from the feedback spring balances the force of the electromagnetic coil.

Digital Positioner

The digital positioner is a single-acting, two stage, force balance instrument. It receives a digital electrical signal and positions a pneumatic control valve. Figure 13 is a photograph of a digital pneumatic positioner and Figure 14 is a block diagram of the digital positioner in a control loop.

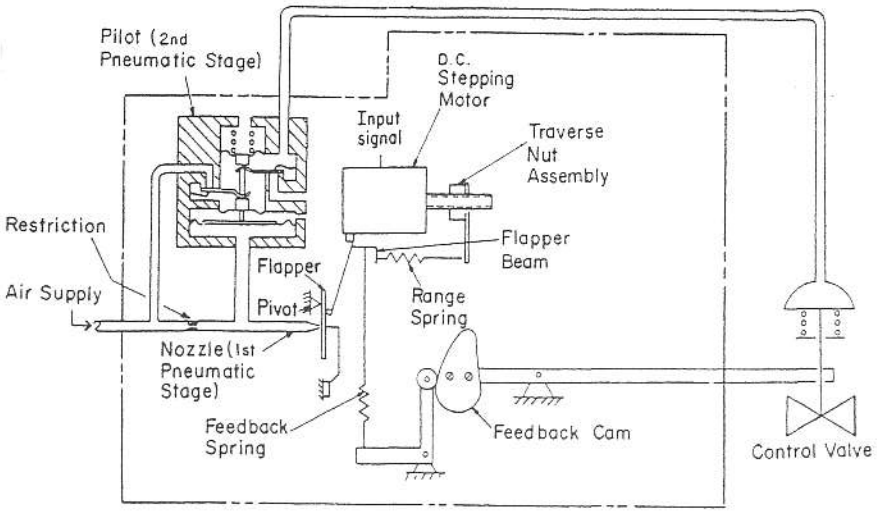

Figure 15. SCHEMATIC DIAGRAM, DIGITAL PNEUMATIC POSITIONER.

Figure 15 is a schematic illustration of a digital positioner. The input signal goes to a dc stepping motor which rotates in discrete steps. As the threaded motor shaft rotates, a traverse nut assembly travels on it varying the tension of a range spring. If this tension decreases, it causes a clockwise rotation of the flapper beam and a counter-clockwise rotation of the flapper allowing more air to flow out of the nozzle. This effect reduces the back pressure applied to the diaphragm of the second stage pilot, causing a reduction of output pressure applied to the control valve diaphragm. Reduced pressure on the control valve diaphragm will cause the valve stem to move up and the feedback cam to rotate counter-clockwise. Action of the feedback cam and feedback spring rotates the flapper beam in a counter-clockwise direction and the flapper in a clockwise direction. This action decreases the amount of air flowing out of the nozzle. When the moment on the flapper beam caused by the range spring, equals that of the feedback spring, the valve has reached the new position called-for by the signal. The valve will remain in this position until the signal is changed.

PNEUMATIC VOLUME AND PRESSURE BOOSTERS (See Fig. 16)

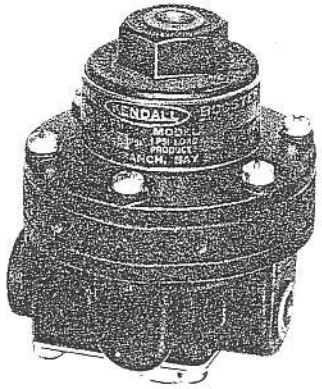

Figure 16. VOLUME AND/OR PRESSURE BOOSTER.

Courtesy Kendall, Inc.

Volume Boosters

A volume booster can be used to increase the speed of operation of the diaphragm control valve. A typical arrangement is shown in Figure 17.

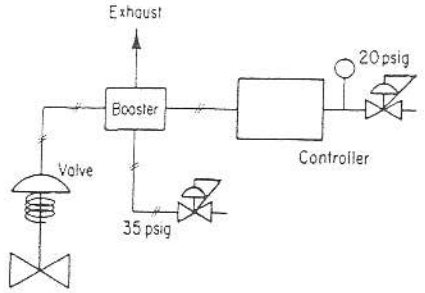

Figure 17. VOLUME BOOSTER IN CONTROL VALVE LOOP.

Figure 17 shows a volume booster being used in conjunction with a controller and a pneumatic valve. The controller applies its output signal to the booster instead of to the control valve. Only about one cubic inch of air is required to position the pilot in the booster, thus, the volume of air moved through the connecting tubing is small. The air that operates the valve comes through the pilot in the booster and since this pilot has a large capacity (35 scfm), the stroking time for the control valve is substantially reduced.

If a pressure reducing valve is used for the supply pressure to the booster, it should be of the high-capacity type so as not to limit the output capacity of the booster.

Pressure Boosters

Pressure boosters are generally also volume boosters; however, their main function is to increase the pressure from the controller to above 20 psig in certain valve applications.

Figure 18 shows a single-seated, air-to-open valve with a 6-30 psi spring. The heavy spring may be used to close the valve against a high upstream pressure; however, the normal controller output (20 psi maximum) cannot open the valve. A 2:1 booster makes the system operational.

The booster may be preferred over a positioner because of its lower cost. Also, the booster does not close a loop around the valve, a fact which may enhance stability in fast-response systems.

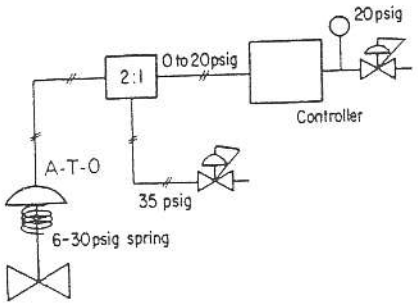

Figure 18. PRESSURE BOOSTER IN CONTROL VALVE LOOP.

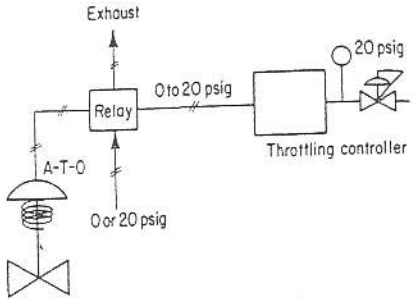

Figure 20. THROTTLING SERVICE, TWO-POSITION PNEUMATIC RELAY FOR EMERGENCY SHUT-DOWN.

ON-OFF AIR RELAYS

An on-off air relay may be used in some applications where the maximum output (20 psi) of the controller is insufficient to operate a diaphragm control valve. Figure 19 shows an on-off pneumatic controller operating an air relay. The relay in turn applies 35 psig or 0 psig to the diaphragm control valve to open or close the valve. The 35 psi pressure allows the single-seated diaphragm control valve to close against higher line pressure than would be possible with the 20 psig output from the controller.

The pneumatic relay may be used with a throttling controller where the relay is used in emergency service, as shown in Figure 20. In this application the relay is normally positioned so that the exhaust port is closed and the path through the relay from controller to the diaphragm control valve is kept open. With proper piping, an emergency signal (0 or 20 psi) can be applied to the relay to close off the controller signal and to open the exhaust port. With the relay exhaust port connected to the control valve, the air-to-open valve is rapidly closed by the valve spring.

SOLENOID VALVES

An electric solenoid valve, as shown in Figure 21, can be used to advantage in combination with diaphragm control valves in some applications.

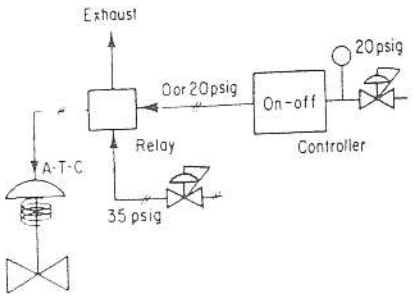

Figure 19. ON-OFF PNEUMATIC RELAY IN

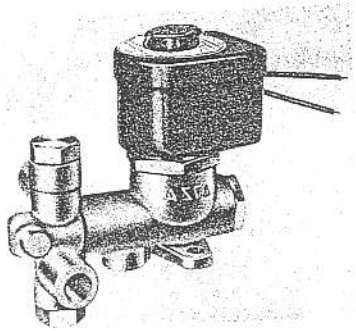

Figure 21. THREE-WAY ELECTRIC SOLENOID VALVE.
Courtesy Automatic Switch Company

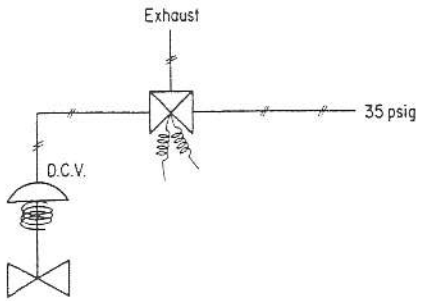

Figure 22. THREE-WAY SOLENOID VALVE USED TO ACTUATE A DIAPHRAGM CONTROL VALVE FOR ON-OFF SERVICE.

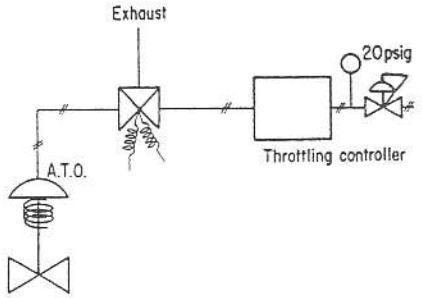

Figure 23. THREE-WAY SOLENOID VALVE IN A DIAPHRAGM CONTROL VALVE LOOP FOR THROTTLING SERVICE.

Figure 22 illustrates a three-way solenoid valve supplying or exhausting air from a diaphragm control valve to achieve on-off operation. Depending on the size, this arrangement may be considerably less expensive and faster responding than an all electric valve.

The three-way solenoid valve may be used with a pneumatic throttling controller where the solenoid is used in emergency service, as illustrated by Figure 23. In this application the solenoid three-way valve is normally positioned so that the exhaust port is closed and the path between the controller and diaphragm control valve is kept open. With proper piping an emergency signal, in the form of electric current through the solenoid, can cause the controller output to be blocked and the exhaust port to be opened. With the solenoid valve exhaust port connected to the diaphragm control valve, the air-to-open valve is rapidly closed by the valve spring.

POSITION INDICATING SWITCHES

For remote, two-position indication of valve stem position, either plunger or roller-actuated switches (see examples in Figure 24) can be mounted directly on the actuator yoke. This indication is limited to two-position and is usually accomplished by having the switch operate red and green indicating lights.

AIR LOCKS

This device is used for applications that require a control valve to hold its position in the event that the plant air supply pressure falls below a given level.

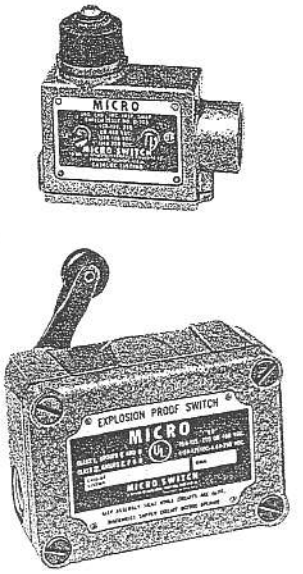

Figure 24. VALVE POSITION INDICATOR SWITCHES.
Courtesy Honeywell, Inc.

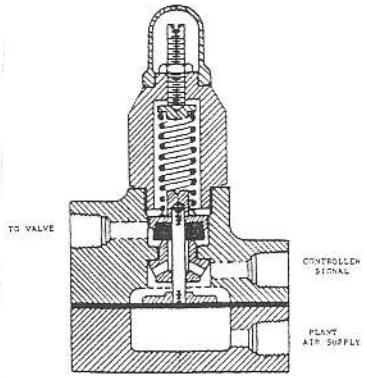

Figure 25. AIR LOCK.

One type is shown in Figure 25. The plant air supply is fed into a chamber sealed by a spring-opposed diaphragm. In the event that the plant air decreases to a predetermined lower limit, the spring closes the connection to the actuator and locks the existing controller-signal pressure in the line connecting the valve operator.

MOTION TRANSMITTERS

The pneumatic motion transmitter (Figure 26) is a device for continuous indication of valve stem position. It is generally yoke mounted and attached to the valve stem through a linkage. Through this linkage, it senses valve stem position and transmits a proportional 3-15 psig signal to a remotely located receiver.

Devices which transmit the valve stem position electrically are also available.

ELECTROPNEUMATIC TRANSDUCERS

Electropneumatic transducers convert the electrical output signals from electric controllers into pneumatic signals which may be used to operate diaphragm control valves. They may be either current-to-pressure (I/P) transducers, or voltage-to-pressure (E/P) transducers. A typical (I/P) transducer is schematically shown in Figure 27.

The input current is applied to the magnet unit coil. The combined action of the input current and the magnetic field of the permanent magnets creates a force which is applied to the beam through the strap connecting it to the magnet unit. The beam, which is pivoted at one end, is deflected by this force. In operation, an increase in input current pivots the magnet coil

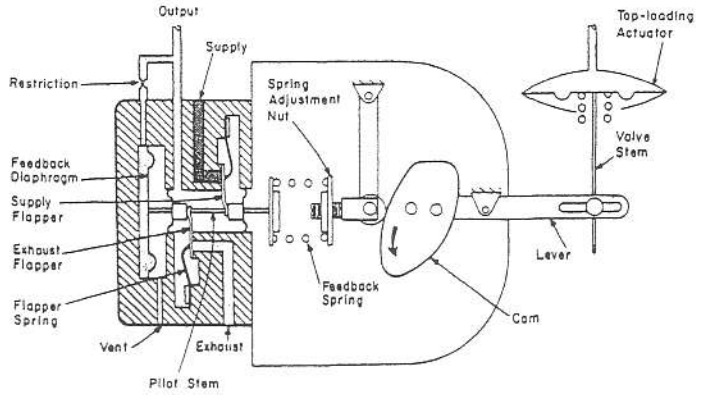

Figure 26. SCHEMATIC DIAGRAM OF A PNEUMATIC MOTION TRANSMITTER.

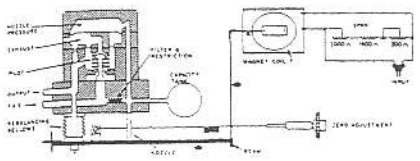

Figure 27. SCHEMATIC DIAGRAM OF AN ELECTROPNEUMATIC (I/P) TRANSDUCER.

clockwise pulling up the strap and moving the beam closer to the nozzle. This increases the nozzle pressure transmitted to the pilot valve and, therefore, increases the output pressure from the pilot valve to both the output line and the rebalancing bellows. At the bellows, the increased output pressure is changed into a balancing force which is applied to the beam to counteract the unbalanced force applied by the magnet unit, thus rebalancing the beam. Zero adjustment is obtained by applying a zeroing force to the beam through the zero spring.

REFERENCES

1. Lloyd, S.G., May, 1969, *Guidelines for the Application of Valve Positioners*, Paper No. TM-23, Fisher Controls, Inc., Marshalltown, Iowa.
2. Lloyd, S.G., October, 1968, *Guidelines for the Use of Valve Positioners and Booster Amplifiers*, Paper No. 68-922, Instrument Society of America, Pittsburgh.

CHAPTER 10

SELECTION OF VALVE MATERIALS OF CONSTRUCTION

C. F. King
E. L. Dixson
D. J. Gaffney

Fisher Controls Company
Marshalltown, Iowa

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INTRODUCTION

Material of construction is of the utmost importance in control valve selection since control valves are called upon to handle all types of fluids, from clean dry air to corrosive chemicals at temperatures ranging from near absolute zero to well above 1000° F., and pressures from near a perfect vacuum to 50,000 psig or higher. There are many materials that can be used and some services require the use of exotic alloys and metals to withstand corrosive fluids. However, since the majority of applications are relatively non-corrosive and at reasonable pressures and temperatures, cast iron and carbon steel are the most common valve body materials.

Most control valve materials can be placed in two categories:

1. The pressure containment materials for the valve body, bonnet, bottom flange and bolting.
2. The valve trim materials for the valve plug, seat ring, cage, valve stem, guide bushings, and packing box parts.

Table I. VALVE BODY (PRESSURE CONTAINMENT) MATERIALS

General Classification	*Applicable Material Specification*	
	Forgings	*Castings*
Cast Iron		ASTM A126
Ductile Iron		ASTM A395
Carbon Steel	ASTM A105	ASTM A216 Grade WCB
Carbon Steel (Low Temp)	ASTM A350 Grade LF1 & LF2	ASTM A352 Grade LCB
Carbon Moly	ASTM A182 Grade F1	ASTM A217 Grade WC1
1-1/4 Cr - 1/2 Mo	ASTM A182 Grade F11	ASTM A217 Grade WC6
2-1/4 Cr - 1 Mo	ASTM A182 Grade F22	ASTM A217 Grade WC9
5 Cr - 1/2 Mo	ASTM A182 Grade F5a	ASTM A217 Grade C5
9 Cr - 1 Mo	ASTM A182 Grade F9	ASTM A217 Grade C12
Type 304	ASTM A182 Grade F304	ASTM A351 Grade CF8
Type 347	ASTM A182 Grade F347	ASTM A351 Grade CF8C
Type 316	ASTM A182 Grade F316	ASTM A351 Grade CF8M
3-1/2 Ni	ASTM A350 Grade LF3	ASTM A 352 Grade LC3
Aluminum	ASTM B247	ASTM B26
Bronze		ASTM B61 ASTM B62
Inconel	ASTM B166	
Monel		ASTM A296-M35
Hastelloy B		ASTM A296-N-12M-1
Hastelloy C		ASTM A494 (CW-12M-1)
Titanium	ASTM B381	ASTM B367
Nickel		ASTM A296-CZ-100
Alloy 20		ASTM A296-CN-7M

MATERIALS FOR PRESSURE CONTAINMENT PARTS

The selection of materials for the pressure containment parts is primarily dependent upon the following conditions of the fluid media: pressure, temperature, corrosion properties, and erosion properties. Sometimes a compromise must be made in material selection. For example, a material with good erosion resistance may not be satisfactory because of poor corrosion properties when handling a particular fluid.

Pressure and Temperature Requirements

Pressure and temperature ratings for the pressure containment parts have been established for the more common materials by the American National Standards Institute (ANSI). Other materials may be used, but are generally confined to the materials and allowable stresses listed in Section VIII of the American Society of Mechanical Engineers, *Boiler and Pressure Vessel Code.*

The recommended materials for specified physical and chemical requirements are given in standards established by the American Society for Testing and Materials (ASTM). It is considered good practice, especially for the pressure containment parts, to specify materials by the proper ASTM designation. Table I lists the more popular pressure containment materials and their respective ASTM specification designations.

Control valves are used in services with a wide variety of temperatures. Table II lists valve body (pressure containment) materials with their respective upper and lower temperature limits.

Since most materials have a reduction in allowable working stress at elevated temperatures, the pressure-temperature ratings must also be considered in choice of materials. These are tabulated for cast iron (Table III), ductile iron (Table IV), steel and steel alloys (Table V), and bronze (Table VI).

Control valve body bolting materials are selected on considerations of strength and temperature. Table VII indicates materials used for stuffing box, bonnet, and bottom flange bolting for various temperature ranges.

Table II. VALVE BODY MATERIAL TEMPERATURE LIMITS DEGREES F.

Material	*Lower*	*Upper*
Cast Iron	-20	410
Ductile Iron	-20	650
*Carbon Steel (Grade WCB)	-20	1000
Carbon Steel (Grade LCB)	-50	650
Carbon Moly (Grade WC1)	-20	850
1-1/4 Cr - 1/2 Mo (Grade WC6)	-20	1000
2-1/4 Cr - 1 Mo (Grade WC9)	-20	1050
5 Cr - 1/2 Mo (Grade C5)	-20	1100
9 Cr - 1 Mo (Grade C12)	-20	1100
Type 304 (Grade CF8)	-425	1500
Type 347 (Grade CF8C)	-425	1500
Type 316 (Grade CF8M)	-425	1500
3-1/2 Ni (Grade LC3)	-150	650
Aluminum	-325	400
Bronze	-325	550
Inconel	-325	1200
Monel	-325	900
Hastelloy B	-325	700
Hastelloy C	-325	1000
Titanium		600
Nickel	-325	500
Alloy 20	-50	300

*The carbon phase of carbon steel may be converted to graphite upon long exposure to temperatures above 775° F. (Check applicable codes for maximum temperature rating of various materials.)

Table III. PRESSURE-TEMPERATURE RATINGS OF CAST IRON PIPE FLANGES AND FLANGED FITTINGS*

	125 lb ASTM A126			250 lb ASTM A126		
Temperature Degrees F.	*Class A Sizes 1-12*	*Class B Sizes 14-24*	*Class B Sizes 30-48*	*Class A Sizes 1-12*	*Class B Sizes 14-24*	*Class B Sizes 30-48*
-20 to 150	175	150	150	400	300	300
200	165	135	115	370	280	250
225	155	130	100	355	270	225
250	150	125	85	340	260	200
275	145	120	65	325	250	175
300	140	110	50	310	240	150
325	130	105		295	230	125
350[1]	125	100		280	220	100
375				265	210	
400[2]				250	200	

All pressure ratings given in PSIG. Allowable pressures may be interpolated between temperatures.

[1]353 F (max) to reflect temperature of saturated steam at 125 psig.
[2]406 F (max) to reflect temperature of saturated steam at 250 psig.

Note: Class A Cast Iron - 21,000 psi min. tensile strength
Class B Cast Iron - 31,000 psi min. tensile strength

LIMITATIONS: 250 lb. - when used for liquid service the tabulated pressure-temperature ratings in sizes 14 in. and larger are applicable to 250 lb. flanges only and not to 250 lb. fittings. (Consult manufacturer for ratings of valves 14" and larger).

*Extracted from ANSI Standard Cast Iron Pipe Flanges and Flanged Fittings (ANSI B16.1-1967), with the permission of the publisher, the American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, N.Y. 10017.

Table IV. PRESSURE-TEMPERATURE RATINGS FOR NODULAR IRON COMPONENTS MANUFACTURED IN COMPLIANCE WITH ANSI B16.5. †

	Type of Flange		
Service Temperature Degrees F	*150 lb. Steel*	*300 lb. Steel*	*600 lb. Steel*
-20 to 100	220	576	1000
150	204	568	1000
200	192	560	1000
250	180	552	1000
300	168	544	1000
350	156	540	1000
400	144	532	1000
450	132	520	1000
500	120	500	1000
550	112	472	944
600	104	444	888
650	96	412	824

Cast Nodular Iron Must Conform To ASTM A395.
Pressure ratings in this table are 80% of those in ANSI B16.5-1968 for carbon steel.

† Extracted in part from American National Standards Institute "Standard Code for Pressure Piping, Power Piping," ANSI B31.1-1973, with the permission of the publisher, The American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, N.Y. 10017.

Table V. PRESSURE-TEMPERATURE RATINGS FOR STEEL AND STEEL ALLOYS* ANSI B16.5-1968

The pressure and temperature ratings given below apply to all flange facings used on steel valve bodies. Screwed steel fittings have the same rating as shown under 600 pound. All pressures are in pounds per square inch gauge (psig)

Service Temperature Degrees F.	150 POUND									300 POUND								
	WCB	WCI	WC6	WC9	C5	C12	304	347	316	WCB	WCI	WC6	WC9	C5	C12	304	347	316
-20 to 100	275									720						615		720
150	255									710						585		710
200	240									700						550		700
250	225									690						520		690
300	210									680						495		680
350	195									675						470		675
400	180									665						450		665
450	165									650						430		650
500	150									625						410		625
550	140									590						395		590
600	130									555						380		555
650	120									515						370		515
700	110									470	480			485		355		495
750	100									425	445			450		340		470
800	92									365	410			415		330		450
850	[82	82								[300	370			385		320		425
875	75	[75								260	[355			365		315		415
900	70]	70]								225]	335]			350		310		400
925	60	60								190	320			335		305		390
950	55	55								155	300			315		305		380
975	50]	50]								120]	280]			300		300		370
1000	[40	[40								[85	[215	265	265	250	290	300		355
1025	–	–	–	–	–	–	–	–	–	–	–	230	235	215	240	295		345
1050	–	–	–	–	–	–	–	–	–	–	–	190	200	190	190	290		335
1075	–	–	–	–	–	–	–	–	–	–	–	165	170	145	150	275		325
1100	–	–	–	–	–	–	–	–	–	–	–	135	145	115	115	255		310
1125	–	–	–	–	–	–	–	–	–	–	–	110	125	95	95	225		300
1150	–	–	–	–	–	–	–	–	–	–	–	85	105	75	75	195	260	290
1175	–	–	–	–	–	–	–	–	–	–	–	65	85	65	65	175	215	260
1200	–	–	–	–	–	–	–	–	–	–	–	40	70	50	50	155	170	235
1225	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	135	140	205
1250	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	110	115	180
1275	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	100	95	160
1300	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	85	75	135
1325	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	75	65	115
1350	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	60	50	95
1375	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	55	45	80
1400	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	50	40	70
1425	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	40	35	60
1450	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	35	30	50
1475	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	30	30	45
1500	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–	25	25	35

* Extracted from ANSI Standard Steel Pipe Flanges and Flanged Fittings (ANSI B16.5 - 1968) with the permission of the publisher, the American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, New York, 10017.

Note: Figures set in brackets must be used in accordance with the Introductory Notes, Paragraph 2.3, of ANSI B16.5-1968. (Applicable codes warn of possible graphitization upon prolonged exposure to temperatures corresponding to bracketed figures.) Abbreviations used in the column headings may be identified by referring to Table 2.

Table V. Cont'd. PRESSURE-TEMPERATURE RATINGS FOR STEEL AND STEEL ALLOYS* ANSI B16.5-1968

The pressure and temperature ratings given below apply to all flange facings used on steel valve bodies. Screwed steel fittings have the same rating as shown under 600 pound. All pressures are in pounds per square inch gauge (psig)

Service Temperature Degrees F.	600 POUND									900 POUND								
	WCB	WCI	WC6	WC9	C5	C12	304	347	316	WCB	WCI	WC6	WC9	C5	C12	304	347	316
-20 to 100	1440						1235		1440	2160						1850		2160
150	1420						1165		1420	2130						1750		2130
200	1400						1095		1400	2100						1645		2100
250	1380						1040		1380	2070						1565		2070
300	1365						985		1365	2050						1480		2050
350	1350						945		1350	2025						1415		2025
400	1330						900		1330	2000						1350		2000
450	1305						860		1305	1955						1290		1955
500	1250						825		1250	1875						1235		1875
550	1180						795		1180	1775						1190		1775
600	1110						765		1110	1660						1145		1660
650	1030						735		1030	1550						1105		1550
700	940	960			965		710		985	1410	1440			1450		1065		1480
750	850	890			900		685		940	1275	1330			1360		1025		1410
800	730	815			835		660		895	1100	1225			1250		985		1345
850	[600	[745			765		640		850	[900	1115			1150		960		1275
875	525	710			735		630		825	785	1060			1100		945		1240
900	445]	670]			700		620		805	670]	1010]			1050		930		1205
925	375	635			665		615		780	565	955			1000		920		1175
950	310	600			635		610		760	465	900			950		915		1140
975	240	555			600		605		735	360	835			900		905		1105
1000	[170	[430	535	535	500	585	600		715	[255	[645	800	800	750	875	900		1070
1025	–	–	[455	465	430	480	595		690	–	–	[685	700	645	720	890		1035
1050	–	–	375]	400	355	375	585		670	–	–	565]	595	535	565	875		1000
1075	–	–	325	[345	290	300	550		645	–	–	490	[515	435	455	825		970
1100	–	–	275	290	225	225	515		625	–	–	410	430	340	340	770		935
1125	–	–	225	245]	190	190	455		600	–	–	335]	370]	285	285	680		900
1150	–	–	170	205	150	150	395	520	585	–	–	255	310	225	225	590	780	875
1175	–	–	125	170	125	125	350	430	525	–	–	190	255	190	190	525	650	785
1200	–	–	80]	135]	105	105	310	345	465	–	–	125]	205]	155	155	465	515	700
1225	–	–	–	–	–	–	265	285	415	–	–	–	–	–	–	400	425	620
1250	–	–	–	–	–	–	225	225	365	–	–	–	–	–	–	335	340	545
1275	–	–	–	–	–	–	195	190	320	–	–	–	–	–	–	295	285	480
1300	–	–	–	–	–	–	170	150	275	–	–	–	–	–	–	250	225	410
1325	–	–	–	–	–	–	145	125	230	–	–	–	–	–	–	220	190	345
1350	–	–	–	–	–	–	125	105	185	–	–	–	–	–	–	185	155	280
1375	–	–	–	–	–	–	110	95	160	–	–	–	–	–	–	165	140	240
1400	–	–	–	–	–	–	95	80	135	–	–	–	–	–	–	145	125	205
1425	–	–	–	–	–	–	80	70	120	–	–	–	–	–	–	125	110	180
1450	–	–	–	–	–	–	70	60	105	–	–	–	–	–	–	105	95	155
1475	–	–	–	–	–	–	60	55	85	–	–	–	–	–	–	90	85	130
1500	–	–	–	–	–	–	50	50	70	–	–	–	–	–	–	75	75	105

* Extracted from ANSI Standard Steel Pipe Flanges and Flanged Fittings (ANSI B16.5 - 1968) with the permission of the publisher, the American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, New York, 10017.

Note: Figures set in brackets must be used in accordance with the Introductory Notes, Paragraph 2.3, of ANSI B16.5-1968. (Applicable codes warn of possible graphitization upon prolonged exposure to temperatures corresponding to bracketed figures.) Abbreviations used in the column headings may be identified by referring to Table 2.

2500 POUND

Temperature Degrees F.	WCB	WCI	WC6	WC9	C5	C12	304	347	316
-20 to 100	3600						3085		3600
150	3550						2915		3550
200	3500						2740		3500
250	3450						2605		3450
300	3415						2470		3415
350	3375						2360		3375
400	3330						2245		3330
450	3255						2150		3255
500	3125						2055		3125
550	2955						1985		2955
600	2770						1910		2770
650	2580						1845		2580
700	2350	2400			2415		1775		2465
750	2125	2220			2250		1710		2355
800	1830	2040			2080		1645		2240
850	[1500	1860			1915		1595		2125
875	1305	[1770			1830		1570		2070
900	1115	1680			1750		1545		2010
925	945	1590			1665		1535		1955
950	770	1500			1585		1525		1900
975	600	1395			1500		1510		1840
1000	[430	[1070	[1335	[1335	[1250	[1455	[1500	[1305	[1785
1025	—	—	1140	1165	1070	1200	1485	1080	1725
1050	—	—	945	995	890	945	1455	855	1670
1075	—	—	[815	[855	730	755	1370	710	1615
1100	—	—	685	720	565	565	1285	565	1555
1125	—	—	555	615	470	470	1135	470	1500
1150	—	—	[430	[515	375	375	985	375	1455
1175	—	—	315	430	315	315	880	315	1310
1200	—	—	205	345	255	255	770	255	1165
1225	—	—	—	—	—	—	665	710	1035
1250	—	—	—	—	—	—	555	565	910
1275	—	—	—	—	—	—	490	470	795
1300	—	—	—	—	—	—	420	375	685
1325	—	—	—	—	—	—	365	315	575
1350	—	—	—	—	—	—	310	255	465
1375	—	—	—	—	—	—	275	230	405
1400	—	—	—	—	—	—	240	205	345
1425	—	—	—	—	—	—	205	180	300
1450	—	—	—	—	—	—	170	155	255
1475	—	—	—	—	—	—	150	140	215
1500	—	—	—	—	—	—	130	130	170

Temperature Degrees F.	WCB	WCI	WC6	WC9	C5	C12	304	347	316
-20 to 100	6000						5145		6000
150	5915						4855		5915
200	5830						4565		5830
250	5750						4340		5750
300	5690						4115		5690
350	5625						3930		5625
400	5550						3745		5550
450	5430						3585		5430
500	5210						3430		5210
550	4925						3305		4925
600	4620						3180		4620
650	4300						3070		4300
700	3920	4000			4025		2960		4110
750	3550	3700			3745		2850		3920
800	3050	3400			3470		2745		3730
850	[2500	3100			3190		2660		3540
875	2180	2950			3055		2620		3445
900	1865	2800			2915		2580		3350
925	1570	2650			2775		2560		3260
950	1285	2500			2640		2540		3165
975	1000	2320			2500		2520		3070
1000	[715	[1785	[2230	[2230	[2085	[2430	[2500	[2170	[2975
1025	—	—	1900	1945	1785	2000	2470	1800	2880
1050	—	—	1570	1655	1485	1570	2430	1430	2785
1075	—	—	[1355	[1430	1215	1255	2285	1185	2690
1100	—	—	1145	1200	945	945	2145	945	2595
1125	—	—	930	1030	785	785	1895	785	2500
1150	—	—	[715	[855	630	630	1645	2170	2430
1175	—	—	530	715	530	530	1465	1800	2185
1200	—	—	345	570	430	430	1285	1430	1945
1225	—	—	—	—	—	—	1105	1185	1730
1250	—	—	—	—	—	—	930	945	1515
1275	—	—	—	—	—	—	815	785	1330
1300	—	—	—	—	—	—	700	630	1145
1325	—	—	—	—	—	—	605	530	955
1350	—	—	—	—	—	—	515	430	770
1375	—	—	—	—	—	—	455	385	670
1400	—	—	—	—	—	—	400	345	570
1425	—	—	—	—	—	—	345	300	500
1450	—	—	—	—	—	—	285	255	430
1475	—	—	—	—	—	—	250	235	355
1500	—	—	—	—	—	—	215	215	285

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Note: Figures set in brackets must be used in accordance with the Introductory Notes, Paragraph 2.3, of ANSI B16.5-1968.
(Applicable codes warn of possible graphitization upon prolonged exposure to temperatures corresponding to bracketed figures.)
Abbreviations used in the column headings may be identified by referring to Table 2.

Table VI. PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED VALVES AND FITTINGS. *

ANSI B16.5-1973

A105
A155-CM75
A203-B
A350-LF2
A515-70

A106-C
A155-KC70
A203-E
A350-LF3
A516-70

A155-CM70
A155-KCF70
A216-WCB

TEMPERATURE IN°F	WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	285	740	985	1480	2220	3705	6170
200	260	675	900	1350	2025	3375	5625
300	250	655	875	1310	1970	3280	5470
400	245	635	845	1265	1900	3170	5280
500	230	600	800	1195	1795	2995	4990
600	210	545	730	1095	1640	2735	4560
650	205	535	715	1075	1610	2685	4475
700	205	535	710	1065	1600	2665	4440
750	195	505	670	1010	1510	2520	4200
800	160	410	550	825	1235	2055	3430
850	105	265	355	535	800	1335	2230
900	65	170	230	345	515	855	1430
950	40	105	135	205	310	515	855
1000	20	50	70	105	155	255	430

A182-F1
A217-WC1

A204-A
A352-LC1

A204-B

TEMPERATURE IN°F	WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	265	695	925	1390	2085	3470	5785
200	260	680	905	1360	2035	3395	5655
300	250	650	870	1305	1955	3260	5435
400	245	640	855	1280	1920	3200	5330
500	230	620	830	1245	1865	3105	5175
600	210	605	805	1210	1815	3025	5040
650	205	595	795	1195	1790	2985	4970
700	205	585	780	1170	1755	2920	4870
750	195	555	740	1110	1665	2775	4630
800	160	535	715	1070	1605	2675	4455
850	105	495	660	985	1480	2470	4115
900	65	430	570	855	1285	2145	3570
950	40	290	390	585	875	1455	2430
1000	20	190	250	375	565	945	1570

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Table VI Cont'd. PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED VALVES AND FITTINGS.* ANSI B16.5-1973

TEMPERATURE IN °F	A182-F11, A387-C-NT WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	285	750	1000	1500	2250	3750	6250
200	260	710	950	1425	2135	3560	5930
300	250	675	895	1345	2020	3365	5605
400	245	660	880	1315	1975	3290	5485
500	230	640	855	1285	1925	3210	5350
600	210	605	805	1210	1815	3025	5040
650	205	590	785	1175	1765	2940	4905
700	205	570	755	1135	1705	2840	4730
750	195	530	710	1065	1595	2655	4430
800	160	505	675	1015	1520	2535	4230
850	105	485	650	975	1460	2435	4055
900	65	450	600	900	1345	2245	3745
950	40	375	505	755	1130	1885	3145
1000	20	225	295	445	670	1115	1855
1050		135	185	275	410	685	1145
1100		95	130	190	290	480	800
1150		50	70	105	155	255	430
1200		35	45	70	105	170	285

TEMPERATURE IN °F	A182-F22, A217-WC9, A387-D-NT WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	285	750	1000	1500	2250	3750	6250
200	260	715	955	1430	2150	3580	5965
300	250	675	900	1355	2030	3385	5640
400	245	650	865	1295	1945	3240	5400
500	230	640	855	1280	1920	3200	5330
600	210	640	855	1280	1920	3200	5330
650	205	635	850	1270	1905	3180	5295
700	205	635	845	1265	1900	3170	5280
750	195	600	800	1200	1800	3000	5000
800	160	585	775	1165	1750	2915	4855
850	105	540	720	1085	1625	2710	4515
900	65	480	640	960	1440	2400	4000
950	40	375	505	755	1130	1885	3145
1000	20	265	355	535	800	1335	2230
1050		200	265	400	595	995	1655
1100		115	150	225	340	565	945
1150		105	135	205	310	515	855
1200		70	90	135	205	345	570

Table VI Cont'd. PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED VALVES AND FITTINGS.* ANSI B16.5-1973

TEMPERATURE IN °F	A182-FSA, A217-C5 WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	285	750	1000	1500	2250	3750	6250
200	260	750	1000	1500	2250	3750	6250
300	250	745	995	1490	2235	3725	6205
400	245	730	975	1465	2195	3660	6105
500	230	710	945	1420	2130	3550	5915
600	210	690	920	1380	2075	3455	5760
650	205	680	910	1360	2045	3405	5675
700	205	665	890	1335	2000	3335	5555
750	195	550	730	1095	1645	2745	4570
800	160	495	665	995	1490	2485	4145
850	105	440	585	880	1315	2195	3655
900	65	355	470	705	1060	1765	2945
950	40	260	345	520	780	1305	2170
1000	20	190	255	385	575	960	1600
1050		140	185	280	420	705	1170
1100		105	135	205	310	515	855
1150		70	90	135	205	345	570
1200		45	60	90	135	225	370

TEMPERATURE IN °F	A182-F9, A217-C12 WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	285	750	1000	1500	2250	3750	6250
200	260	750	1000	1500	2250	3750	6250
300	250	745	995	1490	2235	3725	6205
400	245	730	975	1465	2195	3660	6105
500	230	710	945	1420	2130	3550	5915
600	210	690	920	1380	2075	3455	5760
650	205	680	910	1360	2045	3405	5675
700	205	665	890	1335	2000	3335	5555
750	195	600	800	1200	1800	3000	5000
800	160	585	775	1165	1750	2915	4855
850	105	530	705	1055	1585	2640	4400
900	65	450	600	900	1345	2245	3745
950	40	370	495	740	1110	1850	3085
1000	20	290	390	585	875	1455	2430
1050		190	250	375	565	945	1570
1100		115	150	225	340	565	945
1150		75	100	150	225	375	630
1200		50	70	105	155	255	430

*Extracted from ANSI Standard Steel Pipe Flanges, Flanged Valves and Fittings (ANSI B16.5-1973) with the

Table VI Cont'd. PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED VALVES AND FITTINGS. * ANSI B16.5-1973

A351-CF3 A351-CF8

TEMPERATURE IN °F	WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	235	615	825	1235	1850	3085	5145
200	195	515	685	1030	1545	2570	4285
300	175	455	605	905	1360	2265	3770
400	155	405	540	805	1210	2015	3360
500	145	375	500	750	1125	1870	3120
600	135	355	475	710	1070	1780	2965
650	135	350	465	700	1050	1750	2915
700	130	345	460	690	1035	1730	2880
750	130	340	455	685	1025	1705	2845
800	130	335	450	675	1010	1685	2810
850	105	335	445	665	1000	1665	2775
900	65	330	440	660	985	1645	2745
950	40	320	430	640	965	1605	2675
1000	20	310	415	620	930	1555	2590
1050		310	410	615	925	1545	2570
1100		255	345	515	770	1285	2145
1150		195	260	390	585	975	1630
1200		155	205	310	465	770	1285
1250		110	145	220	330	550	915
1300		80	110	165	245	410	685
1350		60	80	125	185	310	515
1400		50	65	95	145	240	400
1450		35	45	70	105	170	285
1500		25	30	50	70	120	200

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Table VI Cont'd. PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED VALVES AND FITTINGS.* ANSI B16.5-1973

A182-F-347 A182-F-348 A240-347
A312-TP347 A351-CF8C A358-347
A376-TP347 A430-FP347
A479-347 A479-348

TEMPERATURE IN °F	WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	235	615	825	1235	1850	3085	5145
200	210	545	725	1090	1635	2725	4545
300	195	505	670	1010	1510	2520	4200
400	180	475	635	950	1425	2375	3960
500	170	445	590	890	1335	2220	3705
600	160	420	560	845	1265	2110	3515
650	160	410	550	825	1235	2055	3430
700	155	400	535	800	1205	2005	3345
750	150	395	525	790	1185	1975	3290
800	150	390	520	780	1175	1955	3255
850	105	380	505	760	1140	1905	3170
900	65	370	495	740	1110	1850	3085
950	40	370	490	735	1105	1840	3070
1000	20	350	465	700	1050	1750	2915
1050		350	465	700	1050	1750	2915
1100		310	415	625	935	1560	2600
1150		210	280	420	625	1045	1745
1200		150	200	300	455	755	1225
1250		115	150	225	340	565	945
1300		75	100	150	225	375	630
1350		50	70	105	155	255	430
1400		40	55	80	125	205	345
1450		30	40	60	95	155	255
1500		25	30	50	70	120	200

* Extracted from ANSI Standard Steel Pipe Flanges, Flanged Valves and Fittings (ANSI B16.5-1973) with the permission of the publisher, the American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, New York, 10017.

Table VI Cont'd. PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED VALVES AND FITTINGS. * ANSI B16.5-1973

A351-CF3M A351-CF8M

TEMPERATURE IN°F	WORKING PRESSURES IN PSI						
	150	300	400	600	900	1500	2500
-20 TO 100	235	615	825	1235	1850	3085	5145
200	205	535	715	1070	1605	2675	4455
300	185	485	645	970	1455	2425	4045
400	175	455	610	915	1370	2285	3805
500	170	450	600	895	1345	2240	3735
600	165	435	580	870	1310	2180	3635
650	165	430	570	855	1285	2140	3565
700	160	420	560	845	1265	2110	3515
750	160	410	550	825	1235	2055	3430
800	155	405	540	810	1215	2025	3375
850	105	400	530	800	1195	1995	3325
900	65	400	530	800	1195	1995	3325
950	40	400	530	795	1195	1990	3315
1000	20	390	520	780	1175	1955	3255
1050		365	485	725	1090	1815	3030
1100		320	430	645	965	1610	2685
1150		275	365	550	825	1370	2285
1200		205	275	410	615	1030	1715
1250		180	240	365	545	910	1515
1300		135	185	275	410	685	1145
1350		105	135	205	310	515	855
1400		75	100	150	225	375	630
1450		60	80	115	175	290	485
1500		40	55	80	125	205	345

* Extracted from ANSI Standard Steel Pipe Flanges, Flanged Valves and Fittings (ANSI B16.5-1973) with the permission of the publisher, the American Society of Mechanical Engineers, United Engineering Center, 345

Table VII. PRESSURE-TEMPERATURE RATINGS BRONZE FLANGES AND FLANGED FITTINGS *
150 and 300 lb.

	Material	*Pressure psi (Max)*	*Temperature Deg. F (Max)*
150 lb	ASTM B62	150 225	406 150
	ASTM B61	150 225	500 150
300 lb	ASTM B62	300 500	422 150
	ASTM B61	300 500	550 150

* Extracted from American Standard Bronze Flanges and Flanged Fittings, 150 and 300 Lb (ANSI B16.24-1971) with the permission of the Publisher, The American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York 17, N.Y. 10017.

Table VIII. VALVE BODY BOLTING MATERIALS

Body Materials	*Temperature Degrees F.*	*Material Specifications* Bolts	Nuts
Cast Iron	-20 to 410	ASTM A307 Gr B	ASTM A307 Gr B
Ductile Iron	-20 to 650	ASTM A193 Gr B7	ASTMN A194 Gr 2H
Carbon Steel (Grade WCB)	-20 to 800	ASTM A193 Gr B7	ASTM A194 Gr 2H
Carbon Steel (Grade LCB)	-50 to 650	ASTM A193 Gr B7	ASTM A194 Gr 2H
Carbon Moly (Grade WC1)	-20 to 800 801 to 850	ASTM A193 Gr B7 ASTM A193 Gr B7	ASTM 194 Gr 2H ASTM A194 Gr 7
1¼ Cr - ½ Mo (Grade WC6)	-20 to 800 801 to 1000	ASTM A193 Gr B7 ASTM A193 Gr B7	ASTM A194 Gr 2H ASTM A194 Gr 7
2¼ Cr - 1 Mo (Grade WC9)	-20 to 800 801 to 1000 1001 to 1050	ASTM A193 Gr B7 ASTM A193 Gr B7 ASTM A193 Gr B16	ASTM A194 Gr 2H ASTM A194 Gr 7 ASTM A194 Gr 7
5 Cr - ½ Mo (Grade C5)	-20 to 800 801 to 1000 1001 to 1100	ASTM A193 Gr B7 ASTM A193 Gr B7 ASTM A193 Gr B16	ASTM A194 Gr 2H ASTM A194 Gr 7 ASTM A194 Gr 4
9 Cr - 1 Mo (Grade C12)	-20 to 800 801 to 1000 1001 to 1100	ASTM A193 Gr B7 ASTM A193 Gr B7 ASTM A193 Gr B16	ASTM A194 Gr 2H ASTM A194 Gr 7 ASTM A194 Gr 4
Type 304 (Grade CF8)	-425 to 100 100 to 1500	ASTM A320 Gr B8 * ASTM A193 Gr B8	ASTM A194 Gr 8 * ASTM A194 Gr 8
Type 347 (Grade CF8C)	-425 to 100 100 to 1500	ASTM A320 Cr B8 * ASTM A193 Gr B8	ASTM A194 Gr 8 * ASTM A194 Gr 8
Type 316 (Grade CF8M)	-325 to 100 100 to 1500	ASTM A320 Gr B8 * ASTM A193 Gr B8M	ASTM A194 Gr 8 * ASTM A194 Gr 8M
3½ Ni (Grade LC3)	-150 to -50	ASTM A320 Gr L7	ASTM A194 Gr 4 With Charpy Test or 8M
		ASTM A193 Gr B7	ASTM A194 Gr 2H

Corrosion Resistance

The corrosion properties of the fluid media are important factors in material selection. Table IX is intended to give only a general indication of how a limited selection of materials will react when in contact with certain fluids. The recommendations cannot be absolute because concentration, temperature, contaminants, pressure and other conditions may alter the suitability of a particular material. For example, Table IX shows chlorine gas as 'A' for carbon steel and 'C' for titanium. While this is true for dry gas, the opposite is true if the gas is wet. It is therefore emphasized that this Table should be used only as a guide. References 1-4 provide additional reading on the subject of corrosion.

Erosion Resistance

Erosion is damage caused by the impingement of high velocity particles on the material surfaces. Entrained sand, slurries, catalyst fines, and flashing liquids are sometimes associated with this type of wear.

The selection of materials for the pressure containment parts is rather limited and the use of sacrificial liners at the areas of impingement have been successful. Figure 1 illustrates an angle type control valve and Figure 2 illustrates a ball type control valve with this type of construction. Experience has shown that longer life can be expected from a flashing liquid application by using materials alloyed with chromium, such as ASTM A217 Grade C5 or ASTM A351 Grade CF8M.

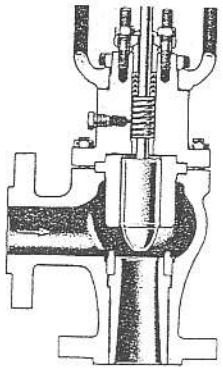

Figure 1. ANGLE VALVE.

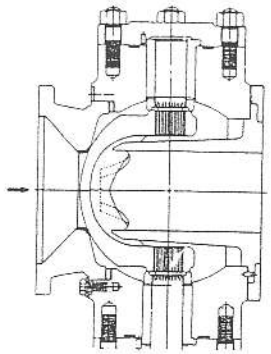

Figure 2. BALL VALVE.

MATERIALS FOR TRIM PARTS

The control valve trim parts which include valve plug, seat ring, cage, valve stem, guide bushings, and packing box parts have a wider lattitude in the selection of materials as they do not comprise the pressure containment parts of the control valve assembly and, therefore, do not come under the jurisdiction of various codes. The selection of these materials is generally predicated by the factors of corrosion, erosion, wear and galling, and temperature.

Corrosion Resistance

Table IX may be used as a general guide for the corrosion resistance of trim materials. A typical valve plug failure caused by corrosion is illustrated in Figure 3.

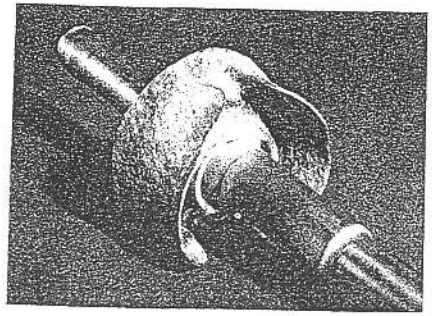

Figure 3. VALVE PLUG FAILURE CAUSED BY CORROSION.

Table IX. CORROSION SELECTION

(PLEASE READ PARAGRAPH ENTITLED CORROSION RESISTANCE BEFORE USING THIS TABLE)

	Carbon Steel	Cast Iron	302 or 304 Stainless Steel	316 Stainless Steel	Bronze	Monel	Hastelloy "B"	Hastelloy "C"	Durimet 20	Titanium	Alloy 6 (co-cr)	416 Stainless Steel	440 C Hard Stainless Steel	17-4 Ph Hard Stainless Steel
Acetaldehyde	A	A	A	A	A	A	I.L.	A	A	I.L.	I.L.	A	A	A
Acetic Acid, Air Free	C	C	B	B	B	B	A	A	A	A	A	C	C	B
Acetic Acid, Aerated	C	C	A	A	A	A	A	A	A	A	A	C	C	B
Acetic Acid Vapors	C	C	A	A	B	B	I.L.	A	B	A	A	C	C	B
Acetone	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Acetylene	A	A	A	A	I.L.	A	A	A	A	I.L.	A	A	A	A
Alcohols	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Aluminum Sulfate	C	C	A	A	B	B	A	A	A	A	I.L.	C	C	I.L.
Ammonia	A	A	A	A	C	A	A	A	A	A	A	A	A	I.L.
Ammonium Chloride	C	C	B	B	B	B	A	A	A	A	B	C	C	I.L.
Ammonium Nitrate	A	C	A	A	C	C	A	A	A	A	A	C	B	I.L.
Ammonium Phosphate (Mono-Basic)	C	C	A	A	B	B	A	A	B	A	A	B	B	I.L.
Ammonium Sulfate	C	C	B	A	B	A	A	A	A	A	A	C	C	I.L.
Ammonium Sulfite	C	C	A	A	C	C	I.L.	A	A	A	A	B	B	I.L.
Aniline	C	C	A	A	C	B	A	A	A	A	A	C	C	I.L.
Asphalt	A	A	A	A	A	A	A	A	A	I.L.	A	A	A	A
Beer	B	B	A	A	B	A	A	A	A	A	A	B	B	A
Benzene (Benzol)	A	A	A	A	A	A	A	A	A	A	A	A	A	A

KEY: A - Can be or is being successfully used.
B - Proceed with caution.
C - Should not be used.
I.L. - Information lacking.

Table IX. cont'd. CORROSION SELECTION

	Carbon Steel	Cast Iron	302 or 304 Stainless Steel	316 Stainless Steel	Bronze	Monel	Hastelloy "B"	Hastelloy "C"	Durimet 20	Titanium	Alloy 6 (co-cr)	416 Stainless Steel	440 C Hard Stainless Steel	17-4 Ph Hard Stainless Steel
Benzoic Acid	C	C	A	A	A	A	I.L.	A	A	A	I.L.	A	A	A
Boric Acid	C	C	A	A	A	A	A	A	A	A	A	B	B	I.L.
Butane	A	A	A	A	A	A	A	A	A	I.L.	A	A	A	A
Calcium Chloride (Alkaline)	B	B	C	B	C	A	A	A	A	A	I.L.	C	C	I.L.
Calcium Hypochlorite	C	C	B	B	B	B	C	A	A	A	I.L.	C	C	I.L.
Carbolic Acid	B	B	A	A	A	A	A	A	A	A	A	I.L.	I.L.	I.L.
Carbon Dioxide, Dry	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Carbon Dioxide, Wet	C	C	A	A	B	A	A	A	A	A	A	A	A	A
Carbon Disulfide	A	A	A	A	C	B	A	A	A	A	A	B	B	I.L.
Carbon Tetrachloride	B	B	B	B	A	A	B	A	A	A	I.L.	C	A	I.L.
Carbonic Acid	C	C	B	B	B	A	A	A	A	I.L.	I.L.	A	A	A
Chlorine, Gas (Dry)	A	A	B	B	B	A	A	A	A	C	B	C	C	C
Chlorine, Gas (Wet)	C	C	C	C	C	C	C	B	C	A	B	C	C	C
Chlorine, Liquid	C	C	C	C	B	C	C	A	B	C	B	C	C	C
Chromic Acid	C	C	C	B	C	A	C	A	C	A	B	C	C	C
Citric Acid	I.L.	C	B	A	A	B	A	A	A	A	I.L.	B	B	B
Coke Oven Gas	A	A	A	A	B	B	A	A	A	A	A	A	A	A
Copper Sulfate	C	C	B	B	B	C	I.L.	A	A	A	I.L.	A	A	A
Cottonseed Oil	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Creosote	A	A	A	A	C	A	A	A	A	I.L.	A	A	A	A
Ethane	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Ether	B	B	A	A	A	A	A	A	A	A	A	B	B	I.L.
Ethyl Chloride	C	C	A	A	A	A	A	A	A	A	A	A	A	A
Ethylene	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Ethylene Glycol	A	A	A	A	A	A	I.L.	I.L.	A	I.L.	A	A	A	A

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Table IX. cont'd. CORROSION SELECTION

	Carbon Steel	Cast Iron	302 or 304 Stainless Steel	316 Stainless Steel	Bronze	Monel	Hastelloy "B"	Hastelloy "C"	Durimet 20	Titanium	Alloy 6 (co-cr)	416 Stainless Steel	440 C Hard Stainless Steel	17-4 Ph Hard Stainless Steel
Ferric Chloride	C	C	C	C	C	C	C	B	C	A	B	C	C	I.L.
Formaldehyde	B	B	A	A	A	A	A	A	A	A	A	A	A	A
Formic Acid	I.L.	C	B	B	A	A	A	A	A	C	B	C	C	B
Freon, Wet	B	B	B	A	A	A	A	A	A	A	A	I.L.	I.L.	I.L.
Freon, Dry	B	B	A	A	A	A	A	A	A	A	A	I.L.	I.L.	I.L.
Furfural	A	A	A	A	A	A	A	A	A	A	A	B	B	I.L.
Gasoline, Refined	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Glucose	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Hydrochloric Acid (Aerated)	C	C	C	C	C	C	A	B	C	C	B	C	C	C
Hydrochloric Acid (Air Free)	C	C	C	C	C	C	A	B	C	C	B	C	C	C
Hydrofluoric Acid (Aerated)	B	C	C	B	C	C	A	A	B	C	B	C	C	C
Hydrofluoric Acid (Air Free)	A	C	C	B	C	A	A	A	B	C	I.L.	C	C	I.L.
Hydrogen	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Hydrogen Peroxide	I.L.	A	A	A	C	A	B	B	A	A	I.L.	B	B	I.L.
Hydrogen Sulfide, Liquid	C	C	A	A	C	C	A	A	B	A	A	C	C	I.L.
Magnesium Hydroxide	A	A	A	A	B	A	A	A	A	A	A	A	A	I.L.
Mercury	A	A	A	A	C	B	A	A	A	A	A	A	A	B
Methanol	A	A	A	A	A	A	A	A	A	A	A	A	B	A
Methyl Ethyl Ketone	A	A	A	A	A	A	A	A	A	A	I.L.	A	A	A
Milk	C	C	A	A	A	A	A	A	A	A	A	C	C	C
Natural Gas	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Nitric Acid	C	C	A	B	C	C	C	B	A	A	C	C	C	B

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Table IX. cont'd. CORROSION SELECTION

	Carbon Steel	Cast Iron	302 or 304 Stainless Steel	316 Stainless Steel	Bronze	Monel	Hastelloy "B"	Hastelloy "C"	Durimet 20	Titanium	Alloy 6 (co-cr)	416 Stainless Steel	440 C Hard Stainless Steel	17-4 Ph Hard Stainless Steel
Oleic Acid	C	C	A	A	B	A	A	A	A	A	A	A	A	I.L.
Oxalic Acid	C	C	B	B	B	B	A	A	A	B	B	B	B	I.L.
Oxygen	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Petroleum Oils, Refined	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Phosphoric Acid (Aerated)	C	C	A	A	C	C	A	A	A	B	A	C	C	I.L.
Phosphoric Acid (Air Free)	C	C	A	A	C	B	A	A	A	B	A	C	C	I.L.
Phosphoric Acid Vapors	C	C	B	B	C	C	A	I.L.	A	B	C	C	C	I.L.
Picric Acid	C	C	A	A	C	C	A	A	A	I.L.	I.L.	B	B	I.L.
Potassium Chloride	B	B	A	A	B	B	A	A	A	A	I.L.	C	C	I.L.
Potassium Hydroxide	B	B	A	A	B	A	A	A	A	A	I.L.	B	B	I.L.
Propane	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Rosin	B	B	A	A	A	A	A	A	A	I.L.	A	A	A	A
Silver Nitrate	C	C	A	A	C	C	A	A	A	A	B	B	B	I.L.
Sodium Acetate	A	A	B	A	A	A	A	A	A	A	A	A	A	A
Sodium Carbonate	A	A	A	A	A	A	A	A	A	A	A	B	B	A
Sodium Chloride	C	C	B	B	A	A	A	A	A	A	A	B	B	B
Sodium Chromate	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Sodium Hydroxide	A	A	A	A	C	A	A	A	A	A	A	B	B	A
Sodium Hypochloride	C	C	C	C	B-C	B-C	C	A	B	A	I.L.	C	C	I.L.
Sodium Thiosulfate	C	C	A	A	C	C	A	A	A	A	I.L.	B	B	I.L.
Stannous Chloride	B	B	C	A	C	B	A	A	A	A	I.L.	C	C	I.L.
Stearic Acid	A	C	A	A	B	B	A	A	A	A	B	B	B	I.L.

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Table IX. cont'd. CORROSION SELECTION

	Carbon Steel	Cast Iron	302 or 304 Stainless Steel	316 Stainless Steel	Bronze	Monel	Hastelloy "B"	Hastelloy "C"	Durimet 20	Titanium	Alloy 6 (co-cr)	416 Stainless Steel	440 C Hard Stainless Steel	17-4 Ph Hard Stainless Steel
Sulfate Liquor (Black)	A	A	A	A	C	A	A	A	A	A	A	I.L.	I.L.	I.L.
Sulfur	A	A	A	A	C	A	A	A	A	A	A	A	A	A
Sulfur Dioxide, Dry	A	A	A	A	A	A	B	A	A	A	A	B	B	I.L.
Sulfur Trioxide, Dry	A	A	A	A	A	A	B	A	A	A	A	B	B	I.L.
Sulfuric Acid (Aerated)	C	C	C	C	C	C	A	A	A	B	B	C	C	C
Sulfuric Acid (Air Free)	C	C	C	C	B	B	A	A	A	B	B	C	C	C
Sulfurous Acid	C	C	B	B	B	C	A	A	A	A	B	C	C	I.L.
Tar	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Trichloroethylene	B	B	B	A	A	A	A	A	A	A	A	B	B	I.L.
Turpentine	B	B	A	A	A	B	A	A	A	A	A	A	A	A
Vinegar	C	C	A	A	B	A	A	A	A	I.L.	A	C	C	A
Water, Boiler Feed	B	C	A	A	C	A	A	A	A	A	A	B	A	A
Water, Distilled	A	A	A	A	A	A	A	A	A	A	A	B	B	I.L.
Water, Sea	B	B	B	B	A	A	A	A	A	A	A	C	C	A
Whiskey and Wines	C	C	A	A	A	B	A	A	A	A	A	C	C	I.L.
Zinc Chloride	C	C	C	C	C	C	A	A	A	A	B	C	C	I.L.
Zinc Sulfate	C	C	A	A	B	A	A	A	A	A	A	B	B	I.L.

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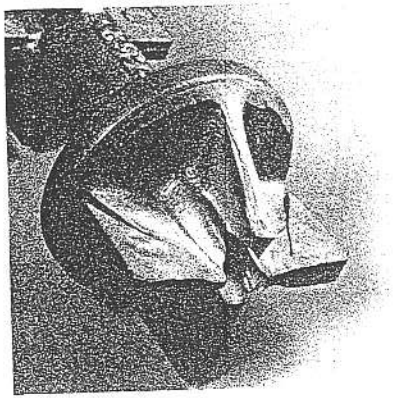

Figure 4. DAMAGE OF VALVE PLUG SEAT BY WIRE-DRAWING.

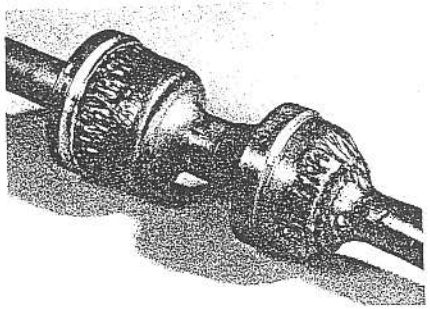

Figure 5. EROSION DAMAGE OF VALVE PLUG BY CATALYST FINES.

Erosion Resistance

The valve plug and seat ring are probably the control valve components that are most susceptible to damage by erosion caused by impingement of high velocity particles. Figure 4 illustrates valve plug seat damage where wet-steam liquid droplets impinged when the valve plug was nearly closed. This type of damage is commonly referred to as "wire drawing."

Figure 5 illustrates a damaged plug which was handling catalyst fines. The typical appearance of erosion damage is the smooth polished wear areas.

Following is a tabulation of various materials listed in order of increasing relative resistance to erosion (the first material would be rated relatively poor and the last relatively good).

Bronze
Aluminum Bronze
Nickel
Alloy 20
Monel
Hastelloy B & C
Type 316
Type 304
K Monel
17-4PH 40RC
Type 416 40RC
Inconel
Type 304 & Type 316 Alloy 6 faced
Type 440 60RC
Chrome & Tungsten Carbide

Wear and Galling Resistance

Wear and galling of valve plug post and bushing materials are usually encountered in applications where high mass flow rates exist. A factor that may contribute to wear is the instability of the plug caused by buffeting forces. Temperature and lubricating qualities of the flowing media and material selection are contributing factors to galling.

Figure 6 illustrates wear damage and Figure 7 illustrates guide post galling. Material selection for both wear and galling resistance are generally in the same category; and better life can be expected by the use of two different materials between the contact surface areas. If like materials must be used, different hardnesses may be beneficial.

Table X lists the wear and galling resistance of various combinations of materials. In addition to these materials, plastic lined bushings have been found to be effective when service conditions permit their usage.

The lubricating qualities of the flowing media also affects wear and galling and such media as dry superheated steam and certain heat exchange liquids have a notable tendency toward this type of failure. As would be expected, lubricating oils have the least tendency.

Temperature Requirements

The service temperature is also a factor in selection of trim materials. Table XI lists the more common trim materials with recommended upper and lower service temperatures. Note that some platings are listed that are occasionally used on valve guiding surfaces. Elastomers that are used for seat discs and packing are included.

Figure 6. VALVE PLUG WEAR DAMAGE.

Elastomers have a variety of usages in a control valve assembly in such applications as actuator diaphragms, piston seals, seat discs, O-ring seals, etc. The choice of materials depends upon service media, pressures and temperatures. Table XII lists the general properties of several materials that are used.

Table XIII illustrates maximum and minimum service temperature limitations for several diaphragm materials. Note that the limits are the extreme temperatures. For practical service life, the maximum temperature should be reduced by at least 25° F. and minimum temperatures raised by 20° to 50° F.

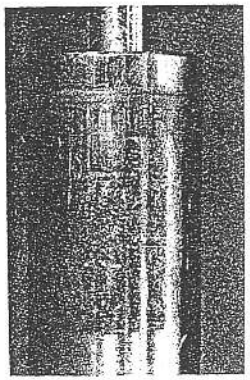

Figure 7. GALLING OF GUIDE POST IN CONTROL VALVE.

Pressures and temperatures exceed the ratings covered by established codes and standards in some control valve applications. In these cases it is recommended that the manufacturer be contacted as these conditions generally warrant control valves of special construction.

REFERENCES

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2. Nelson, G.A., ed., 1967, *Corrosion Data Survey*, National Association of Corrosion Engineers, Houston, Texas.
3. Polar, J.P., ed., 1961, *A Guide to Corrosion Resistance*, Climax Molybdenum Company.
4. Uhlig, H.H., ed., 1948, *The Corrosion Handbook*, John Wiley & Sons, Inc., New York.

Table X. WEAR & GALLING RESISTANCE CHART OF MATERIAL COMBINATIONS

	304 SS	316 SS	BRONZE	INCONEL	MONEL	HASTELLOY B	HASTELLOY C	TITANIUM 75A	NICKEL	ALLOY 20	TYPE 416 HARD	TYPE 440 HARD	17-4 PH	ALLOY 6 (co-cr)	ENC*	Cr PL	AL BRONZE
304 SST	P	P	F	P	P	P	F	P	P	P	F	F	F	F	F	F	F
316 SS	P	P	F	P	P	P	F	P	P	P	F	F	F	F	F	F	F
BRONZE	F	F	S	S	S	S	S	S	S	S	F	F	F	F	F	F	F
INCONEL	P	P	S	P	P	P	F	P	F	F	F	F	F	F	F	F	S
MONEL	P	P	S	P	P	P	F	F	F	F	F	F	F	S	F	F	S
HASTELLOY B	P	P	S	P	P	P	F	F	S	F	F	F	F	S	F	S	S
HASTELLOY C	F	F	S	F	F	F	F	F	F	F	F	F	F	S	F	S	S
TITANIUM 75 A	P	P	S	P	F	F	F	P	F	F	F	F	F	S	F	F	S
NICKEL	P	P	S	F	F	S	F	F	P	P	F	F	F	S	F	F	S
ALLOY 20	P	P	S	F	F	F	F	F	P	P	F	F	F	S	F	F	S
TYPE 416 HARD	F	F	F	F	F	F	F	F	F	F	F	F	F	S	S	S	S
TYPE 440 HARD	F	F	F	F	F	F	F	F	F	F	S	F	S	S	S	S	S
17-4 PH	F	F	F	F	F	F	F	F	F	F	F	S	P	S	S	S	S
ALLOY 6 (co-cr)	F	F	F	F	S	S	S	S	S	S	S	S	S	F	S	S	S
ENC*	F	F	F	F	F	F	F	F	F	F	S	S	S	S	P	S	S
Cr PLATE	F	F	F	F	F	S	S	F	F	F	S	S	S	S	S	P	S
AL BRONZE	F	F	F	S	S	S	S	S	S	S	S	S	S	S	S	S	P

*Electroless Nickel Coating
S - Satisfactory
F - Fair
P - Poor

Table XI. VALVE TRIM MATERIAL TEMPERATURE LIMITS

DEGREES F.

MATERIAL	LOWER	UPPER
Type 304 Stainless Steel	-450	600
Type 316 Stainless Steel	-450	600
Bronze	-460	450
Inconel	-400	1200
K Monel	-400	900
Monel	-400	900
Hastelloy B		700
Hastelloy C		1000
Titanium		600
Nickel	-325	600
Alloy 20	-50	600
Type 416 Stainless Steel 40RC	-20	800
Type 440 Stainless Steel 60RC	-20	800
17-4 PH	-40	800
Alloy 6 (co-cr)	-460	1500
Electroless Nickel Plating	-450	800
Chrome Plating	-450	1100
Aluminum Bronze	-460	600
Nitrile (Buna N)	-40	200
Fluoroelastomer (Viton[1] and Fluorel[2])	-10	400
TFE	-450	450
Nylon	-100	200
Polyethylene	-100	200
Neoprene	-40	180

1. Registered Trademark - DuPont 2. Registered Trademark - 3M Co.

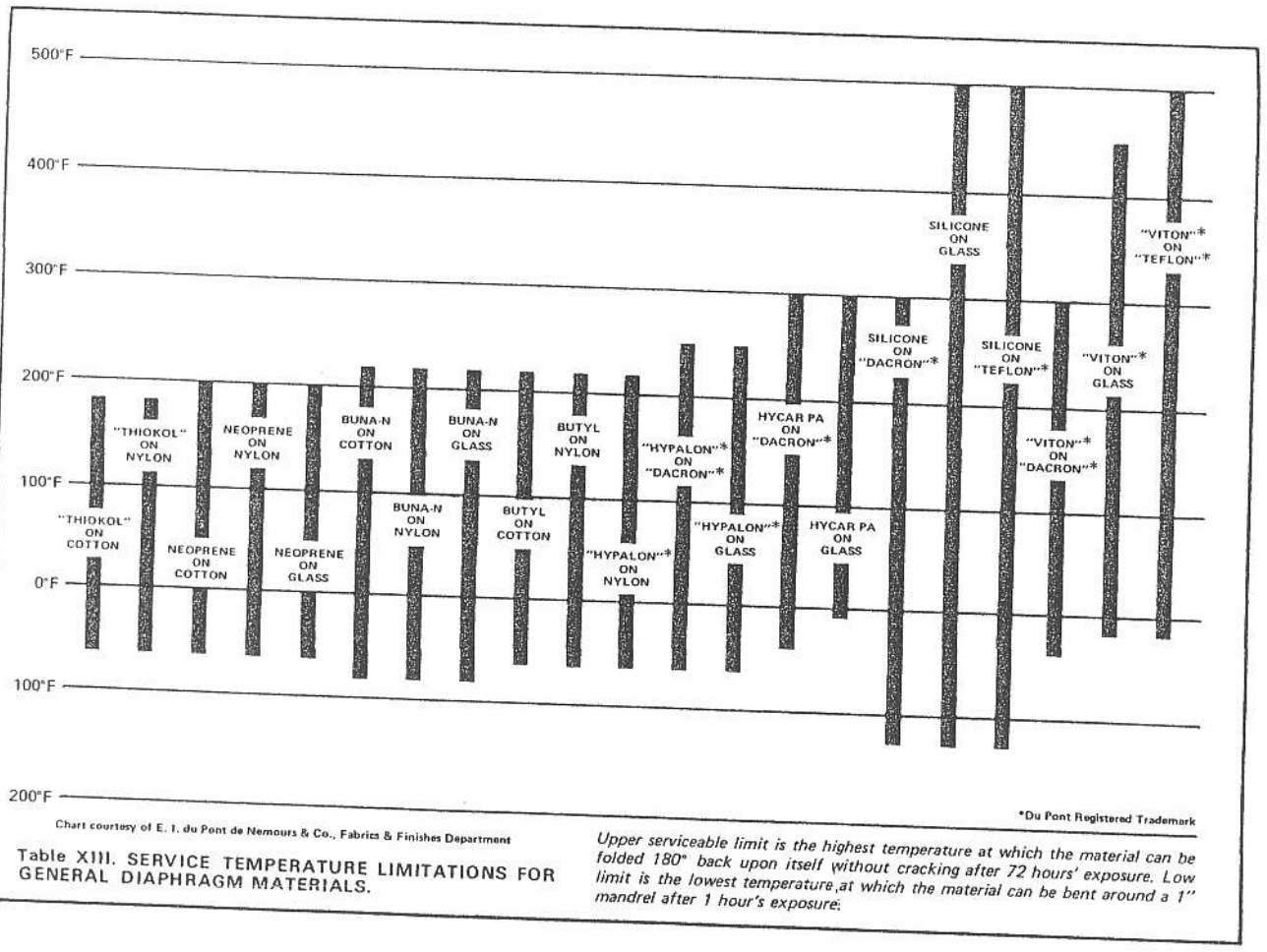
Table XII. GENERAL PROPERTIES OF ELASTOMERS.

