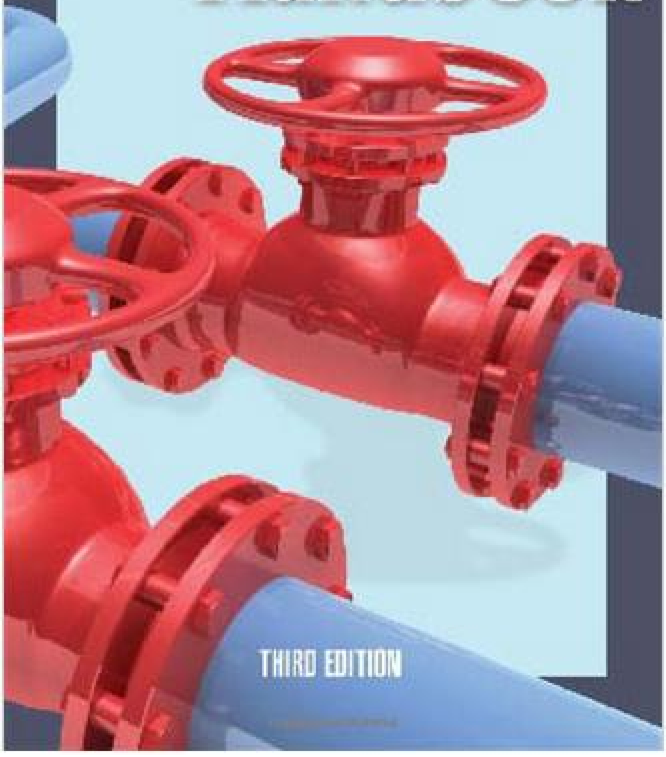
Philip L. Skousen

VALVE Handbook



THIRD EDITION

1 Introduction to Valves

1.1 The Valve

1.1.1 Definition of a Valve

By definition, *valves* are mechanical devices specifically designed to direct, start, stop, mix, or regulate the flow, pressure, or temperature of a process fluid. Valves can be designed to handle either liquid or gas applications.

By nature of their design, function, and application, valves come in a wide variety of styles, sizes, and pressure classes. The smallest industrial valves can weigh as little as 1 lb (0.45 kg) and fit comfortably in the human hand, while the largest can weigh up to 10 tons (9070 kg) and extend in height to over 24 ft (6.1 m). Industrial process valves can be used in pipeline sizes from 0.5 in [nominal diameter (DN) 15] to beyond 48 in (DN 1200), although over 90 percent of the valves used in process systems are installed in piping that is 4 in (DN 100) and smaller in size. Valves can be used in pressures from vacuum to over 13,000 psi (897 bar). An example of how process valves can vary in size is shown in Fig. 1.1.

Today's spectrum of available valves extends from simple water faucets to control valves equipped with microprocessors, which provide single-loop control of the process. The most common types in use today are gate, plug, ball, butterfly, check, pressure-relief, and globe valves.

Valves can be manufactured from a number of materials, with most valves made from steel, iron, plastic, brass, bronze, or a number of special alloys.

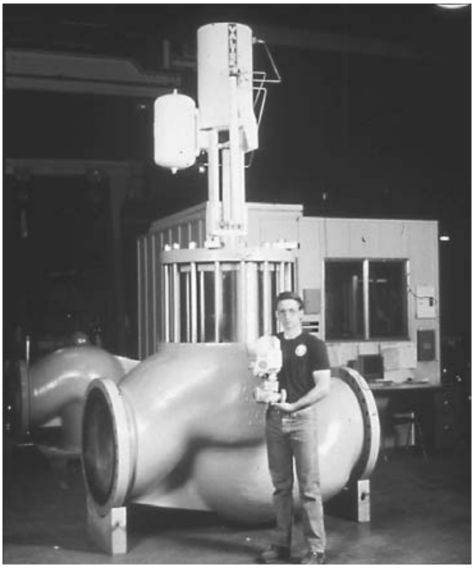


Figure 1.1 Size comparison between 30-in and 1-in globe valves. (*Courtesy of Valtek International*)

1.2 Valve Classification According to Function

1.2.1 Introduction to Function Classifications

By the nature of their design and function in handling process fluids, valves can be categorized into three areas: *on–off valves,* which handle the function of blocking the flow or allowing it to pass; *nonreturn valves,* which only allow flow to travel in one direction; and *throttling valves,* which allow for regulation of the flow at any point between fully open to fully closed.

One confusing aspect of defining valves by function is that specific valve-body designs—such as globe, gate, plug, ball, butterfly, and pinch styles—may fit into one, two, or all three classifications. For example, a plug valve may be used for on–off service, or with the addition of actuation, may be used as a throttling control valve. Another example is the globe-style body, which, depending on its internal design, may be an on–off, nonreturn, or throttling valve.

Therefore, the user should be careful when equating a particular valve-body style with a particular classification.

