ACKNOWLEDGEMENTS

The author is deeply indebted to his supervisor, Dr. C.K.Kwok, for his inspiration, guidance and advice throughout the course of this work.

Acknowledgements are due to Fisher Control of Iowa, Masoneilan of Boston, and Hammel-Dahl of Rhode Island, USA, for numerous discussions and valuable help.

Also, acknowledgements are due to Messrs. A.J. Birchenough and M. Morin of Canatom Limited for their encouragement in the completion of this work.

Acknowledgement is also due to Mrs. P. Soné for typing the thesis.

TABLE OF CONTENTS

Chapter 4

THE FLOW CHARACTERISTICS OF CONTROL VALVES

Chapter 5

CONTROL VALVE ACTUATOR

LIST OF FIGURES

Figure | Page

LIST OF TABLES

NOMENCLATURE

Symbol	Description
B	Contact material properties of Poisson's ratio and Young's moduli
C_1	$C_g/C_v = 36.5\ C_f$
C_2	Correction factor for variation in the specific heat ratio, k
C_f	Critical flow factor
C_v	Valve sizing coefficient (see ISA-S39.2)
C_g	Gas sizing coefficient (ISA-S39.4)
d	Nominal valve size
D	Internal diameter of the piping
E	Young's modulus
f_c	Fraction of critical flow rate
F_d	Valve Style Modifier, dimensionless
F_F	Liquid Critical Pressure Ratio Factor, dimensionless
F_k	Ratio of specific heats factor
F_L	Liquid Pressure Recovery Factor, dimensionless
F_{LP}	Combined Liquid Pressure Recovery Factor and geometry factor of valve with reducers
F_L'	F_L for valve in standard manifold design as specified in ISA S39.2, dimensionless
F_P	Piping Geometry Factor, dimensionless
F_R	Reynolds Number Factor, dimensionless
F_y	Liquid Choked Flow Factor, dimensionless

G	Specific gravity of gas relative to air, with both at standard conditions, dimensionless
G_f	Specific Gravity (ratio of densities) of the liquid at flowing temperature compared to water at 60°F for English units or 4°C for metric units, dimensionless
h	Height of Spherical cap
k	Ratio of specific heats
K	Velocity head coefficient, dimensionless
K_m	Valve recovery coefficient
M	Molecular weight, dimensionless
N	Numerical Constant
p_1	Absolute static pressure upstream of valve
p_2	Absolute static pressure downstream of valve
P_1	Static pressure immediately ahead of the valve or valve and reducer combination
P_2	Static pressure immediately after the valve and reducer combination
P_A	Apparent contact pressure
P_c	Thermodynamic Critical Pressure, outlet pressure at critical flow
P_v	Vapour pressure of liquid at inlet temperature
P_{vc}	Vena Contracta Pressure
ΔP	Pressure drop (P_1-P_2), (p_1-p_2) across the valve
ΔP_c	Critical pressure drop
ΔP_m	Maximum allowable pressure drop
Q	Volumetric leakage rate per unit length of seal

q	Flow rate
q_g	Gas flow rate
q_{max}	Maximum flow rate (choked flow conditions) at a given upstream pressure
Re_v	Valve Reynolds Number, dimensionless
R_e	Reynolds number, dimensionless
R	Radius of spherical cap
T	Absolute temperature (Degrees F + 460 or degrees C + 273)
W	Weight rate of flow, seal width
x	Ratio of pressure drop to absolute inlet static pressure ($\Delta p/p_1$), dimensionless
x_T	Pressure drop ratio factor, dimensionless
x_{TP}	Value of x_T for valve/fitting assembly, dimensionless
Y	Expansion factor. Ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number, dimensionless
Z	Compressibility factor, dimensionless
α	Deformation of spherical cap
γ	Specific weight
ν	Kinematic viscosity, centistokes
Σ	Summation
δ	Gap height

INTRODUCTION

The main purpose of this dissertation is to study in depth control valve seats and discs which are the core of a control valve. These are wetted parts and called trim of a control valve.

Basically, what we are looking for is a situation where two bodies (seat and disc or plug) are contacting each other. By doing so they are modulating and controlling the path of fluids. This can be considered as an immersed body restricting the flow of fluid. The seat is the fixed body, whereas the disc is the moving part.

In Chapter.1,a control valve as described consists of the main sub-assembly such as valve body including trims. The valve body and assembly is the portion that actually controls the passing fluid. It consists of a housing, internal trim (which regulates the fluid flow), bonnet, and sometimes a bottom flange. The valve body sub-assembly is a pressure carrying part which must meet all of the applicable service conditions such as pressure, temperature, elevation, corrosion, etc.

There are various types of control valve body sub-assemblies to suit individual service conditions and piping requirements. Each type has certain advantages and disadvantages for given service requirements and should therefore be selected with care. Also, special attention should be given to radioactive and corrosive (heavy water) fluids, since these have a devastating effect on the wetted parts of the valve materials.

In chapter I, for the sake of better understanding, valve types are grouped by mechanical functions, such as linear motion stem-plug to seat (e.g., globe, angle, cage) and rotary action to stem-disc to seat (e.g., ball, butterfly).

In chapter 2, valve trims, especially seat and disc, have been discussed in detail. The primary function of a control valve trim, which is the movable element (disc) of the valve is to restrict the flow. Its shape fixes the relationship between flow and plug-lift position. A secondary function of valve trims may be to shut off tightly and present a complete blockage to the flow. These functions are accomplished in conjunction with body-shape, actuator (operator), 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. The following parts of the valve constitute valve trims which are in contact with process fluids (wetted parts):

Plug, seat (s), stem, bushings, cages.

Stuffing box components (usually within bonnet) considered as trim on the packing (usually circular rings on round stem) follower, spring, lantern ring and packing retaining ring.

Secondary trim parts use stem-to-plug attachments, seat retaining rings, seat to body seals, spacers. Parts excluded are packing, bonnet, bottom flange, and gasket sealing bonnet and bottom flange to the valve body.

However, it is clearly evident from the description of seat disc control mechanics that an absolute leak tightness is an idealized goal. Also, a judgment has to be made regarding the optimal finish on a seat and disc. For example, if the parts are too finely finished, they will be easily subjected to damage by any suspended particle.

In seat-disc contact theory, an attempt has been made to quantify mathematically valve seat-disc design to satisfy stringent service requirements for long term leak tightness. The conclusion from this section can be drawn by remembering the basic fundamental difference between "CONTACT PRESSURE" seals - high pressure and low area, and "AREA" seals - low pressure and high area.

Finally, in seat and disc design for the globe valves section, various designs have been described; particularly flat-faced seat and disc, included-angle seat and disc, seat and disc alignment, disc guidance have been analysed.

In chapter 3, sizing techniques for control valves have been described. It is evident from the analysis that sizing and selecting a correct control valve and trim for the system depends upon a complete knowledge of the process conditions that the valve will experience in service. A "cook book" solution can create undue problems. It is essential to understand and appreciate that a control valve including trims is only part of a complex system which contains fluids and various hardware, therefore, the method described in chapter 3 to size a control valve is based upon the clear understanding of fluid thermodynamics and valve hardware design.

In chapter 4, a brief resumé of the various flow characteristics of control valves has been depicted. It has been established that the interaction between the control valve flow characteristic and friction loss in the system is very critical. This should be considered and, if possible, analysed theoretically, at the design stage of the system (on loop). The main problem involved is the basic assumption that the flow through the control valve remains the same at wide open position. This is not true in an actual situation. The only parameters of the valve specification sheet that remain fixed are the flow rate and pressure drop.

In chapter 5, the control valve actuator has been analysed. The purpose of the actuator is to provide the required force to operate the valve through the range of the valve travel. The diaphragm actuators are economical and utilitarian while the electro-hydraulic actuators are used for the high thrust applications and may offer economic advantages even with the inclusion of a positioner. The size of the actuator and the operating pressure range determine the available thrust. In addition, other important actuator characteristics like accuracy, load sensitivity, dynamic stiffness, stroking speed and input impedance are discussed.

Chapter 1 THE DESCRIPTION OF CONTROL VALVES

1.1 Introduction

(a) Globe Valves

The designation globe valve encompasses a large number of designs including the hand-operated and power-actuated types used for automatic control. The common feature of all is the internal construction, consisting of a disc or plug or cage moving within the valve body and mating with a seat to effect closure. Globe valves are normally used for throttling applications and the design usually incorporates a tortuous flow path - often 90-degree change in the direction of flow. Most designs are unidirectional; a flow arrow is indicated on the valve body. Pressure drop across the valve is often high in a globe valve and in order to minimise the loss, many manufacturers offer Y-pattern and angle-pattern valves.

(b) Ball Valves

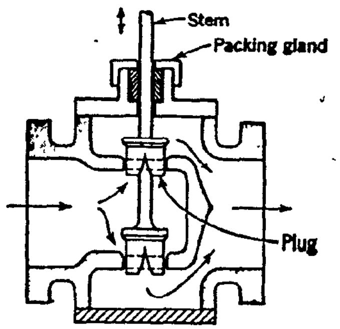
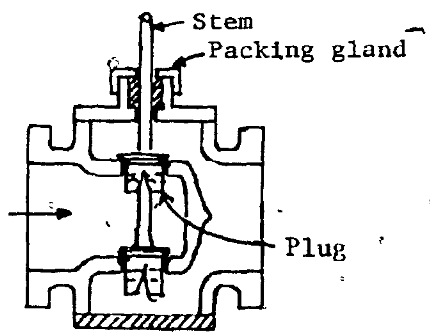
Although ball valves have been available for some time, they have only found widespread acceptance in the chemical, petro-chemical, process industries and power plants within the past fifteen years. This has been due both to advances in elastomer and plastic technology and the development of machine tools capable of mass producing the valve balls. Early designs were metal-to-metal seating and were not bubble tight. The crude packing and metallic seats of the early types have been replaced with materials such as fluorinated polymers and nylon, and the balls currently produced are much better than those in earlier designs. The ball valve is basically an adaptation of the plug valve. Instead of a plug, it has a ball with a hole through one axis to connect the inlet and outlet parts of the body. In the "open" position, the flow is straight through, whereas turning the ball 90 degrees effects shut off. Ball valves come in venturi, reduced and full port pattern. Pressure drop is a function of the port selected; in the full port pattern, it is appreciably the same as a gate valve of the same size and rating. As with the plug valve, ball valves may often be obtained in multiport patterns, effecting savings in valving and other piping components.

(c) Butterfly Valves

The design principle of the butterfly valve is the same as that of a damper in a stovepipe. The disc or flow-control element has about the same diameter as the connected piping. Butterfly valves are made in the swing through style or with seats for shut off. In the shutoff valve, the disc is rotated against a seat that generally incorporates an elastomeric seal. Low-working pressure valves often are completely lined with rubber or other material in closing. The disc compresses the lining around the entire 360-degrees. A number of engineers feel that when the seal of the disc against its lining is directly in line with the stem, it is very difficult to maintain the seal in the area of the stem. In other words, if the seal of the disc is in line with a projection of the centre line of the stem, a problem may result in the disc seal. Because of this design feature, some manufacturers cause the seal of the disc to be offset (from the projection of the centre line of the stem). Butterfly valve bodies in small sizes are of the threaded type, that is screwed directly on to the connecting pipe. In larger sizes, the valves are designed for installation between a pair of pipe flanges and are classified as wafer style or lug style. Where the wafer valve body is used, the bolts span the body. The lug style was developed to facilitate dead-end service. Such a valve body is equipped with lugs that are usually drilled and tapped. Bolts can be installed from either side and thus the valve can be left in the line when piping on one side is removed.

Butterfly valves are good for ON-OFF or THROTTLING service, and cause low fluid-function loss. They are fast acting, since one quarter of the stem will change the disc position from fully opened to fully closed. The lined valves are limited to pressure not more than 150 pounds per square inch, and all butterflies are limited in temperature by their seat and seal materials. Valves that rely on the elastometer seal are available in pressure classes, including ANSI 600 pound. Valves for high-temperature service are available with metal-to-metal seats, but are not tight-shutoff valves.

Chapter 1.2 TYPES OF CONTROL VALVES

VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP. °F	LEAKAGE	RATED C_v RANGE	REMARKS
 Plug "OPEN" Position  Plug 'CLOSE" Position *Fig. 1.1 Double-seated Globe Valves	3/4" - 16" (24" Max.)	(i) Body Rating 150-1500 lb (ii) ΔP Capability Medium	-420 to 1500	Minimum Leakage of 0.5% of Rated C_v	Liq. 2600 Gas: 81000 Vap: 4000	Stem action is Linear Motion. Smaller Actuator force requirement than single seated globe valve.

Notes: The following design parameters are applicable to all Globe Valves.

a) Arduous Duty

(i) Corrosion : Material Selection

(ii) Abrasion : Hardened Trims

(iii) Flashing : Material Selection and hardened Trim

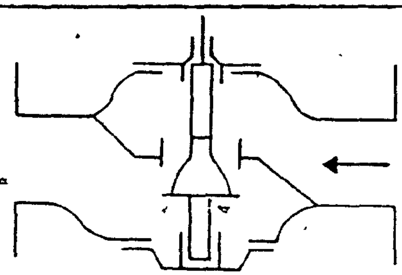
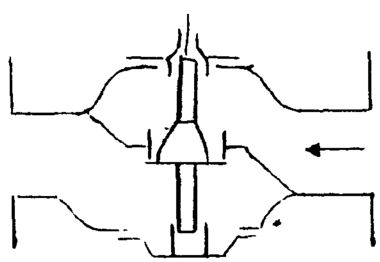
(iv) Cavitation: Special

(v) Toxicity : Bellow Seal

(vi) Slurries : Limited

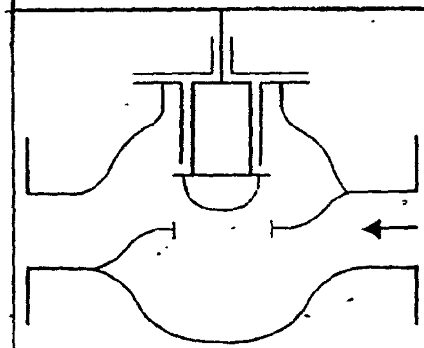
* Hammel-Dahl, U.S.A.

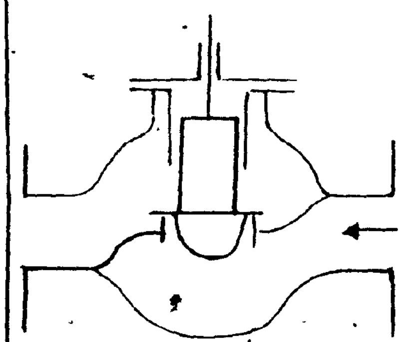
Chapter 1.2 TYPES OF CONTROL VALVES

VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP. °F	LEAKAGE	RATED C_v RANGE	REMARKS
 Plug "Open Position  Plug "CLOSE" Position * Fig. 1.2 Single-seated Top & Bottom Guided Globe Valve	1"- 16"	(i) Body Rating: 150 - 1500 lb.	-420 to 1500	Minimum Leakage .01% of Rated C_v or less with a more powerful actuator	Liq: 2500 Gas: 82000 Vap: 4200	General Service
	b) INHERENT FLOW CHARACTERISTIC		Various			
	c) RANGEABILITY		50:1			

* Masoneilan International, U.S.A.

TYPES OF CONTROL VALVES

VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP. °F	LEAKAGE	RATED C_v RANGE	REMARKS
 Plug "OPEN" Position	1/2" - 16" (30" Max.)	BODY RATING 150-600 lbs. (10"), 900-2500 lb (6")	-420 to 1500	Minimum Leakage .01% of Rated C_v	Liq: 1700 Gas: 57000 Vap: 2600	General Service. Tight Shutoff Available.



Plug "CLOSE" Position

*Fig. 1.3 Single-seated Top Guided Globe Valve

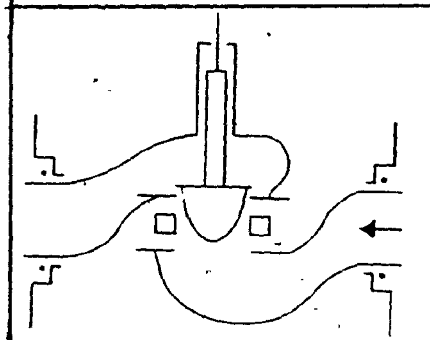
d) INSTALLATION

(i)	Valve quantity wt/space ratio)	High
(ii)	Trim maintenance	In-line
(iii)	Actuator orientation	Fixed

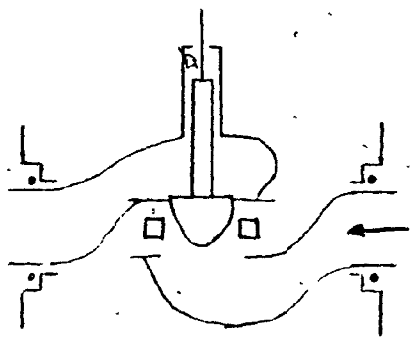
* Fisher Controls, U.S.A.

Chapter 1.2 TYPES OF CONTROL VALVES

VALVE TYPE



Plug "OPEN" Position



Plug "CLOSE" Position

*Fig. 1.4 Split Body Globe Valve

VALVE TECHNICAL SPECIFICATION

SIZE	PRESSURE	TEMP. °F	LEAKAGE	RATED C_v RANGE	REMARKS
½" - 10"	(i) Body Rating 150-600 lb(12") 900-1500 lb(10")	- 420 to 1500	Minimum Leakage of 0.5% of Rated C_v	Liq. 800 Gas 30000 Vap. 1400	General Service Seat ring readily replaced Application limited by temperature and Piping Strain globe; angle assembly

e) ENVIRONMENT

(i)	Physical environment	Material selection and surface treatment
(ii)	Noise abatement	Source and path treatment

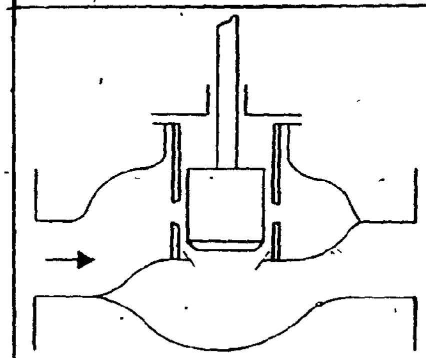
f) ACTUATOR FAILURE MODE

(i)	Fail Open	Yes
(ii)	Fail Close	Yes
(iii)	Fail Fix	Yes

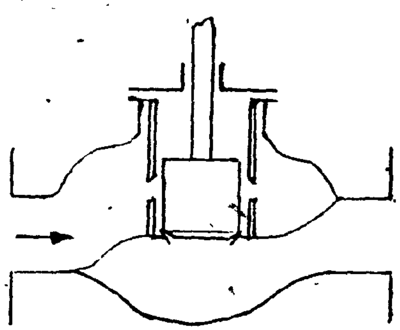
* Masoneilan International, U.S.A.