Property	Natural Rubber	Buna-S	Buna-N	Neoprene	Butyl	Thiokol	Silicone	Hypalon	①② Viton	② Polyurethane	① Polyacrylic	③ Ethylene Propylene
Tensile: Pure Gum	3000	400	600	3500	3000	300	200-450	4000	----	----	100	----
Reinforced	4500	3000	4000	3500	3000	1500	1100	4400	2300	6500	1800	2500
Tear Resistance	Excellent	Poor-Fair	Fair	Good	Good	Fair	Poor-Fair	Excellent	Good	Excellent	Fair	Poor
Abrasion Resistance	Excellent	Good	Good	Excellent	Fair	Poor	Poor	Excellent	Very Good	Excellent	Good	Good
Aging: Sunlight	Poor	Poor	Poor	Excellent	Excellent	Good	Good	Excellent	Excellent	Excellent	Excellent	Excellent
Oxidation	Good	Fair	Fair	Good	Good	Good	Very Good	Very Good	Excellent	Excellent	Excellent	Good
Heat (Max. Temp.)	200°F.	200°F.	250°F.	200°F.	200°F.	140°F.	450°F.	300°F.	400°F.	200°F.	350°F.	350°F.
Static (Shelf)	Good	Good	Good	Very Good	Good	Fair	Good	Good	----	----	Good	Good
Flex Cracking Resistance	Excellent	Good	Good	Excellent	Excellent	Fair	Fair	Excellent	----	Excellent	Good	----
Compression Set Resistance	Good	Good	Very Good	Excellent	Fair	Poor	Good	Poor	Poor	Good	Good	Fair
Solvent Resistance:												
Aliphatic Hydrocarbon	Very Poor	Very Poor	Good	Fair	Poor	Excellent	Poor	Fair	Excellent	Very Good	Good	Poor
Aromatic Hydrocarbon	Very Poor	Very Poor	Fair	Poor	Very Poor	Good	Very Poor	Poor	Very Good	Fair	Poor	Fair
Oxygenated Solvent	Good	Good	Poor	Fair	Good	Fair	Poor	Poor	Good	Poor	Poor	---
Halogenated Solvent	Very Poor	Very Poor	Very Poor	Very Poor	Poor	Poor	Very Poor	Very Poor	----	----	Poor	Poor
Oil Resistance:												
Low Aniline Mineral Oil	Very Poor	Very Poor	Excellent	Fair	Very Poor	Excellent	Poor	Fair	Excellent	----	Excellent	Poor
High Aniline Mineral Oil	Very Poor	Very Poor	Excellent	Good	Very Poor	Excellent	Good	Good	Excellent	----	Excellent	Poor
Synthetic Lubricants	Very Poor	Very Poor	Fair	Very Poor	Poor	Poor	Fair	Poor	----	----	Fair	Poor
Organic Phosphates	Very Poor	Very Poor	Very Poor	Very Poor	Good	Poor	Poor	Poor	Poor	Poor	Poor	Very Good
Gasoline Resistance:												
Aromatic	Very Poor	Very Poor	Good	Poor	Very Poor	Excellent	Poor	Poor	Good	Fair	Fair	Fair
Non-Aromatic	Very Poor	Very Poor	Excellent	Good	Very Poor	Excellent	Good	Fair	Very Good	Good	Poor	Poor
Acid Resistance:												
Dilute (Under 10%)	Good	Good	Good	Fair	Good	Poor	Fair	Good	Excellent	Fair	Poor	Very Good
▲Concentrated	Fair	Poor	Poor	Fair	Fair	Very Poor	Poor	Good	Very Good	Poor	Poor	Good
Low Temperature Flexibility (Max.)	-65°F.	-50°F.	-40°F.	- 40°F.	-40°F.	-40°F.	-100°F.	-20°F.	-30°F.	-40°F.	-10°F.	-50°F.
Permeability to Gases	Fair	Fair	Fair	Very Good	Very Good	Good	Fair	Very Good	Good	Good	Good	Good
Water Resistance	Good	Very Good	Very Good	Fair	Very Good	Fair	Fair	Fair	Excellent	Fair	Fair	Very Good
Alkali Resistance:												
Dilute (under 10%)	Good	Good	Good	Good	Very Good	Poor	Fair	Good	Excellent	Fair	Poor	Excellent
Concentrated	Fair	Fair	Fair	Good	Very Good	Poor	Poor	Good	Very Good	Poor	Poor	Good
Resilience	Very Good	Fair	Fair	Very Good	Very Good	Poor	Good	Good	Good	Fair	Very Poor	Very Good
Elongation (Max.)	700%	500%	500%	500%	700%	400%	300%	300%	425%	625%	200%	500%

▲ Except Nitric and Sulfuric.

NOTES

1. Do not use with steam.
2. Do not use with ammonia.
3. Do not use with petroleum base fluids. Use with ester Base (non-flammable) hydraulic oils and low pressure steam applications to 300°F.



*Du Pont Registered Trademark

Chart courtesy of E. I. du Pont de Nemours & Co., Fabrics & Finishes Department

Table XIII. SERVICE TEMPERATURE LIMITATIONS FOR GENERAL DIAPHRAGM MATERIALS.

Upper serviceable limit is the highest temperature at which the material can be folded 180° back upon itself without cracking after 72 hours' exposure. Low limit is the lowest temperature at which the material can be bent around a 1" mandrel after 1 hour's exposure.

Elastomer Selection Guide

This table (XIV) rates and compares the compatibility of elastomers with specific fluids. The tabulation is based on published literature of various polymer suppliers and rubber manufacturers, laboratory tests, and records of actual service performance.

Note that this information should be used as a *general guide only*. An elastomer which is compatible with a fluid may not be suitable over the entire range of its temperature capability. In general, chemical compatibility decreases with an increase in service temperature.

Selection of an elastomer for a valve disc seat or liner can only be as accurate as the information on which it is based. Known factors should include: (1) temperature, (2) pressure, (3) all chemicals, primary as well as trace, in the flowing fluid, (4) rate of flow, and (5) type of valve action, throttling or on/off.

Table XIV. FLUID COMPATIBILITY TABLE.

Fluid	Elastomer Ratings For Compatibility With Fluid								
	Natural Rubber	Neoprene	Nitrile	EPT	Poly-urethane	Viton	Hypalon	Butyl	Silicone
tic Acid (30%)	B	C	B	A+	C	B	B	A	A
tone	B	B	C	A	C	C	B	A	B
ambient	B	A	A	A	A	A	A	A	A
hot (200°F)	C	C	A	A	B	A	A	A	C
, hot (400°F)	C	C	C	C	C	B	C	C	C
ohol, ethyl	A	A	A	A	B	B	A	A	A
hol, methyl	A	A+	A	A	C	C	A	A	A
nonia, anhydrous	C	A	C	A	C	C	B	A	C
monia, gas (hot)	C	B	C	B	C	C	B	C	B
r (beverage)	A	A	B	A	C	A	A	A	A
zene	C	C	C	C	C	A	C	C	C
k Liquor	B	B	A	B	C	A+	C	C	C
st Furnace Gas	C	C	B	C	C	A	C	C	A
ne (calcium chloride)	A	A	A	A	A	B	A	A	A
adiene Gas	C	B	C	C	C	B	B	C	C
ane, gas	C	A	A+	C	B	A	A	C	C
ane, liquid	C	B	A	C	C	A	C	C	C
bon Tetrachloride	C	C	C	C	C	A	C	C	C
orine, dry	C	C	C	C	C	A	C	B	C
orine, wet	C	C	C	C	C	A	C	C	C
e Oven Gas	C	C	B	C	C	A+	B	C	B
therm A	C	C	C	C	C	A	C	C	C
yl Acetate	C	C	C	B	C	C	C	B	B
ylene Glycol	A	A	A	A+	B	A	A	A	A

NOTE: These recommendations are to be used as a *general guide only*. Full details regarding pressure, temperature, chemical considerations, and the mode of operation must be considered when selecting an elastomer.

\+ - Best possible selection.

RATINGS: A. RECOMMENDED
B. MINOR TO MODERATE EFFECT

Table XIV. FLUID COMPATIBILITY TABLE (continued)

Fluid	Elastomer Ratings For Compatibility With Fluid								
	Natural Rubber	Neoprene	Nitrile	EPT	Poly-urethane	Viton	Hypalon	Butyl	Silicone
Freon 11	C	B	A	C	C	A+	A	C	C
Freon 12	B	A+	A	B	A	B	A	B	C
Freon 22	C	A+	C	A	C	C	A	A	C
Freon 114	A	A	A	A	A	B	B	A	C
Gasoline	C	B	A+	C	B	A	B	C	C
Hydrogen Gas	B	A	A	A	A	A	A	A	C
Hydrogen Sulfide (dry)	B	A	C	A+	B	C	A	A	C
Hydrogen Sulfide (wet)	C	B	C	A+	C	C	C	A	C
Jet Fuel (JP-4)	C	C	A	C	B	A	C	C	C
Methylene Chloride	C	C	C	C	C	B+	C	C	C
Milk	A	A	A+	A	C	A	A	A	A
Natural Gas	C	A	A+	C	B	A	A	C	C
Natural Gas + H_2S (Sour Gas)	C	A	B	C	B	C	A	C	C
Natural Gas, Sour + Ammonia	C	B	B	C	C	C	B	C	C
Naphthalene	C	C	C	C	B	A	C	C	C
Nitric Acid (10%)	C	B	C	B	C	A	B	B	B
Nitric Acid (50-100%)	C	C	C	C	C	A	C	C	C
Nitric Acid Vapor	C	B	C	B	C	A	B	B	C
Nitrogen	A	A	A	A	A	A	A	A	A
Oil (fuel)	C	B	A+	C	C	A	A	C	C
Ozone	C	B	C	A	A	A	A	B	A
Paper Stock	C	B	B	B	C	A	B	B	C
Propane	C	A	A	C	B	A	A	C	C
Sea Water	B	B	A	A	B	A	B	A	B
Sea Water +Sulfuric Acid	C	B	C	B	B	A	B	B	C
Soap Solutions	B	A	A	A	A	A	A	A	A
Steam	C	C	C	B+	C	C	C	B	C
Sulfur Dioxide	C	A	C	A+	B	A	C	B	A
Sulfuric Acid (to 50%)	C	B	C	B	B	A	B	B	C
Sulfuric Acid (50 -100%)	C	C	C	B	C	A	C	B	C
Water (ambient)	A	A	A	A	A	A	A	A	A
Water (200°F)	C	C	B	A+	C	B	B	B	C
Water (300°F)	C	C	C	B+	C	C	C	B	C
Water (de-ionized)	A	A	A	A	A	A	A	A	A
Water, white	B	B	A	A	B	A	B	A	B

CHAPTER 11

MAINTENANCE OF CONTROL VALVES

J. A. Parker and C. R. Thornton

E. I. Du Pont de Nemours & Co.
Wilmington, Delaware

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INTRODUCTION

The diaphragm control valve is very often the most costly element in a control loop. As a part of the process stream, it is also the element most likely to cause process down-time in the event of malfunction. These reasons alone provide the justification for a conscientious control valve maintenance program.

Control valve maintenance is generally divided into two categories. The first is preventive maintenance, which involves the installation precautions taken and the inline follow-up of the valve's performance and environmental deterioration while in service. The second is the overhaul of a control valve when it is no longer able to perform satisfactorily. The latter phase is generally a shop repair, but in some industries, it is performed in the pipeline, under controlled shut-down conditions.

Control valve manufacturers provide detailed installation and overhaul instructions for every type of control valve they produce. This chapter is not intended to replace these manufacturers' instructions. Instead, it is intended to impart a general knowledge of control valve maintenance requirements.

PREVENTIVE MAINTENANCE

Preventive maintenance is a term used to identify a program of scheduled overhaul, prior to failure. The schedule is usually based on previous knowledge of performance in a given service and is such, that a maximum safe in-time is established to minimize the chance of unexpected breakdown. Repair effected under such a program for control valves is a shop overhaul, and will be described later in this chapter. Because of this, the term *preventive maintenance* is being used here to describe the work, which should be done during installation and operation to prevent the untimely breakdown of a control valve, ahead of schedule.

Prior to Start-Up

The preventive maintenance of a properly specified control valve begins at the time of installation. While the body and connections of a control valve are indeed rugged structures, they are not intended to be the means of aligning improperly fitted pipe. Care must be taken to insure that any stresses caused by improper pipe alignment are relieved elsewhere in the pipe system. Pipeline stresses on a control valve may result in misalignment of the stem, guide, and seat system of the valve and, in the case of a split body valve, may cause separation of body flanges. This may cause hysteresis, leak-through in the seat or flanges and possibly packing leakage.

Every effort should be made to install control valves so that their stems travel in a vertical plane with the diaphragm housing above the body. If a valve must be installed with the stem traveling in a horizontal plane, support braces should be provided to support the diaphragm housing. An unsupported valve may have stem misalignment, resulting in unacceptable hysteresis and packing leakage. Control valves should be located so they are accessible for periodic inspection and adjustment. Access ladders and platforms may be used for this purpose.

Prior to start-up, it should be determined if the control valve is to be the collection point for line trash such as rust, weld slag, and other foreign objects. If so, temporary screens or strainers should be considered for installation immediately upstream. A well-lapped seat may be ruined by one small piece of slag in the valve, when the valve is trying to close. If the process stream may normally contain scale, dirt or other hard foreign matter, the installation of permanent-type strainers or filters should be considered.

Control valve diaphragm housings are not air consumers. They are, however, collectors of liquids and other foreign matter in the air system. A clean, dry, oil-free source of air should be selected to operate the control valve. Valves with positioners or booster relays attached are particularly susceptible to failure when exposed to unclean air systems.

After Start-Up

After start-up, periodic field inspection of the control valve installation should be made. The packing will usually require adjustment after a short period in service and the lubricators will require readjusting. Dusty or dirty conditions may require a plastic or rubber boot to be placed around the stem to protect the packing gland and/or the guides in the bottom of the diaphragm housing. Corrosive fume conditions or process drippings may require additional special protection for the control valve. Plastic bags are often an easy solution to this problem since they may be used to completely enclose a valve without affecting its performance. Installation accessories such as lagging or tracing should be inspected and repaired, because these usually have a major effect on a control valve's performance.

Certainly, it is not possible to describe every installation or in-service potential failure situation in this writing. These more-common situations provide the basis for a good control valve preventive maintenance program and, together with the experience gained in any plant situation, will aid in extending the useful life of a control valve between overhauls.

SHOP OVERHAUL

Decontamination

The control valve that has been removed from the line for overhaul poses a special problem to the maintenance organization. This component has been a part of the process piping and may be contaminated with the process fluid. Certainly, a valve in water, steam, air, inert gas, etc., service is no problem and may be overhauled immediately; however, many fluids used in modern processing are corrosive, explosive, radioactive, or, in other ways, hazardous to personnel or equipment. Such contamination must be removed from all process-wetted parts of the valve before it can enter a modern repair shop and before it can be handled by personnel unfamiliar with such process hazards.

The actual decontamination treatment of the wetted parts will vary, depending on the nature of the process fluid. Many times, only a thorough water-wash or steaming of the parts will suffice, while other contaminants may require acid washing or even a special heat treatment of several hundred degree-hours, as is the case with some heavy metal compounds. The required treatment must be determined by someone familiar with the nature of the contaminant. Facilities for control valve decontamination will vary, depending on the type of contaminants involved. Location of the facilities should be chosen so that cross-contamination of other locations or facilities is avoided. This usually means a site remote from the operation and the repair facilities.

Since decontamination applies to the wetted parts of the control valve, disassembly of the body and bonnet components is required. Facilities for this work must be provided. These always include lifting and transporting equipment for handling valves during disassembly. An air supply is essential to stroke the valves to prevent further seat damage during stem and plug removal. Air is also useful for pneumatic tools and, in some cases, is needed for health reasons such as breathing air or chemical suit purging. The need for weather protection will depend upon the need for uninterrupted use of the decontamination facilities. If weather protection is provided, then good lighting and ventilation become essential. The need for tanks or vats, heating facilities, steam hoses and other washing or treating equipment can be determined only after the nature of the contaminant is known. Consideration should be given to maintaining exclusive tool kits and wearing apparel for use at the decontamination site.

Special attention must be given to the selection of the personnel who will do the decontamination service. Because of the generally dirty nature of the work, decontamination is an undesirable job. It is extremely essential that the employee who is disassembling a control valve for decontamination, thoroughly understands the control valve, as well as the nature of the chemical hazards involved, in order to avoid personal injury. It is also important that the details of construction be understood, to avoid further deterioration or damage to the control valve during disassembly. Such things as whether or not bellows seals are involved, whether the plug and stem are removed from the top or bottom of the body, if the seats are threaded into the body or otherwise retained, will affect the actual disassembly procedure. Many expensive bellows seals, for instance, have been destroyed by unsuspecting mechanics trying to improperly remove them. It is further essential that the location of all blind passages in the body or bonnet be known so that they may be thoroughly cleaned. The precision-finished surfaces, such as the stem, plug facing, seat surfaces, and guide surfaces, must be protected from further damage to minimize the overhaul costs. Skill in the use of special tools, such as packing ring pullers, seat ring extractors or flange spreaders, is essential to prevent misuse and resulting damage to the valve.

When all of the process wetted parts of the control valve are free of contamination, the valve should be reassembled, finger-tight, to assure that no components are lost during transit to the repair shop. Prior to shipment to the shop, the valve should be marked in some obvious manner to indicate that it has been decontaminated.

This procedure aids in preventing a contaminated valve from being delivered to the repair shop and worked on by unsuspecting personnel.

Disassembly of the Valve

The decontaminated control valve should arrive at the repair shop accompanied by a job card which gives as much information as possible about the valve and its service in the process. Figure 1 is an example of a typical shop record card.

No. 17082 C

Deliver to Spot #_____

INSTRUMENT SHOP RECORD - CONTROL VALVES

Bldg. Name & No._____ Inst. Item No._____
Equip No._____ Serial No._____
Vendor_____ DV No._____
Body Size_____ Type No._____
What is Ailment of Valve_____
Repair Requirements:_____
Special Items Required
Flow Check ( )
Press. Test_____Lbs. Water Action Required ATO ( ) ATC ( ) Change Trim ( )
Deadend Press. Test at_____Lbs. With_____
Type Packing Required_____
Stroke Required_____Inches Dry Valve Completely
Operating Conditions:
Material_____ Normal Flow_____
Press._____ Sp. Gr._____
Press. Drop_____ Max. Flow_____
Issued By_____ Mechanic_____
Date Issued_____ Start: Date_____Time_____
Required Date_____Time_____ Finish: Date_____Time_____
Code_____W. O._____

MU-1932-BR1

Figure 1. A TYPICAL JOB SHOP RECORD CARD FOR CONTROL VALVES.
Courtesy E. I. DuPont DeNemours & Co.

The first step in the overhaul of a control valve is to mark the actuator connection orientation, in relation to the body flanges. A complete disassembly is next. This means, that after separating the actuator from the bonnet, and the bonnet from the body, the actuator is to be disassembled completely. The bonnet and packing gland components are separated; the plug and stem, as well as the blind head flange, are removed from the body. An inspection of all components should be made at this time to determine the extent of repair, reconditioning and replacement that is required.

In many cases, all of the exposed surfaces of the valve components are rusty and unfit for repainting. These parts should be descaled before proceeding. Sand blasting equipment is especially good for this. Care must be taken, before descaling, to protect all of the machined surfaces such as flange faces. This protection can be heavy masking tape for sand-blasting, or it may require a special grease coating if acid washes are to be used.

After the descaling has been completed, the need for any machine work should be determined. The body and trim will most likely require the services of a machinist. All of the flange surfaces should be inspected for damage that would prevent proper gasket seating. Generally, such damage will be in the form of scratches or grooves across the flange face, from the process side to the outside.

Seat rings that are screwed into the body should be given special attention to determine if replacement is necessary. Wire drawing, corrosion and erosion are usually the reason for replacing seat rings, other than to change the capacity of the valve. Small pitting and worn surfaces are usually repaired by machining.

Many manufacturers machine a 60° bevel on the seating edge of the seat ring and a 65° bevel on the seating edge of the plug, as shown in Figure 2. This causes the actual seat contact to be at the top of the bore of the seat ring. This arrangement is satisfactory for a new control valve; however, a control valve that is in need of repair will often have the bore of the seat ring, worn non-uniformly. Restoring concentricity would require enlarging the bore of the seat ring, which will further increase the capacity of the valve, as well as alter its flow characteristic.

One solution to this problem, other than discarding an otherwise reusable seat ring, is to machine a new surface on the face of the seat ring at a 60° bevel, and to change the bevel on the plug from 65° to 59°, as shown in Figure 3. This causes the seating contact to change to the top of the seat ring bevel, which can be re-

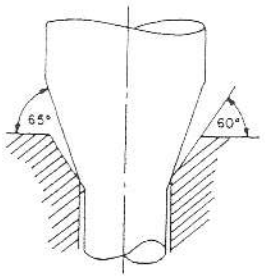


Figure 2. MANUFACTURER'S BEVEL OF 60° ON THE SEAT RING AND 65° ON THE PLUG.

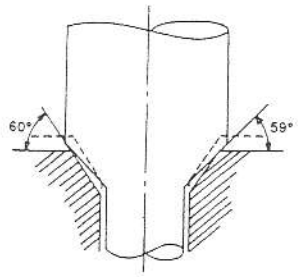


Figure 3. RE-MACHINED SURFACES OF A REPAIRED VALVE, WITH BEVEL OF 60° ON THE SEAT RING AND 59° ON THE PLUG.

machined uniformly without further affecting the valve's capacity or the characteristic of the plug-seat ring contact.

If threaded seat rings are to be replaced, this should be done before any machine work is done on the body. All too often, forces required to tighten a new seat ring will cause distortion of the body. Concentricity must be restored or the valve will never close tightly. Restoration of seat concentricity is best accomplished by using a lathe.

The removal and replacement of threaded seat rings require special tool sets. Most control valve manufacturers produce sets to fit their valves.

Threaded seat rings very often require some type of thread sealing compound to prevent leakage through the threaded area. Not only will such leakage prevent tight shut-off of the flow, but the resulting erosion and wear will also cause irreparable damage to the valve body.

Whenever possible, the valve manufacturer should be consulted for recommendations of sealing compounds for a particular service. If the manufacturer's advice is not available, someone familiar with the process chemistry should be consulted before applying any of the commercially available sealing compounds.

When the necessary machining has been completed, the components are ready for painting. Paint specifications for a control valve should be the same as that for other ing will prevent the paint from being applied to gasket surfaces.

Valve plugs, stems, and seat rings represent some of the most precisely machined parts in the field of instrumentation. The finish on control valve stems is so fine that replacement with a factory-supplied part is the only practical solution to a marred stem surface. This high polish is required to give the best possible packing seal with a minimum amount of hysteresis. The plug and seat ring, of course, have to be perfectly matched, first through precise machining and finally through hand grinding.

Grinding or "lapping" the plug and seat of a control valve requires more skill and knowledge of the craftsman than any other phase of the overhaul, except perhaps, any machining that may be required. It must first be determined that the faults in the seating surfaces can be economically corrected by lapping rather than requiring replacement or remachining, if neither has been done. Alignment of the plug to the seat is absolutely essential for uniform seating. This generally requires that all guides be in place before the lapping operation begins.

Most of the commercially available valve grinding compound sets, which contain a coarse and a fine compound, are satisfactory for control valve lapping.

The coarse compound should be used until a fine, continuous line is produced around the seating edges of the plug and

the coarse compound from the valve, the fine compound should be used to polish this seating seal line.

When the lapping operation has been completed, it is absolutely essential that all of the grinding compound be cleaned from the valve.

Satisfactory flow shut-off and smooth operation of the plug and stem are dependent on the alignment of the plug in the seat ring. Guides are provided to assure excellent alignment. Some valves have guides both below and above the seat ring while others are simply top guided. These guides, in order to accomplish their mission, must be machined to a tolerance close to that of the guide shoulder on the plug-stem assembly. Any corrosion or wear that destroys the close fit renders the guide useless.

Some guides are threaded into place while others are pressed into place. Replacement of threaded guides is straightforward, but pressed guides must be forcibly removed. Various methods work well for this operation. Press-type removal equipment is preferable, if available. Where press equipment is not available, differential heat methods, for expansion of the guide holder and contraction of the guide, will work. This latter method must be used with care in order that permanent damage not be done to the guide holder. New guides can usually be pressed in place with a bench vise.

Disassembly of the Actuator

The control valve actuator, after having been completely disassembled, should be carefully inspected, component by component. Cast steel yokes should be examined for any signs of cracking since all of the valve stroking force is transmitted through this component. Corrosion that either weakens or causes misfitting of components is cause for parts replacement. The diaphragm should be examined for any signs of cracking, hardening or abrasion to indicate possible future failure.

The same surface preparation and painting procedure used on the body components should be applied to the actuator components.

All threaded parts should have the threads dressed for smooth assembly and adjustment. It is generally more profitable to discard all diaphragm case bolts and replace them with new ones.

Reverse-acting actuators should have any stem seals replaced with new ones.

Reassembly of the Valve and Actuator

Reassembly of the control valve body points of concern are that gaskets are used of proper size and material and properly centered; and, that bolts or studs are used of proper size, length and material.

When tightening the bonnet and the blind-head flanges to the body, it is important to tighten diagonally opposite bolts evenly, to assure an even loading on the gasket, thus preventing flange leakage. The valve manufacturer's recommendations for proper torque loading of the particular bolts or studs should be adhered to when tightening the body bolts or studs. Over-tightening will cause excessive elongation, resulting in a weakened flange joint.

The depth of the packing gland should be measured to determine the proper number of packing rings. Preformed packing rings should be used whenever possible. These are commercially available in most materials and sizes. Their uniformity assures the best possible seal in the gland. If a lantern gland is used, enough rings of packing should be placed beneath it, to assure that it will mate with the lubrication port in the gland housing. Packing rings should be inserted one at a time and each should be lightly tapped into place. *Do not pound packing rings.* Pounding distorts the rings and causes them to lose their resilience. When the gland has been filled with the predetermined number of packing rings, the follower and yoke should be installed, finger tight, for hysteresis testing of the assembled valve. It should be noted that special packings, such as Chevron sets, and special spring-loaded glands would be assembled differently. The manufacturer's instructions for these special packings must be followed if a good gland seal is to be expected.

Reassembly of the actuator should include the application of a suitable lubricant to all friction points. These include the stem where it passes through guides and seals, the spring thrust bearings and retainers, and the spring-adjusting nuts.

Test and Adjustment of Reassembled Valve and Actuator

When the reassembly has been completed, the diaphragm case should be leak-tested. This is most easily accomplished by loading the diaphragm chamber with 45 psig of air *(do not exceed the manufacturer's rating for the case)* and leak-testing around the flange edge with a soap solution. A hysteresis test of the actuator should be made before it is mated with the body assembly, to determine that there is no binding of com-

The hysteresis test may be made by attaching a clamp-on arm to the exposed, movable portion of the stem and by attaching a micrometer gauge to the unmovable yoke or spring barrel. The micrometer gauge must, of course, have as much travel capacity as the stem. After zeroing the micrometer gauge to the at-rest position of the stem, record the air-loading pressure required to stroke the actuator at every 10% increment of its travel. Then, while returning the stem to the at-rest position, record the air-loading at every 10% increment of travel. Compare the recorded air-loading pressures in the two directions, for each increment of travel. The differential pressure between the two readings at each increment, should not exceed 0.1 psi. Differential pressures greater than 0.1 psi, indicate an intolerable friction condition, within the actuator assembly, which must be corrected, even if complete disassembly and reassembly of the actuator is required.

The assembled and tested actuator may be mated to the valve body with the hysteresis test equipment attached, because a final-assembly hysteresis test will be made later.

When mating the actuator to the body, it is important that the air connections on the actuator be oriented to the body flanges, exactly as they were before disassembly. This will allow the valve to be reinstalled into the pipeline, without requiring any alterations to the field tubing or piping assembly.

The full-travel loading pressure, required for any control valve, is a function of the diaphragm effective area versus the stem loading (the accumulation of the spring loading and the plug area times the shut-off differential pressure across the plug). Rarely will these factors result in an ideal 15 psi (30 psi for a 6 to 30 psi control system) full-travel, diaphragm pressure.

If the full travel diaphragm pressure requirement is less than 12 psi or more than 18 psi for a 3 to 15 psi control system (24 psi to 36 psi for a 6 to 30 psi control system), and if the valve is not normally equipped with a valve positioner, then consideration should be given to changing either the size of the actuator or the size of the actuator spring.

When the stroke and spring compression have been properly adjusted, a final hysteresis check should be made on the actuator. This is done with the packing loose and with no pressure differential across the valve. This test is to determine that proper alignment of all the components has been obtained.

The test should be run on the complete assembly, just as it was run on the actuator. The hysteresis differential pressure of a properly assembled control valve should not exceed 0.25 psi under these testing conditions. Higher differential pressures indicate that binding is occurring somewhere within the inner valve mechanism. The degree of binding is indicated by the magnitude of the loading-pressure differentials obtained during the test.

Upon completion of a satisfactory hysteresis test, the valve body should be filled with a test fluid with the stem at mid-travel. The test fluid pressure within the body should be raised to the maximum process operating pressure and the packing should be tightened only enough to stop leakage. The stem should be stroked continuously, while the packing is being tightened, to properly seat the packing and to prevent stem binding.

Note: Special packings previously mentioned should be adjusted according to the manufacturer's recommendations. While the packing is being adjusted, the body should be examined for external leaks.

Metal-to-metal seated diaphragm control valves are *not* tight shut-off devices. They will all leak through the seat to some degree. Most manufacturers design and build to a shut-off condition that will allow no more than 0.1% of maximum flow capacity leakage at the shut-off position for a double-seated valve, and will allow no more than 0.01% of maximum flow capacity leakage for a single-seated valve. These leak rates can be duplicated if reasonable care is exercised during the overhaul of a control valve.

Upon the completion of leak testing, any accessories that were a part of the valve assembly should be reconditioned and installed on the valve.

Preparation for Shipment

Preparation of the reconditioned control valve for shipment will vary, depending upon whether the valve is to be reinstalled in the process line from which it was removed or whether it is to be put into storage as a standby spare. To prevent foreign matter from entering, all air connections to the valve should be plugged and the flange openings should be covered. If the valve is to be reinstalled immediately, heavy masking tape will suffice. If the valve is to go to storage, flange guards of wood or heavy cardboard should be fastened to the flange faces. Any protruding accessories such as lubricators or positioner gauges, that can be broken off easily, should be removed from their receptacles and packaged separately to accompany the valve for installation at the field location.

Control valves that are to be stored as standby spares require a clean, dry location for storage. If such facilities are not available, the valve should be completely enveloped in a moisture-proof bag or similar container before storage. The storage area should be such that the valve will not be damaged by other work being performed in the area.

OVERHAUL CHECK-LIST

The step-by-step procedure for control valve overhaul is lengthy. In order to assure that no phase of the overhaul has been overlooked, a careful check-out procedure should be established. The following is a step-by-step check list that one large repair shop foreman has prepared to aid his personnel.

SHOP OVERHAUL DIAPHRAGM VALVES, 1/2 INCH TO 3 INCHES.

Step-by-step procedure and check-list.

1. Obtain job ticket from foreman.
2. Transport valve to work bench or to wash stand if necessary to clean.
3. Install on either control valve vise of repair bench or in standard vise to facilitate disassembly.
4. Mark actuator and body orientation and remove actuator from body. Remove positioner if necessary.
5. Disassemble body by removing bonnet flange, packing, stem and blind flange.
6. Secure actuator, mark for orientation of air connection, back-off the spring adjuster, remove the cap screws from the case assembly with an impact wrench, discard cap screws and nuts.
7. Remove diaphragm, diaphragm plate and actuator stem as a unit.
8. Remove spring adjuster screw nut, spring adaptor screw assembly, spring adjuster, spring seat and spring.
9. Examine diaphragm case assembly and spring barrel case assembly interior; steam-clean if necessary.
10. Remove body from vise; transport body, body bonnets, diaphragm case assembly and spring barrel to sand-blast facility; do general clean-up.
11. Spray-paint body and body components with one coat of silver paint.
12. Spray-paint the diaphragm case assembly and spring barrel exterior with one coat of gray primer. Paint interior of case assembly and spring barrel with one coat of gray primer, if needed. Spray-paint the case and barrel exterior with one coat of orange.
13. Inspect the diaphragm for holes, cracks or excessive wear; replace if needed. Replace all stem seals in reverse actuators; lightly grease the "O" ring before installation.
14. Dress threads on all actuator components; lightly grease.
15. Move parts from the spray booth to the bench. Install the body in a vise.
16. Inspect valve body, flange, ring and seat ring; replace on machine as necessary. Lightly grind with compound to lap-in the seat and ring.
17. Chase the body studs with a die; replace studs as needed; tap-out the stud holes at this time; assemble the flanges to the body; line-up the reference marks for proper orientation.
18. Repack with approved packing; make-up the gland finger tight.
19. Assemble the parts of the actuator barrel; install the diaphragm case, using new cap screws and nuts, and tighten with an impact wrench. Apply air pressure and check for air leaks.
20. Adjust the spring range and stroke of the actuator by applying proper air pressure values.
21. With sufficient air on the actuator to put it at approximately mid-stroke, attach the actuator to the valve body and either bolt or clamp it with a yoke-holding device.
22. Install the actuator and the plug stem connector finger tight; either apply or remove air to adjust the connector for proper seating of the plug. Tighten the jam nuts or the connector. Adjust the travel indicator.
23. Body-test and dead-end test the valve in accordance with information contained on the work card.
24. If the positioner has been removed, reinstall it by first chasing the threads of the positioner bracket or the boss that are on the actuator.
25. Mount the positioner and the positioner arm, and fasten the turnbuckle assembly to the positioner arm.
26. Switch the bypass valve to the instrument and apply sufficient air pressure to the actuator to obtain a mid-stroke position.

27. Adjust the positioner arm turnbuckle to make the front lever parallel to the rear lever of the positioner.
28. Loosen the slider clamp screw and adjust the slider to the approximate actual (in-service) valve travel.
29. Turn the bypass valve back to the positioner and alternately adjust the turnbuckle length and operating spring seat until the valve is at mid-stroke when the applied pressure is at mid-range of the control instrument pressure, in accordance with the information contained on the work card.
30. Check the positioner and the valve for full stroke performance.
31. Complete the necessary paper work in accordance with shop practice; stencil as required.
32. Remove the valve from the vise; mask the valve openings; install proper tag; and prepare it for transport.

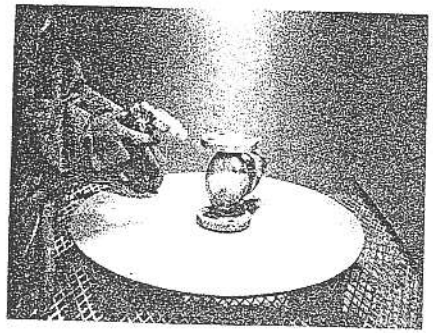


Figure 4. A SMALL SAND-BLAST BOOTH.

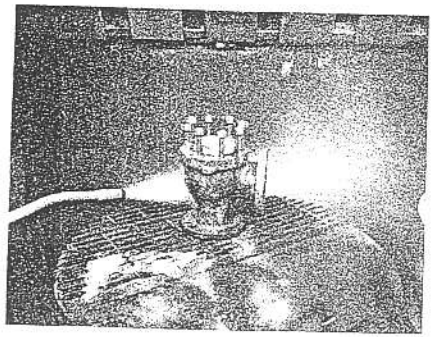
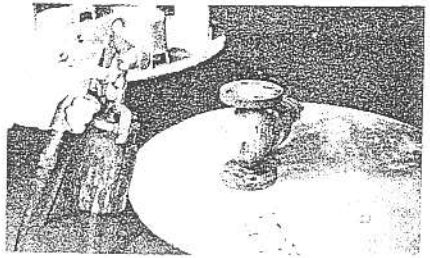


Figure 5. A STEAM-SOLVENT CLEANING



OVERHAUL FACILITIES

Continuous control valve overhaul work requires a work area that is specifically adapted to the tasks. Real efficiency and cost-savings can be realized from the availability of such facilities as a small sand-blast booth, as shown in Figure 4, for surface preparation work. A steam-solvent cleaning booth, similar to that shown in Figure 5, is particularly useful for degreasing and grit removal after sand-blasting. Suitable spray-paint facilities, as shown in Figure 6, provide for fast repainting, with a minimum of fume and dirt contamination to the adjacent work areas.

Proper lifting equipment is particularly useful in handling larger control valves with a minimum number of personnel, as shown in Figure 7.

A holding device, such as that shown in Figure 8, is particularly useful in reassembly work. The valve body can be mounted on the large universal flange, which in turn, can be rotated 360° by turning the crank handle. The universal flange can be piped to a pneumatically-driven pump, which will pump test fluid to the valve body for seat leakage and hydrostatic testing. The valve need not be dismantled from the holding device until the final assembly and test has been completed. A set of blind and screw flanges is a useful accessory to the holding device. The flanges illustrated in Figure 9 are in sizes of one-half inch through six inches, all of 600-pound rating. The bolt holes have

been elongated to the respective 150-pound bolt circle. This allows one set of flanges to be used for flange ratings of from 150 pounds through 600 pounds while being nonstandard enough to discourage their removal for other uses.

Proper seat ring removal tools, as shown in Figure 10, are more than justified by the amount of time that can be saved in seat-ring removal. An additional bonus can be realized by the number of seat rings that are removed undamaged and by avoiding injury to personnel, that could result from using improper rigs.

The importance of a machine tool facility, similar to that shown in Figure 11, cannot be overemphasized for repairing otherwise unusable valve trim.

A test panel, adjacent to the valve-holding device, can save a large amount of time during the final checkout of the control valve. The panel shown in Figure 12 provides gages and regulators for the valve positioner air supply and the pneumatic pump-loading air supply. A gage for indicating hydrotest pressures and a manometer and regulator for the instrument signal pressures are also included.

SUMMARY

A well-planned maintenance program, with trained personnel who have proper job aids, will result in overhauled, diaphragm control valves that will have an inline use-period before breakdown that can be equal to that of new valves, but at a fraction of their cost.

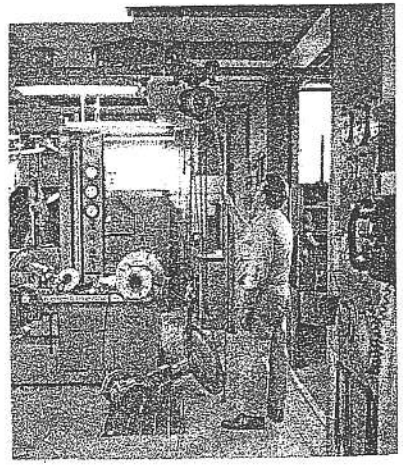


Figure 7. LIFTING EQUIPMENT FOR LARGE CONTROL VALVES.

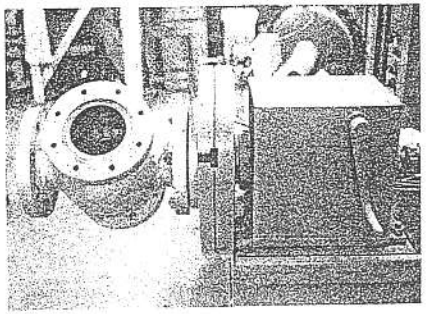


Figure 8. A CONTROL VALVE HOLDING DEVICE.

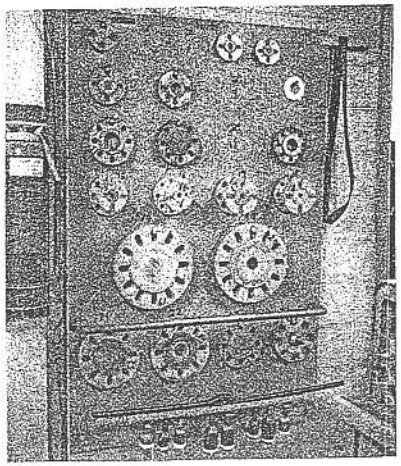


Figure 9. AN ACCESSORY SET OF UNIVERSAL BLIND AND SCREW FLANGES.

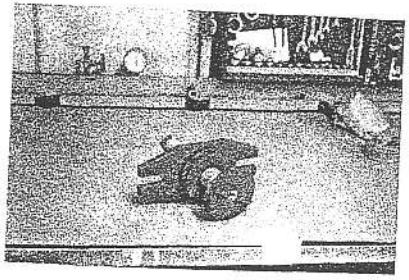


Figure 10. SEAT-RING REMOVAL TOOLS.

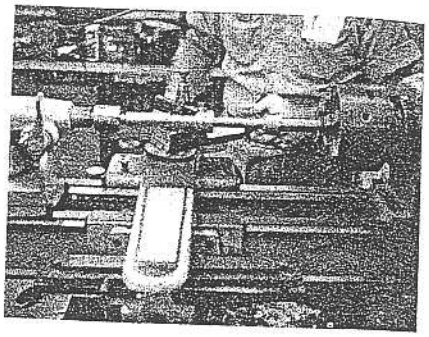


Figure 11. MACHINE TOOL FACILITY FOR REPAIRING VALVE TRIM.

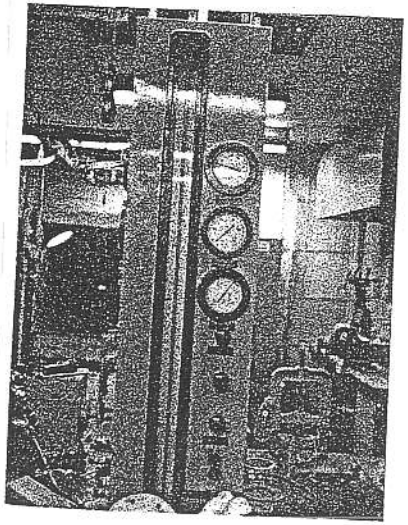


Figure 12. AN INSTRUMENT PANEL FOR CHECK-OUT DURING OVERHAUL.

CHAPTER 12

INSTALLATION PRACTICES

S. F. Luna

General Atomic Co.
San Diego, California

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INTRODUCTION

The engineering technique for selecting the best valve for the job has been treated in depth in the preceding chapters. The fruits of this labor may not be realized unless proper installation practices are followed.

There is but one major factor that can motivate the not so glamorous job of engineering the installation and that factor, of course, is *money*.

1. If the valve to be installed should fail, resulting in a system shutdown for say 10 minutes, or one hour, or four hours, or more, how much money in production and maintenance will this cost; $100, $5,000, or $ _____?
2. If a new plant is being built and the piping installation drawings are incorrect, resulting in an incorrectly piped valve installation, what would it cost to correct the piping? Consider the material cost, cost of labor, drawing changes, overhead, profit, delay charges, etc.
3. Consider the plant start-up situation with the control room full of people; customers, test engineers, standby skilled help, vendor's field personnel, etc., all ready to start-up. If after two to three days of sequentially bringing the plant to a major progress point, the control valve in the next sequence would fail to pass fluid, because it is plugged up with rags, welding rods, left-overs from lunch, etc.; what would this cost? It has been reported that on certain installations the cost during this stage, involving many people, is $2,000 per hour.

The above emphasizes the likelihood of wasting large sums of money, because of the failure to follow good installation practices. There are many other reasons for giving this phase of control valve engineering proper attention—longer valve life, less maintenance, better system performance, etc., but they all boil down to *"money in the hopper, if the installation is proper."*

Good installation practices start with the initial engineering practices, particularly in the preparation of the piping drawings, piping standards and procedures.

Most firms responsible for the installation of control systems have developed control valve installation piping standards and procedures which are either used with specific piping drawings or are used to produce specific piping drawings.

This subject has been covered extensively material, suitable for this handbook, which has, for the most part, appeared in the literature.

The considerations for valve installations may be classified into the following six areas, each of which is discussed in the following sections.

1. The safety of people and equipment in all stages of the valve's life; that is, installation, test, operation, and maintenance.
2. The performance of the control valve as a final control element in a control loop.
3. The requirements of the installation to permit intelligent manual operation of the system, if necessary.
4. The accessibility of the control valve for maintenance.
5. The cost of installation, operation and maintenance.
6. The protection of the valve from either mechanical or environmental forces.

SAFETY CONSIDERATIONS

The following problem areas and solutions must be considered in the valve piping design and installation:

Valves may leak, either initially or through use, at the packing glands, flange gaskets, or at pin holes formed at valve faults. It should be assumed that a leak will develop. What damage can conceivably occur, considering the type of fluid passing through the valve, at the most severe operating conditions (temperature and pressure)? Will a corrosive fluid collect in an electrical wire-way? Will a combustible fluid drop onto a hot tank? Will a hot stream of caustic solution injure an operator?

After a shut-down, valves may hold system pressure for an extended period of time. Means must be provided to safely reduce this pressure, if the valve is to be either opened or removed for maintenance. Likewise, means must be provided for draining trapped damaging fluids.

In the design of the valving system, the following should be considered, to reduce control valve pressure:

1. The design tightness of the control valve. Single seated valves can be furnished to provide a leak rate of less than one drop per hour. Double-seated valves rarely close tightly on both seats; therefore, consideration must be given to possible replacement with a single-seated valve.
2. The volume of trapped fluid between the stop valves. Obviously a large volume of trapped fluid at a high pressure can represent a sizable force, who is careless in the manner in which he relieves control valve pressure.
3. The nature of the fluid. Is it a gas, clean fluid, or fluid with suspended solids? What is the pH of the fluid?

For good design one should assume that it won't be possible to avoid shut-down trapping high pressure fluids; therefore, it is recommended that bleed and/or drain valves be installed on either side of the control valve, if the potential energy of the trapped fluid is dangerously high.

For valves with large trapped volumes at elevated pressures, it may be wise to consider double vent and bleed valves. Figure 1 shows such an arrangement. This arrangement traps small volumes between the two valves and enables pressure relief of the system in steps, by repeated pressure build-up and let-down of the volume between the two drain and/or bleed valves. Note that the control valve must be in open position during this procedure.

If the fluid to be relieved is a hazardous gas, the vent lines must be piped to a safe location. If the gas is not combustible, it still must be piped or directed to avoid a blast of gas carrying rust or other projectiles that can injure the person handling the vent valve.

A let-down valve should be selected that will restrict the flow, thus protecting the careless operator who fails to follow the safe practice of slowly opening the let-down valves. Such restricted let-down valves can plug up easily if the fluid contains suspended solids. It may, therefore, be necessary to let-down pressure from valves located on top of the pipe and then complete the draining operation with larger-sized drain valves. In this case it is very important to train personnel on the safe manipulation of the let-down valves. If the fluid to be drained is a dangerous liquid, the piping system designer should consider piping it to a safe location. During plant operation, valves may be very hot; therefore, provisions must be made to prevent heat damage to the personnel.

Valves handling radioactive fluids require extreme methods of care. The rules and regulations covering this area are very extensive and not covered in this text.

PERFORMANCE

The prime function of the control valve must be considered in the design of the valve installation to avoid imposing dynamic conditions that may detract from the factors used in selecting the valve. The process control engineer is generally very careful in selecting a control valve that will have a specified capacity and a specified flow rate vs. valve lift characteristic. He will normally take into consideration the pressure drop in the piping system, when sizing the valve.

The designer must design the piping system to be consistent with the process con-

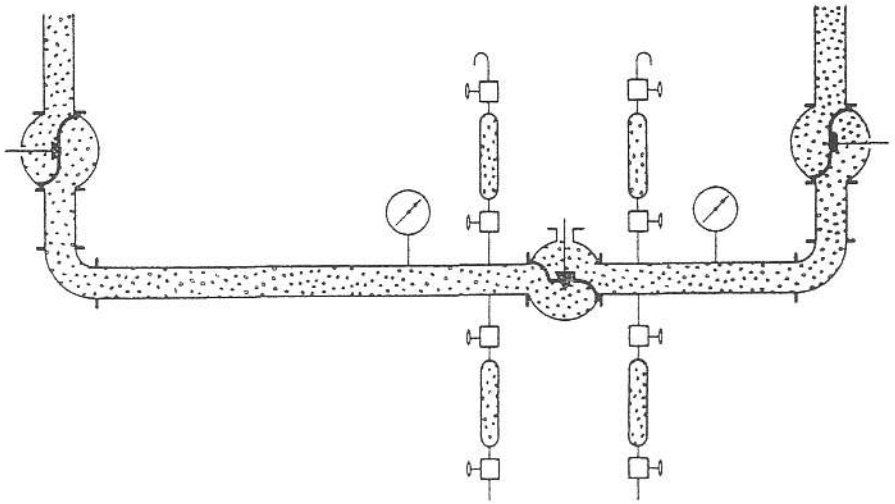


Figure 1. CONTROL VALVE INSTALLATION WITH VENT OR DRAIN CONNECTIONS.

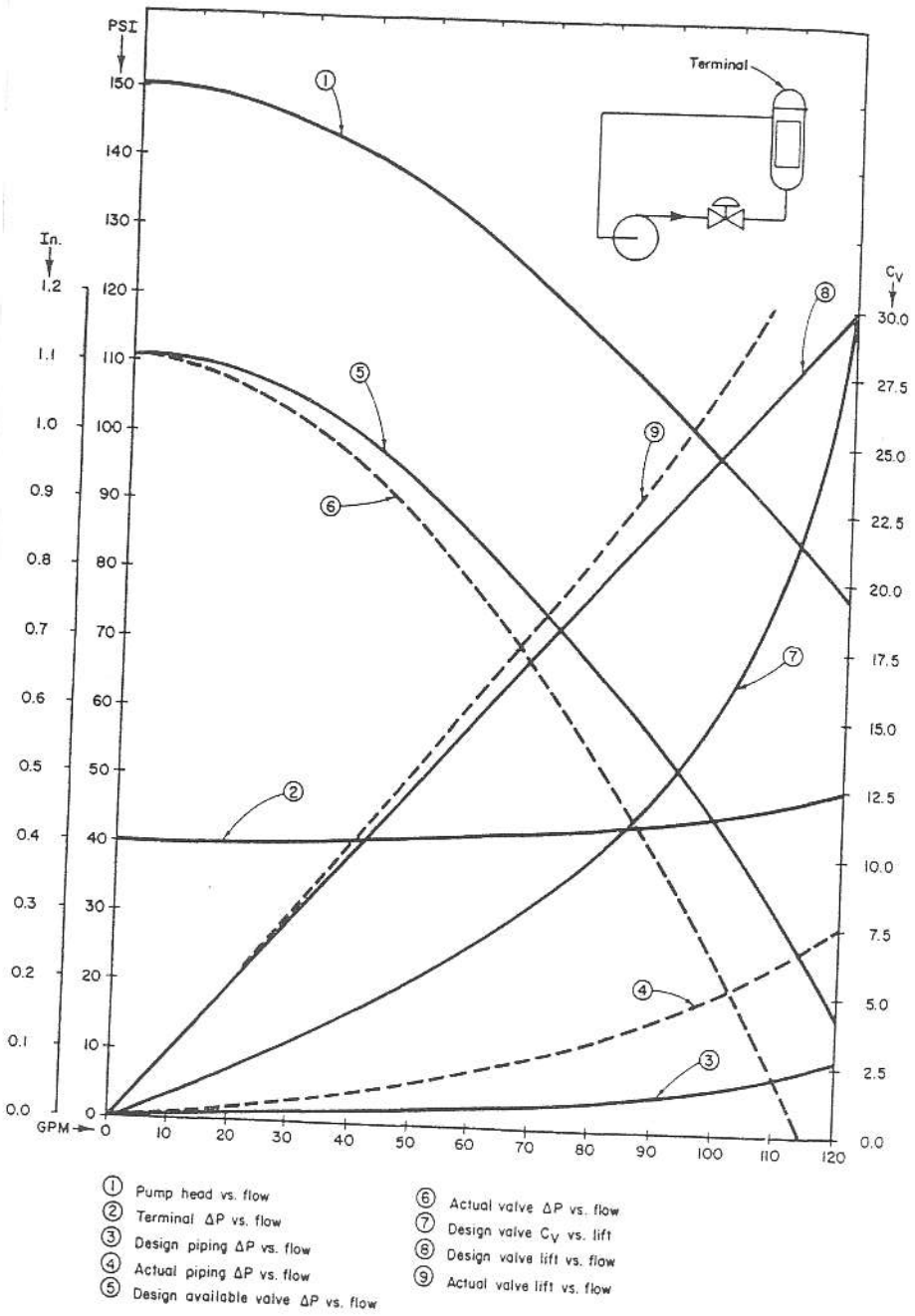


Figure 2. SYSTEM PERFORMANCE CURVES.

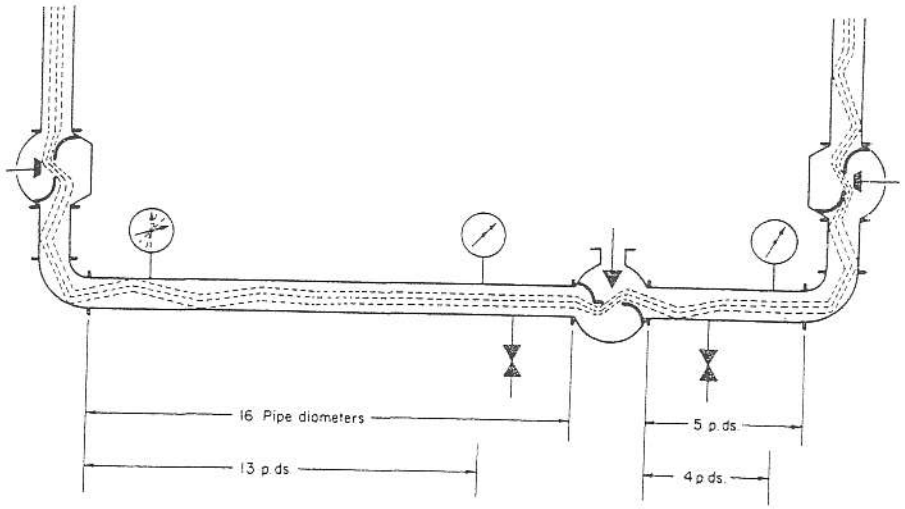


Figure 3. TYPICAL SATISFACTORY ARRANGEMENT OF UPSTREAM AND DOWNSTREAM PIPING.

trol engineer's calculations, or vice versa. Figure 2 shows a method of plotting system pressures to arrive at available pressure drop vs. valve lift. A similar plot will be used by the valve manufacturer to accurately design the valve plug. The piping design engineer should review such plots in designing the piping system. The design friction loss, through the lengths of pipes and fittings selected, can be made consistent with such plots. This calculation should include the friction drop through the fully opened control valve as part of the piping system. The characteristic pump curve is generally available from the pump manufacturer. The terminal pressure drop vs. flow through the process vessel may be more difficult to obtain.

It can be seen from Figure 2 that valve performance—capacity and flow vs. valve lift, will be altered if the actual piping system consumes more pressure drop (curve 4) than that considered in the valve sizing calculation.

In designing the inlet and exit piping at a control valve, a conservative approach is to consider the control valve as a variable orifice; thus, the piping recommendations used to pipe orifice plate assemblies can be applied. The following considerations are recommended:

1. Allow the maximum number of feet of straight run inlet piping that is consistent with other piping demands. A good rule of thumb is 10 to 20 pipe diameters. This is easy to obtain in small size lines,

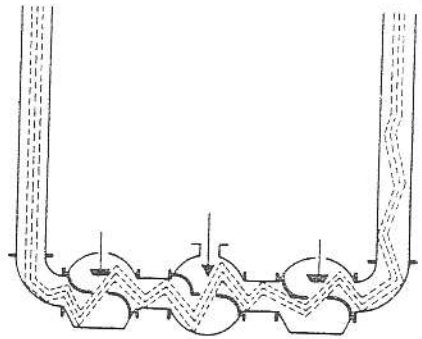


Figure 4. A LESS SATISFACTORY ARRANGEMENT OF UPSTREAM AND DOWNSTREAM PIPING.

but may be impossible to obtain in line sizes larger than nominal 5 inches. Nonetheless the practice of allowing for a maximum straight inlet run will allow better valve performance.
2. Where possible, allow a straight run of 3 to 5 pipe diameters for the outlet piping.
3. The straight run on the inlet to the valve will permit the fluid stream to enter the valve at a steady inlet pressure. Thus, for every position taken by the valve, or in other words, for every new orifice opening, a steady inlet pressure condi-

tion will insure a steady and repeatable flow rate. Figure 3 represents a good flow approach and Figure 4 shows a poor flow approach.

4. It will be noted that the location of the pressure gages in Figure 3 allows a steady, true, pressure head and not pressure head plus and minus the velocity heads due to the non-uniform flow pattern. The accurate measurements of upstream and downstream pressure are particularly important in trouble-shooting fluid system problems. An accurate measurement of control valve pressure drop, plus valve position, can verify if the valve is performing to its design requirements, or if there is internal trouble requiring a valve shut-down and repair.
5. The location of an upstream or downstream tap for a pressure regulating valve should definitely follow the above recommendations to assure sensing accurate and steady pressures undisturbed by perturbations of pipe fittings.

MANUAL OPERATION CONSIDERATIONS

In locating the control valve, consideration should be given to the following factors that will facilitate good manual operation, if required:

1. Is the manually operated valve located where an operator can conveniently manipulate it and at the same time be able to read the indicators displaying the plant parameter he is trying to control?
2. Can the operator see the level gage glass on a tank, if he is manually controlling liquid level?
3. Can the operator determine how much of a change he has made on the valve stem by observing line pressure gages or the valve stem motion indicator?
4. Are indicators of other parameters available so that the operator can anticipate changes? For example, if he is controlling boiler feedwater flow, does he have visual access to a steam pressure gage so he can anticipate the need for more or less feed water if the steam pressure or flow gauge indicates a decrease or increase in steam flow?
5. If the manual operation is performed by the use of a by-pass valve, it is desirable that this valve be selected to have a valve lift vs. flow characteristic that is similar to that of the control valve.

ACCESSIBILITY

There are control valves that have been in operation for 10 to 20 or more years and have never been subject to dissasembly and repair. In fact, some of these valves have been in operation for such a long time that the type of inner valve is unknown. These valves are generally found in service on low pressure systems with non corrosive fluids.

In today's process plants, control valves are required to handle a wide variety of fluids at ever increasing pressures and temperatures; therefore, the piping designer can assume that the control valve will have to be either serviced in-place, or removed, on a routine schedule. The cost for this servicing will depend largely upon the attention given to valve accessibility.

The piping designer should follow a manufacturer's outline drawing that shows the clearances required in the following areas:

1. Above the valve for the removal of the top works with stem and plug as an assembly.
2. Below the valve for removal of the bottom flange and removal of the plug and stem assembly, if required.
3. On the side of the valve for removal of accessories such as manual operators, electric motor drives, solenoid valves, valve positioners, etc.
4. At the body flanges for removal of flange bolts. This dimension is seldom furnished, thus the piping designer must consider this in designing reducers to connect control valves to inlet and outlet piping. (See Figure 18).

Good manufacturer's drawings will permit the preparation of piping drawings that call out clearance requirements. (Figure 20 may be used as a guide in lieu of manufacturer's drawings.) This in itself is not enough since the heating and ventilating designer and the electrical wire routing designer may not avail themselves of the piping drawings. Costly re-routing of heating and ventilating ducts or wire-ways can be avoided if the piping designer advises other group designers of his valve clearance requirements.

CONTROL VALVE MANIFOLDS

The piping arrangement selected for the installation of a control valve involves the consideration of many factors which are tied to the fundamental objective, which is: *Excellent and dependable control valve performance at the minimum overall cost over its design life.*

The designer must, therefore, address himself to the many considerations given in the following check list:

1. How should the inlet and outlet piping to the valve be designed.
2. How much straight run should be used?
3. What line size should be used?
4. Are reducers needed? Does the down stream piping need to be enlarged to allow for expansion of the fluid?
5. Are block valves needed? If so, what size and type?
6. Is a by-pass valve required? If so, what size and type?
7. Where should the valve be located in the line relative to sensing elements?
8. At what elevation shall the control valve be located to avoid obstruction and to provide access.
9. Will the valve be accessible for maintenance and for manual operation? Will the valve stem motion indicator be visible during normal operation?
10. Will the valve be near equipment or indicators that are required for manual operation?
11. Will the valve present a hazard to people or equipment in any phase of its operation?
12. Are vent or drain valves required? If so, how many, what size and type, and where shall they be located? Should these be piped to a safe location?
13. Are pressure gauges required? If so, how many, where should they be located and how should they be piped?
14. Are pressure relieving devices required? If so, what size and where should they be located?
15. Are line strainers needed? If so, for initial plant construction or for re-insertion after a plant shutdown and wet layup?
16. What type of fittings will be needed?
17. What materials of construction should be used?
18. What codes must the installation meet?
19. Will it be necessary to insulate the installation?
20. Will it be necessary to heat trace the installation?
21. If the valve cannot be installed in a vertical position, what orientation should be selected?
22. How should the control valve be supported?
23. What are the requirements for the pneumatic or hydraulic tubing and auxiliaries?
24. Are means available to isolate the pneumatic or hydraulic tubing and auxiliaries and are they independently supported? Electrical hardware associated with electrical driven control valves must be given the same consideration.
25. Are steam traps required? If so, where should they be located?
26. Will pressure taps be required — if so, where should they be located and what size and type of tap valve should be used?
27. Will the valve be subject to damage by moving vehicles? If so, how should it be protected?
28. What about protection against seismic forces or fire?

The design of a control valve manifold starts with an examination of the overall piping design. Piping is generally selected to meet pressure codes, corrosion and wear requirements, noise level requirements and plant expansion requirements. The following fluid velocities are typical: liquids - 5 fps; steam - 100-125 fps; and gases - 250-400 fps.

The size of the pipe selected is greatly influenced by the cost of material and by the cost of fabrication. A high pressure stainless steel piping system requiring welding, stress relieving, and radiography, may be selected to tolerate higher line velocities. Steam flows greater than 400 fps and feed water flows of 40 fps are reported in the literature for high pressure, high temperature power plants.

In many cases the pipe selected is oversized to allow for future plant expansion; thus, the control valve required for initial plant operation may be one or two sizes smaller than the pipe size into which it must be fitted.

There are two alternatives in selecting the control valve, either:

1. a line-sized control valve with reduced trim (for example, a 4 in. valve body with 2 in. trim); or
2. a minimum-sized control valve (for example, a 2 in. valve body with full sized trim).

The selection is generally determined by a cost analysis considering the difference in cost of the valves, piping and pressure drop requirements related to the pumping cost and current financial considerations.

The selection of a line size valve with reduced trim is recommended unless the cost analysis shows a significant advantage for the minimum size valve. It is common practice in many process industries to automatically select line size control valves for line sizes of 2 in. and

With the process piping fixed and the control valve sized, manifold design can proceed. Fortunately, control valve body dimensions have been standardized. (ISA RP 4.1, "Uniform Face to Face Dimensions for Flanged Control Valve Bodies.")

ISA RP 4.2 "Standard Control Valve Manifold Designs" and API RP 550 "Manual on Installation of Refinery Instruments and Control Systems" can be useful in designing a particular manifold, but will probably require some modification to meet specific requirements. Other manifold arrangements and recommendations are given in a series of diagrams appearing later in this chapter. The specific requirements are first determined by a review of a check list as noted above. Modification of a suitable reference design can then be made accordingly.

It is common practice in many processes to install block valves and by-pass valves. These are installed to facilitate continuous operation. The block valves are used to isolate the control valve and the by-pass valve is used while the control valve is being repaired.

There are exceptions where the added expense of one or more block valves precludes their use. Batch processes may be piped so that one block valve is all that is required to isolate a control valve for repair. Certain critical processes cannot tolerate the inadvertent opening of a by-pass valve or the closing of a block valve.

Most companies have adopted recommended minimum sizes similar to Table 1, because of the cost factor. Typical examples of arrangements of the control valves, block valves and by-pass valves are illustrated throughout the Figures in Appendix I of this chapter.

PLANT INSTALLATION PRACTICES

Good plant installation practices start with the proper handling of the control valve from the time it is received at the site.

It is most important that the personnel who will be handling the control valve at all stages of its installation realize that control valves are precision devices with finished surfaces that range down to 1 - 2 RMS (roughness in micro-inches). Furthermore, all personnel handling the control valves should be made aware of the cost and lead-time required to secure control valves. Specially designed control valves can take from 6-14 months to deliver after order placement. Careless handling of valves may cause considerable delay in plant startup.

The degree of inspection of a control valve upon receipt will vary with the facility and expertise of personnel handling it. The range of care and degree of inspection varies widely from one purchaser to another. It is not uncommon on a new plant construction site to find control valves unboxed and exposed to the elements. At the other extreme the so called "expert", armed with an assembly drawing and a garden variety of tools, will attempt a complete disassembly of the control valve in order to check the dimensions of all the parts. Somewhere between these extremes is the proper degree of receipt inspection.

Many firms provide for thorough inspection, by the purchaser, at the place of manufacture. In this case, the receipt inspection consists of the minimum effort required to determine that the valve was not damaged in shipment.

The following is considered the minimum inspection criteria for factory assembled

Table I. RECOMMENDED MINIMUM BLOCK AND BYPASS VALVE SIZING

(All Sizes in Inches)

Line Size:	½ Valve		¾ Valve		1 Valve		1½ Valve		2 Valve		3 Valve		4 Valve		6 Valve		8 Valve		10 Valve		12 Valve	
Control Valve Size	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass	Block	Bypass
½	½	½	¾	¾	1	1	1½	1½														
¾			¾	¾	1	1	1½	1½	2	2												
1					1	1	1½	1½	2	2	2	2										
1½							1½	1½	2	2	2	2	3	3								
2									2	2	3	3	4	4	4	4						
3											3	3	4	4	6	6	6	6				
4													4	4	6	6	8	8				
6															6	6	8	6	8	8	10	10
8																	8	8	10	8	10	10
10																			10	10	12	10
12																					12	12

control valves:

1. visual examination, using the assembly drawing;
2. hydrostatic test;
3. leakage test;
4. hysteresis check;
5. valve-travel check by the operator;
6. operational check of all accessories limit switches, valve positioners, etc.
7. electrical tests (Megger and Hipot) for electrical devices; and
8. packaging and shipping checks to specifications and procedures.

If these tests are properly performed at the factory the receipt inspection will only require visual examination. If these these tests were not performed at the factory, or if there is reason to suspect problems, these tests should be performed on site, as required.

After the receipt inspection, the control valve should be prepared for shipment either to the plant area for installation, or to storage. The valve body's air or electrical connections should be plugged to keep-out dirt. If the control valve is to be stored for any length of time it should be packed for protection against the elements. Refer to Chapter 11 for a more detailed discussion on preparing a valve for storage.

Summary of Installation Practices

The following detailed items are recommended for consideration in every installation:

1. The air supply used to operate pneumatic control valves should be free from oil and moisture.
2. Follow the manufacturer's recommendations for the hydraulic system to power hydraulically driven control valves.
3. Follow the manufacturer's recommendations for the electronic system for electrically or electro-hydraulically driven control valves.
4. Inlet piping to control valves with small passageways, should be fitted with appropriate filters to eliminate internal valve damage from foreign matter in the piping system.
5. Do not put excessive stress on valve bodies when installing in the system. This is particularly important for split body valves.
6. Before initial start-up and after a maintenance shut-down, install screens ahead of the control valve to collect pipe scale, rust, and other debris. Whenever possible, the piping system should be fitted with a spool piece and flushed out prior to control valve installation.
7. If the valve is to operate in a dusty atmosphere, install a rubber or plastic boot around the stem to protect its polished surface from damage.
8. Be sure to follow all of the manufacturer's instructions for adjustments and switch positions for the accessories. For example, do not leave the valve positioner by-pass switch in the by-pass position.
9. If the control valve is to be removed from the system after installation be sure that all block valves are closed and tagged. If the control valve contains damaging fluids or contaminants, it should be tagged accordingly, for proper cleaning, prior to disassembly.
10. Be sure the valve is installed with the flow direction arrow in the proper direction. Cases have been reported where the manufacturer furnished a valve with the arrow pointing in the wrong direction.
11. Review all of the control valve manufacturer's specific instructions prior to installation. The first page of an instruction sheet provided by the Jordan Valve Division of Richards Industries, Inc. (I & M Procedure #60) is in Figure 5, as an example of the kind of material available from manufacturers.

REFERENCES

1. API RP550, Second Edition, 1965, *Manual on Installation of Refinery Instruments and Control Systems,* Part I, Process Instrumentation and Control, 126 pp. American Petroleum Institute, New York.
2. ISA RP4.1, 1951, *Uniform Face to Face Dimensions for Flanged Control Valve Bodies,* 2 pp, Instrument Society of America, Pittsburgh.
3. ISA RP4.2, 1956, *Standard Control Valve Manifold Designs* (Carbon Steel Valves), 20 pp, Instrument Society of America, Pittsburgh.
4. Masek, John A., January, 1967-September, 1969, "Flow Meter and Control Valve Manifold Piping," a collection of reprints from *Heating, Piping and Air Conditioning,* Reinhold Publishing Company, New York.
5. VC-1006-A, *Piping Pointers,* Crane Company, Chicago.

JORDAN VALVE™

DIVISION OF RICHARDS INDUSTRIES, INC
HELEN & BLADE STS. CINCINNATI, OHIO 45216
Telephone (513) 242-3100

60 I & M

INSTRUCTIONS FOR INSTALLATION AND MAINTENANCE

SERIES 60 PRESSURE REGULATORS 1/4" to 2" (PRV)

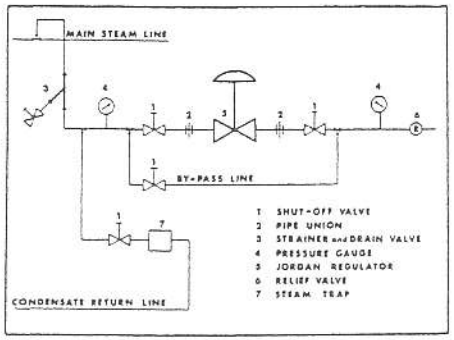
INSTALLATION

YOUR SERIES 60 PRESSURE REGULATOR WILL PROVIDE YOU WITH LONG, TROUBLE-FREE SERVICE IF IT IS CORRECTLY INSTALLED. A FEW MINUTES OF YOUR TIME SPENT READING THESE INSTRUCTIONS MAY SAVE HOURS OF TROUBLE AND DOWN-TIME LATER.

VALVE INSTALLATION (READ CAREFULLY)

1. To protect the regulator from grit, scale, thread chips, and other foreign matter, all pipe lines and piping components should be blown out and thoroughly cleaned before the regulator is installed.
2. Shut-off valves, pressure gages, and by-pass piping should be installed as indicated in the diagram to provide easier adjustment, operation, and testing.
3. In preparing threaded pipe connections, care should be exercised to prevent pipe sealing compound from getting into the pipe lines. Pipe sealing compound should be used sparingly, leaving the two lead threads clean. Jordan uses, and recommends, Seyco #2415 thread sealer Teflon ribbon.
4. **A LINE STRAINER SHOULD BE INSTALLED ON THE INLET SIDE OF THE REGULATOR TO PROTECT IT FROM GRIT, SCALE AND OTHER FOREIGN MATTER. A .033 PERFORATED SCREEN IS USUALLY SUITABLE. LINE STRAINERS ARE AVAILABLE FROM JORDAN VALVE.**
5. Install the regulator in the highest horizontal line of piping to provide drainage for inlet and outlet piping, to prevent water hammer, and to obtain faster regulation.
6. The flow arrow on the regulator body must be pointed in the direction of flow. The regulator may be installed vertically or horizontally without affecting its operation.
7. For best control, 3' - 0" straight sections of pipe should be installed on either side of the regulator.
8. In hot vapor lines, upstream and downstream piping near the regulator should be insulated to minimize condensation.
9. If possible, install a relief valve downstream from the regulator. Set at 15 psi above the control point of the regulator.
10. Expand the outlet piping at least one pipe size if the controlled pressure (downstream) is 25% of the inlet pressure or less. A standard tapered expander connected to the outlet of the regulator is recommended.
11. Where surges are severe, a piping accumulator is recommended.

IDEAL INSTALLATION



TROUBLE SHOOTING

TROUBLE	POSSIBLE CAUSE & CURE
Erratic Control	Oversizing causes cycling and hunting and reduces the rangeability of the valve. Make certain that your sizing is correct.
	Steam traps downstream may need attention.
	Safety valve may be jammed open. Repair as necessary.
	Excessive foreign matter on seats. Clean them.
	Valve stroke out of adjustment. Check and readjust if necessary.
	Valve disc may not be moving freely. Check disc guide clearance.
Will not Operate	Diaphragm ruptured. Replace.
	Adjusting spring broken. Replace.
	Improper spring setting. Reset.

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PROTECT VALVES WITH LINE STRAINERS

Figure 5. SAMPLE INSTALLATION INSTRUCTION SHEET AVAILABLE FROM MANUFACTURERS.