1.2.2 On–Off Valves

Sometimes referred to as *block valves*, on–off valves are used to start or stop the flow of the medium through the process. Common on–off valves include gate, plug, ball, pressure-relief, and tank-bottom valves (Fig. 1.2). A majority of on–off valves are hand-operated, although they can be automated with the addition of an actuator (Fig. 1.3).

On–off valves are commonly used in applications where the flow must be diverted around an area in which maintenance is being performed or where workers must be protected from potential safety hazards. They are also helpful in mixing applications where a number of fluids are combined for a predetermined amount of time and when exact measurements are not required. Safety management systems also require automated on–off valves to immediately shut off the system when an emergency situation occurs.

Pressure-relief valves are self-actuated on–off valves that open only when a preset pressure is surpassed (Fig. 1.4). Such valves are divided into two families: relief valves and safety valves. Relief valves are used to guard against overpressurization of a liquid service. On the other hand, safety valves are applied in gas applications where overpressurization of the system presents a safety or process hazard and must be vented.

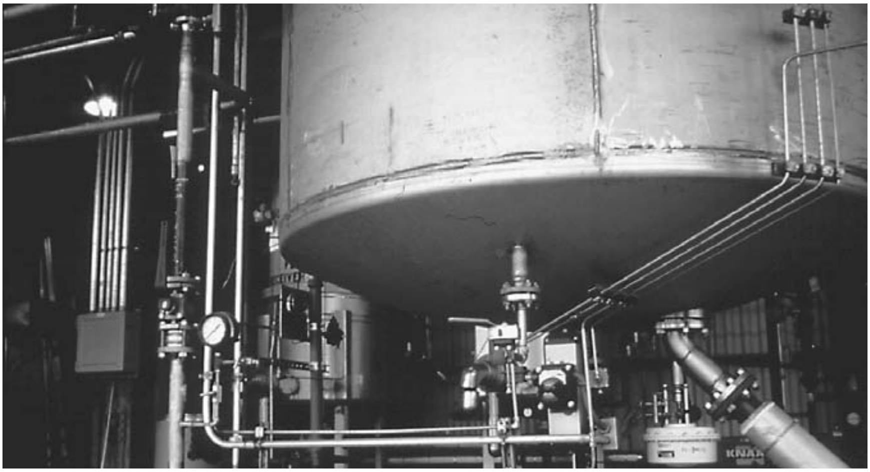


Figure 1.2 Tank bottom valve used in a steel processing application. (*Courtesy of Kammer USA*)

Figure 1.3 Quarter-turn plug valve with rack and pinion actuation system in chemical service. (*Courtesy of Automax, Inc. and The Duriron Company, Valve Division*)

Figure 1.4 Pressure-relief valve being tested for correct cracking pressure. (*Courtesy of Valtek Houston Service Center*)

1.2.3 Nonreturn Valves

Nonreturn valves allow the fluid to flow only in the desired direction. The design is such that any flow or pressure in the opposite direction is mechanically restricted from occurring. All check valves are nonreturn valves (Fig. 1.5).

Nonreturn valves are used to prevent backflow of fluid, which could damage equipment or upset the process. Such valves are especially useful in protecting a pump in liquid applications or a compressor in gas applications from backflow when the pump or compressor is shut down. Nonreturn valves are also applied in process systems that have varying pressures, which must be kept separate.

1.2.4 Throttling Valves

Throttling valves are used to regulate the flow, temperature, or pressure of the service. These valves can move to any position within the stroke of the valve and hold that position, including the full-open or full-closed positions. Therefore, they can act as on–off valves also. Although many throttling valve designs are provided with a hand-operated

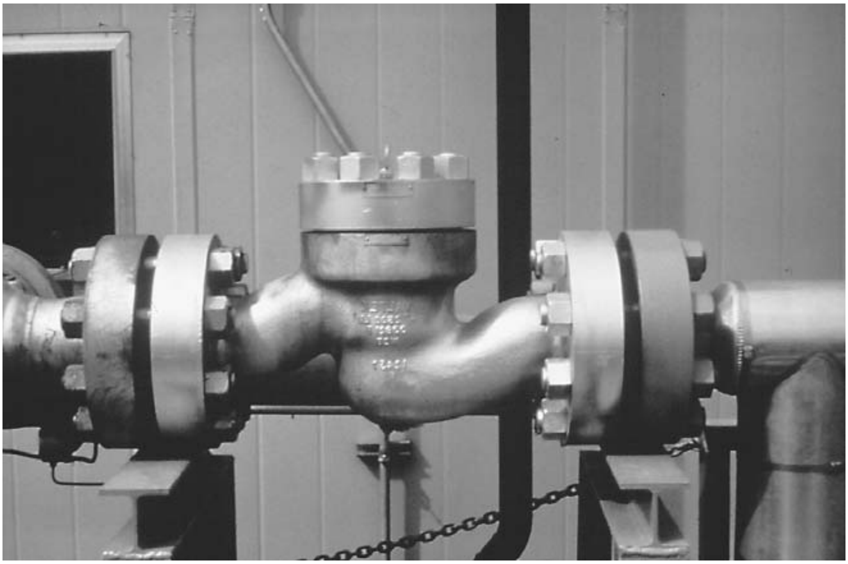


Figure 1.5 Piston check valve in natural gas service. (*Courtesy of Valtek International*)

manual handwheel or lever, some are equipped with actuators or actuation systems, which provide greater thrust and positioning capability, as well as automatic control (Fig. 1.6).