Chapter 1.2 TYPES OF CONTROL VALVES

VALVE TYPE



Plug "OPEN"



Plug "CLOSE"

*Fig. 1.5 Cage Design Globe Valve

VALVE TECHNICAL SPECIFICATION

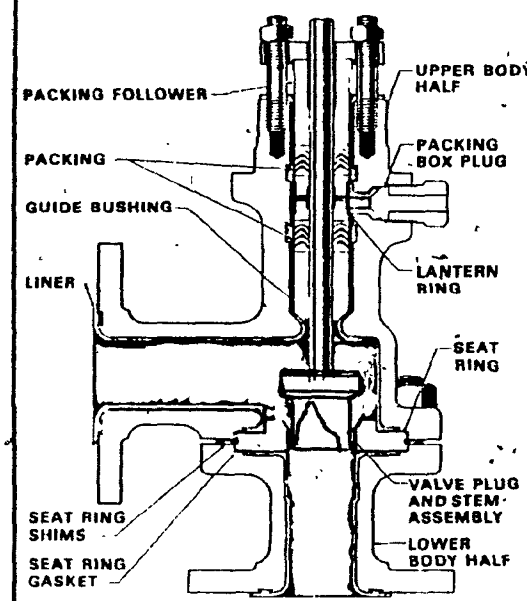
SIZE	PRESSURE	TEMP. °F	LEAKAGE	RATED C_v RANGE	REMARKS
1"–12"	Body Rating 150–2500 lb.	-420 to 1500	minimum Leakage 0.5% of Rated C_v.	Liq: 856 Gas: 29800 Vap: 1490	Quick change trim. More stable throttling. Better plug guiding through bonnet opening

g) CONSTRUCTION MATERIALS

(i)	Standard Body Mat'ls Iron, Steel, S. Steel	Yes
(ii)	Exotic body materials Hastelloy etc.	Yes
(iii)	Integral flange connection	Yes
(iv)	Butt weld end connections	Yes
(v)	Screwed connections	Yes
(vi)	Flangeless connections	Yes

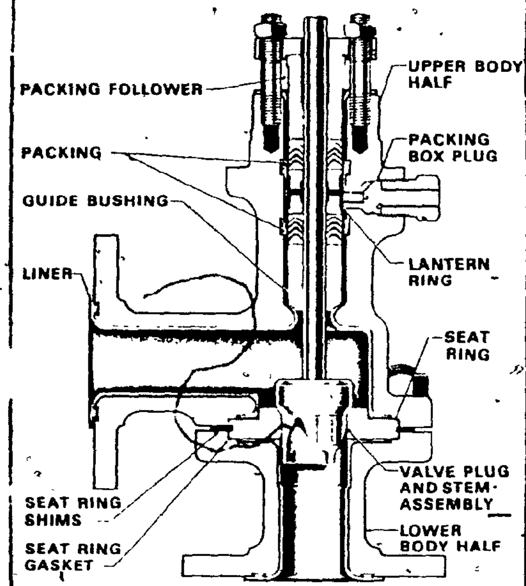
* Hammel-Dahl, U.S.A.

Chapter 1.2 TYPES OF CONTROL VALVES

VALVE TYPE	VALVE, TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP. ^{o}F	LEAKAGE	RATED C_v RANGE	REMARKS
 Plug "OPEN" Position *Fig. 1.6 Angle Body Globe Valve	3/4" - 14". (36 inch max. for steam service).	(i) Body Rating: 150 -600 lb. (14"), 900-1500 lb. (10"). 2500 lb. (6")	-420 to 1500	.01% of Rated C_v.	Liq: 1180 Gas: 27500 Vap: 1810	Low flow Resistance. Used more frequently in On-Off service. Corrosive service (Through 14 inches). Full flushing body Cavity.
	h) ACTUATION (i) Pneumatic Diaphragm — Standard (ii) Electro-Hydraulic — Available (iii) Electrical — Limited					

* Fisher Controls, U.S.A.

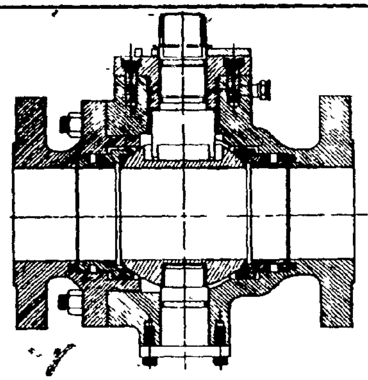
VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP.	LEAKAGE	C_v RANGE	REMARKS
See Fig. 1.6 below						
	Details all the same as those on preceding page.					



Plug "CLOSE" Position

Fig. 1.6 Angle Body Globe Valve

Chapter 1.2 TYPES OF CONTROL VALVES *

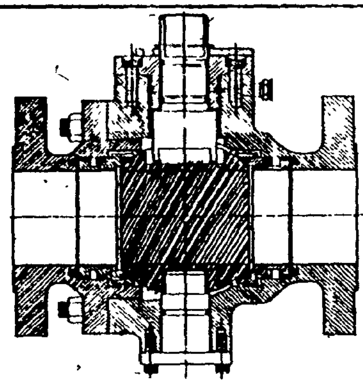
VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP °F	LEAKAGE	RATED C_v RANGE	REMARKS
 Ball "OPEN" Position * Fig. 1.7 Full Ball Valve (supported)	2" - 16"	(1) Body Rating 300 lb.	-420 to 850	Tight shut-off.	Liq: 7050 Gas:128000 Vap: 6400	Tight shut-off. Moderate torque. High pressure drop service Pipeline valve.

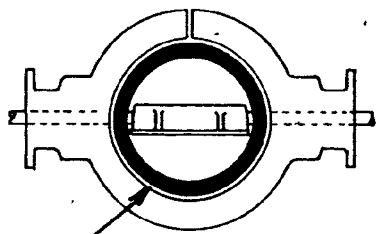
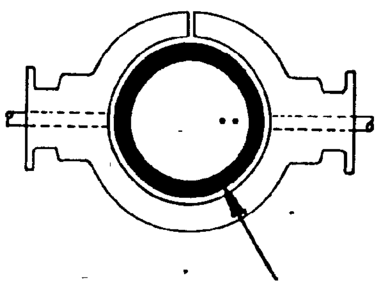
NOTE:

For design parameters applicable to Ball Valves, see page 19.

*Masoneilan International, U.S.A.

Chapter 1.2 TYPES OF CONTROL VALVES

VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP.	LEAKAGE	C_V RANGE	REMARKS
 Ball "CLOSE" Position						
Fig. 1.7 Full Ball Valve (supported)	Details all the same as on preceding page. NOTE: For design parameters applicable to Ball Valves, see page 19.					

VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP. ^{o}F	LEAKAGE	C_v RANGE	REMARKS
 TFE liner *Disc Open, 90° Position, No Pressure in Seal Chamber*  TFE Liner *Disc Closed, Seal Chamber Pressurized—T-Seat is "Pressure Sealed" Against Disc for Extremely Low Leakage Shutoff.* * Fig. 1.8 Lined Body Butterfly Valve	2" – 36"	125-300 lb Body Rating	-20 to 400^{o}F.	Tight Shut-off	Liq: 16100 (60^{o})* 67100 (90^{o})* Gas: 463000 (60^{o})* 1070000 (90^{o})* Vap: 23100 (60^{o})* 53600 (90^{o})*	Bubble tight shutoff. Replaceable liner design available.
	* Valve disc opening in degrees. NOTE: For design parameter applicable to Butterfly Valves, see page 19.					

* Fisher Controls, U.S.A.

VALVE TYPE	VALVE TECHNICAL SPECIFICATION					
	SIZE	PRESSURE	TEMP. °F	LEAKAGE	C_v RANGE	REMARKS
 Disc "OPEN" Position  Disc "CLOSE" Position *Fig. 1.9 Disc Type Butterfly Valve	2" - 72"	(f) Body Rating: 300 lb.	-420 to 2100	Bubble tight shut-off.	Liq: 113000 (60°)* 303000 (90°)* Gas: 3270000 (60°)* 4330000 (90°)* Vap: 163000 (60°)* 217000 (90°)*	Uninterrupted Disc Seating Surface. Fewer Stem Seal Leakage Problems. Available with metal seal ring for high temperature service.

* Valve disc opening in degrees.

NOTE:

For design parameter applicable to Butterfly Valves, see page 19.

*Masoneilan International, U.S.A.

NOTE: The following design parameters are applicable to both Ball Valves (Fig. 1.7) and Butterfly Valves (Figs. 1.8, 1.9) of Chapter 1

PARAMETERS	BALL VALVES	BUTTERFLY VALVES
ARDUOUS DUTY		
(i) Corrosion	Material selection	Material selection
(ii) Abrasion	Hard facing and liner construction	Hard facing
(iii) Flashing	Material selection and hardened trim	Material selection and hardened trim
(iv) Cavitation	Special trim	Avoid
(v) Toxicity	Double packed seal	Double packed seal
(vi) Slurries	Standard construction	Limited
INHERENT FLOW CHARACTERISTIC	Fixed	Fixed
RANGEABILITY	300:1	100:1

NOTE: The following design parameters are applicable to both Ball Valves (Fig. 1.7) and Butterfly Valves (Figs. 1.8, 1.9) of Chapter 1.

PARAMETERS	BALL VALVES	BUTTERFLY VALVES
<u>CONSTRUCTION MATERIALS</u>		
(i) Standard Body Mat'ls Iron, Steel, S. Steel	YES	YES
(ii) Exotic body materials Hastelloy etc.	NO	YES
(iii) Integral flange connection	NO	NO
(iv) Butt weld end connections	NO	YES
(v) Screwed connections	NO	YES
(vi) Flangeless connections	YES	YES

NOTE: The following design parameters are applicable to both Ball Valves (Fig. 1.7) and Butterfly Valves (Figs. 1.8, 1.9) of Chapter 1.

PARAMETERS	BALL VALVES	BUTTERFLY VALVES
INSTALLATION		
(i) Valve quantity wt/space ratio)	Low	Low
(ii) Trim maintenance	Workshop	Workshop
(iii) Actuator orientation	Variable	Variable
ENVIRONMENT		
(i) Physical environment	Material selection and surface treatment	
(ii) Noise abatement	Path treatment	
ACTUATOR FAILURE MODE		
(i) Fail Open	Yes	Yes
(ii) Fail Close	Yes	Yes
(iii) Fail Fix	Yes	Yes

NOTE: The following design parameters are applicable to both Ball Valves (Fig. 1.7) and Butterfly Valves (Figs. 1.8, 1.9) of Chapter 1.

PARAMETERS	BALL VALVES	BUTTERFLY VALVES
ACTUATION		
(i) Pneumatic Diaphragm	Standard	Standard
(ii) Pneumatic Piston	Standard	Standard
(iii) Electro-Hydraulic	Available	Available
(iv) Electrical	Limited	Limited

Chapter 2 DESIGN OF CONTROL VALVE TRIMS

The main purpose of this chapter is to investigate various contact phenomena of seat and disc which form the heart of the control valves.

2.1 Seats and Discs [2]

The leak-tightness of valve seats and discs is a particularly important aspect of valve design. Although the proper initial leak-tightness of a valve may be proved by initial testing, yet it is this extended leak-tightness of a valve that determines its ultimate capabilities for satisfying the system requirements. Extended leak-tightness requiring little or no maintenance takes on added importance in toxic system applications. The design considerations important to achieving this extended leak-tightness of valve seats and discs are discussed in this section. These include the mechanics of seat-disc contact; seat-disc contact theory; valve seat and disc design for globe body valves.

2.2 Seat-Disc Contact Mechanics [2]

Metallurgical contact theory deals with microscopic contact between materials. When magnified, even the smoothest of surfaces appears irregular and rough. Contact between such surfaces therefore consists of contact between only the peaks of the rough irregular surfaces that are called "asperities". Thus, the actual contact area between the surfaces is smaller than the apparent contact area by a large factor. In fact, actual contact is limited to only the spots where the asperities contact.

The importance of this small contact area is apparent for fluid

[ ] See Reference

leak-tightness at the disc-seat contact area in a valve. The fact that the disc and seat are in contact and under high contact pressure does not in itself guarantee that any more than a small percentage of the areas are in contact. Since only a small percentage of the mating surface areas are actually in contact, there is room for the fluid to pass between the contact surfaces. If the mating surfaces are quite rough, that is, if the asperities are relatively large, the flow passages between the mating surfaces will be large and a high leakage rate can be expected. If the surfaces are polished, the flow passages will be small and the leakage rate will be small. Thus, the rate of leakage through the junction is a function of the smoothness and finish of the surfaces.

However, the actual microscopic contact area may be quite small even for two polished surfaces. Furthermore, extending the line width of the apparent mating contact area between the pieces does not necessarily stop leakage; it merely reduces leakage by providing higher flow resistance in the same way that lengthening a pipe increases flow resistance but does not shut off flow. If it is required that the mating surfaces form an absolute leak-tight joint, the surfaces must be forced together with the asperities deforming elastically and/or plastically until a large percentage of the mating surfaces are in actual microscopic contact.

A parameter indicative of good contact is the ratio between the actual microscopic contact area and the apparent contact area of the mating surfaces. Good contact is then defined as the degree to which the actual microscopic contact area and the apparent contact area of the mating surfaces. Good contact is then defined as the degree to which the area ratio approaches one. The following simplified relationship has been used to describe the area ratio between two mating surfaces.[3]

$$\text{Area Ratio} = \frac{\text{Microscopic Contact Area}}{\text{Apparent Contact Area}}$$

where the microscopic contact area is approximately equal to the applied force divided by the material flow pressure. Therefore,

$$\text{Area Ratio} = \frac{\dfrac{\text{Applied Force}}{\text{Material Flow Pressure}}}{\text{Apparent Contact Area}}$$

$$= \frac{\text{Applied Pressure}}{\text{Material Flow Pressure}}$$

The material flow pressure is approximately three times the yield strength of the softer material of the two in contact.[4]

An example will illustrate the smallness of the area ratio obtainable in actual practice. Consider a seal material with a yield strength of approximately 70,000 psi. The flow pressure would then be very nearly 210,000 psi. If an applied pressure of 1000 psi is used to force the surfaces together, the

$$\text{Area Ratio} = \frac{1000}{210000} = \frac{1}{210} = 0.0048$$

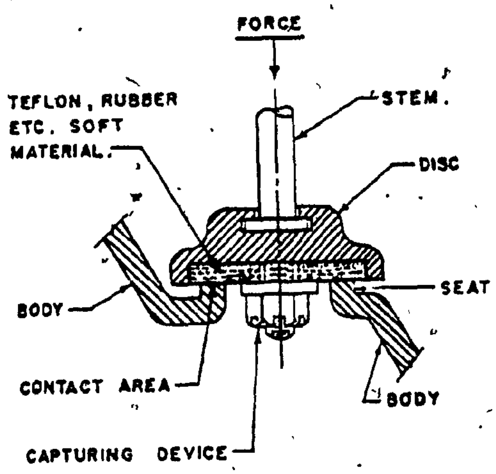
This shows that only a small percentage of the two surfaces ever come together in contact. The forces pushing the two surfaces together are seldom sufficient to deform or flatten more than a small percentage of the high points in the two mating surfaces. Thus, an absolute leak-tight seal is an idealized goal.

Practical characteristics of fluids and seal materials make it possible to achieve a leak-tight seal within the sense of a measurable leakage rate without achieving an area ratio of 1.0. The area ratio relationship is useful in determining how best to minimize leakage by maximizing the area ratio. First, the applied contact force and the applied pressure can be increased to increase the area ratio. This would increase the size of the valve components to provide the additional strength needed.

Second, the applied pressure can be increased by decreasing the contact area. This is done in high-performance valves by designing the valve seating surfaces to mate with "line" contact. Line contact is so named because contact takes place around the seat on only a circular line. Since this line is designed to be as thin as possible, the integrated surface of the contact area is very small. The applied pressure is therefore high, increasing the area ratio. It should be noted that surface deformations, which may occur because of high applied pressure, tend to increase the width of the line as this deformation proceeds. This increase in line width decreases the area ratio. Where this can occur, the valve is generally designed to minimize such deformation through the provision of maximum hardness in the mating surfaces.

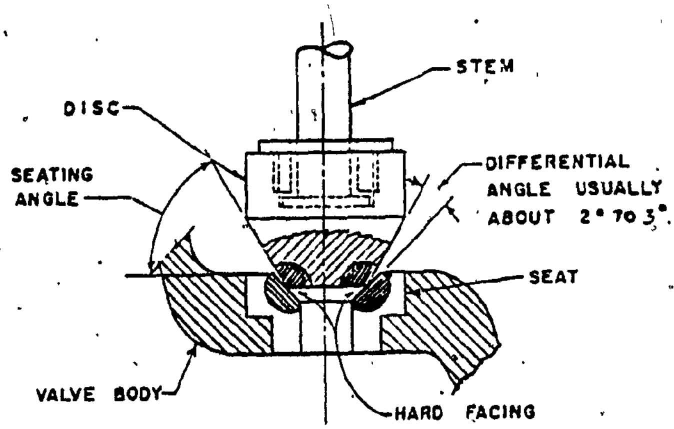
Last, the contact area ratio equation indicates that changing the material flow pressure of the contact surfaces also changes the area ratio. Taken at face value, it would appear that a low material flow pressure is desirable. In fact, this is the case for many valves with soft seats. Usually, soft-seated valves provide a high area ratio because of the readiness with which microscopic deformations take place so that the contact areas may blend together. The geometry of the mating surfaces in soft-seated valves is generally like that depicted in Fig. 2.1.

It is clear that the applied pressure on the contact area varies only with stem force. Deformation of the mating surface does not affect the applied pressure.



Note: EFFECT OF PROCESS FLUID PRESSURE NEGLECTED

* Fig. 2.1 Geometry of Mating Surfaces in Soft-Seated Valves.



NOTE:
THAT ACTUAL CONTACT OCCURS ONLY AT ONE POINT ON THE FACE OF THE SEAT; HENCE LINE CONTACT.

*Fig. 2.2 Included-Angle Globe Valve Designed for Hard Seat Materials.

* Hammel-Dahl Co., U.S.A.

Where soft surfaces will not stand up under severe operating conditions, hard materials and a totally different configuration must be used. A valve designed to use hard seat materials is illustrated in Fig.2,2. In this valve, a differential angle is provided in the geometry between the disc and the seat in order to establish a line contact. This limits actual contact to a very narrow line around the valve seat. Maximum hardness of the seating materials is desirable in this type of valve. If the hardness is not sufficient, the geometry of the mating surfaces is such that deformation will allow the apparent contact area to increase through indentations in both the seat and the disc. An increase in the apparent contact area would mean that for a constant stem force, the applied pressure of contact would decrease. Therefore, increasing the hardness of the mating surface in such a seating arrangement improves the ability of the valve to maintain a narrow line seal with a small apparent contact area over its lifetime. Plastic deformation of a soft material used in this type of seal design can result in an increase in the apprent contact area of a factor of two or three with a corresponding decrease in the applied pressure. Thus, hardness of seating surfaces greatly increases the numerator of the area ratio equation.

On the other hand, hardness and surface strength property changes in the mating surfaces can only affect the material flow pressure of the material by fractions. Thus, an increase in the hardness has only a weak effect on the increase in the denominator of the area ratio equation. The net result is that the higher the hardness, the greater is the area ratio and the more leak-tight the seal is apt to be. This applies to differential-angle seats only. It will be seen that this is a common configuration and an ideal to be attained. Where maximum hardness is not required for contact considerations, it usually is for other reasons such as maximum wear resistance, which is discussed later.

Some important conclusions can be drawn from the preceding discussions. First, it indicates that well-aligned line-contact globe valves will provide good seat leakage characteristics. It also indicates that the leak-tightness of gate valves may not be as good because true line contact is not attainable with the geometry of gate valves. Further, since wedge gate valves apply high pressures through the additional mechanical advantage of the wedge disc, they may have higher area ratios and less leakage than non-wedge types. This, however, is not true for very large parallel disc gate valves. Not only is line contact not possible for them, but additional applied pressure from stem force and wedging action cannot be brought to bear. In very large parallel disc gate valves, any force provided by the stem is usually small and incidental when compared with the very large force from the differential pressure across the large disc. When this condition occurs, the parallel disc gate valves are equally as leak-tight as the wedge gate valves. In addition, parallel disc valves have a seat wiping feature that may provide a measure of seat cleanliness not obtained with wedge gate or globe valves. However, from the standpoint of control theory alone, it appears that such valves cannot as easily bring as many factors to bear in increasing the area ratio as can wedge gate valves or globe valves.

2.3. Theory of Seat and Disc Contact

Applying the mechanics of surfaces in contact presented in the preceding subsection and supporting it with trial and error, ways of improving leak-tightness of valves can be found. A comparative analysis is presented here in which the problem is reduced to a mathematical model and solved to determine the quantitative importance of the various parameters which contribute in valve seat-disc leak-tightness. The new emphasis on analytical theory in valve seat-disc design has resulted from the more stringent requirements for leak-tightness in nuclear plant application.

Surface contact analyses in the literature are presented from several standpoints and with different mathematical models. Yield and plastic deformation are assumed to occur in some cases, and purely elastic deformation is assumed to occur in other cases. The surface is assumed to consist of small wedges in some cases and of small caps of spheres in others. Since surface wave lengths of several magnitudes can be present simultaneously, a "wave" concept is built into some theories. It is unfortunate that most of these mathematical models failed to give a true picture of the phenomenon of surfaces pressed together to form a seal. As our knowledge of the mechanics of surfaces in contact increases, a better mathematical model may be developed to describe this highly complex phenomenon.