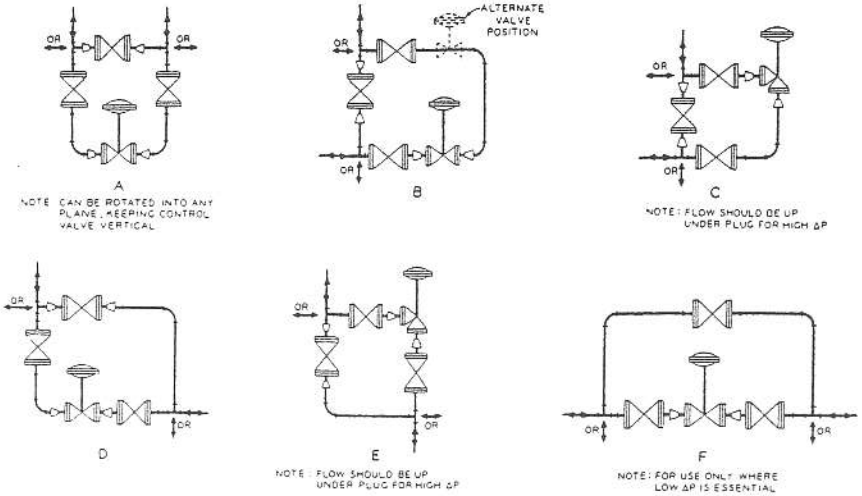
APPENDIX I

CONTROL VALVE MANIFOLD AND PIPING ARRANGEMENTS

Manifold arrangements illustrative of various typical situations are presented in a series of diaphrams on the following pages. These may be quite suitable for use as shown, or may be made suitable for specific requirements by minor modifications.

Figures 6 through 17 are reproduced by permission of American Petroleum Institute, from RP-550, *Manual on Installation of Refinery Instruments and Control Systems;* Part I, Process Instrumentation and and Control, 1965.

Figures 18 through 35 are reproduced by permission of *Heating, Piping and Air Conditioning Journal* from the reprint Collection by John A. Masek, "Flow Meter and Control Valve Piping," 1967, 96 pp.



Arrangement A (ISA RP 4.2, Type 1) is preferred because manifold is compact, control valve is readily accessible for maintenance, and the assembly is easily drained.

Arrangement B is preferred because control valve is more readily accessible.

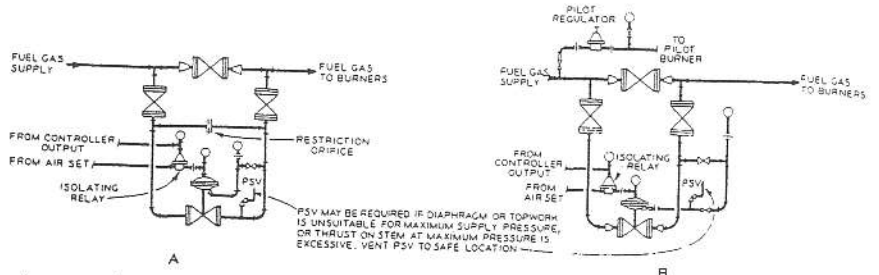
Arrangement C is often used with angle valves. Control valve is self-draining.

Arrangement D (ISA RP 4.2, Type 3) is preferred because the control valve is readily accessible. Bypass is self-draining.

Arrangement E results in a compact manifold, but control valve may not be too accessible.

Arrangement F is preferred because bypass is self-draining; however, requires greater space.

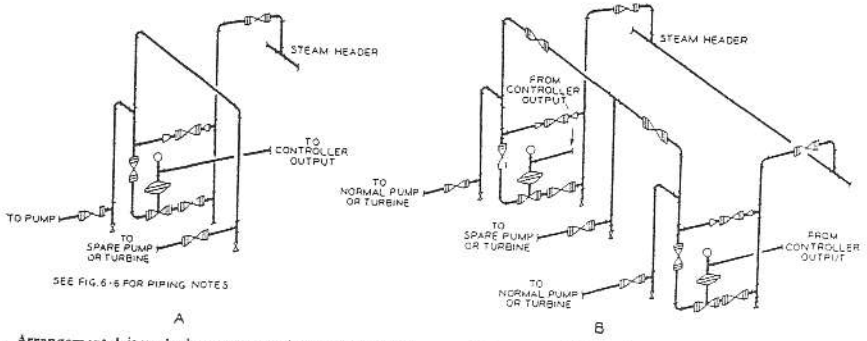
Note: Block and bypass valves should be installed close to tees, as shown, to minimize pockets. Drain and vents on either side of the control valve are not shown, nor are the supports.



Arrangement A shows pressure-balanced valve with isolating relay and restriction orifice for maintaining minimum fires. Restriction orifice is sized for the manufacturer's recommended minimum turndown capacity for the burners. To prevent the control valve from closing completely, travel stops may be used in lieu of the restriction orifice.

Arrangement B shows a pilot regulator which is suggested where minimum fire maintained by restriction would cause furnace tube failure on process flow stoppage. Isolating relay is used to prevent gas flow into controller on diaphragm leakage or breakage. Ratio relay may be used to permit air loading to match fuel gas burner pressure.

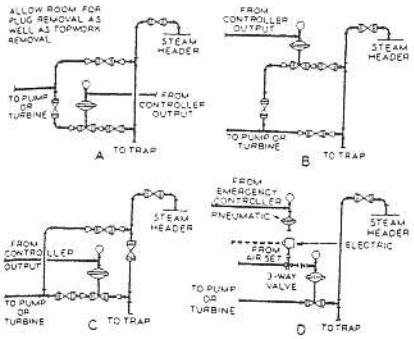
Figure 7. PIPING ARRANGEMENTS FOR PRESSURE-BALANCED VALVE AT PROCESS HEATERS.



Arrangement A is used where one control valve is employed for normal pump, or turbine, and spare.

Arrangement B is used where two normal pumps have a common spare.

Figure 8. STEAM PIPING TO SPARE PUMP OR TURBINE.



Arrangement A makes the control valve readily accessible; the piping is self-draining. Steam inlet may be above minimum distance required for plug removal.

Arrangement B may be used where the bypass can be installed at turbine or pump inlet. Control valve may be too high for easy access; piping is self-draining.

Arrangement C is often used to make control valve more accessible because the control valve can be located at the same elevation as turbine or pump steam inlet.

Arrangement D is often used for emergency start, or stop, of pump or turbine.

Notes:

1. Bypass and bypass valve should not be so located as to form a pocket; they should be self-draining.
2. Location of separators, strainers, and traps not shown because company standards differ on use and placement. The control valve should be located as low as possible for easy access.

Figure 9. CONTROL VALVE ARRANGEMENTS FOR STEAM TO PUMPS OR TURBINES.

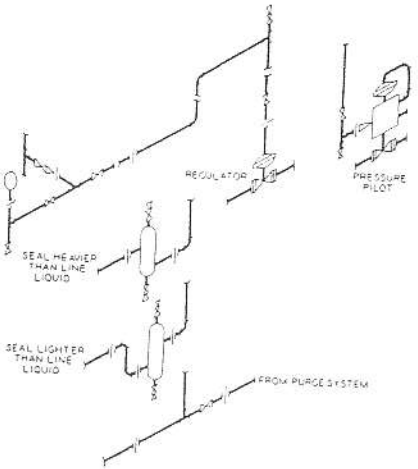
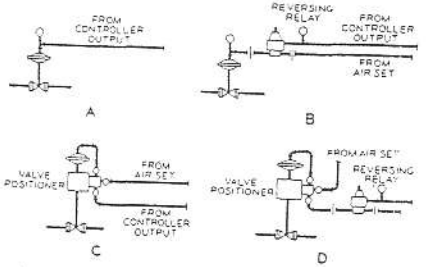


Figure 10. PIPING AT REGULATOR VALVES OR PRESSURE PILOTS.



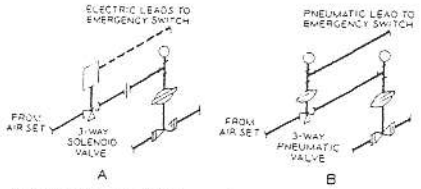
Arrangement A shows nominal control valve piping with gage for reading diaphragm pressure.

Arrangement B shows control valve with reversing relay. The reversing relay is used to reverse controller output to valve, where two valves with different action are connected to one controller, or is used to provide the same action when on automatic or manual control.

Arrangement C shows control valve with positioner.

Arrangement D shows control valve with positioner and reversing relay. The reversing relay is the same as used in **B** with standard control valve; in addition, it allows the positioner to be bypassed. A reverse-acting positioner may be used but it may not be bypassed.

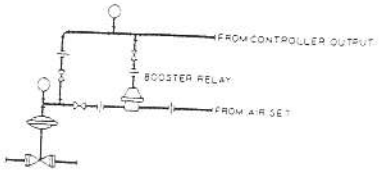
Figure 11. ARRANGEMENTS OF PNEUMATIC PIPING OR TUBING, TO CONTROL VALVES.



Arrangement A shows control valve with emergency solenoid actuator.

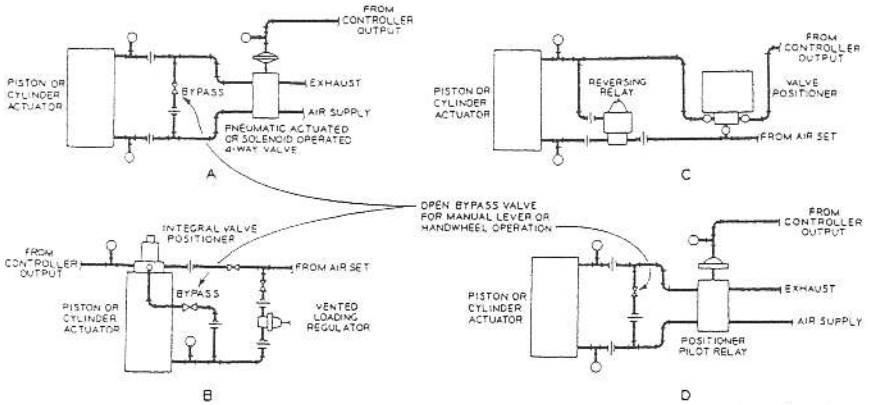
Arrangement B shows control valve with emergency pneumatic actuator.

Figure 12. CONTROL VALVE PIPING WITH EMERGENCY CONTROL ACTION.



Arrangement shows control valve with booster relay. Block valves and bypass valves are for booster servicing.

Figure 13. BOOSTER RELAY PIPING.



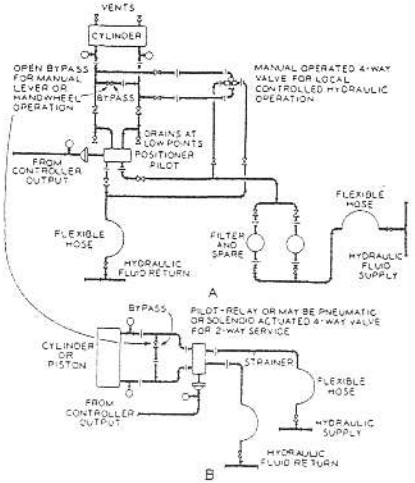
Arrangement A shows the actuator used with four-way valve in two-position service.

Arrangement B shows the actuator with constant pressure loading on one side.

Arrangement C shows the actuator with reversing relay.

Arrangement D shows the actuator with pilot relay for throttling service.

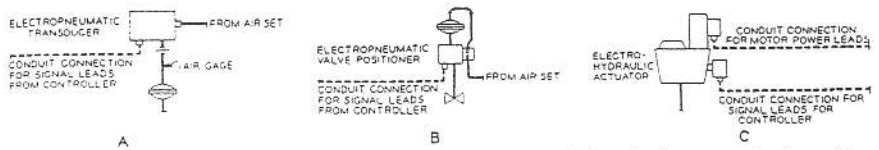
Figure 14. AIR CYLINDER OR PISTON ACTUATORS.



Arrangement A shows the cylinder actuator with provision for local manual hydraulic operation.

Arrangement B shows the cylinder or piston actuator without local hydraulic operation.

Figure 15. HYDRAULIC CYLINDER OR PISTON ACTUATORS.



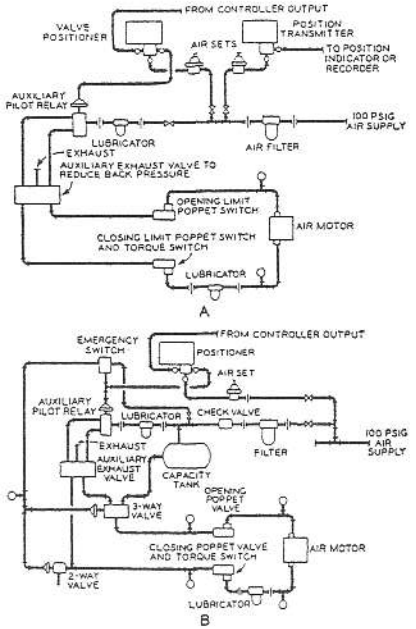
Arrangement A shows the electropneumatic transducer mounted at pneumatic control valve.

Arrangement B shows the electropneumatic valve positioner on valve.

Arrangement C shows the electrohydraulic valve actuator.

Note: Provide seals in electrical leads as required by electrical specifications for area in which actuator is installed.

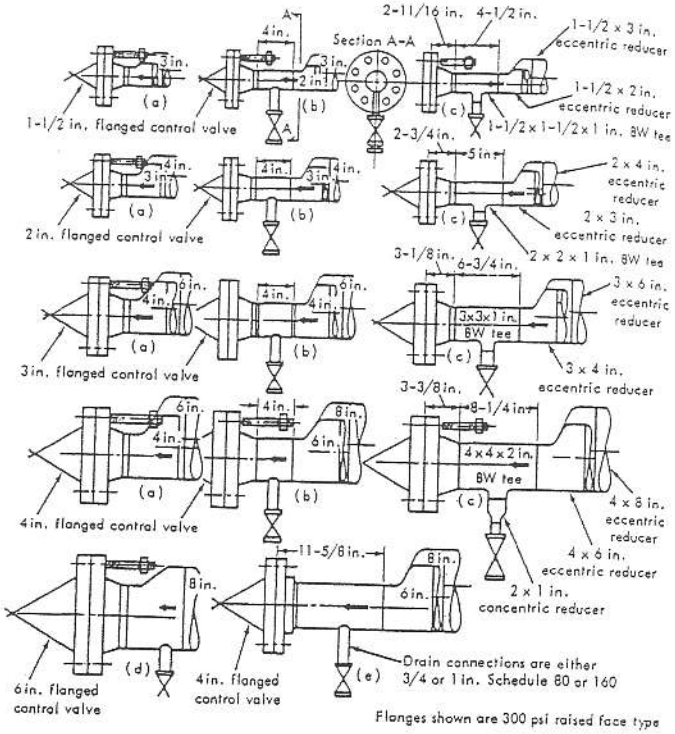
Figure 16. ELECTROPNEUMATIC AND ELECTROHYDRAULIC ACTUATORS.



Arrangement A shows air motor actuator with position transmitter.

Arrangement B shows air motor actuator with auxiliary equipment to close valve on emergency.

Figure 17. AIR MOTOR ACTUATORS.



a) Weld neck flange and eccentric reducer; configuration can prevent stud removal.
b) 4 in. long pipe spool piece welded between flange and reducer, allowing bolt removal space and drain connections.
c) Butt weld reducing tee, providing bolt removal space and drain connections.
d) 6 in. control valve flange with 8 in. eccentric reducer; stud bolts can be removed.
e) same as c) except slip-on flange is used with appropriate length pipe.

FIGURE 18. TYPICAL CONNECTIONS AT CONTROL VALVE IN MANIFOLD ASSEMBLY.

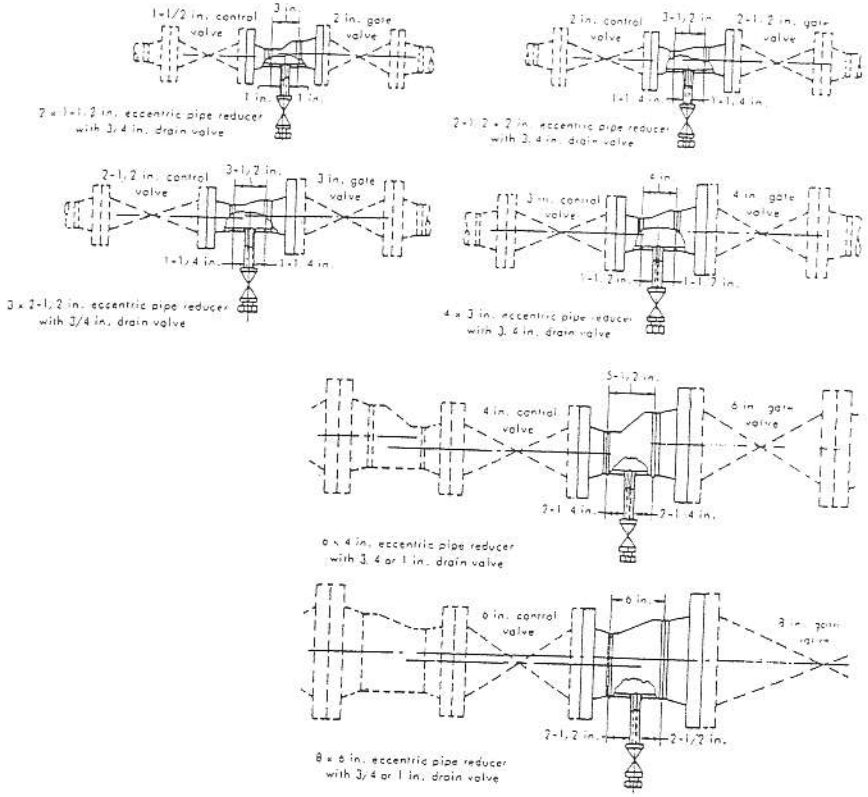


Figure 19. DRAIN VALVES INSTALLED IN BOTTOM OF PIPE REDUCERS ADJACENT TO CONTROL VALVES.

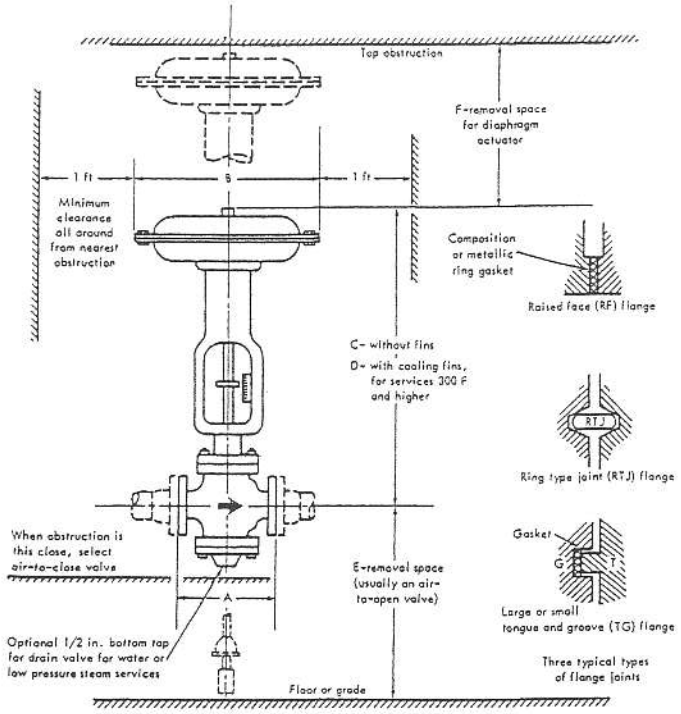
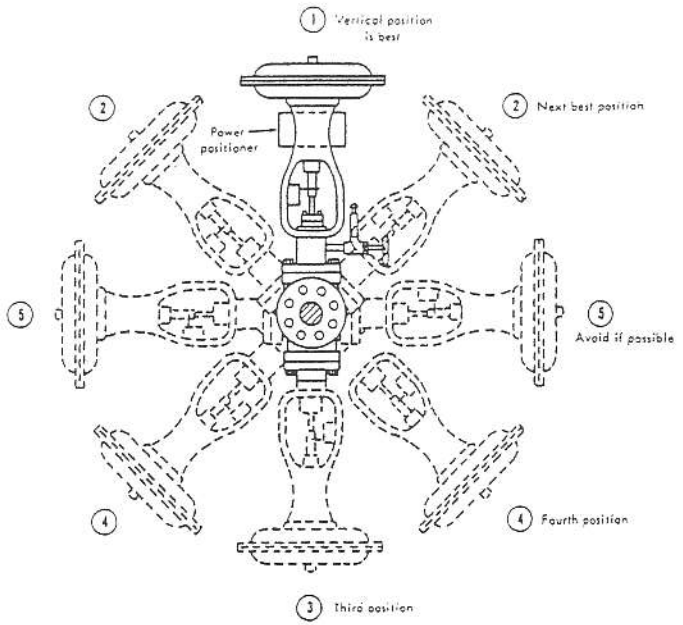


Figure 20. CLEARANCE REQUIREMENTS FOR CONTROL VALVE MANIFOLDS.

Control valve body size, in.	A (face-to-face dimensions) 150 psi raised face flanges	300 psi raised face flanges	400 psi raised face flanges	B	C	D	E*	F*
2**	10	10½	11¼	15½	33	39	21	18
2½	10⅞	11½	12¼	15½	34	40	21	20
3	11¾	12½	13¼	15½	35	40	21	23
4	13⅞	14½	15½	15½	36	42	21	24
6	17¾	18⅝	19½	19⅛	40	43	30	54
8	21⅜	22⅜	23⅜	20¾	46	50	30	54

*Dimensions for stem and trim removal and body removal are minimum; increase if possible.

**Standard screwed control valves should have same clearance dimensions as shown for 2 in. plugged control valves.



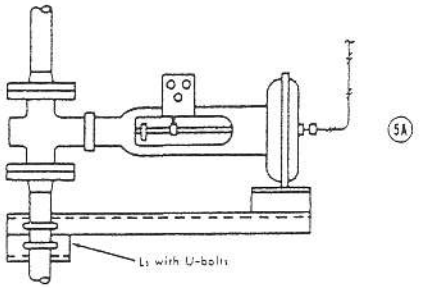


Figure 21. ALTERNATE ORIENTATIONS FOR DIAPHRAGM CONTROL VALVES, POSITION 1 IS PREFERRED.

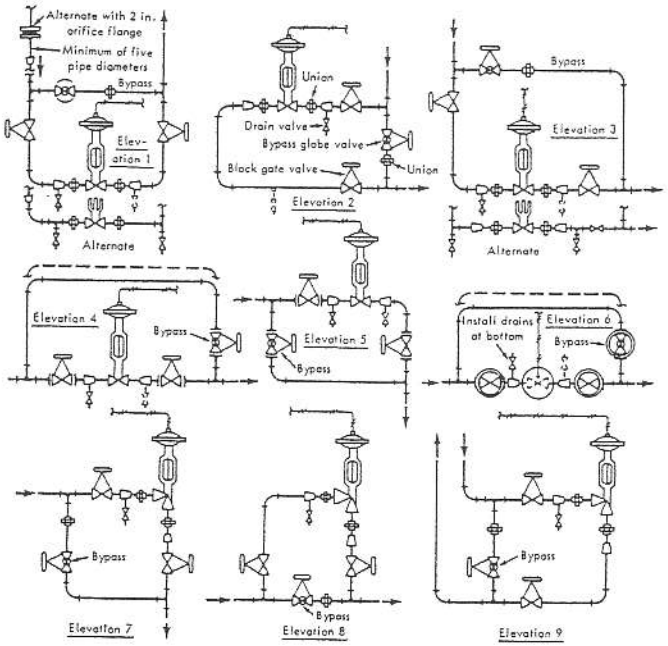
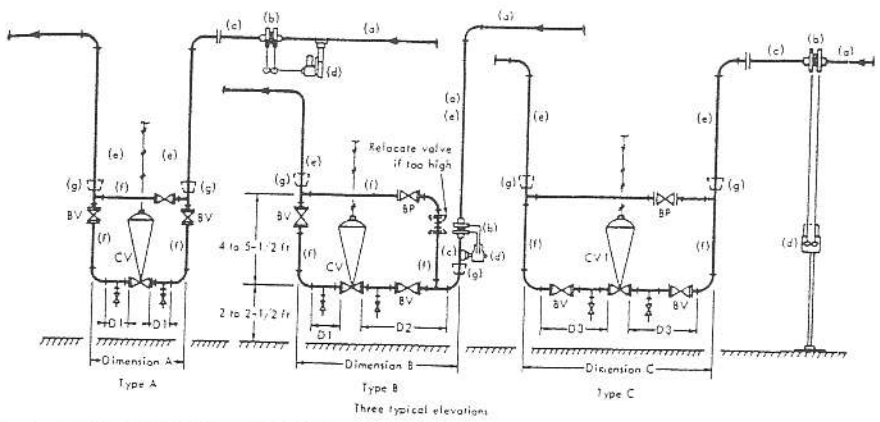


Figure 22. NINE TYPICAL CONTROL VALVE MANIFOLD ARRANGEMENTS FOR THREADED END GLOBE AND ANGLE BODY CONTROL VALVES.



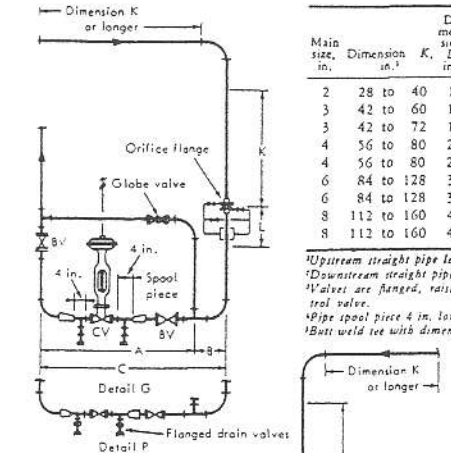
Reducer size (ct., in.		Valve sizes, in.*			Type A manifold			Type B manifold			Type C manifold		
						Straight pipe			Straight pipe			Straight pipe	
Main size (ct., 2 in.	Manifold size (1), in.	CV	BV	BP	Dimension A, in.	D1, in.	D1, pipe diameters	Dimension B, in.	D2, in.	D2, pipe diameters	Dimension C, in.	D3, in.	D3, pipe diameters
2	2	1½	1½	1½	28¼	19	6	45¾	21	16	54¾	20	13
3	2	1½	1½	1½	28¼	19	6	45¾	21	16	54¼	20	13
3	2	2	2	2	32⅛	22	4	51¼	26	12	60⅜	25	12
4	2	2	2	2	32⅛	22	4	51¼	26	12	60⅜	25	12
4	3	3	3	3	41⅛	28	3	65½	37	11	76⅜	32	10
6	3	3	3	3	41⅛	28	3	65½	37	11	76⅜	32	10
6	4	4	4	4	49⅞	33	3	77	39	9	87⅝	37	9
8	4	4	4	4	49⅞	33	3	77	39	9	87⅝	37	9
8	6	6	6	6	73	50	2	101¼	40	6	114½	48	8

*All valves are flanged, raised face, 300 psi ASA rating. CV = control valve, BV = block valve (gate), and BP = bypass valve (globe).

(1) Control valve body (with reduced trim) and manifold all in process line size
(2) Manifold piping all in control valve body size but smaller than process line size

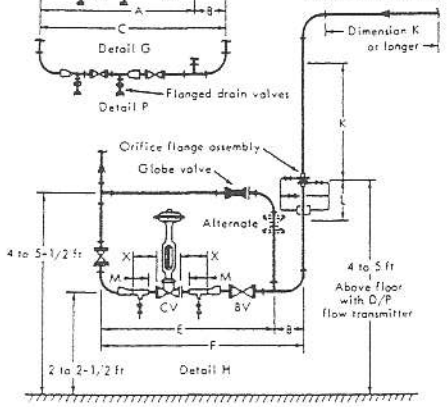
(a) Process line upstream from orifice plate, minimum of 15 nominal pipe diameters of straight pipe
(b) Orifice plate in orifice flange assembly
(c) Process line downstream from orifice plate, minimum of 5 to 10 diameters of straight pipe
(d) Differential pressure flow transmitter
(e) Process line size
(f) Control valve manifold, control valve body pipe size, process line size, or smaller
(g) Pipe reducer (optional when manifold is smaller than process line size)
CV-Diaphragm control valve, CV-1 — Control valve with split body
BV-Block gate valve, control valve size
BP-Bypass globe valve, control valve size
D -Straight pipe section upstream and downstream of control valve

Figure 23. THREE TYPICAL CONTROL VALVE MANIFOLDS.



Main size, in.	Dimension in.[1]	K,	Dimension L, in.[2]	Valve sizes, in.[3] BV	CV	Detail G dimensions, in.[4] A	B	C	Detail H dimensions, in.[5] E	B	F
2	28 to	40	10	2	1½	48⅛	5½	53⅜	49⅛	5½	54⅛
3	42 to	60	15	3	1½	54⅞	7⅞	62¾	55⅞	7⅞	63¾
3	42 to	72	15	3	2	56¼	7⅞	64⅛	61¾	7⅞	69⅛
4	56 to	80	20	4	2	60⅞	10⅛	71	66⅜	10⅛	76½
4	56 to	80	20	4	3	63⅜	10⅛	73¾	72⅛	10⅜	82¼
6	84 to	128	30	6	3	76	14⅞	90⅜	84½	14⅞	99⅛
6	84 to	128	30	6	4	76½	14⅞	93⅛	93	14⅞	107⅛
8	112 to	160	40	8	4	83½	19	104½	100	19	119
8	112 to	160	40	8	6	90⅝	19	109⅜	110⅛	19	129⅜

[1] *Upstream straight pipe length based on 0.70 or 0.80 beta ratio.*
[2] *Downstream straight pipe length based on 5 nominal pipe diameters.*
[3] *Valves are flanged, raised face, 300 psi USASI rating; BV = block valves (gate), CV = control valve.*
[4] *Pipe spool piece 4 in. long on each side of control valve.*
[5] *Butt weld tee with dimension M (see table below) on each side of control valve.*



Control valve flange size, in.	Butt weld tee size, in.*	BW tee dimension M, in.	Approximate straight pipe length, X, nominal pipe diameters
1½	1½×1½×1	4½	4+
2	2 ×2 ×1	5	3+
3	3 ×3 ×1	6¾	3+
4	4 ×4 ×1½	8¼	2+
6	6 ×6 ×2	11¼	2+
8	8 ×8 ×3	14	2+

*With 1 in. drain valve at bottom.

Figure 24. FLANGED CONTROL VALVE MANIFOLDS WITH ORIFICE FLANGE ASSEMBLIES.

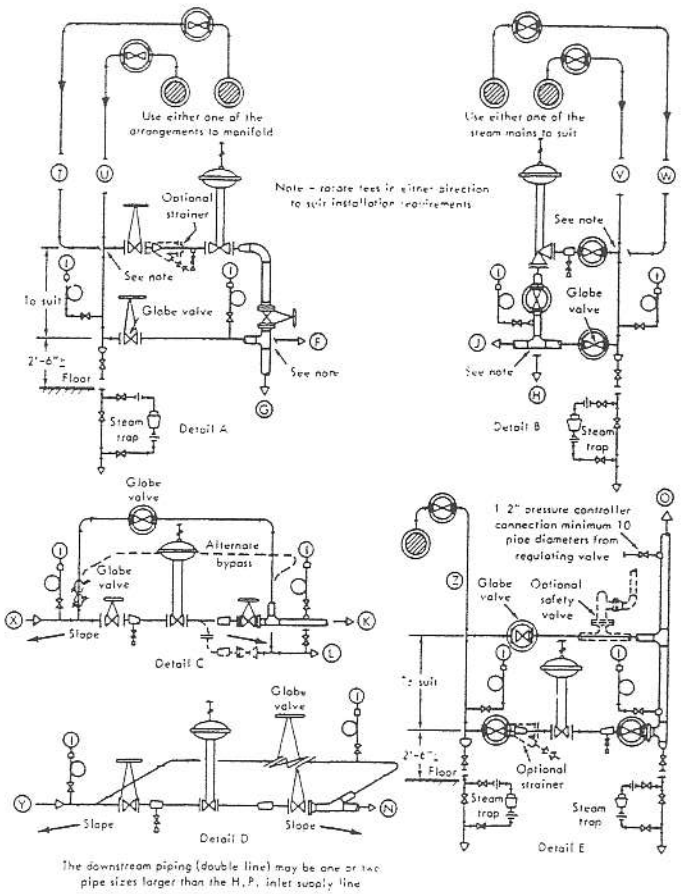


Figure 25. MANIFOLD DESIGN ARRANGEMENTS FOR FLOW AND PRESSURE REDUCING CONTROL VALVES IN LOW PRESSURE STEAM SERVICE USING FLANGED FITTINGS.

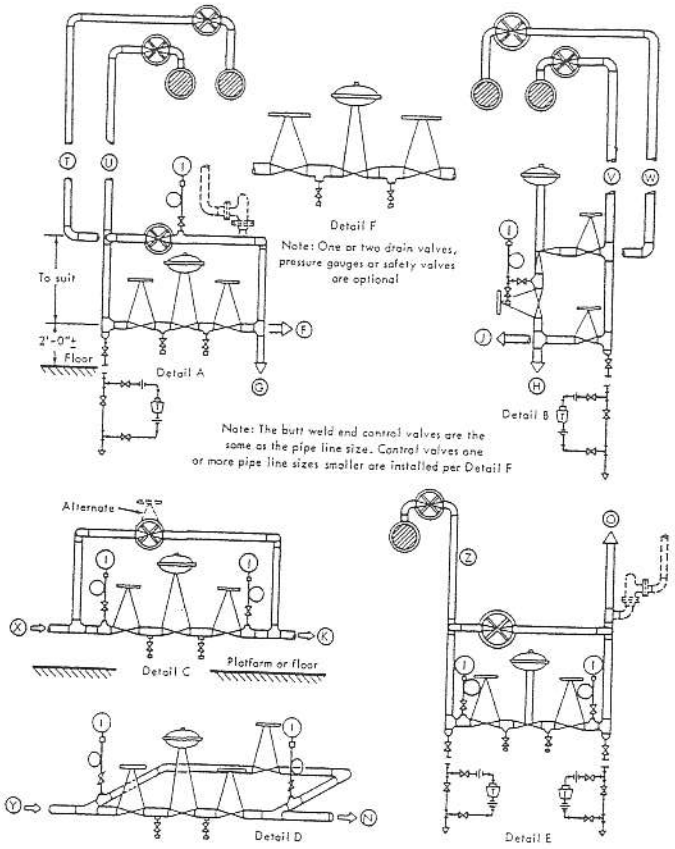


Figure 26. CONTROL VALVE MANIFOLD USING WELDED CONNECTIONS FOR UPPER PRESSURE RANGE SERVICES ON STEAM AND PROCESS STREAMS.

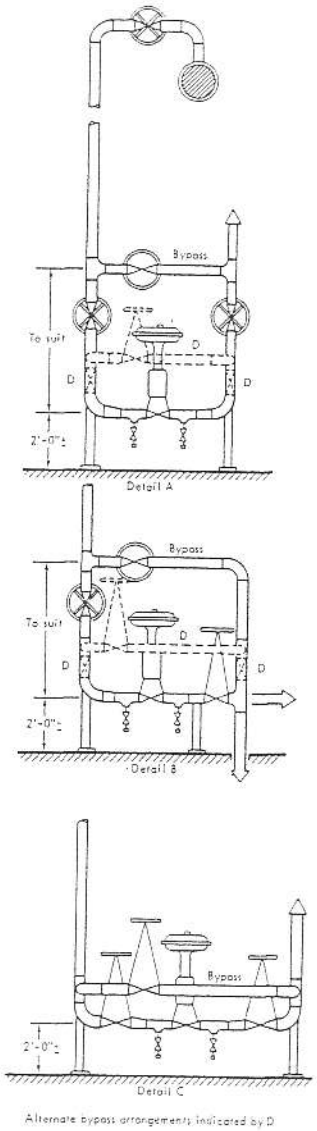


Figure 27. BUTT WELD END TYPE CONTROL, BLOCK, AND BYPASS VALVES IN PIPING MANIFOLDS PROVIDING STEAM OR OTHER FLUID FLOW REGULATION.

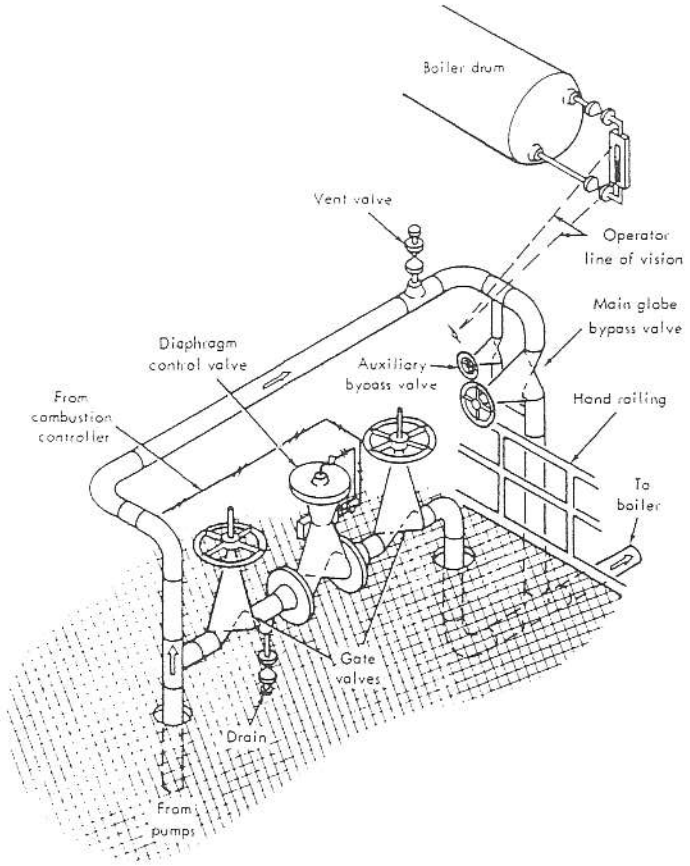


Figure 28. CONTROL VALVE MANIFOLD WITH BYPASS PIPING EXTENDING BEYOND ACCESS PLATFORM.

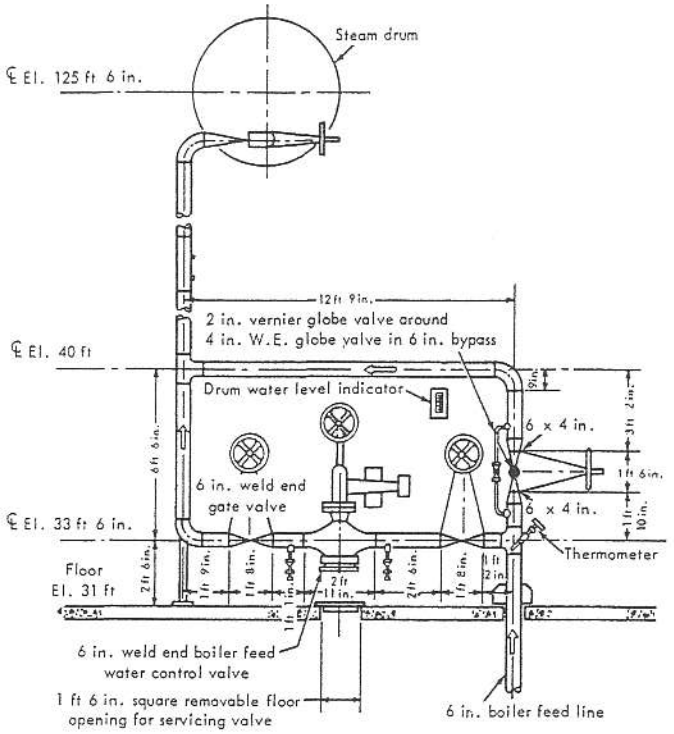


Figure 29. FEED WATER CONTROL VALVE MANIFOLD BELOW INLET TO BOILER STEAM DRUM.

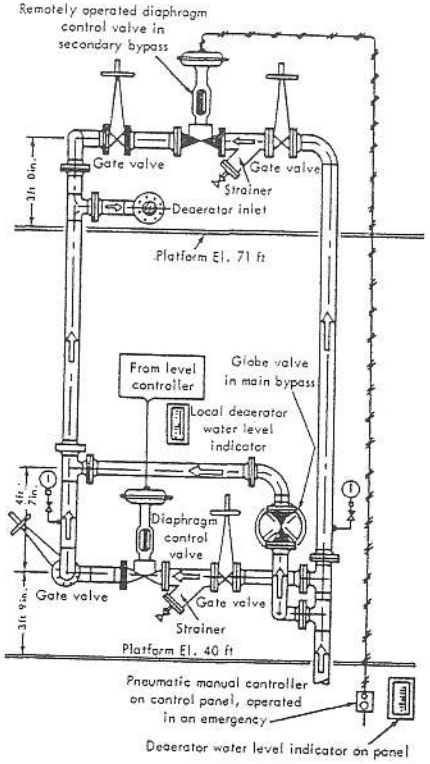


Figure 30. MAKEUP WATER CONTROL VALVE MANIFOLD COMBINED WITH AN AUXILIARY VALVE MANIFOLD, REMOTELY CONTROLLED.

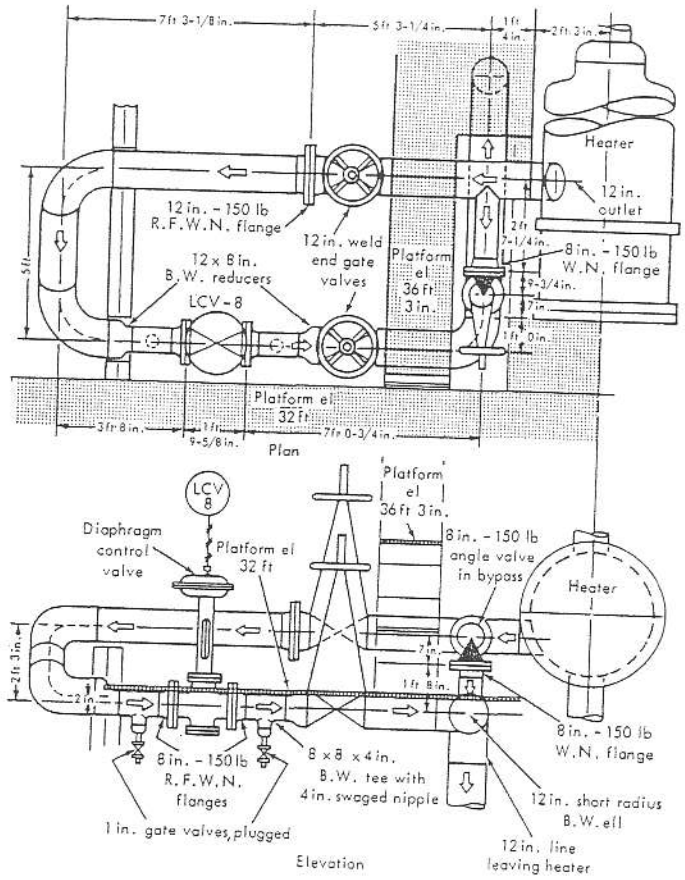


Figure 31. CONTROL VALVE MANIFOLD LOOP AT HEATER OUTLET.

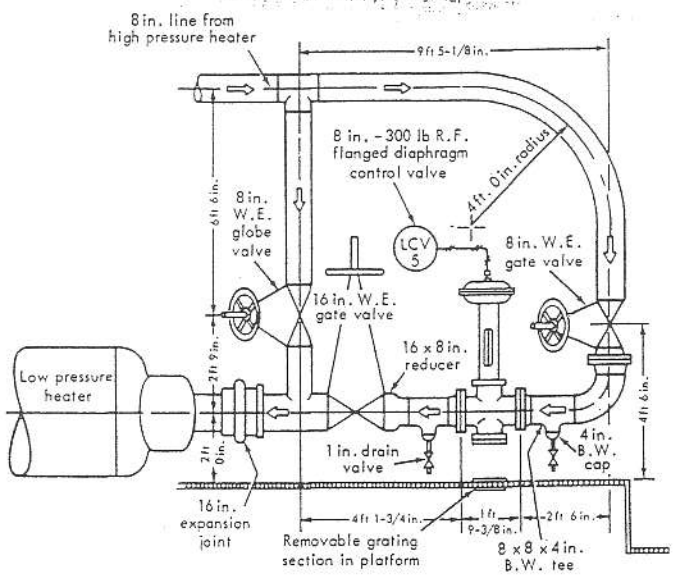


Figure 32. MANIFOLDING OF CONTROL VALVE FOR LOOPED ENTRY TO HEATER FROM OVERHEAD SUPPLY LINE.

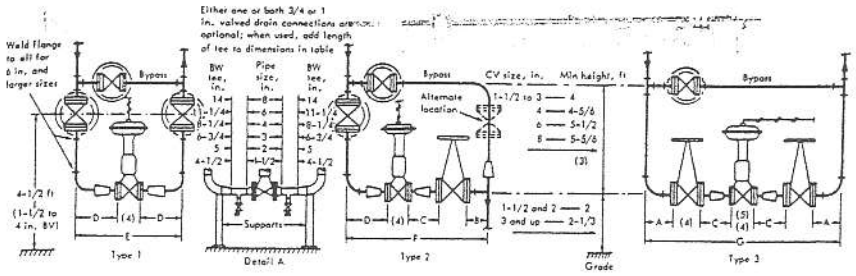
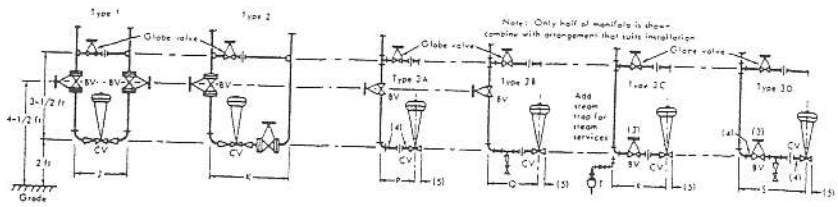


Figure 33. DIMENSIONS FOR FLANGED CONTROL VALVE MANIFOLDS.

Valve sizes, in.		Dimensions, in., with swaged nipples							Dimensions, in., with weld reducers				
BV	CV	A	B	C	D	E	F	G	C	D	E	F	G
150 psi block and control valves—													
2	1½	5½	5	11 7/8	11 3/4	32¼	44⅛	57	7 3/4	8 7/8	[1]	[2]	50⅛
3	1½	7¼	6⅛	13 7/8	14 3/4	38¼	51¼	66	8½	10 7/8	[1]	[2]	42¼	57
3	2	7¼	6⅛	13¼	15	40½	53⅜	67¾	8¼	10½	[2]	[2]	43⅜	58¼
4	2	9	7⅛	14½	17½	45½	58⅜	73⅛	9½	12½	5½	[2]	48⅜	65⅛
4	3	9	7⅛	14¼	17¼	47⅛	60⅜	77⅜	9¼	12⅜	5 7/8	[2]	50⅜	67⅛
6	3	12½	9⅛	18¼	23¼	59⅜	[1]	[1]	11⅜	17¼	46⅞	[2]	60⅜	81⅜
6	4	12½	9⅛	18½	24	62	[1]	[1]	12	17½	49	[2]	63¼	84¼
8	4	16	11	20	28	70	[1]	[1]	15	21	56	[2]	70⅜	95¼
8	6	16	11	20½	28½	74⅞	[1]	[1]	13½	21½	60⅞	[2]	73½	100⅛
10	6	19	12½	22½	33½	84⅞	[1]	[1]	14½	25½	68⅞	[2]	83½	113⅛
10	8	19	12½	23	34	89½	[1]	[1]	15	26	73½	[2]	88⅛	119¾
150 psi block valve, 300 psi control valve—													
2	1½			11 3/4			45⅜	58	8 7/8			[2]	51
3	1½			13 7/8			52¼	67	8 3/4			[2]	43¼	58
3	2			13½			53⅜	68⅛	9			[2]	44⅛	59½
4	2			14¾			59⅜	76⅛	9¾			[2]	48⅞	66⅜
4	3			15⅛			62⅛	79½	10⅛			[2]	52⅛	69⅛
6	3			18⅜			[1]	[1]	12⅜			[2]	62⅛	83⅜
6	4			18⅞			[1]	[1]	12⅜			[2]	64⅜	85⅜
8	4			20⅜			[1]	[1]	13⅜			[2]	72	96⅝
8	6			20⅞			[1]	[1]	13⅞			[2]	77⅛	101⅝
10	6			22⅞			[1]	[1]	14⅞			[2]	85⅜	112¾
10	8			23⅜			[1]	[1]	15⅜			[2]	89⅞	117½
300 psi block and control valves—													
2	1½	5¾	5¼	11 3/4	12 7/8	33¼	47⅜	62	8 7/8	8 3/4	[1]	[2]	55
3	1½	7⅜	6½	13 3/4	15 7/8	39¼	56⅛	74¾	9 7/8	10 3/4	[1]	[2]	47⅛	65¾
3	2	7⅜	6½	13⅞	15¾	41⅛	57½	76⅛	9⅜	10¾	[1]	[2]	48½	67⅛
4	2	9⅜	7½	15⅜	17¾	46⅛	65½	83⅞	10⅜	12¼	36¼	53⅜	73⅞
4	3	9⅜	7½	15½	18⅜	48⅞	65⅞	86⅝	10½	13¼	38⅞	55⅞	76⅜
6	3	12⅞	9½	19	24⅛	60¾	[1]	[1]	12½	17⅝	47⅞	68¼	91½
6	4	12⅞	9½	19¼	24½	63⅜	[1]	[1]	12¾	17⅞	50⅛	70¾	97⅞
8	4	16⅜	11⅜	20¼	28⅛	71⅜	[1]	[1]	13¾	21⅜	57⅜	77¾	108½
8	6	16⅜	11⅜	21¾	28⅞	76½	[1]	[1]	14¼	21⅞	62½	82⅞	113¼
10	6	19⅜	13⅜	23½	33⅞	86½	[1]	[1]	15½	25⅞	70½	91⅜	125¼
10	8	19⅜	13¾	24	34⅛	91¼	[1]	[1]	16	26⅛	75¼	96⅛	130
600 psi block and control valves—													
2	1½	6¼	5⅜	12⅜	12½	35⅛	52⅜	71⅝	9⅛	9	[1]	[2]	64⅛
3	1½	8	6⅞	14½	13½	41⅛	61¼	83⅜	10	11	[1]	[2]	52¾	74⅜
3	2	8	6⅞	14⅜	15⅜	42¾	62¼	85¼	10⅛	11⅛	[1]	[2]	53⅞	76¼
4	2	10¼	8⅜	16⅜	18⅜	47¾	71⅜	99¼	11⅜	13⅜	37¾	61⅜	88¼
4	3	10¼	8⅜	16¼	18½	50½	74⅛	102	11¾	13½	40½	64¾	92
6	3	13⅞	10½	20⅛	24½	62½	[1]	[1]	13⅞	18	49½	78⅛	113½
6	4	13⅞	10½	21⅜	25¼	66¼	[1]	[1]	14⅛	18¾	53¾	81⅞	117¼
8	4	17½	12½	22¾	29¼	74¾	[1]	[1]	15¾	22¼	60¼	92½	134¾
8	6	17½	12½	23⅜	29⅞	80	[1]	[1]	16⅜	22⅞	66	98¼	140½
10	6	21⅛	14⅛	26½	34⅞	90	[1]	[1]	18½	26⅞	74	111¾	161½
10	8	21¼	14¼	26¾	35½	95¼	[1]	[1]	18¾	27½	79¼	116½	166¾

[1] Use butt weld reducers.
[2] Use butt weld swaged pipe nipples.
[3] Add 6 in. for air fins.
[4] Gasket thickness is 1/8 in. for 150 and 300 psi valves; 1/4 in. for 600 psi valves.
[5] Split body valve optional.



Valve sizes, in.		Dimensions, in., for flanged block valve, screwed control valve	K, for ratings, psi, of:			Dimensions, in., for screwed block and control valves (and manifold type)											
BV	CV	J	150	300	600	P	Q	R	S	2P (1)	2Q (1)	2R (3)	3S (3)	P+Q (1)	P+R (2)	P+S (2)	Q+R (2)	Q+S (2)	R+S (1)
¼	½	—	—	—	—	11 7/8	16 11/16	17 1/8	23 3/8	22 1/8	33 3/8	34 7/8	46 3/8	28	28 1/2	34 1/8	34 1/8	40 1/4	40 1/4
1	½	—	—	—	—	13 3/8	17 3/4	18 1/8	24 1/2	23 1/4	33 1/2	36 1/4	49	29 1/8	30	36 1/8	36 1/8	42 3/4	42 3/4
1	¾	—	—	—	—	11 7/8	17 11/16	18 1/8	24 3/8	23 1/8	35 1/8	36 1/8	48 7/8	29 1/4	29 7/8	36	36	42 3/4	42 3/4
1½	¾	24	36	37	40	13 3/4	21 3/8	20 7/8	28 1/4	27 1/2	42 1/4	41 3/4	56 1/2	34 7/8	34 5/8	42	42	49 1/4	49 1/4
1½	1	24	36	37	40	14 7/8	21 13/16	21 7/8	28 1/8	28 7/8	43 5/8	43 3/4	57 7/8	36 1/4	36	43 1/8	43 1/8	50 1/4	50 1/4
2	1	24 1/4	36 5/8	38 3/8	42 1/4	16 3/4	24 1/16	24 1/2	32	33 1/2	48 1/16	49	64	41	41 1/4	48 1/4	48 1/4	56 1/4	56 1/2
2	1½	25 3/8	37 3/4	38 1/4	43 3/8	17 7/8	24 13/16	24 13/16	32 7/8	34 1/8	49 5/8	49 7/8	64 7/8	41 7/8	42 1/8	49 1/8	49 1/8	57 1/4	57 5/8

Pipe size of item	Basic dimensions, in., used to establish dimensions P through S: 3000 psi ell and tee, c/l to end	3000 psi union, end to end	600 psi end to end[3]	4 in. Control valve, gate valve, nipple less threads[4]	c/l to end[4]	Swaged nipple less engagement for small end size of: ½	¾	1	1½	Pipe nipples less thread engagement, in., for nipple length, in., of: 1½	2	2½	3	3½	4	6	8	10
½	—	—	—	—	2½	—	—	—	—	½	1	1½	2	2½	3	5	7	9
¾	1½	2¼	3½	2⅞	2½	1 7/8	—	—	—	⅛	⅞	1⅜	1⅞	2¼	2⅞	4⅞	6⅞	R⅛
1	1¾	2 7/8	4⅜	2⅜	3 7/8	2 7/8	2¼	—	—	—	⅜	1⅛	1⅝	2⅛	2⅝	4⅜	6⅜	R⅝
1½	2⅜	3	4½	2⅝	3¾	—	3¼	3½	—	—	⅜	1⅛	1⅝	2⅛	2⅝	4⅜	6⅜	R⅝
2	2½	3⅜	5¼	2½	4½	—	—	3 7/8	4 3/8	—	—	½	1½	2	2½	4½	6½	R½

[3,4] Refer to drawings above; also, for Note 5, see control valve manufacturer's catalog for exact end-to-end dimension.

Figure 34. DIMENSIONS FOR SCREWED CONTROL VALVE MANIFOLDS.

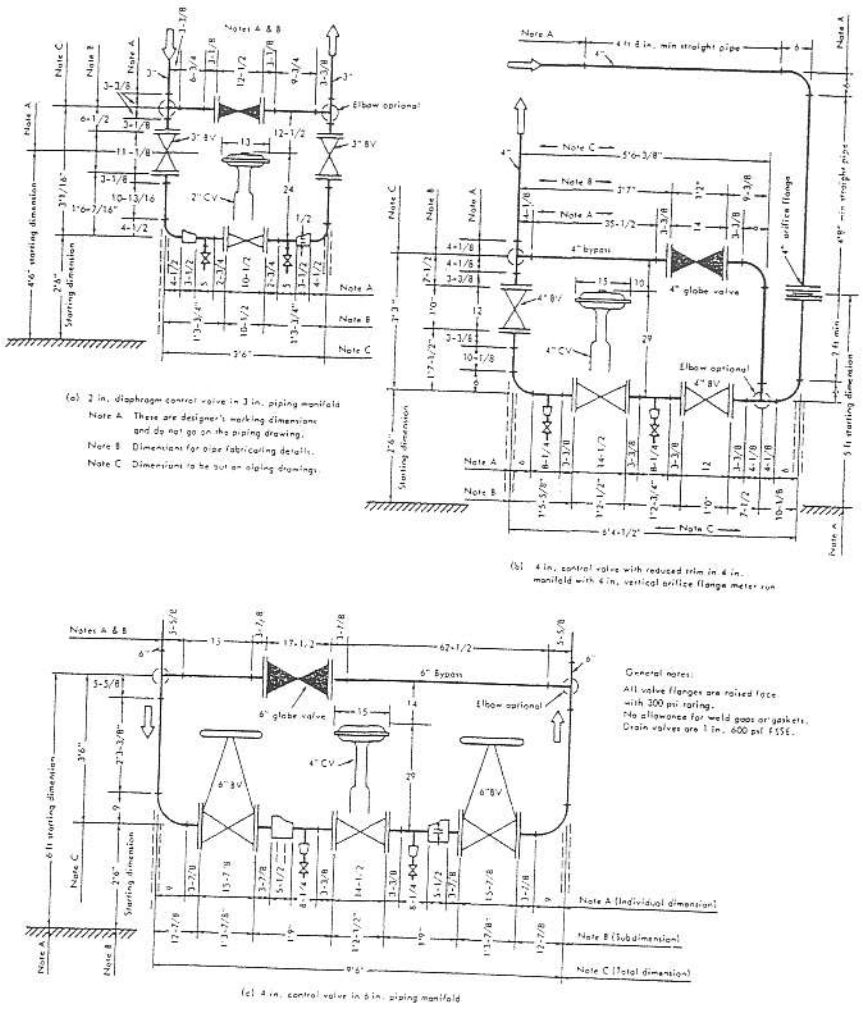


Figure 35. METHODS FOR DIMENSIONING CONTROL VALVE MANIFOLD PIPING. ALL UNDESIGNATED DIMENSIONS ARE IN INCHES.

APPENDIX II
HELPFUL HINTS ON HANDLING TOOLS AND EQUIPMENT

The following piping hints and diagrams were reproduced by permission from Crane Company's Bulletin VC-1006-A, *Piping Pointers.*

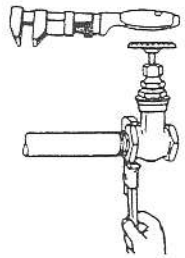
They are included as a helpful supplement since it is felt that in many cases, field problems arise because of failure to observe such simple and obvious practices and procedures.

know your wrenches

easy does it with the proper tools

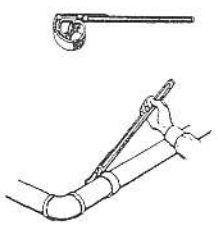
Good handling of piping materials begins with knowing the proper tools to use and how to use them. Wrenches are important tools on every piping job. When used improperly, however, they can cause considerable damage. Two simple rules regarding wrenches can keep you out of trouble:

Rule 1: choose the right type.
Rule 2: choose the right size.



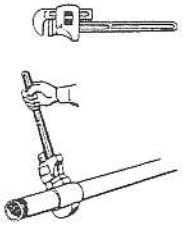
hex and monkey wrench

Your hexagonal or monkey wrench -- with its smooth, square jaws -- is always your best bet for hex end valves and fittings. It gives a better fit on the part to be turned, and doesn't have the crushing effect of a pipe wrench. Because of the shape of their jaws, hex or monkey wrenches obviously cannot be used on pipe or other round objects.



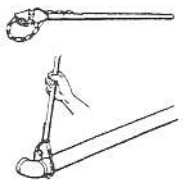
strap wrench

Strap wrenches are used mainly when working with plated or polished materials to avoid marring the surface. They also come in handy in tight places where you can't get in with a pipe wrench.



pipe wrench

Designed for use on pipe and screwed end fittings only. Not as efficient as a monkey wrench on hexagonal ends. The harder you pull a pipe wrench, the tighter it squeezes. This squeezing action can do great damage to a valve. It may cripple or even crush its body.

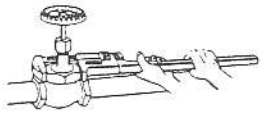


pipe tongs or chain pipe wrench

Pipe tongs are wrenches of the chain strap and lever type. They are generally used for larger sizes of pipe—about 3-inches and above —although they are available for sizes down to 1½-inch.

open end wrenches

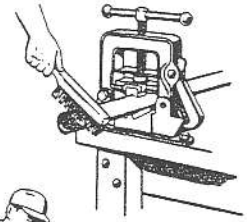
Open end or socket wrenches are most suitable and fastest to work with when pulling up flange bolts and nuts. Getting the right size is the most important consideration here. Slippage of the wrench can cause bruised knuckles and can wear nut and bolt heads round.

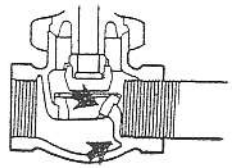


too much leverage is dangerous

Avoid bearing down on the joint with an oversize wrench or a wrench with an extension handle. It often gives you more leverage than you need, and may result in pulling up a joint so tight that the fitting will be cracked, the valve twisted out of shape, or the pipe run clear into the seats.

how to make up a screwed joint





The screwed joint (threaded male and female ends screwed together) is the most common method of joining pipe. A high degree of standardization is maintained in the threading of piping materials. American Pipe Thread Standards guide the piping industry.

The chart below shows the normal engagement male and female threads for making up screwed joints.

use a wire brush

Since dirt can make a good fit impossible, the first thing to do is to wipe both male and female threads clean. Wire brushing is recommended, especially if threaded pipe has been exposed to weather. Running a tap or die over the threads will usually straighten any that may be damaged and will help to assure a good metal-to-metal joint.

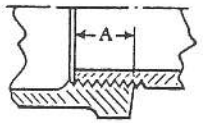
lubricant is good - but not inside a line

Next, use a bit of good thread lubricant or "pipe dope," but put it on the *male threads only*. This prevents any excess from squeezing into the pipe and causing harm to valve seats or other mechanism. Lubricant reduces friction when pulling up a joint. It is not expected to seal the joint, or to compensate for poor workmanship.

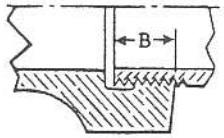
With lubricant applied, start the joint by hand. If the thread engagement feels "right," turn it up as far as it will go. If the threading job is up to standard . . . and if the threads are free of dirt . . . the joint is now started right. Further turns with a wrench will complete the job.

pipe run in too far

But be careful with the wrench. Never use an oversize one that will tempt you to "lean on the joint." Too much pull up can cause damage, especially in the case of valves. Don't try to run all the male threads into the joint. The lead of the die always leaves a few imperfect and unusable male threads which should be left exposed.



American Standard and A.P.I. line pipe threads



shoulder type drainage fitting treads

normal engagement between male and female threads to make tight joints

DIMENSIONS, IN INCHES

Dimensions given do not allow for variations in tapping or threading.

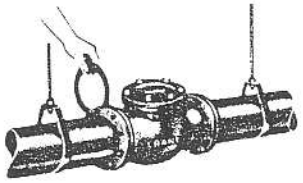
Size	1/8	1/4	3/8	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2
A	1/4	3/8	3/8	1/2	9/16	11/16	11/16	11/16	3/4	15/16
B*						9/16	5/8	5/8	5/8	7/8

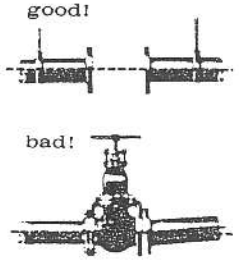
Size	3	3-1/2	4	5	6	8	10	12	14
A	1	1-1/16	1-1/8	1-1/4	1-5/16	1-7/16	1-5/8	1-3/4	
B*	15/16	1	1-1/16	1-3/16	1-1/4	1-3/8	1-9/16	1-11/16	1-7/8

*Using American Standard Taper Male Thread with Crane shoulder type Drainage Fittings. The external thread, however, should not be threaded small to gauge and not more than one turn large.

how to assemble a flanged joint

Flanged joints are made up by bolting two flanges together with a gasket between their machined faces to assure a tight seal. The proper steps in assembling a flanged joint are shown below.



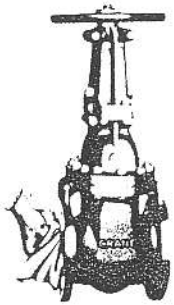


clean all parts

As with screwed joints, all parts should be clean to assure best results. Use a solvent-soaked rag to remove the rust-preventing grease put on flanges at the factory. Next, clean off all dirt and grit particles. Then wipe off the gasket just to be sure it is clean.

align and support pipe

When the pipe is in place, be sure it is properly supported. For example, a valve cannot hold up an unsupported length of pipe without undergoing great strain. The flanges should be accurately aligned—by checking with a spirit level, both horizontally along the pipe and vertically across the flange faces. It is then ready to be bolted tight.

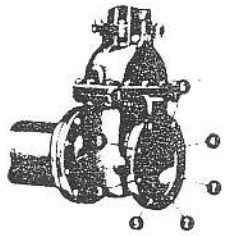


insert gasket

With the flanges secured in position, slip in half the bolts at the bottom. They will hold the gasket in place. Gaskets should be coated with a little graphite and oil or other recommended lubricant before they are inserted. They are then easier to remove if the joint is opened later. Finally, slip in the rest of the bolts, apply a little thread lubricant to each, and then turn up all nuts by hand, far as they'll go.

it's best to tighten bolts in this order

Now start pulling up with a wrench—not in rotation, but by the cross-over method to load the bolts evenly and eliminate any concentrated stresses. Keep repeating by going over and across until the joint is uniformly tight.

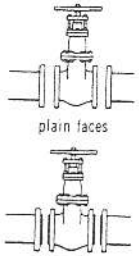


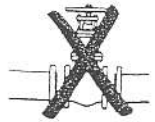
flange facings should be alike

Do not attempt to get a tight flanged joint with flange faces which are not alike. Steel valves and fittings normally are manufactured with raised flange faces. Piping materials in iron and bronze usually have plain or flat flange faces.

As shown, the proper procedure is to join a plain face to a plain face—or a raised face to a raised face. The proper gaskets will assure a tight joint. But never should a raised face be joined to a flat or plain face.

When joining 125-pound iron or any bronze flanges to steel flanges, be sure to remove the raised face on the steel flange. A full faced gasket is preferred for such a joint.





CHAPTER 13

PLANNING AN INSTALLATION AND SELECTING A CONTROL VALVE

Mathew L. Freeman
AMETEK, Inc.
Power Systems Group
Hatfield, Pennsylvania

EDITOR'S COMMENT

Selecting a control valve to serve a particular application may be, to the inexperienced, very difficult and confusing. This is because of the numerous valve types and their respective functions. The availability of many materials, sizes, pressure and temperature ratings and actuators also adds to the problem. The valve specification engineer, by review of this selection guide, can relate his process requirements with valve types suitable to handle them. By further study of the guide plus information in the other handbook chapters, the specifying engineer can be relatively sure that he is making the best functional and economic choice of valve types. The purpose of the first part of this chapter, "Planning a Control Valve Installation", is to have the valve specifying engineer make a mental survey of all service and environmental conditions which affect the valve's performance. The engineer will then be aware when reviewing the second part, "Selecting a Control Valve", to pick only those valve types capable of operating under the specified conditions.

The author has related service conditions, environment, a valve's functional performance, valve availability in required materials, size, pressure and temperature plus the right trim configuration for proper installation.

This chapter increases the usefulness of the handbook immeasurably in that a step-by-step procedure for analysis of valve type selection has been attempted. While it is obviously impossible in a single chapter to make an exhaustive analysis of all of the hundreds of valve types and variations for all conceivable applications in all industries, the methods described are comprehensive and can be easily translated into a format suitable to any application problem.

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PART 1
PLANNING A CONTROL VALVE INSTALLATION

Introduction

The purposes of this chapter is to bring the user's attention to the many factors involved in planning a control valve installation, even before attempting to select a specific valve type, and once these factors are known, to relate the different valve types to their proper applications, relative performance and cost.

First Step

When planning a control valve installation the first step is to gather together all of the information relating to the service conditions and surrounding environment. This includes not only the normal operating conditions but also conditions on startup and severe conditions during periods of abnormal operation. The forces acting on the valve through fluid conditions in the piepline, the piping itself, seismic shock and unusual environment surrounding the valve and its actuator must also be included. The skills and policy of field craftsmen must be considered in regard to factory or field mounting of auxiliary controls.

Failure to consider all of the factors often leads to the installation of valves which do not function properly during the startup period or which experience permature failure due to unusual wear or mechanical breakage.

Valve specifications for normal operation are adequately listed in Chapter 2 and are related to the body, actuator, positioner, transmitter or controller, service conditions and accessories.

The Instrument Society of America has a specification form (S20.6)[1] that can be helpful in noting conditions and valve specifications in a manner which is understood by both user and manufacturer. This will suffice for most valve applications, in that abnormal conditions may not be extreme. However until one lists abnormal conditions and evaluates them, a critical condition may be easily overlooked which may lead to future trouble. Some of these abnormal conditions with brief descriptions of the implications of what might occur in some situations follow.

The objective in citing these unusual conditions and their effect is to stimulate the user into thinking of others for his particular valve application. Think of the valve as part of an overall system and the effect of the system acting on the valve as well as the valve acting on the system. Only then may an adequate valve selection be made which will meet the greatest number of operating requirements. The following items are presented in brief form to provide the reader with a check list of material that may require an in depth study.

I - MAJOR FACTORS

System Design.

The function of a system is to control fluid flow which may be accomplished by means of; (1) a variable speed-drive pump; (2) variable volume positive displacement type pump driving at constant speed; (3) a control valve; (4) or miscellaneous other means. Variable speed pump flow control units are utilized for large flows involving high pumping energy requirements in order to conserve pumping power. Metering type variable volume positive displacement type pump units are frequently used where an exact control of a given volume of one fluid must be injected into a system over a period of time, or to mix with much larger measured volumes of other fluids. The control valve is the simplest of the three and involves the least amount of equipment. In its simplest form it may be a self-contained flow or pressure regulator whereas the valves we are discussing in this handbook receive their control signal from some other measuring device. The control valve has the greatest flexibility, that is, it may control over a broader range of high-to-low flow and rapidly change flow from one extreme to the other more readily than the first two devices. It is relatively inexpensive compared to the other two means of control, and like them, it takes a separate source of energy to operate its actuator. The first two systems put energy into the fluid to move it into the flow stream in the required amount, whereas a control valve dissipates energy from a higher level flow stream to generate the required control action. The control valve transforms part of the flow energy into massive internal friction in the flowing fluid which results in a rise in temperature and a lower pressure of the fluid at the valve outlet.

In planning your control system, the first step is to decide which type of flow control device best suits your needs, and if a control valve is needed, read further in this handbook.

[1] Part of ISA Standard ISA-S20, 1975, specification forms for process measurement and control instruments, primary elements, and control valves.

What Fail-Safe Valve Action Is Required?

Fail-safe action is defined as the position the valve will assume, that is, open or closed upon failure of the input power to the valve actuator. Study your system, and decide whetheryou want the valve to remain open to continue the flow and relieve pressure but at the maximum flow rate or if the valve must close, to prevent loss of fluid, or to prevent excessive flow. Some systems do not require any fail-safe action. Others may require the valve to remain in its last throttling position until control of the process may be handled manually. Many types of actuators are available, but the diaphragm actuator is the one most commonly used. Air pressure is used to effect the opening or the closing of the valve. The energy source that can reverse the above procedure is a compressed spring that supplies the force to shift the valve to the fail-safe position upon loss of actuator air power. Fail-safe action must be consistent throughout the complete set of auxiliary controls such as, controllers, positioners, and transducers in the control system. In other words, loss of air power or loss of air signal power must have the same effect throughout the entire system.

Another factor which must be considered is the effect of fluid forces acting on the valve stem plug assembly and their relation to the available actuator power in the form of spring thrust. Globe valves with flow under the plug are subject to an opening thrust from the buildup of line pressure. The actuator spring for a spring-closed fail-safe valve must be sufficient to overcome these forces. When flow is over the plug, a buildup of these forces tends to maintain the plug in the closed position. If a valve must fail-open, then the spring force of the actuator for fail-safe action must be able to overcome this differential pressure loading on the closed plug. A second point to keep in mind with a fail-closed system is that the liquid volume is locked up betweenthe control valve and any block valve which may be closed upstream. On long lines subject to temperature rise a pressure relief device must be provided.

Most rotary type valves tend to be closed by dynamic fluid flow forces acting upon the valve plug, ball or vane. Once the plug is in the closed position, it is generally balanced to dynamic flow forces. Butterfly valves and trunnion type solid ball valves present no problems. The segmented type, rotary plug or ball valves require special consideration in that flow behind the ball or plug will force it onto the seat and create a higher opening torque requirement than would be the case if the flow would be against the other side of the plug. This must be taken into account in the spring failure action of actuators mounted on rotary valves. Carefully review your system and determine what will happen under the maximum incoming flow rates and pressures should your valve be required to fully close or fully open for fail-safe action.

Sizing Is Essential.

This is one of the most important considerations in selecting a control valve in that neither a valve too large or too small can do an adequate control job. Precision of control is lost when the valve is oversized, and when undersized the process cannot achieve its maximum flow rate without extremely high pressure drop and the possible effects of noise, erosion and cavitation. Control valves, pumps and piping should be sized simultaneously to prevent the continued addition of safety factors, yielding an "oversized" control valve. The term "flow coefficient" (Cv) as defined by ISA is the means of measuring the capacity of one valve against another regardless of type or configuration.It is usable in this simple form only at moderate flow rates. When applying it to a system, flashing, cavitation and aerodynamic noise must be taken into consideration, along with choked gas flow conditions. All of these are adequately described in Chapters 6 and 7 of the Handbook. The central idea is that at normal operating conditions, the valve should be designed to be open approximately 65 to 75% of its full open flow capacity. If this is done then it allows you to operate your system at a higher capacity or to maintain capacity if the inlet pressure falls off somewhat. The valve would also be in its most ideal operating range for adequate flow control to a given flow characteristic: the one which you select to match your system.