Pressure regulators are throttling valves that vary the valve's position to maintain constant pressure downstream (Fig. 1.7). If the pressure builds downstream, the regulator closes slightly to decrease the pressure. If the pressure decreases downstream, the regulator opens to build pressure.

As part of the family of throttling valves, *automatic control valves,* sometimes referred to simply as *control valves,* is a term commonly used to describe valves that are capable of varying flow conditions to match the process requirements. To achieve automatic control, these valves are always equipped with actuators. Actuators are designed to receive a command signal and convert it into a specific valve position

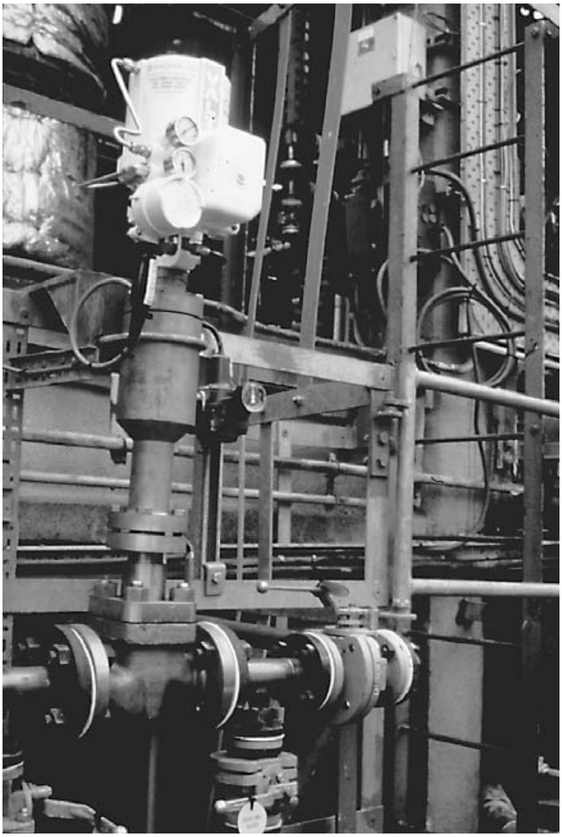


Figure 1.6 Globe control valve with extended bonnet (left) with quarter-turn blocking ball valves (right and bottom) in refining service. (*Courtesy of Valtek International*)

Figure 1.7 Pressure regulator. (*Courtesy of Valtek International*)

using an outside power source (air, electric, or hydraulic), which matches the performance needed for that specific moment.

1.2.5 Final Control Elements within a Control Loop

Control valves are the most commonly used final control element. The term *final control element* refers to the high-performance equipment needed to provide the power and accuracy to control the flowing medium to the desired service conditions. Other control elements include metering pumps, louvers, dampers, variable-pitch fan blades, and electric current-control devices.

As a final control element, the control valve is part of the *control loop,* which usually consists of two other elements besides the control valves: the *sensing element* and the *controller.* The sensing element (or sensor) measures a specific process condition, such as the fluid pressure, level, or temperature. The sensing element uses a transmitter to send a signal with information about the process condition to the controller or a much larger distributive control system. The controller receives the input from the sensor and compares it to the set point, or the desired value needed for that portion of the process. By comparing the actual input against the set point, the controller can make any needed corrections to the process by sending a signal to the final control element, which is most likely a control valve. The valve makes the change according to the signal from the controller, which is measured and verified by the sensing element, completing the loop. Figure 1.8 shows a

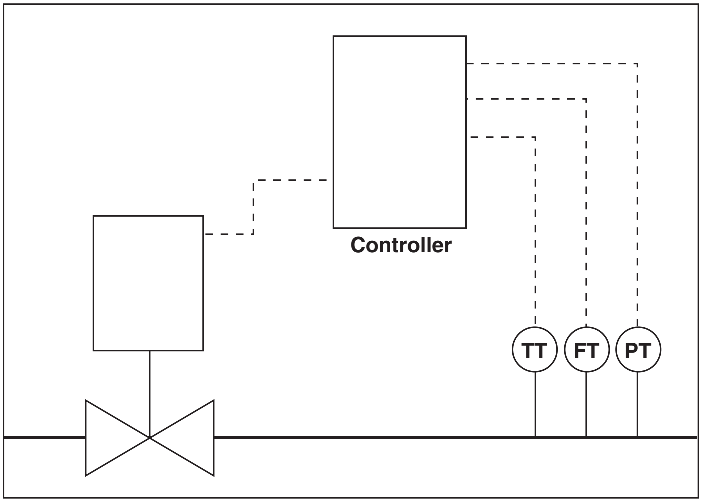


Figure 1.8 Control loop schematic showing the relationship among flow (FT), pressure (PT), and temperature (TT) transmitters, and the controller and control valve. (*Courtesy of Valtek International*)

diagram of a common control loop, which links a controller with the flow (FT), pressure (PT), and temperature transmitters (TT) and a control valve.