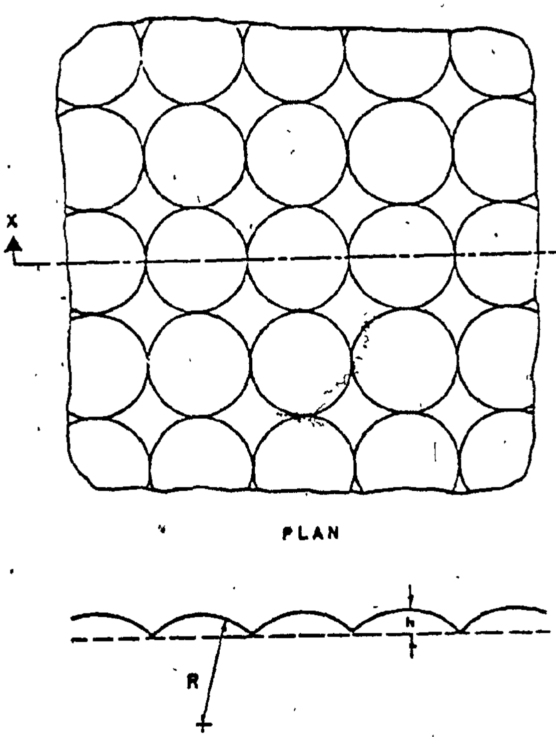
However, such analyses are not absolutely without practical benefit. Since the problem of valve leak-tightness is reduced to a math. model, parameters affecting seat material and geometry can be determined. For instance, the relative merits or rating for each type of seal (line contact or area contact) can be indicated in the analysis, and other variables in the design (material hardness, contact pressure, and surface roughness) may be evaluated. All of these factors and their relationship can be determined within the inherent accuracy of the analysis.

For the applications considered in this document, it will usually be assumed that the seating contact will consist of surfaces with exceptional hardness that are polished to a high degree of smoothness. The seal must also remain tight over a large number of opening-closing cycles. Therefore, the analysis using a mathematical model of sphere caps and assuming only elastic deformation can generally be considered applicable. Such a quantitative analysis was developed by the Berkeley Nuclear Laboratories in England probably because of the high leak-tightness requirements for valves in *HTGR service,[5] and the theory of that analysis is presented here. The treatment is generally applicable to any fluid (liquid, vapor, or gas) with the inclusion of a suitable viscosity constant.

The analysis[5] can be briefly described as follows. First, the geometrical model is established, as illustrated in Fig.2.3. The surface is assumed to consist of small caps of spheres placed adjacent to each other in rows and columns. These small spherical caps are identical in size. The height, h, of each spherical cap is taken as the distance from a line through the point at which the cap connects with the adjacent spherical caps to the upper or outermost point of the spherical

* HTGR - High Temperature Gas-Cooled Reactor

cap. It may be seen in Fig.2.3 that if R is the radius of each spherical cap, a measure of the surface smoothness is h/R. At this point, it is assumed in the analysis that a perfect plane is laid in contact with the tops of the spherical caps. These surfaces are then loaded together so that the tops of the spherical caps are flattened. Consider a point on each body such that these two points approach each other by a distance α as the two bodies are loaded together. In this case, α is the deformation of the spherical caps or the distance by which the two surfaces approach each other as the force pressing these surfaces together is increased. The distance can be calculated by elastic theory as a function of the radius R, the apparent contact pressure P_A and the material characteristics ν and E.



SECTION THROUGH X-X

Fig. 2.3. Model of surface Roughness.[5]

For small apparent contact pressures, a flow path will exist between the rough surface and the perfect plane. The volume of fluid between these two surfaces will depend upon the approach distance α until the spherical caps are flattened. It is assumed in this analysis that α cannot exceed h. The simplification is also made that this fluid volume can be expressed as a uniform gap of height δ. That is, a fictional gap between smooth surfaces of height δ will contain the same volume as the irregular voids remaining between the rough surface and the perfect plane. For this purpose, it is desirable to express the gap in the non-dimensional form δ/h.

It can be shown by fluid flow theory that the quantity of fluid passing the gap between two smooth surfaces is proportional to δ^3 when a given fluid pressure differential from gap inlet to gap outlet is assumed. The quantity of fluid leaking through the gap is also a function of the length of the leakage path or the line width of the apparent contact area. That is,

$$Q \text{ is proportional to } \delta^3/w \quad [5]$$

where

Q = volumetric leakage rate per unit length of seal and

w = seal line width or the straight-line distance leakage must pass going through the seal.

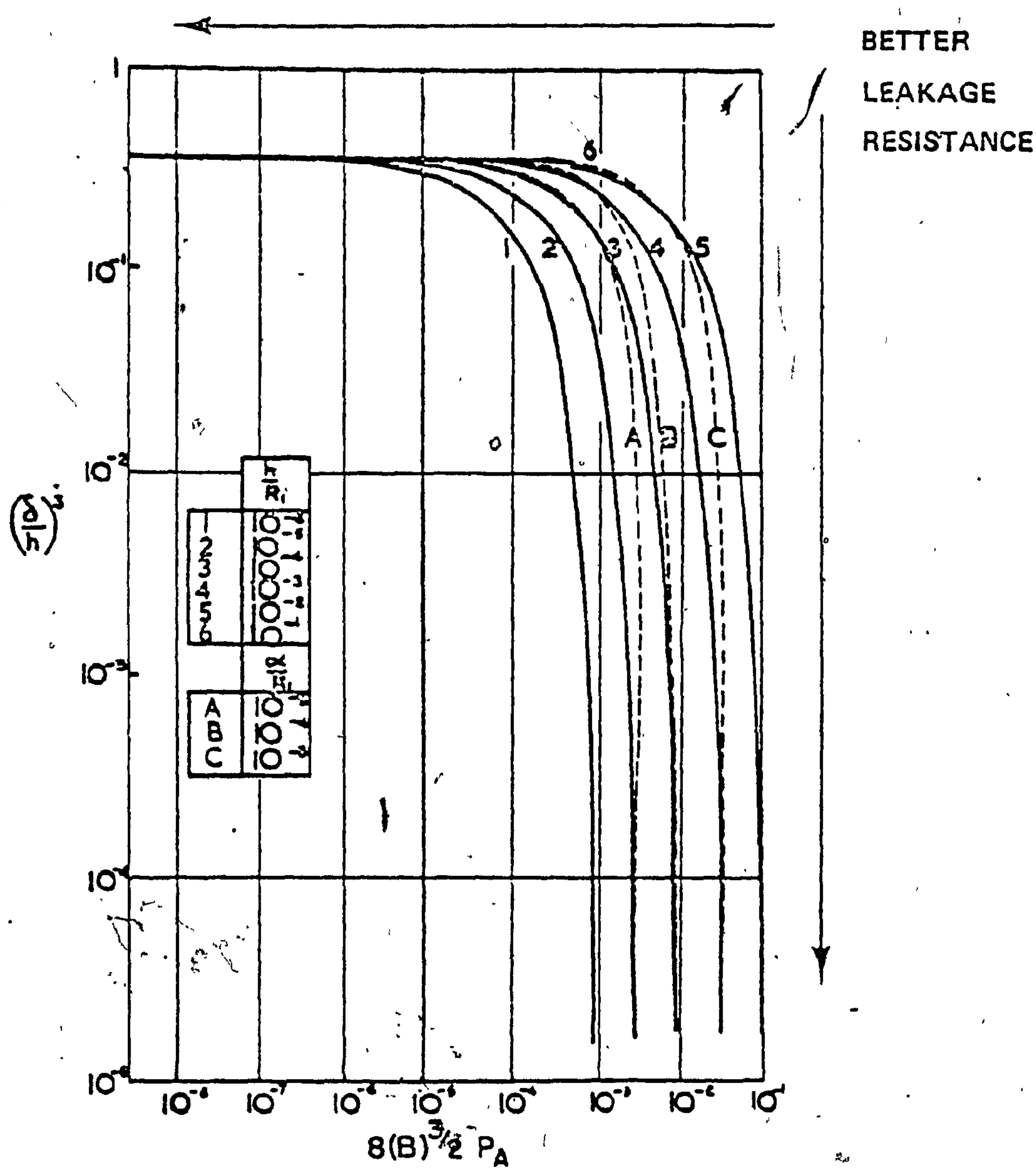
Simultaneous solution of the related equations provides a plot of the cube of the equivalent gap $(\delta/h)^3$ as a function of the apparent contact pressure expressed in non-dimensional form. A family of curves for various ratios of surface roughness given in the h/R relationship is illustrated in Fig 2.4. Since the leakage rate is proportional to δ^3/w,

$$Qw \text{ varies as } \frac{\delta^3}{h^3} \text{ or } \left(\frac{\delta}{h}\right)^3$$

Therefore, the ordinate is also a scale for the product of leakage rate multiplied by seal line width.

This contact theory is useful in evaluating the various factors affecting valve seat leakage and determining how they may be changed to improve an existing valve design. However, the results would not be very accurate when used to determine actual leakage rates from certain given parameters. The contact theory is not sufficiently accurate to permit experimental duplication of the curves illustrated in Fig. 2.4, but experimental work is expected to produce a set of curves of similar geometry to confirm the relationships of the various parameters.

The relationships illustrated by the curves in Fig. 2.4 can be explained as follows. The ordinate, $(\delta/h)^3$, is a dimensional quantity which is directly proportional to the volumetric leakage rate multiplied by the seal line width. The seal line width is the straight-line distance the leakage flow must travel in order to pass through the seal. There are other factors, such as fluid viscosity, p.d., etc. that must be assumed constant for the sake of simplicity. So, $(\delta/h)^3$ should be reduced, if possible, in good seal design to reduce both leakage and seal line-width requirements. Obviously, if Qw is small the leakage rate must be small. However, a high $(\delta/h)^3$ can be tolerated if the seal length is allowed to be large since leakage is also inversely proportional to seal length. One must keep in mind that seal length is dependent upon other factors.



[5]

Fig 2.4. Variation of the cube of the equipment gap with apparent contact pressure.

The abscissa is $8\,(B)^{3/2}\ P_A$, which is a dimensionless quantity representing the contact pressure where B is a parameter defined as a mathematical combination of the contact material properties of Poisson's ratios and Youngs moduli. [5]

$$B = \left[\frac{9}{16}\left(\frac{1-\nu_1^{\,2}}{E_1} + \frac{1-\nu_2^{\,2}}{E_2}\right)^2\right]^{\frac{1}{3}}$$

where

ν_1, ν_2 = Poisson's ratios of the two bodies in contact and

E_1, E_2 = Young's moduli of the two bodies in contact.

The parameter B does not differ greatly for metals. The abscissa is therefore almost a direct function of the apparent contact pressure P_A.

The need to use sufficient apparent contact pressure on the surfaces to deform the spherical caps elastically and reduce the leakage gap is illustrated in the sectional view of Fig.2.3. The resulting curves of $(\delta/h)^3$ as a function of contact pressure illustrated in Fig.2.4 indicate that $(\delta/h)^3$ remains constant up to a critical contact pressure, determined by the degree of smoothness of the surface, at which point $(\delta/h)^3$ decreases rapidly. At this point, the surfaces have been brought into a high degree of contact and leakage is cut off. Thus, the first objective in obtaining leak-tightness is to apply enough apparent contact pressure to operate the seal in the region to the right of the bend of the curves. In this region, small increases in apparent contact pressure produce a rapid decrease in leakage or, specifically, in Qw.

The index h/R is a dimensionless term representing smoothness. The index α/R represents an elastic deformation limit. A surface finish h/R cannot have a deformation greater than $\alpha/R \approx h/R$ and still be considered within the scope of this analysis. Since h/R is not directly convertible to rms surface finish to be specified for the seat surfaces.

The group of curves for h/R illustrated in Fig. 2.4 shows that improving the surface finish moves the bend of the curve to the left. Thus, the surface finish can change the required apparent critical pressure by factors of 10 to 100 for a given leakage rate. This can be a high premium to pay for not polishing the contact surfaces. Rapid multiple-factor increases in leakage are also a high premium to pay for surfaces roughened by erosion and wear or for a design where there is a high degree of sensitivity to slight roughening caused by normal operation. Further, it is interesting to note that surface finish has a large effect on leak-tightness, at least in the elastic theory. Undoubtedly, where small plastic local deformation occurs, the initial surface finish has less of an effect. What actually happens in seat overstress is not clear and has not been well investigated. In any case, the harder and stronger the hard-facing seating material, the more applicable is the elastic analysis, which includes this emphasis on surface polish.

One other factor about Fig. 2.4 should be emphasized. The ordinate $(\delta/h)^3$ is directly proportional to the product of the leakage rate and seal line width. Should structural limitations prevent the use of an apparent contact pressure to the right of the bend of the h/R curves, the leakage rate may be reduced by making the line width of the seal very wide. In this case, the value of Qw remains constant but Q becomes smaller as w increases.

From the preceding discussion, it is apparent that there is some seal line width that will be more apt to leak than others. A way of designing to the right of the bend of the h/r curves is to make the line width of the seal very narrow, increasing the value of P_A'. A conceptual graph of leakage as a function of seal line width for constant stem force is illustrated below in Fig. 2.5. The maximum leakage will occur when the various parameters place the seal design in the bend of the h/R curves of Fig. 2.4.

The conclusion generally reached from the results of the theoretical analysis is that the valve seat should be either very narrow or very wide. Clearly, other factors such as compressive stress resistance and thermal stress design must also be considered. This can

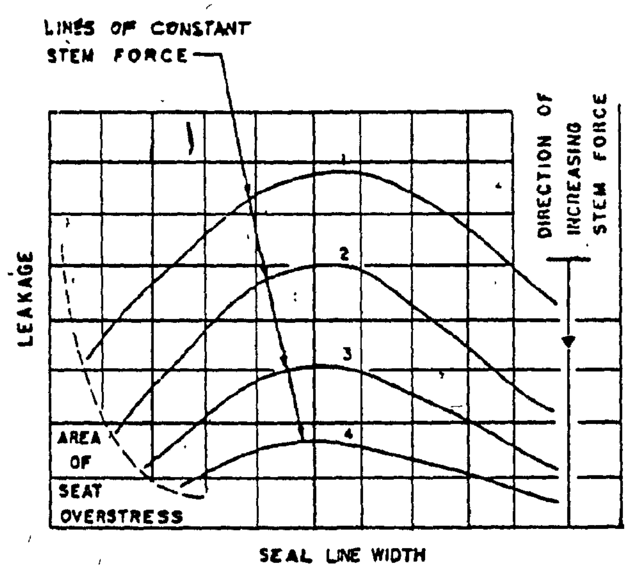


Fig.2.5. Conceptual Graph of Leakage as a Function of the Seal line Width for Constant Stem Force.[5]

be immediately recognized in valve practice as the difference between the approaches taken in globe valve design where line contact is used and in swing check valve design where a large flat-faced seat is used. Each can be designed to have acceptable leak-tightness from the standpoint of the theory, given no other complicating factors. But within the scope of practical limitations, the basic fundamental difference between "contact pressure" seals, high pressure and low area, and "area" seals, low pressure and high area, should be remembered.

2.4 Design of Seat and Disc for Globe Body Valves [2]

Globe valves designed for high-temperature and high-pressure nuclear service normally have either an integral hard-faced seat or a welded-in seat ring. This practice gives almost complete assurance of no leakage around the back of the seat ring. Since integral seats obviously cannot be replaced and welded-in seat rings are difficult to replace at best, the design should provide for in-place lapping of the seat a number of times over the service life of the valve.

If the integral hard-faced seat is used, the requirements for applying a thick, uniform, and hard material to the valve seat should be recognized. The better hard-facing materials are applied by a shielded welding process, and the hard-facing process must be performed to a qualified welding procedure. The base material on which the hard-facing material is to be laid must be sound. Careful control of the temperatures during each step of the process is necessary to assure that the hard-facing material is uncontaminated and free of cracks. Sensitization of the valve body material must also be guarded against.

When the hard-facing is properly applied, the integral hard-faced seat provides a reliable and trouble-free design.

The use of a welded-in seat ring offers a number of advantages. First, the material properties desired at the seal surfaces are different than those desired for the valve body. By using a harder material for the seat, hardness is not a requirement of the valve body, especially if it is achieved at the sacrifice of ductility. The longer cyclic life afforded by the harder seat material will reduce the frequency of lapping required to keep the seat-disc closure leak-tight and might even obviate the need to replace the valve during the service life of the system.

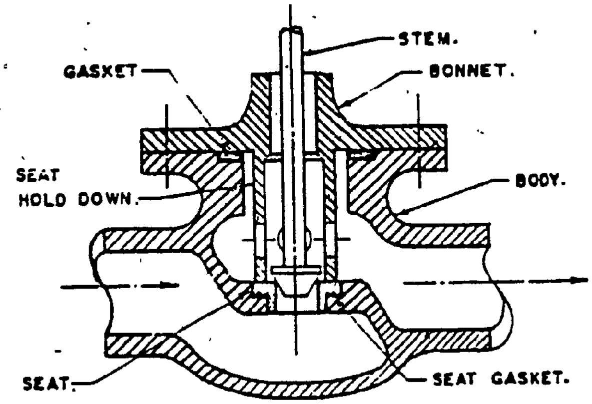
From a fabrication standpoint, the use of a seat ring offers further advantages. Fabrication of the seat ring is easier to control because of the uniformity of material thickness and the smaller size of the item, and the material properties are closer to the desired conditions. If hard-facing is to be applied to the seating surface, the hard-facing procedure is better controlled when working with the smaller item. The separate seat ring is also easier to inspect by radiography, sonic, and dye penetrant methods than seats machined directly in the valve body. Finally, if the valve body is a casting of a globe valve with a line type of seal, even minor porosity at the lip of the seal would require extensive welding repairs or make the whole casting worthless. This problem is avoided by inserting a forged seat ring into the more porous casting.

Valves intended for low-temperature and low-pressure service often have replaceable seat rings to improve the speed and ease with which a leaking valve seat may be made leak-tight again. In theory, replacement of the seat ring is a quick and easy way of correcting a leaking seat condition. In practice, replaceable seats are frequently difficult to remove. At worst, such removable valve seats allow leakage around the back of the seat, thereby eroding the valve body and causing general deterioration that may require replacement of the entire valve. If replaceable valve seats are used, care should be taken in the design and fabrication of the seat, particularly the seal between the seat and the valve body in low-pressure and low-temperature systems.

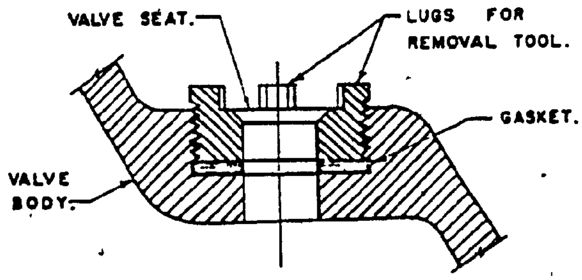
In recent years, there are a number of valves with replaceable seat rings on the market. This type of design has become more reliable in recent years because of the advances made in non-weld seal design. Metallic O-rings and similar types of gaskets now available have generally given good service. From a maintenance standpoint, replacing a valve seat appears to be easier and quicker than either lapping a valve or replacing it, even at the expense of more frequent maintenance. A case in point is the design of high-pressure steam straps. Since steam-trap service is particularly severe, the internals of steam traps do not normally last more than a few years. In such cases, the replaceable seats become a design requirement. At the same time, it is worth noting that maintenance of traps involving replacement of the seat requires care to ensure that the trap does not leak around the back of the new seat and exhibit worse leakage than before repair when the valve leaked by the eroded seat-disc contact surface. In most systems, valves can be designed (allowing for maintenance) to be properly leak-tight for the full service lifetime of the valve. In many valves, the assurance of leak-tightness provided by a seat weld is an important factor in valve design even where good non-weld seals could be used.

There are a number of ways of fastening a seat in a valve. Replaceable type seats may be fastened with a bonnet hold-down or with screw threads, as illustrated in Fig. 2.6. Non-replaceable seats are shrink-fitted, screwed, or fastened by welding. In high-temperature, high-pressure valves, the seats are typically shrink-fitted and fastened by a weld, as shown in Fig. 2.7. When the valve does not have a compressed gasket between the seat and body, the shrink-fit type of fastening provides an easier method than the screwed type of fastening. It also resists loosening under thermal conditions. Once fitted, a shrink-fit type of seat cannot be replaced. However, screwed-in seats are not easily removed either. Corrosion, galling, and freezing of the threads can effectively preclude seat removal. Screwed-in seats can also suffer thermal notch fatigue at the thread roots.