Proper Flow Characteristic Must Be Specified.

After establishing the normal flow rate for the process, it is necessary to establish the maximum and minimum rates for all operating conditions including start-up. Where the maximum and minimum rates require precision control, the manufacturer should be so advised in order that he may recommend a valve which will properly handle all conditions. Standard valve designs have certain limitations in both the low range and high range of flow rate control. Below 10% of Cv valve rating, deviation from the flow characteristic may be very large. Above 90% of Cv valve rating, the rate of Cv increase with unit lift often decreases. Special high quality valve designs may be furnished to overcome these

shortcomings in globe valve applications. Rotary valves present a more difficult problem in these very low and high flow regions. In some cases, however, these may be altered by design variations to give improvement in these areas.

Extreme low flow control accuracy in a large valve is only as accurate as the means of plug positioning by the actuator. Often times positioners are required to give the desired performance without hysterisis which can cause unstable flow in these regions.

The valve manufacturer's application engineers can help the user relate the valve's published inherent rangeability to the practical installed valve turndown and rangeability.

After normal, maximum, and minimum rates of flow have been established, and from them the rangeability determined, the next step would be to critically examine the process itself to see what it's characteristics are or must be. The flow ranges of most processes may vary in the range of 25 to 200 percent of average flow. Normal pressure variation is ± 10 percent, ranging to ±25 percent of average. The object of this analysis is to arrive at the choice of a characteristic that the control valve must have. Should the valve be on-off, linear, or equal percentage? If the choice of valve characteristic is correct and matches the characteristics of the process, then control performance of the overall system should prove to be satisfactory. Chapter 5 contains practical guide-lines as an aid in making this choice.

Required Quality of Control Must Be Evaluated.

This is not only related to the type of valve selected for the service, but also to the price one is willing to pay for the quality desired.

Most valve manufacturers offer several valve lines of the same functional type as well as different types. They can relate the performance limitations of one line and type to another in order that the user may make the most economial selection to match the service requirements.

A typical example occurs when the type of valve performance at one end of the range of a line is overlapped by a higher quality line of control valves.

Compliance With Code Rules.

The various codes specify or restrict the use of materials, design rules, fabrication procedures, and quality assurance procedures for valves. Thus, both user and manufacturer must relate what can be built within code rules to the control valve application and environmental requirements. Many codes apply both to valve fabrication and application pro-

advise the manufacturer concerning the type of plant or field installation and general location in regard to populated areas. The manufacturer must know the industry classification under which the plant or process is being operated and equipped.

Most control valves are built to American National Standards Institute (ANSI) code requirements: B-16.5 for flanged end valves and B16.34 for weld end valves. In addition the ASME Boiler & Pressure Vessel Code, Section VIII, Divisions 1 & 2 and Section III for nuclear service often apply with ANSI B31 specified for some steam turbine control valves. Valve body assemblies are normally tested at 1½ times their cold working pressure rating at ambient temperature.

Valves in petroleum production service must be built to the American Petroleum Institute (API) standards because the probability of severe over pressurization is much greater. In this case it is common practice for the valves to be tested at pressures twice their normal rated working pressures at ambient temperature. Separate API standards are published for Production, Pipeline and Refinery service.

Piping Code ASME B31.1 is divided into: 1) Power Piping, 2) District Heating and Refrigeration, 3) Oil Transmission Lines Outside Village Limits, 4) Oil Lines Within Refinery Limits, 5) Gas Piping Systems Within Village Limits and 6) Gas Transmission Lines outside of highly populated areas. Each division has different maximum operating temperatures and pressures for a given piping thickness of the same material. This affects the types of nipples and reducers welded to valve bodies by the manufacturer.

ASME Section III for nuclear service is specific as to the materials and stresses allowed for pressure-containing parts: the body, bonnet and bolting. The valve plug is also included in this category. The design rules, approved materials, stress levels, and quality assurance requirements vary depending whether a valve application is classified as Nuclear Class 1, 2 or 3 depending upon the importance of its function in the system. Is it the responsibility of the user or his architect engineer to specify in the Design Specification, code-approved materials.

The users valve design specification must be a formal document certified by a licensed professional engineer familiar with nuclear power plant design. ISA valve specification sheets may be included to give the source of the essential data. Once accepted as being in agreement with the Nuclear Code, Section III and possible to manufacture, then all changes by either party must be accepted,

II - PIPE LINE FORCES

Piping Forces Acting Upon the Valve.

Bending moments, compressive and tensile forces acting upon the body and end connections of the valve are generated by cold start-ups and by significant temperature and pressure changes during operation and shutdown.

Normally this presents no problem with full "Line Size" valves except in severe cases, since the ANSI B16.5 and ANSI B16.34 Codes for flanged and weld end valves specify sufficient body wall thickness to make the valve body stronger than the piping.

Flangeless valves are limited in size and pressure rating by the manufacturers so as to provide adequate strength of the assembly and retain gasket tightness.

Consideration should be given when installing less than line-size valve bodies, particularly when the valve body is more than one line-size smaller. The user must evaluate the need of installing piping anchors to isolate the valve body from excessive forces.

Piping Forces Generated by Valve Action Combined With Service Conditions.

1.) Strains on Downstream Piping

Rapid opening of valves whose bodies are drained by steam traps so as to remain at operating line temperature while the valve is closed, results in sudden thermal strains to downstream piping. Valve body strains are minimal in most cases. Frequent operation and the reoccurring thermal ratcheting effect may cause the bolting to creep and lessen gaset sealing. Continuous preheating or cooling may be required in extreme cases. Heat transfer across the bridge wall of the body, ball or butterfly disc causes much less effect on the downstream part of the valve.

2.) Poor Drainage of Body

When the low part of the valve body is not drained, condensate slugs may exist in the upstream and /or downstream flow line. High velocity gas or steam eventually may pick up the slug, propelling it with sufficient velocity to damage piping and some types of valves. Lines should be adequately trapped and drained. Even the accumulation of liquid below an orifice plate in a horizontal line has been known to fracture rotary valve plugs.

3.) Quick Valve Openings

Fast opening of large valves at high differential pressures causes a strong line reactive force. Piping is displaced and vibration and loosened pipe anchors may result. Steam dump valves have been known to back the discharge nozzle out of the collector vent pipe with resulting loss of life. Piping must be securely anchored. Several valves opening in parallel have damaged large headers.

4.) Quick Closing of Valves

Fast closing of valves in liquid service causes water hammer capable of building sufficient line pressure to rupture iron valves and fittings and to separate steel flanges and blow gaskets. Slowing valve closure time by sizing pilot orifices in pneumatic control equipment and adding a throttling contour to the first 15% of stroke of a quick opening type plug may be utilized to greatly reduce this water hammer effect.

5.) High Energy Service

In high energy service, consideration should be given to the type of closure which may occur from a secondary equipment failure such as actuator linkage, spring, or power. Most rotary valves are dynamically closed by flow, and if freed to rotate, close rapidly with severe line shock.

6.) High Flow Rates

High flow rates accompanied by high pressure drop in the valve often create flashing and cavitation which causes severe vibration in downstream piping. In these cases keep downstream pipe runs short, have them enter a large vessel as soon as possible and have as few bends as possible in the line.

7.) Diaphragm Valve Types

Diaphragm valves; pinch valves, and sleeve valves all displace fluid into the line upon closing. They must not be used on dead end service with block valves. Either the valve will fail to close, or if actuator power is sufficient, will rupture the diaphragm.

Valve End Connections Must Match Piping Design.

End connections on valves are usually specified by the piping design engineer. He can recognize the pipe forces that will act on the valve body; therefore he should also be made aware of the forces generated in the piping by valve action.

The type of joint is selected while considering the required strength to maintain the desired gasket compression sealing stress under all operating conditions and the degree of

gasket seal tightness as related to toxicity, amount of product contamination allowed on vacuum service, fire hazard and human burn hazard. Erosion and crevice corrosion may also be factors helping to establish the type of end connection chosen.

For example, the National Association of Corrosion Engineers (NACE) Specification 150 for high pressure high content hydrogen sulphide service requires flanges to be cast or forged integral with the valve body. Crevices are not allowed. Socket weld ends would not be permitted on sizes 2 inches and smaller. Butt weld joints should be used. Weld ends on chrome-moly steel bodies are often welded to carbon steel piping. These bodies are specified for high temperature service and to prevent erosion on flashing service. Welding of the chrome-moly or any piping wall thicker than 1 inch requires stress relieving which should be localized to prevent body bridge wall distortion in globe valves resulting in an ovalized seat joint.

Rather than performing stress relief locally at the joint in the field, many purchasers buy their valves with a carbon steel nipple attached at the factory.

The above practice is also strongly recommended for installations where the piping is Chrome-Moly. This allows furnace stress relief of the body-nipple or reducer assembly (all chrome-moly). The body is then machined for the bonnet gasket and seat joint and is in true alignment.

In corrosion resistant service involving lined bodies, (bodies lined with TFE or other elastomers) care must be taken to insure that the protected end of the body makes up with the protected end of the piping with adequate gasket loading. It is also important that this loading is limited, so that bolt tightening does not extrude the liners excessively. This is the responsibility of the piping designers. Field pipe fitting foremen should be given adequate instructions.

Seismic Forces Strain Actuators and Valve Bonnets.

Valve specifications, particularly for fossil fuel power installations and nuclear power installations, require that the Valve-Actuator-Auxiliary Equipment Unit withstand seismic shock loading along its three principal axes and remain in operating condition. Loading varies with valve elevation in the plant and in some cases goes as high as 3g's. Actual loadings from a 3g input to the valve body may go much higher in the valve actuator, depending upon how well a valve is supported. If the assembly has no calculated frequencies below 33 cycles per second, then a static analysis is acceptable. If low frequencies do exist, actual testing and dynamic operation may be specified since strains are much greater.

Siesmic Forces

Seismic forces vary the bonnet flange bending stresses and bolting strains and sometimes it is necessary for the valve designer to increase the dimensions of these parts. Actuator weight and overhand, including the positioning of auxiliary equipment, may limit the choice of actuator.

Momentary upset of control during seismic shock may be allowed with rapid return to normal operation after the occurence of shock. False indications from electric valve position switches have occurred. Field practice for valves with long actuator extensions is to support the actuator with an anchor which is rigid with the immediate main line piping.

Physical Shock

Severe shock; such as might be experienced in a plant explosion, may require that all valve actuator yokes and brackets be made of ductile materials to prevent failure. Shock may also permanently displace instrument set points and instrument mountings are often loosened affecting the control setting. When in doubt, specify shock testing, but be prepared to pay the higher costs. Most valve designs when closed and pressure loaded will remain tight without loss of pressure during shock. Partially open valves may momentarily change plug position but usually rapidly recover. The valve itself is a rugged member; and failure if it occurs; is usually in the actuator or instrument mounting.

III-PRESSURE CONDITIONS

Pressure Cycling causes Fatigue of Parts and Poor Control.

Rapid pressure fluctuations of great magnitude will fatigue the bonnet bolting and gaskets. Cumulative pressure cycles over the expected life of the valve should be reported to the manufacturer in these cases. On dead end service, butterfly valve shafts and ball valve trunnions and body joints are affected. Control action may be unstable.

Bonnet stud fatigue may occur in valves controlling the discharge pressure of high pressure duplex or triplex reciprocating pumps. Pulsation fatigue tests have shown that an adequate stiff bonnet design and alloy steel bolting stressed to 45,000 psi tensile stress give satisfactory long life. Torque wrenches should be used when remaking the joint in field maintenance to achieve uniform

loading of all studs. Pressure snubbers (hydraulic type) may be required on the valve stem to stabilize erractic pulsations and line pressure variations.

Installation of flow line surge chambers (as shown in Figure 1) between the pump and valve are a great help. These may be "tuned" by varying their air pressure loading and throttling the liquid input and exhaust with a hand valve. Otherwise, high pressure pulses may readily travel through a partially open control valve controlling pressure to an almost dead end system.

Hazards Of Excessive Differential Pressure

Differential pressure usually has little effect on *globe valves* with the exception of those with flow under the plug where differential pressure may overcome the actuator force and cause a false opening. The safety hazard of this condition should be evaluated. Flow over the plug simply forces the plug tighter on the seat of an unbalanced globe valve and seat damage could occur on very narrow seat joints by the plastic yielding if it is of a soft material or cracking if made from hard or brittle materials. Conversely the actuator might not have sufficient power to open the valve if fail open action is required.

Excessive pressure differential across *butterfly valves* may result in a bent stem or catastrophic failure of the pressure boundary upon stem shear. *Ball valves* generally seal tighter, although the floating ball type may damage the downstream seat. Trunnion designs may suffer bearing damage or bent trunnions. *Plug valves* can withstand excessive differential pressure loadings with little damage or leakage; however operating torque increases.

Unusual Effects Of Excessive Line Pressure

Globe valve stem thrust may prevent a low force actuator from closing the valve or cause the plug in a throttling valve to go wide open and create excessive downstream flow. Diaphragm valves may rupture their diaphragm and allow flow downstream to occur. Major external leakage is retained by the metal bonnet in these cases.

Valve bodies will often withstand pressure of several times their ratings before rupture. Piping would normally rupture first. However, before piping rupture, the bonnet studs usually yield and the bonnet flange rotates allowing bonnet leakage or in extreme cases a blown gasket.

Pressure sealed bonnets utilizing a soft metal triangular seal may be used in hazardous exothermic processes on valves 6 inches or larger. This joint will remain tight until the bonnet neck yields at extremely high pressure prior to body rupture and then the entire bonnet blows out. Safety heads with rupture discs are usually installed in these processes to prevent the type of failure just described. Pressure sealed bonnets operate best beginning with the ANSI 900 lb. class valve. Lower ANSI ratings do not provide sufficient thrust on the joint to seal effectively.

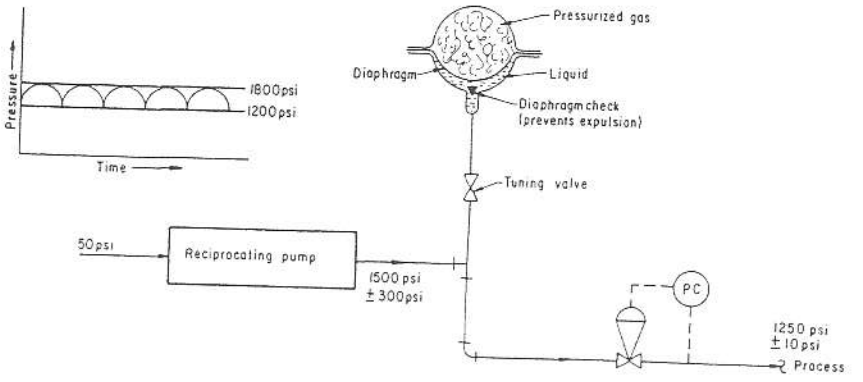


Figure 1. SURGE CHAMBER.

IV-THERMAL EFFECTS.

Temperature Cycling.

Rapid temperature changes of great magnitude create a thermal ratchetting[1] effect which is capable of eventually stress rupturing both body and bolting. Figure 2 illustrates the thermal ratchetting effect. For severe cases advise the manufacturer of the cumulative frequency over the life of the valve; the magnitude of change, and the rate of change with time.

It is also important that the manufacturer have this information for his designers in order to relate the change in valve operating clearances during these thermal cycles. Some internal parts reach operating temperature sooner than others and differential thermal expansion of different trim materials must not be allowed to effect an interference or tight bearings.

Peaking Temperatures.

Time of operation at excessively high temperatures, well above normal operating conditions must be evaluated by the manufacturer with respect to maintaining adequate operating clearances, prevention of galling, and softening of hardened trim without recovery upon cooling.

Many elastomer and plastic compounds used for seals can operate above the temperature ratings established by material suppliers for long service life when high temperature exposure time is limited. Periods of hours to several days at these higher temperatures may cause only minor deterioration. Consult the valve manufacturer in in each case. Published data by the material supplier on most compounds does not represent conditions of valve operation and therefore is not usable. Frequently, long service life ratings established by the valve manufacturer for continuous exposure are at lower temperature and relate to specific fluids.

Ambient Thermal Effects on Double Seated Valves.

Double seated ball valves and plug valves will lock up in the closed position and become imovable if liquid expands in the body cavity on temperature rise from the sun or nearby heat source. Most designs, however, have features which relieve locked-in' pressure downstream; however codes require the purchaser to specify this feature when ordering.

In liquid service, when a block valve is closed upstream of a globe control valve with flow over the plug, pressure relief must be provided to prevent thermal expansion from creating excessive pressures. This is of particular importance in long flow lines.

Formation of Hydrates May Block Valve.

The expansion-cooling effect in gas throttling valves often freezes water or hydrates in the valve body thus blocking flow.

These situations may be remedied by:

1.) Preheating the fluid before expansion.

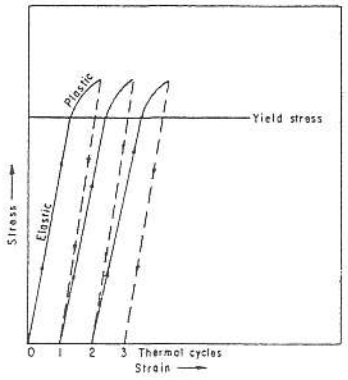


Figure 2. THERMAL RATCHETTING EFFECT.

[1]Rapid differential thermal expansion causes local yielding and permanent specific displacement at the rate of one minute step with each thermal cycle.

2.) Injecting ahead of the valve, compounds which lower the hydrate formation temperature.

3.) Jacketing and heating the valve body.

4.) Projecting the valve into a warm vessel. If the valve has an extension type body, place plug and seat throttling region inside the vessel.

Minimum Temperature Effects.

Specification of minimum temperature, frequency, and time at this teemperature allows the manufacturer to evaluate the low temperature impact strength of both trim and pressure containing parts. Frequent or extended operation at low temperature will require the specification of a material having adequate cold impact strength. Rare low temperature excursions of short duration would only require this in very hazardous service. Elastomer seals harden, take a set, then shrink. As a result, the initial radial seal between seat ring and valve body may not be achieved in ball valves, and between disc and body in butterfly valves. Metal spring loaded seals may be used to overcome this effect. The differential pressure required to effectively seal the upstream seat of a trunnion ball valve increases. Elastomer stem packings should be of the pressure energized type and should also be spring energized to seal as pressure is admitted to the valve.

Cryogenic Temperatures.

Valve body weight is critical in order to prevent excessive flashing in liquid service on startup as the valve body operating temperature is lowered. This avoids excessive loss of cryogenic liquid.

Whenever possible, place the valve stems vertically or at a 45° angle to create a vapor lock in the bonnet and therefore reduce the rate of heat transfer. This will prevent freezing of the stem packing where it is exposed to atmospheric moisture. Should it be necessary to place the valve on its side, vapor seals may be installed which seal in the direction of the bonnet and relieve pressure in the direction of the body.

In liquid oxygen service the user should specify to the valve manufacturer any high energy mechanical shocks which might occur within the valve piping system. This could be, for example, a rapid valve plug closure on foreign metal objects lodged in the seat. The manufacturer will then base his choice of materials to eliminate a fire hazard condition, because high energy impacts may initiate an oxygen fire in the system. Stainless steel valve body and trim materials are satisfactory in most liquid oxygen applications; however in a few cases a considerable amount of plant equipment has been destroyed by oxygen fires initiated by high energy impact. Users may specify brass or Monel body and trim materials to prevent sparking from occuring during a high mechanical impact. All pressure containing parts must be of material which is not brittle at minimum operating temperature. Refer to Chapter 10.

V - SURROUNDING ENVIRONMENT.

Cold Environment.

Many valves operate outdoors on cold plant startups in northern latitudes where temperatures may be a -20 to -40°F. Other valves operate in the arctic where start-up temperatures may be as low as -60°F. Always advise the manufacturer of your start-up temperatures. Some valve body materials have have low impact values and therefore other alloys must be furnished. In a cold environment, trim materials such as the 400 Series stainless steels and tungsten carbides for erosive service are brittle and normal plug closure may chip and fracture the plug or seat ring. For extremely cold service, electro mechanical actuators normally require electric heaters to maintain the sealing elements in a pliable state and to keep grease and oil viscosity low. Elastomer sealing elements within the valve require adequate metal spring backup to remain tight at low pressure at temperatures below -10°F. This is of particular importance in rotary type valves. Breakaway operating torques of ball and butterfly valves may be double that experienced when operating at 70°F and adequate actuator power must be provided.

Electric actuation is preferred for cold climates in order to prevent moisture freeze-up of instrument airlines. Pneumatic controls may be used, but these are usually housed in heated structures. Self draining, or drain tapped valve bodies and control lines must be provided for cold climate operation. In low ambient temperatures actuator diaphragms stiffen and excessive hysteresis will occur in valve operation below -40°F. Silicone diaphragms may be used at lower temperatures. A stiff diaphragm may tear rather than roll and stretch slightly as it does in normal operation.

Hot Environments affect the Actuator.

The principal problem with high ambient temperatures is the limitation of the material used in the diaphragm actuator. The use of silicone and Nordel rubber diaphragms can solve some of these problems. For extreme ambient temperatures above 300°F a metal

bellows diaphragm may be required. The latter is quite useful in areas of high radioactivity which would deteriorate the elastomer diaphragm materials or limit their usable life. These same comments apply to diaphragms of the various instruments associated with control valves.

In nuclear service, tests simulating Loss of Coolant Accident (LOCA) in regard to pressure and temperature, but without radioactivity show that adequate LOCA service may be obtained from Nordel type rubbers. This of course is a relatively short time service where the temperature is above the long life temperature specified for this material by control valve manufacturers. The initial LOCA (55 psia 310°F) drops in 30 minutes to 200°F. Thermal effects on plastic components of auxiliary electrical equipment should be evaluated.

Actuator Protection Against Miscellaneous Environments.

Humidity, such as experienced in semi-tropical environment causes an excessive collection of moisture in any vented space such as actuator spring barrels. Over a period of time these spaces will fill with condensed moisture. Unsealed electrical equipment proves to be totally unsatisfactory and even sealed equipment with vents will normally break down by electrical shorting after approximately three months. Electrical equipment must be totally sealed and preferably heated, although the latter is not essential. Electrical terminals, connections and wiring should be given an anti-fungus treatment since fungus growth (slime) will readily destroy the electrical resistance and strength of plastic components. This can occur while equipment is in storage in the warehouse or field before actual installation. All enclosed spaces which are vented must have screened vents to prevent insects such as dirt dobbers from building their nests inside. An actuator case spring can be immobilized when packed solid with the mud of dirt dobber nests. Instrument pilots have been plugged with insects and cocoons. Make sure all vents are plugged during equipment storage and vent screens are in place when the equipment is installed.

Physical Protection of the Valve Actuator.

Many valve applications require fail-safe action for emergency operating conditions and in these cases, where structural damage may occur around the actuator, the actuator-valve-stem-unit must be protected from physical damage which could cause stem bending and valve stroke jamming. Valves with "breakaway actuator stems" are designed to fail in the closed position. Rotary valves sometimes use a hydraulic vane actuator to prevent sudden or rapid closure by limiting the exit of hydraulic fluid from the vane compartments. Both rotary vane actuators and piston actuators for high energy service applications normally have flow control devices imbedded in their metal body components so that these cannot be broken off by outside physical forces. These controls are usually free flow for the inlet with adjustable restricted flow for the outlet.

Corrosive environment protection.

The valve body actuator stem connection and open yoke diaphragm construction may be subject to severe corrosion in certain chemical and marine services. Marine service valves often have enclosed yoke barrels which are vented in in order to prevent excessive pressure buildup from possible stem packing leakage. Other diaphragm actuated designs are available with the actuator connected directly to the valve bonnet, or forming part of the bonnet so that this exposed connection is eliminated. Surrounding corrosive environments may also affect valve positioners, transducers and other equipment which should be sealed and if necessary, pressurized with clean air.

The Control Valve in a Hazardous Environment.

Vapor proof, intrinsically safe or explosion-proof equipment may be required either due to severely corrosive or explosive atmospheres. These conditions may exist continuously or intermittently. Explosion proof equipment is bulky and switches normally have high high operating forces. This may restrict its use on some smaller valves which have insufficient actuator power or inadequate space for a satisfactory mounting. Intrinsically safe equipment may be used for these cases.

VI - FLUID EFFECTS.

Erosion.

Fluid particle erosion is a function of many variables: particle size, content in the fluid medium, density, velocity, degree of turning in passing through the valve, and valve operating requirements to control the process. As a consequence, only general guide lines can be given by the manufacturer. There is no substitute for actual experience with equipment under similar conditions. Frequently the best valve for a given highly erosive service is determined by plant operators simply installing one after another until they find a product with satisfactory life span. Valve manufacturers follow this testing program and often offer improved trim which in some cases have phenomenally increased valve life.

Operating Wear.

The number of complete operating cycles a valve goes through greatly influences the valve's total service life. In some cases valves dumping high pressure erosive fluids may full stroke many thousands of times per day where valves in other services simply throttle with a few on and off operations. These conditions determine the type of valve trim and actuator to be specified for best service and therefore the manufacturer should be advised when a considerable amount of valve stroking or closure is required by the process.

Internal Corrosion.

The valve manufactuer can can only recommend body and trim material for general service conditions, thus it is the responsibility of the user to specifically advise the manufacturer of materials which have satisfactorily withstood his type of service, if this information is available. The variety of conditions, and the variation in corrosion rates by slight changes in these conditions makes this a difficult problem which in most cases can only be resolved by a knowledge of prior experiences.

Metal alloy insert linings, elastomer linings, and glass linings are available for the internal surfaces of some valve designs to eliminate corrosion. The user must evaluate the hazard of slight failures in these linings and corrosion of the supporting body material. Elastomer and glass lined products may be inspected by the electrostatic method using a high voltage spark discharge tool. *Crevice corrosion* is a problem in some services and this calls for the elimination of all socket welds and body crevices in the valve design. This is particularly important in high pressure service with hydrogen sulphide content and is covered in Specification 150 issued by the National Association of Corrosion Engineers. Corrosion is often related to *fluid velocity* and the erosive content of the fluid. The protective film normally formed on the metal surfaces is rapidly worn away, exposing fresh metal. Under these conditioons, corrosion may proceed at an extremely rapid rate because normal testing, where velocity is not involved, would indicate an adequate material. *Stress corrosion cracking* is a particular evil in many services including use of 316 stainless steel for fluids containing chloride ions. Valve plugs and stems have been known to fail in a matter of hours or weeks from chloride ion initiated stress corrosion cracking in certain services. *Line cleaning fluids* should be evaluated with respect to their corrosion of valve trim. The same thing applies to corrosion inhibitors often injected into the line ahead of the valve which could damage the elastomer seals should the concentration of the inhibiter be too high.

Noise Considerations.

Aerodynamic noise predictions have reached a state of development where they are feasible to be made either by the user or the manufacturer with a fair degree of accuracy.[1] Where acceptable noise levels are exceeded, consider the time that plant personnel will be in this area to see if low noise valves are justified, since they are considerably more expensive than standard equipment. Low noise valves generally have lower flow capacities than standard valves. Their size increases tremendously when discharging to atmosphere or to very low pressures. When noise treatment is to be handled by means of silencers, be sure to allow adequate piping length for their installation. The small flow passages in some designs of low noise valves may become plugged by line debris, thus reducing the valve's flow capacity, The concentration of these particles expected to enter the valve body during its normal life should therefore be considered. Designs are available with large flow passages which do not have this problem, however their ability to lower noise is sometimes restricted by the size of the valve body required to accommodate this type of design. Refer to Chapter 7 for more information.

Valve Plugging.

Gunky materials, precipitates, crystallization of compounds and heavy scale deposits may occasionally plug the valve body flow passages in such a way as to interfere with operation of the valve. Smooth bore body designs with few dead end passages are available to help overcome these conditions.

There is perhaps an overconcern with the possible plugging of vent passages in pressure balanced plug construction. However, keep in mind that this is practically a dead-end service to the cavity above the plug, and with

[1] 1973 accuracy ranges are ± 2 to ± 5 dB.

the valve in a vertical position any particles which move into this area are generally flushed out on the next stroke. Balanced plug valve designs should not be used in services where products polymerize and build up within the valve body, or where extremely heavy plugging would occur from other sources. Petroleum service valves have been torn down which were completely blocked with paraffin and sand that was compacted into the flow passages and yet the balance portion of the trim was free to operate. Special non-closing automatic tank bottom drain valves are available which will pass a considerable amount of sludge without blockage.

Miscellaneous Fluid Effects.

Viscosity affects the choice of valve type, trim selection, and flow capacity for very viscous fluids. Valve body styles with fewer turns in the flow path and relatively large widths for annular flow areas should be selected.

The vapor pressure of the fluid should be considered. Fluids with low *vapor pressure* may readily flash or cavitate when the pressure in the vena contracta at the throttling orifice of the valve approaches or falls below the vapor presssure at operating temperature.

Formulas are available to predict cavitating conditions, and one of the factors contained in these formulas is the vapor pressure at the flowing temperature. Having predicted cavitation, it is then possible for the valve application engineer to eliminate or greatly reduce cavitation by changing the type of valve or certain operating pressure conditions. Chapter 6 has more information about this.

Very light fluids are more difficult to seal, whereas very heavy fluids would present a greater shock for a given flow velocity upon sudden valve closure. A *specific gravity value* would therefore be helpful.

The degree of flashing is affected by *super heating* and *pressure drop*. Considerable noise and vibration in the downstream piping may be created if these factors are not considered. Flashing flow often reduces a valve's flow capacity, however this is not completely predictable. Refer to Chapter 7 for further details. *Iron body valves* are rapidly eroded in flashing service; steel valves also erode, and Chrome-Molybdenum steels are normally required to give satisfactory life.

Food Processing Valves.

An entirely separate set of valve designs is required for this service so as not to allow clogging, fermentation or degradation to occur while allowing the valve to be readily disassembled for cleaning.

Further, the Food and Drug Administration regularly publishes information on the type of chemicals which are prohibited in materials used in food processing equipment and also the maximum allowable percentages of certain other chemicals. This affects the materials used in compounding the various elastomer seals for ball, butterfly, and soft seated globe valves. It is the manufacturer's responsibility to be cognitive of these changing requirements. The user should keep in mind that there are no published specifications, only requirements that certain chemicals may not exist in these elastomers, or if they do, that percentages shall not exceed prescribed limits.

Special Types of Fluids.

Thixotropic fluids tend to create valve instability problems when changing from one form of flow to the other. A thixotropic fluid is a gel which liquefies when subjected to the action of vibratory forces, such as ultra-sonic waves, or even shaking. Upon standing they will again resume their gel form. This occurs when valve throttling changes flow from very low to higher rates and vice versa. It is best to maintain a continuous low flow. Valve actuator position stops and or plugs with drilled holes may be used.

VII-SEALING.

Allowable Leakage Rates.

The user should determine the maximum allowable leakage rate he can tolerate or the degree of tightness required over the expected life of the valve prior to trim replacement maintenance. This serves as a basis for selecting the proper type of valve to meet these conditions. Factory leak test rates of some valves are maintained in the field for long periods, whereas others are quickly scored or slightly damaged by repeated closures or erosion such that the leaks in normal service are higher. Valve manufacturers can provide relative information as to the degree of tightness which may be expected for different service periods and different environments.

Stem Packing Materials.

Many types of packing materials are available. *TFE packings* give the lowest friction, lowest valve hysteresis, and very tight seals.[1] *Graphite-asbestos* packings are suitable for higher temperature service and may be reinforced with Inconel® wire. The graphite-asbestos packing leak rate is considerably higher than that for TFE packing when hysteresis is of an equal value. When graphite-asbestos packing is tightened to approach the TFE value of tightness, hysteresis becomes excessive and unstable valve operation usually results. Newer *graphite ribbon* packings offer solutions to high temperature packings. These packings must be broken in for several hundred cycles before hysteresis is at an acceptable level. Some instrument men do this in the shop before placing the valve on the line, whereas others place the valve on the line and bypass it while they cycle it to break in the packing. *Plastic packing* may be injected through a lubricator fitting and retained in the packing box by double chevron rings set back to back. Too high an injection pressure builds up pressure on the packing in the box which has no mechanism for excess pressure relief. Stem friction then becomes so great, the valve jams.

Rotary valves handling Liquified Petroleum Gases (LPG) should have their bearings set outward of stem seals if roller bearings are to be used. The bearings must be selected and a bleed device provided between the stem packing and bearing seal. LPG products permeate through rubber sealing elements and if not bled off to atmosphere, will penetrate the bearing seal and dilute the bearing lubricants causing bearing failure and a jammed valve. Propane at 1000 psi line pressure will diffuse through rubber duck chevron packings at the rate of 50 to 100 ml per day with the valve stem remaining static.

Low Pressure and Vacuum Service.

Elastomer stem seals must be adequately energized by metal springs in order to seal tightly. When service varies from pressure to vacuum, two sets of chevron type packing may be installed on either side of the lantern ring, back to back, with the top set restraining pressure from the body to atmosphere and the lower set preventing leakage of atmosphere into the body when on vacuum service.

Elastomer Seals Absorb Gases.

Certain elastomer materials absorb gases at high pressures and when valve pressures return to much lower values, release of the gas may blister or rupture the elastomer seals before the entrapped gases can escape. The gases exist in the elastomer interstices in the molecular form rather than bubbles and do not interfere with high pressure sealing operation. A typical example of this is illustrated by a valve operating at 4000 psig with high carbon dioxide (20 to 60%) in the gas flow stream. When shutting down and disassembling a valve that has been used under these conditions, many elastomer seals will pop and blister and rupture their surfaces for several hours after being removed from the valve. Depressurize these service lines gradually to prevent seal damage.

Bellows Stem Seals.

Bellows Stem Seals are used to contain hazardous fluids, or fluids which would deteriorate the normally used packing materials. They are frequently used in radioactive service or in vacuum service to prevent air contamination in the line. Bellows seals are expensive, have limited stroking life, require higher actuator forces to overcome the bellows spring rate, and should be avoided when not essential. Their cycle life depends upon full stroke or partial throttling stroke and few bellows manufacturers will specify a life without proof testing a group of bellows under the exact pressure and temperature of operation. High pressure long life designs are now available; these require long bonnet extensions for installation in a valve. Extra head room should be planned for installations where this type of stem seal is used. The bellows seal is usually backed up by a set of packing with a "tell tale" leak off between the bellows and the packing. A pressure gage is attached to the leak off fitting and is used to indicate bellows failure.

Seal Welded Bonnets.

These are used in toxic or hazardous service where bonnet leakage could lead to serious complications. They are also used on some small screwed bonnet type valves in steam service. The seal weld is simply a projection of two thin metal lips, one from the body and one from the bonnet which are fusion welded without the addition of filler material. Seal welds are usually tested by means of a mass spectrometer and a helium sniffer tube. Figure 3 illustrates the seal-welded bonnet designs.

VIII-VALVE ACTUATOR.

Valve Actuator.

The type of actuator to be selected depends on many factors: force required, type of input power available to the actuator, action or lock-up in case of emergencies, or speed of action, frequency and cycle life before component failure, and/or need of manual overriding handwheel control. The stability of the actuator may be increased by higher spring rates or higher air pressure loadings. In case of severe instability, cycling, initiated by fluid pulsations acting within the valve body on the plug-stem assembly may be reduced by

[1] 10^{-3} to 10^{-4} ml/sec for gases.

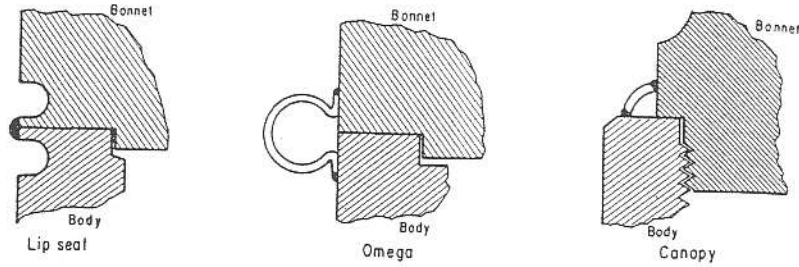


Figure 3. SEAL WELDED BONNET DESIGNS.

hydraulic snubbers mounted on the actuator stem. On-off type electric actuators are rated as capable of continuous cycling or intermittent cycling. In the latter case it is advisable to have a time delay device installed to prevent restart so early that continuous cycling would trip the electrical reset relay protecting the actuator motor from burnout. Motor winding temperature sensitive switches are available to perform this function and will reset upon adequate cooldown.

IX - AUXILIARY ITEMS.

Compatibility of Auxiliary Equipment.

When multiple auxiliary items are to be mounted on the valve actuator, the manufacturer needs to review the compatibility of these units from an overall control system viewpoint and also the availability of physical space to adequately mount them. The controller response to a controlled variable input signal must be faster than the valve actuator response. A type of emergency action should be carefully evaluated for all components. Overriding controls such as electrical signals, pneumatic valve shifts and a bypass for manual operation should be specified.

Valve Position Indication.

In addition to end-of-stroke postion signalling, many instrument engineers like to have the valve's throttling position indicated on a central panelboard. This may be readily accomplished either electrically or pneumatically by a position transmitter attached to the valve stem. Switches must be allowed sufficient switch arm travel for the tripped switch to reset its position. Sometimes short stroke globe valves may magnify the stem motion by means of levers to accomplish this. In other cases it is quite impractical to do so. More expensive sealed-type proximity switches may be used in these applications, although they require additional auxiliary controls.

Valves which stroke to a fixed closure position present problems in that it is difficult to set a switch to trip precisely at the moment of closure when there is no overstroke of the valve. Special globe valve plugs, with built-in overtravel controlled by Belleville springs, overcome this difficulty. Rotary mechanical actuators often build up torque after the closure. The torque is against a mechanical stem stop, allowing the worm drive gear to advance along a shaft compressing a spring so that there is additional motion to insure a switch trip-out. Maximum operator torque is also controlled by this means. Butterfly valves, including lined types, and ball valves have sufficient variation in closure position to allow positive switch tripping.

Handwheels and Stops.

Selection is based on accessibility. The valve's position in the piping and walkway areas is also important when choosing an overriding manual handwheel. Choose the top mounted handwheel for valves mounted relatively low. The more expensive side mounted handwheel should be chosen for valves at higher elevations. The side mounted type may also be operated by a chain fall plus chains to release and rest the locking levers. Yoke designs, incorporating a worm gear drive, have low operating torque and tremendous power output. They require many turns to accomplish closure, which may be a factor to consider for an emergency manual operation.

Valve position stops are often required to limit maximum or minimum flow rates and may be installed on either linear or rotary motion valves. Stops are often available in both opening and closing directions and in addition to handwheel operation.

X - PLANT FACTORS.

Installation Effects

Globe style valves operate with longer packing life and less hysteresis when installed in the vertical position. This does not however mean that proper operation cannot be obtained when they are mounted horizontally. If the horizontal overhang of the actuator is so great that drooping occurs, binding the stem, it will be necessary to support the actuator case.

Characterized ball valves with thin metal seats operate best with the shaft in the horizontal position. This eliminates the weight of the ball having to be forced to the center of the seat orifice by a thin metal seat as would be the case should the valve stem be in a vertical position. Tighter sealing and longer seat life result with horizontal shaft mounting.

Flangeless valves use long through-bolt studs from one pipeline flange across the valve body to the other. When the valve is installed with reducers, that is, expanding from the valve body up to a larger size flow line, it is necessaary to check to make sure that adeqate stud removal distance is available. This is generally no problem when the valve is only one or two sizes smaller than the piping. If the difference is more than two sizes smaller, it may be required that a straight pipe length extend from the valve before being enlarged back to the main line piping. Turn buckle bolting is available at extra cost when this is not feasible. When installing short length flangeless type valves it is necessary for the pipe designer to make sure that the vane or ball will clear the piping. When using heavy wall piping it is sometimes necessary to counterbore the flanges.

When welding valves with *elastomer seals* into flow lines, care should be taken that welding heat does not damage the elastomer. Adequate body extension past the seal area to the point of welding or extension nipples attached to the body at the factory prior to assembly may readily overcome this difficulty. If in doubt, and it is possible to disassemble a bonnet type valve in the field, then the seals may be removed and reinstalled after welding.

Valves must be installed in such a manner that they can be *safely serviced.* This means there should be safe access, adequate clearance to remove parts, and to allow draining of hazardous fluids from the valve manifold. Make sure adequate drains, pressure gages and thermowells are installed in the control valve manifold piping.

Valve Protection During Plant Startups.

Many processes, particularly power generation, start up cold with high pressure drops and low flows which cause the valve to experience cavitation and flashing of a more serious magnitude than that experienced under actual operating conditions. The user should carefully review his startup conditions and the number of times these conditions will occur, particularly the initial startup which is normally much longer than normal plant restarts. The manufacturer should be furnished this information in order that he may provide a valve which will adequately withstand these conditions as well as the long time standard operating conditions. Hot cleaning flushes may be made with valve trim removed to eliminate the possibility of seat joint damage by welding beads, scale and sand.

Maintenance Planning Pays Off.

Planning activities for a valve installation should involve thought about the subject of maintenance. For example, it should be known beforehand where, how, and by whom the valve is going to be maintained. Early settlement of these points can be quite advantageous since they can influence the answers for these two questions: What type of valve should I select and how should I install it? Some types of valves are serviceable in the pipe, some must be removed from the pipe and serviced in the field, while others will be serviceable only in a maintenance shop. Relating to this concept, a valve maintenance location versus valve type chart can prove useful. It lists the components which may be removed for each particular location for adequate servicing. The chart is found in PART II of this chapter.

In the planning stages on the subject of maintenance it is advantageous to approximate the life span of the trim parts of the valve. Try to develop an economic evaluation of several alternatives, if possible. For example, would a higher-cost longer-life type of trim be any more economical than frequent replacement of standard type trim? The anser might well hinge on whether the trim replacement is serviceable with the valve in the pipe, or whether it requires servicing to be done in a maintenance shop.

IX - SUMMARY.

Summary.

With adequate planning now accomplished, proceed to read Part II of this chapter. It concerns the selection of control valve types that best suit specific applications. Factors discussed in Part I which directly apply to your valve application may be pursued further in other chapters of the handbook. Knowledge of your process and industry practice, combined with the control valve manufacturers experience should be sufficient to evaluate the possible effect of the many factors on a specific valve application. A correct valve specification can be made, provided pertinent information is at hand and

PART 2
SELECTION OF A CONTROL VALVE

INTRODUCTION

The following section deals specifically with all types of valves such as globe, ball, butterfly, rotary eccentric plug, diaphragm and other miscellaneous types of valves used in control service. This information allows the user to be aware of the range of operating conditions and size for each type of valve and how one type differs in terms of performance to another with different environments and service. Performance of a type of valve is eventually related to relative price and quality. The quality of control refers to crude; moderate or precise control of flow with varying degrees of stability; rangeability (turndown ratio) and valve trim life. The correct valve must be used with the proper instruments to function adequately in the dynamics of the control loop. Bear in mind that selection of control valves involves so many variables that only general guide lines may be given. The following table is given to indicate the typical distribution of control valves by size.

TYPICAL DISTRIBUTION OF CONTROL VALVE SIZE IN PROCESS PLANTS*

SIZE	CUMULATIVE PERCENTAGES
1½ inches or smaller	65% of total control valves
2 inches or smaller	83% of total control valves
3 inches or smaller	91% of total control valves
4 inches or smaller	96% of total control valves

*Source: ISA Paper 34.2.62
Forman & Oriolo.

VALVE SELECTION

General Considerations

The following operating variables are used to select valve types which can handle a specified set of service conditions:

1) Line Pressure (valve body pressure rating).
2) Flow Rate (C_V under flowing conditions related to valve size).
3) Differential Pressure (Allowable ΔP for throttling stability, low noise, anti-cavitation, and minor erosive wear).
4) Operating Temperature Range (Related to design and available materials).
5) Corrosion Rate (Related to

Evaluating Factors

Which type is the best valve for a specific application will depend upon the relative importance of the following factors:

1) Noise Level - Less than 90 dbA and/or choked flow if sonic velocity is achieved. (Limits flow increase with decreasing downstream pressure.
2) Cavitation - Greater than incipient (minor). Choked flow under cavitating conditions.
3) Flashing - Valve is sized to handle choked flow and body materials to withstand greater erosion.
4). Erosion - Minor or offset by design and hardened trim.
5) Throttling Stability - To meet process flow and pressure changes.
6) Cost - Total cost includes: purchase, installation, operating power, and maintenance.
7) Size - To fit space available. Consider piping strength, seismic forces, and pipe reducers versus a line size valve body with reduced flow area trim.

Computer programs are being developed by major valve manufacturers and large user industries to make this evaluation while leaving the final selection to the skilled judgement of the control valve application engineer. Some programs also include pumps, piping and manifolds. Many process conditions are general enough that the average user engineer can make an adequate valve selection. For severe or unusual conditions it is wise to consult with the valve manufacturer.

Computer programs now available often calculate, specify or call attention to:

1) Body material.
2) Pressure rating.
3) Temperature limitations.
4) Valve capacity (C_V) and size.
5) Cavitating conditions.
6) Flashing conditions.
7) Excessive velocity.
8) Noisy valves and an indication of sound pressure level (dbA).
9) C_V losses due to pipe reducers.
10) Correct C_V for choked flow based on valve opening.
11) Actuator size.
12) Whether or not valve meets all computer input constraints including differential pressure limitations for throttling certain fluids in specific valve types, plus compatibility with auxiliary equipment (positioners, pos-

13) End connections, based on piping specifications and line sizing.
14) Most economic selection. (Now being developed).
15) Economical hydraulic gradient. This helps to calculate the most effective piping dimensions and differential pressure across the valve as it relates to the loss of pressure in the system and pumping energy requirements.
16) Manifold factors.

KEY FACTORS TO CONSIDER IN THE SELECTION OF CONTROL VALVES

The following thirteen sets of tables are designed to be used in sequence and will serve as a guide to allow the user to delimit the type or types of valves best suited for a particular set of general operating conditions. These tables are not a substitute for a thorough reading and understanding of the material presented in the handbook, but they do assist the reader in relating the performance and general application of one valve type to another.

VALVE SELECTION TABLES

1) Operating Pressure and Temperature versus Available Sizes and Flow Capacity.
2) Allowable Differential Pressure for Continuous Throttling.
3) Operating Temperature Range.
4) Inherent Flow Characteristic and Rangeability.
5) Leakage.
6) Valve End Connections.
7) Actuators.
8) Valve Types versus Compactness, Weight, Relative Cost and Actuator Power.
9) Ease and Location of Maintenance.
10) Anticavitation Valves.
11) Low Aerodynamic Noise Valves.
12) Valve Body Types (size; pressure class; temperature range; sectional views).
13) Readily Available Body Materials and Linings.

SECTION I
PRESSURE-TEMPERATURE RATING VERSUS VALVE TYPE AND SIZE RANGE

It is first necessary to translate process data for the specific application into the valve flow coefficient (C_V). Chapter 6 contains information as to how to do this. The manufacturer's flow coefficient rating (C_V) is defined as the number of gallons per minute of 60°F water flowing through any valve in wide-open position with a 1.0 psi differential pressure. It is simply a means of comparing the capacity of one valve with that of another of the same type while operating under the same service conditions. The manufacturer's rated C_V for valves of different types may be equal but under actual operating conditions due to different effects on each type of valve by pipe reducers, cavitation or critical gas flow, the C_V value may be quite different. Section 8 of this chapter has more information about this. Methods of calculating C_V for both liquids and gases are given in Chapter 6. The wide open C_V value (manufacturer's rating) is usually selected so that normal process flow is about 65 to 75 percent or more of the rated C_V which will allow adequate control with some variation in process flow.[1] Examine Figures 4 and 5.

THE C_V OF THE LARGEST SIZE OF EACH TYPE OF VALVE IS SHOWN AT THE RIGHT END OF THE BAR GRAPH LINES.

The second step is to refer to the Pressure-Temperature tables in Chapter 9 and from these determine the correct rating for the specific application. Piping design may upgrade the valve rating. Also, end to end installation space in the piping may determine pressure rating if piping design has already been completed. Third; after the pressure-temperature rating has been established (i.e. - ANSI 150 lb. 600 lb. etc) entry can be made into the series of bar-graphs. "Pressure Rating versus Valve Type and Size Range" to determine whether the valve size at this rating is available in a given type or types.

[1] For pipeline valves the value sometimes reaches 90 percent. Sizing may also be based on ultimate

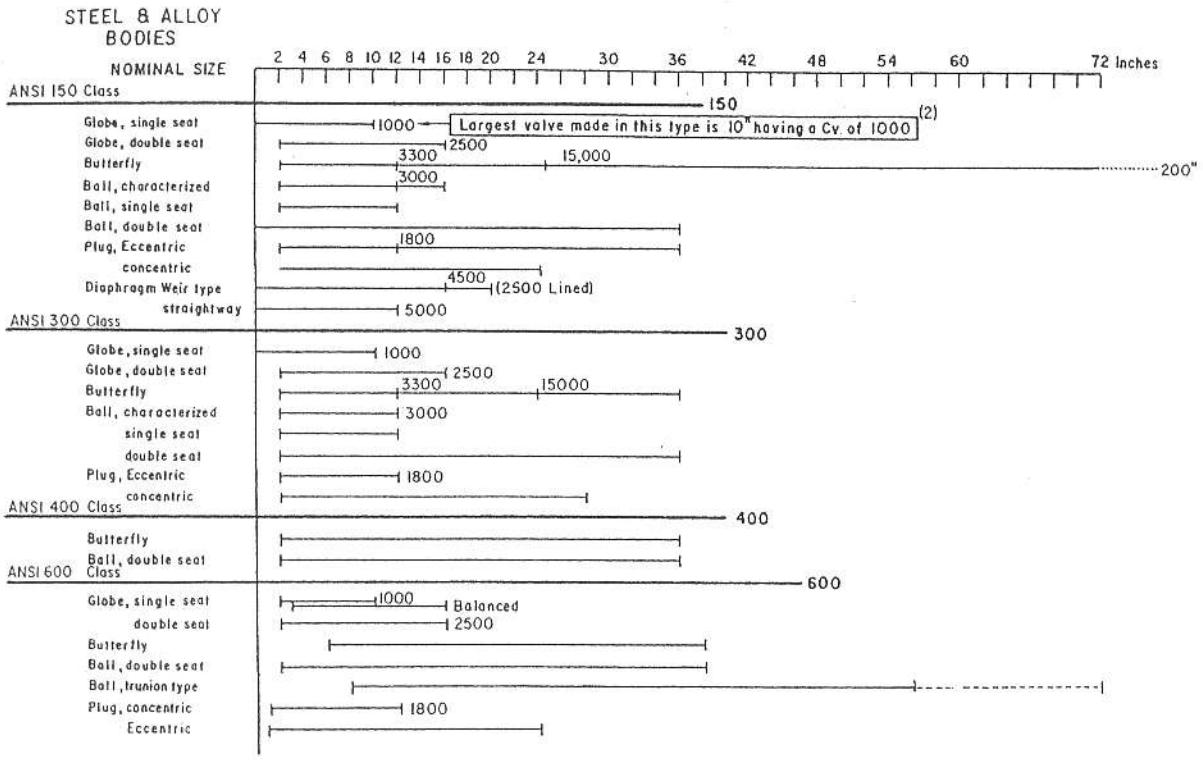


Figure 4. CONTROL VALVE SIZE, TYPE AND PRESSURE RATING.(1)
Includes Cv Rating of Largest Size for Steel and Alloy Body Materials.

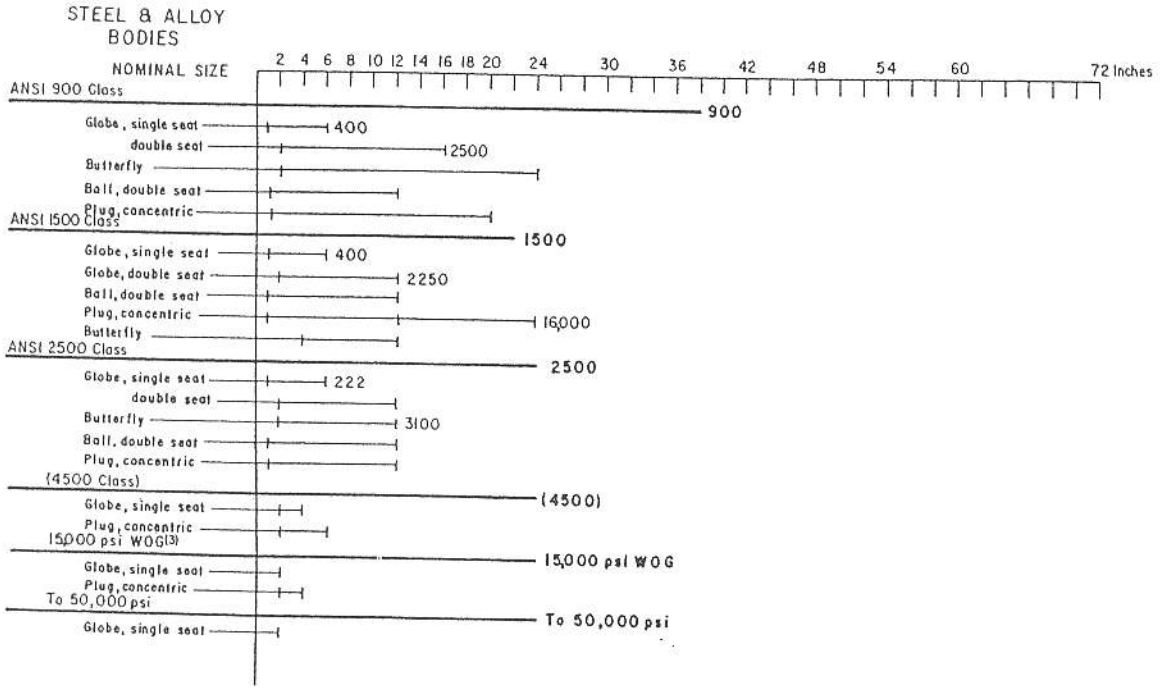


Figure 4. CONT'D. CONTROL VALVE SIZE, TYPE AND PRESSURE RATING.(1)

1) Information available as of January 1974. Consult manufacturers for latest data.
2) The 10-inch size limitation applies to valves with standard actuators. Larger sizes to 18 inches are available with high powered actuators particularly in the Globe Angle Style. Internal pressure balanced design allows sizes above 10 inches with standard actuators having a high degree of stiffness (high spring rate for a pneumatic diaphragm actuator). The latter types have been manufactured up to 30 inch port diameters and are powered with electro-mechanical, electrohydraulic actuators, or very large pneumatic cylinders.
3) Refers to water, oil or gas pressure rating at 100°F.

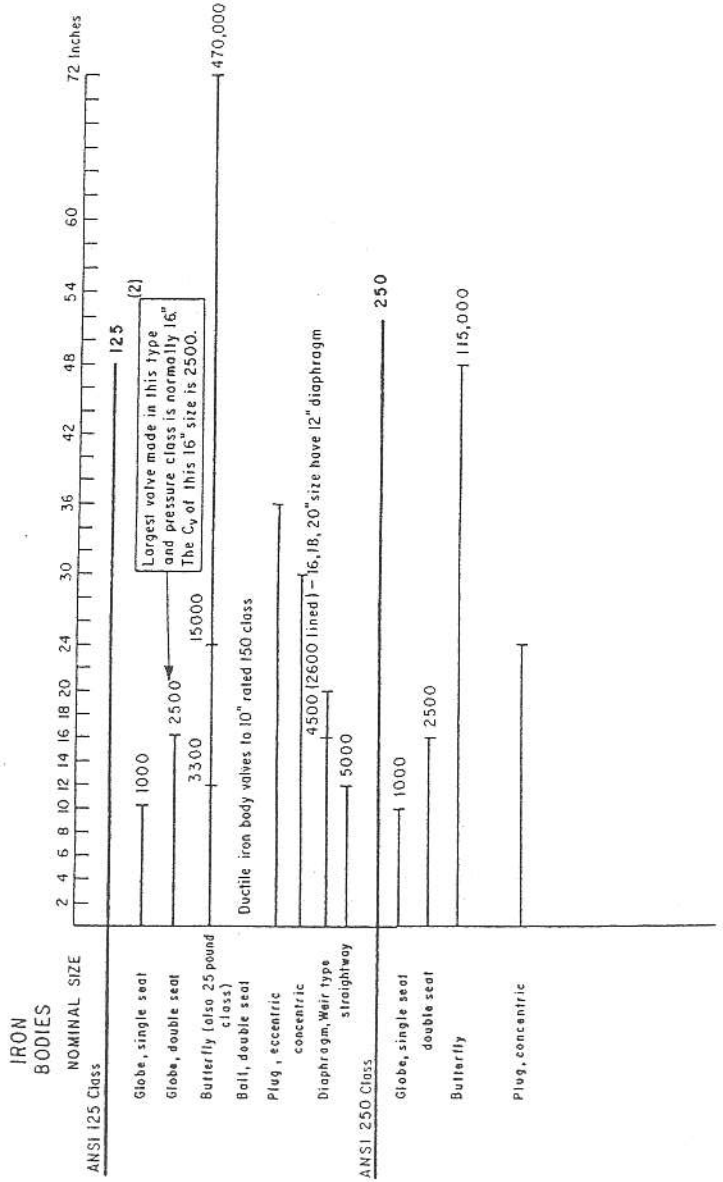


Figure 5. CONTROL VALVE SIZE, TYPE AND PRESSURE RATING.(1)

Include Cv Rating of Largest Size for Iron Body Materials.

1) Information available as of January 1974. Consult Manufacturers for latest data.
2) Sizes above 16 inches are available as specials with high powered actuators and in a non-seating design have been built to 24 inches in size.

SECTION II
SHUT-OFF DIFFERENTIAL PRESSURE VERSUS VALVE TYPE AND SIZE

From data given in this section the proper valve type with adequate shut-off differential pressure may be selected in relation to the size required as previously determined by C_V in Section Number 1. The type of valve chosen must be such that the allowable pressure drop shown in the Maximum Differential Pressure Shut-Off With Standard Actuator Graph is not exceeded in normal process operating conditions or by build-up of pressure during shut-off periods. Examine Figure 6. These three graphs are based on maximum shut-off capability resulting from the best combination of variables for each type of valve with a standard actuator. These variables are listed later in the text. They show the factors which must be considered by a manufacturer in establishing a shut-off differential pressure rating for a given valve. Examples of differential pressure ranges are given for specific valves to show the wide variation. C_V ratings are shown for one size of a valve, but different types. Be aware that the ratings are not obtainable under all flow and piping conditions.

An actuator of sufficient power may be installed on a valve to raise its differential pressure shut-off value to the limit of the mechanical strength of the bonnet and internal components. Limitations are: 1) allowable seat joint load at higher differential pressures, 2) rupture of elastomer diaphragms, and 3) reduction of flow capacity of ball and butterfly valves as the strength of internal parts is increased. Pressure balanced globe valves offer one solution to higher differential pressures with moderate sized actuators. Double seated globe valve types approach static balance when closed, however they are far from being dynamically balanced while throttling. Modern cage design globe valves usually eliminate or minimize dynamic balancing considerations.

VARIABLES AFFECTING DIFFERENTIAL PRESSURE CAPABILITY FOR SHUT-OFF

1) Flow Capacity in the seat orifice area.
2) Flow tending to open or flow tending to close the valve plug.
3) Diaphragm action-air to open or air to close.
4) Actuator size.
5) Degree of plug pressure balance both static and dynamic.
6) Stem size.
7) Actuator supply pressure.
8) Actuator spring range.

Examples of Differential Pressure Drop Capability Achieved by Changing The Variables For 6 Inch Valves With Standard Actuators

TYPE OF VALVE	DIFFERENTIAL PRESSURE RANGE	C_V[1]
Solid Ball	1440 psi.	1500
Unlined Butterfly	21 psi. – 200 psi.	1750
Characterized Ball	175 psi. – 320 psi.	1200
Diaphragm	125 psi. – 125 psi.	600
Globe, double seated	400 psi. – 400 psi.	540
Globe, single seated (non pressure balanced)	90 psi. – 210 psi.	400

To use the three graphs of Figure 6 enter on the left your maximum shut in pressure and from the bottom with the valve size. Make a valve type selection whose curve lies above and to the right of the intersection point of your pressure and size.

MAXIMUM ALLOWABLE THROTTLING DIFFERENTIAL PRESSURE FOR LONG SERVICE LIFE AND THROTTLING STABILITY

Throttling erosion shortens valve trim life and in severe cases even the body life. Abrasive particles in liquids, or liquid droplets in gases, coupled with high flow velocity through the vena contracta area of the valve to plug seat flow area causes erosion. Cavitation erosion and or corrosion-erosion caused by the scouring away of the protective oxide metal coating are not considered in this section.

Figure 7 (A-D) lists valve types and the allowable maximum differential pressure depending upon abrasive conditions. Listings are for clean non-abrasive liquids and gases for reference, for abrasive fines in liquids and gases, and for both moderate and harshly abrasive slurries. Size your valve so as not to exceed these pressure drops in order to achieve the required flow capacity. Information is very limited on this subject as indicated in Figure 7.

When an excessively high liquid pressure drop cannot be avoided, select a valve having the fewest turns in its fluid flow path, (*) a generous body flow area feeding fluid to the seat joint area, and use valve trim harder than the abrasive materials in the fluid. Refer to Chapter 4, the Trim Erosion section for more information.

[1]The magnitude and relative order of these flow capacities may change considerably as throttling pressure

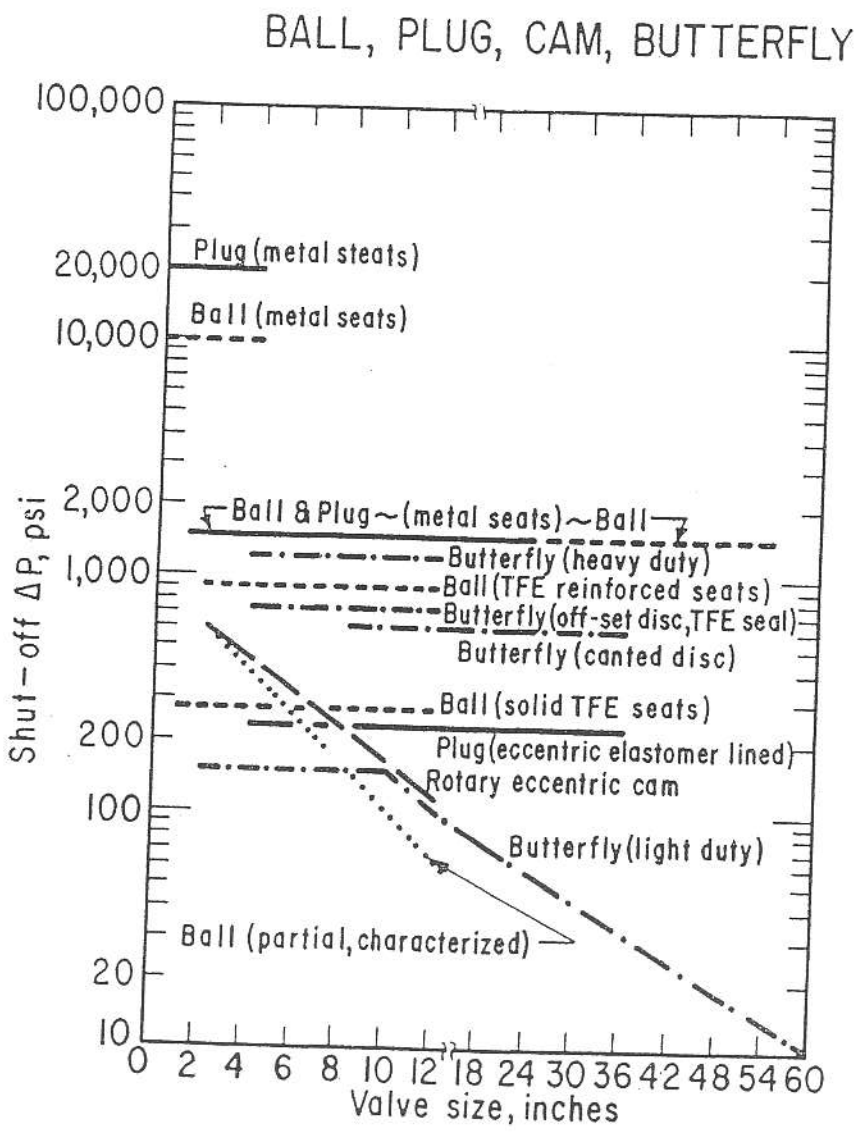


Figure 6A. MAXIMUM SHUT-OFF DIFFERENTIAL PRESSURE WITH STANDARD ACTUATOR VERSUS VALVE SIZE.*

Use only for valve types listed above. See restrictions under Figure 6C.

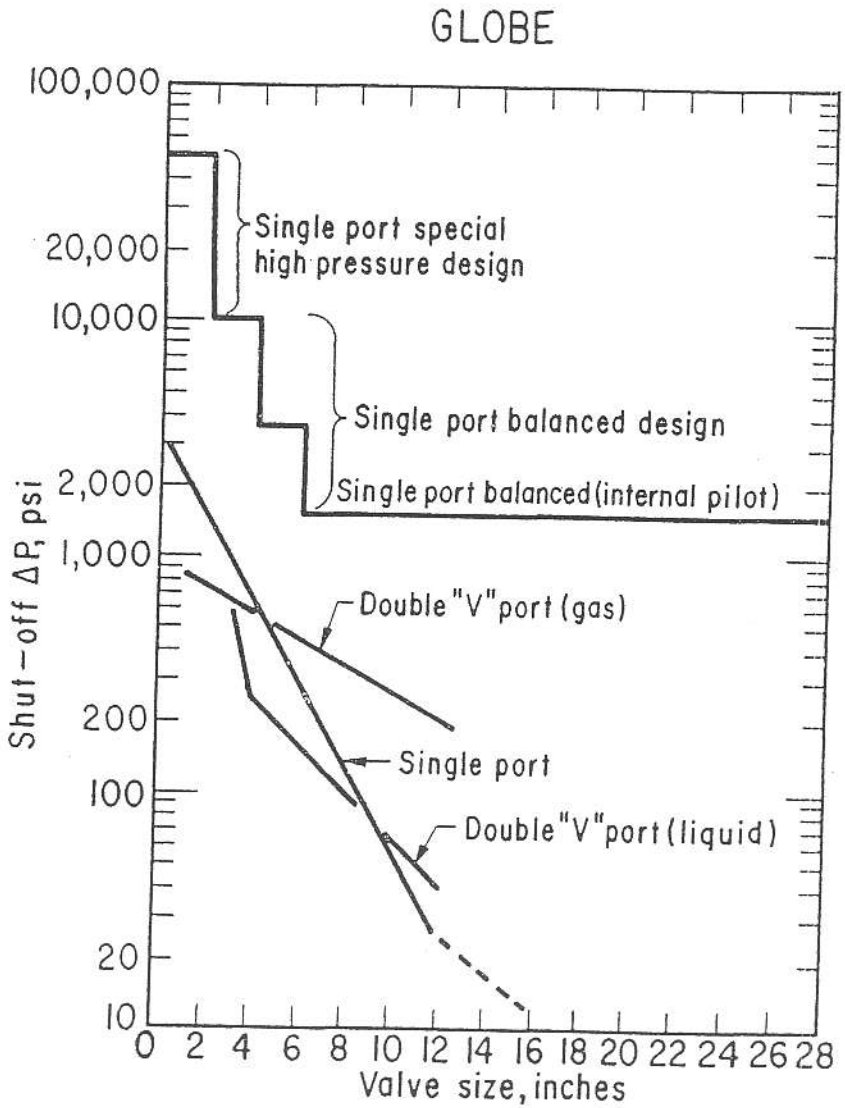


Figure 6B. MAXIMUM SHUT-OFF DIFFERENTIAL PRESSURE WITH STANDARD ACTUATOR VERSUS VALVE SIZE.*

Use only for globe valves. See restrictions under Figure 6C.

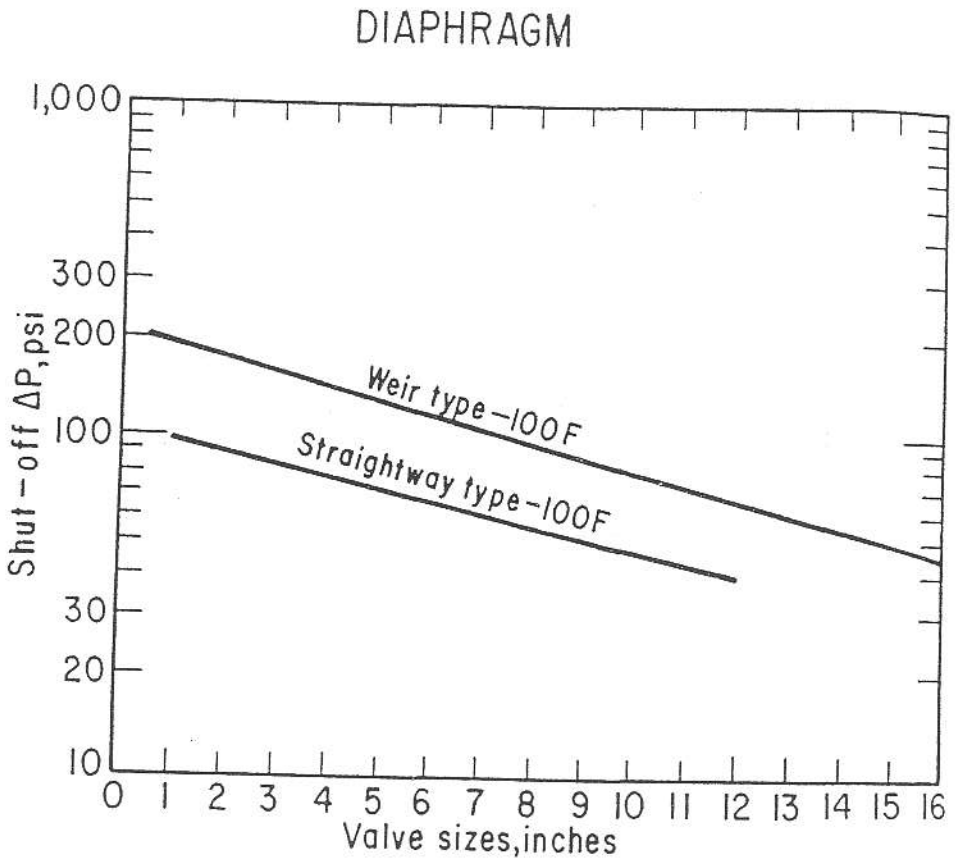


Figure 6C. MAXIMUM SHUT-OFF DIFFERENTIAL PRESSURE WITH STANDARD ACTUATOR VERSUS VALVE SIZE.*

Use only for diaphragm valves.

* 1) DO NOT USE FOR CONTINUOUS THROTTLING ΔP RATING. Graphs show shut-off capability only.

2) Differential pressure ratings are given with valve assembled with standard actuator. Larger actuators and higher ratings are available in some cases. Consult your valve supplier.

3) Differential pressure ratings are on the basis of full port valve orifices approximating 0.8 nominal pipe size and considering wall thickness for the pressure involved. Butterfly valves are full port line size.

4) Globe valve double "V" port ratings for liquids are lower than for gas due to the greater rotary force imparted to the valve plug by the higher density fluid upon valve opening.

LINE FLUIDS	CLEAN NON-ABRASIVE		EROSIVE FINES IN FLUIDS		ABRASIVE SLURRIES	
	LIQUID	GAS	LIQUIDS	GASES	MODERATELY ABRASIVE	HARSHLY ABRASIVE
BALL, solid type [1] Standard trim, hard chrome plated ball, TFE seats	300 psi without allowing cavitation	200 psi	Consult Manufacturer for Recommendations		15 psi	5 psi
SPECIAL TRIM Hard chrome plated ball and Stellite seats	Special trim is not required for these fluids		Consult Manufacturer for Recommendations		Consult Manufacturer for Recommendations	
ULTIMATE TRIM Hard chrome plated ball and non-contacting Stellite seat orifice ring			Consult Manufacturer for Recommendations		Consult Manufacturer for Recommendations	

Figure 7A. BALL VALVES.
Maximum Allowable Differential Pressure for Continuous Throttling.
(For adequate trim life and throttling stability)

[1] For segmented type ball with "vee" or contour edge contact manufacturer for recommendation.