1.3 Classification According to Application

1.3.1 Introduction to Application Classifications

Although valves are often classified according to function, they are also grouped according to the application, which often dictates the features of the design. Three classifications are used: *general service valves,* which describes a versatile valve design that can be used in numerous applications without modification; *special service valves,* which are specially designed for a specific application; and *severe service valves,* which are highly engineered to avoid the side effects of difficult applications.

1.3.2 General Service Valves

General service valves are those valves that are designed for the majority of commonplace applications that have lower-pressure ratings between American National Standards Institute Class 150 and 600 (between PN 16 and PN 100), moderate-temperature ratings between −50 and 650°F (between −46 and 343°C), noncorrosive fluids, and common pressure drops that do not result in cavitation or flashing. General service valves have some degree of interchangeability and flexibility built into the design to allow them to be used in a wider range of applications. Their body materials are specified as carbon or stainless steels. Figure 1.9 shows an example of two general service valves, one manually operated and the other automated.

1.3.3 Special Service Valves

Special service valves is a term used for custom-engineered valves that are designed for a single application that is outside normal process applications. Because of its unique design and engineering, it will only function inside the parameters and service conditions relating to that particular application. Such valves usually handle a demanding temperature, high pressure, or a corrosive medium. Figure 1.10 shows a control valve designed with a sweep-style body and ceramic trim to handle an erosive mining application involving sand particulates and high-pressure air.

Figure 1.9 Wedge gate valves used in a blocking service to bypass general service control valves in a gasification process. (*Courtesy of Valtek International*)

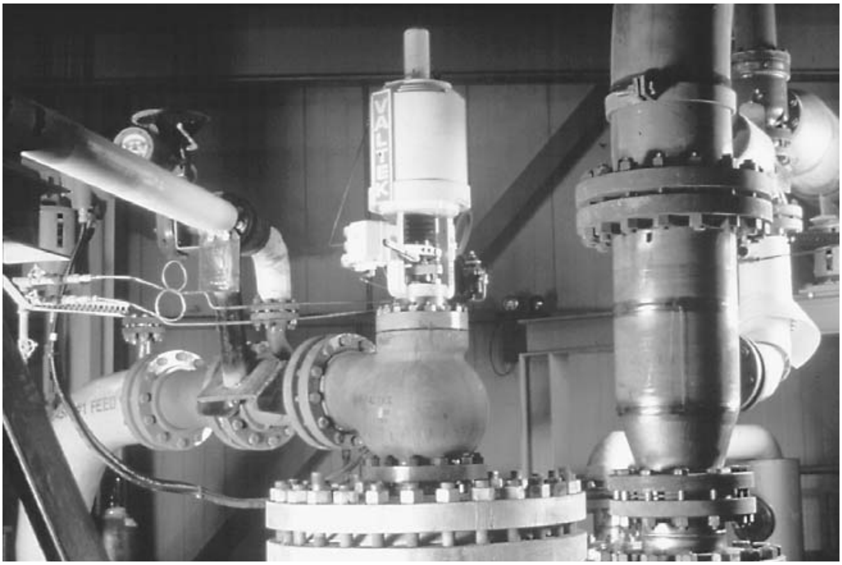


Figure 1.10 Sweep-style globe valve used in an erosive mining application involving high-pressure air and sand particulates. (*Courtesy of Valtek International*)

1.3.4 Severe Service Valves

Related to special service valves are *severe service valves,* which are valves equipped with special features to handle volatile applications, such as high pressure drops that result in severe cavitation, flashing, choking, or high noise levels (which is covered in greater detail in Chap. 9). Such valves may have highly engineered trims in globe-style valves, or special disks or balls in rotary valves to either minimize or prevent the effects of the application.

In addition, the service conditions or process application may require special actuation to overcome the forces of the process. Figure 1.11 shows a severe service valve engineered to handle 1100°F (593°C) liquid-sodium application with multistage trim to handle a high pressure drop and a bonnet with special cooling fins. The electrohydraulic actuator was capable of producing 200,000 lb (889,600 N) of thrust.