If the seat is not properly designed, high temperature can lead to cracking of the seat and overstress of the valve body. Thermal stresses may be caused by the differences in temperatures and thermal expansion coefficients of the valve seat and body, or the disc seating force may be increased because of the differences in temperatures and thermal expansion coefficients of the valve

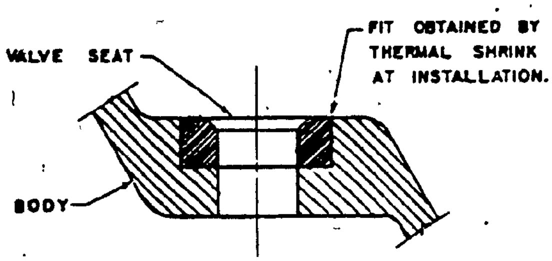


BONNET HOLD DOWN SEAT

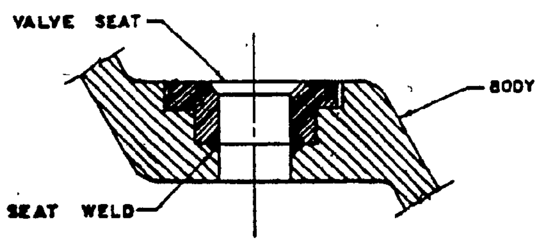


SCREW IN SEAT

* Fig. 2.6 Methods of securing replaceable seats



SHRINK FITTED SEAT



WELD FASTENED SEAT

* Fig. 2.7 Methods of securing non-replaceable seats

* Masoneilan International, U.S.A.

seat and body, or the disc seating force may be increased because of the differences in temperatures and thermal expansion coefficients of the valve body, bonnet, and stem. The design for the valve must include provisions for these thermal stresses, particularly if the fluid temperatures are subject to rapid fluctuations. In valves with screwed-in seats, the thread clearance should not be so small that an interference fit or high constraint is provided. In valves with shrink-fit seats, the degree to which the seat is compressed should be limited so that thermally induced stresses will not crack the seat.

In valves with welded seats, the seat-to-body weld should not be exposed to the thermal stresses just discussed. Normally, the seat is designed with a shoulder, as illustrated in Fig. 2.8, so that axial expansion is constrained by the shoulder and not the weld. When fitting up the seat in the body, the seat should be in full contact with the shoulder before welding. Since it is important that thermal stresses be kept off the weld, the weld is normally placed on the bottom of the seat adjacent to the seat shoulder. If the welds were placed at the top of the seat, undesirable thermal stresses could result. Since it becomes difficult to place the weld below the seat in smaller valves, it must be placed above the seat for better accessibility. The seat design then should have provision to allow for thermal expansion. In this case, the shoulder should be moved up to the top of the seat, and a gap provided between the bottom of the seat and the body, as illustrated in Fig. 2.8.

Radial thermal expansion of the seat must be carefully controlled to preclude seat warpage and the consequent loss of tightness.

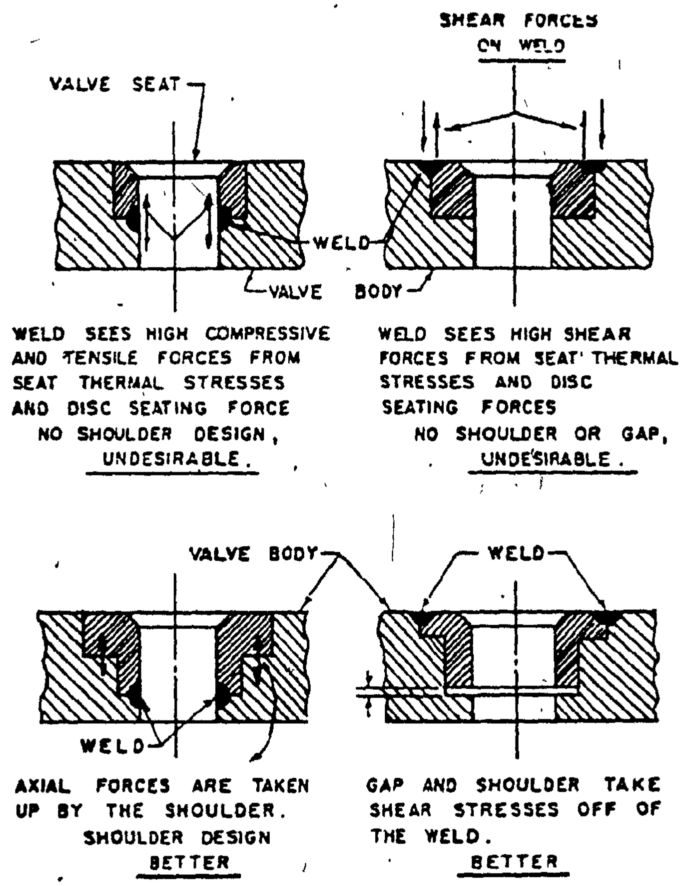


*Fig 2.8 Design for Shoulder in Valves With Welded Seats.

Seats are usually well constrained by the body, and thermal expansion in the radial direction is generally taken up by internal stresses with perhaps some strain ratcheting. Some attempts have been made to reduce the possibility of seat warpage by thermal effects through the use of a "floating" seat design, as illustrated in fig. 2.9. Such a design provides

*Note: Masoneilan International, Inc.

an annular gap between the seat and the body to free the seat from radial body constraint. Thermal expansion of the valve body can sometimes be uneven due to its geometric configuration. It will cause compression of the seat ring and eventual warpage during thermal cycling. A "floating" design may be considered. However, this design suffers from other drawbacks, including those normally associated with any deep crevice within a valve. In addition, there is the possibility that the weld will be subject to even greater stress.

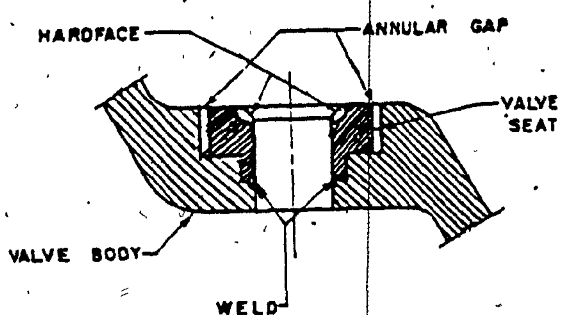


*Fig. 2.9 "Floating" Seat Design.

2.4.1 Seat and Disc with flat face design. Along with the material requirements for valve seats and discs, mating geometry is an important part of valve design for leak-tightness. Globe valve seats may be of either the flat-faced type illustrated in Fig. 2.1 or the included seat angle type illustrated in Fig. 2.2. The included seat angle types may have shallow or deep angles and differential

*Note: Fisher Controls Company, U.S.A.

angle features. The widths of the seat-disc mating surfaces may be small or large, and the width of the mating surface of the seat may be different from that of the disc. It may be important in some applications to allow for lapping the hard-facing as the lifetime of the valve progresses. The type of mating contact used should be considered in providing disc guidance since certain types of mating contact may have very low tolerance to a slight out-of-alignment condition of the disc.

The case for the flat-faced seat and disc geometry is straightforward. This geometry has the advantage of being simple and not susceptible to alignment problems. Contact is usually of the "area" type, and machining of the flat and right-angled dimensions to good tolerances is relatively easy. The seat can be machined and inspected to a given flatness tolerance. Almost no disc alignment is necessary because the disc is guided by the mating surface of the seat. Since area contact is used, lateral guidance is controlled only by the amount of overhang provided by the disc.

The disadvantages of the flat-faced type of seat-disc mating surfaces are also quite clear. First, design is more amenable to the area type of seal. For a flat-faced type of seat and disc to have a line-contact pressure type of seal implies either a thin seat surface or a small differential angle on the face of either the disc or seat. A thin seat involves the risk of deformation and failure, and a small disc or seat differential angle takes away the advantage of flatness. Further, line-contact pressure seals require a certain amount of seat elasticity that is not easily obtainable in a flat-faced seating design. As a result, a minute out-of-flatness condition in a line-contact pressure seal on the part of either the seat or the disc would require excessive stem forces to achieve the necessary complete contact. For these reasons, the flat-faced type of seat-disc geometry generally employs area contact and a wide seat.

2.4.2 Included-Angle Seat and Disc. Since mating conditions cannot be controlled precisely, valves with included-angle seats and discs are more amenable to line-contact pressure mating because of the additional mechanical advantage built into the seat and disc contact through the wedge design. Some typical included-angle valve seat configurations are illustrated in Fig. 2.10. However, slight out-of-tolerance conditions in the angles or the circularity of either the seat or disc can leave large portions of the mating surfaces out of contact and result in leakage through the seat.

Included-angle seats and discs increase the contact pressure by factors of about 1.5 to 5 without the use of heavier and larger stems and yokes. Included-angle seats have greater elasticity than flat-faced seats because of the elastic properties of the inner diameter of thick-walled cylinders. Included-angle seats also have some well-known deficiencies. If the angle is narrow, the disc can seriously impair seating. Care must be taken to limit the applied seating force or the valve body may be overstressed as a consequence of the mechanical advantage inherent in the wedge design.

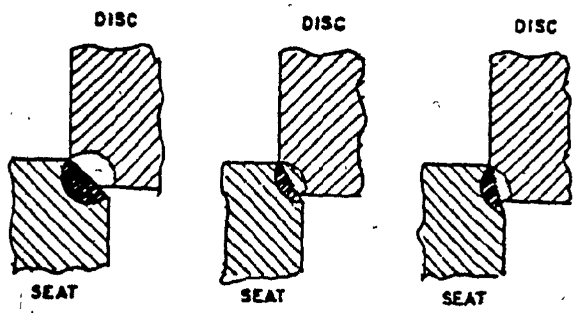


Fig. 2.10. Typical Included-Angle Valve Seat Configurations.

The mechanical advantage obtainable with included-angle globe valves as a function of the included angle is illustrated in Fig. 2.11. Surface friction, which reduces contact pressure, has been considered. If mechanical advantage increases rapidly as the included angle becomes larger than 40° or 45°, especially for the low friction factor curves. However, above 60° approximately, there is an increasing probability that friction can be high enough to cause sticking of the disc within the seat. The boundary at which sticking is apt to occur is superimposed on the graph. It is clear that above 70° or 80°, surfaces can stick together even with low friction.

From the curves shown in Fig. 2.11, it may be concluded that there is not much to be gained from using a shallow angle. Included-angle seats are practicable for included angles of from approximately 40° – 70° or 80°. In order to avoid all possibility of disc sticking accidentally in situations where thermal effects are present and the mating surfaces are roughened by erosion, the maximum practicable

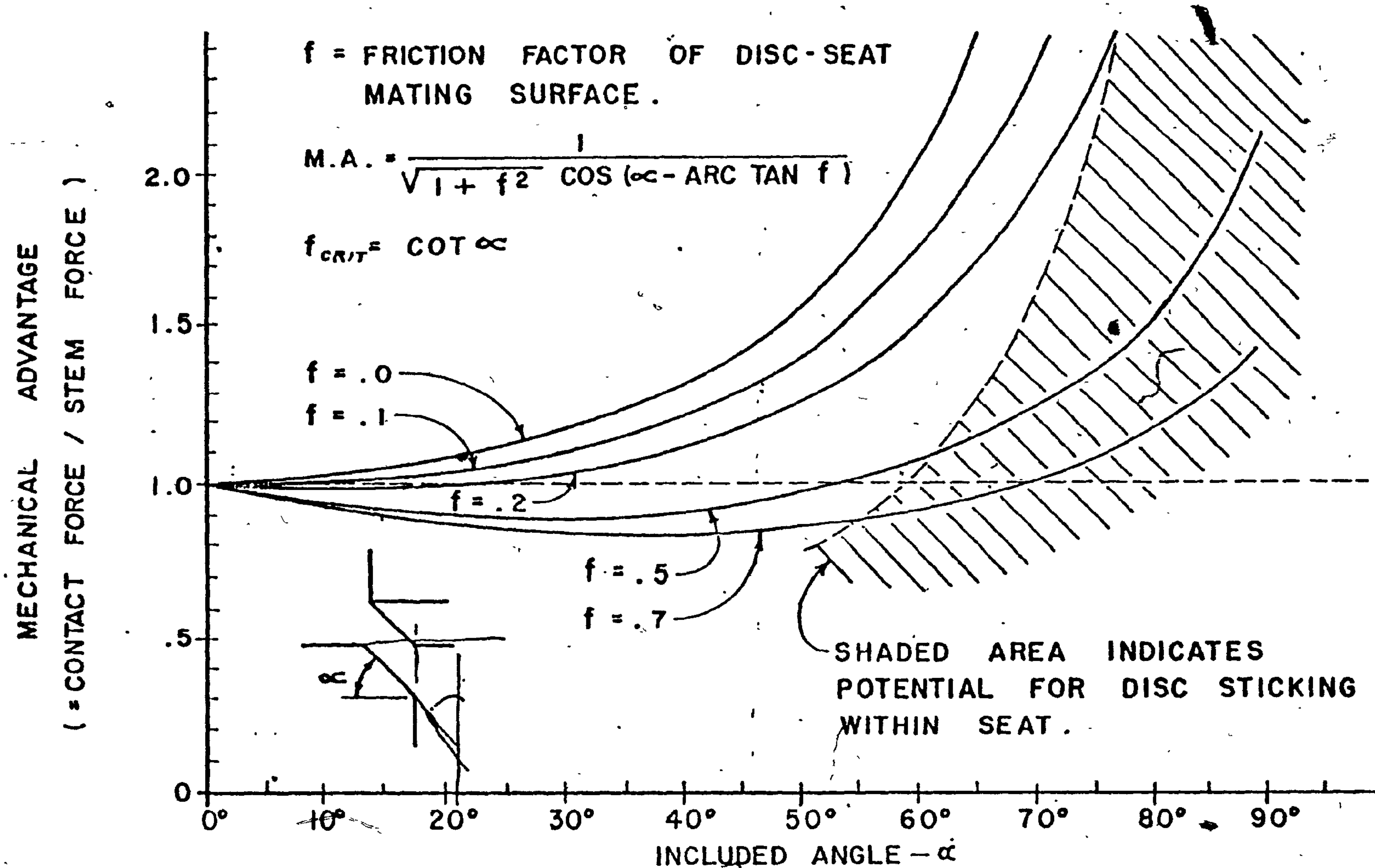


Fig. 2.11. Mechanical Advantage Obtainable With Included-Angle Globe Valve.[26]

angle is about 55^{o} to 60^{o}. It should be noted that the friction forces occurring in valve seats result from very high contact pressures. In addition, such friction factors may vary over the wear lifetime of the surfaces if disc-seat galling occurs. The results of friction tests made on some materials under high contact forces conditions indicate that while friction factors can be below 0.2 and even as low as 0.02, they will increase rapidly in the event of seizing or galling.[26]

Actual valve practice reflects these conclusions drawn from Fig. 2.11. Most included-angle globe valves have seat angles of approximately 45^{o}. This makes substantial use of the mechanical advantage while precluding the possibility of disc sticking. Nor does it put a restrictive limitation on the design of the seat for resistance to overstress under the combined influence of stem and pressure forces.

The maintenance to be performed on seats and discs should be given consideration in the design of included-angle globe valves. The cross-sectional dimensions of the seat and disc will change progressively after successive refinishing operations, as illustrated in Fig. 2.12. As may be seen, the hard-facing is applied to the seat in a manner to account for the included angle. Allowance should be made for several refinishing operations throughout the service lifetime of the valve with 10 to 20 mils removed at each operation. Line contact should be maintained throughout the lifetime of the valve. The small differential angle must be reestablished in the refinishing operations. This angle usually allows the line contact to occur on the lower or inside portion of the seat. Normal valve practice dictates that the differential angle be a positive one that allows contact on the inner edge.

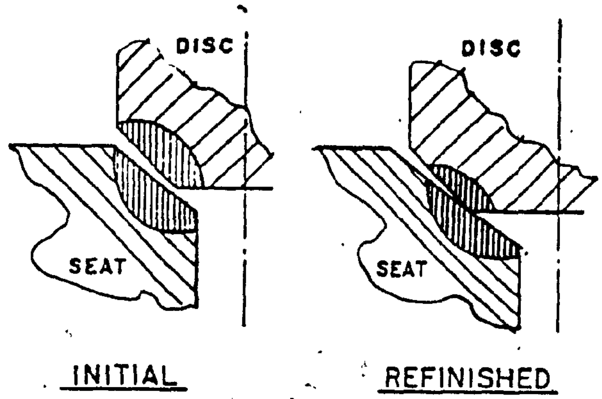


Fig. 2.12. Geometry Changes Caused By Refinishing Operations.

2.4.3 Alignment of Seat and Disc. With respect to the alignment of the disc within the seat, the area contact type of disc is inherently more self-aligning than the line-contact pressure type. The two types of line-contact included-angle seat closures are illustrated in Fig. 2.13. One type employs the knife edge of the disc to achieve line contact, and the other has a wide disc sealing face but makes use of a small differential angle between the seat and disc to establish line-contact. In either case, when the valve is being closed, the centre lines of the disc and seat must be aligned when the seal mating surfaces come into contact. If the disc guide is not sufficient to do this, the disc will not contact the seat uniformly around the periphery of the mating area and good pressure contact will not be obtained. The self-alignment properties of discs with line-contact closure are not as good as those of discs with area contact closure.

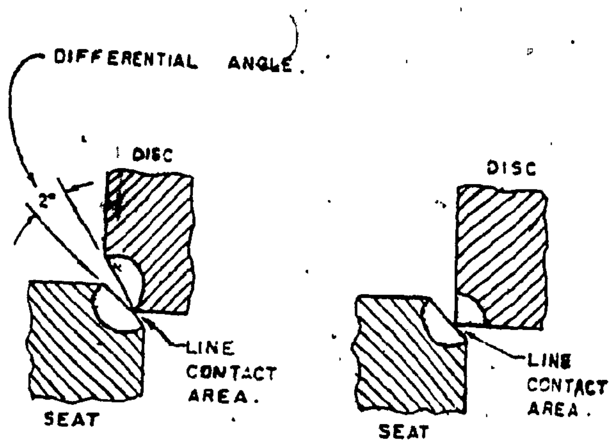


Fig. 2.13. Line-Contact Pressure Type of Included-Angle Seat Closure.

The line-contact pressure type of included-angle globe valve seat requires disc guidance of both the stem and the seat for best results.

It should be noted that one disadvantage of the line-contact included-angle seat without the differential angle is that lapping operations will tend to extend the area of contact. This tends to reduce contact effectiveness as well as the allowable time between lapping operations. Perhaps some margin could be gained by designing the seat and disc contact initially as a non-differential type of contact and proceed through lapping operations to the line-contact differential-angle type of seat and disc contact. However, this is somewhat exotic and more complicated than should be necessary.

Discs are sometimes designed to be self-aligning on the seat. This means that as the disc approaches the seat, a lower tailpiece or disc guide contacts the inner edge of the seat and aligns the disc seating area on the seat seating area, as illustrated in Fig. 2.14.

Such a disc guide is considered useful for large valves where misalignment between the body and the bonnet could cause poor seating of the disc. The disc guide normally has a slight taper and should be designed so that it cannot contribute to disc sticking within the seat. Disc guides can be used on globe and check valves but not on gate valves. Since these guides can provide an extra measure of leak tightness, especially for larger valves, greater seat tightness is usually achieved in globe valves than in gate valves.

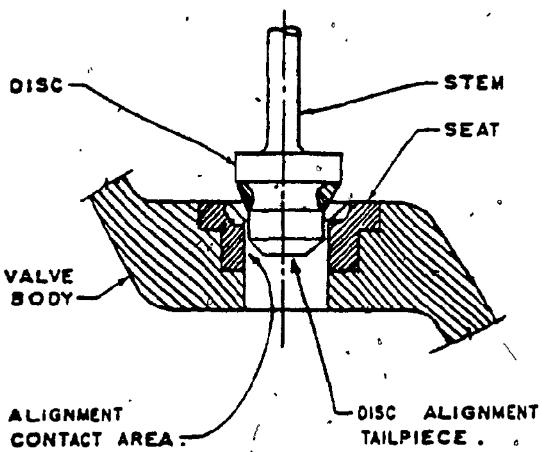


*Fig. 2.14. Disc Alignment on the Seat.