LINE FLUIDS	CLEAN NON-ABRASIVE		EROSIVE FINES IN FLUIDS		ABRASIVE SLURRIES	
	LIQUID	GAS	LIQUIDS	GASES	MODERATELY ABRASIVE	HARSHLY ABRASIVE
STANDARD TRIM	No problems at rated differential pressures — full open or fully closed. Do not allow valve to cavitate		NO RATINGS HAVE BEEN ESTABLISHED Consult valve manufacturer who will consider differential pressure in relation to upstream pressure (erosion on closure), angle of opening during throttling, and corrosive properties (for corrosive-erosion) by line fluid. Then special or ultimate trim may be recommended.			
SPECIAL TRIM, Lined body and hardened disc edge	Special trim is not required for these fluids					
ULTIMATE TRIM, Hardfaced disc and body with hard chrome plated shaft	Special trim is not required for these fluids					

Figure 7B. BUTTERFLY VALVES.
Maximum Allowable Differential Pressure for Continuous Throttling.
(For adequate trim life and throttling stability)

LINE FLUIDS	CLEAN NON-ABRASIVE		EROSIVE FINES IN FLUIDS		ABRASIVE SLURRIES	
	LIQUID	GAS	LIQUIDS	GASES	MODERATELY ABRASIVE	HARSHLY ABRASIVE
STANDARD TRIM	Use manufacturer's charts based on valve size, diaphragm material, fluid and temperature to arrive at a differential pressure limit		Limit Fluid velocity to 6 ft./sec.	Limit Fluid velocity to 6 ft./sec.	Limit slurry velocity to 6 ft./sec.	Limit slurry velocity to 6 ft./sec.
SPECIAL TRIM, Elastomer lined body	Special trim is not required for these fluids		CONSULT MANUFACTURER Manufacturer may raise allowable differential pressure given in his tables after consideration of hardness and velocity of particles in the line fluid in relation to both diaphragm life and body lining material.			
ULTIMATE TRIM, Glass lined body, Tefzel® lined body	Special trim is not required for these fluids					

® Trademark of Du Pont Co.

Figure 7C. DIAPHRAGM VALVES.
Maximum Allowable Differential Pressure for Continuous Throttling.
(For adequate trim life and throttling stability)

LINE FLUIDS	CLEAN NON-ABRASIVE		EROSIVE FINES IN FLUIDS		ABRASIVE SLURRIES	
	LIQUID	GAS	LIQUIDS	GASES	MODERATELY ABRASIVE	HARSHLY ABRASIVE
STANDARD TRIM, 316 stainless steel plug and seat to 12 inches	200 psi	300 psi	Not suitable	Not suitable	Not suitable	Not suitable
SPECIAL TRIM, Stellite #6 hardfaced plug and seat joint to 24 inches	500 psi	1000 psi	100 psi	200 psi	Not suitable	Not suitable
STAINLESS STEEL, 416 Plug and seat with 17-4 PH hardened cage	1000 psi	1000 psi	250 psi	500 psi	25 psi	Not suitable
FULLY STELLITED plug contour seat and joint, seat orifice to 4 inches	2000 psi	5000 psi	500 psi	1000 psi	25 psi	Not suitable (body erosion)
ULTIMATE TRIM, 440-C stainless steel plug and seat (also cage) to 4 inches Hardened 58-62 R_C	1000 psi Several years if no cavitation	1000 psi Several years	5000 psi (3 mo. life)	10,000 psi (6 mo. life)	50 psi	Not suitable (body erosion)
TUNGSTEN CARBIDE plug and seat to 2 inches	1000 psi Several years if no cavitation	1000 psi Several years	5000 psi (6-12 mos. life)	10,000 psi (6-12 mos. life)	100 psi	Not suitable (body erosion)

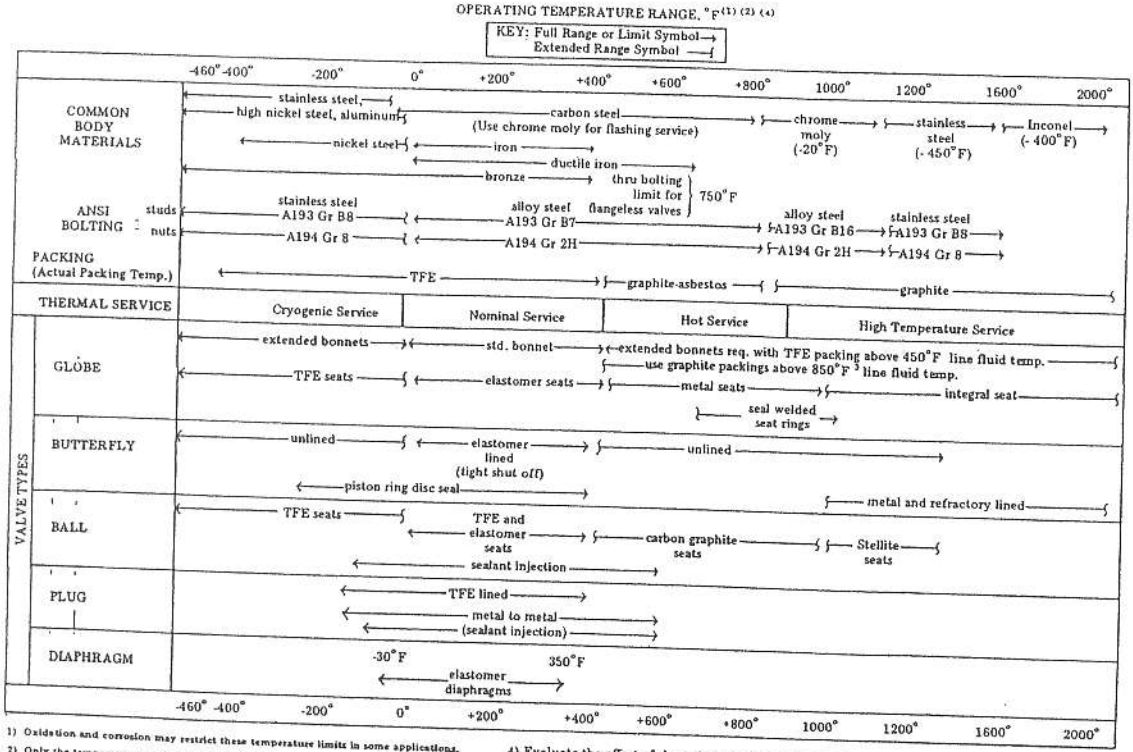
Figure 7D. GLOBE VALVES.
Maximum Allowable Differential Pressure for Continuous Throttling.
(For adequate trim life and throttling stability)

SECTION III

VALVE OPERATING TEMPERATURE RANGE

Figure 8 illustrates valve types normally available for use in certain temperature ranges, and lists general construction and materials for these specific valves. Refer to Chapter 9 for more information.

Figure 8 is used by finding the maximum and minimum process temperatures (°) on the chart and then by inspection identify the valve types that are made for the temperature range. Note material and construction which must be specified by the purchaser or the manufacturer. When a temperature range covers two or more material classifications, consult the manufacturer.



1) Oxidation and corrosion may restrict these temperature limits in some applications.

2) Only the temperature range where usage is common or essential for service materials. Full temperature ranges are given in the materials chapter of this handbook.

3) Safety consideration to prevent minor packing leakage from overheating bonnet, thus destroying TFE packing and causing excessive leakage. No extension is needed with graphite-asbestos packing below 850°F. No extension needed with all graphite packing.

4) Evaluate the effect of short time excursions into regions of higher and lower temperature on a basis of frequency of occurance, safety, operating reliability, and deterioration or shortening the life of the valve and or principal parts.

Figure 8. TYPICAL OPERATING TEMPERATURE VERSUS VALVE TYPE AND COMMON MATERIALS.

SECTION IV

INHERENT FLOW CHARACTERISTIC AND RANGEABILITY

Figure 9 illustrates the flow characteristic (s) of each of the type of valves listed. Corresponding to each of these is a column listing the rangeabilities available, and a column giving a guide to control characteristics at low flow rates.

When differential pressure drops severely with valve opening, the equal percentage characteristic approaches linear and a linear characteristic approaches quick opening. A quick opening characteristic approaches extreme quick opening.

How To Use The Table:

1) Select the types(s) of valve on the basis of the type of flow characteristic that the process requires.

2) Calculate the "Rangeability" required by the process. Refer to Chapter 5 for this. This figure must be a lower value than those listed in the "Rangeability" column.

3) Check the information in the "Low Flow Control" column to make sure no problems or limitations exist.

Other characteristics that are not listed, can be achieved by adding a valve positioner with a suitable characterized cam. However its effect on the time constant usually destroys improvement in gain. Cams are effective when the valve actuator is not the first or second time constant. The characteristic is then not important. This recommendation does not apply to high pressure drop gas or steam valves. Refer to Section 11 for more information.

Select on basis of "Inherent" characteristic changing to "Installed" characteristic in process having large decrease in pressure drop with valve opening. Refer to text Chapter V. Pick overall rangeability* and low flow control required. Many processes vary flow rate by a factor of 4:1 whereas pressure may fluctuate ±25%. The maximum combined effect is a turndown ratio of 5:1. Other characteristics are available by adding a valve positioner and characterized cam but these are often not effective.

NOTE: When upstream pressure drops with valve opening the equal percentage approaches linear and linear approaches quick opening. Quick opening becomes extreme quick opening (high gain).

Valve Type	Inherent Flow Characteristic	Rangeability	Low Flow Control (Down to 5% of C_v Rating)	Remarks
Linear Motion Type				
Globe Style (with characterized plug or characterized cage)	Linear	30:1 to 300:1	Good (See Chapter IV)	—
	Equal Percentage	30:1 to 60:1	Good	—
	Parabolic	—	Good	—
	Quick Opening	100:1	—	—
Sliding Gate (Multiple Orifice Type)	Linear	100:1	—	—
V-Orifice Gate	Equal Percentage	—	—	—
	Linear	—	Equal Percentage first 15% opening	—
Rotary Motion Type				
Butterfly	Equal Percentage	100:1 to 12 inches[1]	Symmetrical disc is equal percentage; offset disc is quick opening first 10-15°, then equal percentage	Reduced port available as special
Cam Plug	—	33:1 to 100:1 above 12 inches[1]		
Cam Disc	Linear	100:1	—	—
Plug Characterized Plug (or cage surrounding plug)	—	—	—	—
Full Ball	Modified linear and semi-parabolic	—	—	—
Characterized and "V" Ball	Equal Percentage 9° to 80° open	50:1	Good	—
	Modified equal percentage	350:1 (with seat) 100:1 (w/o seat for erosive service)	Good	—
Diaphragm				
Pinch	—	—	Good	—
Weir type	—	Standard, 5:1 Dual range plug, 15:1	Not available Fair	Use dual range plug to increase rangeability of Weir type.
Sleeve	—	—	—	—

*Rangeability is defined as ratio of maximum to minimum flow controllable to the desired characteristic without sudden changes or flattening of the flow curve between selected minimum and maximum.

[1]This data is for an in-line disc. Offset disc rangeability is 20:1.

Figure 9. INHERENT FLOW CHARACTERISTICS AND RANGEABILITY.

SECTION V

SPECIFIED MAXIMUM LEAKAGE

Decide how large a leak rate the specific process application can safely and economically tolerate at the maximum differential pressure build up against the closed valve. Keep in mind that continuous leakage at sufficiently high differential pressure may eventually "wire draw" the plug-to-seat joint causing excessive leakage.

Next, calculate the maximum allowable leakage rate in terms of percent of valve maximum C_v value. Another method is to specify the leakage rate per inch of valve seat orifice diameter per pound of differential pressure to compare with water or air leakage test specifications. These specifications are given in ISA recommended practice RP-39.6. Also refer to Chapter 4.

Now, refer to Figure 10. It lists leak rates for various valve types. Select a valve type having a lower leak rate than the maximum process leak rate allowed. The given rates are based on factory tests of new valves so an allowance must be made for leakage to increase with service usage.

Leakage Class ISA RP39.6	Allowable Leakage Rate Air or Water	Valve Types	Remarks
Class I	Category II, III or IV, but no test required by agreement between user and supplier.	Valve Types listed in Category II, III & IV	Quality of mfg. implies that these valves do not exceed leakage classes II, III & IV, but no guarantee is stipulated.
Class II	0.5% rated valve capacity, (maximum C_v)	Globe, double seated. Globe, single seated, balanced with stepped metal piston seat. Butterfly, metal lined.	
Class III	0.1% of rated valve capacity	High quality globe double seated. Globe, single seated, balanced with continuous metal piston seals.	
Class IV	.01% of rated valve capacity	Globe, single seated. Globe, single seated, balanced with elastomer piston seals. Rotary eccentric cam type. Ball valves with metal seat.	
Class V	$5x10^{-4}$ cc/min. of water per inch of orifice diameter per psi differential pressure	Globe valves in Class IV with heavy duty actuators to increase seating force.	Few valves continue to remain this tight in service unless the seat plastically deforms to maintain contact with the plug.
Class VI	Maximum permissible leakage associated with resilient seating valves. Expressed as bubbles per min as per RP39.6[1]	Globe with resilient seat. Butterfly, elastomer lined. Rotary eccentric cam with elastomer seat. Ball with resilient seat, solid ball type. Diaphragm, Weir type. Plug valves, elastomer seated or sealant injection sealing system.	Elastomer sealed valves remain this tight for many thousands of cycles until the seal is worn or cut.

[1] Example — 0.45 cc/min for a 2-inch port orifice diameter in a globe, butterfly or ball valve with 50 psi differential pressure air. Equivalent to 3 bubbles per minute from a ¼-inch O.D., .032-inch wall tube, ¼-inch under water surface.

NOTE: The terms bubble tight and drop tight are meaningless unless some leakage rate is specified. Lack of visible air bubbles using soap solution indicates leakage of less than $1x10^{-3}$ to $1x10^{-4}$ cc/sec.

Figure 10. VALVE LEAKAGE RATES.

SECTION VI
VALVE END CONNECTIONS

Figure 11 lists the generic type of valve connections available for specific valve types and sizes. The reasons for selecting a particular type may be found in the "Usage" column. A listing of each specific type of valve connection and its use, size range, temperature rating, pressure rating and gasket (if used) can be found in the Figure 12. To use these tables, first read over the usage column of Figure 11 and gain an idea of the type of connection best suited for your service conditions. Locate this generic type in the following Figure 12 and select a specific design while making sure that its ratings exceed operating conditions.

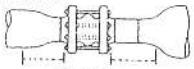
Type and Description	Valve Types Used On	Sizes	Usage
Screwed	All	1/4 in. to 2 inches. Some low pressure 3 & 4 inch connections.	General: Not suitable for hot thermal cyclic service or high pressure gas. Usually limited to 5000 psi ambient liquids & 3000 psi gas.
Flanged	All	1/2 in. & larger	General: Allows removal from piping without a union required for screwed.
Loose Flange (Flange is retained to valve body pipe hub with split key ring) High pressure flanges may be screwed to body piping hub (usually service above 15,000 psi)	Primarily used on Split Body Valves (loose flange & split body applications limited by piping strains in some systems) (Also on some globe valves)	1 in. to 4 inches 150 lb. 300 lb. 600 lb. (Same body – different flanges) Non-standard E to E dimensions for 150 & 300 lb. optional	1) Allows use of low cost carbon steel flanges with an alloy body. 2) One body may be used with several flange ratings – 150, 300 & 600 lb.* *(usually maximum due to pipe bending moments) 3) End-to-end dimension is the same for all ratings (usually the ISA or ANSI standard for 600 lb.) However, this is optional.
Flangeless (Through bolts from pipe flanges surround and hold body between pipe flanges) Allow piping clearance for long studs when body size is smaller than line size.  Clearance	Butterfly Ball, Solid Type Ball, Characterized Rotary – Cam Type Globe	2 to 24 inches 2 to 16 inches 1 in. to 24 inches 1 in. to 12 inches 1 in. to 4 inches	1) One body serves several pressure ratings 150 lb., 250 lb., 300 lb., 400 lb., & 600 lb. with the same end-to-end dimension. 2) Less metal in alloy bodies. 3) Ball and cam action valves are smaller than line size to allow bolting to pass body. (Balls are 80% line flow area). 4) Do not use with iron flanges as bolting strains will break flange which overhangs valve body. 5) Butterfly discs and solid balls may overhang valve end on short pattern valve bodies.

Figure 11. GENERIC TYPES OF VALVE CONNECTIONS.
Must match piping specifications.

Figure 12. PRINCIPAL VALVE END CONNECTIONS, FLANGE FACINGS AND GASKETS.

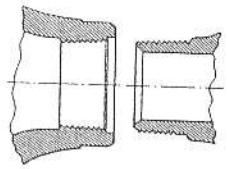
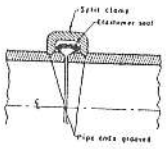
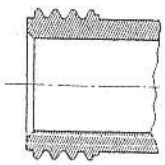
Connection	Application: Use	Size Range	Pressure Ratings	Temp. Limits	Gasket & Remarks
SCREWED 	1) General service at moderate pressure & steady temperature w/o severe shock or vibration. 2) Male threads used on a few valves through 2 inches.	¼ in. to 4 inches	Gas 3000 psi Liq. 5000 psi	400°F	1) Metal loaded thread seal to 600°F. 2) TFE tape thread seal to 450°F. Provide union in piping for valve removal for service.
GROOVED  	1) Valve easily replaceable in piping. 2) Lightweight joint. 3) Used to construct manifolds.	¾ in. to 24 inches 2, 2½ & 3 inches	1000* psi W.P. 2500* psi W.P.	Rating of Elastomer seal 0° to 300°F	1) Working pressure w/o external piping loads.* 2) Lightweight. 3) Pressure energized seal.
FOOD INDUSTRY Male Half Union 	Quick Disconnect for Cleaning For Thin Wall Stainless Pipe	1 in. to 4 inches		Elastomer Limit	

Figure 12. (Continued)

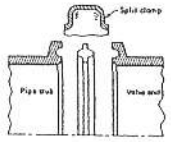
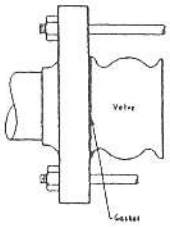
Connection	Use	Application: Size Range	Application: Pressure Ratings	Application: Temp. Limits	Gasket & Remarks
Grooved Flange Hub with Split Ring Clamp 	For Thin Wall Stainless Pipe	1 in. to 4 inches		Elastomer Limit	Rubber O-ring w/wings to eliminate crevices. Also with TFE outer jacket.
FLANGELESS 	1) Allows one body to serve 150, 300 & 600 lb. ratings. 2) Savings in cost of both valve & installation. 3) Used on: Butterfly to 24 inches Ball to 16 inches Rotary Eccentric plug to 12 inches Globe to 4 inches	1 to 12 inches	to 600 lb. Not suitable for Iron Flanges	—	Standard Flange Gaskets (Through bolting eliminates outer flange on valve body. Standard raised face on body for standard flange gasket).

Figure 12. PRINCIPAL VALVE END CONNECTIONS, FLANGE FACINGS AND GASKETS.

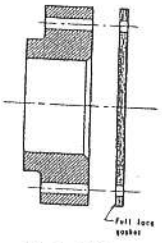
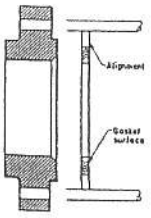
Connections	Use	Application: Size Range	Application: Pressure Ratings	Application: Temp. Limits	Gaskets & Remarks
FLANGED					Serrated facing on flange aids in sealing gaskets. Concentric "V" groove is standard. Phonographic finish is special.
Flat Face 	Iron, bronze & aluminum flanges to prevent breakage or bending	½ in. to 24 inches	125 lb. 150 lb.	400°F	1) Use full dia. gasket to flange O.D. 2) Rubberized asbestos. 3) Elastomer, TFE.
Raised Face (Includes loose flange held on flange hub by split key ring) 	General Service	½ in. to 24 inches	250 lb. to 2500 lb.	See ANSI B16.5	1) Rubberized asbestos to 500°F. 2) Metal-spiral wound, with asbestos to 1050°F. Flange face finish 125 microinch.

Figure 12. (Continued)

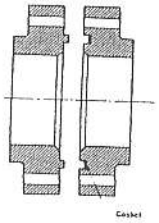
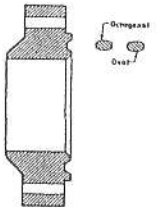
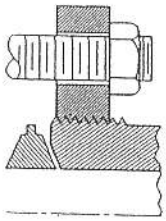
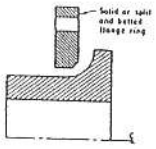
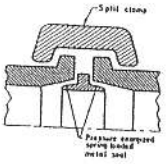
Connection	Application: Use	Size Range	Pressure Ratings	Temp. Limits	Gasket & Remarks
Tongue & Groove 	1) Chemical Service. 2) To restrain gaskets which cold flow.	½ in. to 24 inches	250 to 2500 lb.	See ANSI B16.5	1) TFE, filled TFE 2) Elastomer
Ring Joint 	1) High pressure. 2) High temperature (1500°F)	1 in. to 24 inches	150 to 2500 lb. 10,000 & 15,000 W.O.G. API Flgs.	-320°F to 1500°F	1) Oval or octagonal soft metal ring. 2) Ring may have bonded elastomer seals to reduce corrosion. 3) Requires axial movement of piping to disassemble.
Lens Ring with Threaded Flange 	Extreme pressure	—	to 50,000 psi	—	Soft metal tapered lens ring.

Figure 12. PRINCIPAL VALVE END CONNECTIONS, FLANGE FACINGS AND GASKETS.

Connection	Application: Use	Size Range	Pressure Ratings	Temp. Limits	Gasket & Remarks
Van Stone or Lapped Joint Flange 	Thin wall stainless steel or alloy piping for paper or chemical industry service handling corrosive products.[1]	3-12 inches	125 psi W.P. 250 psi W.P.	200°F 200°F	Rubber or TFE
SPLIT CLAMP 	1) Pressure energized metal seal. 2) Much lighter in weight than flanged. 3) Stronger than pipe in bending, compression and tension. 4) Vibration will not cause leakage. 5) Withstands 500°F instantaneous thermal shock w/o leaking. (Valve body & flange hub of piping should be of similar mat'l). 6) Meets ANSI & ASME code requirements. 7) Has higher pressure rating than standard flanged joints since load is less due to smaller sealing diameter of gasket.	½ in. to 30 inches	to 2500 lb. in some sizes	to 1800°F in special matl's	1) Carbon steel gasket. 2) 17-4PH gasket.

[1] A loose steel or ductile iron flange ring (solid or split in half and bolted) backs up the Van Stone lapped joint pipe stub. Make sure the pipe I.D. will allow clearance for vane of butterfly valve.

Figure 12. (Continued)

Connection	Use	Application Size Range	Pressure Ratings	Temp. Limits	Gasket & Remarks
WELDED Socket	Tight Joint for: 1) High temperature. 2) Vibration. 3) Hazardous Fluids.	¼ in. to 2 inches	Piping	Piping	Note: Carbon steel nipples often furnished with Chrome-Moly steel body valves to simplify welding into steel piping.
Butt	1) High Temperature. 2) Vibration. 3) Hazardous Fluids. *Also for smaller sizes for severe thermal cyclic and corrosive service.	2½* & up	Piping	Piping	Note: Carbon steel nipples often furnished with Chrome-Moly steel body valves to simplify welding into steel piping.

SECTION VII

ACTUATOR SELECTION FACTORS

1) The available powering medium - air, electric, hydraulic.
2) Thrust requirement for maximum valve differential pressure operation and tightness of shut-off required to overcome valve operating friction versus the size available for each type of actuator.
3) The ability of the actuator to hold the plug in a fixed position (stiffness) through surges of line pressure. This is achieved by a high rate mechanical spring, low case air volume, or hydraulic fluid loading. Also in extreme cases by using a hydraulic snubber on an air diaphragm type actuator. The rate of force change of the actuator with stroke should be at least twice the rate of change of the line pressure dynamic unbalance forces within the valve acting on the plug and stem.
4) The required actuator action upon failure of powering medium; valve fails open, closed or holds position.
5) Adequate frequency response to satisfy process dynamic and/or safe emergency full open or closing times.
6) Temperature limits of the Elastomer Diaphragm Materials.

Neoprene	-20°F. to 180°F.
Silicone	-30°F. to 300°F.
Viton	-10°F. to 350°F.
Metal Bellows	Use above 350°F. and for high levels of radiation.

Note: Low temperature performance can be altered by a different formulation of the elastomer. If the temperature goes below minimum temperature, high hysteresis, brittleness and tearing may occur.

7) Actuator cost escalating in this order:

1. air diaphragm type,
2. air piston,
3. air rotary,
4. electric, and
5. electrohydraulic type.

Now that you have read over the factors that affect the selection of a valve actuator, establish the requirements for the valve type previously selected. Do not forget about service, environmental conditions, emergency action and reliability.

This will require the calculation of forces or use of manufacturer's valve-actuator rating tables. See Chapter 7 for more information. Refer to Chapter 10 for service temperature limitations of general diaphragm materials.

Second, be certain that the actuator selected is mountable on the particular valve size chosen and will fit into the space available with adequate room to remove it for maintenance. The actuator must not be too powerful for the valve components.

Figure 13 schematically illustrates a great many generic types of actuators as well as many modifications which are normally available.[1] Stroke and thrust values are listed for linear motion types, whereas degrees of rotation and torque are listed for rotary motion valves. Speed of stroking, power supply range, action on power failure and valve types and sizes are listed together with advantages and limitations of each type. Refer to the *ISA Control Valve Compendium* for more extensive information listed by manufacturers.

[1]It is impossible to list complete information or even all types of the many and varied actuators on the market; however Figure 13 gives a good idea of functions and style of actuators available. Consult each manufacturer for the specific range of actuators and the valves they are used on.

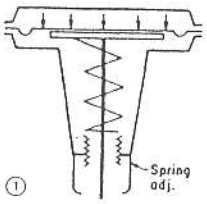
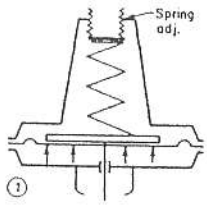
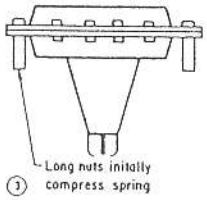
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
PNEUMATIC DIAPHRAGM — Linear Control Valve Stem Motion							
Air Extends Spring Retracts		to 4 inches 25 to 400 sq. in.	15 to 60 psi air	moderate	Spring controls	Globes to 16"	1) Reliable simple actuator.
Air Retracts Spring Extends		to 4 inches 25 to 400 sq. in.	15 to 60 psi air	moderate	Spring controls	Globes to 16"	1) Reliable simple actuator.
Non-Adj. Spring Both air extend & retract		to 2½ inches to 140 sq. in.	15 to 30 psi air	moderate	Spring controls	Globes to 4"	1) Less complex. 2) Positioner required in most applications.

Figure 13. TYPES OF ACTUATORS.

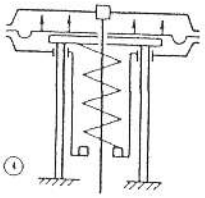
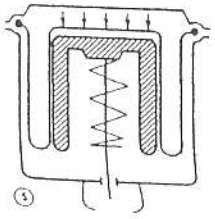
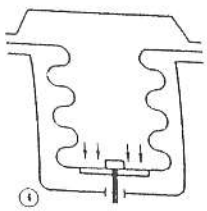
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Stem attached to case which moves; diaphragm plate is fixed.		to 4 inches to 400 sq. in.	to 125 psi air	slow	Spring controls	Globes to 16"	1) Optional design. 2) Case moves.
Pneumatic Rolling Diaphragm Air extends stem		Strokes to 7" Thrusts to 2500 lbs. to 7" diameter head	to 100 psi air	Fast due to low volumes	Spring controls	1-12" rotary valves 1-6" globe valves	1) Low friction. 2) Compact.
Metal Bellows		to 200 sq. in.	air	Slow	Not fail safe without a spring	—	1) Suitable for use in high radioactive environment. 2) Used in hot or cold environments exceeding elastomer usage range.

Figure 13. (Continued)

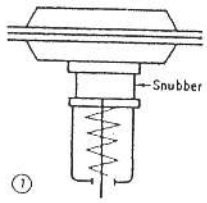
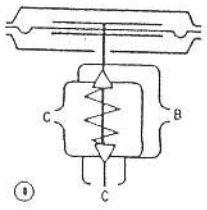
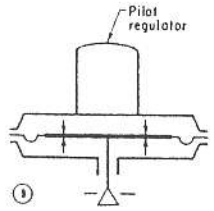
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust sizes	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valves Types & Sizes Used On	Advantages and Limitations
Hydraulic Snubber added for greater stability		Diaphragm cases to 400 sq. in.	air to 60 psi	Snubbing action adjustable Speed vs valve throttling stability	Spring controls	Globe	1) Installed on fluctuating pressure service.
Springless Used on 3-way, 3-pos. globe valve		Strokes to 3 inches 100 sq. in.	air to 60 psi	Fast	Valve spring closes both seat ports	Globes 1 to 4"	1) Simple 3-way action plus shut off.
Used on self-contained pressure regulator		Strokes to 3 inches 100 sq. in.	air to 60 psi	Fast		Globe type pressure regulators	1) Self-contained control with manual setting of pilot regulator.

Figure 13. TYPES OF ACTUATORS.

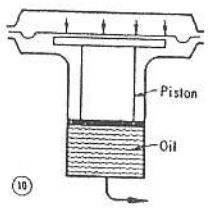
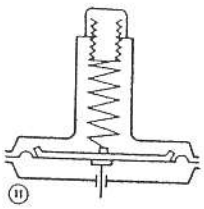
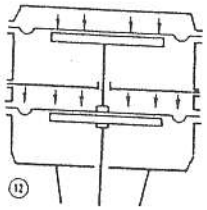
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust, Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Pneumatic Diaphragm Hydraulic Thrust Transfer Air Extends		to 150 sq. in. actuator	to 100 psi air	Fast	Not Fail Safe	1" to 8" inline valves with hydraulic plug positioning cylinder in flow stream	1) Thrust intensifier action with simple pneumatic control.
Super Power Pneumatic Diaphragm Spring Extends, Air Retracts		3" stroke 400 sq. in. 47,600 lb. thrust	125 psi air (maximum)	Slow unless boosters are used.	Spring thrusts down		1) Largest pneumatic diaphragm actuator — most powerful. 2) Size and weight may limit applications.
Double Pneumatic Diaphragm Giant power		3" thrusts to 80,000 lbs.	to 100 psi air	Slow	Not Fail Safe	Fits only large globe valves	1) Compact and powerful. 2) Very heavy. 3) Special yoke required.

Figure 13. (Continued)

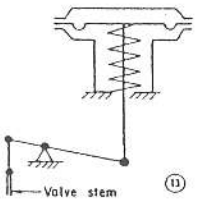
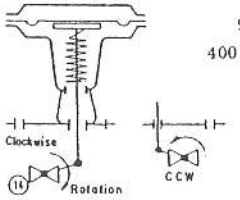
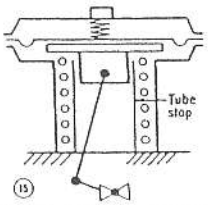
Type / Description of Operation	Schematic	Motion: Stroke, in. / Rotation, degrees / Thrust sizes	Power Source & Level	Stroking Speed: Inches/sec. / sec./degree span / RPM	Action on Power Failure	Valves Types & Sizes Used On	Advantages and Limitations
Short stroke lever arm magnifies power		Valve strokes to ¾ inch 240 sq. in 6:1 lever ratio	30 psi air	moderate	Spring controls	Heavy-duty low-noise valves 2-8" Bantam 1" low flow globe valves	1) Economical means of increasing stem thrust for short stroke valves.
PNEUMATIC DIAPHRAGM — Rotary Valve Stem Motion							
Linear to rotary motion Changing mounting position reverses action		90° 400 sq. in.	15-60 psi air	medium	Spring controls	to 12" ball to 20" butterfly unless very low pressure	1) Powerful. 2) Excessive height.
Rocking stem close coupled Changing mounting position reverses action		90° 140 sq. in.	15-60 psi air	fast	Spring controls	to 12" butterfly & ball valves	1) Compact. 2) One less link in operating levers.

Figure 13. TYPES OF ACTUATORS.

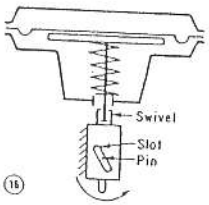
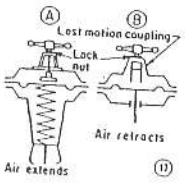
Type / Description of Operation	Schematic	Motion: Stroke, in. / Rotation, degrees / Thrust Size	Power Source & Level	Stroking Speed: Inches/sec. / sec./degree span / RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Rotary Output Pneumatic Diaphragm		90° Both rotation and lifting action may be combined	to 60 psi air	medium	Spring controls	Eccentric ball & plug valves to 4 inches	1) Combined lift and rotary action reduces plug valve wear and opening torque.
HANDWHEEL OVERRIDING ACTUATORS — (Actuator may be set as stem travel stop — partially open valve position) (#18 & #19 allow diaphragm and/or spring replacement while holding valve stem position — consult manufacturer)							
Top Mounted Threaded Stem		Allows any valve to full stroke with actuator	Manual input Usually 50 ft. lb. torque on handwheel rim is maximum	Slow	Must be manually engaged	Any diaphragm actuator or piston actuator	1) Simple. 2) Lowest cost. 3) Difficult to reach. 4) Subject to corrosion if not totally enclosed. 5) Requires head room.
Side Mounted Lever Action		Allows any valve to full stroke with actuator	Manual input Usually 50 ft. lb. torque on handwheel rim is maximum	Slow	Must be manually engaged	Designed to mount only on diaphragm type actuators	1) Accessible. 2) Heavy — may cause high yoke stress on seismic loading. 3) Subject to corrosion.

Figure 13. (Continued)

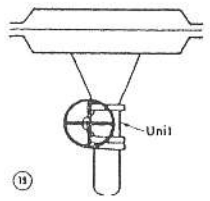
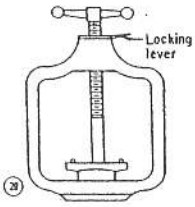
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Yoke Mounted Worm Gear Drive		Allows any valve to full stroke with actuator	25 ft. lb. torque max. input required due to gear ratio	Very slow	Must be manually engaged	Used with both diaphragm & piston actuators	1) Completely sealed & greased. 2) Extends actuator height. 3) For manual emergency closure consider time required to make total number of handwheel turns.
Manual Only		—	100 ft. lb. torque input on handwheel rim on valves over 4 inches	—	—	Globe all sizes	1) Inexpensive manual throttling. 2) Use of "cheaters" (bar in wheel) often damages valve trim or fractures iron yokes.

Figure 13. TYPES OF ACTUATORS.

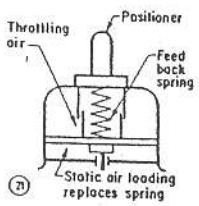
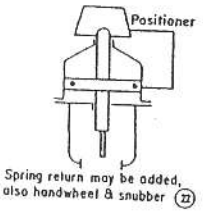
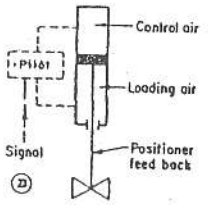
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust, Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
PNEUMATIC PISTON — Linear valve stem motion (#21 & 22 also used on rotary valves)							
Piston with Dual Air Loading		Strokes to 6" 30-200 sq. in. piston to 20,000 lbs. thrust	100 psi air	Fast	Not Fail Safe without pneumatic lock up	½ to 14" globe rotary valves (see #14 mounting)	1) High thrust. 2) Springless.
Piston with Dual Air Loading		2 to 8 inches	2400 to 25,000 lbs.* 4¾-17" dia. cylinder *w/o spring	— 35 psi min. to 150 psi	Valve stem thrust controls actuator w/o spring. Spring controls Air balast may be used to lock up position of on-off springless actuators.	Linear and Rotary Motion Valves	1) Compact. 2) Good frequency response.
Long Stroke Air Loaded Cylinder		to 6" stroke 4-8" bore 2500 lb. thrust	100 psi air	Fast	Not Fail Safe	Globe valves Rotary valves	1) Compact. 2) Springless.

Figure 13. (Continued)

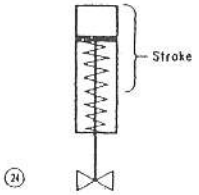
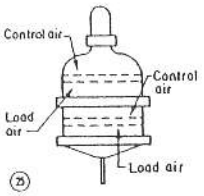
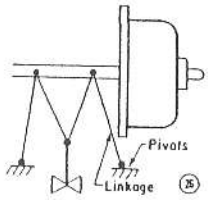
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust, Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Spring Opposed Air Cylinder		2½" stroke 12-100 sq. in. to 15,000 lbs. thrust	150 psi air	Fast	Spring controls	½ to 8" globe	1) Powerful.
Double Piston Dual Air Loading Super Power		to 6" stroke Combined area 400 sq. in. 40,000 lb. thrust	100 psi air	Fast	Not Fail Safe without pneumatic lockup	1-14" globe	1) Powerful. 2) Compact. 3) Not Fail Safe.
Toggle Action Dual Air Loaded Piston		100,000 lb. thrust with double 400 sq. in. piston area	100 psi air		Not Fail Safe	Globe	1) Power magnification by levers. 2) Make sure levers do not bind.

Figure 13. TYPES OF ACTUATORS.

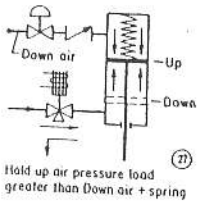
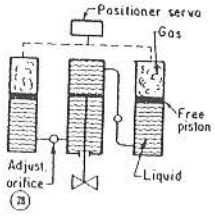
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Rapid Stroke Piston Open-close action Optional Spring Loading		—	2000 psi nitrogen	Extra Fast	Thrust Down	2-12" Globe	1) High reliability. 2) Extremely rapid closure.
Pneumatic Hydraulic High Stability Type Actuator		—	to 2000 psi nitrogen over oil	Fast closing and slow opening Vice versa	Drifts	Globe 2-12"	1) Precision throttling with varying high flow line pressures 1000 to 5000 psi
Handwheels Top Mounted and In-Yoke Available		—	—	—	—	—	See previous handwheel comments.

Figure 13. (Continued)

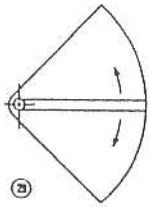
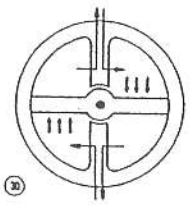
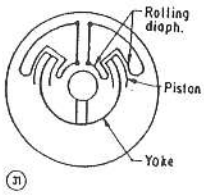
Type / Description of Operation	Schematic	Motion: Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed: Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
ROTARY VANE ACTUATORS							
Flapper Vane Open-Close action & throttling with positioner		90°	100 psi air	Fast	Valve stem controls Not Fail Safe	1-6" Rotary valves	1) Moderate cost.
Double Vane Air or Hydraulic		90°	Air & oil to 1000 psi	Fast	Oil may be locked up to hold position Not Fail Safe on air	Large pipeline ball valves & plug valves	1) Powerful. 2) Leakage limits throttling action.
Curved Piston Rolling Diaphragm		90° to 2400 in./lbs. single 4800 in./lbs. duplex mounting	80 psi air	Fast	Spring return Fail Safe		1) Low friction.

Figure 13. TYPES OF ACTUATORS.

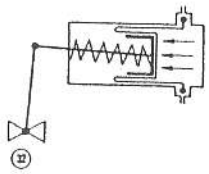
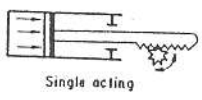
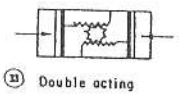
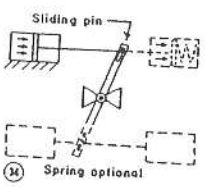
Type / Description of Operation	Schematic	Motion: Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed: Inches/sec. sec./degree span RPM	Action Power Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
PISTON ACTUATORS — Rotary Stem Motion Valves							
Rocking Piston Rolling Diaphragm		50°	15-60 psi air in 6.5" dia. cylinder	Fast	Spring controls	Rotary eccentric plug 1-12"	1) Less linkage. 2) Less friction. 3) Compact.
Ratchet Drive	 	90°	100 psi air	Fast	Spring controls	Rotary ½ to 36"	1) Simple. 2) Powerful. 3) Some lost motion restricts fine throttling.
Scotch Yoke Double Acting Cylinders		90°	100 psi air	Fast	Spring controls	Rotary 4-36"	1) Single or multiple cylinder. 2) Powerful. 3) Fast action. 4) Slight backlash in yoke.

Figure 13. (Continued)

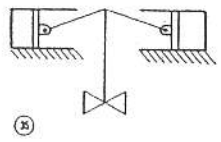
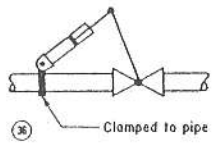
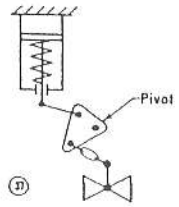
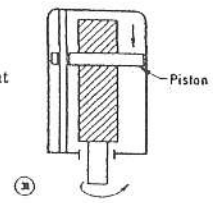
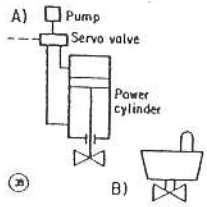
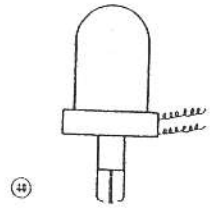
Type — Description of Operation	Schematic	Motion — Stroke, in. Rotation, degrees Thrust sizes	Power Source & Level	Stroking Speed — Inches/sec. sec./degree span RPM	Action on Power Failure	Valves Types & Sizes Used On	Advantages and Limitations
Crank Arm		90°	100 psi air	Fast	Not Fail Safe	Rotary valves	1) No lost motion in linkage — fine positioning possible.
Pivoting Cylinder Double Acting		90°	Hydraulic or 100 psi air	Fast	Not Fail Safe	Rotary valves 2-72"	1) Simple mounting. 2) Designed to provide greatest torque at point of valve opening.
Variable-Rate Rotary Output		90° thrust based on cylinder diameter & operating pressure	—	Fast	Spring controls	Pipeline ball valves 6-36"	1) Linkage designed to give rapid rotation near valve full open position to gain control of flow at a very low differential pressure while in liquid pipeline service

Figure 13. TYPES OF ACTUATORS.

Type — Description of Operation	Schematic	Motion — Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed — Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
Axial Thrust Piston-to-Rotary Output		90°	100 psi air	Fast	Valve dynamic forces control Actuator not Fail Safe	Rotary valves	1) Compact actuator. 2) Simple mounting.
		ELECTRO HYDRAULIC — Linear Valve Stem Motion with Power Cylinder. Rotary Valve Stem Motion with Vane Actuator (see No. 30)					
Continuous Throttling Type Moderate to High Speed Stroking		to 125,000 lbs. thrust	electric, ¼ to 600 HP	Fast	Requires auxiliary valves to lock up stem position	Reciprocating Stem — all sizes	1) High power. 2) Fast stroking. 3) Expensive. 4) Complex.
Slow Speed, Open-Closed Valve Action		Linear ½" to 3½" to 3000 lb. thrust	110 Volt AC or DC	30 to 60 sec. per inch	Spring strokes valve stem open or closed	½" to 6" globe valves in moderate pressure service	1) Compact. 2) Explosion proof housing available. 3) Nominal cost. 4) Slow stroke only. 5) No lock up on power failure.

(a) Electronic amplification with LVDT Feedback (very expensive)
(b) Has mechanical type servo valve (moderate price)

Figure 13. (Continued)

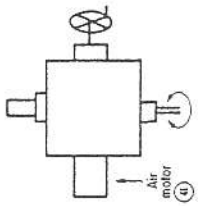
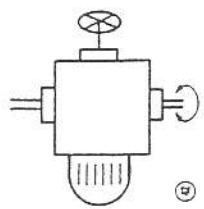
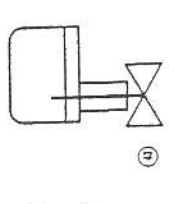
Type Description of Operation	Schematic	Motion Stroke, in. Rotation, degrees Thrust Size	Power Source & Level	Stroking Speed Inches/sec. sec./degree span RPM	Action on Power Failure	Valve Types & Sizes Used On	Advantages and Limitations
CONTINUOUS ROTARY OUTPUT — Worm Gear Driving Threaded Linear Valve Stem or Rotary Valve Stem.							
Pneumatic Motor Intermittant Operation		Rotary Reversible No limit to to rotations	100 psi air & 1000 psi gas	Fast	Holds valve stem position	to 36" rotary & reciprocating type valves	1) High power. 2) High speed. 3) Uses a large volume of air or gas. 4) Exhausts gas.
Electro-Mechanical (A) Intermittant Operation		Rotary Reversible No limit to rotations May drive bull gear	110 Volt AC & DC 440 Volt, three phase, A.C. power	Moderate	Holds valve stem position	2" to 34" globe valves in high-pressure service	1) High power. 2) Sealed explosion proof housing. 3) De-clutchable handwheel. (handwheel idle when motor operates)
(B) Continuously Powered Synchronous Electric Motor		¼" to 2" linear 500 lb. thrust Also rotary output. Drives until stalled open or closed.	to 400 oz. in torque in small sizes. More powerful models for plug valves. 115 Volt AC stall current does not damage motor Also digital pulse stepping.	Slow	Holds position	1/8" to 2" globe 4"-20" Segmented plug	1) Compact. 2) Simple controls. 3) Low power requirement.

Figure 13. TYPES OF ACTUATORS.

SECTION VIII

MATCHING VALVE TYPES WITH SERVICE CONDITIONS

This section will help to illustrate the results of specific valve performance when all types are reviewed for use in specific service applications. It calls attention to the many factors which must be considered in attempting to select the best valve.

The charts and tables for Section 8 show relative valve side view dimensions, weight, actuator power requirement, and ratioed cost increase with severity of service. Examples have been deliberately selected to show how each valve type is superior to others for specific conditions. Keep in mind that each valve type serves best only in a limited segment of the total market applications.

Charts illustrate how cost can vary with valve types used in the same type of service. The relative cost of any specific valve type may hold fixed or increase with increasing severity of service conditions. Scale drawings of valve cross sections illustrate relative size versus service, weight and actuator size and power. This information is not presented in an effort to show preference to any particular valve type. The examples cited are only a few of the millions of possible service conditions and therefore are presented only as a means of edification.

SELECTING A CONTROL VALVE FOR LOWEST PUMPING HORSEPOWER REQUIREMENT

EXAMPLE: 1

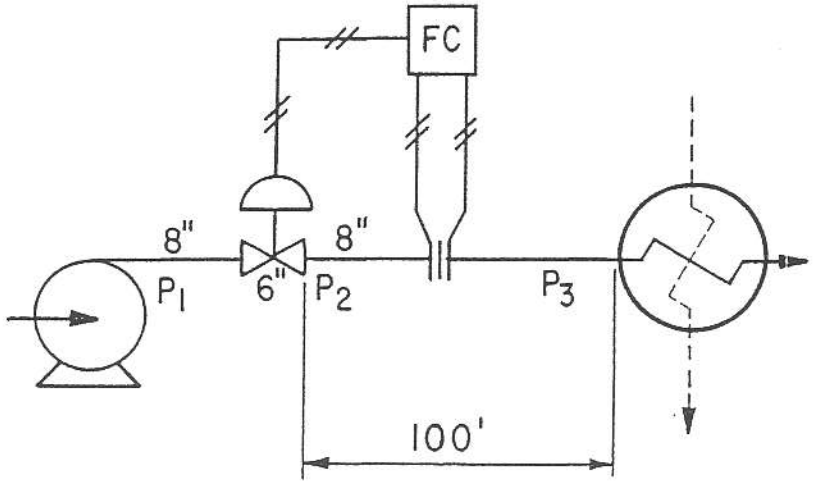


Figure 14. CENTRIFUGAL PUMP SUPPLYING WATER TO HEAT EXCHANGER.

Q = 3600 GMP ± 20%, 23 ft/sec velocity in an 8 inch line.
T = 60°F.
P_1 = Select the valve for the minimum pump discharge pressure.
P_2 = 44 psig at 3600 GPM with 9 psi drop in 100 ft. of 8 inch line.
P_3 = 35 psig minimum delivery pressure specification.

To select the P_1 minimum for lowest pump discharge pressure and pumping horsepower requirement, find the control valve type having the lowest differential pressure when flowing at a rate of 3600 gpm. The C_v value cannot be used directly because additional frictional losses are caused by the reducer going from an 8 inch line to 6 inch valve size and back to the 8 inch line. Also, some types of valves may be operating in the region of cavitation which further increases flow resistance. The data contained in Figures 15 and 16 illustrates these points. The butterfly valve has the lowest pump discharge pressure with the ball valve a close second. A globe style is about twice that of a butterfly valve. However, both ball and butterfly valves are at the point of incipient cavitation when flow reaches 3600 GPM.

Valve Type	Manufacturer's Rated C_v for 6-inch Valves	C_v Installed in 8-inch Line ($F_p C_v$)	Differential Pressure Full Open Valve to Flow 3600 GPM (1)	Required P_1 psia	Remarks
Globe, single seated	400	392	84 psi	143	
Globe, double seated	450	439	67	126	
Rotary Eccentric Plug	500	485	55	114	
Diaphragm, Weir type	600	575	39	98	Butterfly Valve has lowest ΔP for required flow and thus requires lowest pump discharge pressure. Therefore the system requires the lowest energy input.
Ball, characterized (segmented ball)	990	887	16	75	
Butterfly (75° open)	1150	966	13	72	

Figure 15. PERFORMANCE OF VALVE TYPES.
To be used with example one.

1) Note: Full open position at normal flow chosen only to illustrate pump discharge pressure relationship in that complete Cv versus opening curves were not available for all valve types. Actual practice would be to size the valve to be 65 to 75 percent open at 3600 GPM to allow for a 20 percent flow increase.

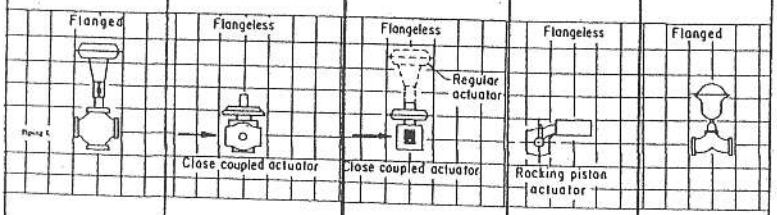
Item	Globe(1) Single/Double Seated	Characterized(2) Ball	Butterfly(3)	Rotary(4) Eccentric Plug	Saunders(5) Diaphragm	Keyed Remarks
Each Square is 10" on a side						1) Flow to open, air to close. Iron body, 125 lb. 316 stainless steel trim. 2) Steel body, 150 lb., flangeless, chrome plated, stainless steel ball, stainless steel seal ring. 3) Iron body, 125 lb. stainless steel vane. 4) Steel body, 150 to 600 lb. flangeless, stellited stainless steel plug and stainless steel seat. 5) Iron body, rubber diaphragm.
Actuator sq. in. Power psi Weight, Lbs. (Valve Plus Actuator)	200/145 sq. in. 35 psi 450 lbs.	140 sq. in. 35 psi 185 lbs.	140 sq. in. 30 psi 72 lbs.	33 sq. in. 40 psi 178 lbs.	200 sq. in. 35 psi 300 lbs.	

Figure 16. RELATIVE SIZE OF VALVE ACTUATOR FOR A 6-INCH VALVE.
Includes actuator power required and respective weight. To be used with example one.

SELECTING A VALVE TO DISSIPATE PRESSURE

EXAMPLE 2:

Transferring a liquid from a pressure vessel at a constant pressure to a pressure vessel maintained at a lower constant pressure. Objective: control the liquid level in the first vessel.

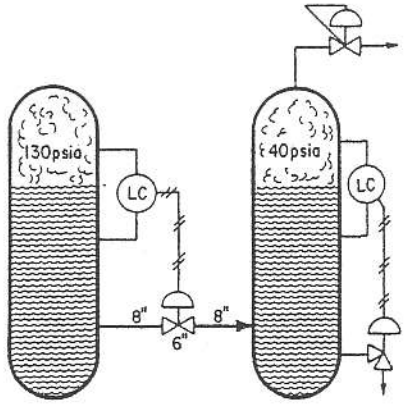


Figure 17. PROCESS CITED IN EXAMPLE TWO.

Q = 3600 gpm, water
T = 60°F.
P_1 = 130 psia constant
P_2 = 40 psia constant
ΔP = 90 psi constant

Calculate the required C_v under flowing conditions to handle 3600 gpm. This value must be less than the rated C_v for the selected 6 inch valve otherwise an 8 inch valve would be needed. This is done to prevent throttling near the full open position and therefore produce better control. Figure 18 contains the needed information.

REDUCING VALVE SIZE VERSUS THE INCREASE IN NOISE LEVEL

EXAMPLE 3:

In reference to Figure 19, note that the required installed C_v[8] value is less than the rated C_v[1] value so that all valves will handle the flow of 90,000 lbs./hr with adequate excess flow capacity. Also note that the butterfly and ball valves appear to be excessively oversized. What happens to the sound pressure level when these two valves are reduced to the 4 inch size? To avoid sonic velocity at the valve outlet maintain the valve's outlet diameter $\geq d$.

$$d = 0.118(1)\sqrt{\frac{W}{P_2}} = 4.5 \text{ inches which is too small.}$$

This exit velocity would be too high and noise calculations indicate these 4 inch valves would have SPL's of 135 dbA which is painful to the human ear. Exit velocity from 6 inch valves is approximately 230 ft./sec. This is well below 1/3 of the sonic value, which yields a less noisy operation. Since all valve types exceed the 90 dbA OSHA limit, if exposure of operating personnel is continuous over an eight hour period, then select a special low noise type of valve. Refer to Section XII. Otherwise, limit exposure time to maximum allowed in OSHA standards for dbA level.

[1] For steam only

Valve Type	Manufacturer's Rated C_v for 6 inches	F_L*	C_v Installed in 8-inch line	Remarks
Globe, single seated	440	0.90	387	Uses 97% of available C_v at 3600 GPM. Higher flow rates limited by beginning of cavitation.
Globe, double seated	450	0.98	389	Uses 86% of available C_v at 3600 GPM.
Rotary Eccentric Plug	500	0.85	391	Uses 78% of available C_v at 3600 GPM.
Diaphragm	600	0.70	477	P_1 is too high a value for diaphragm to atmosphere rating of valve.
Ball, characterized (segmented ball)	990	0.60	586	Cavitation will damage valve.
Butterfly (75° open)	1150	0.65	562	Cavitation will damage valve.

*As F_L approaches 1.0, a valve's pressure recovery value becomes lower and a tendency to cavitate lessens.

Figure 18. PERFORMANCE OF VALVE TYPES.
To be used with example two.

EXAMPLE 3:

Saturated Steam: W = 90,000 lbs./hr.
T = 400°F
P_1 = 250 psia
P_2 = 110 psia
ΔP = 140 psi

Valve Type	Mfg's Rated C_v[1] for 6-inch Valve	F_L[2]	Flow[3]	Calculated C_v[4]	Required C_v Installed in 8-inch Line F_{LP}	C_v[5]	Noise SPL 6 inch dbA[6]	Noise SPL 4 inch dbA	Remarks
Globe, Single Seated	400	0.90 Flow to Open Top Guided Contoured	Choked	221	0.86	223	100	—	6 inches, will handle. Excessive Noise.
Globe, Double Seated	450	0.98 V-Port	Choked	209	0.94	210	99	—	6 inches, will handle. Excessive Noise.
Rotary Eccentric Plug	500	.85 Flow to Open	Choked	231	0.80	233	100	—	6 inches, will handle. Excessive Noise
Butterfly (75° open)	1150	0.65	Choked	303	0.60	348	103	135 dbA[6]	Unsatisfactory-ΔP throttling too high.[7]
Characterized Ball	990	0.60	Choked	327	0.55	360	101	135 dbA[6]	Service is severe for thin metal seat ring and it is not recommended.
Diaphragm	600	0.70	Choked	282	—	295	101	—	Pressure and temperature too high for diaphragm material.

FOOTNOTES:
[1] C_v from manufacturer's rating tables.
[2] F_L = choked flow factor from manufacturer's engineering data.
[3] Flow is choked when $\frac{\Delta P}{P_1} > X_T$ or X_{TP}
[4] C_v (choked flow) saturated steam = $C_v = \frac{W}{2F_P P_1 \sqrt{X_{TP}}}$ Per RP-39.1
[5] To account for losses in reducers under choked flow conditions.
[6] Exceeds OSHA standard limit of 90 dbA for an 8-hour exposure. Low noise type valves are required.
[7] ΔP throttling limited to 28 psi in vane open position with standard weight valve and actuator.
NOTE: Reduction of valve size may lead to a higher noise level.

Figure 19. RELATIONSHIP OF VALVE SIZE AND NOISE GENERATED IN GENERAL STEAM SERVICE.
To be used with example three.

EXAMPLE 4:

Fluid: Superheated Steam
900° F
90,000 lbs./hr.
P_1 = 250 psia
ΔP = 140 psi

Maximum Leakage Allowed: 0.01% of Rated C_v.

Valve Types	Remarks
Single Seated Globe	Will handle conditions.
Characterized Ball	Not suitable, temperature limit of standard 316 stainless steel seat ring is 500°F. Will handle conditions with Inconel seat ring.
Double Seated Globe	Unsatisfactory, minimum leakage 0.5% of rated C_v.
Rotary Eccentric Plug	Unsatisfactory, 750°F temperature limit exceeded.
Butterfly	No suitable liner. Unlined valve will handle but minimum leakage 0.5% of rated C_v is excessive.
Floating Ball and Trunion Ball	Temperature limit exceeds elastomer seat materials required for tight shutoff. Stellite seats available in special high temperature valves do not give tight enough shutoff. Some special designs will handle conditions.
Diaphragm	No suitable elastomer material. Displacement is too great for a metal diaphragm.

Figure 20. SERVICE TEMPERATURE AND SHUTOFF REQUIREMENT NARROWS THE VALVE CHOICE.
(ISA-RP leakage class IV; measured with 50 psi air ΔP.)

EXAMPLE 5:
Natural Gas: P_1 = 1000 psia
P_2 = 14.7 psia
ΔP = 985.3 psi
T = 60°F
Q = 6,600,000 SCFH

Allowable leakage 0.1% of rated C_v.

Valve Types	Remarks
Double Seated Globe (high quality)	Will handle conditions but with a high noise level. ΔP shutoff is too high to throttle near the seat at low flow rates. Unstable if contoured plug design.
Cage, Balanced Single Seated Globe	Will handle conditions.
Single Seated Globe (non-balanced)	Could handle but actuator size, bulk and cost are excessive for a non-balanced design.
Solid Ball	
Floating Ball	Will handle conditions.
Trunnion Ball	Will handle the conditions with 1/3 to 1/2 the torque of Floating Ball Valve.
Characterized Ball	Body pressure rating exceeded. ΔP rating exceeded.
Butterfly	Pressure rating of standard valve exceeded. A valve built with heavy vane and extra heavy shaft would have a very low flow capacity. High ΔP BFV's are available in larger sizes.

Figure 21. HIGH PRESSURE SHUT-OFF NARROWS THE VALVE CHOICE.
(ISA-RP 39.6 leakage Class III; measured with 50 psi air ΔP.)

EXAMPLE 6:

Valve Types	Remarks
Double Seated Globe	Unsatisfactory, leakage excessive, 0.1% to 0.5% of rated C_v. Could be handled with elastomer soft seats on both plugs but expensive.
Solid Ball	
Floating Ball	Will handle conditions. Bubble tight shutoff.
Trunnion Ball	Will handle conditions.

Figure 22. CHANGING ALLOWABLE LEAKAGE TO 0.01% OF RATED C_v.
(ISA-RP 39.6 leakage Class IV; measured with 50 psi air ΔP.)

EXAMPLE 7:

Valve Types	Remarks
Solid Ball with metal seats (Lubricated with viscous sealant) Trunnion type preferred	Will handle conditions. Sealant good to 600°F.

Figure 23. CHANGING TEMPERATURE TO 500°F AND MAINTAINING ALLOWABLE LEAKAGE AT 0.1% OF RATED C_v.
(ISA-RP 39.6 leakage Class IV; measured with 50 psi air ΔP.)

EXAMPLE 8:

Valve Types	Remarks
Pilot Operated Single Seated Cage Balanced Globe Valve	1) Will handle examples 3 to 8 with smallest actuator. 2) Will throttle to full ΔP at any flow rate. 3) Higher cost than other types due to complexity of internal pilot valve which is a part of the main valve plug. 4) Temperature to 1050°F.

Figure 24. A SPECIAL TYPE OF GLOBE VALVE.

EXAMPLE 9:

Valve Types	Remarks
Globe Types	Rapid erosion,[1] unsatisfactory.
Rotary Eccentric Plug (metal seated)	Handles some services with limited life.
Ball	
Characterized	Excessive erosion. Longer life without seat ring if higher leakage is permitted. Special designs available to extend life are costly.
Solid	Limited life with standard valves. Longer life with special designs.[2]
Butterfly	Excessive vane erosion, unsatisfactory.
Diaphragm (with elastomer lined body)	Will handle conditions with long life providing the velocity is less than 6 ft./sec.
Pinch Type Valve	Will handle conditions.

Figure 25. ABRASIVE SLURRY CONSIDERATIONS.

[1]Special angle type globe valves with ceramic or tool steel seats and plugs often give adequate service. Use a sweep (long radius) angle body with seat orifice having a rounded entrance and long bore. An oilfield adjustable choke is an example of this type.

[2]Use integral ball and stem valves to prevent abrasive wear at the joint, hardened ball surface and solid stellite seat rings. This design will handle sand and gravel, flyash, iron ore dust, tailings. For more information, see Chapter 4, the Ball Valve Section.

EXAMPLE 10:

Process: Manufacture of "Food Board" that is used as a sanitary cardboard for food containers such as milk cartons.

Fluid: Fibrous Slurry Paper Stock, 4.5% in water.

Requirement: Stock must not hang up in valve where it will then ferment and sour product.

Valve Types	Remarks
Characterized Ball Valve	Will handle conditions. Any fibers lodged on ball or thin seat ring are sheared on closure.
Butterfly Valve	Stock above 2% will hang up on vane.
Diaphragm Valve	Fibers hanging on bridgewall are not sheared when elastomer diaphragm seals to bridgewall on closure.
Rotary Eccentric Plug	Not satisfactory — clogs.
Globe Valves	Not satisfactory — clogs.
Solid Ball (Floating or Trunnion Type)	Unsatisfactory — body cavity clogs.

Figure 26. VALVE COMPATIBILITY WITH FLUID CONTENT

EXAMPLE 11:

Process: Pulp and Paper Manufacture.

Fluid: "Black Liquor" from chip digester: Content is injurious to elastomer materials.

Valve Material: 316 stainless steel required.

1.35 spg.
50 to 60% solids.
190 gpm (3-4 inch Valve)
200°F.
ΔP = 8 psi

Valve Types	Remarks
Characterized Ball Valve	Will handle conditions.
Butterfly Valve	Fluid destroys elastomer lined. TFE lined too expensive, leakage rate of unlined valve too high.
Diaphragm Valve	Fluid destroys elastomer. Could use TFE faced diaphragm.
Globe Valve	Solids content too high — will clog.
Solid Ball (Floating or Trunnion Type)	Unsatisfactory — body cavity clogs.

Figure 27. VALVE COMPATIBILITY WITH FLUID CONTENT.

EXAMPLE 12: RATIOS OF VALVE COSTS WITHIN GIVEN TYPES

Example	Body*	Single Seated Top Guided 200 sq. in. Diaph. Act.	Double Seated Top & Bottom Guided 145 sq. in. Diaph. Act.
1)	Iron, 125 lb. Standard Construction 150 psi ΔP shutoff	1.1	1.0 (base)
2)	Steel, 150 lb. Standard Construction 150 psi ΔP shutoff	1.7	1.6
3)	Steel, 300 lb. Standard Construction 150 psi ΔP shutoff	1.7	1.6
4)	Chrome Moly Steel, 300 lb., Construction for 900°F, Extension Finned Bonnet Stellited Seat Joint Stellited Guide Bushings Stellited Plug Guide Posts	2.9	3.0
5)	Steel, 600 lb., 1200 psi ΔP, 60°F, 0.5% C_v maximum leakage Stellited Seat Joint	2.1	1.8
6)	Steel, 600 lb., 1200 psi ΔP, tight shutoff, 60°F. Stellited Seat Joint High Pressure Piston Actuator (90 psi) Not Fail Safe	2.1	—
7)	Chrome Moly, 600 lb., 1200 psi ΔP, 600°F. Pilot Operated Balanced Design Highly Reliable Fail Safe Action	2.8	—

*This information is from a 6-inch globe valve.

Figure 28. UPGRADING FOR INCREASING SEVERITY OF SERVICE.

Use With Globe Valves.

Body	Cost Ratio*
Steel	1.0 (base)
Stainless	1.5
Alloy 20	2.7
Hastelloy C	3.7

*This information is for a 3-inch flangeless single seated globe valve, ANSI 300 lb., with a 140 sq. in. diaphragm actuator.

Figure 29. STANDARD AND HIGH ALLOY VALVES FOR EROSIVE SERVICE.

Use with Flangless Globe Valves, 1 to 4 inches.

Body	Liner	Cost Ratio[1]
Iron	Unlined	1.0 (base)
Iron	Buna-N	1.0
Steel	Unlined	1.2
Steel	Buna-N	1.4
Stainless	Unlined	1.5
Iron	Viton[2]	1.7
Steel	Viton[2]	2.0
Steel	Unlined	1.2 +
(Heavy Duty — Higher ΔP, Smaller Flow Capacity)		
Stainless	Unlined	1.5 +
(High Temperature — Piston Ring Seal)		

[1]This information is for a 6-inch butterfly valve, ANSI 125, 150 and 300 lb., with a 140 sq. in. diaphragm actuator.
[2]Higher temperature operation.

Figure 30. LINED AND UNLINED BUTTERFLY VALVES.

Body	Cost Ratio[1]
Steel	1.00 (base)
Steel, Stellited Seat Joint	1.05
Steel, Soft Seat, TFE	1.06
Stainless	1.30

[1]This information is for a 6-inch flangeless rotary eccentric plug valve with a 20 sq. in. cylinder operator in the ANSI 150, 300 and 600 lb. class.
[2]One body serves all ratings to 600 lb.

Figure 31. ROTARY ECCENTRIC PLUG VALVE.

Body	Cost Ratio*
Steel, 150 lb.	1.0 (base)
Stainless, 150 lb.	1.2
Steel, 300 lb.	1.4
Stainless, 300 lb.	1.9

*This information is for a 6-inch characterized ball valve with a stainless seal ring and a 140 sq. in. diaphragm actuator.

Figure 32. CHARACTERIZED BALL VALVES.

Body	Cost Ratio[1]
Iron	1.0 (base)
Rubber Lined (Iron)	1.3
Carbon Steel	1.5
Glass Lined (Iron) (with TFE Diaphragm)	1.7
Stainless Steel	2.1

[1]This information is for a 6 inch Saunders Type Diaphragm Valve with standard diaphragm actuator.

Figure 33. SAUNDERS TYPE DIAPHRAGM VALVE.

SECTION IX
EASE AND LOCATION OF MAINTENANCE

To successfully use a control valve the user must first decide where, how and who will maintain it. Control valves constantly throttle fluid flow and are thus subject to wear and the need of replacement of trim. Primary maintenance is performed in the field with the valve either in the line or removed if required to reach the internal parts. Secondary maintenance is performed in the users valve repair shop whereas tertiary maintenance is the responsibility of the manufacturer at his plant or licensed field repair service shops. Make these decisions before attempting to select a particular type of valve.

Figure 34 lists specific valve types and the type of maintenance that may be performed by the three groups. The degree of probable maintenance in the specific service areas is best known by user experience or may be based on the manufacturer's recommendations. Select the valve type which can be matched with appropriate plans for its maintenance.

Degree of Maintenance	Means of Maintenance	Valve Types	Trim Parts Replaceable	Remarks
Primary	Field Maintenance Quick Change Trim (Valve body remains in piping)	Globe — top entry cage	All	
		Barstock-bottom entry cage	"	
		Plug	"	
		Ball — top entry design	"	
		standard Globe	Plug, stem	Screwed seat rings often require shop removal.
		Saunders	All	
	Remove Body from piping	Split Body	All	1) Certain liner designs are removable without disturbing stem & disc. Both in-line & offset.
		Angle w/tail pc. seat	"	
		Globe w/remov. bridgewall	"	
		Ball — 2 or 3 pc. bolted body	"	2) Disc & stem when bolted may be removed.
		Butterfly	See remarks	3) Wrap around and slip-in liners may be replaced provided disc is pinned or bolted to shaft. Liners vulcanized to body are not replaceable. Liner in offset disc design is replaceable without removing stem or disc.
		Pinch	All	
		In Line Gate	"	
		Gate (sliding gate with slots)	Disc, stem	Some seats are slip-in-type which are removable.
Secondary	Users Valve Maintenance Shop	Globe	Screwed in seat rings, welded seat rings	
		Butterfly	Stem, disc	Liner if not bonded to body.
		Gate (sliding gate with slots)	Seat — pressed in type	
Tertiary	Return to Manufacturer	Globe	Replacement of hard-faced integral seat joint	
		Ball — all welded body	All	It is often economical to overhaul large pipe line ball and plug valves even after 10 to 20 years of service.
		TFE Lined Valves	Replace lining or scrap	

Figure 34. EASE AND LOCATION OF MAINTENANCE.
Select valves depending on where and how the user must maintain them to keep his process in operation with minimum down time.

SECTION X

ANTI-CAVITATION VALVES

The tendency of a valve to cavitate under given service conditions may be predicted with the method given in Chapter 6. Standard valve types which have a high dimensionless liquid pressure recovery factor (F_L) have less cavitation than valves with a low (F_L). In other words, valves which recover downstream of the seat orifice vena contracta, a substantial part of the pressure drop lost across the seat has a greater tendency to cavitate than valves with low pressure recovery. These (F_L) factors listed by valve type and their respective variation with percent of valve rated C_v flow rates are illustrated in Figure 35. A chart of F_L values (at full open) versus specific valve type and body designs is also included. (See Figure 36.) Note that globe type valves have a lesser tendency to cavitate than ball and butterfly valves.

Figure 37 lists valve types in order of their non-cavitation performance and follows with body type, size range, differential pressure limit, and a relative cost factor over conventional valves of the same capacity. Specific designs shown in various locations in this handbook are referred to by Chapter and Figure Number.

The type of valve selected for best service life depends upon the degree of cavitation above incipent cavitation as illustrated in Chapter 6. Consult the valve manufacturer for additional specific information.

First, calculate whether or not cavitation is predicted for the particular valve type you are interested in. If cavitation is imminent, select a valve having a higher F_L factor at normal flow conditions (not rated C_v) and repeat the calculation. If cavitation still is predicted, it is preferable to go to the use of a special anti-cavitation valve. Use of hard trim in standard valves adds additional cost for the original trim, and for scheduled replacement of trim. Also, body erosion may occur. While mechanical damage may be acceptable with some valve styles, on some services, this cannot be predicted with any reliability.

The following information gives an overall view on the concept of special valves and their principles of operation which reduce the tendency to cavitate in liquid flow.

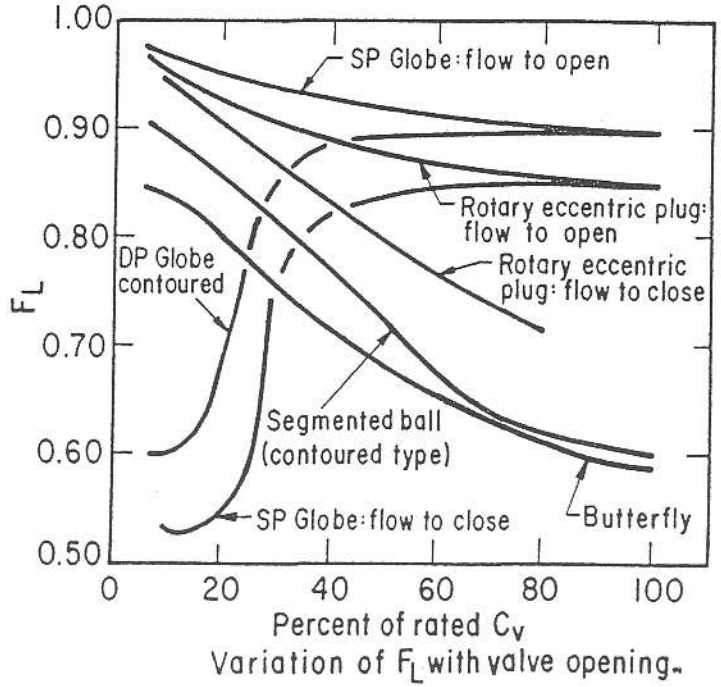


Figure 35. VARIATION OF VALVE PRESSURE RECOVERY FACTOR (F_L) WITH PERCENT OF C_V FLOW RATE.

Courtesy of Masoneilan International Inc.