1.4 Classification According to Motion

1.4.1 Introduction to Motion Classifications

Some users classify valves according to the mechanical motion of the valve. *Linear-motion valves* (also commonly called *linear valves*) have a sliding-stem design that pushes a closure element into an open or closed position. (The term *closure element* is used to describe any internal valve device that is used to open, close, or regulate the flow.) Gate, globe, pinch, diaphragm, split-body, three-way, and angle valves all fit into this classification. Linear valves are known for their simple design, easy maintenance, and versatility with more size, pressure class, and design options than other motion classifications—therefore, they are the most common type of valve in existence today.

On the other hand, *rotary-motion valves* (also called *rotary valves*) use a closure element that rotates—through a quarter-turn or 45° range—to open or block the flow. Rotary valves are usually smaller in size and weigh less than comparable linear valves, size for size. Application-wise, they are limited to certain pressure drops and are prone to cavitation and flashing problems. However, as rotary-valve designs have matured, they have overcome these inherent limitations and are now being used at an increasing rate.

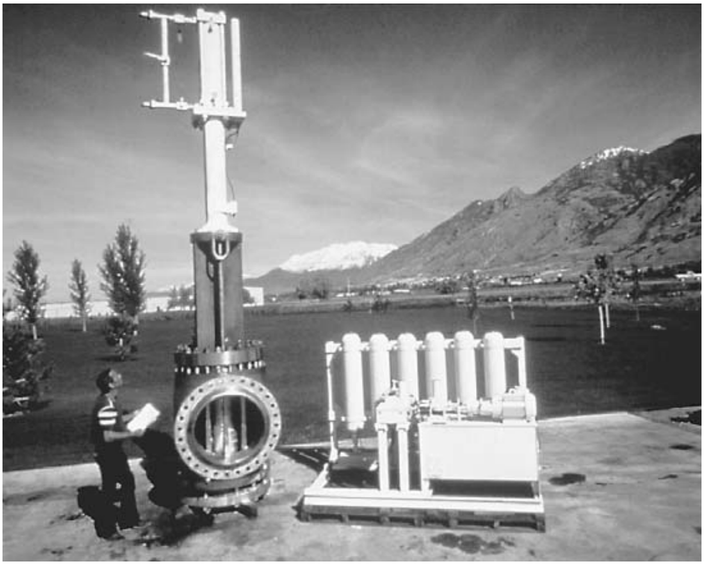


Figure 1.11 Severe service valve designed to handle high-pressure-drop, high-temperature liquid-sodium application. (*Courtesy of Valtek International*)

1.5 Classification According to Port Size

1.5.1 Full-Port Valves

In process systems, most valves are designed to restrict the flow to some extent by allowing the flow passageway or area of the closure element to be smaller than the inside diameter of the pipeline. On the other hand, some gate and ball valves can be designed so that internal flow passageways are large enough to pass flow without a significant restriction. Such valves are called *full-port valves* because the internal flow is equal to the full area of the inlet port.

Full-port valves are used primarily with on–off and blocking services, where the flow must be stopped or diverted. Full-port valves also allow for the use of a *pig* in the pipeline. The pig is a self-driven (or flow-driven) mechanism designed to scour the inside of the pipeline and to remove any process buildup or scale.

1.5.2 Reduced-Port Valves

On the other hand, *reduced-port valves* are those valves whose closure elements restrict the flow. The flow area of that port of the closure element is less than the area of the inside diameter of the pipeline. For example, the seat in linear globe valves or a sleeve passageway in plug valves would have the same flow area as the inside of the inlet and outlet ports of the valve body. This restriction allows the valve to take a pressure drop as flow moves through the closure element, allowing a partial pressure recovery after the flow moves past the restriction.

The primary purpose of reduced-port valves is to control the flow through reduced flow or through throttling, which is defined as regulating the closure element to provide varying levels of flow at a certain opening of the valve.

1.6 Common Piping Nomenclature

1.6.1 Introduction to Piping Nomenclature

Although a complete glossary is included in this handbook, the reader should be acquainted with the piping nomenclature commonly used in the global valve industry. Because the valve industry, along with a good portion of the process industry, has been driven by developments and companies originating in North America over the past 50 years, valve and piping nomenclature has been heavily influenced by the imperial system, which uses such terms as *pounds per square inch* (*psi*) to refer to pressure or *nominal pipe size* (*NPS*) to refer to valve and pipe size (in inches across the pipe's inside diameter). These terms are still in use today in the United States and are based upon the nomenclature established by the American National Standards Institute (ANSI).

Outside of the United States, valve and piping nomenclature is based on the International System of Units (metric system), which was established by the International Standards Organization (ISO). According to the metric system, the basic unit measurement is a *meter,* and distances are related in multiples of meters (kilometers, e.g.) or as equal units of a meter (centimeters, millimeters). Typically metric valve measurements are called out in millimeters and pressures are noted in *kilopascal* (kPa) (or *bar*). ISO standards refer to pipe diameter as *nominal diameter* (DN) and pressure ratings as *nominal pressure* (PN). Tables 1.1 and 1.2 provide quick reference for both ANSI and ISO standards.