If the disc is guided by the stem, more allowance in the guide design should be given for transverse disc freedom than angular freedom. Stem transverse misalignment and slight misalignment caused by cocking of the disc require take-up in transverse freedom, as is illustrated in Fig. 2.15. It should be noted that angular misalignment between the seat and disc will always occur, although it is usually

*Note: Fisher Control Company, U.S.A.

small. The geometrical analysis of the reduction in effective contact is difficult to obtain accurately, however, this type of effect can be

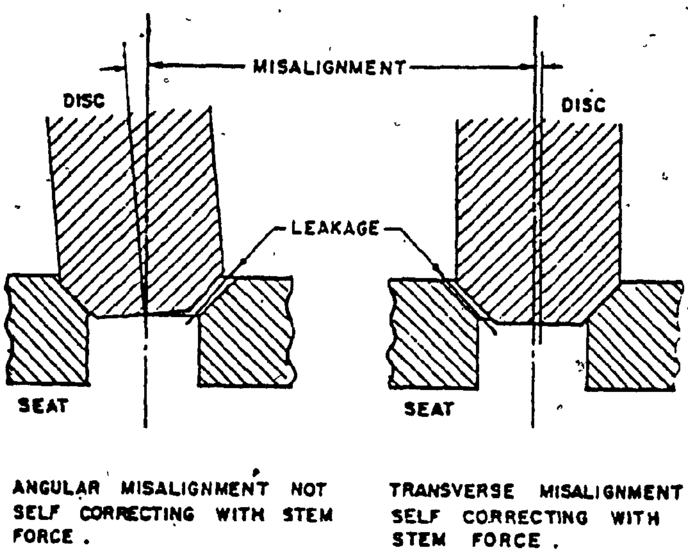


Fig. 2.15. Leakage Caused By Disc Misalignment.

easily measured in a function type of test performed during the initial valve leak-tightness qualification. If necessary, this type of test could be performed by conducting valve opening and closing cycles with the stem in a horizontal position and subsequently testing leak tightness.

The conclusions that may be drawn from the preceding discussion of seat and disc geometry are as follows. In high-pressure and high-temperature applications, the line-contact pressure type of seating with hard-facing is mandatory. An included-angle seat with a wide seating surface could reduce flow erosion, particularly at the seal contact line. Thus, a differential-angle line-contact type of geometry might be selected. The seat angle would be 45° to preclude the possibility of disc seizing and to reduce hoop stresses in the valve seat and body. This type of arrangement requires very accurate disc alignment. Therefore, the disc would be aligned on both the stem, with a slight amount of lateral allowance, and on the seat with an extended tailpiece below the seat.

2.4.4 Guidance for Valve Disc. In mechanical movement of the valve, the disc is normally moved from a position out of the flow stream to a position in the flow stream and then to a position against the valve seat. Movement of the disc must be guided to preclude vibrations, disc cocking and hangup, and to promote good and accurate seating. Within these guidelines, there is some margin for slack. Some valve designs require close-tolerance disc guide and others permit relative freedom of the disc. Whatever the valve design, there are usually very specific requirements for disc guidance.

Generally for all valves, the disc is guided either on the stem, the bonnet, the body, the seat, or on a combination of these parts. Some valves do not require disc guidance for seating purposes but merely to prevent excessive vibration of the disc while the valve is open. However, disc guidance is an important factor in the leak-tightness of globe valves. The disc guides necessary to provide good seating of the disc and proper seat motion in globe valves are discussed here. The disc guidance for proper seating in included-angle valves is also discussed.

Since globe valve discs, especially the line-contact variety, require close guidance during valve operation, guide design is an important part of the valve. In smaller globe valves, the stack-up of tolerances is not large enough to seriously misalign the bonnet from the body of the valve, and the disc can be accurately guided from the stem. Virtually all globe valve designs have the swivel disc feature; that is, the disc is allowed to rotate with respect to the stem. This is important in the design of globe valves since it has been proved that sliding contact of seating surfaces does not contribute as much to leak tightness as it does to wear of the seating surfaces. This applies regardless of the size of the valve or the material of the seating surfaces. Swivel type of seating contact is therefore recommended for globe valve seats except for the backseats where heavy wear action does not occur.

Since swivel disc design is an integral part of globe valves, the dimensional tolerances of the swivel disc connection require careful analysis. Several considerations are of utmost importance. Stem forces can be generous to prevent an excessively high contact pressure between the stem and disc. Otherwise, this area is subject to galling and seizing that result in loss of swivel action and rapid wear of the seat contact surfaces during valve operation. At the same time, the tolerances must be tight enough to provide good disc alignment and seating, especially if the disc is aligned off the stem. A careful balance of disc tolerances must therefore be obtained.

As previously noted, it is most important to preclude disc cocking and angular misalignment with the seat. A certain amount of lateral misalignment is allowable and necessary to account for bonnet misalignment with the body and to make up for whatever angular misalignment occurs. Given this fact, it becomes clear that the proper alignment can best be achieved by using a deep-disc design with the stem inserted deeply within the disc, as illustrated in Fig. 2.16. Not much angular misalignment can be expected to occur in this design and there is still some freedom for transverse of lateral misalignment.

In larger globe valves, guidance of the disc by the stem is less reliable because the stem itself may be misaligned with respect to the seat. This is brought about by the naturally larger stack-up

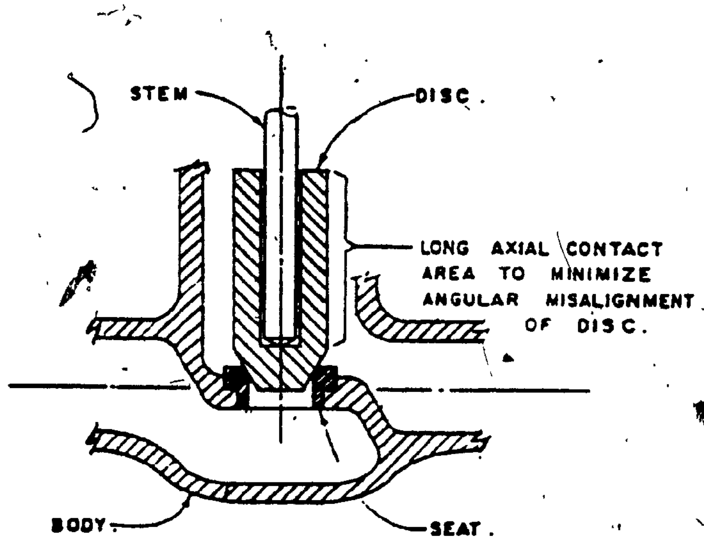


*Fig. 2.16. Design for Globe Valve Stem-Guided Disc

*Note: Rockwell Company

of tolerances between the seat, bonnet, and the stem. In these valves, other methods of guiding the disc are important. The disc may be guided by the body, bonnet, or by the seat itself. The disc may be guided off the body or bonnet of the valve by using the pressure boundary walls of the body or bonnet, as illustrated in Fig. 2.17, or by using a spindle

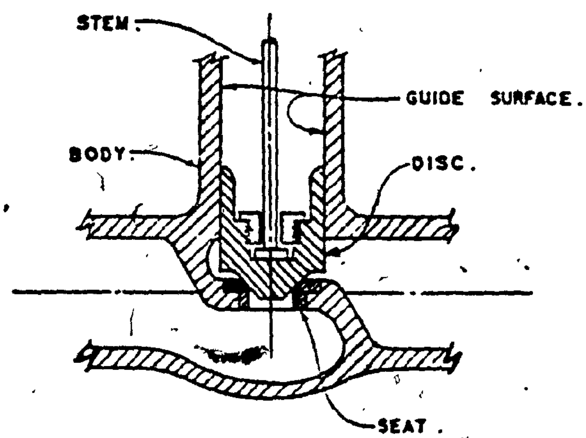


*Fig. 2.17. Design for Body-Guided Disc Globe Valve.

arrangement, as illustrated in Fig. 2.18. Problems which may arise when these types of guides are used generally involve the cocking and sticking variety. Measures which may be used to preclude these problems include the provision of deep engagement between the guide surfaces at all times. In addition, corrosion must be minimized and entrapment of corrosion particles must be reduced. Carbon-steel wear surfaces are not satisfactory in this application, and they must be supplemented with a corrosion resistant liner. Stainless steel may be more satisfactory in this application, especially if the wear surface is polished.

*Note: Rockwell Company

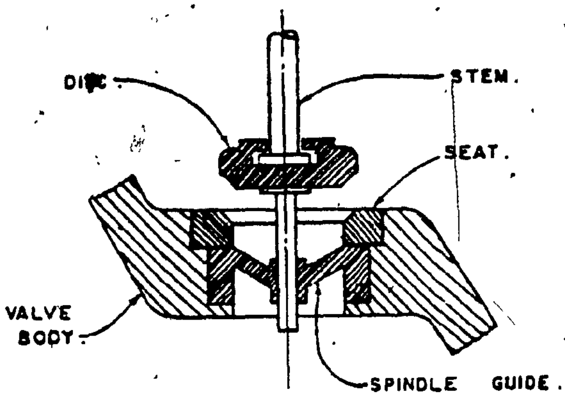


*Fig. 2.18. Spindle-Guided Disc.

Guide spindles are frequently used in throttle valves, and this spindle arrangement for disc guidance is generally quite satisfactory and trouble-free. It allows for a tight dimensional tolerance and at the same time provides for disc swivel freedom. The parts involved are not pressure boundaries and any wear does not subject the valve to the possibility of body leakage. Since the wear surfaces in this arrangement are smaller, it is not as difficult to provide hard-facing or a special liner for wear surfaces. On the other hand, the spindle type of guide is somewhat more difficult to arrange properly within the valve because of the complicated geometry involved. Unless the possibility of casting defects in the pieces can be accepted, internal fasteners and locking devices are necessary and these are undesirable because of the possibility of their coming loose and being swept into the flow stream.

Some success in disc guidance in globe valves has been achieved through the use of disc tailpieces that align off the valve seat. This

* Note: Fisher Controls Company

arrangement has the big advantage that the disc is guided off the part to which it must be aligned, as illustrated in Fig. 2.19. If the tailpiece is not deep and tapered at the end, it can be subject to cocking and sticking caused by thermal cycling. In general, good results can be obtained with this type of guide without the expense

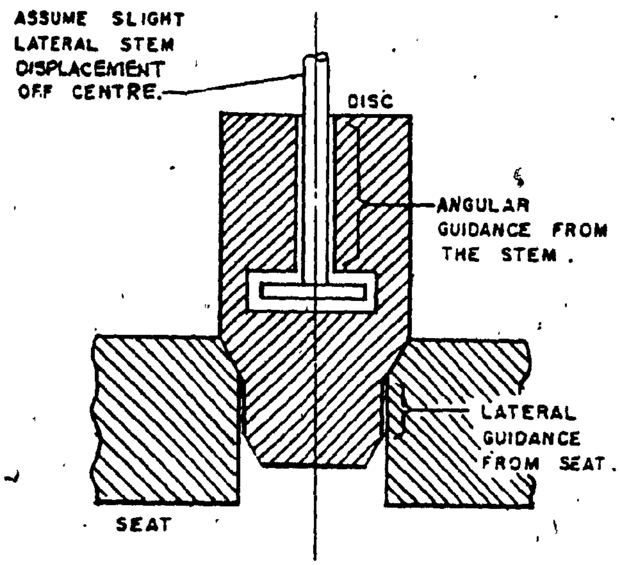


Fig. 2.19. Disc Guidance Off the Stem and the Seat.

of providing a disc spindle arrangement. However, this type of guide may not be as desirable as the spindle type of guide for trip valves since the disc tailpiece disengages and the spindle does not. Engagement during rapid tripping action may either damage the seat or increase tripping time. These considerations can largely be eliminated through proper geometrical dimensioning and tolerance design, with proper control over the dimensions during the life of the valve.

Chapter 3 THE SIZING OF CONTROL VALVES

3.1 Introduction

The need for an accurate valve sizing equation is necessary for manufacturers and users of control valves. It is recommended that when selecting a control valve for a system, its capacity calculations must meet the requirements of the actual installation. The actual sizing techniques for control valves should include the conditions "at" and "in" the valve and adjacent piping, because when the valves are mounted between pipe reducers to cut cost, there is a decrease in "inherent" valve capacity. The reducers create an additional pressure drop in the system by acting as sudden contractor or enlargement in series with valves.

In this chapter fundamentals of control valve sizing for liquids and gases will be discussed.

3.2 Sizing techniques for liquids

a) General sizing equation

The following formulae are based on Instrument Society of America's (ISA)-S 39.1, 1973. For explanation of symbols, see Nomenclature.

For the flow of single *Newtonian liquid through a control valve, use the general formula

$$q = N_1 F_p F_y F_R C_v \sqrt{\frac{P_1 - P_2}{G_f}} \qquad (3.1)$$

* Newtonian fluid deforms continuously under shear stresses.

* Table 3.1 Numerical constants used in the equations

N	q	C_v	P_1, P_2	d, D	ν
$N_1 = 1$	gpm	gpm $/\sqrt{psi}$	psia	....	....
$= 0.865$	m^3/h	gpm $/\sqrt{psi}$	bar	....	....
$N_2 = 890$	....	gpm $/\sqrt{psi}$	....	in.	....
$= 0.00214$	....	gpm $/\sqrt{psi}$	....	mm	....
$N_3 = 1$	....	gpm $/\sqrt{psi}$	....	in.	....
$= 645$	....	gpm $/\sqrt{psi}$	....	mm	....
$N_4 = 17{,}300$	gpm	gpm $/\sqrt{psi}$	....	in.	Centistokes
$= 76{,}200$	m^3/h	gpm $/\sqrt{psi}$	....	mm	Centistokes

b) Reynolds Number Corrections

The valve sizing coefficient (C_v) is useful in sizing valves for flow of liquids. But when viscous conditions occur, significant sizing error will occur. The coefficient C_v is determined using the standard liquid sizing equation 3.1.

This C_v, along with the flow and viscosity can be used to determine a fluid Reynolds Number Factor. F_R can be obtained from Fig. 3.1. This factor is required when non-turbulent flow conditions are established through a control valve. This is due to a low pressure drop, a high viscosity fluid or small valve C_v. The F_R factor can be determined by ISA Standard S39.2. The Valve Reynolds Number can be calculated using the following ISA formula:

$$Re_v = \frac{N_4 F_d q}{\nu \sqrt{F_P F_L C_v}} \left[\frac{F_P^2 F_L^2 C_v^2}{N_2 D^4} + 1 \right]^{\frac{1}{4}} \quad - (3.2)$$

*From ISA Standard S39.1

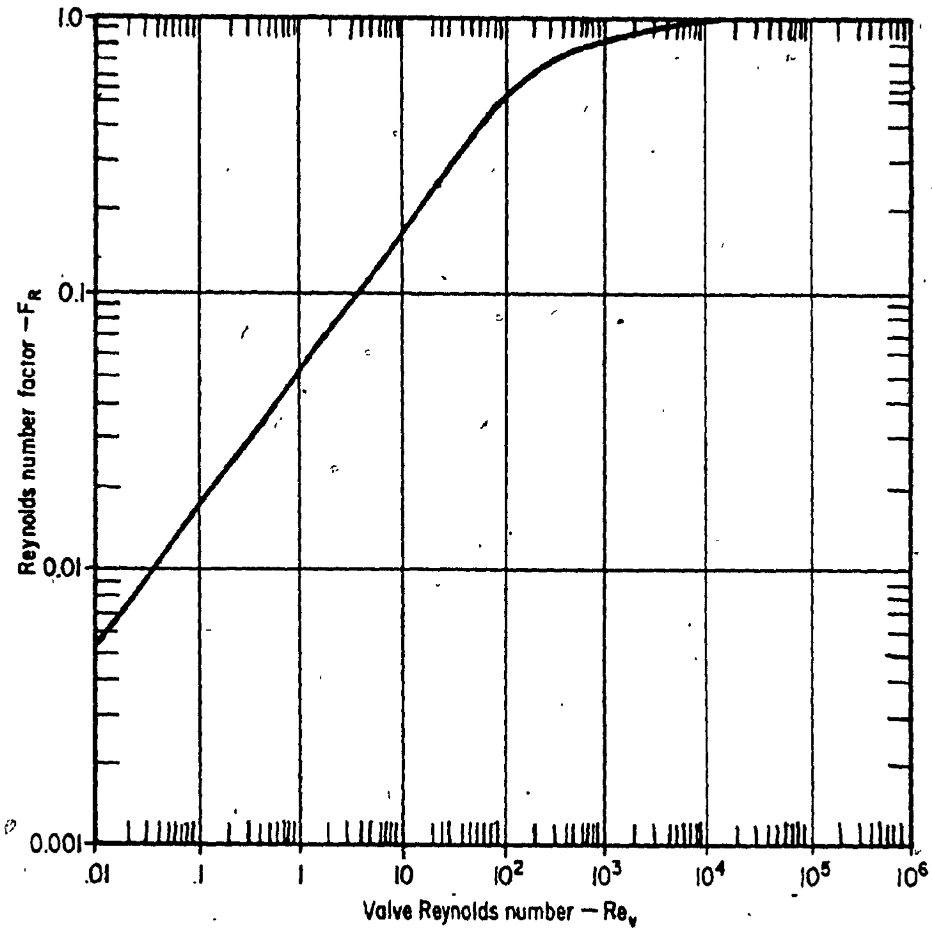


Figure 3.1: Reynolds Number Factor for Liquid Sizing [2]

The bracketed quantity in the preceding equation accounts for the "velocity of approach". Except for wide-open ball or butterfly valves, this refinement has only a slight effect on the Re_v calculations and can be neglected. Also, the Reynolds number R_e expresses the ratio of inertial forces to viscous forces in the fluid. At high Reynolds number, inertial forces predominate, and flow is proportional to tha square root of the pressure drop. At low Reynolds number, viscous forces predominate, and flow is linear with the pressure drop.

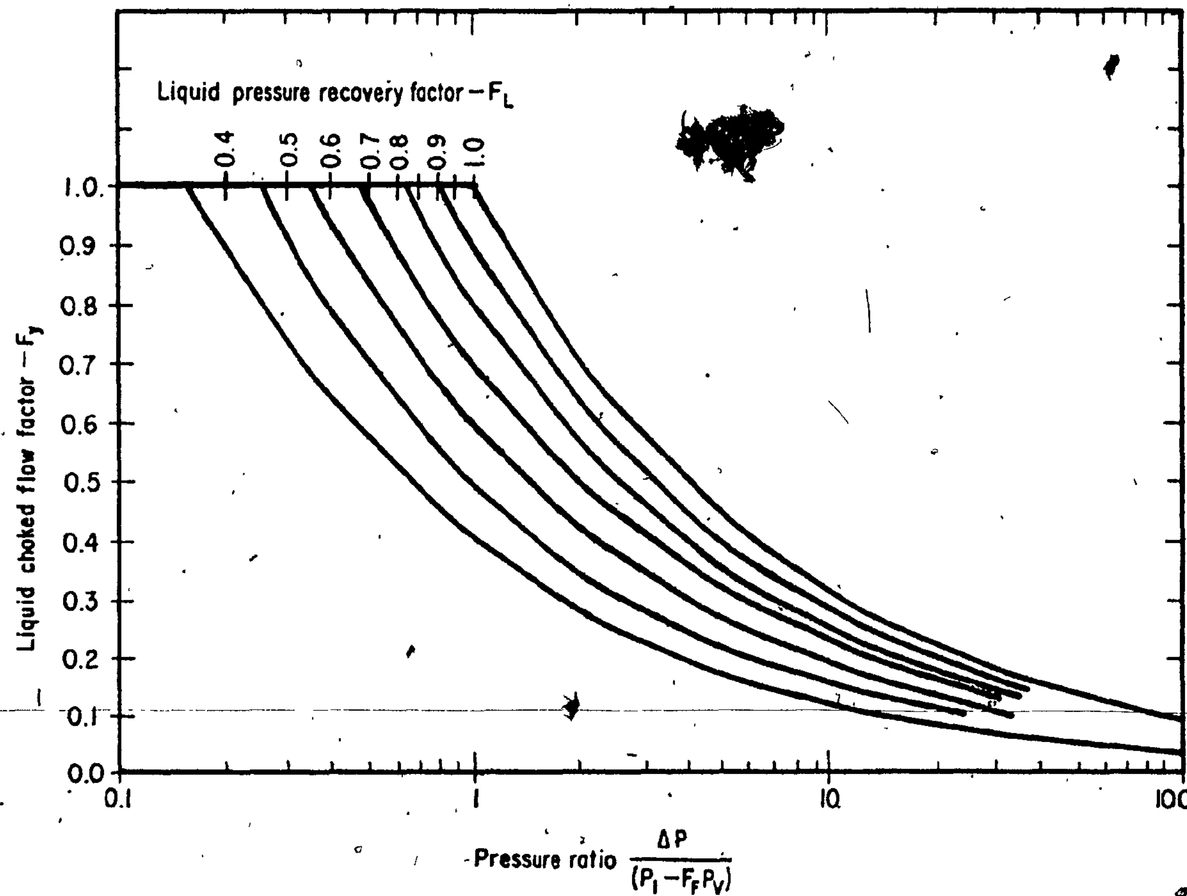


Figure 3.2: Liquid Choked Flow Factor - Fy [2]

The following formulae are suggested by Brodgesell.[24]

$$\Delta P = K\, Q_L\, \mu \quad \ldots\ldots\ldots\ldots\ldots (3.2a)$$

Such a linear flow law makes calculations simple. The complexity comes in trying to estimate the laminar flow characteristic from the given values of C_v for turbulent conditions. One method, as suggested by Liptak,[25] of viscous flow sizing, proceeds as follows.