Valve Type	Trim Size	Flow To	F_L
Single Seated Globe	A	Close	.85
		Open	.90
	B	Close	.80
		Open	.90
Rotary Eccentric Plug	A	Close	.68
		Open	.85
	B	Close	.70
		Open	.88
Double Seated Globe	A	Contoured V-Port	.90
			.98
	B	Contoured V-Port	.80
			.95
Split Body Globe Valves	A	Close	.80
		Open	.75
	B	Close	.80
		Open	.90
Sweep Angle	A	Close	.48
		Open	.90
	B	Close	.55
		Open	.95
Butterfly	A	Flow in Either Direction	.65
Ball, Characterized (Contoured)	A		.60
Globe, Balanced Quick-Change Trim	A	Close	.90
	B	Close	.90
Regular Angle	A	Close	.81
		Open	.89
	B	Close	.80
		Open	.90
Y Valves	A	Close	.75
		Open	.75

Anti-Cavitation Valve Characteristics

General Information

1) Flow capacity is about 1/5 of that of a conventional valve; however, if differential pressure falls off as the valve opens, the stacked disc design may have an open cage at the top. This design has about 1/2 the capacity of a conventional valve.
2) May be pressure balanced to improve throttling stability.
3) Normally built in 600 to 4500 lb. ratings.
4) TFE soft seats are available for drop tight closure.
5) Much more expensive than conventional valves.

Principles of Reducing Cavitation

1) 90° bends in the flow path create a series of velocity head losses reducing pressure gradually.
2) Pressure drop may be divided across a series of orifices.
3) A combination of principles 1 and 2 having multiple small differential pressures rather than one larger differential pressure to keep the liquid above its vapor pressure so that cavitation will not occur.
4) Flow may be jetted against flow and swirled to create a massive turbulence and internal friction to dissipate the energy as heat.
5) Cavitation as in 4, results in collapse of bubbles in the center of the cage away from the seat joint.
6) Using the service conditions calculate whether or not the valve selected will cavitate. If cavitation is greater than incipient (minor) then select equipment from Figure 37.

VALVE TYPES AND DESCRIPTION Listed in order of Non-Cavitating Performance. Best valves are listed first.	Body Style	Body Size Range	Body ΔP Limit, psi	Relative Cost Factor[1]	Reference Chapter	Reference Figures	Remarks
Cage type stacked discs (having multi-flow channels with 90° bends to create velocity head losses)	Globe Angle	—	—	3-10	IV	34	Flow direction is over the plug. Must handle relatively clean fluids to prevent clogging of small orifices. Fits in standard globe valve body.
Multi Port (plugs and orifices in series) (Also known as cascade flow with additional configurations to increase turbulence)	Long Body Angle	1-6"	6000 psi	1.5-3.0	IV	19, 20	Divides overall ΔP across four orifices.
Ball, characterized (segmented ball with tube bundle in flow-way)	Ball	6-16"	500 psi	1.5-2.0	IV	—	Tube bundle-like orifice flow straightening vanes fits into standard ball flow-way.
Ported Flow Cage (holes in opposite sides of cage jet flow together to create turbulence in center of cage. Holes may be stepped or threaded for additional effect)	Globe Angle	2-18"	—	1.5	IV	18 (a)	Fits in standard cage type globe valve body.
Two Valves in series	Globe	1-12"	—	2.0	—	—	Dividing ΔP.
Orifice Choke installed downstream of valve (hardened orifice — tungsten carbide or Stellite if hot service)	—	—	—	1.5	—	—	Divided pressure drop creating back pressure on valve to reduce cavitation.

[1]Based on standard control valve equals 1.0.

Figure 37. RELATIVE VALVE CAVITATION PERFORMANCE.

SECTION XI
LOW AERODYNAMIC NOISE VALVES

Valve noise is normally generated by the following three conditions: 1) aerodynamic noise from gas flows, 2) flow induced vibration of valve parts, and 3) cavitation noise in liquid flow. This section deals only with aerodynamic noise from gas or steam flow. Chapter 7 has more information concerning the others.

General Characteristics

The following information is in reference to the special valves that are designed to reduce aerodynamic noise levels:

1.) Flow capacity is about 1/5 to 3/4 that of conventional valves.
2) May be readily pressure balanced to improve throttling stability.
3) Normally built in 300 to 2500 lb. ratings.
4) Effectively handle gas or steam at high temperature.
5) More expensive than conventional valves.
6) Auxiliary equipment may be added to increase attenuation.

The principles of reducing aerodynamic noise are as follows:

1) 90 degree turns create a series of velocity head losses reducing pressure.
2) Expanding area with pressure drop and volume increase keeps gas velocity well below sonic. (<1/3 Mach 1.0).
3) Multiple orifices raise the frequency level allowing better attenuation by metal components.

To use Figure 38, first one must calculate the noise level predicted for your specific service conditions as discussed in Chapter 7. Next, decide what level of noise is acceptable or permitted for your installation. The difference is the amount of attenuation in dbA required. Figure 40 list values of attenuation required under a heading in the left hand column. Valve types which are capable of obtaining this degree of attentuation while under certain flow conditions are listed in the second column. Body style, size range, differential pressure limit and cost factor over a conventional valve of the same size follow. By referring to the chapter and figure number, more information may be located in other parts of the handbook on a particular type of valve.

As a final check, recalculate the attentuation for the service conditions applied to the particular valve selected using the specific manufacturer's noise methods. If insufficient, pick out a higher attentuation range anti-noise valve.

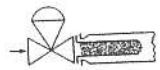
Attenuation Required dbA	Valve Type	Style	Body Size	ΔP psi	References Chapter	References Figures	Remarks
25	Multi-turn expanding area flow cage	Globe	2-12"	3,000	IV	34	See note 2.
		Angle	2-6"	3,000	III	37	See note 2.
		Fabricated Angle	—	10,000			Readily built in sizes to 36" for any P_1/P_2 ratio and noise attenuation.
10 to 25	Multi-turn expanding area tapered plug	Angle	2-6"	5,000/ 4,000	IV	35	See note 3.
10 to 20	Multi-orifice double flow cage	Globe	2-12"	1,500	IV	33 (d)	
10 to 12	Multi-orifice flow cage (round or rectangular orifices)	Globe	2-12"	3,600	IV	33 (a) (b)	Fits in conventional valve body. Uses conventional actuator.
	Multi-V-Port plug (generating triangular orifices with seat)	Globe	1/2-4"	500	IV	33 (c)	Used in gas regulators but may be applied to any globe valve.
	Multi-turn expanding area	Double Seated Globe	4-14"	1,100/ 270	III		Fits conventional double seated valve body. Has multiple orifice plate in outlet. Uses conventional actuator.
	Multiple orifice in series plus expanding area	Long Body Angle	1-6"	6,000/ 1,440	IV	19 (b)	Non-clogging design. TFE metal reinforced drop-tight soft seat available.
6 to 9	Multi-orifice sliding gate valve	Gate		200	IV	9 (g)	Positively loaded non-vibrating gate.
	Auxiliary Equipment						
to 25	Multiple orifice plates (one, two, three sets)	Installed between flanges	3-12"	1,500	—	—	Install first set on valve outlet flange, 10-15 dbA attenuation per set. Multiple sets to 25 dbA. Plate set is two plates with larger orifices in downstream plate.
		Welded-in Nipple	4-36"	1,500/ 300	—	—	For large piping and high temperature service.
	 Multiple orifice thimble	Installed between flanges	—	500	—	—	Install on valve outlet flange. Single stage of orifices in thimble.

Figure 38. LEVEL OF ATTENUATION OF NOISE BY VALVE TYPE AND AUXILIARY EQUIPMENT.

Attenuation Required dbA	Valve Type	Style	Body Size	ΔP psi	References Chapter	References Figures	Remarks
to 20	Silencers	Packed	No limitation except cost	Designed for minimum drop	VII	—	Packed designs should not be used with moisture in fluid or they will clog and may be damaged by increased ΔP. Acoustical type not subject to moist steam restriction. Occupy considerable space - and diameter. Large sizes are expensive.
	Downstream [4]	"	"	"	VII	—	Attenuates noise proceeding upstream when flow thru valve is below critical (sonic).
to 8	Insulation (applied to valve body and piping)	—	No limit	—	VII	—	Provides noise reduction only in area of application.
to 5	Heavy wall piping	—	Cost & availability limitations	—	VII	—	Attenuation in area of heavier piping but noise exits from lighter weight piping downstream.
USE METHODS GIVEN IN NOISE SECTION IN TEXT TO CALCULATE NOISE LEVEL OF CONVENTIONAL VALVE, THEN SELECT ANTI-NOISE VALVES & EQUIPMENT BASED ON ATTENUATION REQUIRED[1] FOR SERVICE APPLICATION.							

[1] Find attenuation required in (dbA) by subtracting allowable noise level from noise level calculated for standard valve in this application.

[2] Flow capacity may be increased to 50% that of a conventional valve by having the upper half of the cage open if ΔP falls off with valve opening. Characterized flow is obtained by varying the number of orifices in each disc as the valve strokes. A long stroke piston actuator is often required. Subject to plugging even with relatively clean fluids. Example: — Boiler feedwater pump recirculation valve designed to operate at 40% opening with full flow when new operates at 60% opening two years later. Oversizing extends life without degrading flow control.

[3] Non-clogging design is suitable for erosive high pressure drop service. A short stroke is required on lever type diaphragm actuator. Example: — One valve handled sand fines in gas well flow lines at drops to 5000 psi several years with minor trim wear (316 stainless steel trim with hard-faced seat joint). This design is also used in high pressure steam pressure reduction.

[4] Sizes range from 6 inches to 6 feet in diameter and 4 to 40 inches in length. Relates to downstream silencers only.

Figure 38. LEVEL OF ATTENUATION OF NOISE BY VALVE TYPE AND AUXILIARY EQUIPMENT. CONT'D.

SECTION XII
VALVE TYPES

Valve types are selected to best handle some specific type of fluid and/or control action. An alternate selection may be required if the type desired is not available in the proper material, size, and pressure-temperature rating.

Tables and diagrams for this section present types of control valve configurations and lists their respective size, pressure-temperature range, plus important application information. Valve types are grouped by mechanical function, such as linear motion of plug to seat and rotary motion of plug to seat. In-line and diaphragm valves are sub groups of linear motion valves. Review Chapter 3 before making a selection.

Valve sizes, pressure ratings, temperature range, leakage rates and other data are given for quick reference to aid in selecting a particular design within a given valve type.

LINEAR MOTION TYPE

Globe Style

Double Seated

¾-16" (24" Max.)
150-1500 lb.
General Service.

Minimum Leakage 0.5% of Rated C_v.

Smaller Actuator Force Requirement than Single Seat.

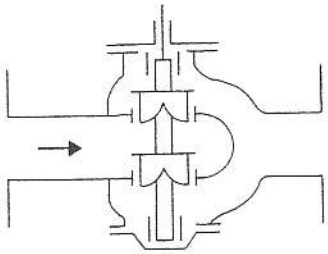


Figure 39.

Single Seated — Top and Bottom Guided

1-16"
150-1500 lb.
General Service.

Minimum Leakage .01% of Rated C_v or less with a more powerful actuator.

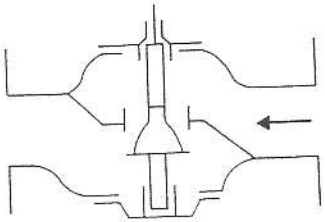


Figure 40.

Single Seated — Top Guided

1/2*-16" (30" Max.)
150-600 lb. (10"), 900-2500 lb (6")
General Service.

Minimum Leakage: .01% of Rated C_v. Tight Shutoff Available.

*Smaller sizes in ¼" bodies

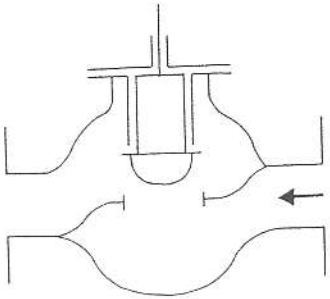


Figure 41.

Removable Bridge Wall Design

1-4"
150-300 lb.
Moderate General Service.

Corrosive service in Alloy Construction.

Quick Change Trim Through Valve Pipe End.

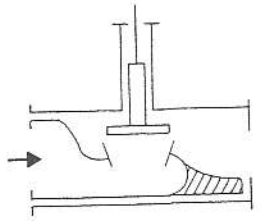


Figure 42.

Split Body Design

½-12"
150-600 lb (12"), 900-1500 lb (10")
2500 lb. (6")
Chemical & General Service.

Seat Ring readily Replaced.

Application limited by Temperature and Piping Strain.

Globe, angle, or right angle assembly.

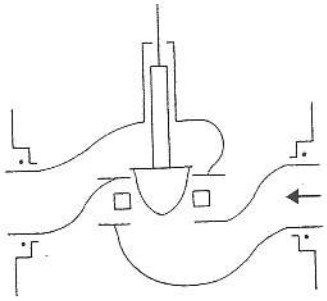


Figure 43.

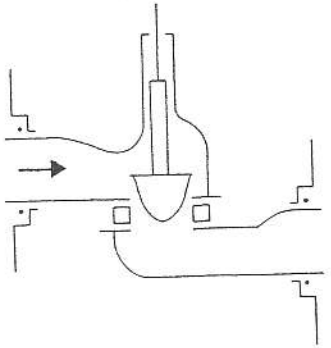
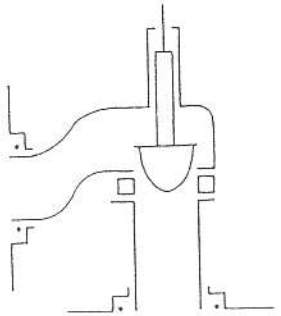


Figure 44.

Split Body Design

Full Draining Offset

½-4"
150-600 lb.



Angle

¼-1½"

Pilot Operated Cage Design

6-12"
600-1500 lb.
Steam Dump Service in Fossil Fuel & Nuclear Power Stations.

Tight Shutoff.

Pressure Balanced.

Low Actuator Force Requirement.

Pilot opens first to establish balance across main plug.

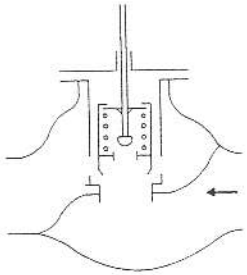


Figure 46.

Cage Design

1-12"
150-2500 lb.
Better Plug Guiding.

More Stable Throttling.

Quick Change Trim Through Bonnet Opening.

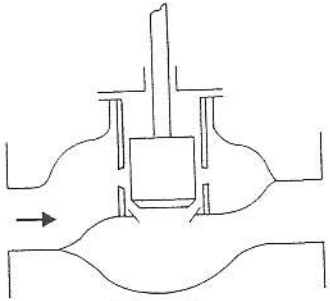


Figure 47.

1-12"
150-4500 lb.
Balanced Construction for High Pressure Drop Applications and Smaller Actuators.

Low Overall Weight and Size.

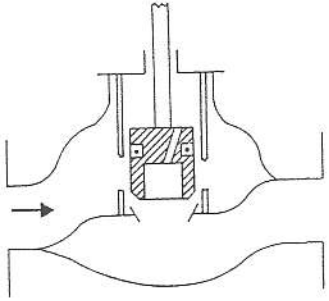


Figure 48.

1-4"
150-4500 lb.
Cage Design for Erosive Service.
Note annulus between cage and plug.

Plug is top guided above cage with a heavy guide combined with

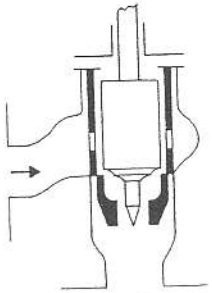


Figure 49.

Angle

½-12"
150-600 lb. (12"), 900-1500 lb. (10"), 2500 lb. (6")
Special designs to 50,000 psi.

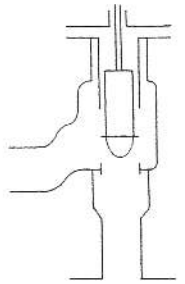


Figure 50.

Sweep Angle

2-16"
300-1500 lb.
High Drop Dirty and Erosive Service.

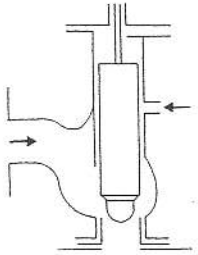


Figure 51.

"Y" Style

¾-14" (36 inch max. for steam service)
150-600 lb. (14"), 900-1500 lb. (10"), 2500 lb. (6")
Low Flow Resistance.

Used More Frequently in On-Off Service.

Corrosive Service. (Through 14 inches)
Full Flushing Body Cavity.

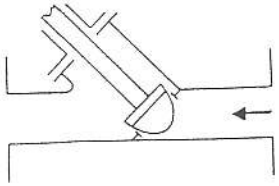
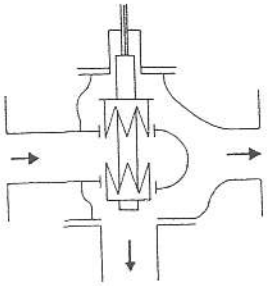


Figure 52.

3-Way Valves

Diverting Service

1-16"
150-600 lb.
Non Shut off.



Mixing Service

1-16"
150-600 lb.
Non Shut off.

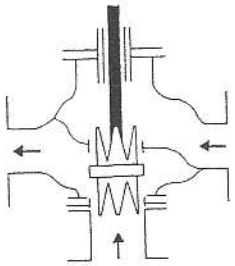


Figure 54.

Split Body

1-10"
150-600 lb, 2500 lb (1½")
Combined Diverting
and Mixing.

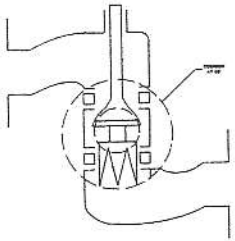


Figure 55.

Balanced Design

½-6"
150-600 lb.
Differential pressure to 1440
psi with standard actuator

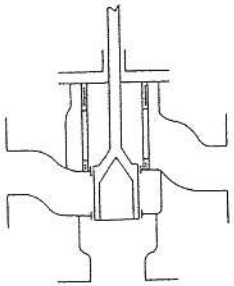


Figure 56.

3-Way 3-Position

1-4"
To 5000 psi W.O.G.
Full Off Position — All
Ports Simultaneously.
Combined Diverting & Mixing.
High Pressure Balanced
Designs Available.

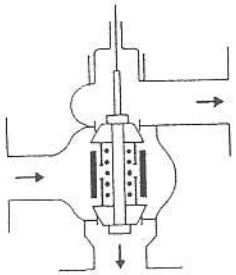


Figure 57.

Anticavitation Valve Types

Turbo Cascade

1-4"
To 4500 lb.

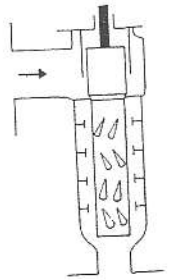


Figure 58.

Multiple Orifice Restrictions

1-6"
6000 psi

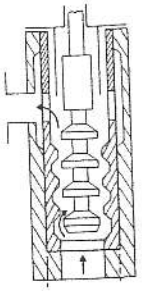


Figure 59.

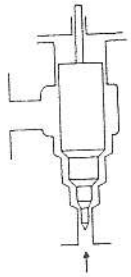


Figure 60.

Multiple Velocity Head Loss

Multiple Orifice Cage

To 2500 lb.
To 18"

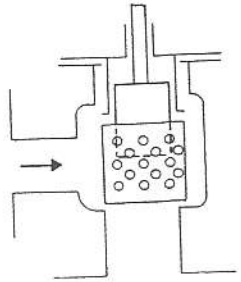


Figure 61.

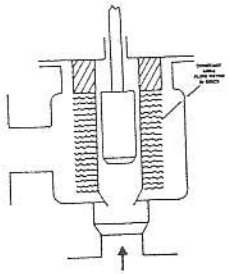


Figure 62.

Cage Disc with Tortuous Flow Paths

150 to 2500 lb.

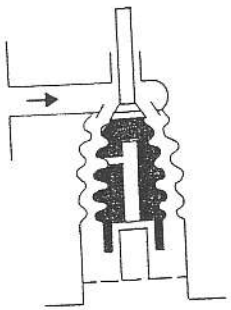


Figure 63.

Low Aerodynamic Noise Valve Types

Angle With Expanding Area

Flow cone and right angle turns each creating a velocity head loss.

2-8"
To 6000 psi

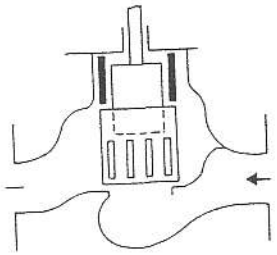


Figure 64.

Slotted Cage

2-12"
150 to 1500 lb.

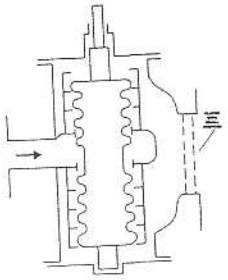


Figure 65.

Double Seated with Double Plugs Having Right Angle Turns
Expanding Area Flow Path.

4-14"
300 to 600 lb.

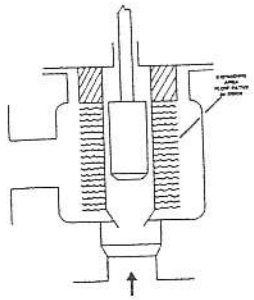


Figure 66.

Cage with Disc Having Expanding Area Flow Path

Right angle turns create a velocity head loss.

To 2500 lb.

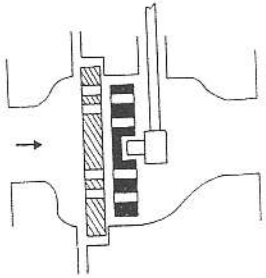


Figure 67.

Throttling Gate Valves

Sliding Gate

1 to 6"
300 lb. Tight Shut Off.
General & Mild Erosive Service.

Some Noise Reduction.
Orifices in gate and seat plates.

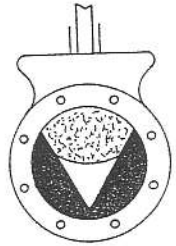


Figure 68.

"V" Orifice Gate

2-24", 150 psi W.O.G.
30-36", 125 psi W.O.G.

(W.O.G. means water, oil, gas non-shock rating at 100°F.)

Used for Paper Stock.

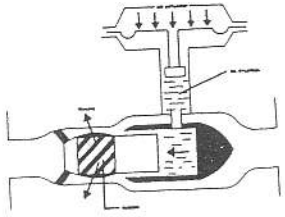


Figure 69.

IN-LINE VALVES

Swell Plug

1-6"
150 to 600 lb.
Tight Shut Off.

Gas Pressure Regulating Service.
Some Noise Reduction.
Erosive Liquid Service.

Hollow Mandrel

2"
10,000 psi

Flow Rate Energized.
Shut Off & Throttling.

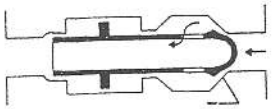


Figure 70.

Digital Controlled Multi-Poppet

Instantaneous flow rate adjustment from computer signal.

Solenoid poppet valves control on-off as many as 8 orifices of different sizes. By varying combination of orifices multi-step precision flow is obtained with almost instantaneous change to new set point.

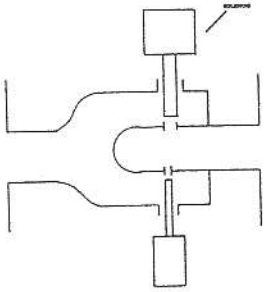


Figure 71.

ROTARY MOTION

Butterfly

1-150"
-460 to 2100°F.
25-125-300-600 lb.
Lower Pressure Ratings as Size Increases
High Flow Capacity with
Moderate Pressure Drop.
Available lined to Prevent Corrosion

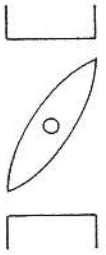


Figure 72.

Three-Way Valve Action

Building two valves into a flow tee and linking their vane arms for combined actuation.

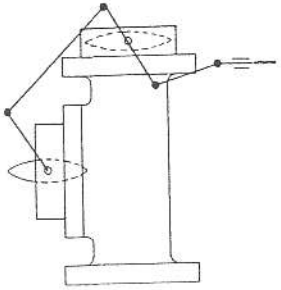


Figure 73.

Offset Disc And Shaft

2-20", 150 lb.
Double Seat Available.

Uninterrupted Disc Seating Surface.

Fewer Stem Seal Leak Problems.

Available with metal seal ring for high temperature service.

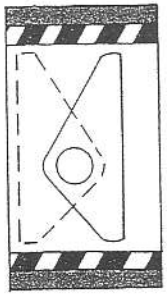


Figure 74.

Canted Disc

2-36"
300 lb.

Uninterrupted Disc Seating Surface.

Fewer Stem Leak Problems.

Available with metal seal ring for high temperature service.

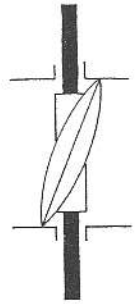


Figure 75.

Canted Stem

Uninterrupted Disc Seating Surface.

Fewer Stem Seal Leakage Problems.

Awkward actuator mounting.

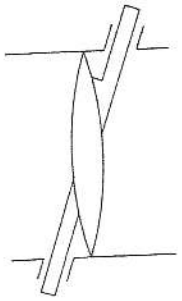


Figure 76.

Externally Pressure Energized Liner Sealing

To 600 psi differential pressure.

Tight Shutoff at higher pressures than can be handled by liner compression by disc interference alone.

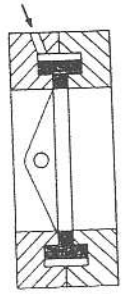


Figure 77.

Unlined Body

2-150"
200 psi
-460 to 1500°F.

Minimum Leakage Rate 0.5% of Rated C_v.

Used for services not requiring tight shutoff and whose line fluids would degrade elastomer liners.

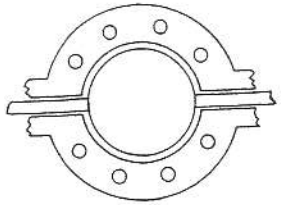


Figure 78.

Lined Body

Elastomer

2 to 72"
-20 to 400°F.

Bubble tight shutoff.

Replaceable liner design available.

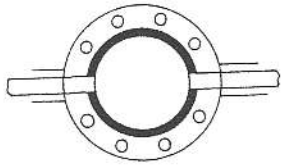


Figure 79.

TFE Lined (Split Clamp Design)

Integral disc & stem
-20 to 300°F.
3 to 36"

For corrosive fluid.

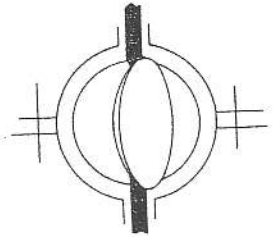


Figure 80.

Three-Way Valve Action

Tee body with two BFV's linked together so that one closes as the other opens.

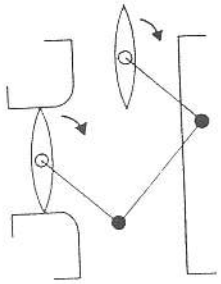


Figure 81.

CAM ACTION PLUG VALVES

Eccentric Cylindrical Plug (Elastomer Faced)

½ to 36"
To 250 psi
Tight shutoff.

Corrosion resistant linings available.

Also available with V-Port Plug.

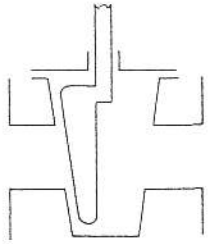


Figure 82.

Eccentric Disc Plug

1-12"
150 to 600 lb.
-320° to +750°F

General Services.

Flangeless design.

Tight shutoff available.

Low actuating force requirement.

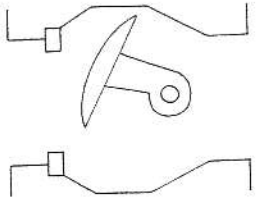


Figure 83.

THROTTLING PLUG

Cylindrical Plug

1 to 30"
150 to 600 lb.

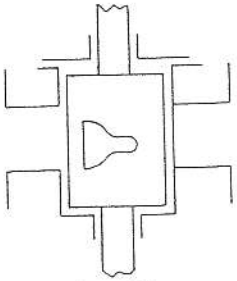


Figure 84.

Tapered Plug

Mainly pipeline applications.

1 to 30"
150 to 600 lb.

Also available in stainless body with TFE seat inserts.
1-6 inches, 150 lb.

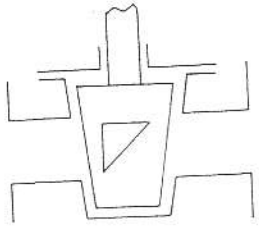


Figure 85.

Cage Plug

1 to 6"
150 & 300 lb.

Throttling cage fixed within a rotating plug.

400°F Max.
150 psi differential pressure Max.

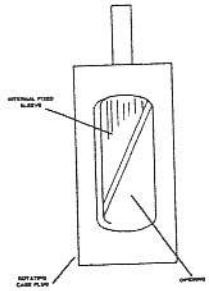


Figure 86.

BALL VALVE

Full Ball

¼ to 72"
150 to 2500 lb.

Least flow resistance of any type.

Tight shutoff.

Block and bleed action to check seat tightness.

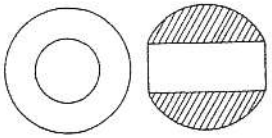


Figure 87.

Floating Ball

¼ to 36"
To 2500 lb.

Tightest shutoff.
Highest torque.

General Service,
Chemical & Pipeline Valve.

TFE Lined Valves.

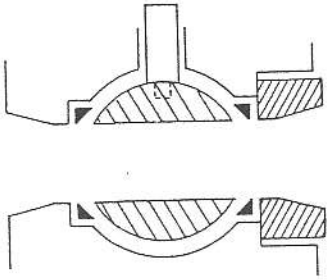


Figure 88.

Trunnion Supported Ball

2 to 72"
300, 900, 1500 lb.
Tight shutoff.

Moderate torque.

High pressure drop Service.

Pipeline Valve.

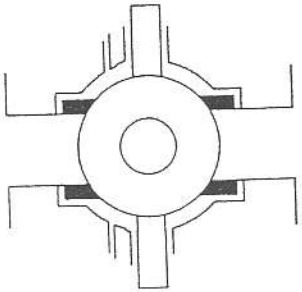


Figure 89.

Characterized Segmented Ball

2 to 16"
150-300 lb.

Shear action closure and non-clogging features make this design suitable for paper pulp service.

Also general service.

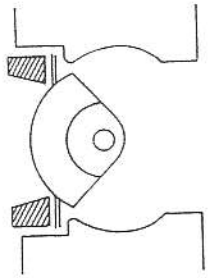


Figure 90.

Three-Way Ball

Floating Ball Design.

2 to 12"
150-300 lb.

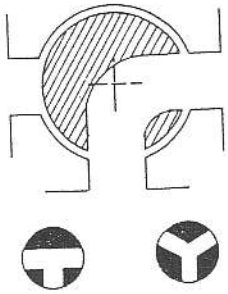


Figure 91.

DIAPHRAGM ACTION

Saunders

¼ to 20"

Full vacuum to 150 psi.
Tight shutoff.

Available with many different linings and diaphragm materials to resist corrosion and erosion.

Limited throttling characteristic.

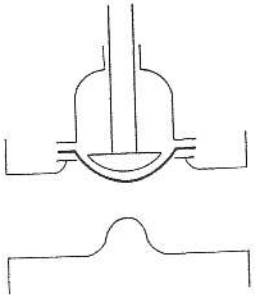


Figure 92.

Wedge Plug

½ to 12"

Full vacuum to 100 psi.

Highest flow capacity of this type.

Lined bodies available.

Lower differential pressure limits than Saunders type.

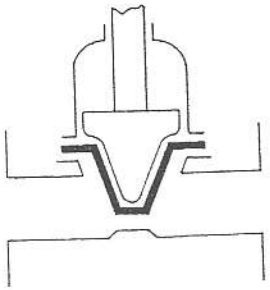


Figure 93.

Pinch

1 to 12"

Full vacuum to 150 psi.
10° to 350°F.

Slurry service.

TFE liner available for corrosive service.

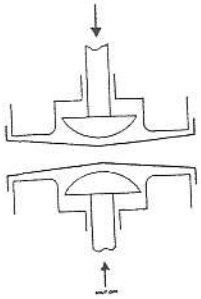


Figure 94.

Sleeve

1 to 12"
to 600 lb.

Gas regulating service.

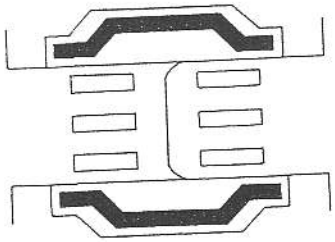


Figure 95.

JACKETED VALVES

Steam Jacketed

For viscous fluids and high melting point materials.

Built in all sizes and valve types, 150 to 300 lb.
Jacket design may require over size end connections or studs.
Consult jacket fabricator.
Most jackets have a pressure rating of 150 psig saturated steam.

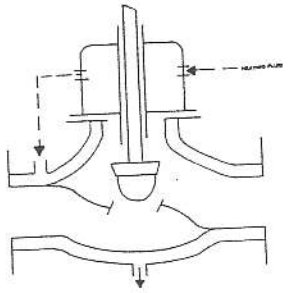


Figure 96.

Vacuum Jacketed

Thermal barrier for cryogenic service.

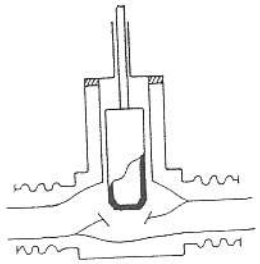


Figure 97.

SECTON XIII

BODY MATERIALS

One can use Figure 98 to select the required body material for a valve based on the pressure-temperature rating requirements and corrosion and erosion resistance. Readily available materials are listed in Figure 98 for each type of valve and size. Consider also the relative material cost, fire safety, and shock impact resistance against breakage. Consider special industry codes and process requirements. Non-metallic seating and linings commonly available for each valve type are also listed.

(√) – AVAILABLE IN ALL SIZES. (–) – NOT AVAILABLE OR LONG PROCUREMENT DELAY.

Material	General Use	Globe	Butterfly[4]	Solid Ball	Plug	Diaphragm
Standard Metals Cast Iron	1) General Service 2) Mild Corrosion	√	√	to 12"	√	√
Ductile Iron	1) to 650°F. 2) to 1000 psi max. 3) Cost Reduction	√	√	to 12"	√	√
Bronze and Aluminum Bronze	1) Mild Service 2) Cryogenic 3) Oxygen	√	Vanes to 12"	to 4"	to 2"	√
Aluminum	1) Light Weight 2) Cryogenic	–	–	to 4"	to 2"	√
Steel	General Service	√	√	√	√	√
Alloy Steel	High Temperature 600 to 1050°F.	√	–	–	–	–
Stainless Steel	1) Cryogenic 2) 1050 to 1200°F. 3) Corrosive Service	√	to 12"	to 16"	to 12"	√
Linings Elastomer	1) Erosive 2) Tight Shutoff	–	√	– [2,3]	√ [1]	√ [1]
Plastic[2]	Corrosive	–	–	– [2,3]	√ [2]	√
TFE	Severely Corrosive	1 & 2"	4 to 12"	2 to 8"	2 to 12"	√ (with TFE faced diaphragm)
Glass		–	–	–	–	√
Corrosion Resistant Body Alloys						
Inconel[5]	Chemical Service	1 to 4"	–	–	–	–
Monel[5]	Chemical Service	1 to 4"	–	–	–	–
Hastelloys[5]	Chemical Service	1 to 4"	–	–	–	–
Alloy 20[5]	Chemical Service	1 to 4"	–	–	–	–
Titanium[5]	Chemical Service	1 to 4"	–	–	–	–
Glass[3]	Chemical Service	–	–	to 4"	–	–

[1] Special split body plug valves w/anti-corrosive metal linings are also available.
[2] Solid PVC valves available in plug, ball & diaphragm types. Fiberglass plastic valves in ball & plug types.
[3] Some TFE faced ball valves are available.
[4] Made from any castable alloy or material available in plate form.
[5] Usually lined iron bodies are preferred because of lower cost.

Figure 98. AVAILABLE BODY MATERIALS AND LININGS.

Refer to Chapter 10 for temperature ranges and other specific limitations.

APPENDIX

Numerous tables, charts, graphs, and standards are presented in the Appendix. It was impractical to include every item of information that an engineer may need; however, the attempt was made to select the most commonly referred-to data that would be required in applying the contents of the foregoing Chapters.

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CONTROL VALVES IN NUCLEAR POWER*

M. L. Freeman
AMETEK, Inc.

Introduction

Control valves are incorporated into nuclear auxiliary systems with design and materials specifications established by Section III of the *ASME Boiler & Pressure Vessel Code*.

Design and material criteria established by Code Publication or Addenda become mandatory on all valve orders accepted by ASME certified nuclear valve manufacturers (holders of the "N" stamp for new valves or the "NPT" stamp for valve parts) six months after they are published. Any valves or valve parts made in accordance with Code cases are stamped with the Code case number.

Control Valve Applications

Control valves for nuclear auxiliary services usually are found in sizes from ¾ to 4 in. Pressure ratings range from ANSI Class-150 to ANSI Class-1500 for nuclear Classes 1, 2, and 3. The three nuclear classes essentially reflect different degrees of material quality assurance. Class 1 is the toughest, meaning that materials and valves built to these standards often can be subjected to higher stress levels than the two lower classes. Few control valves, other than sampling valves and valves in special process equipment, require Class 1 certification. The numbers of Class 2 and 3 valves used in nuclear power plant systems are about equal.

Large control valves, in 6 to 12-in. sizes generally are used on heat exchangers and coolant spray headers. But there also are a small number of ANSI Class-600 steam-dump valves in 6 to 10-in. sizes.

Globe valves are specified for nearly all principal and auxiliary control valve services. Butterfly valves are used to control air pressure in the containment shell and to regulate liquid flow in low pressure systems. In large sizes, up to 36 in., they serve in safety systems—such as those that regulate containment air recirculation. For these applications, butterfly valves have emergency closure features that are designed for use in the unlikely event of a loss-of-coolant accident (LOCA).

A loss of coolant accident is considered to be the most severe credible accident that could occur in nuclear power plants. Essentially, it is a double-ended rupture of a pipe that circulates cooling water to the reactor. An accident of this type would cause a large quantity of water to escape from the primary coolant system and flash into steam thus filling the containment vessel. This would pose severe environmental problems for valves within the vessel.

Design Criteria

Nuclear control-valve design criteria are similar to those used for valves in fossil-fired systems. Some notable differences: (1) Operating pressures and temperatures are higher in fossil applications, but the number of valve operating cycles is greater in nuclear systems over the life of the plant. (2) Higher reliability requirements, containment of radioactive fluids, and the radioactive environment in nuclear plants also cause some design variation. To illustrate the second point: Valves for on/off service normally are supplied with back seats so that, in the open position, radioactive fluids cannot leak through the packing to the atmosphere.

In addition, modulating valves usually have two sets of stem packing, with a leakoff between them which feeds into a low pressure waste gas and/or liquid collection system. Three-way valves have leakoffs and double-packing because there is no way to back-seat these valves. Bellows seal valves can be specified where fluid radioactivity is high or where a leakoff collection system is not practical. Bellows differential pressure requirements range from very low to as high as 600 psi in a few instances.

Materials Considerations

Several materials used for commercial valves are not Code approved for nuclear-valve construction. Reasons: (1) The material may not be completely stable in its service environment. (2) Strength may deteriorate rapidly beyond certain limits that might be exceeded in case of an operating accident. (3) No adequate inspection method can insure the material's reliability.

Further, some conventional valve materials are listed in the Code only in the wrought form. Castings cannot be used, because material properties of the castings and/or methods of inspection may be inadequate, or the probability of pouring a defective casting in these materials is high. ASME's rigid quality assurance inspections, along with acceptable materials and material preparations for nuclear service are given in Section III of the codes.

Detailed inspections are required on valves larger than 4 in., less severe inspection on 2 to 4 in. sizes, and no inspection requirements for valves 2 in. and smaller other than those inspection procedures that are followed in good commercial valve-manufacturing practice.

The most severe inspection criteria are for Class 1 valves to insure the highest degree of reliability. Paradoxically, the designer may sometimes find that the material thickness calculated to meet the lower allowable stress levels for Class 2 requirements may actually give a heavier part than a Class 1 valve with the same pressure and temperature rating. Reason: For Class 1 use, the material has a higher allowable stress because of the more stringent inspection requirements.

Body and Bonnet Materials

Body and bonnet materials for nuclear and commercial valves are generally the same, except that the ASME material designation "S" replaces the ASTM "A" prefix in material identity.

Bolt materials do not differ except for the more-stringent inspection requirements.

Trim Materials

The valve plug is classified in the Code as a pressure-retaining part, and must be made of Code approved materials— that is, except for valves 2 in. and smaller. Small plugs can be made from solid Stellite and the 400-series hardenable stainless steels, in accordance with good commercial practice. Larger plugs may have a hard-faced inlay on the seat joint and/or guiding surface of any Code acceptable base material. In valves larger than 2-in. nominal size, the 400 series hardenable and the precipitation hardened stainless steels may not be used.

Code Case 1621 provides extensive rules for both internal and external parts of valves, which are directly related to valve performance. Valve components are divided into eight categories, depending on importance of function and area groupings. Special rules apply for each category with regard to materials, design and quality assurance.

Stem Packings and Seals

Teflon packings are not suitable for radioactive service, and are replaced with nonlubricated graphite or graphite/asbestos type. The sealing element for soft seat construction is generally Nordel® rubber, which shows less deterioration from radioactive conditions than many other elastomer materials. For steam dump valves, which operate infrequently, some manufacturers use a close-fitting, long-bushing type of leakoff gland, instead of stem packing, to reduce break away friction.

A radioactive environment affects all elastomer and plastic components to varying degrees. Therefore, it is the responsibility of the user to change these parts during regular maintenance periods, as determined by the concentration of radioactivity at the valve location and the total time of exposure. Replacements are made on the basis that enough operating life must always be present to sustain the high amount of radioactivity that would be released in case of a LOCA.

Repair welds in bodies and bonnets for valves 2 in. and smaller are examined in accordance with commercial material specifications, valves over 2 in. are checked by magnetic particle or liquid penetrant, as indicated in Section III. Excessive depth of repair in a pressure-retaining part for valves larger than 2 in. requires radiography and documentation of the repair. Weld-repair restrictions, radiography, and the documentation involved with the use of castings has influenced some manufacturers to offer forged valve products in sizes through 4 in.

Engineering Requirements

Nuclear valves, like those used in fossil-fired stations, are designed to operate safely and meet performance specifications under both normal and abnormal service conditions. Nuclear design criteria, however, usually are more stringent. For example, seismic forces are normally specified at a higher acceleration or "g" level (3-g horizontal, 2-g vertical), and stress calculations often dictate heavier bolts and bonnet flange depending on the size of the actuator and its extension.

With respect to seismic design criteria, present practice of most architect/engineer firms is to permit, in the valve specifications a static analysis to prove that the resonant frequency of the actuator is above 33 Hz. If it is lower, a dynamic analysis or vibration test must be conducted because of the high stress imposed by higher amplitudes of vibration. Since dynamic analysis requires an extensive computer program and is costly, most pneumatic diaphragm actuators are designed to have a resonant frequency above 33 Hz.

Vibration shaker-table tests, when used, usually are conducted on the actuator with auxiliary components, to simulate the worst-case condition. The valve body can be included in the test, at added expense (because of the greater weight), but it is so rigid that little useful information is obtained.

Auxiliary components generally are given

* Reprinted in part from the article "Designing Nuclear Control Valves," with permission from *Power*, Oct. 1974, copyright McGraw-Hill, Inc., Oct. 1974.

a vibration test in the x, y and z planes by their fabricators, and are certified to the valve manufacturer as meeting ASME requirements. Often, during a test of the complete actuator and auxiliary components, both air and electrical signals are imposed to check operations of the combined system. Some specifications even require the valve to stroke with the actuator displaced to its maximum amplitude. Do not confuse Seismic Category I, II, or III with Nuclear Class 1,2 or 3. The Atomic Energy Commission's Regulatory Guides specify the seismic category depending upon location of equipment and safety requirement.

Allowable Leakage

The allowable leakage through the plug-to-seat joint often is specified by the architect/engineer at a very low rate: hence it may involve the application of a higher than normal plug seating force to insure valve tightness. Calculations must be made to insure that the large actuator does not generate excessive seismic stresses in the bolts or bonnet flange. The actuator can be braced to adjacent piping if necessary.

Also, specifications often indicate the amount of leakage allowed from the back seat, and the customer may even ask for leakage rates from the stuffing box as a function of packing tightness and valve hysteresis. Figure 1 shows how much leakage can be expected from a typical stuffing box for graphite/asbestos packing in nuclear service. TFE packing is not used because radiation causes it to deteriorate. The chart shows that TFE has a much lower coefficient of friction against the stem, and seals more tightly for a given hysteresis. For an on/off valve, hysteresis is not critical and leakage can be reduced by increasing packing tightness.

Special forms of all-graphite ribbon packing are being used successfully to achieve low leakage rates and minimum hysteresis. Most types of ribbon packing have a high initial hysteresis (3-4 psi) and require about a 500-stroke break-in period (at operating pressure) to achieve 0.25-psi hysteresis, the preferred maximum for good throttling service.

Loss-of-Coolant Accident

Calculations show that under LOCA conditions containment pressure can reach 50 psia in as little as 10 sec. with containment temperature reaching 300°F. Emergency cooling systems are designed to lower this temperature below about 200°F within 30 min; the remaining heat decays at a much slower rate.

Valve actuators and auxiliary equipment can be affected adversely by this sudden change in the ambient environment if they are not properly designed. A typical diaphragm actuator illustrates this point (Figure 2). The 50-psia ambient pressure enters the actuator case through the spring-barrel vent. If the valve is energized with a lower control-air pressure on the diaphragm, the buildup to 50 psia will overcome this opposing force and aid the spring in its action to stroke the valve. (The control-air signal normally varies from 15 to 45 psia.) In the case of a reverse actuator this overpowering action would close the valve while for the direct actuator it would open the valve.

A pressure differential across the diaphragm in the wrong direction tends to pull the diaphragm from under the stem hold-down washer. A larger plate washer normally would be required to insure continued

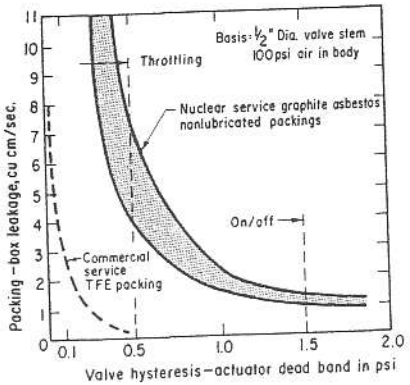
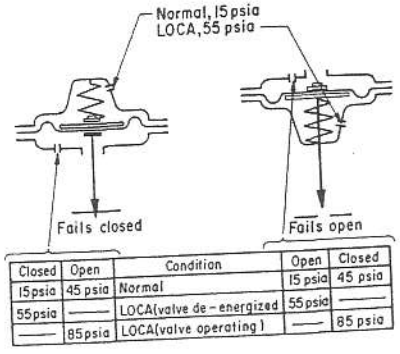


Figure 1. EXPECTED PACKING BOX LEAKAGE.



Closed	Open	Condition	Open	Closed
15psia	45 psia	Normal	15 psia	45 psia
55psia	——	LOCA(valve de-energized	55psia	——
——	85psia	LOCA(valve operating)	——	85 psia

Figure 2. TYPICAL DIAPHRAGM ACTUATORS.

operation without leakage. Most valves, however, would be de-energized at the onset of a LOCA by their three-way solenoids, and thus the postulated diaphragm movement would not occur. Shifting the solenoid to the vent position by de-energizing its coil allows air to leave the diaphragm case faster than ambient pressure builds upon the spring-barrel side because there is a volume differential.

But realize that venting is only partial because the LOCA ambient pressure soon enters the solenoid vent and contacts the diaphragm case where it balances pressure across the diaphragm. Valve-stem position then is controlled by the spring, and the valve fails closed in the case of a reverse actuator and fails open in the case of a direct actuator. There may be a transient pressure differential across the diaphragm before final equalization because of volume difference, solenoid capacity and/or vent-hold size, etc. This difference is not greater than the spring force and the valve strokes in the correct direction.

The LOCA peak temperature is short-lived and thermal deterioration of elastomer materials is not a very important factor. Pressure and temperature tests conducted at maximum LOCA conditions for 30 hours shows that Neoprene diaphragm material with a rayon insert will withstand the thermal cycle and operate the valve properly.

Valve position indication switches have presented some problems with respect to electrical shorts in the high-moisture LOCA atmosphere. Temperature has had some effect on plastic switch mountings. These problems have been resolved with improved equipment.

Valve Design Specification

The valve-design specification, which is prepared and signed by a registered professional engineer, indicates service conditions and ambient environment for each control valve. Unusual operating conditions and performance during a LOCA are often specified, as are auxiliary equipment functions and the timing of valve operation.

For Class 1 valves the number of operating cycles, pressure and temperature peaks, and the rate of change of these variables over the life of the valve must be provided by the purchaser. Thus data permits the manufacturer to calculate fatigue stresses that must be included in the detailed design report that is required by the Code for valves 4 in. and larger. Specification also describes the Code-acceptable materials desired for the pressure-retaining parts—bonnet, body, bolts and valve plug (in sizes above 2 in.).

Manufacturing and Quality Assurance

After a valve manufacturer accepts an order he submits his procedures for quality assurance, inspection, cleaning and testing which were prepared in accordance with the specification. They must be approved and/or negotiated to resolve differences of opinion. Once the procedures are approved by the customer, an inspector from an ASME authorized inspection agency reviews the contract to insure conformance to Section III and assigns inspection hold points.

Next, the manufacturer's quality assurance staff reviews materials purchase orders and clears them for placement. Note that orders have ASME material designations when placed with a vendor having an NPT stamp. The vendor is responsible for conforming to the Code and for furnishing the required material identity and physical and chemical test data. If the order is placed with a vendor who does not have an NPT stamp, then these requirements must be included in the purchase order and the vendor's facility and material control procedures must be approved by the valve manufacturer after a physical inspection of the vendor's plant.

When all valve parts have been completed they are laid out for assembly and inspected. Parts then are cleaned and assembled in a clean room. The lead seal commonly used in commercial service to seal the seat ring to the body cannot be used for nuclear valves, because of the long radio active half-life of the heavy-metal constituents of this viscous seal. Silicone sealants cannot be used because they decompose in radioactive environments and lose their sealing capacity. A suspension of graphite in isopropyl alcohol has been used effectively, but in the newer valve designs, the cage holds down the seat and seat-to-body seal with a spiral-wound gasket creating a tight joint without sealant.

Standard Valve Test

The manufacturer often conducts his own test and inspection of the assembled valve in the presence of the customer's inspector. This usually involves a body hydro test, tests for seat leakage, packing tightness, hysteresis, stroking time, and a review of the specifications with respect to valve's required performance.

Performance testing sometimes is required on certain types of valves. Packing and back-seat leakage are also often checked. Bear in mind that it is practical to check packing leakage volumetrically on production valves only if they have leakoff connections. Backseat leakage is so small that it must be checked by the soap-bubble detection method around the valve stem. If no bubbles are visible, leakage is less than 1×10^{-4} ml/sec.

CONTROL VALVES

NO	BY	DATE	REVISION

SHEET OF

SPEC. NO | REV.

DATE | CONTRACT

REQ. P.O.

BY | CHK'D | APPR.

GENERAL	1	Tag No.				
	2	Service				
	3	Line No./Vessel No.				
	4	Line Size/Sched. No.				
BODY	5	Type of Body				
	6	Body Size / Port Size				
	7	Guiding / No. of Ports				
	8	End Conn. & Rating				
	9	Body Material				
	10	Packing Material				
	11	Lubricator / Iso. Valve				
	12	Bonnet Type				
	13	Trim Form				
	14	Trim Material / Seat/Plug / Shaft Mtl				
	15	Required Seat Tightness				
	16	Max. Allow. Sound Level DBA				
ACTUATOR	17	Model No. & Size				
	18	Type of Actuator				
	19	Close at / Open at				
	20	Flow Action to				
	21	Fail Position				
	22	Handwheel & Location				
POSIT.	23	Mfr. & Model No.				
	24	Filt. Reg. / Gages / Bypass				
	25	Input Signal				
	26	Output Signal				
	27	Air Supply Pressure				
TRANSDUCER	28	Make & Model No.				
	29	Input Signal				
	30	Output Signal				
OPTIONS	31					
	32					
	33					
SERVICE	34	FLOW UNITS	LIQUID	STEAM	GAS	
	35	Fluid				
	36	Quant. Max. / C_v				
	37	Quant. Oper. / C_v				
	38	Valve C_v / Valve F_L / ΔP				
	39	Norm. Inlet Press.				
	40	Max. Inlet Press.				
	41	Max. Shut Off ΔP				
	42	Temp. Max. / Operating				
	43	Oper SPGR / Mol. Wt.				
	44	Oper Visc. / % Flash				
	45	% Superheat / % Solids				
	46	Vapor Press. / Crit. Press				
	47	Predicted Sound Level DBA				
	48	Manufacturer				
	49	Model No.				

Notes.

ISA FORM S20-61

CONTROL VALVES

Specification sheet instructions for ISA Form S20.61

4. Can be used for extra identification or in case of duplicate valves indicate quantity.

5. State type of body; Globe, Angles, 3-way, Y-type, Butterfly, Ball, Saunders, Plug Cock, 4-way,

6. State inlet and outlet connection size, internal port size.

7. Type of Guiding — Top, top and bottom, skirt or port, cage.

8. State Type of connection, i.e.,: screwed (NPT), flanges, weld end. State rating and standard such as 300# ANSI.

9. Material: Iron, steel, alloy plastic, bronze, etc.

10. Packing: Teflon, teflon-asbestos, graphite asbestos, durametallic, etc.

11. If required, State Yes.

12. Bonnet: Standard, plain extension, finned extension, bellows seal.

13. State Characteristic:
- L = Linear
- LV = Linear V Port
- EP = Equal Percentage
- EPT = Equal Percentage Turned
- EPB = Equal Percentage Balanced
- Q = Quick Opening

14. Material: State type of stainless steel or alloy, Alloy #6, composition, hardened, other.

15, 16. None

17. Actuator manufacturers model number and size if not included in the model number — this is usually supplied by the vendor after purchase of actuator.

18. Type of actuator diaphragm, cylinder, motor Electro/Hydraulic or other.

19. Actuator Signal Open/Closed could be 3-15 PSIG. 0-90 PSIG, etc. whatever is required to cause full stroke of the valve.

20. Flow action — tends to "open" or "close" valve.

21. Failure position — open, closed, or last position.

22. Handwheel Location — Top or side mounted and on rotary valves such as Butterfly's which end of the shaft.

23. If a positioner is required, state manufacturer. Model number can be supplied by the vendor.

24. Check if required.

25. Positioner Input Signal. State each end 3-15 4-20 ma, etc.

26. Positioner Output signal for each end of input signals, i.e., 0-30 PSIG.

27. Air supply to positioner.

28. If a separate transducer is required (not integral with the positioner), give manufacturer. The supplier can give the model number.

29. Input signal to the transducer.

30. Output signal from the transducer.

31, 32, 33. For options such as travel stops, mounted Solenoid Valves, speed control valves, limit switches, etc. Use a letter code and identify under notes, i.e., LS — Limit Switch, SC — Speed Control.

34. State liquid, steam, gas units GPM, LB/Hr, FT^3/min, etc.

35. Name of fluid and state whether vapor or liquid if not apparent.

36. State maximum quantity required by process and corresponding C_v.

37. State operating quantity required by process and corresponding C_v.

38. The manufacturer shall fill in the valve C_v and F_l (Liquid Pressure) Recovery Factor without reducers or other accessories.

39. Operating inlet pressure and pressure differential with units (psia, psig, inches H_2O or H_g). Note at this point that one might consider how minimum conditions will fit the sizing.

40. Maximum inlet pressure if different from normal.

41. State the maximum pressure drop in shut-off position to determine proper actuator size. This is actual difference in inlet and outlet pressure stated in psi, inches of H_2O or H_g, etc.

42. State °F. or °C.

43. State operating specific gravity and molecular weight.

44. State operating viscosity and its units. State flash at valve outlet, i.e., of max flow that will be flashed to vapor because of the valve pressure drop.

45. In the case of vapors, state superheat and in the cases of liquids, state the solids, if present.

46. Note vapor pressure of fluid as well as the critical pressure.

47. Give manufacturers predicted sound level DBA.

48. Complete when available.

ISA – RP4.1

1950

RECOMMENDED PRACTICE

UNIFORM FACE TO FACE DIMENSIONS FOR FLANGED CONTROL VALVE BODIES

Sponsor

INSTRUMENT SOCIETY of AMERICA
400 Stanwix Street
Pittsburgh, Pennsylvania 15222

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(With Tentative Status Removed)

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FOREWORD

This Recommended Practice has been prepared as a part of the service of the Instrument Society of America toward a goal of uniformity in the field of instrumentation. To be of real value this report should not be static, but should be subjected to periodic review. Toward this end the Society welcomes all comments and criticisms, and asks that they be addressed to Secretary, Standards & Practice Board, Instrument Society of America, 400 Stanwix Street, Pittsburgh, Pennsylvania 15222.

This report was prepared by the Sub-Committee on Control Valve and By-pass Manifolds (RP-4).

C. W. Bates, Chairman 1949-50—Humble Oil & Refining Co.
W. H. Fortney, Chairman 1947-49—Humble Oil & Refining Co.
M. K. Anderson—Carbide & Carbon Chemicals Corp.
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E. D. Mattix—Cities Service Refining Corporation
J. A. Parker—Shell Chemical Corporation
R. R. Proctor—The Pure Oil Company
H. T. Wicker—Standard Oil Company

The assistance of the Fluid Controls Institute in developing this Recommended Practice is gratefully acknowledged.

PURPOSE

This report is presented as a means of giving official endorsement to the control valve face-to-face dimensions shown herein. These dimensions were developed by the majority of control valve manufacturers with the assistance of the National Steam Specialty Club, when it was called to their attention as a result of the committee's initial questionnaire, that the user industry was 100% in agreement that uniform face to face dimensions for control valves were desirable. The adoption of these dimensions by all control valve manufacturers will allow piping design, where control valves are involved, without giving special consideration to the make of equipment to be used.

SCOPE

The sizes shown are those most widely used throughout industry. As indicated in the title only flanged valves are covered. The flange ratings are those most frequently used in industry. Only valves which are in the category of diaphragm actuated control valves of the plug type are included.

UNIFORM FACE TO FACE DIMENSIONS FOR FLANGED CONTROL VALVE BODIES

PIPE SIZE INCHES	125 PSI IRON 150 PSI STEEL SEE NOTE 1	250 PSI IRON 300 PSI STEEL SEE NOTE 2	600 PSI STEEL SEE NOTE 3
½	....	7½	8
¾	....	7⅞	8⅛
1	7¼	7¾	8¼
1½	8¾	9¼	9⅞
2	10	10½	11¼
2½	10⅞	11½	12¼
3	11¾	12½	13¼
4	13⅞	14½	15½
6	17¾	18⅝	20
8	21⅛	22⅜	24

NOTE 1—Cast iron bodies with 125 PSI ASA B16A-1939 flanges, and steel bodies with 150 PSI raised face ASA-B16E-1939 flanges.
NOTE 2—Cast iron bodies with 250 PSI ASA B16B-1944 flanges, and steel bodies with 300 PSI raised face B16E-1939 flanges.
NOTE 3—Steel bodies with 600 PSI raised face ASA B16E-1939 flanges.

AMERICAN PIPE FLANGE DIMENSIONS
Diameter of Bolt Circle - Inches

Nominal Pipe Size	125 lb. Cast Iron	ANSI 150 lb.	250 lb. Cast Iron	ANSI 300 lb.	ANSI 400 lb.	ANSI 600 lb.	ANSI 900 lb.	ANSI 1500 lb.	ANSI 2500 lb.
1	3-1/8	3-1/8	3-1/2	3-1/2	3-1/2	3-1/2	4	4	4-1/4
1-1/4	3-1/2	3-1/2	3-7/8	3-7/8	3-7/8	3-7/8	4-3/8	4-3/8	5-1/8
1-1/2	3-7/8	3-7/8	4-1/2	4-1/2	4-1/2	4-1/2	4-7/8	4-7/8	5-3/4
2	4-3/4	4-3/4	5	5	5	5	6-1/2	6-1/2	6-3/4
2-1/2	5-1/2	5-1/2	5-7/8	5-7/8	5-7/8	5-7/8	7-1/2	7-1/2	7-3/4
3	6	6	6-5/8	6-5/8	6-5/8	6-5/8	7-1/2	8	9
4	7-1/2	7-1/2	7-7/8	7-7/8	7-7/8	8-1/2	9-1/4	9-1/2	10-3/4
5	8-1/2	8-1/2	9-1/4	9-1/4	9-1/4	10-1/2	11	11-1/2	12-3/4
6	9-1/2	9-1/2	10-5/8	10-5/8	10-5/8	11-1/2	12-1/2	12-1/2	14-1/2
8	11-3/4	11-3/4	13	13	13	13-3/4	15-1/2	15-1/2	17-1/4
10	14-1/4	14-1/4	15-1/4	15-1/4	15-1/4	17	18-1/2	19	21-1/4
12	17	17	17-3/4	17-3/4	17-3/4	19-1/4	21	22-1/2	24-3/8
16	21-1/4	21-1/4	22-1/2	22-1/2	22-1/2	23-3/4	24-1/4	27-3/4	-
18	22-3/4	22-3/4	24-3/4	24-3/4	24-3/4	25-3/4	27	30-1/2	-
20	25	25	27	27	27	28-1/2	29-1/2	32-3/4	-
24	29-1/2	29-1/2	32	32	32	33	35-1/2	39	-

AMERICAN PIPE FLANGE DIMENSIONS
Number and Diameter of Stud Bolts - Inches

Nominal Pipe Size	125 lb. Cast Iron		ANSI 150 lb.		250 lb. Cast Iron		ANSI 300 lb.		ANSI 400 lb.		ANSI 600 lb.		ANSI 900 lb.		ANSI 1500 lb.		ANSI 2500 lb.	
	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.
1	4	1/2	4	1/2	4	5/8	4	5/8	4	5/8	4	5/8	4	7/8	4	7/8	4	7/8
1-1/4	4	1/2	4	1/2	4	5/8	4	5/8	4	5/8	4	5/8	4	7/8	4	7/8	4	1
1-1/2	4	1/2	4	1/2	4	3/4	4	3/4	4	3/4	4	3/4	4	1	4	1	4	1-1/8
2	4	5/8	4	5/8	8	5/8	8	5/8	8	5/8	8	5/8	8	7/8	8	7/8	8	1
2-1/2	4	5/8	4	5/8	8	3/4	8	3/4	8	3/4	8	3/4	8	1	8	1	8	1-1/8
3	4	5/8	4	5/8	8	3/4	8	3/4	8	3/4	8	3/4	8	7/8	8	1-1/8	8	1-1/4
4	8	5/8	8	5/8	8	3/4	8	3/4	8	7/8	8	7/8	8	1-1/8	8	1-1/4	8	1-1/2
5	8	3/4	8	3/4	8	3/4	8	3/4	8	7/8	8	1	8	1-1/4	8	1-1/2	8	1-3/4
6	8	3/4	8	3/4	12	3/4	12	3/4	12	7/8	12	1	12	1-1/8	12	1-3/8	8	2
8	8	3/4	8	3/4	12	7/8	12	7/8	12	1	12	1-1/8	12	1-3/8	12	1-5/8	12	2
10	12	7/8	12	7/8	16	1	16	1	16	1-1/8	16	1-1/4	16	1-3/8	12	1-7/8	12	2-1/2
12	12	7/8	12	7/8	16	1-1/8	16	1-1/8	16	1-1/4	20	1-1/4	20	1-3/8	16	2	12	2-3/4
16	16	1	16	1	20	1-1/4	20	1-1/4	20	1-3/8	20	1-1/2	20	1-5/8	16	2-1/2	-	-
18	16	1-1/8	16	1-1/8	24	1-1/4	24	1-1/4	24	1-3/8	20	1-5/8	20	1-7/8	16	2-3/4	-	-
20	20	1-1/8	20	1-1/8	24	1-1/4	24	1-1/4	24	1-1/2	24	1-5/8	20	2	16	3	-	-
24	20	1-1/4	20	1-1/4	24	1-1/2	24	1-1/2	24	1-3/4	24	1-7/8	20	2-1/2	16	3-1/2	-	-

Reprinted from "Control Valve Handbook," 1965, Fisher Controls Company, Marshalltown, Iowa.

AMERICAN PIPE FLANGE DIMENSIONS
Flange Thickness - Inches

Nominal Pipe Size	125 lb. Cast Iron	ANSI 150 lb.		250 lb. Cast Iron	ANSI 300 lb.		ANSI 400 lb.		ANSI 600 lb.		ANSI 900 lb.		ANSI 1500 lb.		ANSI 2500 lb.	
	FF*	RF	RTJ	RF	RF	RTJ	RF	RTJ	RF	RTJ	RF	RTJ	RF	RTJ	RF	RTJ
1	7/16	7/16	11/16	11/16	11/16	15/16	15/16	15/16	15/16	15/16	1-3/8	1-3/8	1-3/8	1-3/8	1-5/8	1-5/8
1-1/4	1/2	1/2	3/4	3/4	3/4	1	1-1/16	1-1/16	1-1/16	1-1/16	1-3/8	1-3/8	1-3/8	1-3/8	1-3/4	1-13/16
1-1/2	9/16	9/16	13/16	13/16	13/16	1-1/16	1-1/8	1-1/8	1-1/8	1-1/8	1-1/2	1-1/2	1-1/2	1-1/2	2	2-1/16
2	5/8	5/8	7/8	7/8	7/8	1-3/16	1-1/4	1-5/16	1-1/4	1-5/16	1-3/4	1-13/16	1-3/4	1-13/16	2-1/4	2-5/16
2-1/2	11/16	11/16	15/16	1	1	1-5/16	1-3/8	1-7/16	1-3/8	1-7/16	1-7/8	1-15/16	1-7/8	1-15/16	2-1/2	2-5/8
3	3/4	3/4	1	1-1/8	1-1/8	1-7/16	1-1/2	1-9/16	1-1/2	1-9/16	1-3/4	1-13/16	2-1/8	2-3/16	2-7/8	3
4	15/16	15/16	1-3/16	1-1/4	1-1/4	1-9/16	1-5/8	1-11/16	1-3/4	1-13/16	2	2-1/16	2-3/8	2-7/16	3-1/4	3-7/16
5	15/16	15/16	1-3/16	1-3/8	1-3/8	1-11/16	1-3/4	1-13/16	2	2-1/16	2-1/4	2-5/16	3-1/8	3-3/16	3-7/8	4-1/8
6	1	1	1-1/4	1-7/16	1-7/16	1-3/4	1-7/8	1-15/16	2-1/8	2-3/16	2-7/16	2-1/2	3-1/2	3-5/8	4-1/2	4-3/4
8	1-1/8	1-1/8	1-3/8	1-5/8	1-5/8	1-15/16	2-1/8	2-3/16	2-7/16	2-1/2	2-3/4	2-13/16	3-7/8	4-1/16	5-1/4	5-9/16
10	1-3/16	1-3/16	1-7/16	1-7/8	1-7/8	2-3/16	2-3/8	2-7/16	2-3/4	2-13/16	3	3-1/16	4-1/2	4-11/16	6-3/4	7-3/16
12	1-1/4	1-1/4	1-1/2	2	2	2-5/16	2-1/2	2-9/16	2-7/8	2-15/16	3-3/8	3-7/16	5-1/8	5-7/16	7-1/2	7-15/16
16	1-7/16	1-7/16	1-11/16	2-1/4	2-1/4	2-9/16	2-3/4	2-13/16	3-1/4	3-5/16	3-3/4	3-15/16	6	6-7/16	-	-
18	1-9/16	1-9/16	1-13/16	2-3/8	2-3/8	2-11/16	2-7/8	2-15/16	3-1/2	3-9/16	4-1/4	4-1/2	6-5/8	7-1/16	-	-
20	1-11/16	1-11/16	1-15/16	2-1/2	2-1/2	2-7/8	3	3-1/8	3-3/4	3-7/8	4-1/2	4-3/4	7-1/4	7-11/16	-	-
24	1-7/8	1-7/8	2-1/8	2-3/4	2-3/4	3-3/16	3-1/4	3-7/16	4-1/4	4-7/16	5-3/4	6-1/8	8-1/4	8-13/16	-	-

Abbreviations Used: FF = Flat Face; RF = Raised Face; RTJ = Ring Type Joint.

AMERICAN PIPE FLANGE DIMENSIONS
Flange Diameter - Inches

Nominal Pipe Size	125 lb. Cast Iron	ANSI 150 lb.	250 lb. Cast Iron	ANSI 300 lb.	ANSI 400 lb.	ANSI 600 lb.	ANSI 900 lb.	ANSI 1500 lb.	ANSI 2500 lb.
1	4-1/4	4-1/4	4-7/8	4-7/8	4-7/8	4-7/8	5-7/8	5-7/8	6-1/4
1-1/4	4-5/8	4-5/8	5-1/4	5-1/4	5-1/4	5-1/4	6-1/4	6-1/4	7-1/4
1-1/2	5	5	6-1/8	6-1/8	6-1/8	6-1/8	7	7	8
2	6	6	6-1/2	6-1/2	6-1/2	6-1/2	8-1/2	8-1/2	9-1/4
2-1/2	7	7	7-1/2	7-1/2	7-1/2	7-1/2	9-5/8	9-5/8	10-1/2
3	7-1/2	7-1/2	8-1/4	8-1/4	8-1/4	8-1/4	9-1/2	10-1/2	12
4	9	9	10	10	10	10-3/4	11-1/2	12-1/4	14
5	10	10	11	11	11	13	13-3/4	14-3/4	16-1/2
6	11	11	12-1/2	12-1/2	12-1/2	14	15	15-1/2	19
8	13-1/2	13-1/2	15	15	15	16-1/2	18-1/2	19	21-3/4
10	16	16	17-1/2	17-1/2	17-1/2	20	21-1/2	23	26-1/2
12	19	19	20-1/2	20-1/2	20-1/2	22	24	26-1/2	30
16	23-1/2	23-1/2	25-1/2	25-1/2	25-1/2	27	27-3/4	32-1/2	-
18	25	25	28	28	28	29-1/4	31	36	-
20	27-1/2	27-1/2	30-1/2	30-1/2	30-1/2	32	33-3/4	38-3/4	-
24	32	32	36	36	36	37	41	46	-

Reprinted from "Control Valve Handbook," 1965, Fisher Controls Company, Marshalltown, Iowa.

Commercial Wrought Steel Pipe Data

Schedule Wall Thickness—Per ASA B36.10-1950*

Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D					a	A
Inches	Inches	Inches	Inches	Feet	d^2	d^3	d^4	d^5	Sq. In.	Sq. Ft.
Schedule 10	14	14	0.210	13.58	1.132	184.42	2504.4	34010.	461856.	144.75	1.005
	16	16	0.250	15.5	1.291	240.25	3723.9	57720.	894660.	188.69	1.310
	18	18	0.250	17.5	1.4583	306.25	5359.4	93789.	1641309.	240.53	1.670
	20	20	0.250	19.5	1.625	380.25	7414.9	144590.	2819500.	298.65	2.074
	24	24	0.250	23.5	1.958	552.25	12977.	304980.	7167030.	433.74	3.012
	30	30	0.312	29.376	2.448	862.95	25350.	744288.	21864218.	677.76	4.707
Schedule 20	8	8.625	0.250	8.125	0.6771	66.02	536.38	4359.3	35409.	51.85	0.3601
	10	10.75	0.250	10.25	0.8542	105.06	1076.9	11038.	113141.	82.52	0.5731
	12	12.75	0.250	12.25	1.021	150.06	1838.3	22518.	275855.	117.86	0.8185
	14	14	0.312	13.376	1.111	178.92	2393.2	32012.	428185.	140.52	0.9758
	16	16	0.312	15.376	1.281	236.42	3635.2	55894.	859442.	185.69	1.290
	18	18	0.312	17.376	1.448	301.92	5246.3	91156.	1583978.	237.13	1.647
	20	20	0.375	19.250	1.604	370.56	7133.3	137317.	2643352.	291.04	2.021
	24	24	0.375	23.25	1.937	540.56	12568.	292205.	6793832.	424.56	2.948
	30	30	0.500	29.00	2.417	841.0	24389.	707281.	20511149.	660.52	4.587
Schedule 30	8	8.625	0.277	8.071	0.6726	65.14	525.75	4243.2	34248.	51.16	0.3553
	10	10.75	0.307	10.136	0.8447	102.74	1041.4	10555.	106987.	80.69	0.5603
	12	12.75	0.330	12.09	1.0075	146.17	1767.2	21366.	258304.	114.80	0.7972
	14	14	0.375	13.25	1.1042	175.56	2326.2	30821.	408394.	137.88	0.9575
	16	16	0.375	15.25	1.2708	232.56	3546.6	54084.	824801.	182.65	1.268
	18	18	0.438	17.124	1.4270	293.23	5021.3	85984.	1472397.	230.30	1.599
	20	20	0.500	19.00	1.5833	361.00	6859.0	130321.	2476099.	283.53	1.969
	24	24	0.562	22.876	1.9063	523.31	11971.	273853.	6264703.	411.00	2.854
	30	30	0.625	28.75	2.3958	826.56	23764.	683201.	19642160.	649.18	4.508
Schedule 40	⅛	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.005242	0.00141	0.057	0.00040
	¼	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
	⅜	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
	½	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.09310	0.304	0.00211
	¾	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
	1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
	1¼	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
	1½	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
	2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
	2½	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
	3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
	3½	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
	4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
	5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
	6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
	8	8.625	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
	10	10.75	0.365	10.02	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
	12	12.75	0.406	11.938	0.9965	142.5	1701.3	20306.	242470.	111.93	0.7773
	14	14.0	0.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394
	16	16.0	0.500	15.000	1.250	225.0	3375.0	50625.	759375.	176.72	1.2272
	18	18.0	0.562	16.876	1.4063	284.8	4806.3	81111.	1368820.	223.68	1.5533
	20	20.0	0.593	18.814	1.5678	354.0	6659.5	125320.	2357244.	278.00	1.9305
	24	24.0	0.687	22.626	1.8855	511.9	11583.	262040.	5929784.	402.07	2.7921
Schedule 60	8	8.625	0.406	7.813	0.6511	61.04	476.93	3725.9	29113.	47.94	0.3329
	10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.66	0.5185
	12	12.75	0.562	11.626	0.9688	135.16	1571.4	18268.	212399.	106.16	0.7372
	14	14.0	0.593	12.814	1.0678	164.20	2104.0	26962.	345480.	128.96	0.8956
	16	16.0	0.656	14.688	1.2240	215.74	3168.8	46544.	683618.	169.44	1.1766
	18	18.0	0.750	16.500	1.3750	272.25	4492.1	74120.	1222982.	213.83	1.4849
	20	20.0	0.812	18.376	1.5313	337.68	6205.2	114028.	2095342.	265.21	1.8417
	24	24.0	0.968	22.064	1.8387	486.82	10741.	236994.	5229036.	382.35	2.6552
Schedule 80	⅛	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
	¼	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
	⅜	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03200	0.01354	0.141	0.00098
	½	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
	¾	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
	1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
	1¼	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.00891

(continued on the next page)

Courtesy Crane Co., Reproduced from Technical Paper No. 410, "Flow of Fluids."