Table 1.1 Nominal Pipe Size vs. Nominal Diameter*

Nominal Pipe Size (NPS) *(inches)*	Nominal Diameter (DN) *(millimeters)*
0.25	6
0.5	15
0.75	20
1.0	25
1.25	32
1.5	40
2.0	50
2.5	65
3.0	80
4.0	100
6.0	150
8.0	200
10.0	250
12.0	300
14.0	350
16.0	400
18.0	450
20.0	500
24.0	600
36.0	900
42.0	1000
48.0	1200

**Data courtesy of Kammer Valve.*

Table 1.2 ANSI Pressure Class vs. Nominal Pressure*

ANSI Pressure Class *pounds of force per square inch of surface area*	Nominal Pressure (PN) *allowable pressure in bar*
150	16
300	40
600	100
900	160
1500	250
2500	400
4500	700

Note: PN is an approximation to the corresponding ANSI pressure class, and should not be used as an exact correlation between the two standards. PN correlates to DIN (Deutsche Industrie Norme) pressure–temperature rating standards, which may vary significantly from ANSI pressure–temperature ratings.

**Data courtesy of Kammer Valve.*

2 Valve Selection Criteria

2.1 Valve Coefficients

2.1.1 Introduction to Valve Coefficients

The measurement commonly applied to valves is the *valve coefficient* (C_v), which is also known as the *flow coefficient*. When selecting a valve for a particular application, the valve coefficient is used to determine the valve size that will best allow the valve to pass the required flow rate, while providing stable control of the process fluid. Valve manufacturers commonly publish C_v data for various valve styles, which are approximate in nature and can vary—usually up to 10 percent—according to the piping configuration or trim manufacture.

If the C_v is not calculated correctly for a valve, the valve usually experiences diminished performance in one of two ways: If the C_v is too small for the required process, the valve itself or the trim inside the valve will be undersized, and the process system can be starved for fluid. In addition, because the restriction in the valve can cause a buildup in upstream pressure, higher back pressures created before the valve can lead to damage in upstream pumps or other upstream equipment. Undersized C_v's can also create a higher pressure drop across the valve, which can lead to cavitation or flashing.

If the C_v is calculated too high for the system requirements, a larger, oversized valve is usually selected. Obviously, the cost, size, and weight of a larger valve size are a major disadvantage. Besides that consideration, if the valve is in a throttling service, significant control problems can occur. Usually the closure element, such as a plug or a disk, is located just off the seat, which leads to the possibility of creating a high pressure drop and faster velocities—causing cavitation, flashing, or erosion of the trim parts. In addition, if the closure element is closure to the seat and the operator is not strong enough to hold that position, it may be sucked into the seat. This problem is appropriately called *the bathtub stopper effect.*

2.1.2 Definition of C_v

One C_v is defined as one U.S. gallon (3.78 liters) of 60°F (16°C) water that flows through an opening, such as a valve, during 1 min with a 1-psi (0.1-bar) pressure drop. As specified by the Instrument Society of America (ANSI/ISA Standard S75.01), the simplified equation for C_v is

$$C_v = \text{flow} \times m \sqrt{\frac{\text{specific gravity at flowing temperature}}{\text{pressure drop}}}$$

A step-by-step process for calculating C_v is found in Chap. 7.

2.2 Flow Characteristics

2.2.1 Introduction to Flow Characteristics

Each throttling valve has a *flow characteristic,* which describes the relationship between the valve coefficient (C_v) and the valve stroke. In other words, as a valve opens, the flow characteristic—which is an inherence to the design of the selected valve—allows a certain amount of flow through the valve at a particular percentage of the stroke. This attribute allows the valve to control the flow in a predictable manner, which is important when using a throttling valve.

The flow rate through a throttling valve is not only affected by the flow characteristic of the valve, but also by the pressure drop across the valve. A valve's flow characteristic acting within a system that allows a varying pressure drop can be much different or can vary significantly from the same flow characteristic in an application with a constant pressure drop. When a valve is operating with a constant pressure drop without taking into account the effects of piping, the flow characteristic is known as *inherent flow characteristic.* However, if both the valve and piping effects are taken into account, the flow characteristic changes from the ideal curve and is known as the *installed flow characteristic.* Usually, the entire system must be taken into account to determine the installed flow characteristic, which is discussed further in Sec. 2.2.5. Some rotary valves—such as butterfly and

Figure 2.1 Characterizable quarter-turn plug. (*Courtesy of The Duriron Company, Valve Division*)

ball valves—have an inherent characteristic that cannot be changed because the closure element cannot be modified easily. For that reason, rotary control valves in a throttling application can modify this inherent characteristic using a characterizable cam with the actuator's positioner, or by changing the shape of the closing device, such as a V-notched ball valve. Quarter-turn plug and ball valves can modify the characteristic by varying the opening on the plug (Fig. 2.1). On the other hand, linear valves usually have a flow characteristic designed into the trim, by determining either the size and shape of the holes in a cage (Fig. 2.2) or the shape of the plug head (Fig. 2.3).