If μ_L is the viscosity corresponding to a fictional transition point between laminar and turbulent flow, then the ratio of μ_L to μ will also be the ratio of the available and sizing pressure drops.

*TABLE 3.2 REPRESENTATIVE F_L FACTORS AT FULL RATED TRAVEL FOR FULL SIZE TRIM

Valve Type	Flow Direction	Trim Type	F_L
Single Port or Double Port Globe	Either	Ported Plug	0.85 - 0.95
Double Port Globe	Either	Contoured Plug	0.85 - 0.90
Single Port Globe	Flow-To-Open	Contoured Plug	0.85 - 0.90
Single Port Globe	Flow-To-Close	Contoured Plug	0.75 - 0.85
Angle	Flow-To-Open	Contoured Plug	0.85 - 0.90
Angle	Flow-To-Close	Venturi	0.45 - 0.55
Ball		Characterized	0.60 - 0.65
Ball		Reduced & Full Bore	0.55 - 0.60
Butterfly			0.55 - 0.65

The above values are typical and do not include variations between valve sizes, other types of trim, full or reduced ports, and manufacturers.

$$\frac{\mu_L}{\mu} = \frac{\text{available } \Delta P}{\text{Sizing } \Delta P} \quad \ldots\ldots (3.2b)$$

From the above relationship, a viscosity correction factor $F_{visc.}$ is defined:

$$F_{visc.} = \sqrt{\frac{\mu}{\mu_L}} \quad \ldots\ldots (3.2c)$$

Therefore, C_v (laminar) $= F_{visc.}\ C_v$ (turbulent) . . . . (3.2.d)

The start of laminar flow in the valve is taken at valve Reynolds number $Re_v = 500$, and the pipe flow Reynolds number Re_p corresponding to the start of laminar flow in the valve is calculated:

$$Re_{Pipe} = \frac{d}{D}\sqrt{\frac{C_d}{110}}\ Re_{valve} \quad \ldots\ldots (3.2.e)$$

Once the Reynolds number in the pipe corresponding to the start of

* ISA STANDARD

*Table 3.3 Typical values of F_p for Reducer and Expander of the same size.

d/D	N_3C_v/d^2				
	10	15	20	25	30
0.67	0.98	0.95	0.91	0.87	0.83
< 0.50	0.96	0.91	0.85	0.79	0.73

laminar flow in the pipe is known, $\mu_L = \dfrac{3160\ Q_L G}{D\ Re_p}$ . . . . .(3.2c)

With a value for μ_L, the correction factor $F_{visc.}$ can be determined from equation (3.2c). If the "VISCOUS" C_v calculated does not match the C_v of the valve picked at the start of the calculation, the procedure should be repeated for another valve size.

c) Liquid Choked Flow Factor, Fy.**

The liquid choked flow factor, Fy, covers for the effect of different valve geometries and fluid properties on choked flow. The factor Fy is the ratio of actual choked flow through the valve to the calculated flow through the valve assuming nonvapourising, incompressible flow. It may be caluclated from ISA S39.1 formula:

$$F_y = F_L \sqrt{\frac{P_1 - F_F P_v}{\Delta P}} \quad \text{. . . . . (3.3)}$$

Limiting value of Fy = 1 or read Fig. 3.2.

d) Liquid Pressure Recovery Factor, F_L.

The liquid pressure recovery factor, F_L, accounts for the influence of the valve internal geometry on the choked flow valve

* ISA STANDARD S39.1

** SEE APPENDIX C-1

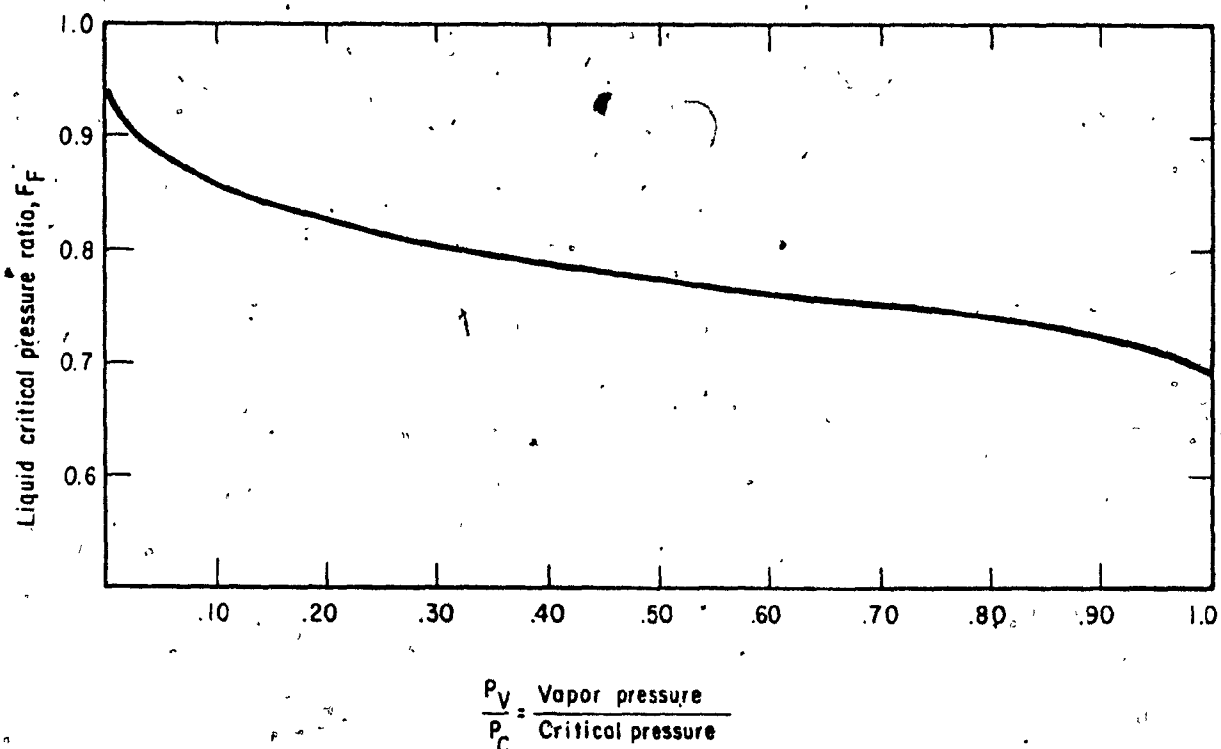


Figure 3.3 Liquid Critical Pressure Ratio F_F [2]

capacity. In table 3.2, typical values of Factor F_L at full rated travel for full size trim have been quoted by ISA standard. Also, the following equation as recommended by ISA can be used to determine F_L.

$$F_L = \sqrt{\frac{\Delta P}{P_1 - P_{vc.}}} \quad \text{--------- (3.4)}$$

The above equation is used for the test specimen under incompressible flow conditions. The factor F_L can be determined experimentally using ISA standard S39.2 formula $F_L = \frac{q_{max.}}{N_1 C_v \sqrt{P_1 - 0.96 P_v}}$ ...... (3.5)

To meet the maximum deviation of ± 5% normally required by ISA standard S39.2, F_L must be determined by testing. When the installed valve is of other than line size, and when estimated values are permissible, the following ISA formula may be used:

$$F_L F_P = F_{LP} = F_L' \left[\frac{(F_L')^2 K_i}{N_2} \left(\frac{C_v}{d^2} \right)^2 + 1 \right]^{-1/2} \quad \text{(3.6)}$$

In the above equation, K_i is the head loss coefficient $(K_1 + K_{B1})$ of the reducer or other devices between the upstream pressure tap and inlet face of the control valve.

e) <u>Piping Geometry Factor, F_p</u>

The piping geometry factor, F_p, accounts for reducer and/or expanders attached to the control valve body. The F_p factor is the ratio of flow through the valve as installed with reducers to the flow that would result if the valve was installed without reducers in a standard manifold as described in the ISA standard S39.2 and tested under identical service conditions. The pressure drop must be limited to valves such that no choking of the flow takes place.

To meet a maximum deviation of ± 5 per cent as allowed by the ISA Standard, F_p must be determined by actual test.

When estimated values are permitted, the followins ISA equation be used -

$$F_p = \frac{1}{\sqrt{1 + \frac{\Sigma K}{N_2} \left(\frac{C_v}{d^2} \right)^2}} \quad \text{(3.7)}$$

In the above equation the factor, K, is the algebraic sum of the effective velocity head coefficients of the reducers and/or expanders attached to, but not including, the valve and within the space allocated to the pressure drop of the valve. Thus K* is defined as

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2} \quad \text{(3.8)}$$

---

* SEE APPENDIX D-1

When the inlet and out fittings of the control valves are identical.

$$K_{B1} = K_{B2} \quad \text{(3.9)}$$

In those cases, where inlet and outlet fittings are different,

$$K_{B1} \text{ and/or } K_{B2} = 1-\left(\frac{d}{D}\right)^4 \quad \text{(3.10)}$$

where experimental K_1 and K_2 values are not available, the following ISA formulae can be used to calculate the approximate values:

$$K_1 + K_2 = 1.5\left(1-\frac{d^2}{D^2}\right)^2 \quad \text{(3.11)}$$

For reducers only,

$$K_1 = 0.5\left(1-\frac{d^2}{D^2}\right)^2 \quad \text{(3.12)}$$

For expander only,

$$K_2 = 1.0\left(1-\frac{d^2}{D^2}\right)^2 \quad \text{(3.13)}$$

f) Liquid Critical Pressure Ratio Factor, F_F

The ratio factor, F_F as defined by ISA standard, is the ratio of *vena contracta pressure at choked flow conditions to the vapour pressure of the liquid at the inlet temperature. Values of F_F may be determined from figure 3.3 of ISA Standard. Also, it can be calculated from the following ISA standard recommended equation,

* apparent

$$F_F = 0.96 - 0.28\sqrt{\frac{P_v}{P_c}} \quad \text{(3.14)}$$

g) Valve Style Modifier Factor, F_d

F_d is described by ISA standard as a valve modifier factor.

Test results of a number of valves show similarity of the characteristic curve of Fig. 3.1. This permits generalization of valves with different style, size or port opening (as ~~described~~ in Chapter 1) with a standard curve of Fig. 3.1. The curve represents, as per ISA

Standard an F_x value of 1.0 for V-notch ball valves and for single-ported globe valves. However, a F_d value of 0.71 may be used for valves with two parallel flow paths such as double-ported globe (see chapter 1, Fig. 1.1) and butterfly valves. Nevertheless, caution must be used in applying the figure to other valve styles since all valves may not show the same characteristics.

h) <u>Cavitation and Flashing</u>

A more common problem develops when cavitation or flashing conditions exist. These two related phenomena can place a significant limitation on the basic liquid sizing equation. Since both cavitation and flashing occur in many practical applications, they are worth considering in some detail. Both cavitation and flashing tend to limit the flow and must therefore be taken into account in order to accurately size a control valve. When either of these two phenomena are present, the basic liquid sizing equation is not a valid representation of what actually exists in the valve. Flashing of a liquid to a vapour within a valve is related to pressure recovery because a liquid static pressure exists within the valve rather than downstream. When the static pressure is reduced to the vapour pressure of the liquid, flashing occurs. If pressure recovery causes the downstream pressure to exceed the vapour pressure, vapour bubbles formed in the region of high velocity will collapse in the downstream piping. This two-stage phenomenon is termed cavitation. Physical damage to valve trim, valve body or downstream piping may occur.

With an increasing pressure drop across the valves, vapour cavities as described above will form when vena contracta pressure drops below the liquid's vapour pressure. The proportionality between increasing flow rate and increasing pressure drop to the one-half power deteriorates with the vapour cavity formation. When sufficient vapour has been formed, the flow will become completely "choked" so that there will be no increase in flow as the pressure drop (ΔP) is increased. The following ISA formulae should be used to determine the maximum allowable pressure drop that is effective in producing flow. It should be noted, however, that this limitation on the sizing pressure drop, $\Delta P_{(allow)}$, does not imply that this is the maximum drop that may be handled by the valve.

$$\Delta P_{(allow)} = F_L^2 (P_1 - F_F P_V) \quad \ldots\ldots\ldots\ldots (3.15)$$

ISA has recommended three alternative methods of sizing control valve. The first equation is for a Newtonian fluid (liquid) under non-cavitating or non-flashing conditions. The second and third equations give the maximum flow rate at choked conditions and with ISA standard and non-standard test manifolds respectively.

$$q = N_1 F_P F_R C_V \sqrt{\frac{P_1 - P_2}{G_f}} \quad \ldots\ldots\ldots\ldots (3.16)$$

$$q_{max} = N_1 F_R F_L C_V \sqrt{\frac{P_1 - F_F P_V}{G_f}} \quad \ldots\ldots\ldots\ldots (3.17)$$

$$q_{max} = N_1 F_R F_{LP} C_V \sqrt{\frac{P_1 - F_F P_V}{G_f}} \quad \ldots\ldots\ldots\ldots (3.18)$$

Equation (3.16) yields a flow rate of q which must be compared with the q_{max} determined from equations (3.17) or (3.18); which equation is used depends on the piping. The q and q_{max} values are compared and the smaller value is used to determine the size of the control valve.

3.2 Sizing techniques for gases and vapours

a) General Sizing equations

The following ISA flow equations through a control valve presented are for use with gas or vapour, and are not intended for use with gas or vapour, and are not intended for use with multi-phase streams such as vapour-liquid or gas-solid mixtures.

$$w = N_6 F_p C_v Y \sqrt{x p_1 \gamma_1} \quad \ldots\ldots (3.19)$$

$$q = N_7 F_p C_v p_1 Y \sqrt{\frac{x}{G T_1 Z}} \quad \ldots\ldots (3.20)$$

$$w = N_8 F_p C_v p_1 Y \sqrt{\frac{xM}{T_1 Z}} \quad \ldots\ldots (3.21)$$

$$q = N_9 F_p C_v p_1 Y \sqrt{\frac{x}{M T_1 Z}} \quad \ldots\ldots (3.22)$$

$$\text{where } Y = 1 - \frac{x}{3 F_k x_T} \quad \ldots\ldots (3.23)$$

$$\text{and } F_k = k/1.40 \quad \ldots\ldots (3.24)$$

In all the above formulae, x may not exceed $F_k X_T$ even though the actual pressure drop ratio is greater.

The numerical constant N is a function of the measurements used. Values for N are listed in Table 3.4

b) Expansion Factor, y and critical flow factor C_f

The factor y accounts for the change in fluid density as it passes from the valve inlet to the vena contracta. It also accounts for the change in area of the vena contracta as the pressure drop is varied (contraction coefficient).

TABLE 3·4 NUMERICAL CONSTANTS*

N	w	q*	p	Y	T	d
N_2 = 0.00214 = 890	- - - - - - - -	- - - - - - - -	- - - - - - - -	- - - - - - - -	- - - - - -	mm in
N_5 = 0.00241 = 1000	- - - - - - - -	- - - - - - - -	- - - - - - - -	- - - - - - - -	- - - - - -	mm in
N_6 = 27.3 = 63.3	kg/h lb/hr	- - - - - - - -	bar psia	kg/m^3 lb/ft^3		
N_7 = 417 = 1360	- - - - - - - -	m^3/h scfh	bar psia	- - - - - - - -	K R	
N_8 = 94.8 = 19.3	kg/h lb/hr	- - - - - - - -	bar psia	- - - - - - - -	K R	
N_9 = 2240 = 7320	- - - - - - - -	m^3/h scfh	bar psia	- - - - - - - -	K R	

*The standard cubic foot is taken at 14.73 psia and 60°F, and the standard cubic meter at 1013 millibar and 15°C.

Theoretically, y is affected by all of the following:

1) Ratio of port area to body inlet area;
2) Shape of the flow path;
3) Pressure drop, x;
4) Ratio of specific heats, k.

The influence of 1), 2) and 3) is defined by the sizing factor x_{Tm}, which may be established by air test in accordance with ISA standard S39.4.

The same critical-flow factor, C_f, is useful for flashing liquids or critical compressible flow. Results of tests by manufacturers and ISA Standard(ISA 39.2 and S39.4) with water and air show surprisingly close correlation when determining C_f. This implies the same general degree of pressure recovery for liquids, vapours or gases, because the critical-flow factor is a measure of the tendency toward pressure recovery. Table 3.5 shows critical-flow factors for typical control valve.

* ISA STANDARD

Table 3.5 Critical flow factors for control valves at 100% lift. (C_f) [6]

Body	Trim	Plug / condition	C_f
Double-port, globe body	A	Parabolic plug	0.90
	A	V-Port plug	1.00
	B	Parabolic plug	0.62
	B	V-Port plug	0.95
Single-port, globe body	A	Flow to close plug	0.85
	A	Flow to open plug (Parabolic plug only)	0.90
	B	Flow to close plug	0.50
	B	Flow to open plug (Parabolic plug only)	0.90
Split body	A	Flow to close plug	0.80
	A	Flow to open plug (Parabolic plug only)	0.75
	B	Flow to close plug	0.50
	B	Flow to open plug (Parabolic plug only)	0.90
Angle body	A	Flow to close plug	0.48
	A	Flow to open plug (Parabolic plug only)	0.90
	B	Flow to close plug	0.55
	B	Flow to open plug (Parabolic plug only)	0.95
Butterfly	D/d=1	α=60°	0.68
	D/d=1	α=90°	0.58
	D/d=2	α=60°	0.62
	D/d=2	α=90°	0.50

(A) Full capacity trim, orifice diameter ~ 0.8 valve diameter

(B) Reduced capacity trim, 50% of (A) and less.

Note: The listed values apply for equal port area valves only and do not include corrections for pipe friction.

From table 3.5 it is evident that a low number implies high pressure recovery, whereas a number approaching unity implies very little recovery.