*The latest revision of this standard is ANSI B36.10-1970; therefore, the schedule wall thickness used to derive the given data may vary slightly from the current standard in some cases.

Commercial Wrought Steel Pipe Data

Schedule Wall Thickness—Per ASA B36.10-1950 *

Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D					a	A
Inches	Inches	Inches	Inches	Feet	d^2	d^3	d^4	d^5	Sq. In.	Sq. Ft.
Schedule 80—cont.	1½	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
	2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
	2½	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
	3	3.5	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
	3½	4.0	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
	4	4.5	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
	5	5.563	0.375	4.813	0.4011	23.16	111.49	536.38	2583.	18.194	0.1263
	6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
	8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
	10	10.75	0.593	9.564	0.7970	91.47	874.82	8366.8	80020.	71.84	0.4989
	12	12.75	0.687	11.376	0.9480	129.41	1472.2	16747.	190523.	101.64	0.7058
	14	14.0	0.750	12.500	1.0417	156.25	1953.1	24414.	305176.	122.72	0.8522
	16	16.0	0.843	14.314	1.1928	204.89	2932.8	41980.	600904.	160.92	1.1175
	18	18.0	0.937	16.126	1.3438	260.05	4193.5	67626.	1090518.	204.24	1.4183
	20	20.0	1.031	17.938	1.4948	321.77	5771.9	103536.	1857248.	252.72	1.7550
	24	24.0	1.218	21.564	1.7970	465.01	10027.	216234.	4662798.	365.22	2.5362
Schedule 100	8	8.625	0.593	7.439	0.6199	55.34	411.66	3062.	22781.	43.46	0.3018
	10	10.75	0.718	9.314	0.7762	86.75	807.99	7526.	69357.	68.13	0.4732
	12	12.75	0.843	11.064	0.9220	122.41	1354.4	14985.	165791.	96.14	0.6677
	14	14.0	0.937	12.126	1.0105	147.04	1783.0	21621.	262173.	115.49	0.8020
	16	16.0	1.031	13.938	1.1615	194.27	2707.7	37740.	526020.	152.58	1.0596
	18	18.0	1.156	15.688	1.3057	246.11	3861.0	60572.	950250.	193.30	1.3423
	20	20.0	1.281	17.438	1.4532	304.08	5302.6	92467.	1612438.	238.83	1.6585
	24	24.0	1.531	20.938	1.7448	438.40	9179.2	192195.	4024179.	344.32	2.3911
Schedule 120	4	4.50	0.438	3.624	0.302	13.133	47.595	172.49	625.1	10.315	0.07163
	5	5.563	0.500	4.563	0.3802	20.82	95.006	433.5	1978.	16.35	0.1136
	6	6.625	0.562	5.501	0.4584	30.26	166.47	915.7	5037.	23.77	0.1650
	8	8.625	0.718	7.189	0.5991	51.68	371.54	2671.	19202.	40.59	0.2819
	10	10.75	0.843	9.064	0.7553	82.16	744.66	6750.	61179.	64.53	0.4481
	12	12.75	1.000	10.750	0.8959	115.56	1242.3	13355.	143563.	90.76	0.6303
	14	14.0	1.093	11.814	0.9845	139.57	1648.9	19480.	230137.	109.62	0.7612
	16	16.0	1.218	13.564	1.1303	183.98	2495.5	33849.	459133.	144.50	1.0035
	18	18.0	1.375	15.250	1.2708	232.56	3546.6	54086.	824804.	182.66	1.2684
	20	20.0	1.500	17.000	1.4166	289.00	4913.0	83521.	1419857.	226.98	1.5762
	24	24.0	1.812	20.376	1.6980	415.18	8459.7	172375.	3512313.	326.08	2.2645
Schedule 140	8	8.625	0.812	7.001	0.5834	49.01	343.15	2402.	16819.	38.50	0.2673
	10	10.75	1.000	8.750	0.7292	76.56	669.92	5862.	51291.	60.13	0.4176
	12	12.75	1.125	10.500	0.8750	110.25	1157.6	12155.	127628.	86.59	0.6013
	14	14.0	1.250	11.500	0.9583	132.25	1520.9	17490.	201136.	103.87	0.7213
	16	16.0	1.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394
	18	18.0	1.562	14.876	1.2396	221.30	3292.0	48972.	728502.	173.80	1.2070
	20	20.0	1.750	16.5	1.3750	272.25	4492.1	74120.	1222981.	213.82	1.4849
	24	24.0	2.062	19.876	1.6563	395.06	7852.1	156069.	3102022.	310.28	2.1547
Schedule 160	½	0.840	0.187	0.466	0.0388	0.2172	0.1012	0.04716	0.02197	0.1706	0.00118
	¾	1.050	0.218	0.614	0.0512	0.3770	0.2315	0.1421	0.08726	0.2961	0.00206
	1	1.315	0.250	0.815	0.0679	0.6642	0.5413	0.4412	0.3596	0.5217	0.00362
	1¼	1.660	0.250	1.160	0.0966	1.346	1.561	1.811	2.100	1.057	0.00734
	1½	1.900	0.281	1.338	0.1115	1.790	2.395	3.205	4.288	1.406	0.00976
	2	2.375	0.343	1.689	0.1407	2.853	4.818	8.138	13.74	2.241	0.01556
	2½	2.875	0.375	2.125	0.1771	4.516	9.596	20.39	43.33	3.546	0.02463
	3	3.50	0.438	2.624	0.2187	6.885	18.067	47.41	124.4	5.408	0.03755
	4	4.50	0.531	3.438	0.2865	11.82	40.637	139.7	480.3	9.283	0.06447
	5	5.563	0.625	4.313	0.3594	18.60	80.230	346.0	1492.	14.61	0.1015
	6	6.625	0.718	5.189	0.4324	26.93	139.72	725.0	3762.	21.15	0.1469
	8	8.625	0.906	6.813	0.5677	46.42	316.24	2155.	14679.	36.46	0.2532
	10	10.75	1.125	8.500	0.7083	72.25	614.12	5220.	44371.	56.75	0.3941
	12	12.75	1.312	10.126	0.8438	102.54	1038.3	10514.	106461.	80.53	0.5592
	14	14.0	1.406	11.188	0.9323	125.17	1400.4	15668.	175292.	98.31	0.6827
	16	16.0	1.593	12.814	1.0678	164.20	2104.0	26961.	345482.	128.96	0.8956
	18	18.0	1.781	14.438	1.2032	208.45	3009.7	43454.	627387.	163.72	1.1369
	20	20.0	1.968	16.064	1.3387	258.05	4145.3	66590.	1069715.	202.67	1.4074
	24	24.0	2.343	19.314	1.6095	373.03	7204.7	139152.	2687582.	292.98	2.0346

Courtesy Crane Co., Reproduced from Technical Paper No. 410, "Flow of Fluids."

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Commercial Wrought Steel Pipe Data

(Per ASA B36.10-1950)*

Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D	d^2	d^3	d^4	d^5	a	A
Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.
Standard Wall Pipe										
⅛	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.00524	0.00141	0.057	0.00040
¼	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
⅜	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
½	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.0931	0.304	0.00211
¾	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
1¼	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
1½	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
2½	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
3½	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
8	8.625	0.277	8.071	0.6725	65.14	525.75	4243.0	34248.	51.161	0.3553
	8.625S	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
10	10.75	0.279	10.192	0.8493	103.88	1058.7	10789.	109876.	81.585	0.5666
	10.75	0.307	10.136	0.8446	102.74	1041.4	10555.	106987.	80.691	0.5604
	10.75S	0.365	10.020	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
12	12.75	0.330	12.090	1.0075	146.17	1767.2	21366.	258300.	114.80	0.7972
	12.75S	0.375	12.000	1.000	144.0	1728.0	20736.	248800.	113.10	0.7854
Extra Strong Pipe										
⅛	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
¼	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
⅜	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03201	0.01354	0.141	0.00098
½	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
¾	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
1¼	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.00891
1½	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
2½	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
3	3.500	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
3½	4.000	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
4	4.500	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
5	5.563	0.375	4.813	0.4011	23.16	111.49	536.6	2583.	18.194	0.1263
6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.662	0.5185
12	12.75	0.500	11.750	0.9792	138.1	1622.2	19072.	223970.	108.434	0.7528
Double Extra Strong Pipe										
½	0.840	0.294	0.252	0.0210	0.0635	0.0160	0.004032	0.00102	0.050	0.00035
¾	1.050	0.308	0.434	0.0362	0.1884	0.0817	0.03549	0.01540	0.148	0.00103
1	1.315	0.358	0.599	0.0499	0.3588	0.2149	0.1287	0.07711	0.282	0.00196
1¼	1.660	0.382	0.896	0.0747	0.8028	0.7193	0.6445	0.5775	0.630	0.00438
1½	1.900	0.400	1.100	0.0917	1.210	1.331	1.4641	1.611	0.950	0.00660
2	2.375	0.436	1.503	0.1252	2.259	3.395	5.1031	7.670	1.774	0.01232
2½	2.875	0.552	1.771	0.1476	3.136	5.554	9.8345	17.42	2.464	0.01710
3	3.500	0.600	2.300	0.1917	5.290	12.167	27.984	64.36	4.155	0.02885
3½	4.000	0.636	2.728	0.2273	7.442	20.302	55.383	151.1	5.845	0.04059
4	4.500	0.674	3.152	0.2627	9.935	31.315	98.704	311.1	7.803	0.05419
5	5.563	0.750	4.063	0.3386	16.51	67.072	272.58	1107.	12.966	0.09006
6	6.625	0.864	4.897	0.4081	23.98	117.43	575.04	2816.	18.835	0.1308
8	8.625	0.875	6.875	0.5729	47.27	324.95	2234.4	15360.	37.122	0.2578

Courtesy Crane Co., Reproduced from Technical Paper No. 410, "Flow of Fluids."

*The latest revision of this standard is ANSI B36.10-1970; therefore, the schedule wall thickness used to derive the given data may vary slightly from the current standard in some cases.

Stainless Steel Pipe Data

Schedule Wall Thickness — Per ASA B36.19-1957*

Nominal Pipe Size	Outside Diameter	Thickness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D	d^2	d^3	d^4	d^5	a	A
Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.
Schedule 5 S										
½	0.840	0.065	0.710	0.0592	0.504	0.358	0.254	0.1804	0.396	0.00275
¾	1.050	0.065	0.920	0.0767	0.846	0.779	0.716	0.659	0.664	0.00461
1	1.315	0.065	1.185	0.0988	1.404	1.664	1.972	2.337	1.103	0.00766
1¼	1.660	0.065	1.530	0.1275	2.341	3.582	5.480	8.384	1.839	0.01277
1½	1.900	0.065	1.770	0.1475	3.133	5.545	9.815	17.37	2.461	0.01709
2	2.375	0.065	2.245	0.1871	5.040	11.31	25.40	57.03	3.958	0.02749
2½	2.875	0.083	2.709	0.2258	7.339	19.88	53.86	145.9	5.764	0.04003
3	3.500	0.083	3.334	0.2778	11.12	37.06	123.6	411.9	8.733	0.06065
3½	4.000	0.083	3.834	0.3195	14.70	56.36	216.1	828.4	11.545	0.08017
4	4.500	0.083	4.334	0.3612	18.78	81.41	352.8	1529.	14.750	0.1024
5	5.563	0.109	5.345	0.4454	28.57	152.7	816.2	4363.	22.439	0.1558
6	6.625	0.109	6.407	0.5339	41.05	263.0	1685.	10796.	32.241	0.2239
8	8.625	0.109	8.407	0.7006	70.68	594.2	4995.	41996.	55.512	0.3855
10	10.750	0.134	10.482	0.8375	109.9	1152.	12072.	126538.	86.315	0.5994
12	12.750	0.156	12.438	1.0365	154.7	1924.	23933.	297682.	121.50	0.8438
Schedule 10 S										
⅛	0.405	0.049	0.307	0.0256	0.0942	0.0289	0.00888	0.00273	0.074	0.00051
¼	0.540	0.065	0.410	0.0342	0.1681	0.0689	0.02826	0.01159	0.132	0.00092
⅜	0.675	0.065	0.545	0.0454	0.2970	0.1619	0.08822	0.04808	0.233	0.00162
½	0.840	0.083	0.674	0.0562	0.4543	0.3062	0.2064	0.1391	0.357	0.00248
¾	1.050	0.083	0.884	0.0737	0.7815	0.6908	0.6107	0.5398	0.614	0.00426
1	1.315	0.109	1.097	0.0914	1.203	1.320	1.448	1.589	0.945	0.00656
1¼	1.660	0.109	1.442	0.1202	2.079	2.998	4.324	6.235	1.633	0.01134
1½	1.900	0.109	1.682	0.1402	2.829	4.759	8.004	13.46	2.222	0.01543
2	2.375	0.109	2.157	0.1798	4.653	10.04	21.65	46.69	3.654	0.02538
2½	2.875	0.120	2.635	0.2196	6.943	18.30	48.21	127.0	5.453	0.03787
3	3.500	0.120	3.260	0.2717	10.63	34.65	112.9	368.2	8.347	0.05796
3½	4.000	0.120	3.760	0.3133	14.14	53.16	199.9	751.5	11.11	0.07712
4	4.500	0.120	4.260	0.3550	18.15	77.31	329.3	1403.	14.26	0.09899
5	5.563	0.134	5.295	0.4413	28.04	148.5	786.1	4162.	22.02	0.1529
6	6.625	0.134	6.357	0.5298	40.41	256.9	1633.	10382.	31.74	0.2204
8	8.625	0.148	8.329	0.6941	69.37	577.8	4813.	40083.	54.48	0.3784
10	10.750	0.165	10.420	0.8683	108.6	1131.	11789.	122840.	85.29	0.5923
12	12.750	0.180	12.390	1.0325	153.5	1902.	23566.	291982.	120.6	0.8372
Schedule 40 S										
⅛ to 12	Values are the same, size for size, as those shown on the facing page for Standard Wall Pipe (heaviest weight on 8, 10, and 12-inch sizes).									
Schedule 80 S										
⅛ to 12	Values are the same, size for size, as those shown on the facing page for Extra Strong Pipe.									

Courtesy Crane Co., Reproduced from Technical Paper No. 410, "Flow of Fluids."

*The latest revision of this standard is ANSI B36.19-1965; therefore, the schedule wall-thickness used to derive the given data may vary slightly from the current standard in some cases.

DIN STANDARDS*

DIN Pressure-Temperature Ratings Carbon Steel Flanges and Pipes

Nominal Pressure	Maximum Permissible Working Pressure				Hydraulic Test Pressure
	I (W) For Fluids, Gases and Vapours up to 120° C.	II (G) For Fluids, Gases and Vapours up to 300° C.	III (H) For Fluids, Gases and Vapours up to 400° C.		
	Flanges & Pipes	Flanges & Pipes	Flanges	Pipes	
16	16	13	13	10	25
25	25	20	20	16	40
40	40	32	32	25	60
64	64	50	40	40	96
100	100	80	64	64	150
160	160	125	100	100	240
250	250	200	160	160	375
320	320	250	200	200	480
400	400	320	250	250	600

DIN Cast Steel Flange Standard Nenndruck 16

(Nominal Pressure 16 Kgf/cm²)

Nominal Bore	Pipe		Flange			Bolting		
	Outside Dia.	Thickness	Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	22	6	90	16	60	4	M12	14
15	25	6	93	16	65	4	M12	14
20	33	6.5	105	18	75	4	M12	14
25	39	7	115	18	85	4	M12	14
32	46	7	140	16	100	4	M16	18
40	55	7.5	150	16	110	4	M16	18
50	65	7.5	165	18	125	4	M16	18
65	86	8	185	18	145	4	M16	18
80	97	8.5	200	20	160	8	M16	18
100	119	9.5	220	20	180	8	M16	18
125	145	10	250	22	210	8	M16	18
150	172	11	285	22	240	8	M20	23
175	199	12	315	24	270	8	M20	23
200	224	12	340	24	295	12	M20	23
250	278	14	405	26	355	12	M24	27
300	330	15	460	28	410	12	M24	27

DIN Cast Steel Flange Standard Nenndruck 25

(Nominal Pressure 25 Kgf/cm²)

Nominal Bore	Pipe		Flange			Bolting		
	Outside Dia.	Thickness	Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	22	6	90	16	60	4	M12	14
15	25	6	95	16	65	4	M12	14
20	33	6.5	105	18	75	4	M12	14
25	39	7	115	18	85	4	M12	14
32	46	7	140	18	100	4	M16	18
40	55	7.5	150	18	110	4	M16	18
50	65	7.5	165	20	125	4	M16	18
65	86	8	185	22	145	8	M16	18
80	97	8.5	200	24	160	8	M16	18
100	119	9.5	235	24	190	8	M20	23
125	145	10	270	26	220	8	M24	27
150	172	11	300	28	250	8	M24	27
175	199	12	330	28	280	12	M24	27
200	224	12	360	30	310	12	M24	27
250	278	14	425	32	370	12	M27	30
300	330	15	485	34	430	16	M27	30

DIN Cast Steel Flange Standard Nenndruck 40

(Nominal Pressure 40 Kgf/cm²)

Nominal Bore	Pipe		Flange			Bolting		
	Outside Dia.	Thickness	Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	22	6	90	16	60	4	M12	14
15	25	6	95	16	65	4	M12	14
20	33	6.5	105	18	75	4	M12	14
25	39	7	115	18	85	4	M12	14
32	46	7	140	18	100	4	M16	18
40	55	7.5	150	18	110	4	M16	18
50	66	8	165	20	125	4	M16	18
65	87	8.5	185	22	145	8	M16	18
80	98	9	200	24	160	8	M16	18
100	120	10	235	24	190	8	M20	23
125	147	11	270	26	220	8	M24	27
150	174	12	300	28	250	8	M24	27
175	201	13	350	32	295	12	M27	30
200	228	14	375	34	320	12	M27	30
250	282	16	450	38	385	12	M30	33
300	334	17	515	42	450	16	M30	33

All dimensions in mm.

All pressures in Kgf/cm².

*German flange standards that are widely used in Europe.

Reprinted from "Control Valve Handbook," 1965, Fisher Controls Company, Marshalltown, Iowa.

DIN STANDARDS

DIN Cast Steel Flange Standard Nenndruck 64

(Nominal Pressure 64 Kgf/cm²)

Nominal Bore	Pipe Thickness	Flange			Bolting		
		Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	8	100	20	70	4	M12	14
15	8	105	20	75	4	M12	14
25	9	140	24	100	4	M16	18
40	9	170	26	125	4	M20	23
50	10	180	26	135	4	M20	23
65	10	205	26	160	8	M20	23
80	11	215	28	170	8	M20	23
100	12	250	30	200	8	M24	27
125	13	295	34	240	8	M27	30
150	14	345	36	280	8	M30	33
175	15	375	40	310	12	M30	33
200	16	415	42	345	12	M33	36
250	19	470	46	400	12	M33	36
300	21	530	52	460	16	M33	36

DIN Cast Steel Flange Standard Nenndruck 100

(Nominal Pressure 100 Kgf/cm²)

Nominal Bore	Pipe Thickness	Flange			Bolting		
		Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	10	100	20	70	4	M12	14
15	10	105	20	75	4	M12	14
25	11	140	24	100	4	M16	18
40	11	170	26	125	4	M20	23
50	12	195	28	145	4	M24	27
65	13	220	30	170	8	M24	27
80	14	230	32	180	8	M24	27
100	15	265	36	210	8	M27	30
125	16	315	40	250	8	M30	33
150	18	355	44	290	12	M30	33
175	20	385	48	320	12	M30	33
200	21	430	52	360	12	M33	36
250	25	505	60	430	12	M36	39
300	29	585	68	500	16	M39	42

DIN Cast Steel Flange Standard Nenndruck 160

(Nominal Pressure 160 Kgf/cm²)

Nominal Bore	Pipe Thickness	Flange			Bolting		
		Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	10	100	20	70	4	M12	14
15	10	105	20	75	4	M12	14
25	10	140	24	100	4	M16	18
40	10	170	28	125	4	M20	23
50	10	195	30	145	4	M24	27
65	11	220	34	170	8	M24	27
80	12	230	36	180	8	M24	27
100	14	265	40	210	8	M27	30
125	16	315	44	250	8	M30	33
150	18	355	50	290	12	M30	33
175	19	390	54	320	12	M33	36
200	21	430	60	360	12	M33	36
250	24	515	68	430	12	M39	42
300	28	585	78	500	16	M39	42

DIN Cast Steel Flange Standard Nenndruck 250

(Nominal Pressure 250 Kgf/cm²)

Nominal Bore	Pipe Thickness	Flange			Bolting		
		Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	10	125	24	85	4	M16	18
15	10	130	26	90	4	M16	18
25	11	150	28	105	4	M20	23
40	13	185	34	135	4	M24	27
50	13	200	38	150	8	M24	27
65	14	230	42	180	8	M24	27
80	16	255	46	200	8	M27	30
100	19	300	54	235	8	M30	33
125	22	340	60	275	12	M30	33
150	25	390	68	320	12	M33	36
175	29	430	74	355	12	M36	39
200	32	485	82	400	12	M39	42
250	38	585	100	490	16	M45	48
300	47	690	120	590	16	M48	52

All dimensions in mm.

Reprinted from "Control Valve Handbook," 1965, Fisher Controls Company, Marshalltown, Iowa.

DIN STANDARDS

DIN Cast Steel Flange Standard Nenndruck 320

(Nominal Pressure 320 Kgf/cm²)

Nominal Bore	Pipe Thickness	Flange			Bolting		
		Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	11	125	24	85	4	M16	18
15	11	130	26	90	4	M16	18
25	11	160	34	115	4	M20	23
40	14	195	38	145	4	M24	27
50	15	210	42	160	8	M24	27
65	18	255	51	200	8	M27	30
80	19	275	55	220	8	M27	30
100	24	335	65	265	8	M33	36
125	27	380	75	310	12	M33	36
150	32	425	84	350	12	M36	39
175	35	485	95	400	12	M39	42
200	38	525	103	440	16	M39	42
250	49	640	125	540	16	M48	52

DIN Cast Steel Flange Standard Nenndruck 400

(Nominal Pressure 400 Kgf/cm²)

Nominal Bore	Pipe Thickness	Flange			Bolting		
		Outside Dia.	Thickness	Bolt Circle Dia.	No. of Bolts	Thread	Bolt Hole Dia.
10	11	125	28	85	4	M16	18
15	11	145	30	100	4	M20	23
20	11	160	34	115	4	M20	23
25	12	180	38	130	4	M24	27
32	14	205	44	150	4	M27	30
40	15	220	48	165	4	M27	30
50	18	235	52	180	8	M27	30
65	22	290	64	225	8	M30	33
80	25	305	68	240	8	M30	33
100	30	370	80	295	8	M36	39
125	36	415	92	340	12	M36	39
150	41	475	105	390	12	M39	42
175	47	545	120	450	12	M45	48
200	53	585	130	490	16	M45	48

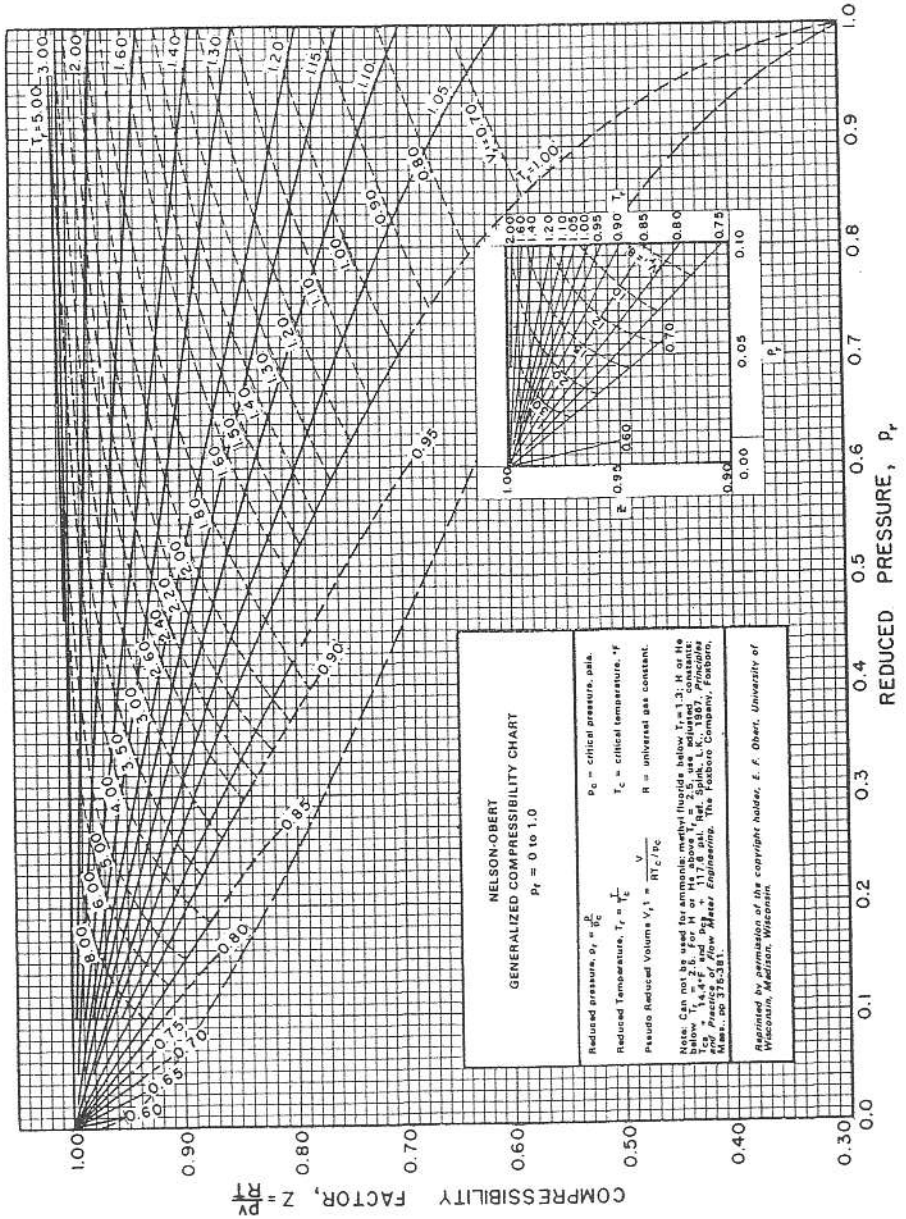
All dimensions in mm.

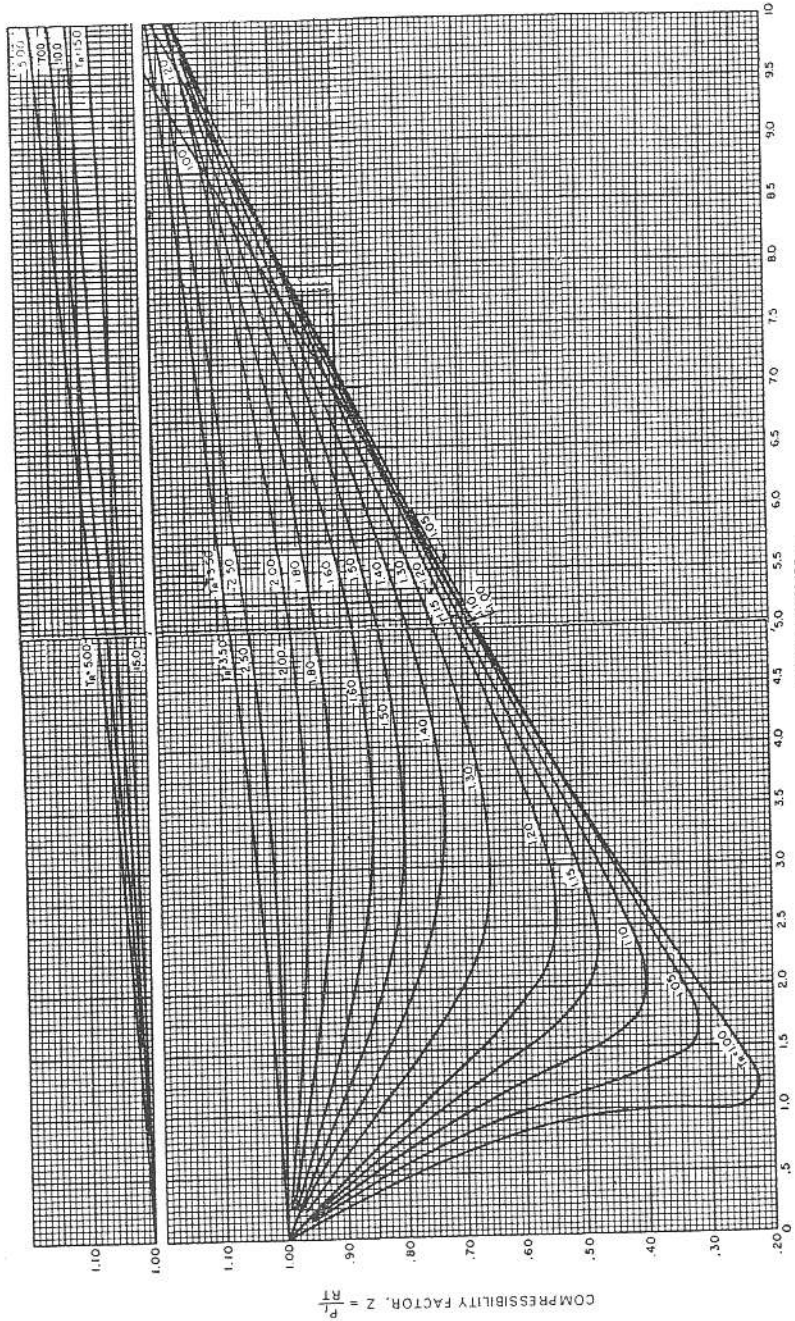
Reprinted from "Control Valve Handbook," 1965, Fisher Controls Company, Marshalltown, Iowa.

VALVE SIZING DATA
REPRESENTATIVE VALVE FACTORS

BODY & TRIM TYPE	FLOW DIRECTION	LINE SIZE BODY (D=d)						HALF SIZE (D=2d)			
		C_d	F_L	X_T	F_d*	F_s*	K_c	$\frac{N_3 C_v}{D^2}$	F_{LP}	x_{TP}	F_s
Single Seat Globe											
Wing Guided	Either	11	.90	.75	1.0	1.05	c	2.8	.85	.75	1.04
V-Skirt	Either	9	.90	.75	1.5	1.38	c	2.3	.86	.75	1.36
Contoured	Open	11	.90	.72	1.0	1.05	.65	2.8	.85	.73	1.04
Contoured	Close	11	.80	.55	1.0	1.09	.58	2.8	.76	.57	1.08
V-Plug	Either	9.5	.90	.75	1.0	1.05	.80	2.4	.86	.75	1.04
Cage	Open	14	.90	.75	1.0	1.06	.65	3.5	.82	.75	1.04
Cage	Close	16	.80	.70	1.0	1.11	c	4.0	.72	.71	1.08
Double Seat Globe											
Wing Guided	--	14	.90	.75	.71	0.84	c	3.5	.82	.75	0.83
V-Skirt	--	13	.90	.75	.71	0.84	c	3.3	.83	.75	0.83
Contoured	--	13	.85	.70	.71	0.85	.70	3.3	.79	.71	0.84
V-Plug	--	12.5	.90	.75	.71	0.84	.80	3.1	.83	.75	0.84
Angle											
Full Port Contour	Close	20	.80	.65	1.0	1.12	.53	5.0	.69	.68	1.08
Full Port Contour	Open	17	.90	.72	1.0	1.08	.64	4.3	.78	.73	1.04
Restricted Contour	Close	≥6	.70	.55	1.0	1.13	c	1.5	.69	.56	1.13
Restricted Contour	Open	≥5.5	.95	.80	1.0	1.02	c	1.3	.93	.80	1.02
2:1 Tapered Orif.	Close	12	.45	.15	1.0	1.31	c	3.0	.44	.17	1.31
Cage	Open	12[a]	.85	.65	1.0	1.08	c	3.0	.80	.66	1.06
Cage	Close	12[a]	.80	.60	1.0	1.10	c	3.0	.75	.62	1.08
Venturi	Close	22	.50	.20	1.0	1.29	.17	5.5	.46	.26	1.26
Ball											
Std. Bore[b]	--	30	.55	.15	1.0	1.28	.25	7.5	.47	.24	1.22
Characterized	--	25	.57	.25	1.0	1.25	.22	6.3	.50	.33	1.21
Butterfly											
60-Deg. Open	--	17	.68	.38	.71	0.92	.3	4.3	.63	.43	0.91
90-Deg. Open	--	>30	.55	.20	.71	1.01	c	>7.5	.45	.33	0.97

* The values of F_d are based on limited test data which have not been corroborated by independent laboratories. F_s is computed from F_d. Key: a = Variable, b = Orif. ≅ 0.8d, c = Unavailable.





NELSON-OBERT GENERALIZED COMPRESSIBILITY CHART, P_r = 0 to 10.

Note: Refer to page 488 for explanation of P_r, T_r and $V_r 1$ and for limitations with certain gases.

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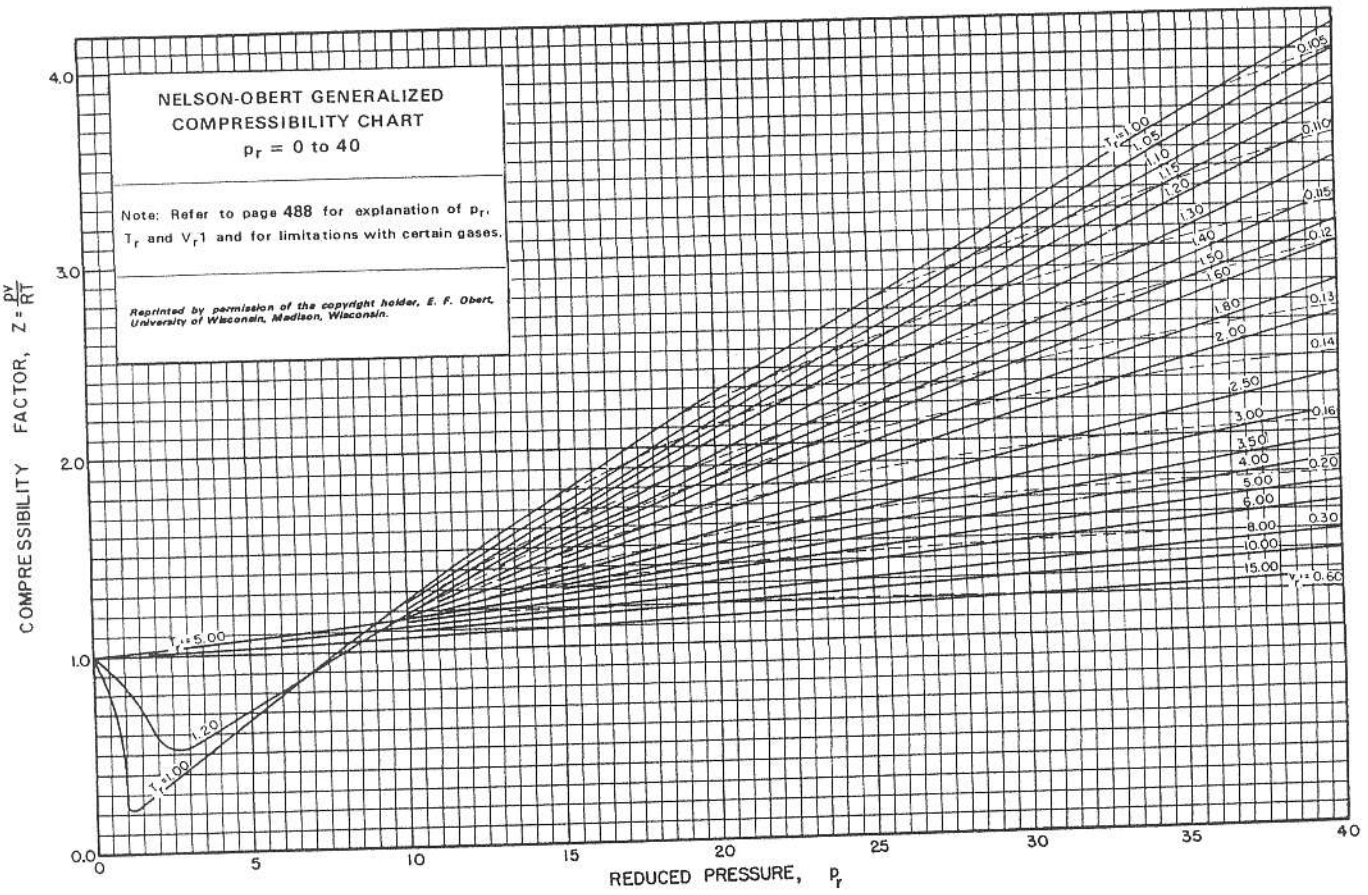
NELSON-OBERT GENERALIZED
COMPRESSIBILITY CHART
p_r = 0 to 40
Note: Refer to page 488 for explanation of p_r, T_r and V_r1 and for limitations with certain gases.
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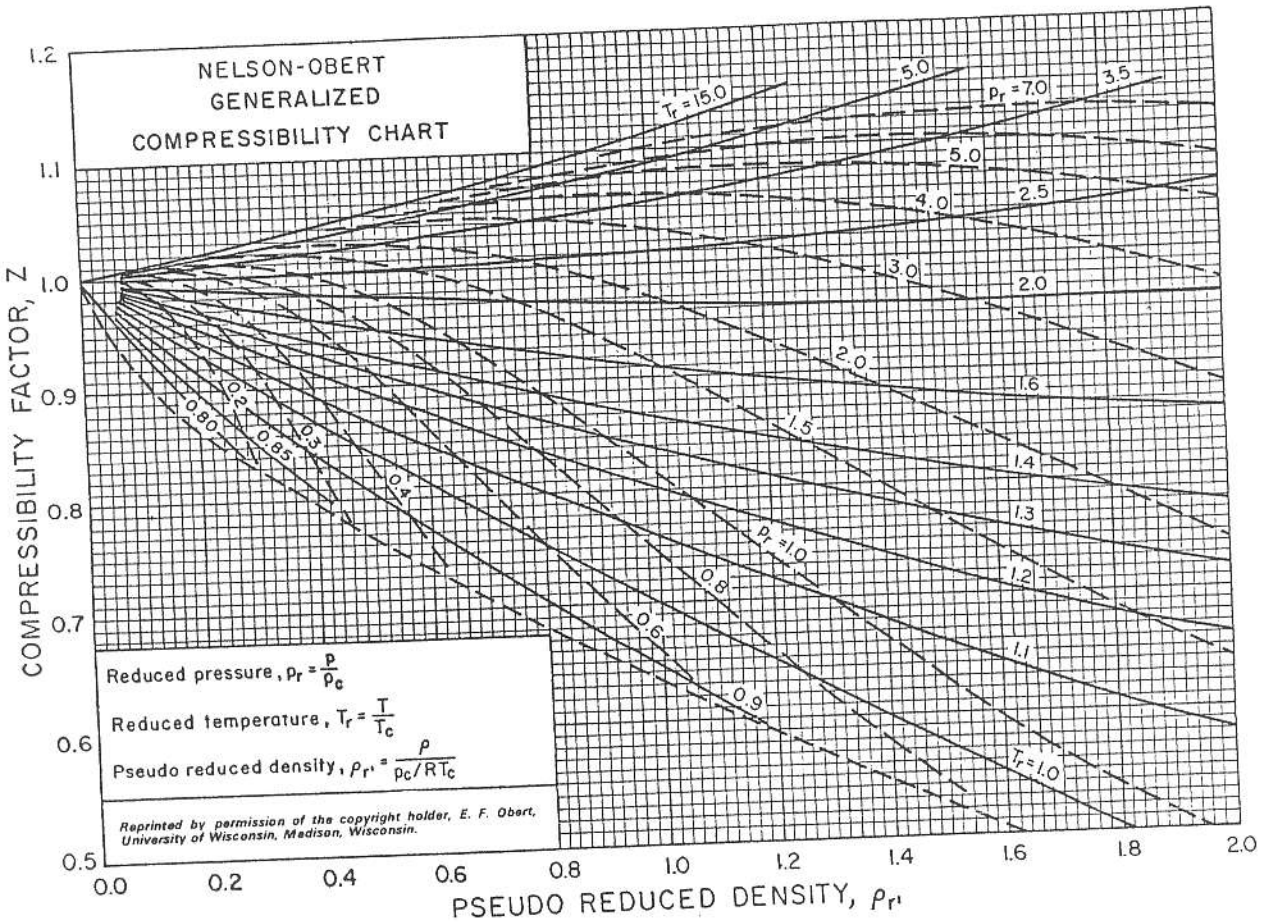
COMPRESSIBILITY FACTOR, $Z = \frac{pV}{RT}$
REDUCED PRESSURE, p_r
T_r=5.00
1.20
1.00
T_r=1.00
1.05
1.10
1.15
1.20
1.30
1.40
1.50
1.60
1.80
2.00
2.50
3.00
3.50
4.00
5.00
6.00
8.00
10.00
15.00
V_r=0.60
0.105
0.110
0.115
0.12
0.13
0.14
0.16
0.20
0.30


NELSON-OBERT
GENERALIZED
COMPRESSIBILITY CHART
COMPRESSIBILITY FACTOR, Z
PSEUDO REDUCED DENSITY, $\rho_{r'}$
Reduced pressure, $p_r = \frac{p}{p_c}$
Reduced temperature, $T_r = \frac{T}{T_c}$
Pseudo reduced density, $\rho_{r'} = \frac{\rho}{p_c/RT_c}$
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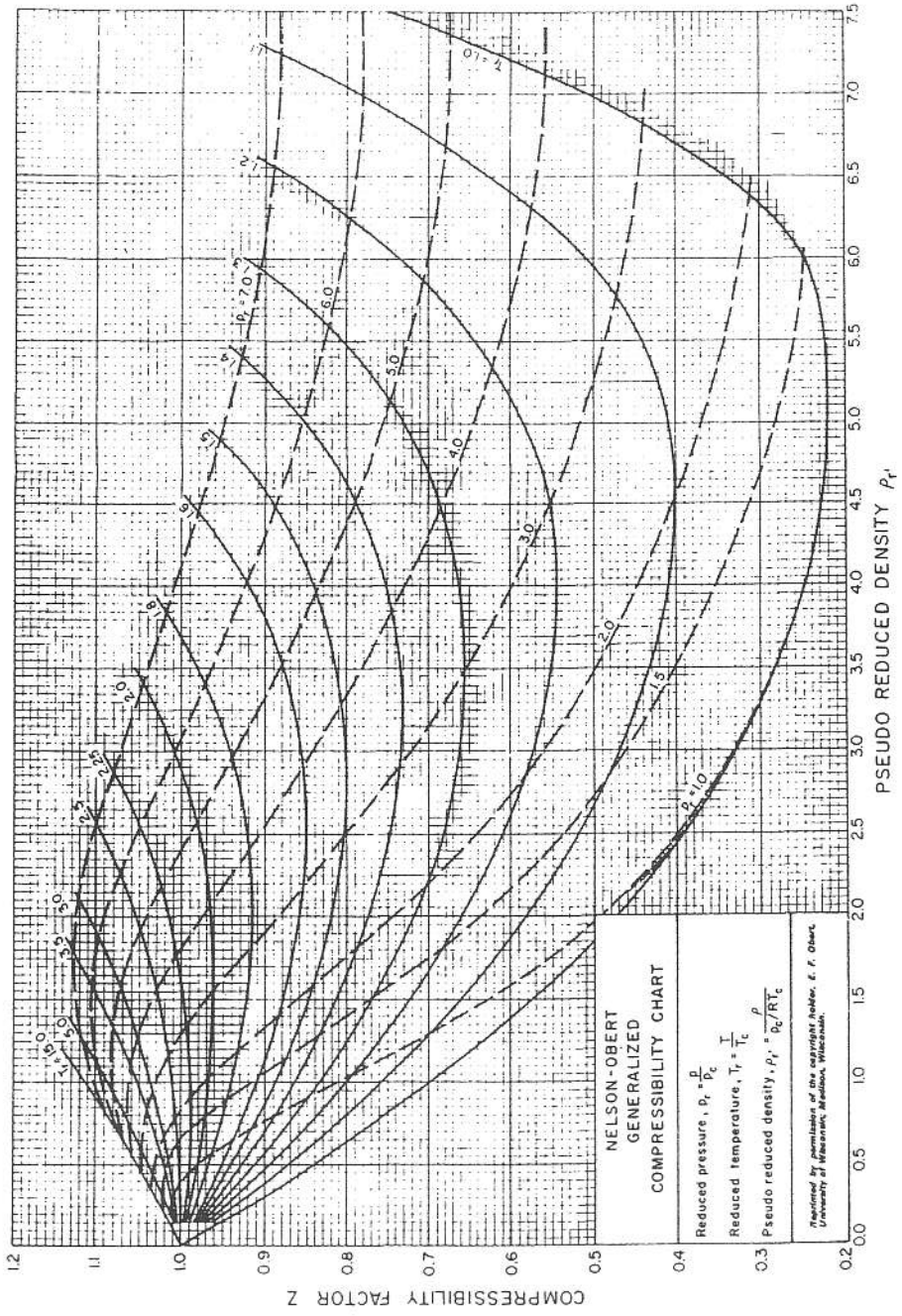
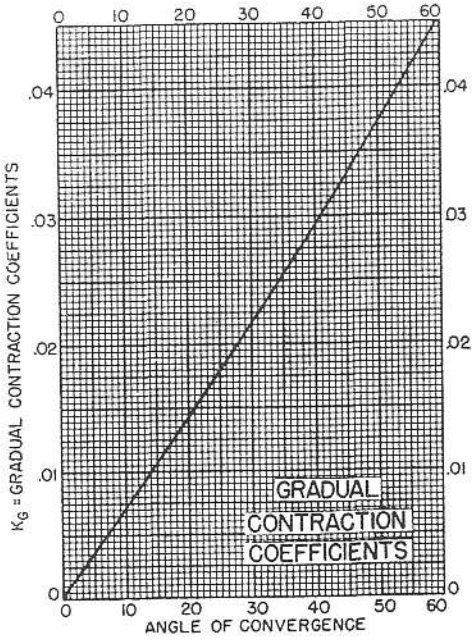
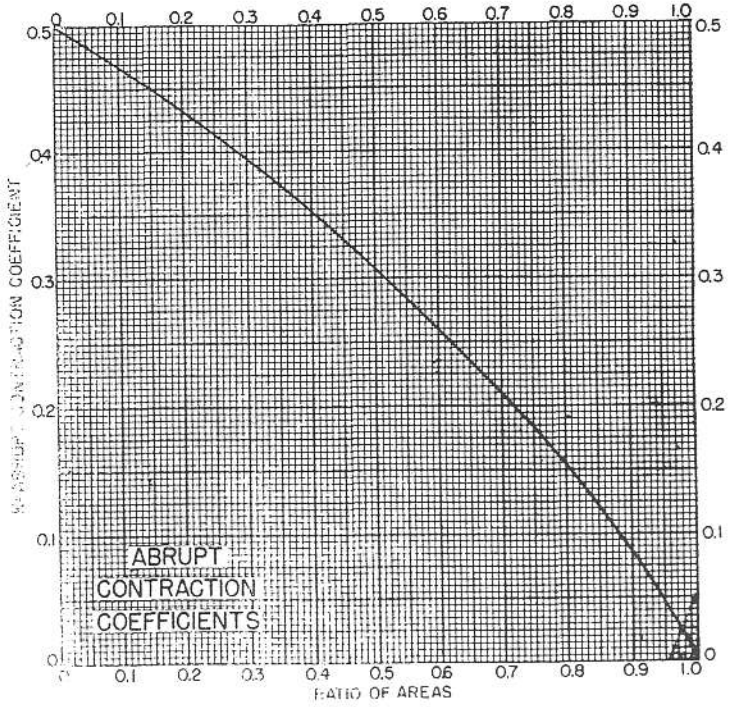
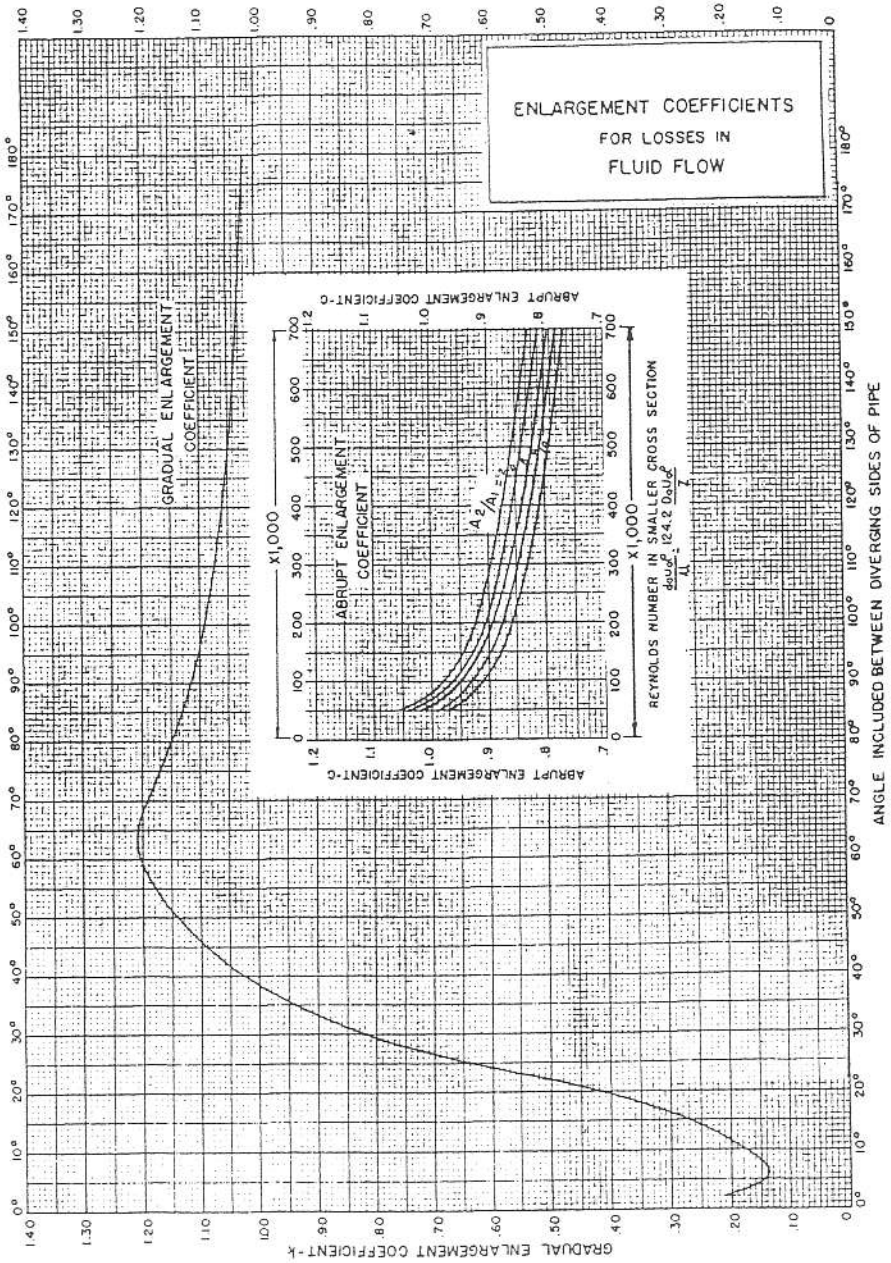
T_r=15.0
5.0
3.5
p_r=7.0
5.0
2.5
4.0
3.0
2.0
2.0
1.6
1.5
1.4
1.3
p_r=1.0
1.2
0.8
1.1
0.6
0.9
T_r=1.0
0.2
0.80
0.85
0.3
0.4



COEFFICIENTS FOR CONTRACTION LOSSES IN FLUID FLOW







Specific Gravity and Viscosity of Liquids

Liquid	Boiling point at atm press °F	Specific gravity: Temp °F	Specific gravity: Water at 4°C = 1	Kinematic viscosity: Temp °F	Kinematic viscosity: Centistokes	Kinematic viscosity: SSU	Coordinates chart, pg 497: a	Coordinates chart, pg 497: b
Acetaldehyde	69	68	.782	68	.295	—	3.0	0
Acetic acid	245	68	1.049	68	1.17	—	3.3	5.1
Acetone	133	68	.79	68	.41	—	2.8	1.5
Aero motor oil (typical)	—	60	.895	100	—	1100		
	—	—	—	210	—	95	2.7	12.2
Alcohol, allyl-n	207	68	.855	68	1.60	—	3.9	5.6
Alcohol, butyl-n	243	68	.81	68	3.64	38	3.3	7.31†
	243	158	.78	158	1.17	—	4.65	7.05††
Alcohol, ethyl-n (grain)	172	68	.789	68	1.52	—	3.35	5.67
Alcohol, methyl-n (wood)	151	68	.79	59	.74	—	3.8	3.8
Alcohol, propyl-n	207	68	.804	68	2.8	35	3.2	6.95
Ammonia (liquid)	-28	0	.662	0	.30	—	—	—
Aniline	363	68	1.022	68	4.37	40	4.2	7.15
Automobile crankcase oils, SAE 10	—	60	.88–.94	130	—	90–120	3.0	10.7
SAE 20	—	60	.88–.94	130	—	120–185	3.0	11.3
SAE 30	—	60	.88–.94	130	—	185–225	3.0	11.65
SAE 40	—	60	.88–.94	130	—	225–400	2.9	12.0
SAE 50	—	60	.88–.94	210	—	80–105	2.8	12.2
SAE 60	—	60	.88–.94	210	—	105–125	2.65	12.65
SAE 70	—	60	.88–.94	210	—	125–150	2.6	12.7
Automobile transmission lub, SAE 80	—	60	.88–.94	0	—	100,000 max	—	—
SEA 90	—	60	.88–.94	100	—	800–1500	—	—
SAE 140	—	60	.88–.94	210	—	120–200	—	—
SAE 250	—	60	.88–.94	210	—	200 min	—	—
Beer	—	60	1.01	68	—	32 (est)	—	—
Benzol (Benzene)	176	68	.879	68	.744	—	3.6	3.85
Brine, calcium chloride, 25%	—	60	1.23	60	—	39	—	—
Brine, sodium chloride, 25%	—	60	1.19	60	—	34	—	—
Bromine	142	68	2.9	68	.34	—	—	—
Butyric acid-n	316	68	.959	68	1.61	—	3.0	5.95
Carbolic acid (phenol)	360	65	1.08	65	11.83	65	4.5	7.9
Carbon disulphide	115	68	1.263	68	.298	—	-.9	8.9*
Carbon tetrachloride	170	68	1.594	68	.612	—	4.0	3.25
Castor Oil	—	68	.96	68	—	4600	2.75	12.2
Chloroform	142	68	1.489	68	.38	—	3.05	1.2
Compounded steam cyl oil (5% tallow)	—	60	.90	110	—	1800	2.4	12.8
				210	—	145		
Decane-n	343	68	.73	68	1.24	—	2.95	5.35
Diethyl ether	94.4	68	.714	68	.32	—	3.0	.6
Ethyl acetate	171	68	.90	68	.49	—	3.7	2.5
Ethyl bromide	101	59	1.45	68	.27	—	-.2	8.3*
Ethylene bromide	269	68	2.18	68	.787	—	3.6	3.95
Ethylene chloride	183	68	1.246	68	.668	—	3.5	3.5
Formic acid	213	68	1.221	68	1.48	—	3.9	5.5

†Accurate for the range between 32 and 158°F.
††Accurate for the range between 139 and 158°F.
*Divide figure obtained with these coordinates by 10 to obtain viscosity in centistokes.

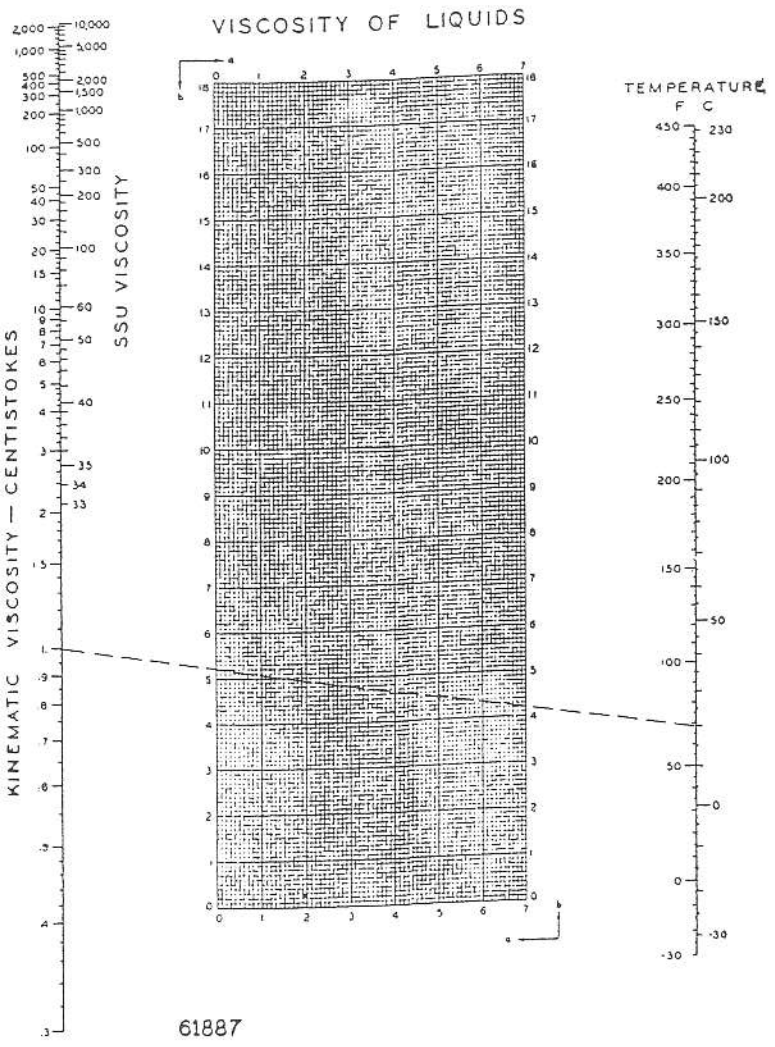
Reprinted from "Cameron Hydraulics Data," 1958, Cameron Pump Division, Ingersoll Rand Company, Phillipsburg, New Jersey.

Specific Gravity and Viscosity of Liquids

Liquid	Boiling point at atm press °F	Specific gravity		Kinematic viscosity			Coordinates chart, pg 497	
		Temp °F	Water at 4°C—1	Temp °F	Centi stokes	SSU	a	b
Freon 11	—	70	1.49	70	.32	—	—	—
Freon 12	—	70	1.33	70	.21	—	—	—
Freon 21	—	70	1.37	70	.27	—	—	—
Fuel Oil, No. 1	—	60	.82–95	100	—	30–40	—	—
No. 2	—	60	.82–95	100	—	35–50	—	—
No. 3	—	60	.82–95	100	—	55 max	—	—
No. 5	—	60	.82–95	100	—	60–450	—	—
No. 6	—	60	.82–95	122	—	430–2900	—	—
Gasoline, typical (a)	—	6	.74	60	.88	—	3.15	4.15
(b)	—	6	.72	60	.64	—	3.7	3.2
(c)	—	6	.68	60	.46	—	2.6	1.7
Glycerine, 100%	554	68	1.26	68.6	—	2950	3.25	11.5
Glycerine and water, 50%	—	68	1.13	68	—	43	—	—
Glycol, Ethylene	—	68	1.125	70	17.8	88.4		
Heptane-n	209	68	.684	68	.60	—	2.9	3.1
Hexane-n	156	68	.66	68	.49	—	3.0	2.3
Hydrochloric acid, 31.5%	—	68	1.05	68	1.9	—	—	—
Kerosene	—	60	.78–.82	68	—	35	—	—
Lard oil	—	60	.91–.92	100	—	180–200	—	—
Linseed oil (raw)	538	60	.92–.94	100	—	143	2.0	11.1
Marine engine oil (20% blown rape)	—	60	.94	210	—	70	2.5	11.9
Methy acetate	135	68	.93	68	.44	—	3.1	1.6
Methy iodide	108	68	2.28	68	.213	—	.35	7.5*
Milk	—	68	1.02–1.04	68	—	32 est	—	—
Naphthalene	424	68	1.145	176	.9	—	—	—
Neatsfoot oil	—	60	.91–.92	60	107-123	—	—	—
Nitric acid, 60%	—	68	1.37	68	1.87	—	—	—
Nitrobenzene	412	68	1.203	68	1.67	—	3.1	6.0
Nonane-n	302	68	.718	68	.97	—	1.6	4.7
Octane-n	258	68	.70	68	.77	—	3.0	3.95
Olive oil	(570)	68	.91	68	93.0	—	2.4	11.4
Pentane-n	97	68	.63	68	.37	—	2.8	1.1
Petroleum either (benzine)	—	60	.64	60	—	31 est	—	—
Propionic acid	286	68	.99	68	1.13	—	3.2	5.05
Quenching oil (typical)	—	60	.86–89	100	—	100–120	—	—
Rapeseed oil	—	68	.91	68	180	—	2.6	11.5
Soya bean oil	—	60	.924	60	86	—	2.1	11.3
Sperm oil	(209)	77	.88	100	21	—	2.3	10.7
Sugar, 20%	—	68	1.08	68		33.5	—	—
40%	—	68	1.18	68	5.3	43	—	—
60%	—	68	1.29	68	44	202	—	—
Sulfuric acid, 100%	640	68	1.83	68	14.6	76	—	—
95%	—	68	1.83	68	14.5	75	—	—
60%	—	68	1.50	68	4.4	41	—	—
Turbine oil (typical medium)	—	60	91	100	—	270	—	—
Turpentine	320	60	.86–87	60	1.83	—	2.9	6.1
Water (fresh)	212	60	1.0	60	1.1	31.5	5.0	4.5
Water (sea)	—	60	1.03	60	—	31.5	—	—
Xylene-O	287	68	.87	68	93	—	2.8	4.5

*Divide figure obtained with these coordinates by 10 to obtain viscosity in centistokes.

Reprinted from "Cameron Hydraulics Data," 1958, Cameron Pump Division, Ingersoll Rand Company, Phillipsburg, New Jersey.



If viscosity is known at two temperatures, plot these points on the graph. The intersection of the lines from temperature to viscosity will give a point. Lines through this point will give viscosity at any other temperature. The point shown is for water. Other points are given in the tables pages 495 and 496. These data are approximate. However, coordinates have been given only for those liquids upon which consistent data over the normal temperature range was available.

Reprinted from "Cameron Hydraulics Data," 1958, Cameron Pump Division, Ingersoll Rand Company, Phillipsburg, New Jersey.

Physical Properties of Water

Temperature of Water t Degrees Fahrenheit	Saturation Pressure P' Pounds per Square Inch Absolute	Specific Volume $\overline{V}$ Cubic Feet Per Pound	Weight Density ρ Pounds per Cubic Foot	Weight Pounds Per Gallon
32	0.08859	0.016022	62.414	8.3436
40	0.12163	0.016019	62.426	8.3451
50	0.17796	0.016023	62.410	8.3430
60	0.25611	0.016033	62.371	8.3378
70	0.36292	0.016050	62.305	8.3290
80	0.50683	0.016072	62.220	8.3176
90	0.69813	0.016099	62.116	8.3037
100	0.94924	0.016130	61.996	8.2877
110	1.2750	0.016165	61.862	8.2698
120	1.6927	0.016204	61.7132	8.2498
130	2.2230	0.016247	61.550	8.2280
140	2.8892	0.016293	61.376	8.2048
150	3.7184	0.016343	61.188	8.1797
160	4.7414	0.016395	60.994	8.1537
170	5.9926	0.016451	60.787	8.1260
180	7.5110	0.016510	60.569	8.0969
190	9.340	0.016572	60.343	8.0667
200	11.526	0.016637	60.107	8.0351
210	14.123	0.016705	59.862	8.0024
212	14.696	0.016719	59.812	7.9957
220	17.186	0.016775	59.613	7.9690
240	24.968	0.016926	59.081	7.8979
260	35.427	0.017089	58.517	7.8226
280	49.200	0.017264	57.924	7.7433
300	67.005	0.01745	57.307	7.6608
350	134.604	0.01799	55.586	7.4308
400	247.259	0.01864	53.648	7.1717
450	422.55	0.01943	51.467	6.8801
500	680.86	0.02043	48.948	6.5433
550	1045.43	0.02176	45.956	6.1434
600	1543.2	0.02364	42.301	5.6548
650	2208.4	0.02674	37.397	4.9993
700	3094.3	0.03662	27.307	3.6505

Specific gravity of water at 60 F = 1.00

Weight per gallon is based on 7.48052 gallons per cubic foot.

All data on volume and pressure are abstracted from ASME Steam Tables (1967), with permission of publisher, The American Society of Mechanical Engineers, 345 East 47th Street, New York, N.Y. 10017.

Courtesy Crane Co., Reproduced from Technical Paper No. 410, "Flow of Fluids."

PROPERTIES OF SATURATED STEAM

Absolute Pressure		Vacuum Inches of Hg	Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Lbs. Per Sq. In. P'	Inches of Hg		t Degrees F.	Btu/lb.	Btu/lb.	H_g Btu/lb.	$\overline{V}$ Cu. Ft. per Lb.
0.20	0.41	29.51	53.14	21.21	1063.8	1085.0	1526.0
0.25	0.51	29.41	59.30	27.36	1060.3	1087.7	1235.3
0.30	0.61	29.31	64.47	32.52	1057.4	1090.0	1039.5
0.35	0.71	29.21	68.93	36.97	1054.9	1091.9	898.5
0.40	0.81	29.11	72.86	40.89	1052.7	1093.6	791.9
0.45	0.92	29.00	76.38	44.41	1050.7	1095.1	708.5
0.50	1.02	28.90	79.58	47.60	1048.8	1096.4	641.4
0.60	1.22	28.70	85.21	53.21	1045.7	1098.9	540.0
0.70	1.43	28.49	90.08	58.07	1042.9	1101.0	466.9
0.80	1.63	28.29	94.38	62.36	1040.4	1102.8	411.7
0.90	1.83	28.09	98.24	66.21	1038.3	1104.5	368.4
1.0	2.04	27.88	101.74	69.70	1036.3	1106.0	333.6
1.2	2.44	27.48	107.92	75.87	1032.7	1108.6	280.9
1.4	2.85	27.07	113.26	81.20	1029.6	1110.8	243.0
1.6	3.26	26.66	117.99	85.91	1026.9	1112.8	214.3
1.8	3.66	26.26	122.23	90.14	1024.5	1114.6	191.8
2.0	4.07	25.85	126.08	93.99	1022.2	1116.2	173.73
2.2	4.48	25.44	129.62	97.52	1020.2	1117.7	158.85
2.4	4.89	25.03	132.89	100.79	1018.3	1119.1	146.38
2.6	5.29	24.63	135.94	103.83	1016.5	1120.3	135.78
2.8	5.70	24.22	138.79	106.68	1014.8	1121.5	126.65
3.0	6.11	23.81	141.48	109.37	1013.2	1122.6	118.71
3.5	7.13	22.79	147.57	115.46	1009.6	1125.1	102.72
4.0	8.14	21.78	152.97	120.86	1006.4	1127.3	90.63
4.5	9.16	20.76	157.83	125.71	1003.6	1129.3	81.16
5.0	10.18	19.74	162.24	130.13	1001.0	1131.1	73.52
5.5	11.20	18.72	166.30	134.19	998.5	1132.7	67.24
6.0	12.22	17.70	170.06	137.96	996.2	1134.2	61.98
6.5	13.23	16.69	173.56	141.47	994.1	1135.6	57.50
7.0	14.25	15.67	176.85	144.76	992.1	1136.9	53.64
7.5	15.27	14.65	179.94	147.86	990.2	1138.1	50.29
8.0	16.29	13.63	182.86	150.79	988.5	1139.3	47.34
8.5	17.31	12.61	185.64	153.57	986.8	1140.4	44.73
9.0	18.32	11.60	188.28	156.22	985.2	1141.4	42.40
9.5	19.34	10.58	190.80	158.75	983.6	1142.3	40.31
10.0	20.36	9.56	193.21	161.17	982.1	1143.3	38.42
11.0	22.40	7.52	197.75	165.73	979.3	1145.0	35.14
12.0	24.43	5.49	201.96	169.96	976.6	1146.6	32.40
13.0	26.47	3.45	205.88	173.91	974.2	1148.1	30.06
14.0	28.50	1.42	209.56	177.61	971.9	1149.5	28.04

Reprinted from "Control Valve Handbook," 1965, Fisher Controls Company, Marshalltown, Iowa.