The three most common types of flow characteristics are *equal percentage, linear,* and *quick-open.* The ideal curves for these three flow characteristics are shown in Fig. 2.4. However, the inherent characteristic of these curves can be affected by the body style and design, and piping factors.

Figure 2.2 Characterizable cages. (*Courtesy of Fisher Controls International, Inc.*)

2.2.2 Equal-Percentage Flow Characteristic

Of the three common flow characteristics, the equal-percentage characteristic is the most frequently specified with throttling valves. With an equal-percentage characteristic, the change in flow per unit of valve stroke is directly proportional to the flow occurring just before the change is made. With an inherent equal-percentage characteristic, the flow rate is small at the beginning of the stroke and increases to a larger magnitude at the end of the stroke. This provides good, exact control of the closure element in the first half of the stroke, where control is harder to maintain because the closure element is more apt to be affected by process forces. On the other hand, an equal-percentage characteristic provides increased capacity in the second half of the stroke, allowing the valve to pass the required flow. An equal-percent-

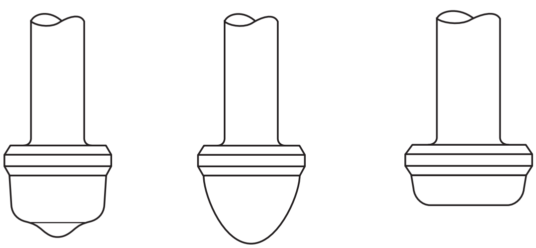


Figure 2.3 Characterizable linear plugs. (*Courtesy of Valtek International*)

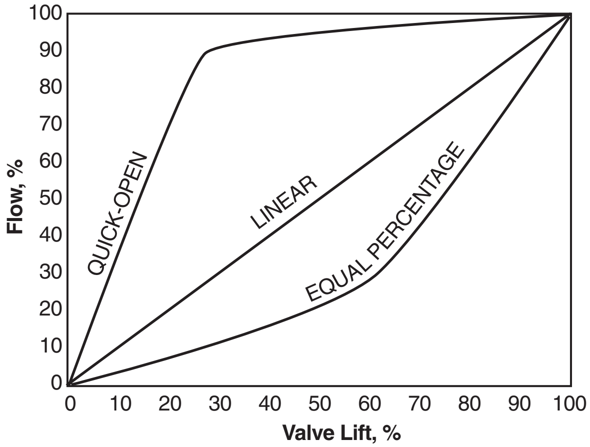


Figure 2.4 Typical inherent flow characteristics. (*Courtesy of Valtek International*)

age characteristic results in improved rangeability (Sec. 2.2.9) for a particular valve, as well as better repeatability and resolution in the first half of the stroke.

The mathematical formula for an equal-percentage characteristic is

$$Q = Q_0\, e^{nL}, \qquad \frac{dQ}{dL} = nQ$$

where Q = flow rate
L = valve travel
e = 2.718
Q_0 = minimum controllable flow
n = constant

Although the flow characteristic of the valve itself is equal percentage, the installed flow characteristic is closer to the linear flow characteristic. This is usually the case when the process system's pressure drop is larger than the pressure drop across the valve. Figure 2.5 shows two flow curves for an equal-percentage characteristic: the inherent flow characteristic and the installed characteristic that takes into account piping effects. The addition of the piping effects has a tendency to move the flow characteristic away from the ideal equal-percentage characteristic toward the inherent linear characteristic.

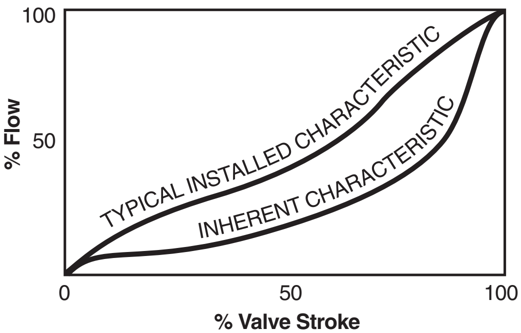


Figure 2.5 Typical inherent and installed equal-percentage flow characteristics. (*Courtesy of Valtek International*)

2.2.3 Linear Flow Characteristic

The inherent linear flow characteristic produces equal changes in flow per unit of valve stroke, regardless of the position of the valve. Linear flow characteristics are usually specified in those process systems where the majority of the pressure drop is taken through the valve. For the most part, linear flow characteristics provide better flow capacity throughout the entire stroke, as opposed to equal-percentage characteristics.

The mathematical formula for the linear characteristic is

$$Q = kL, \qquad \frac{dQ}{dL} = k$$

where Q = flow rate
L = valve travel
k = constant of proportionality

Figure 2.6 shows the inherent linear flow characteristic, as well as the installed characteristic (taking into account piping effects). As can be seen by this figure, the piping effects have a tendency to push the linear flow characteristic toward the quick-open characteristic.