Baumann suggested the following formula: [25]

$$q_g = 1360\, C_f\, C_v \sqrt{\frac{P_1 - P_2}{GT}} \sqrt{\frac{P_1 + P_2}{2}} \quad \ldots\ldots\ldots\ldots (3.25)$$

which accurately predicts the flow rate at critical compressible flow, but some measurable error exists in the transition from early incompressible flow at a very low pressure differential to critical flow. The following formula was introduced by Buresh and Schuder [28] to eliminate the error mentioned above:

$$q_g = \sqrt{\frac{520}{GT_1}}\, C_1\, C_2\, C_v\, P_1 \sin\left[\frac{3417}{C_1}\sqrt{\frac{\Delta P}{P_1}}\right]_{deg} \quad \ldots\ldots\ldots (3.26)$$

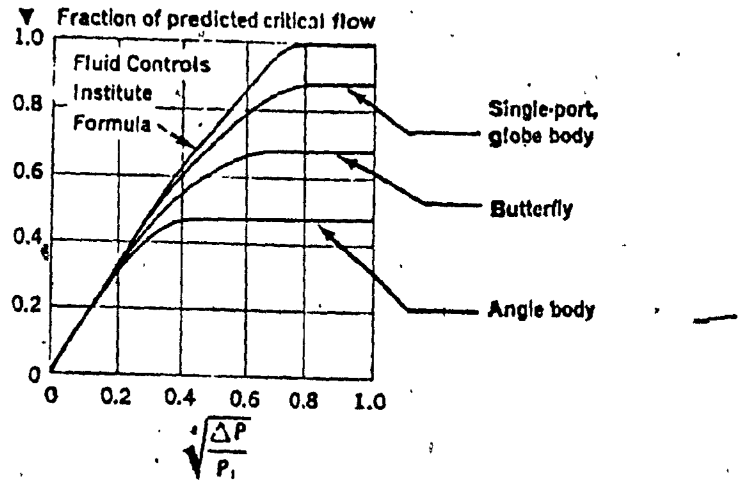


Fig. 3.4 DEVIATION of actual flows for selected valves at maximum opening when compared to flow predicted by formula. [25]

The previous equation (3.26) is an empirical expression designed to predict maximum flow, and was presented by Buresh and Schuder as the simplest empirical statement fitting closely to an accumulation of experimental data. Another form of such an empirical relation has been developed by various manufacturers and individuals. The method used is outlined here. Fig. 3.4 shows flow-test results purposely selected to illustrate the range of deviation from predicted rates of flow when using the *FCI sizing formula for compressible fluids. The FCI formula is as follows.

$$q = 963\, C_v \sqrt{\frac{(P_1 - P_2)(P_1 + P_2)}{GT}} \quad \ldots\ldots\ldots\ldots (3.27)$$

Pressure differential for critical flow has been taken as 0.5P. The control valves that produce a critical flow rate much lower than

* FCI: FLUID CONTROL INSTITUTE, U.S.A.

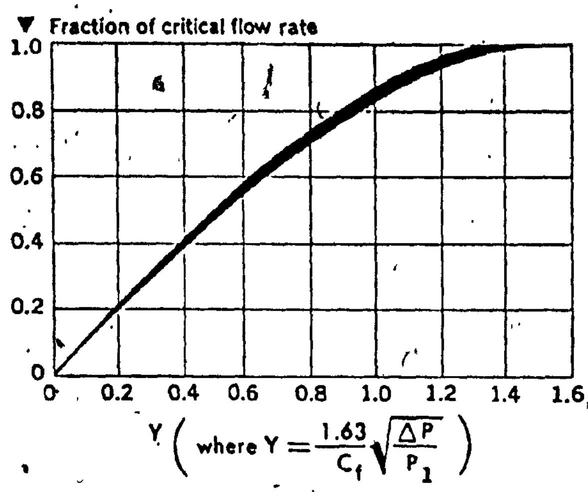


Fig. 3.5 FLOW correlation established from actual test data [25] for many valve configurations at maximum valve opening

predicted have high pressure recovery. A unified pattern of all available data relating to critical compressible flow has been established by Boger.[6] He plotted the fraction of critical flow rate f_f versus a new term y where

$$y = \frac{1.63}{C_f}\sqrt{\frac{\Delta P}{P_1}} \quad \ldots\ldots\ldots\ldots (3.28)$$

The resulting dimensional plot is shown in Fig. 3.5. The shaded area in the figure denotes the spread of test data. The least polynomial fit to the mean value was taken to establish the following relation by Liptak:[25]-

$$f_c = y - 0.148\,y^3 \quad \ldots\ldots\ldots\ldots (3.29)$$

When y has a maximum value of 1.50, the above equation becomes equal to one. However, an equation to predict critical flow rate may be

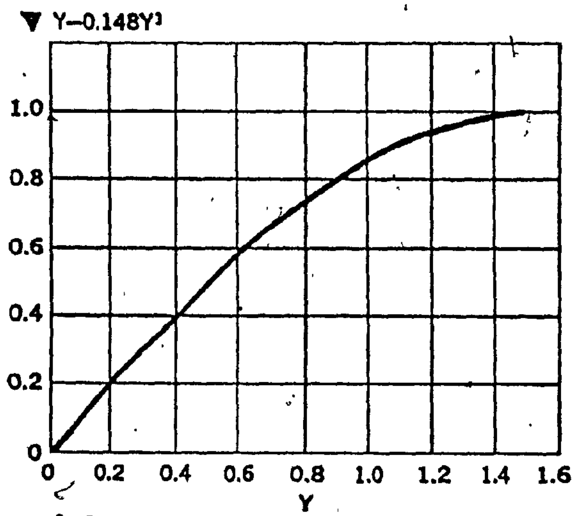


Fig. 3.6. GRAPH establishes value for correction factor. [6]

derived from the basic FCI equation (3.25) by using critical pressure drop. So derived and adding C_f, the equation becomes

$$q_g = \frac{834\, C_f C_v P_1}{\sqrt{GT}} \qquad \ldots\ldots\ldots\ldots (3.30)$$

The expression derived from Figure 3.6 may be used to modify equation (3.28) for subcritical flow. The completed empirical equation now becomes

$$q_g = 834 \frac{C_f C_v P_1}{\sqrt{GT}} (y - 0.148 y^3) \qquad \ldots\ldots\ldots\ldots (3.31)$$

Both equations (3.33) and (3.28) require a preliminary calculation and inspection of the result before continuing with the complete solution. Although equation (3.30) can hardly be considered of less complexity, it does contain C_f rather than a new factor.

Again, a correlation of factors may be drawn. [27]

$$C_1 = 36.5\, C_f \qquad \ldots\ldots\ldots\ldots (3.32)$$

Tests by various manufacturers and ISA standards have shown that critical-flow factor C_f determined with cavitating water flow and critical air flow, are similar enough to be accepted as one factor for present purposes.

c) <u>Pressure drop ratio factor, x_T</u> [2]

If the inlet pressure (P_1) is held constant and the outlet pressure (P_2) is progressively lowered, the mass flow rate through a control valve will increase to a maximum limit. A further reduction in P_2 will produce no further increase in flow. This limit is reached when x reaches a value of $F_k X_T$. The value of x is used in any of the control valve ISA sizing equations (3.19) through (3.23) must be held to this limit even though the actual pressure drop is greater. This means that numerically the value of y ranges between 1.0 and 0.667 and the slope of the y versus x curve is established by the product $F_x X_T$.

If a valve is installed with reducers, expanders, or other devices attached to it, the value of X_T is affected. To meet the specified tolerance limitation, the valve and attached fittings must be tested as a unit. When calculated values are acceptable, the following ISA formula may be used:

$$x_{TP} = \frac{x_T}{F_P^2}\left[1 + \frac{x_T K_i}{N_5}\left(\frac{C_v}{d^2}\right)^2\right]^{-1} \quad \ldots\ldots\ldots\ldots\ldots (3.33)$$

In this equation, x_{TP} and x_T are the installed value and rated value respectively of the pressure drop ratio factor, and K_i is the inlet velocity head coefficient ($K_2 + K_{B1}$) of the reducer or other device between the upstream pressure tap and the inlet face of the control valve.

d) Ratio of specific heats factor, F_k [2]

The factor x_T is based on air as the flowing fluid. Air has a ratio of specific heats k of 1.40. If the flowing medium differs in this respect, the factor F_K is used to adjust x_T for the flowing fluid. Both theoretical and experimental evidence indicate that a linear correction is adequate for control valve sizing calculations, and that $F_k \simeq \frac{K}{1.40}$.

e) Compressibility factor, Z

The ISA equations (3.20), (3.21), (3.22) do not contain a term for the actual specific weight of the fluid of upstream conditions. Instead, this term is derived from the laws of perfect gases, based on the inlet temperature and pressure. Under certain conditions, deviation from these laws can result in gross error. The compressibility factor must be included to rectify this error. The typical value can be found from Fig. 3.7.

3.3 Sizing Application Procedure Recommendation

A method of control valve sizing for liquids and gas that is as simple as possible is recommended. The following preliminary procedures are recommended:

For liquids - by observation, it may be evident whether flashing exists. If there is any doubt or change in data available, calculation can be made by ISA equation, $\Delta P_{(allow)} = F_L^2 (P_1 - F_F P_V)$ . . . . . .(3.34)

or equation suggested by Boger $\Delta P_m = (C_f)^2 (P_1 - P_v)$ . . . . .(3.35) [6]

If the actual ΔP is less than $\Delta P_{(allow)}$, use the ΔP. If actual ΔP exceeds $\Delta P_{(allow)}$, use $\Delta P_{(allow)}$, to prevent cavitation damage, the maximum allowable pressure differential should be limited to $\Delta P_{(allow)}$. This may be accomplished, in some instances, by selecting a control valve with a high critical flow factor C_f or by installing control valves in series.

<u>For Gases</u> –for most purposes, the use of the basic FCI sizing formula, together with equation (3.28) for critical flow, result in acceptable accuracy. By inspecting, it may be evident whether or not critical flow is indicated. If there is any doubt, calculate the control pressure drop ΔP_c as suggested by Boger,[6]

$$\Delta P_c = 0.5(C_f)^2 P_1 \qquad (3.36)$$

If $(P_1 - P_2)$ is less than ΔP_c, use the actual ΔP to calculate C_v. If $(P_1 - P_2)$ exceeds ΔP_c, then use equation (3.28) for calculating critical flow, and divide the resulting C_v by C_f to increase the required C_v. If greater accuracy is desired in the transition region, equation (3.26) may be used. Calculate y as per equation (3.28). If y is less than 1.50, compare with Fig. 3.6, which shows graphically the relation of $(y - 0.148y^3)$ to Y. Substitute this value for $(y - 0.148y^3)$ in equation (3.26).

travel always produces the same equal percentage change in the existing flow.

4.4 Modified Parabolic Characteristic

A valve with a modified parabolic flow characteristic is sometimes referred to as a throttle plug. Figure 4.2 shows that the flow increases rather slowly at small travels, but progressively increases more rapidly as the travel increases. In this respect, it is similar to the equal percentage flow characteristic, but the rate of flow increase is more moderate.

The valve trim parts are contoured so that the exposed flow area varies as a parabolic function of the valve travel.

Assuming that the flow is proportional to the flow area, the flow equation can be expressed as: [19]

$$Q = kx^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (4.5)$$

The slope of this flow curve maintains the same constant proportionality to the travel at every flow condition.

The valve plug can be shaped so that the area exposed to flow is a parabolic function of the travel, but in real life, the flow is not necessarily proportional to the area. This results in an actual flow characteristic which departs somewhat from a true parabolic relationship, thus giving rise to the name modified parabolic. As mentioned earlier, the modified parabolic flow characteristic is now used less commonly than in the past.

4.5 Linear Characteristic

Proper contouring of the valve trim parts can maintain the linear flow characteristic throughout the full travel of the valve. This contouring reduces the quick opening aspects of the valve so that less flow area per increment of travel is obtained in the low flow region. This accounts for the slope of the linear flow curve in Figure 4.2 being less than that of the quick opening curve. Note, however, that the slope of the linear flow curve is the same for the entire travel. Equal increments of valve travel result in equal increments of flow.[19] An equation that expresses this linear flow characteristic is:

$$Q = kx \qquad (4.6)$$

where

Q = Flow rate

x = Valve travel.

k = A proportionality constant which depends upon the units for Q and x.

4.6 Recommendation

Some guidelines that will help in the selection of the proper flow characteristics have been given in Table 4.1. It is important to remember, however, that there will be occasional exceptions to most of these recommendations and that a positive result is possible only by means of a complete dynamic analysis of the control loop. It should also be noted that where a linear characteristic is recommended, a quick opening control valve plug could be

Flow Control Processes

FLOW MEASUREMENT SIGNAL TO CONTROLLER	LOCATION OF CONTROL VALVE IN RELATION TO MEASURING ELEMENT	BEST INHERENT CHARACTERISTIC	
		Wide Range of Flow Set Point	Small Range of Flow but Large ΔP Change at Valve with Increasing Load
Proportional To Flow	In Series	Linear	Equal-Percentage
	In Bypass*	Linear	Equal-Percentage
Proportional To Flow Squared	In Series	Linear	Equal-Percentage
	In Bypass*	Equal-Percentage	Equal-Percentage

*When control valve closes, flow rate increases in measuring element.

Pressure Control Systems

Application	Best Inherent Characteristic
Liquid Process	Equal-Percentage
Gas Process, Small Volume, Less Than 10 ft. of Pipe Between Control Valve and Load Valve	Equal-Percentage
Gas Process, Large Volume (Process has a Receiver, Distribution System or Transmission Line Exceeding 100 ft. of Nominal Pipe Volume) Decreasing ΔP with Increasing Load, ΔP at Maximum Load > 20% of Minimum Load ΔP	Linear
Gas Process, Large Volume, Decreasing ΔP with Increasing Load, ΔP at Maximum Load < 20% of Minimum Load ΔP	Equal-Percentage

Liquid Level Systems

Control Valve Pressure Drop	Best Inherent Characteristic
Constant ΔP	Linear
Decreasing ΔP with Increasing Load, ΔP at Maximum Load > 20% of Minimum Load ΔP	Linear
Decreasing ΔP with Increasing Load, ΔP at Maximum Load < 20% of Minimum Load ΔP	Equal-Percentage
Increasing ΔP with Increasing Load, ΔP at Maximum Load < 200% of Minimum Load ΔP	Linear
Increasing ΔP with Increasing Load, ΔP at Maximum Load > 200% of Minimum Load ΔP	Quick Opening

[19]

Table 4.1. Illustration of Control Valve Flow Characteristics

used, and while the controller will have to operate on a wider proportional band setting, the same degree of control accuracy may be expected.

Chapter 5 CONTROL VALVE ACTUATOR

5.1 Introduction

The prime function of a valve actuator is to move the stem under all conditions of opposing force. The force required to move the stem vary greatly and are dependent mainly on the system pressure, fluid motion, and the type and tightness of the packing gland.

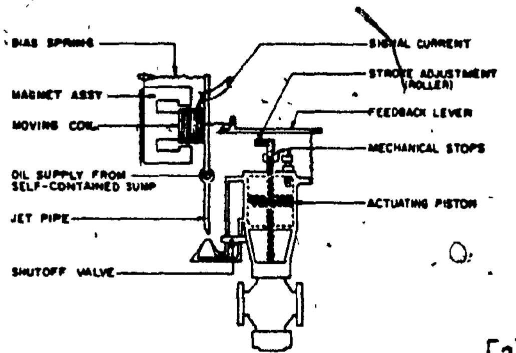
The frequently used types of remote actuators for large valves are pneumatic and electric motor actuator. Although hydraulic and electro-hydraulic actuators are rapidly gaining ground.

Hydraulic and pneumatic actuators can be controlled by the operating pressure of the fluid and the area of the operating piston or diaphragm. Motor actuators, which act through gear drives on the stem, are controlled by limit torque switches that are actuated by valve position and developed torque.

It is essential to ensure that these devices are reliable, to avoid damage to the valve or actuator. When closing and opening speeds are high, the valve design must have an adequate provision for shock loading. It may be important to provide a means of reducing the high transient forces through the use of energy absorbing devices. Dashpots or springs may be used to reduce the velocity just prior to disc contact with the seat. Three basic types of actuators have been discussed in this chapter.

5.2 Electro-hydraulic actuatot

A typical electro-hydraulic valve actuator has been shown in Fig. 5.1



[2]

Figure 5.1 AN ELECTRO-HYDRAULIC VALVE ACTUATOR.

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 or when combined with an electro-hydraulic valve actuator, can make a single control loop without any additional 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 5.1, 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.

5.3 Electric Motor Actuator

Electric motor actuators are suited to those applications where the energy requirements to operate the valve are large and a slower operating time is acceptable to the fluid system. Electric motor actuated control valve has been described below in Fig. 5.2

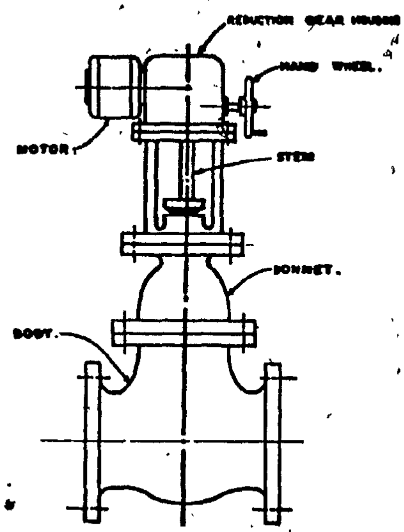


Fig. 5.2 Electric Motor Actuated Control Valve [2]

Electric motor actuators consist of a reversing motor, reduction gearing, and the various switches to start and stop the motor.

The limit switches are capable of limiting travel in both the open and close directions, and most actuators are also equipped with torque switches capable of limiting the applied stem thrust in both opening or closing. This design feature is for the protection of the valve and actuator should any foreign matter obstruct movement of the valve disc.

A handwheel is provided on the valve actuator. The motor is normally disconnected from the drive when the valve is operated manually and the handwheel is disconnected from the drive train when the valve is operated by motor. The drive train is provided with a "lost-motion" device that will allow the motor and gearing to pick up full speed and momentum before torque is applied to the valve stem nut. Such a device will increase the effectiveness of the unit in unseating sticky valves and it should be effective for both manual and electrical operation.

5.4 Pneumatic Diaphragm Actuator

Pneumatic diaphragm actuators are widely used in nuclear and fossil power stations for remote operations of valves.

Actuators provide high operation speeds, in a single and reliable actuator. The air supply needed to actuate the diaphragm is readily available from the plant air system, but care should be taken to operate with air which is free of moisture and oil.

Pneumatic diaphragm actuators are often used with throttling control valves, and they provide a responsive and accurate actuator for this important function.

A typical Penumatic Diaphragm Actuator and Valve Assembly is shown in Fig. 5.3.

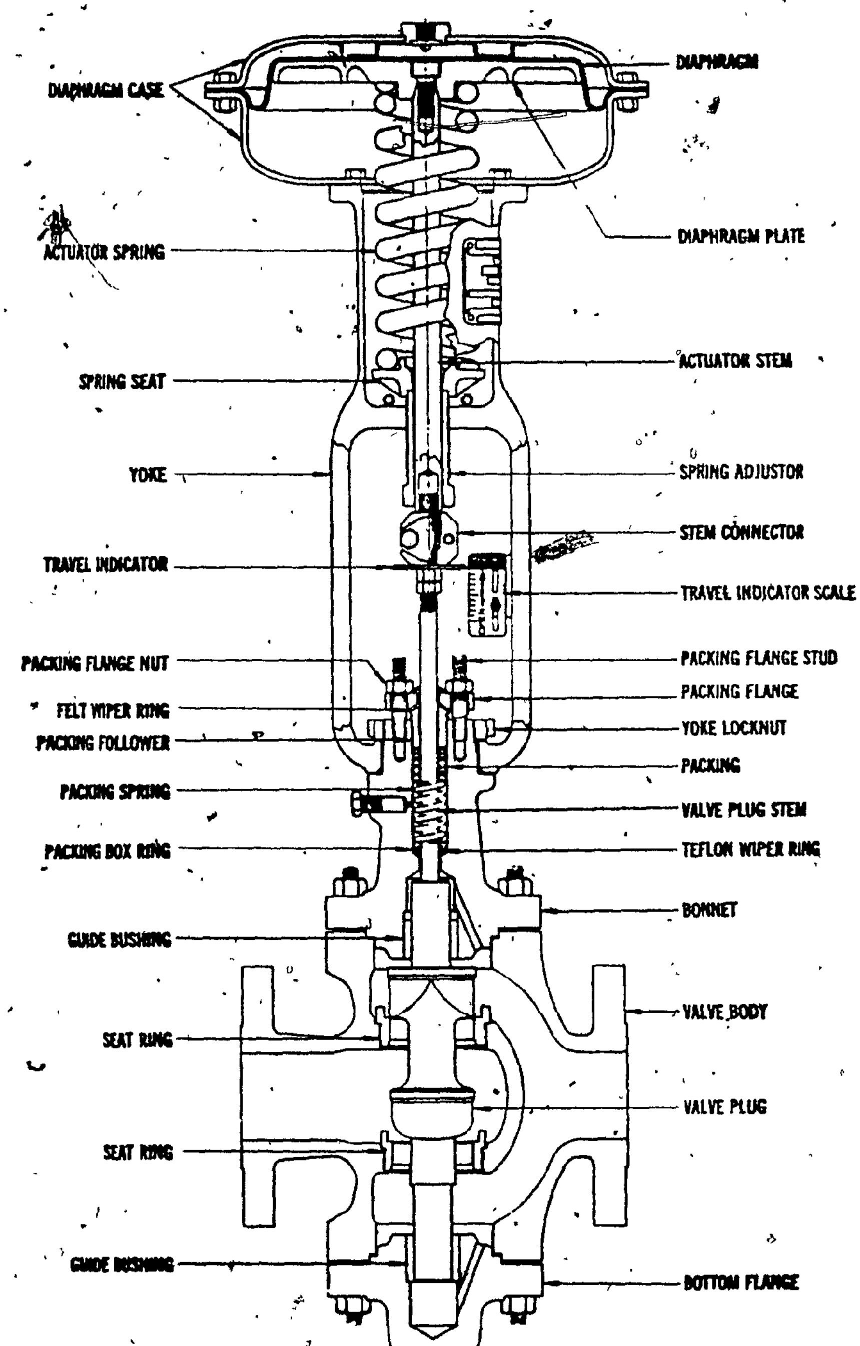


*Fig. 5.3 Pneumatic Diaphragm Actuator and Valve Assembly.