PROPERTIES OF SATURATED STEAM

Pressure Lbs. per Sq. In.		Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Absolute P'	Gauge P	t			H_g	$\overline{V}$
		Degrees F.	Btu/lb.	Btu/lb.	Btu/lb.	Cu. Ft. per Lb.
14.696	0.0	212.00	180.07	970.3	1150.4	26.80
15.0	0.3	213.03	181.11	969.7	1150.8	26.29
16.0	1.3	216.32	184.42	967.6	1152.0	24.75
17.0	2.3	219.44	187.56	965.5	1153.1	23.39
18.0	3.3	222.41	190.56	963.6	1154.2	22.17
19.0	4.3	225.24	193.42	961.9	1155.3	21.08
20.0	5.3	227.96	196.16	960.1	1156.3	20.089
21.0	6.3	230.57	198.79	958.4	1157.2	19.192
22.0	7.3	233.07	201.33	956.8	1158.1	18.375
23.0	8.3	235.49	203.78	955.2	1159.0	17.627
24.0	9.3	237.82	206.14	953.7	1159.8	16.938
25.0	10.3	240.07	208.42	952.1	1160.6	16.303
26.0	11.3	242.25	210.62	950.7	1161.3	15.715
27.0	12.3	244.36	212.75	949.3	1162.0	15.170
28.0	13.3	246.41	214.83	947.9	1162.7	14.663
29.0	14.3	248.40	216.86	946.5	1163.4	14.189
30.0	15.3	250.33	218.82	945.3	1164.1	13.746
31.0	16.3	252.22	220.73	944.0	1164.7	13.330
32.0	17.3	254.05	222.59	942.8	1165.4	12.940
33.0	18.3	255.84	224.41	941.6	1166.0	12.572
34.0	19.3	257.58	226.18	940.3	1166.5	12.226
35.0	20.3	259.28	227.91	939.2	1167.1	11.898
36.0	21.3	260.95	229.60	938.0	1167.6	11.588
37.0	22.3	262.57	231.26	936.9	1168.2	11.294
38.0	23.3	264.16	232.89	935.8	1168.7	11.015
39.0	24.3	265.72	234.48	934.7	1169.2	10.750
40.0	25.3	267.25	236.03	933.7	1169.7	10.498
41.0	26.3	268.74	237.55	932.6	1170.2	10.258
42.0	27.3	270.21	239.04	931.6	1170.7	10.029
43.0	28.3	271.64	240.51	930.6	1171.1	9.810
44.0	29.3	273.05	241.95	929.6	1171.6	9.601
45.0	30.3	274.44	243.36	928.6	1172.0	9.401
46.0	31.3	275.80	244.75	927.7	1172.4	9.209
47.0	32.3	277.13	246.12	926.7	1172.9	9.025
48.0	33.3	278.45	247.47	925.8	1173.3	8.848
49.0	34.3	279.74	248.79	924.9	1173.7	8.678
50.0	35.3	281.01	250.09	924.0	1174.1	8.515
51.0	36.3	282.26	251.37	923.0	1174.4	8.359
52.0	37.3	283.49	252.63	922.2	1174.8	8.208
53.0	38.3	284.70	253.87	921.3	1175.2	8.062
54.0	39.3	285.90	255.09	920.5	1175.6	7.922
55.0	40.3	287.07	256.30	919.6	1175.9	7.787
56.0	41.3	288.23	257.50	918.8	1176.3	7.656
57.0	42.3	289.37	258.67	917.9	1176.6	7.529
58.0	43.3	290.50	259.82	917.1	1176.9	7.407
59.0	44.3	291.61	260.96	916.3	1177.3	7.289
60.0	45.3	292.71	262.09	915.5	1177.6	7.175
61.0	46.3	293.79	263.20	914.7	1177.9	7.064
62.0	47.3	294.85	264.30	913.9	1178.2	6.957
63.0	48.3	295.90	265.38	913.1	1178.5	6.853
64.0	49.3	296.94	266.45	912.3	1178.8	6.752

PROPERTIES OF SATURATED STEAM

Pressure Lbs. per Sq. In.		Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Absolute P'	Gauge P	t			H_g	$\overline{V}$
		Degrees F.	Btu/lb.	Btu/lb.	Btu/lb.	Cu. Ft. per lb.
65.0	50.3	297.97	267.50	911.6	1179.1	6.655
66.0	51.3	298.99	268.55	910.8	1179.4	6.560
67.0	52.3	299.99	269.58	910.1	1179.7	6.468
68.0	53.3	300.98	270.60	909.4	1180.0	6.378
69.0	54.3	301.96	291.61	908.7	1180.3	6.291
70.0	55.3	302.92	272.61	907.9	1180.6	6.206
71.0	56.3	303.88	273.60	907.2	1180.8	6.124
72.0	57.3	304.83	274.57	906.5	1181.1	6.044
73.0	58.3	305.76	275.54	905.8	1181.3	5.966
74.0	59.3	306.68	276.49	905.1	1181.6	5.890
75.0	60.3	307.60	277.43	904.5	1181.9	5.816
76.0	61.3	308.50	278.37	903.7	1182.1	5.743
77.0	62.3	309.40	279.30	903.1	1182.4	5.673
78.0	63.3	310.29	280.21	902.4	1182.6	5.604
79.0	64.3	311.16	281.12	901.7	1182.8	5.537
80.0	65.3	312.03	282.02	901.1	1183.1	5.472
81.0	66.3	312.89	282.91	900.4	1183.3	5.408
82.0	67.3	313.74	283.79	899.7	1183.5	5.346
83.0	68.3	314.59	284.66	899.1	1183.8	5.285
84.0	69.3	315.42	285.53	898.5	1184.0	5.226
85.0	70.3	316.25	286.39	897.8	1184.2	5.168
86.0	71.3	317.07	287.24	897.2	1184.4	5.111
87.0	72.3	317.88	288.08	896.5	1184.6	5.055
88.0	73.3	318.68	288.91	895.9	1184.8	5.001
89.0	74.3	319.48	289.74	895.3	1185.1	4.948
90.0	75.3	320.27	290.56	894.7	1185.3	4.896
91.0	76.3	321.06	291.38	894.1	1185.5	4.845
92.0	77.3	321.83	292.18	893.5	1185.7	4.796
93.0	78.3	322.60	292.98	892.9	1185.9	4.747
94.0	79.3	323.36	293.78	892.3	1186.1	4.699
95.0	80.3	324.12	294.56	891.7	1186.2	4.652
96.0	81.3	324.87	295.34	891.1	1186.4	4.606
97.0	82.3	325.61	296.12	890.5	1186.6	4.561
98.0	83.3	326.35	296.89	889.9	1186.8	4.517
99.0	84.3	327.08	297.65	889.4	1187.0	4.474
100.0	85.3	327.81	298.40	888.8	1187.2	4.432
101.0	86.3	328.53	299.15	888.2	1187.4	4.391
102.0	87.3	329.25	299.90	887.6	1187.5	4.350
103.0	88.3	329.96	300.64	887.1	1187.7	4.310
104.0	89.3	330.66	301.37	886.5	1187.9	4.271
105.0	90.3	331.36	302.10	886.0	1188.1	4.232
106.0	91.3	332.05	302.82	885.4	1188.2	4.194
107.0	92.3	332.74	303.54	884.9	1188.4	4.157
108.0	93.3	333.42	304.26	884.3	1188.6	4.120
109.0	94.3	334.10	304.97	883.7	1188.7	4.084
110.0	95.3	334.77	305.66	883.2	1188.9	4.049
111.0	96.3	335.44	306.37	882.6	1189.0	4.015
112.0	97.3	336.11	307.06	882.1	1189.2	3.981
113.0	98.3	336.77	307.75	881.6	1189.4	3.947
114.0	99.3	337.42	308.43	881.1	1189.5	3.914

PROPERTIES OF SATURATED STEAM

Pressure Lbs. per Sq. In. Absolute P'	Pressure Lbs. per Sq. In. Gauge P	Temperature t Degrees F.	Heat of the Liquid Btu/lb.	Latent Heat of Evaporation Btu/lb.	Total Heat of Steam H_g Btu/lb.	Specific Volume V Cu. Ft. per lb.
115.0	100.3	338.07	309.11	880.6	1189.7	3.882
116.0	101.3	338.72	309.79	880.0	1189.8	3.850
117.0	102.3	339.36	310.46	879.5	1190.0	3.819
118.0	103.3	339.99	311.12	879.0	1190.1	3.788
119.0	104.3	340.62	311.78	878.4	1190.2	3.758
120.0	105.3	341.25	312.44	877.9	1190.4	3.728
121.0	106.3	341.88	313.10	877.4	1190.5	3.699
122.0	107.3	342.50	313.75	876.9	1190.7	3.670
123.0	108.3	343.11	314.40	876.4	1190.8	3.642
124.0	109.3	343.72	315.04	875.9	1190.9	3.614
125.0	110.3	344.33	315.68	875.4	1191.1	3.587
126.0	111.3	344.94	316.31	874.9	1191.2	3.560
127.0	112.3	345.54	316.94	874.4	1191.3	3.533
128.0	113.3	346.13	317.57	873.9	1191.5	3.507
129.0	114.3	346.73	318.19	873.4	1191.6	3.481
130.0	115.3	347.32	318.81	872.9	1191.7	3.455
131.0	116.3	347.90	319.43	872.5	1191.9	3.430
132.0	117.3	348.48	320.04	872.0	1192.0	3.405
133.0	118.3	349.06	320.65	871.5	1192.1	3.381
134.0	119.3	349.64	321.25	871.0	1192.2	3.357
135.0	120.3	350.21	321.85	870.6	1192.4	3.333
136.0	121.3	350.78	322.45	870.1	1192.5	3.310
137.0	122.3	351.35	323.05	869.6	1192.6	3.287
138.0	123.3	351.91	323.64	869.1	1192.7	3.264
139.0	124.3	352.47	324.23	868.7	1192.9	3.242
140.0	125.3	353.02	324.82	868.2	1193.0	3.220
141.0	126.3	353.57	325.40	867.7	1193.1	3.198
142.0	127.3	354.12	325.98	867.2	1193.2	3.177
143.0	128.3	354.67	326.56	866.7	1193.3	3.155
144.0	129.3	355.21	327.13	866.3	1193.4	3.134
145.0	130.3	355.76	327.70	865.8	1193.5	3.114
146.0	131.3	356.29	328.27	865.3	1193.6	3.094
147.0	132.3	356.83	328.83	864.9	1193.8	3.074
148.0	133.3	357.36	329.39	864.5	1193.9	3.054
149.0	134.3	357.89	329.95	864.0	1194.0	3.034
150.0	135.3	358.42	330.51	863.6	1194.1	3.015
152.0	137.3	359.46	331.61	862.7	1194.3	2.977
154.0	139.3	360.49	332.70	861.8	1194.5	2.940
156.0	141.3	361.52	333.79	860.9	1194.7	2.904
158.0	143.3	362.53	334.86	860.0	1194.9	2.869
160.0	145.3	363.53	335.93	859.2	1195.1	2.834
162.0	147.3	364.53	336.98	858.3	1195.3	2.801
164.0	149.3	365.51	338.02	857.5	1195.5	2.768
166.0	151.3	366.48	339.05	856.6	1195.7	2.736
168.0	153.3	367.45	340.07	855.7	1195.8	2.705
170.0	155.3	368.41	341.09	854.9	1196.0	2.675
172.0	157.3	369.35	342.10	854.1	1196.2	2.645
174.0	159.3	370.29	343.10	853.3	1196.4	2.616
176.0	161.3	371.22	344.09	852.4	1196.5	2.587
178.0	163.3	372.14	345.06	851.6	1196.7	2.559

PROPERTIES OF SATURATED STEAM

Pressure Lbs. per Sq. In. Absolute P'	Pressure Lbs. per Sq. In. Gauge P	Temperature t Degrees F.	Heat of the Liquid Btu/lb.	Latent Heat of Evaporation Btu/lb.	Total Heat of Steam H_g Btu/lb.	Specific Volume V Cu. Ft. per lb.
180.0	165.3	373.06	346.03	850.8	1196.9	2.532
182.0	167.3	373.96	347.00	850.0	1197.0	2.505
184.0	169.3	374.86	347.96	849.2	1197.2	2.479
186.0	171.3	375.75	348.92	848.4	1197.3	2.454
188.0	173.3	376.64	349.86	847.6	1197.5	2.429
190.0	175.3	377.51	350.79	846.8	1197.6	2.404
192.0	177.3	378.38	351.72	846.1	1197.8	2.380
194.0	179.3	379.24	352.64	845.3	1197.9	2.356
196.0	181.3	380.10	353.55	844.5	1198.1	2.333
198.0	183.3	380.95	354.46	843.7	1198.2	2.310
200.0	185.3	381.79	355.36	843.0	1198.4	2.288
205.0	190.3	383.86	357.58	841.1	1198.7	2.234
210.0	195.3	385.90	359.77	839.2	1199.0	2.183
215.0	200.3	387.89	361.91	837.4	1199.3	2.134
220.0	205.3	389.86	364.02	835.6	1199.6	2.087
225.0	210.3	391.79	366.09	833.8	1199.9	2.0422
230.0	215.3	393.68	368.13	832.0	1200.1	1.9992
235.0	220.3	395.54	370.14	830.3	1200.4	1.9579
240.0	225.3	397.37	372.12	828.5	1200.6	1.9183
245.0	230.3	399.18	374.08	826.8	1200.9	1.8803
250.0	235.3	400.95	376.00	825.1	1201.1	1.8438
255.0	240.3	402.70	377.89	823.4	1201.3	1.8086
260.0	245.3	404.42	379.76	821.8	1201.5	1.7748
265.0	250.3	406.11	381.60	820.1	1201.7	1.7422
270.0	255.3	407.78	383.42	818.5	1201.9	1.7107
275.0	260.3	409.43	385.21	816.9	1202.1	1.6804
280.0	265.3	411.05	386.98	815.3	1202.3	1.6511
285.0	270.3	412.65	388.73	813.7	1202.4	1.6228
290.0	275.3	414.23	390.46	812.1	1202.6	1.5954
295.0	280.3	415.79	392.16	810.5	1202.7	1.5689
300.0	285.3	417.33	393.84	809.0	1202.8	1.5433
320.0	305.3	423.29	400.39	803.0	1203.4	1.4485
340.0	325.3	428.97	406.66	797.1	1203.7	1.3645
360.0	345.3	434.40	412.67	791.4	1204.1	1.2895
380.0	365.3	439.60	418.45	785.8	1204.3	1.2222
400.0	385.3	444.59	424.0	780.5	1204.5	1.1613
420.0	405.3	449.39	429.4	775.2	1204.6	1.1061
440.0	425.3	454.02	434.6	770.0	1204.6	1.0556
460.0	445.3	458.50	439.7	764.9	1204.6	1.0094
480.0	465.3	462.82	444.6	759.9	1204.5	0.9670
500.0	485.3	467.01	449.4	755.0	1204.4	0.9278
520.0	505.3	471.07	454.1	750.1	1204.2	0.8915
540.0	525.3	475.01	458.6	745.4	1204.0	0.8578
560.0	545.3	478.85	463.0	740.8	1203.8	0.8265
580.0	565.3	482.58	467.4	736.1	1203.5	0.7973
600.0	585.3	486.21	471.6	731.6	1203.2	0.7698
620.0	605.3	489.75	475.7	727.2	1202.9	0.7440
640.0	625.3	493.21	479.8	722.7	1202.5	0.7198
660.0	645.3	496.58	483.8	718.3	1202.1	0.6971
680.0	665.3	499.88	487.7	714.0	1201.7	0.6757

PROPERTIES OF SATURATED STEAM

Pressure Lbs. per Sq. In.		Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Absolute P'	Gauge P	t			H_g	$\overline{V}$
		Degrees F.	Btu/lb.	Btu/lb.	Btu/lb.	Cu. Ft. per lb.
700.0	685.3	503.10	491.5	709.7	1201.2	0.6554
720.0	705.3	506.25	495.3	705.4	1200.7	0.6362
740.0	725.3	509.34	499.0	701.2	1200.2	0.6180
760.0	745.3	512.36	502.6	697.1	1199.7	0.6007
780.0	765.3	515.33	506.2	692.9	1199.1	0.5843
800.0	785.3	518.23	509.7	688.9	1198.6	0.5687
820.0	805.3	521.08	513.2	684.8	1198.0	0.5538
840.0	825.3	523.88	516.6	680.8	1197.4	0.5396
960.0	845.3	526.63	520.0	676.8	1196.8	0.5260
880.0	865.3	529.33	523.3	672.8	1196.1	0.5130
900.0	885.3	531.98	526.6	668.8	1195.4	0.5006
920.0	905.3	534.59	529.8	664.9	1194.7	0.4886
940.0	925.3	537.16	533.0	661.0	1194.0	0.4772
960.0	945.3	539.68	536.2	657.1	1193.3	0.4663
980.0	965.3	542.17	539.3	653.3	1192.6	0.4557
1000.0	985.3	544.61	542.4	649.4	1191.8	0.4456
1050.0	1035.3	550.57	550.0	639.9	1189.9	0.4218
1100.0	1085.3	556.31	557.4	630.4	1187.8	0.4001
1150.0	1135.3	561.86	564.6	621.0	1185.6	0.3802
1200.0	1185.3	567.22	571.7	611.7	1183.4	0.3619
1250.0	1235.3	572.42	578.6	602.4	1181.0	0.3450
1300.0	1285.3	577.46	585.4	593.2	1178.6	0.3293
1350.0	1335.3	582.35	592.1	584.0	1176.1	0.3148
1400.0	1385.3	587.10	598.7	574.7	1173.4	0.3012
1450.0	1435.3	591.73	605.2	565.5	1170.7	0.2884
1500.0	1485.3	596.23	611.6	556.3	1167.9	0.2765
1600.0	1585.3	604.90	624.1	538.0	1162.1	0.2548
1700.0	1685.3	613.15	636.3	519.6	1155.9	0.2354
1800.0	1785.3	621.03	648.3	501.1	1149.4	0.2179
1900.0	1885.3	628.58	660.1	482.4	1142.4	0.2021
2000.0	1985.3	635.82	671.7	463.4	1135.1	0.1878
2100.0	2085.3	642.77	683.3	444.1	1127.4	0.1746
2200.0	2185.3	649.46	694.8	424.4	1119.2	0.1625
2300.0	2285.3	655.91	706.5	403.9	1110.4	0.1513
2400.0	2385.3	662.12	718.4	382.7	1101.1	0.1407
2500.0	2485.3	668.13	730.6	360.5	1091.1	0.1307
2600.0	2585.3	673.94	743.0	337.2	1080.2	0.1213
2700.0	2685.3	679.55	756.2	312.1	1068.3	0.1123
2800.0	2785.3	684.99	770.1	284.7	1054.8	0.1035
2900.0	2885.3	690.26	785.4	253.6	1039.0	0.0947
3000.0	2985.3	695.36	802.5	217.8	1020.3	0.0858
3100.0	3085.3	700.31	825.0	168.1	993.1	0.0753
3200.0	3185.3	705.11	872.4	62.0	934.4	0.0580
3206.2	3191.5	705.40	902.7	0.0	902.7	0.0503

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PROPERTIES OF SUPERHEATED STEAM

$\overline{V}$ = specific volume, cubic feet per pound
h_g = total heat of steam, Btu per pound

Pressure Lbs. per Sq. In.		Sat. Temp.		Total Temperature - Degrees Fahrenheit (t)										
Absolute P'	Gauge P	t		360°	400°	440°	480°	500°	600°	700°	800°	900°	1000°	1200°
14.696	0.0	212.00	$\overline{V}$	33.03	34.68	36.32	37.96	38.78	42.86	46.94	51.00	55.07	59.13	67.25
			h_g	1221.1	1239.9	1258.8	1277.6	1287.1	1334.6	1383.2	1432.3	1482.3	1533.1	1637.5
20.0	5.3	227.96	$\overline{V}$	24.21	25.43	26.65	27.86	28.46	31.47	34.47	37.46	40.45	43.44	49.41
			h_g	1220.3	1239.2	1258.2	1277.1	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	1637.4
30.0	15.3	250.33	$\overline{V}$	16.072	16.897	17.714	18.528	18.933	20.95	22.96	24.96	26.95	28.95	32.93
			h_g	1218.6	1237.9	1257.0	1276.2	1285.7	1333.8	1382.4	1431.7	1481.8	1532.7	1637.2
40.0	25.3	267.25	$\overline{V}$	12.001	12.628	13.247	13.862	14.166	15.688	17.198	18.702	20.20	21.70	24.69
			h_g	1216.9	1236.5	1255.9	1275.2	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	1637.0
50.0	35.3	281.01	$\overline{V}$	9.557	10.065	10.567	11.062	11.309	12.532	13.744	14.950	16.152	17.352	19.747
			h_g	1215.2	1235.1	1254.7	1274.2	1283.9	1332.5	1381.4	1430.9	1481.1	1532.1	1636.8
60.0	45.3	292.71	$\overline{V}$	7.927	8.357	8.779	9.196	9.403	10.427	11.441	12.449	13.452	14.454	16.451
			h_g	1213.4	1233.6	1253.5	1273.2	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	1636.6
70.0	55.3	302.92	$\overline{V}$	6.762	7.136	7.502	7.863	8.041	8.924	9.796	10.662	11.524	12.383	14.097
			h_g	1211.5	1232.1	1252.3	1272.2	1282.0	1331.1	1380.4	1430.1	1480.5	1531.6	1636.3
80.0	65.3	312.03	$\overline{V}$	5.868	6.220	6.544	6.862	7.020	7.797	8.562	9.322	10.077	10.830	12.332
			h_g	1209.7	1230.7	1251.1	1271.1	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	1636.2
90.0	75.3	320.27	$\overline{V}$	5.208	5.508	5.799	6.084	6.225	6.920	7.603	8.279	8.952	9.623	10.959
			h_g	1207.7	1229.1	1249.8	1270.1	1280.1	1329.8	1379.4	1429.3	1479.8	1531.0	1635.9
100.0	85.3	327.81	$\overline{V}$	4.663	4.937	5.202	5.462	5.589	6.218	6.835	7.446	8.052	8.656	9.860
			h_g	1205.7	1227.6	1248.6	1269.0	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1635.7
120.0	105.3	341.25	$\overline{V}$	3.844	4.081	4.307	4.527	4.636	5.165	5.683	6.195	6.702	7.207	8.212
			h_g	1201.6	1224.4	1246.0	1266.9	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	1635.3
140.0	125.3	353.02	$\overline{V}$	3.258	3.468	3.667	3.860	3.954	4.413	4.861	5.301	5.738	6.172	7.035
			h_g	1197.3	1221.1	1243.3	1264.7	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	1634.9
160.0	145.3	363.53	$\overline{V}$	...	3.008	3.187	3.359	3.443	3.849	4.244	4.631	5.015	5.396	6.152
			h_g	...	1217.6	1240.6	1262.4	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	1634.5
180.0	165.3	373.06	$\overline{V}$	...	2.649	2.813	2.969	3.044	3.411	3.764	4.110	4.452	4.792	5.466
			h_g	...	1214.0	1237.8	1260.2	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	1634.1
200.0	185.3	381.79	$\overline{V}$	...	2.361	2.513	2.656	2.726	3.060	3.380	3.693	4.002	4.309	4.917
			h_g	...	1210.3	1234.9	1257.8	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1633.7
220.0	205.3	389.86	$\overline{V}$	...	2.125	2.267	2.400	2.465	2.772	3.066	3.352	3.634	3.913	4.467
			h_g	...	1206.5	1231.9	1255.4	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5	1633.3
240.0	225.3	397.37	$\overline{V}$	...	1.9276	2.062	2.187	2.247	2.533	2.804	3.068	3.327	3.584	4.093
			h_g	...	1202.5	1228.8	1253.0	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9	1632.9
260.0	245.3	404.42	$\overline{V}$	...	...	1.8882	2.006	2.063	2.330	2.582	2.827	3.067	3.305	3.776
			h_g	...	...	1225.7	1250.5	1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	1632.5
280.0	265.3	411.05	$\overline{V}$	...	...	1.7388	1.8512	1.9047	2.156	2.392	2.621	2.845	3.066	3.504
			h_g	...	...	1222.4	1247.9	1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	1632.1
300.0	285.3	417.33	$\overline{V}$	...	...	1.6090	1.7165	1.7675	2.005	2.227	2.442	2.652	2.859	3.269
			h_g	...	...	1219.1	1245.3	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1631.7
320.0	305.3	423.29	$\overline{V}$	...	...	1.4950	1.5985	1.6472	1.8734	2.083	2.285	2.483	2.678	3.063
			h_g	...	...	1215.6	1242.6	1255.2	1313.2	1367.2	1419.8	1472.1	1524.7	1631.3
340.0	325.3	428.97	$\overline{V}$	...	...	1.3941	1.4941	1.5410	1.7569	1.9562	2.147	2.334	2.518	2.881
			h_g	...	...	1212.1	1239.9	1252.8	1311.6	1366.1	1419.0	1471.5	1524.1	1630.9
360.0	345.3	434.40	$\overline{V}$	...	...	1.3041	1.4012	1.4464	1.6533	1.8431	2.025	2.202	2.376	2.719
			h_g	...	...	1208.4	1237.1	1250.3	1310.1	1365.0	1418.1	1470.8	1523.5	1630.5

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PROPERTIES OF SUPERHEATED STEAM

V = specific volume, cubic feet per pound
h_g = total heat of steam, Btu per pound

Pressure, Lbs. per Sq. In.		Sat. Temp.		Total Temperature - Degrees Fahrenheit (t)										
Absolute P	Gauge P	t		500°	540°	600°	640°	660°	700°	740°	800°	900°	1000°	1200°
380.0	365.3	439.60	V	1.3616	1.4444	1.5605	1.6345	1.6707	1.7419	1.8118	1.9149	2.083	2.249	2.575
			h_g	1247.7	1273.1	1308.5	1331.0	1342.0	1363.8	1385.3	1417.3	1470.1	1523.0	1630.0
400.0	385.3	444.59	V	1.2851	1.3652	1.4770	1.5480	1.5827	1.6508	1.7177	1.8161	1.9767	2.134	2.445
			h_g	1245.1	1271.0	1306.9	1329.6	1340.8	1362.7	1384.3	1416.4	1469.4	1522.4	1629.6
420.0	405.3	449.39	V	1.2158	1.2935	1.4014	1.4697	1.5030	1.5684	1.6324	1.7267	1.8802	2.031	2.327
			h_g	1242.5	1268.9	1305.3	1328.3	1339.5	1361.6	1383.3	1415.5	1468.7	1521.9	1629.2
440.0	425.3	454.02	V	1.1526	1.2282	1.3327	1.3984	1.4306	1.4934	1.5549	1.6454	1.7925	1.9368	2.220
			h_g	1239.8	1266.7	1303.6	1326.9	1338.2	1360.4	1382.3	1414.7	1468.1	1521.3	1628.8
460.0	445.3	458.50	V	1.0948	1.1685	1.2698	1.3334	1.3644	1.4250	1.4842	1.5711	1.7124	1.8508	2.122
			h_g	1237.0	1264.5	1302.0	1325.4	1336.9	1359.3	1381.3	1413.8	1467.4	1520.7	1628.4
480.0	465.3	462.82	V	1.0417	1.1138	1.2122	1.2737	1.3038	1.3622	1.4193	1.5031	1.6390	1.7720	2.033
			h_g	1234.2	1262.3	1300.3	1324.0	1335.6	1358.2	1380.3	1412.9	1466.7	1520.2	1628.0
500.0	485.3	467.01	V	0.9927	1.0633	1.1591	1.2188	1.2478	1.3044	1.3596	1.4405	1.5715	1.6996	1.9504
			h_g	1231.3	1260.0	1298.6	1322.6	1334.2	1357.0	1379.3	1412.1	1466.0	1519.6	1627.6
520.0	505.3	471.07	V	0.9473	1.0166	1.1101	1.1681	1.1962	1.2511	1.3045	1.3826	1.5091	1.6326	1.8743
			h_g	1228.3	1257.7	1296.9	1321.1	1332.9	1355.8	1378.2	1411.2	1465.3	1519.0	1627.2
540.0	525.3	475.01	V	0.9052	0.9733	1.0646	1.1211	1.1485	1.2017	1.2535	1.3291	1.4514	1.5707	1.8039
			h_g	1225.3	1255.4	1295.2	1319.7	1331.5	1354.6	1377.2	1410.3	1464.6	1518.5	1626.8
560.0	545.3	478.85	V	0.8659	0.9330	1.0224	1.0775	1.1041	1.1558	1.2060	1.2794	1.3978	1.5132	1.7385
			h_g	1222.2	1253.0	1293.4	1318.2	1330.2	1353.5	1376.1	1409.4	1463.9	1517.9	1626.4
580.0	565.3	482.58	V	0.8291	0.8954	0.9830	1.0368	1.0627	1.1131	1.1619	1.2331	1.3479	1.4596	1.6776
			h_g	1219.0	1250.5	1291.7	1316.7	1328.8	1352.3	1375.1	1408.6	1463.2	1517.3	1626.0
600.0	585.3	486.21	V	0.7947	0.8602	0.9463	0.9968	1.0241	1.0732	1.1207	1.1899	1.3013	1.4096	1.6208
			h_g	1215.7	1248.1	1289.9	1315.2	1327.4	1351.1	1374.0	1407.7	1462.5	1516.7	1625.5
620.0	605.3	489.75	V	0.7624	0.8272	0.9118	0.9633	0.9880	1.0358	1.0821	1.1494	1.2577	1.3628	1.5676
			h_g	1212.4	1245.5	1288.1	1313.7	1326.0	1349.9	1373.0	1406.8	1461.8	1516.2	1625.1
640.0	625.3	493.21	V	0.7319	0.7963	0.8795	0.9299	0.9541	1.0008	1.0459	1.1115	1.2168	1.3190	1.5178
			h_g	1209.0	1243.0	1286.2	1312.2	1324.6	1348.6	1371.9	1405.9	1461.1	1515.6	1624.7
660.0	645.3	496.58	V	0.7032	0.7670	0.8491	0.8985	0.9222	0.9679	1.0119	1.0759	1.1784	1.2778	1.4709
			h_g	1205.4	1240.4	1284.4	1310.6	1323.2	1347.4	1370.8	1405.0	1460.4	1515.0	1624.3
680.0	665.3	499.88	V	0.6759	0.7395	0.8203	0.8690	0.8922	0.9369	0.9800	1.0424	1.1423	1.2390	1.4269
			h_g	1201.8	1237.7	1282.5	1309.1	1321.7	1346.2	1369.8	1404.1	1459.7	1514.5	1623.9
700.0	685.3	503.10	V	...	0.7134	0.7934	0.8411	0.8639	0.9077	0.9498	1.0108	1.1082	1.2024	1.3853
			h_g	...	1235.0	1280.6	1307.5	1320.3	1345.0	1368.7	1403.2	1459.0	1513.9	1623.5
750.0	735.3	510.86	V	...	0.6540	0.7319	0.7778	0.7996	0.8414	0.8813	0.9391	1.0310	1.1196	1.2912
			h_g	...	1227.9	1275.7	1303.5	1316.6	1341.8	1366.0	1400.9	1457.2	1512.4	1622.4
800.0	785.3	518.23	V	...	0.6015	0.6779	0.7223	0.7433	0.7833	0.8215	0.8763	0.9633	1.0470	1.2088
			h_g	...	1220.5	1270.7	1299.4	1312.9	1338.6	1363.2	1398.6	1455.4	1511.0	1621.4
850.0	835.3	525.26	V	...	0.5546	0.6301	0.6732	0.6934	0.7320	0.7685	0.8209	0.9037	0.9830	1.1360
			h_g	...	1212.7	1265.5	1295.2	1309.0	1335.4	1360.4	1396.3	1453.6	1509.5	1620.4
900.0	885.3	531.98	V	...	0.5124	0.5873	0.6294	0.6491	0.6863	0.7215	0.7716	0.8506	0.9262	1.0714
			h_g	...	1204.4	1260.1	1290.9	1305.1	1332.1	1357.5	1393.9	1451.8	1508.1	1619.3
950.0	935.3	538.42	V	...	0.4740	0.5489	0.5901	0.6092	0.6453	0.6793	0.7275	0.8031	0.8753	1.0136
			h_g	...	1195.5	1254.6	1286.4	1301.1	1328.7	1354.7	1391.6	1450.0	1506.6	1618.3
1000.0	985.3	544.61	V	...	...	0.5140	0.5546	0.5733	0.6084	0.6413	0.6878	0.7604	0.8294	0.9615
			h_g	...	...	1248.8	1281.9	1297.0	1325.3	1351.7	1389.2	1448.2	1505.1	1617.3

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PROPERTIES OF SUPERHEATED STEAM

V = specific volume, cubic feet per pound
h_g = total heat of steam, Btu per pound

Pressure, Lbs. per Sq. In.		Sat. Temp.		Total Temperature - Degrees Fahrenheit (t)										
Absolute P	Gauge P	t		660°	700°	740°	760°	780°	800°	860°	900°	1000°	1100°	1200°
1100.0	1085.3	556.31	V	0.5110	0.5445	0.5755	0.5904	0.6049	0.6191	0.6601	0.6866	0.7503	0.8117	0.8716
			h_g	1288.5	1318.3	1345.8	1358.9	1371.7	1384.3	1420.8	1444.5	1502.2	1558.8	1615.2
1200.0	1185.3	567.22	V	0.4586	0.4909	0.5206	0.5347	0.5484	0.5617	0.6003	0.6250	0.6843	0.7412	0.7967
			h_g	1279.6	1311.0	1339.6	1353.2	1366.4	1379.3	1416.7	1440.7	1499.2	1556.4	1613.1
1300.0	1285.3	577.46	V	0.4139	0.4454	0.4739	0.4874	0.5004	0.5131	0.5496	0.5728	0.6284	0.6816	0.7333
			h_g	1270.2	1303.4	1333.3	1347.3	1361.0	1374.3	1412.5	1437.0	1496.2	1553.9	1611.0
1400.0	1385.3	587.10	V	0.3753	0.4062	0.4338	0.4468	0.4593	0.4714	0.5061	0.5281	0.5805	0.6305	0.6789
			h_g	1260.3	1295.5	1326.7	1341.3	1355.4	1369.1	1408.2	1433.1	1493.2	1551.4	1608.9
1500.0	1485.3	596.23	V	0.3413	0.3719	0.3989	0.4114	0.4235	0.4352	0.4684	0.4893	0.5390	0.5862	0.6318
			h_g	1249.8	1287.2	1320.0	1335.2	1349.7	1363.8	1403.9	1429.3	1490.1	1548.9	1606.8
1600.0	1585.3	604.90	V	0.3112	0.3417	0.3682	0.3804	0.3921	0.4034	0.4353	0.4553	0.5027	0.5474	0.5906
			h_g	1238.7	1278.7	1313.0	1328.8	1343.9	1358.4	1399.5	1425.3	1487.0	1546.4	1604.6
1700.0	1685.3	613.15	V	0.2842	0.3148	0.3410	0.3529	0.3643	0.3753	0.4061	0.4253	0.4706	0.5132	0.5542
			h_g	1226.8	1269.7	1305.8	1322.3	1337.9	1352.9	1395.0	1421.4	1484.0	1543.8	1602.5
1800.0	1785.3	621.03	V	0.2597	0.2907	0.3166	0.3284	0.3395	0.3502	0.3801	0.3986	0.4421	0.4828	0.5218
			h_g	1214.0	1260.3	1298.4	1315.5	1331.8	1347.2	1390.4	1417.4	1480.8	1541.3	1600.4
1900.0	1885.3	628.58	V	0.2371	0.2688	0.2947	0.3063	0.3173	0.3277	0.3568	0.3747	0.4165	0.4556	0.4929
			h_g	1200.2	1250.4	1290.6	1308.6	1325.4	1341.5	1385.8	1413.3	1477.7	1538.8	1598.2
2000.0	1985.3	635.82	V	0.2161	0.2489	0.2748	0.2863	0.2972	0.3074	0.3358	0.3532	0.3935	0.4311	0.4668
			h_g	1184.9	1240.0	1282.6	1301.4	1319.0	1335.5	1381.2	1409.2	1474.5	1536.2	1596.1
2100.0	2085.3	642.77	V	0.1962	0.2306	0.2567	0.2682	0.2789	0.2890	0.3167	0.3337	0.3727	0.4089	0.4433
			h_g	1167.7	1229.0	1274.3	1294.0	1312.3	1329.5	1376.4	1405.0	1471.4	1533.5	1593.9
2200.0	2185.3	649.46	V	0.1768	0.2105	0.2400	0.2514	0.2621	0.2721	0.2994	0.3159	0.3536	0.3887	0.4218
			h_g	1147.8	1217.4	1265.7	1286.3	1305.4	1323.3	1371.5	1400.8	1468.2	1531.1	1591.8
2300.0	2285.3	655.91	V	0.1575	0.1978	0.2247	0.2362	0.2468	0.2567	0.2835	0.2997	0.3365	0.3703	0.4023
			h_g	1123.8	1204.9	1256.7	1278.4	1298.4	1316.9	1366.6	1396.5	1464.9	1528.5	1589.6
2400.0	2385.3	662.12	V	...	0.1828	0.2105	0.2221	0.2327	0.2425	0.2689	0.2848	0.3207	0.3534	0.3843
			h_g	...	1191.5	1247.3	1270.2	1291.1	1310.3	1361.6	1392.2	1461.7	1525.9	1587.4
2500.0	2485.3	668.13	V	...	0.1686	0.1973	0.2090	0.2196	0.2294	0.2555	0.2710	0.3061	0.3379	0.3678
			h_g	...	1176.8	1237.6	1261.8	1283.6	1303.6	1356.5	1387.8	1458.4	1523.2	1585.3
2600.0	2585.3	673.94	V	...	0.1549	0.1849	0.1967	0.2074	0.2172	0.2431	0.2584	0.2926	0.3236	0.3526
			h_g	...	1160.6	1227.3	1252.9	1275.8	1296.8	1351.4	1383.4	1455.1	1520.6	1583.1
2700.0	2685.3	679.55	V	...	0.1415	0.1732	0.1853	0.1960	0.2059	0.2315	0.2468	0.2801	0.3103	0.3385
			h_g	...	1142.5	1216.5	1243.8	1267.9	1289.7	1346.1	1378.9	1451.8	1518.0	1580.9
2800.0	2785.3	684.99	V	...	0.1281	0.1622	0.1745	0.1854	0.1953	0.2208	0.2356	0.2685	0.2979	0.3254
			h_g	...	1121.4	1205.1	1234.2	1259.6	1282.4	1340.8	1374.3	1448.5	1515.4	1578.7
2900.0	2885.3	690.26	V	...	0.1143	0.1517	0.1644	0.1754	0.1853	0.2108	0.2254	0.2577	0.2864	0.3132
			h_g	...	1095.9	1193.0	1224.3	1251.1	1274.9	1335.3	1369.7	1445.1	1512.7	1576.5
3000.0	2985.3	695.36	V	...	0.0984	0.1416	0.1548	0.1660	0.1760	0.2014	0.2159	0.2476	0.2757	0.3018
			h_g	...	1060.7	1180.1	1213.8	1242.2	1267.2	1329.7	1365.0	1441.8	1510.0	1574.3
3100.0	3085.3	700.31	V	...	...	0.1320	0.1456	0.1571	0.1672	0.1926	0.2070	0.2382	0.2657	0.2911
			h_g	...	...	1166.2	1202.9	1233.0	1259.3	1324.1	1360.3	1438.4	1507.4	1572.1
3200.0	3185.3	705.11	V	...	...	0.1226	0.1369	0.1486	0.1585	0.1843	0.1986	0.2293	0.2563	0.2811
			h_g	...	...	1151.1	1191.4	1223.5	1251.1	1318.3	1355.5	1434.9	1504.7	1569.9
3206.2	3191.5	705.40	V	...	...	0.1220	0.1363	0.1480	0.1583	0.1838	0.1981	0.2288	0.2557	0.2806
			h_g	...	...	1150.2	1190.6	1222.9	1250.5	1317.9	1355.2	1434.7	1504.5	1569.8

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PHYSICAL CONSTANTS OF VARIOUS FLUIDS

Fluid	Formula	Molecular Weight	Boiling Point °F. at 14.696 psia	Specific Gravity	
				Liquid 60/60 °F.	Gas
Acetic Acid	$HC_2H_3O_2$	60.05	245	1.05	
Acetone	C_3H_6O	58.08	133	.79	2.01
Air	N_2O_2	28.97	-317	.86	1.0
Alcohol, Ethyl	C_2H_6O	46.07	173	.79	1.59
Alcohol, Methyl	CH_4O	32.04	148	.79	1.11
Ammonia	NH_3	17.03	-28	.62	59
Ammonium Chloride*	NH_4Cl			1.07	
Ammonium Hydroxide*	NH_4OH			.91	
Ammonium Sulfate*	$(NH_4)_2SO_4$			1.15	
Aniline	C_6H_7N	93.12	365	1.02	
Argon	A	39.94	-302	1.65	1.38
Beer				1.01	
Bromine	Br_2	159.84	138	2.93	5.52
Calcium Chloride*	$CaCl_2$			1.23	
Carbon Dioxide	CO_2	44.01	-109		1.52
Carbon Disulfide	CS_2	76.1	115	1.29	2.63
Carbon Monoxide	CO	28.01	-314	.80	.97
Carbon Tetrachloride	CCl_4	153.84	170	1.59	5.31
Chlorine	Cl_2	70.91	-30	1.42	2.45
Chromic Acid	H_2CrO_4	118.03		1.21	
Citric Acid	$C_6H_8O_7$	192.12		1.54	
Copper Sulfate*	$CuSO_4$			1.17	
Ether	$(C_2H_5)_2O$	74.12	34	.74	2.55
Ferric Chloride*	$FeCl_3$			1.23	
Fluorine	F_2	38.00	-305	1.11	1.31
Formaldehyde	H_2CO	30.03	-6	.82	1.08
Formic Acid	HCO_2H	46.03	214	1.23	
Furfural	$C_5H_4O_2$	96.08	324	1.16	
Glycerine	$C_3H_8O_3$	92.09	554	1.26	
Glycol	$C_2H_6O_2$	62.07	387	1.11	
Helium	He	4.003	-454	.18	.14
Hydrochloric Acid	HCl	36.47	-115	1.64	

* Aqueous Solution - 25% by weight of compound.

PHYSICAL CONSTANTS OF VARIOUS FLUIDS

Fluid	Formula	Molecular Weight	Boiling Point °F. at 14.696 psia	Specific Gravity	
				Liquid 60/60 °F.	Gas
Hydrofluoric Acid	HF	20.01	66	.92	
Hydrogen	H_2	2.016	-422	.07	.07
Hydrogen Chloride	HCl	36.47	-115		1.26
Hydrogen Sulfide	H_2S	34.07	-76	.79	1.17
Isopropyl Alcohol	C_3H_8O	60.09	180	.78	2.08
Linseed Oil			538	.93	
Magnesium Chloride*	$MgCl_2$			1.22	
Mercury	Hg	200.61	670	13.6	6.93
Methyl Bromide	CH_3Br	94.95	38	1.73	3.27
Methyl Chloride	CH_3Cl	50.49	-11	.99	1.74
Naphthalene	$C_{10}H_8$	128.16	424	1.14	4.43
Nitric Acid	HNO_3	63.02	187	1.5	
Nitrogen	N_2	28.02	-320	.81	.97
Oil, Vegetable				.91 - .94	
Oxygen	O_2	32	-297	1.14	1.105
Phosgene	$COCl_2$	98.92	47	1.39	3.42
Phosphoric Acid	H_3PO_4	98.00	415	1.83	
Potassium Carbonate*	K_2CO_3			1.24	
Potassium Chloride*	KCl			1.16	
Potassium Hydroxide*	KOH			1.24	
Sodium Chloride*	NaCl			1.19	
Sodium Hydroxide*	NaOH			1.27	
Sodium Sulfate*	Na_2SO_4			1.24	
Sodium Thiosulfate*	$Na_2S_2O_3$			1.23	
Starch	$(C_6H_{10}O_5)_x$			1.50	
Sugar Solutions*	$C_{12}H_{22}O_{11}$			1.10	
Sulfuric Acid	H_2SO_4	98.08	626	1.83	
Sulfur Dioxide	SO_2	64.6	14	1.39	2.21
Turpentine			320	.87	
Water	H_2O	18.016	212	1.00	.62
Zinc Chloride*	$ZnCl_2$			1.24	
Zinc Sulfate*	$ZnSO_4$			1.31	

* Aqueous Solution - 25% by weight of compound.

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PHYSICAL CONSTANTS OF COMMON HYDROCARBONS

No.	Compound	Formula	Molecular Weight	Boiling Point °F., 14.696 psi, abs.	Specific Gravity Liquid 60/60 °F.	Specific Gravity Gas
1	Methane	CH_4	16.042	-258.69	...	0.555
2	Ethane	C_2H_6	30.068	-127.53	....	1.046
3	Propane	C_3H_8	44.094	-43.73	0.5077	1.547
4	n-Butane	C_4H_{10}	58.120	31.10	0.5844	2.071
5	2-Methylpropane (isobutane)	C_4H_{10}	58.120	10.89	0.5631	2.067
6	n-Pentane	C_5H_{12}	72.146	96.933	0.63116	2.4906
7	2-Methylbutane (isopentane)	C_5H_{12}	72.146	82.134	0.62476	2.4906
8	2,2-Dimethylpropane (neopentane)	C_5H_{12}	72.146	49.105	0.601	2.4906
9	n-Hexane	C_6H_{14}	86.172	155.736	0.66405	2.9749
10	2-Methylpentane	C_6H_{14}	86.172	140.488	0.65790	2.9749
11	3-Methylpentane	C_6H_{14}	86.172	145.908	0.66902	2.9749
12	2.2-Dimethylbutane (neohexane)	C_6H_{14}	86.172	121.534	0.65399	2.9749
13	2,3-Dimethylbutane	C_6H_{14}	86.172	136.378	0.66639	2.9749
14	n-Heptane	C_7H_{16}	100.198	209.169	0.68819	3.4591
15	2-Methylhexane	C_7H_{16}	100.198	194.094	0.68299	3.4591
16	3-Methylhexane	C_7H_{16}	100.198	197.330	0.69151	3.4591
17	3-Ethylpentane	C_7H_{16}	100.198	200.251	0.70257	3.4591
18	2.2-Dimethylpentane	C_7H_{16}	100.198	174.560	0.67833	3.4591
19	2.4-Dimethylpentane	C_7H_{16}	100.198	176.900	0.67723	3.4591
20	3,3-Dimethylpentane	C_7H_{16}	100.198	186.917	0.69767	3.4591
21	2.2,3-Trimethylbutane (triptane)	C_7H_{16}	100.198	177.584	0.69454	3.4591
22	n-Octane	C_8H_{18}	114.224	258.197	0.70677	3.9432
23	2,5-Dimethylhexane (diisobutyl)	C_8H_{18}	114.224	228.385	0.69795	3.9432
24	2.2.4-Trimethylpentane ("isooctane")	C_8H_{18}	114.224	210.628	0.69625	3.9432
25	n-Nonane	C_9H_{20}	128.250	303.436	0.72171	4.4275
26	n-Decane	$C_{10}H_{22}$	142.276	345.2	0.73413	4.9118
27	Cyclopentane	C_5H_{10}	70.130	120.672	0.75048	2.4211
28	Methylcyclopentane	C_6H_{12}	84.156	161.262	0.75354	2.9053
29	Cyclohexane	C_6H_{12}	84.156	177.328	0.78344	2.9053
30	Methylcyclohexane	C_7H_{14}	98.182	213.681	0.77398	3.3896
31	Ethene (ethylene)	C_2H_4	28.052	-154.68	...	0.9684
32	Propene	C_3H_6	42.078	-53.86	0.5218	1.4526
33	1-Butene	C_4H_8	56.104	20.73	0.6011	1.9368
34	cis-2-Butene	C_4H_8	56.104	38.70	0.6272	1.9368
35	trans-2-Butene	C_4H_8	56.104	33.58	0.6100	1.9368
36	2-Methylpropene (isobutene)	C_4H_8	56.104	19.68	0.6002	1.9368
37	1-Pentene	C_5H_{10}	70.130	85.95	0.6461	2.4210
38	1,2-Butadiene	C_4H_6	54.088	50.5	0.658	1.8673
39	1,3-Butadiene	C_4H_6	54.088	24.06	0.6272	1.8673
40	2-Methyl-1.3-butadiene (isoprene)	C_5H_8	68.114	93.34	0.6851	2.3515
41	Ethyne (acetylene)	C_2H_2	26.036	-119	0.615	0.8988
42	Benzene	C_6H_6	78.108	176.185	0.88458	2.6965
43	Toluene	C_7H_8	92.134	231.121	0.87190	3.1808
44	Ethylbenzene	C_8H_{10}	106.160	277.137	0.87175	3.6650
45	1,2-Dimethylbenzene (o-xylene)	C_8H_{10}	106.160	291.95	0.88482	3.6650
46	1,3-Dimethylbenzene (m-xylene)	C_8H_{10}	106.160	282.38	0.86880	3.6650
47	1,4-Dimethylbenzene (p-xylene)	C_8H_{10}	106.160	281.03	0.86576	3.6650
48	Styrene (Phenyl Ethylene)	C_8H_8	104.144	293.4	0.91112	3.5954
49	Isopropylbenzene (cumene)	C_9H_{12}	120.186	306.31	0.86642	4.1492

Properties of Cryogenic Fluids

Gas	He	H_2	CH_4	N_2	O_2	F_2
Density, NTP* lb./cu. ft.	.0103	.0052	.0415	.0724	.0827	.0982
Boiling Point 1 atm. °F.	-452.1	-423.2	-258.7	-320.4	-297.4	-306.6
Vapor Density at B.P. lb./cu. ft.	1.06	.084	.111	.288	.296	---
Liquid Density at B.P. lb./cu.ft.	7.62	4.37	26.46	50.41	71.27	94.2
Heat of Vaporization at B.P. Btu/lb.	8.8	193	219.2	85.2	91.7	74.1
Critical Temperature °F.	-450.3	-400.3	-116.5	-232.8	-181.1	-200.2
Critical Pressure, psia	33.2	187.7	673.1	492.3	736.9	808.3

NTP = Normal Temperature and Pressure, 70°F. and 14.696 psia.

Area Equivalents

Convert From \ Convert To	Square Meters	Square Inches	Square Feet	Square Yards	Square Miles	Square Kilometers
Square Meters	1	1549.99	10.7639	1.19599	3.861×10^{-7}	1×10^{-6}
Square Inches	0.0006452	1	6.944×10^{-3}	7.716×10^{-4}	2.491×10^{-10}	6.452×10^{-10}
Square Feet	0.0929	144	1	0.11111	3.587×10^{-8}	9.29×10^{-8}
Square Yards	0.83613	1296	9	1	3.228×10^{-7}	8.361×10^{-7}
Square Miles	2,589,999	-	27,878,400	3,097,600	1	2.59
Square Kilometers	1,000,000	-	10,763,867	1,195,985	0.3861	1

1 square meter = 100 square decimeters = 10,000 square centimeters

1 square millimeter = 0.01 square centimeter = 0.00155 square inches

Volume Equivalents

Convert From \ Convert To	Cubic Decimeters	Cubic Inches	Cubic Feet	U. S. Quart	U. S. Gallon	Imperial Gallon	U. S.* Barrel
Cubic Decimeters	1	61.0234	0.03531	1.05668	0.264178	0.220083	0.00629
Cubic Inches	0.01639	1	5.787×10^{-4}	0.01732	0.004329	0.003606	0.000103
Cubic Feet	28.317	1728	1	29.9221	7.48055	6.22888	0.1781
U.S. Quart	0.94636	57.75	0.03342	1	0.25	0.2082	0.00595
U.S. Gallon	3.78543	231	0.13368	4	1	0.833	0.02381
Imperial Gallon	4.54374	277.274	0.16054	4.80128	1.20032	1	0.02877
U.S.* Barrel	158.98	9,702	5.6146	168	42	34.973	1

1 cubic meter = 1000 cubic decimeters = 1,000,000 cubic centimeters
1 liter = 10 deciliters = 100 centiliters = 1000 milliliters = 1000 cubic centimeters
1 liter = 1 cubic decimeter.
* Petroleum

Volume Rate Equivalents

Convert From \ Convert To	Liters Per Minute	Cubic Meters Per Hour	Cubic Feet Per Hour	Imperial Gallon Per Minute	U.S. Gallon Per Minute	U.S. Barrel Per Day
Liters Per Minute	1	0.06	2.1189	0.21997	0.264178	9.057
Cubic Meters Per Hour	16.667	1	35.314	3.667	4.403	151
Cubic Feet Per Hour	0.4719	0.028317	1	0.1038	0.1247	4.2746
Imperial Gallon Per Minute	4.546	0.2727	9.6325	1	1.20032	41.1
U.S. Gallon Per Minute	3.785	0.2273	8.0208	0.8326	1	34.28
U.S. Barrel Per Day	0.1104	0.006624	0.23394	0.02428	0.02917	1

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Pressure Equivalents

Convert From \ Convert To	Kg. Per Sq. Cm.	Lb. Per Sq. In.	Atm.	Meters of Hg.	In. of Hg.	Meters of Water	In. of Water	Ft. of Water
Kg. Per Sq. Cm.	1	14.22	0.9678	0.7355	28.96	10.01	394.05	32.84
Lb. Per Sq. In.	0.07031	1	0.06804	0.05171	2.036	0.7037	27.7	2.309
Atm.	1.0332	14.696	1	0.76	29.92	10.34	407.14	33.93
Meters of Hg.	1.3596	19.34	1.316	1	39.37	13.61	535.7	44.64
In. of Hg.	0.03453	0.4912	0.03342	0.0254	1	0.3456	13.61	1.134
Meters of Water	0.09991	1.421	0.0967	0.07349	2.893	1	39.37	3.281
In. of Water	0.002538	0.0361	0.002456	0.001867	0.07349	0.0254	1	0.0833
Ft. of Water	0.03045	0.4332	0.02947	0.0224	0.8819	0.3048	12	1

1 ounce/sq.inch = 0.0625 lbs./sq.inch

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TEMPERATURE CONVERSION FORMULAS

To Convert From	To	Substitute in Formula
Degrees Centigrade	Degrees Fahrenheit	(°C x 9/5) + 32
Degrees Centigrade	Degrees Kelvin	(°C - 273.16)
Degrees Fahrenheit	Degrees Centigrade	(°F - 32) x 5/9
Degrees Fahrenheit	Degrees Rankine	(°F - 459.69)

Temperature Interpolation Factors

°C.	Degree	°F.
0.56	1	1.8
1.11	2	3.6
1.67	3	5.4
2.22	4	7.2
2.78	5	9.0
3.33	6	10.8
3.89	7	12.6
4.44	8	14.4
5.00	9	16.2

Equivalents of Degrees API, Degrees Baumé, Specific Gravity, Weight Density, and Pounds Per Gallon at 60F/60F

Degrees on API or Baumé Scale	Values for API Scale — Oil			Values for Baumé Scale — Liquids Lighter Than Water			Values for Baumé Scale — Liquids Heavier Than Water		
	Specific Gravity S	Weight Density, Lb/Ft³ ρ	Pounds per Gallon	Specific Gravity S	Weight Density, Lb/Ft³ ρ	Pounds per Gallon	Specific Gravity S	Weight Density, Lb/Ft³ ρ	Pounds per Gallon
0	...	...	...	...	...	...	1.0000	62.36	8.337
2	...	...	...	...	...	...	1.0140	63.24	8.454
4	...	...	...	...	...	...	1.0284	64.14	8.574
6	...	...	...	...	...	...	1.0432	65.06	8.697
8	...	...	...	...	...	...	1.0584	66.01	8.824
10	1.0000	62.36	8.337	1.0000	62.36	8.337	1.0741	66.99	8.955
12	0.9861	61.50	8.221	0.9859	61.49	8.219	1.0902	67.99	9.089
14	0.9725	60.65	8.108	0.9722	60.63	8.105	1.1069	69.03	9.228
16	0.9593	59.83	7.998	0.9589	59.80	7.994	1.1240	70.10	9.371
18	0.9465	59.03	7.891	0.9459	58.99	7.886	1.1417	71.20	9.518
20	0.9340	58.25	7.787	0.9333	58.20	7.781	1.1600	72.34	9.671
22	0.9218	57.87	7.736	0.9211	57.44	7.679	1.1789	73.52	9.828
24	0.9100	56.75	7.587	0.9091	56.70	7.579	1.1983	74.73	9.990
26	0.8984	56.03	7.490	0.8974	55.97	7.482	1.2185	75.99	10.159
28	0.8871	55.32	7.396	0.8861	55.26	7.387	1.2393	77.29	10.332
30	0.8762	54.64	7.305	0.8750	54.57	7.295	1.2609	78.64	10.512
32	0.8654	53.97	7.215	0.8642	53.90	7.205	1.2832	80.03	10.698
34	0.8550	53.32	7.128	0.8537	53.24	7.117	1.3063	81.47	10.891
36	0.8448	52.69	7.043	0.8434	52.60	7.031	1.3303	82.96	11.091
38	0.8348	52.06	6.960	0.8333	51.97	6.947	1.3551	84.51	11.297
40	0.8251	51.46	6.879	0.8235	51.36	6.865	1.3810	86.13	11.513
42	0.8155	50.86	6.799	0.8140	50.76	6.786	1.4078	87.80	11.737
44	0.8063	50.28	6.722	0.8046	50.18	6.708	1.4356	89.53	11.969
46	0.7972	49.72	6.646	0.7955	49.61	6.632	1.4646	91.34	12.210
48	0.7883	49.16	6.572	0.7865	49.05	6.557	1.4948	93.22	12.462
50	0.7796	48.62	6.499	0.7778	48.51	6.484	1.5263	95.19	12.725
52	0.7711	48.09	6.429	0.7692	47.97	6.413	1.5591	97.23	12.998
54	0.7628	47.57	6.359	0.7609	47.45	6.344	1.5934	99.37	13.284
56	0.7547	47.07	6.292	0.7527	46.94	6.275	1.6292	101.60	13.583
58	0.7467	46.57	6.225	0.7447	46.44	6.209	1.6667	103.94	13.895
60	0.7389	46.08	6.160	0.7368	45.95	6.143	1.7059	106.39	14.222
62	0.7313	45.61	6.097	0.7292	45.48	6.079	1.7470	108.95	14.565
64	0.7238	45.14	6.034	0.7216	45.00	6.016	1.7901	111.64	14.924
66	0.7165	44.68	5.973	0.7143	44.55	5.955	1.8354	114.46	15.302
68	0.7093	44.23	5.913	0.7071	44.10	5.895	1.8831	117.44	15.699
70	0.7022	43.79	5.854	0.7000	43.66	5.836	1.9333	120.57	16.118
72	0.6953	43.36	5.797	0.6931	43.22	5.778	...	...	...
74	0.6886	42.94	5.741	0.6863	42.80	5.722	...	...	...
76	0.6819	42.53	5.685	0.6796	42.38	5.666	...	...	...
78	0.6754	42.12	5.631	0.6731	41.98	5.612	...	...	...
80	0.6690	41.72	5.577	0.6667	41.58	5.558	...	...	...
82	0.6628	41.33	5.526	0.6604	41.19	5.506	...	...	...
84	0.6566	40.95	5.474	0.6542	40.80	5.454	...	...	...
86	0.6506	40.57	5.424	0.6482	40.42	5.404	...	...	...
88	0.6446	40.20	5.374	0.6422	40.05	5.354	...	...	...
90	0.6388	39.84	5.326	0.6364	39.69	5.306	...	...	...
92	0.6331	39.48	5.278	0.6306	39.33	5.257	...	...	...
94	0.6275	39.13	5.231	0.6250	38.98	5.211	...	...	...
96	0.6220	38.79	5.186	0.6195	38.63	5.165	...	...	...
98	0.6166	38.45	5.141	0.6140	38.29	5.119	...	...	...
100	0.6112	38.12	5.096	0.6087	37.96	5.075	...	...	...

Courtesy Crane Co., Reproduced from Technical Paper No. 410, "Flow of Fluids."

OTHER USEFUL CONVERSIONS

To Convert From	To	Multiply By
Cu. Ft. (Methane)	B.T.U.	1000 (approx.)
Cu. Ft. of Water	Lbs. of Water	62.4
Degrees	Radians	0.01745
Gals.	Lbs. of Water	8.336
Grams	Ounces	0.0352
Horsepower (mech.)	Ft. Lbs. per Min.	33,000
Horsepower (elec.)	Watts	746
Kg.	Lbs.	2.205
Kg. per Cu. Meter	Lbs. per Cu. Ft.	0.06243
Kilowatts	Horsepower	1.34
Lbs.	Kg.	0.4536
Lbs. of Air (14.7 psia and 60° F.)	Cu. Ft. of Air	13.1
Lbs. per Cu. Ft.	Kg. per Cu. Meter	16.0184
Lbs. per Hr. (Gas)	Std. Cu. Ft. per Hr.	$\frac{13.1}{\text{Specific Gravity}}$
Lbs. per Hr. (Water)	Gals. per Min.	0.002
Lbs. per Sec. (Gas)	Std. Cu. Ft. per Hr.	$\frac{46{,}160}{\text{Specific Gravity}}$
Radians	Degrees	57.3
Scfh Air	Scfh Propane	0.81
Scfh Air	Scfh Butane	0.71
Scfh Air	Scfh 0.6 Natural Gas	1.29

PREFIXES

Prefix	Meaning	Numerical Value
Micromicro-	One-trillionth	0.000,000,000,001 or 10^{-12}
Millimicro-	One-billionth	0.000,000,001 10^{-9}
Micro-	One-millionth	0.000,001 10^{-6}
Milli-	One-thousandth	0.001 10^{-3}
Centi-	One-hundredth	0.01 10^{-2}
Deci-	One-tenth	0.1 10^{-1}
Deka-	Ten	10 10
Hecto-	One hundred	100 10^{2}
Kilo-	One thousand	1,000 10^{3}
Mega-	One million	1,000,000 10^{6}

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GREEK ALPHABET

A	α	Alpha	N	ν	Nu
B	β	Beta	Ξ	ξ	Xi
Γ	γ	Gamma	O	o	Omicron
Δ	δ	Delta	Π	π	Pi
E	ϵ	Epsilon	P	ρ	Rho
Z	ζ	Zeta	Σ	σ	Sigma
H	η	Eta	T	τ	Tau
Θ	θ	Theta	Υ	υ	Upsilon
I	ι	Iota	Φ	ϕ	Phi
K	κ	Kappa	X	χ	Chi
Λ	λ	Lambda	Ψ	ψ	Psi
M	μ	Mu	Ω	ω	Omega

ABOUT THE AUTHORS

James B. Arant is a 1946 graduate of the University of Texas with a Bachelor of Science in Chemical Engineering. He was associated with Dow Chemical Company at Freeport, Texas, and Standard Oil and Gas Company in Tulsa, Oklahoma, before joining E. I. du Pont de Nemours in 1950. From 1950 to 1967 he was a Division Engineer at Du Pont's Sabine River Works in Orange, Texas. At that location he was involved in plant project instrument design and maintenance engineering covering a wide range of processes and operating conditions. In 1967 he moved to Du Pont's Engineering Department, Design Division, in Wilmington, Delaware, where he is a Senior Engineer responsible for instrument control design for major project facilities for several Du Pont operating departments. He also provides consultation on special control valve requirements for severe service and noise. Mr. Arant has published a number of papers and given talks on the subjects of control valve noise, turbine and compressor instrumentation and flow metering. He is a Senior Member of the Instrument Society of America and a member of the RP59 committee on control valve noise testing and prediction. He is also a Registered Professional Engineer in the State of Texas.

Ernest E. Allen is presently Manager of Valve Research with Fisher Controls, Marshalltown, Iowa. At Fisher, he has directed an extensive research study on valve noise. A graduate of Iowa State University, Mr. Allen has authored numerous technical articles on valve noise. He is a a member of the ISA Standards and Practice Committee on Valve Noise, SP-59.

Earl A. Bake is New Product Engineering Manager with the Flow Control Division of Rockwell International, Pittsburgh, Pennsylvania. His previous experience included four years as Research Project Engineer at Rockwell, developing experimental and analytical information on valve flow and cavitation and evolving methods for design to optimize flow performance. Prior to that he worked at Westinghouse Electric Corporation for twelve years as technical administrator of research and development on valves for nuclear power plants. Mr. Bake, who received his BSME from Case Western Reserve, is a member of ASME and has published an ASME paper on high velocity gas flow in reduced port valves.

Hans D. Baumann; P.E., is an independent Consultant on Automatic Control Systems and specifically on Control Valves and Regulators. Until recently he was Vice President of Technology, heading all US and worldwide engineering activities, including R & D for Masoneilan International, Inc. In prior assignments he served as Chief Engineer for Welland and Tuxhorn in Germany and Director of Engineering for Cashco, Inc. A Senior member of ISA, Mr. Baumann also belongs to ASME and holds over 50 US and foreign patents. He has authored or coauthored more than 30 technical articles in leading journals here and abroad and is Contributing Author to the "Instrument Engineers Handbook" by Liptak. In addition he serves as the sole US Representative to the International Standards Committee on Control Valves. Educated in Germany, Mr. Baumann has taken graduate courses at Western Reserve and Northeastern University.

Everett L. Dixson is employed by the Fisher Controls Company as a research engineer. At Fisher, he is responsible for the investigation and testing of such non-metallic materials as elastomers, plastics, adhesives, etc., with special emphasis on the practical application of these materials to service conditions in assemblies. Previously, Mr. Dixson served as a design and development engineer with Lennox Industries, Marshalltown, Iowa, and as a flight test engineer with the Boeing Co. of Seattle, Wash. A member of the American Society of Mechanical Engineers, Mr. Dixson holds a BS degree in Mechanical Engineering from Iowa State University.

Leslie R. Driskell is currently with Dravo Corporation, Chemical Plants Division, Pittsburgh, Pa. Here, he directs the activities of the instrument and electrical Engineering Departments. His previous professional associations include E. I. DuPont de Nemours

Co., Riggs Distler and Company, Purdue University and Seagram-Calvert, Inc. An ISA Fellow Member, Mr. Driskell is the author of numerous technical papers; in 1968, he won the Instrumentation Technology Award for an article published in that journal. Mr. Driskell holds a BS degree in Mechanical Engineering from the University of Louisville.

Matthew L. Freeman is Engineering Manager for the Power Systems Group of AMETEK, Inc., Hatfield, Pennsylvania. He is responsible for engineering off-gas nuclear and solar process systems involving products of several AMETEK divisions. His previous professional associations include Masoneilan International, Rockwell International, Honeywell, McEvoy and Atlantic Richfield Co. He is experienced in the design and application of many different types of valves and controls used in difficult service. He has been a leader in ISA and ASME Sections and is active on several ANSI code committees. A graduate of Mississippi State University; Mr. Freeman holds ten valve patents and is the author of numerous technical articles on control valves.

Donald J. Gaffney is currently manager of Engineering Codes and Standards with the Fisher Controls Company, Marshalltown, Iowa. He is responsible for product design and improvement, materials, Codes and Standards, Seismic and Structural Analysis. Mr. Gaffney has been associated with Fisher Controls Company since receiving his B.S. Degree in Mechanical Engineering from the University of Nebraska. He has been Chairman of the Control Valve Committee of the PMC Section of SAMA for several years.

William H. Hopf is presently Manager of Engineering of Irvington-Moore in Jacksonville, Florida, a division of United States Natural Resources Inc., Menlo Park, California. Previously, he was Chief Engineer of Jamesbury Corporation, with 10 years of valve design and application experience. Mr. Hopf holds a MS degree in Mechanical Engineering from Worcester Polytechnic Institute, Worcester, Massachusetts, is a member of ASME, and has been active on several valve standards committees in ANSI, API, MSS as well as the ISA.

Melville W. Hott, Jr., is President of the M. W. Hott Company, Cincinnati, Ohio. The M. W. Hott Company represents several prominent manufacturers of control valves, instrumentation hardware and process control devices. He was formerly Executive Vice President of Jordan Valve Division of Richards Industries, Cincinnati, Ohio. An ISA Member, Mr. Hott also belongs to the American Society of Civil Engineers and holds several patents in valve design.

James W. Hutchison was a member of the staff of the Honeywell Education Center. He retired in March, 1974. Mr. Hutchison's previous professional associations include the Allied Chemical Corporation; E. I. duPont deNemours & Company and Powers Regulator Company. As an ISA Senior Member, Mr. Hutchison has served as Editor-in-Chief of the *ISA Handbook of Control Valves*, and as both Secretary and Education Chairman of the Philadelphia Section. He is a registered Pennsylvania Professional Engineer in the state of Pennsylvania.

Charles F. King is currently a technical sales consultant with the Fisher Controls Company, Marshalltown, Iowa. Here, he acts in an advisory capacity on control valve applications and new products. Mr. King has served the Fisher Controls Company in various capacities—as project engineer, research engineer and director of research. He is an ISA Senior Member. The author of several publications, Mr. King holds a BSME from Iowa State University.

Orval P. Lovett is Section Supervisor of Instrumentation for E. I. DuPont de Nemours & Company, Wilmington, Delaware. He is in charge of instrumentation design for DuPont's expanded and new plant facilities. Prior to his association with DuPont, Mr. Lovett was with the Stone & Webster Engineering Corporation. An ISA Fellow Member, Mr. Lovett has served as Past Section President, Wilmington Section, Past Director of ISA's Process Measurement and Control Division and has received a Special Services Award from ISA's Wilmington Section. The author or co-author of four technical papers on control valves published in *Instrumentation Technology*. Mr. Lovett is also a contributor to the *Control Engineers Handbook*. He is a registered Professional Engineer in the state of Delaware. He is chairman of ANSI B16 Subcommittee on Control Valve Standards. He holds a BS degree in Civil Engineering from Drexel University.

Sal F. Luna is Section Leader-General Instrumentation and Electrical for General Atomic Company. He had been associated with Atomic Power Divisions of Westinghouse since 1955 where he was primarily concerned with the application of advanced technology to a wide variety of reactor plant control systems. Previously, he was a principal instrument engineer with the Catalytic Construction Company of Philadelphia. Mr. Luna has authored a chapter in the *ISA Transducer Compendium* and has authored several technical papers. He has received numerous patent disclosure awards. An ISA Fellow Member, he is presently serving as Vice President of the ISA Industries and Sciences Dept. Mr. Luna holds a BS degree in Chemistry from Niagara University.

Jerry L. Lyons, CMfgE, P.E., is a Senior Project Engineer at Chemetron Fluid Controls Division, a graduate from Oklahoma Institute of Technology, and is a certified manufacturing engineer in the field of product engineering. Mr. Lyons has worked for the past thirteen years as a project engineer in design, research and development of valves. In this capacity, Mr. Lyons has written a book and numerous articles on valves. Mr. Lyons is also serving as a committeeman on pressure ratings for the National Fluid Power Association for the American Standard Institute (ANSI), and is a chairman of the A.S.M.E., OAC subcommittee for valves. He is a Senior member of S.M.E. and I.S.A. (Society of Manufacturing Engineers and Instrument Society of America).

Ralph L. Moore is presently Senior Instrument Engineer with E. I. DuPont de Nemours & Company, Inc. His responsibilities include the simulation of physical and chemical processes by mathematical modeling and the verification of operation of the proposed instrumentation. Prior to his association with DuPont, Mr. Moore was an instructor in instrumentation at Case Western Reserve. An ISA Fellow Member, he is Vice President of the Education and Research Department. He is also a member of the National Society of Professional Engineers, Tau Beta Pi, Sigma Xi, and Phi Kappa Phi. The author or co-author of several articles, Mr. Moore received the ISA *Instrumentation Technology* Award for the article, "The Computer / Team Approach to Control-System Design". Mr. Moore holds Masters Degrees from both Drexel University and Case Western Reserve.

Joseph O'Connor, Jr. is President of Kieley & Mueller, Inc. of Middletown, New York. His previous experience includes two years with Fluid Controls Company, Philadelphia and since 1953 with Kieley & Mueller in several capacities from sales applications, through Vice President, to his present position as President. Mr. O'Connor, who received his BSME from Princeton University, is an ISA Senior Member.

John A. Parker is presently an Instrument Specialist Engineer with E. I. DuPont de Nemours & Company, Inc., Wilmington, Delaware. Here, he is responsible for the design, procurement and start-up of instrument systems for chemical facilities. An ISA Member, Mr. Parker has authored papers for the First ISA Final Control Elements Symposium and the Fisher Controls Company's 1970 Process Seminar.

William L. Scull is Manager Of Research And Development with the Leslie Co., Parsippany, New Jersey. He is responsible for the development of new and improved products in the field of fluid control. These programs have included basic research in the area of valve noise and the development of valve noise prediction techniques. Mr. Scull is a member of the ASME and Acoustical Society of America. He is also a member of the ISA Standards and Practices Valve Noise Committee SP-59. He holds a BS Degree in Mechanical Engineering from Fairleigh Dickinson University and is a registered Professional Engineer in the State of New Jersey.

Gordon F. Stiles is a Technology Consultant in the Research Department of Fisher Controls Co., Marshalltown, Iowa. Mr. Stiles is the author of several articles on flashing, cavitation and control valve sizing. He holds a BS degree in Mechanical Engineering from Iowa State University.

Edward L. Vollmer, III, presently handles technical writing and editing for Honeywell, Inc., Fort Washington, Pa. He is responsible for initiating, writing and editing technical literature. Previously, Mr. Vollmer spent five years as a valve sales engineer and seven years as an instructor in control theory and instrumentation. He is the co-author of *Introduction to Process Control Dynamics*, a text used at the Honeywell Education Center. Mr. Vollmer holds a BS degree in Science, Mathematics and Education from West Chester State College, Pennsylvania.

Charles R. Thornton is currently with E. I. DuPont de Nemours & Company, Inc., as an Instrument engineer. Here, he is responsible for the design and maintenance of instrument systems for a continuous stream, integrated multi-product, chemical complex. Prior to his association with DuPont, Mr. Thornton spent fourteen years in instrument maintenance in the chemical and petroleum industries and ten years in the design of instrument systems including hardware evaluation and selection. Mr. Thornton is an ISA Senior Member.

Rolf B. Weil served as Product Manager of Control Valves for the Jamesbury Corporation from 1966-1971. Prior to coming to Jamesbury, Mr. Weil was a sales engineer for the Milton Roy Company. An ISA Member and a member of ISA's Valve Sizing Committee (SP39). Mr. Weil holds a BS degree from the University of the Witwatersrand, Johannesburg, South Africa.

Paul Wing is currently Vice President Engineering for Masoneilan International, Inc. with Headquarters at Norwood, Massachusetts. He now directs research and development at Masoneilan International and coordinates engineering activities at the 11 manufacturing locations throughout the world. Mr. Wing has served 30 years with Masoneilan in various key engineering positions. An ISA Senior Member, Mr. Wing is the author of numerous articles on control valve design and application. He holds a BS degree in Chemical Engineering from Massachusetts Institute of Technology.

ALPHABETICAL INDEX TO MAJOR CHARTS, GRAPHS AND TABLES

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