* FISHER CONTROL CO., U.S.A.

CONCLUSIONS

Any attempt to subdivide control valve into various types will be less than perfect. Nevertheless, in chapter 1, the selection into globe, ball and butterfly valves comes closest to describing the broad categories of control valves presently in use. One should appreciate that there are numerous variations within these broad categories. For example, the globe valve category includes three-way and angle style bodies and is not restricted to designs having reciprocating stem movement. There is also a definite trend towards flangeless (clamped between line flanges) body styles with either reciprocating or rotary stem motion. While these valves do not look like conventional flanged globe style designs, they fulfill the same function and have similar performance characteristics.

Early sixties, the acceptance of a ball valve as a modulating device has gained confidence. Two very distinct concepts developed in the evolution of the ball control valve. One type which evolved from the hand valve utilises a full ball, supported entirely by the seats. The second concept which developed along the lines of the concentric plug valve utilises a hollowed-out, spherical segment, supported by two stub shafts running in bearings. For the purpose of clarity, this latter design has been referred to as a characterised ball valve. The characterised ball valve is so named because a spherical segment with a V-notched or parabolic leading edge will produce an essentially equal percentage characteristic. The full ball valve differs from the characterised ball valve in that its plug is made from a solid sphere with a passage drilled through it. The ball is normally supported entirely by a pair of seat rings usually made of Teflon. The limitation of the ball depends 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 1000 pounds per square inch.

The most common type of rotary valve used for control is the butterfly valve. The typical application range as shown in Chapter 1 varies from 2 to 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 degrees opening, resulting in a Cv rating of 18. In recent designs. special shaped vanes are used to reduce the dynamic torque and to permit wide open operation of increased capacity on certain installations. The most common body design is the flangeless "wafer" type for bolting between line flanges. American National Society standard body ratings are used, but the valves are also rated for maximum pressure drop in the closed position and in the sixty degree open position. Properly selected, the butterfly valve, generally in sizes 4 inch and above, offers the advantages of simplicity, low cost, light weight and space saving, in combination with good flow control characteristics.

It is beyond the scope of this report to attempt a complete study of the seat-disc design. But a reasonable discussion of the many known technical variables which must be controlled, of the design have been discussed in chapter 2. It is clear from chapter 2 that this is a difficult and complex subject which has been solved acceptably only in a few applications. Although certain facts are well established in the valve industries, no rigorous mathematical analysis and experimental data are readily available to public. The argument put forward by the manufacturer for not publishing data is that these are proprietary information. A long range research and development effort is recommended to test various material combinations and seat-disc designs to determine limits of known technology for achieving ~~better results~~ in steam, gas, liquids and high pressure, high temperature water applications specifically for nuclear generating stations. Flow erosion, seating surface wear, corrosion, contact pressure at seating and sealing, alignment of moving and stationary parts of the valve, radiation damage effects, dimensional changes of valve parts from ambient to operating temperatures, and geometry of sealing and seating surfaces and other pertinent parameters should be quantified as to limits which determine repetitive seat-disc tightness over a prolonged period of service. The end product of this investigation is envisaged as a series of technical specification inserts for direct incorporation by the users into specifications used to purchase important key valves.

In essence, chapter 3 is the analytic section of the valve. Improper valve sizing can be both expensive and inconvenient. A valve that is too small will not pass the required flow and the process will be affected. A valve that is oversize will be unnecessarily expensive and it can also lead to instability and other critical problems. The days of selecting a valve based upon the size of the pipeline are gone forever. Selecting the correct valve size for a given application requires a knowledge of process conditions that the valve will actually see in service. The techniques for using this information for sizing valves have been discussed in chapter 3. It is apparent from chapter 3 that accurate valve sizing for liquids requires use of the dual coefficients C_v and K_m. A single coefficient is not sufficient to describe both the capacity and the recovery characteristics of the valve. Similarly, for accurate valve sizing for compressible fluid, the use of the dual coefficients C_g and C_1 is required. It can be concluded that the proper selection and sizing of a control valve for incompressible and compressible fluids is a highly technical problem with many parameters to be considered; however, accurate valve sizing can save money through improved performance and lower initial cost. Technical information, test data, sizing catalogues, nomographs, sizing slide rules, and even computer programmes are available that remove the guesswork and make valve sizing a simple, accurate procedure.

It is obvious from chapter 4, which describes the valve characterisation. As with all technical subjects, the study of valve characterisation can be as complex as one desires to make it, but the fundamental discussed in the chapter can go a long way towards helping one to select the right valve characteristic for the application. To conclude, guidelines described in the chapter that will help in the selection of the proper flow characteristic.

In Chapter 5, various basic types of actuators have been discussed. It is evident from the discussion that to work successfully with a control valve, an actuator must be able to move the valve stem continually. Dynamic performance can be important, too. Resistance to oil and water in air supply lines and to corrosion from the environment is often necessary. Special design considerations must be considered when the actuators are installed in the radioactive environment because frequent maintenance is not possible. Pneumatic actuators are commonest today, with electric-motor drive a distant second and electro-hydraulic third. In the pneumatic group, linear motion units predominate and can adapt to rotary motion to actuate 90-deg valves. Simple crank arrangement permits this. The electric-motor drive adapts more readily to rotary motion than to linear, but some linear actuators are available, chiefly for slow motion. Finally, it is very important to remember in the design stage that actuators represent extended mass (cantilevered loads), and where the actuator is heavy, there may be problems from its weight or from such outside loads as seismic shock. Nuclear valves and actuators, especially, are analysed from this standpoint.

Finally, a short range research is recommended to abate the noise generated from Condenser Steam Dump Valves (also known as Turbine by-pass Valves). Normally, steam is collected in a header before final dumping into the condenser. At the junction, where all the steam entering into the header creates an organ pipe effect. This generates noise and vibration. Both phenomena have a serious consequence on the safety of the system and personal safety. In view of the stringent requirements of the regulatory authorities, some timely fundamental research work should be conducted by the University.

REFERENCES

1. American National Standard Institute — ANSI B16,5, B,16,10,

2. Instrument Society of America — ISA Handbook of Control Valves by J.W. Hutchison.

3. F.F. Ling — "Some Factors influencing the Area-load Characteristics for semi-smooth Continuous Surfaces under Static Loading" Transactions of the American Society of Mechanical Engineers, pages 1113-1119, July 1958.

4. D.J. De Paul, Editor "Corrosion and Wear Handbook" — USAEC Report TID-7006, Bettis Atomic Power Laboratory, Westinghouse Electric Corporation, March 1957.

5. M.D. Rowe, L.A. Mitchell — "Leakage between the Surfaces in Elastic Contact", pages 203-218 in Bearing and Seal Design in Nuclear Power Machinery, The American Society of Mechanical Engineers, New York, 1967.

6. H.W. Boger — Flow Characteristics for Control Valve Installations. ISA Conference, October, 1966, New York City.

7. D.J.L. Lin, A.J. Hanssen — The Application Limits and Accuracies of Control Valve Flow Coefficient, Cv. ASME paper No. 56.- IRD-22 (Sept. 17 1956).

8. O.P. Lovett — Comparison of Control Valve Sizing Calculations, ISA paper 72.59 (Sept. 21, 1959).

9. Fluid Control Institute, Inc., U.S.A. — Quality Control Standard for Control Valve Seat Leakage - FCI 70-2 .

10. G.F. Brockett — Correlation of Valve Sizing Methods. ISA Symposium (April 1, 1952).

11. P. Wing — Practical Determination of Cv Rating. ISA Journal, September 1960.

12. FCI (November 1958) FCI 52-2 Recommended Voluntary Standards for Measurement Procedure for Determining Control Valve Flow Capacity.

13. Fisher Controls, U.S.A. Sizing and Selection Data Catalog 10.

14. W.G. Holzbock How to Select Control Valve Size. Chemical Engineering, Vol. 66, No. 8. April 20, 1959.

15. J.E. Valstar A Fresh Look at Selecting Control Valve Characteristics. Control Engineering, Vol. 6. March 1959.

16. P.S. Buckley Selection of Optimum Final Element Characteristics. Instrumentation in the Chemical and Petroleum Industries. Vol. 1. ISA, Pittsburg, U.S.A.,1964.

17. J.G. Ziegler, N.B. Nichols Valve Characteristics and Process Control. Instruments 22: 75-81. January 1949.

18. C.B. Schuder How to Size High Recovery Control Valves correctly. Instrument Technology, page 46. February 1968.

19. C.B. Schuder Control Valve Characteristics. Instruments and Control Systems. Page 79, March 1967.

20. H.W. Boger The Effect of Installed Flow Characteristic on Control Valve Gain. Paper No. 68-920, Proceedings of the 1968 ISA Conference and Exhibit. Oct. 1968.

21. G.F. Stiles, C.W. Sheldon Liquid Viscosity Effects on Control Valve Sizing.

22. D.P. Eckman Automatic Process Control, J. W. Wiley & Sons, Inc., 1958.

23. US Atomic Energy Commission — Large Globe Valves in Water-cooled Nuclear Reactor system. USA E.C. Report # 3221.

24. Brodgesell, A. — Control valve Sizing Formulas, Instrument Technology, January 1972.

25. Liptak, B.G. — Instrument Engineers' Handbook, Vol. II, Chilton Book Company, Philadelphia.

26. Lamvermeyer, D.J. and Emmons — Friction Evaluation of Materials for Nuclear Rocket Valve Application, pages 272-287 in Bearing and Seal Design in Nuclear Power Machinery, ASME, New York, 1967.

27. Baumann, H.D. — The Introduction of a Critical Flow Factor for Valve Sizing, ISA Trans., Apr. 1963.

28. Buresh, J.K. and Schuder, C.B. — The Development of a Universal Gas Sizing Equation for Control Valves, ISA Trans., Oct. 1964.

APPENDIX - A1

REPRESENTATIVE x_T FACTORS AT FULL RATED OPENING FOR FULL SIZE TRIM [2]

Valve Type	Trim Type	Flow Direction	x_T
Globe			
single port	ported plug	either	0.75
	contoured plug	flow-to-open	0.72
		flow-to-close	0.55
	characterized cage	flow-to-open	0.75
		flow-to-close	0.70
double port	ported plug	either	0.75
	contoured plug	either	0.70
Angle	contoured plug	flow-to-open	0.72
	characterized cage	flow-to-open	0.65
		flow-to-close	0.60
	Venturi	flow-to-close	0.20
Ball	characterized regular port	----	0.25
	(Diam. ≃ 0.8d)	----	0.15
Butterfly	60 Deg opening	----	0.38
	90 Deg opening	----	0.20

The Recommended Voluntary Standard Formulas for Sizing Control Valves contains the only prior U.S. standard control valve sizing formulas for compressible fluids. This work was published by the Fluid Controls Institute in 1962, and is designated FCI-62-1. In U.S. conventional units the basic gas equation is:

$$q = 963\, C_v \sqrt{\frac{(p_1 - p_2)(p_1 + p_2)}{GT}}$$

where $p_2 \geq 0.5\, p_1$

Flow predicted by this equation is almost always more than the true flow, the size of the error depending upon the x_T value of the valve under consideration. If the error is to be limited to 10 percent, the pressure-drop ratio must not exceed the following limits: [2]

Rated Valve x_T	0.80	0.75	0.70	0.60	0.50	0.40	0.30	0.20
Maximum x	1.0	0.44	0.38	0.28	0.21	0.15	0.10	0.065

APPENDIX B1

Examples of control valve sizing calculations.

Cavitation of liquid. Let us select a valve type where cavitation may occur. The fluid is steam condensate.

P_1 - inlet pressure - 167 psia

ΔP - 105 psi

T_1 - inlet temperature - $180^{\circ}F$

P_v - vapour pressure - 7.5 psia

First, if otherwise suitable, a butterfly valve is acceptable on a steam condensate condition. From Table 3.5, find $C_f = 0.68$ for a butterfly valve with 60 degree operation. Hence, by equation (3.35)

$$\Delta P_m = C_f^2 (P_1 - P_v) = (0.68)^2 (167 - 7.5) = 74 \text{ psi}.$$

Since the actual pressure drop (105 psi) exceeds the allowable drop (74 psi), cavitation will occur. Secondly, for a single-port, top-guided valve as described in Chapter 1, flow to open, find $C_f = 0.9$ from Table 3.5.

$$\Delta P_m = (0.90)^2 (167 - 7.5) = 129 \text{ psi}.$$

In this case, the allowable pressure drop (129 psi) exceeds the actual pressure drop (105 psi) by 23 per cent, which is a good margin. This valve is a better selection than a butterfly valve, because cavitation will be avoided. A double-port control valve may be used. The single-port control valve offers lower seat leakage. The double-port control valve offers the possibility of a more economical actuator, especially in larger valve size.

<u>Subcritical gas flow</u>. The following data are given -

q_g - 160,000 cu. f./hr.

P_1 - inlet pressure - 275 psia

ΔP - 90 psi

T_1 - 60^{o} F

Find valve capacity.

(i) Use the FCI formula for subcritical gas flow.

$$C_v = \frac{q_g}{1360\sqrt{\frac{\Delta P}{GT}}\sqrt{\frac{P_1+P_2}{2}}} = \frac{160\,000}{1360\sqrt{\frac{90}{520}}\sqrt{\frac{275+185}{2}}}$$

$$= 18.6$$

(ii) Use the unified gas-sizing equation for greater accuracy and assume a single-port, top-guided control valve (see chapter 1 Fig. 1.2).From Table 3.5 find C_f = 0.90.

Then from equation 3.28,

$$y = \frac{1.63}{C_f}\sqrt{\frac{\Delta P}{P_1}} = \frac{1.63}{0.90}\sqrt{\frac{90}{275}} = 1.04$$

From Figure 3.6, find $(y-0.148y^3) = 0.87$.

Therefore, $$C_v = \frac{q_g\sqrt{GT}}{834\, C_f\, P_1\,(y-0.148y^3)} = \frac{160{,}000\sqrt{520}}{834(0.90)(275)(0.87)}$$

$$= 20.4$$

This value represents an error of approximately 10% in the use of the FCI formula.

<u>Critical vapour flow</u>. The following data are given -

Fluid - saturated steam

W - 78,000 lb./hr.

P_1 - inlet pressure - 1260 psia.

P_2 - outlet pressure - 300 psia.

A heavy-duty valve is suggested for a steam pressure-reducing system application. First, assume a reduced trim for a heavy-duty angle control valve, flow to close, as per Table 3.5. $C_f = 0.55$.

Then, by equation (3.36)

$$\Delta P_c = 0.5\,(C_f)^2\,P_1 = 0.5\,(0.55)^2\,1260 = 191 \text{ psi}.$$

using the critical-flow formula (3.40) for steam given:

$$C_v = \frac{w}{1.83\,C_f\,P_1} = \frac{78000}{1.83\,(0.55)\,1260} = 61.5$$

A lower C_v could be attained by using the valve, flow to open, but a more economical choice is a single-port, top-guided control valve as described in Chapter 1; installed flow to open. Secondly, for a single-port, top-guided control valve, flow to open, we find $C_f = 0.9$ from Table 3.5. Hence

$$C_v = \frac{78000}{1.83\,(0.90)\,1260} = 37.6$$

A lower capacity is required at critical flow for a valve with less pressure recovery. Although this may not lead to a smaller control valve body size because of velocity and stability considerations, the choice of a more economical control valve body type and a smaller acuator requirement is alternative. The heavy-duty, angle valve requirement is the alternative. The heavy-duty angle valve finds its application generally on flashing-hydrocarbon liquid service with a choking tendency or condenser steam dump valves (turbine by-pass valves).

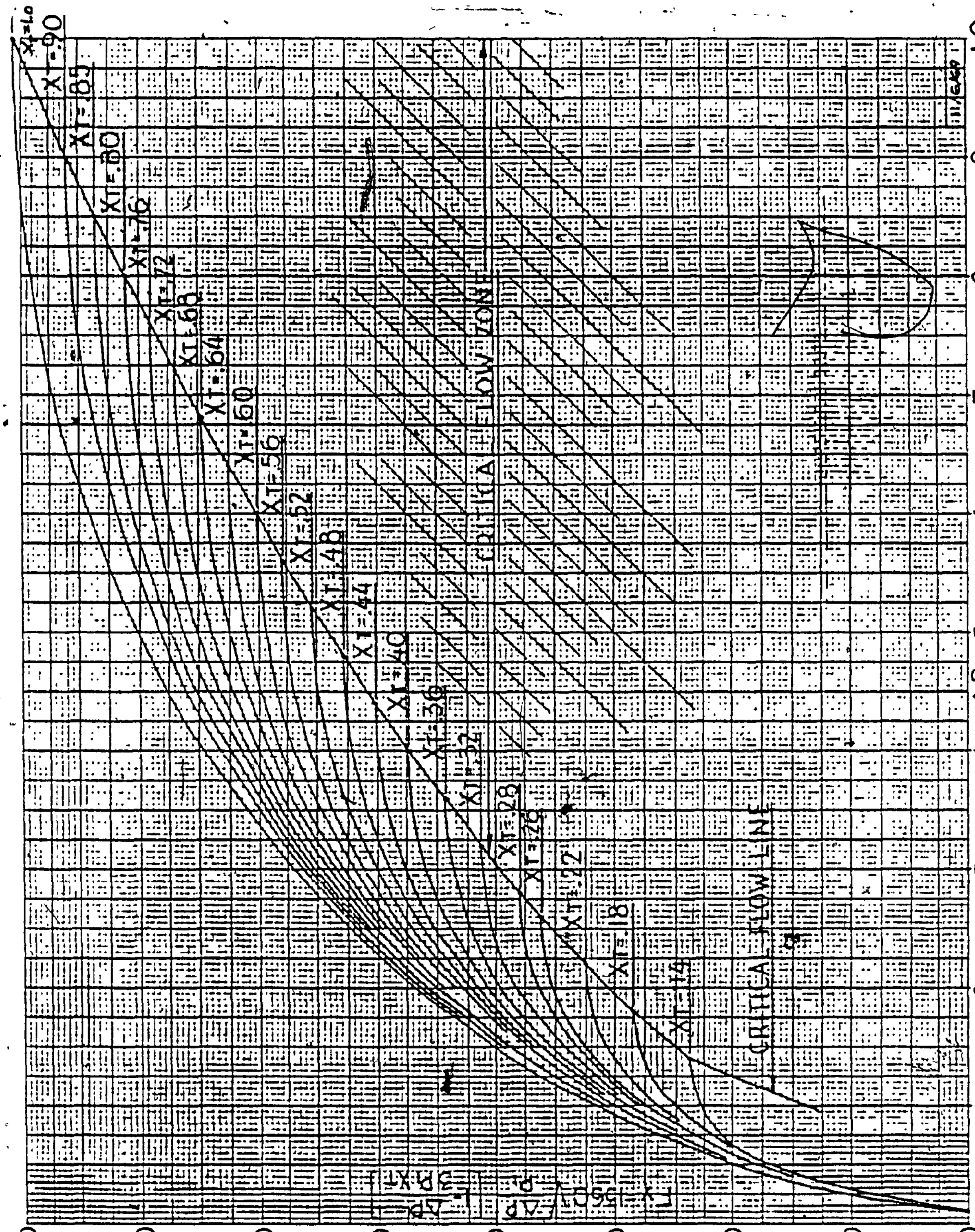


Fig. C-1 : F_y values of valves [2]

$$F_y = 1360 \sqrt{\frac{\Delta p}{P_1}} \left[1 - \frac{\Delta p}{3 P_1 X_T} \right]$$