2.2.4 Quick-Open Flow Characteristic

The quick-open characteristic is used almost exclusively for on–off applications, where maximum flow is produced immediately as the valve begins to open (Fig. 2.7). Because of the extreme nature of the

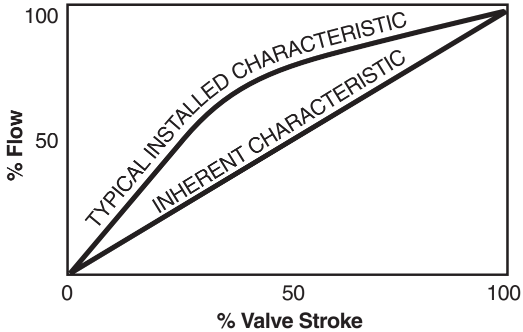


Figure 2.6 Typical inherent and installed linear flow characteristics. (*Courtesy of Valtek International*)

quick-open characteristic, the inherent and installed characteristics are similar.

2.2.5 Determining Installed Flow Characteristics

As discussed earlier, the inherent flow characteristic can change dramatically when the valve is installed in a process system. When the system's piping effects are taken into account, the equal-percentage characteristic moves toward linear, and the linear characteristic moves toward quick-open. Two examples of installed applications follow, one without piping effects and the other with piping effects.

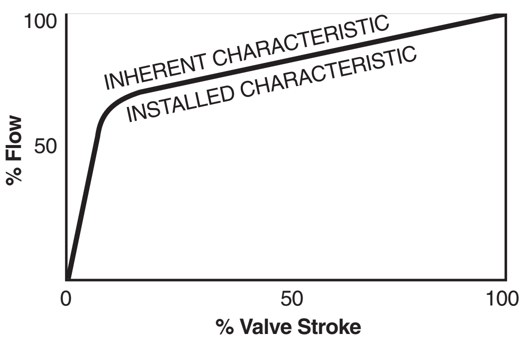


Figure 2.7 Typical inherent and installed quick-open flow characteristics. (*Courtesy of Valtek International*)

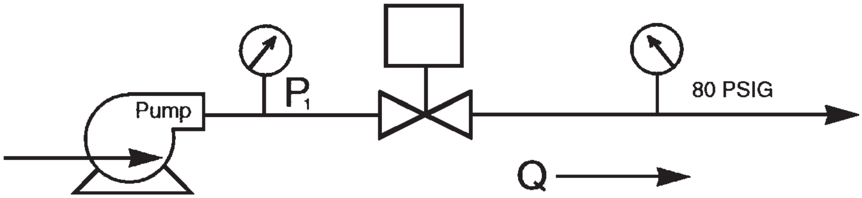


Figure 2.8 Typical flow schematic showing no piping losses. (*Courtesy of Valtek International*)

2.2.6 Flow Characteristic Example A (without Piping Effects)

Figure 2.8 shows a schematic of a process system that includes a centrifugal pump and a valve, which is used to maintain the pressure downstream to 80 psi or 5.5 bar. For illustration purposes, Fig. 2.9 provides the pump's relationship between the pump output (psi) and the flow (gal/min).

For this example, piping losses are assumed to be minimal. A total of 200 gal/min (757 liters/min) is required for the maximum flow rate. From Fig. 2.9, at 200 gal/min, the pump discharge pressure (P_1) is found to be 100 psi (6.9 bar) upstream of the valve, while 80 psi (5.5 bar) is required downstream (or, in other terms, a 20-psi or 1.4-bar

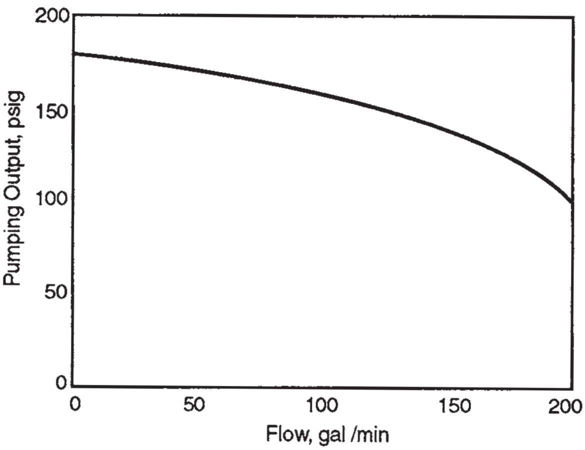


Figure 2.9 Flow chart of typical pump characteristics. (*Courtesy of Valtek International*)

pressure drop). Using the sizing formula for C_v (Sec. 2.1.2), we determine the C_v required for this application, which is

$$C_v = Q\sqrt{\frac{G_F}{\Delta P}} = 200\sqrt{\frac{1}{20}} = 45$$

Assuming that the C_v of 45 is the maximum C_v, several values of flow can now be estimated, along with the required valve C_v and the percent of maximum C_v the valve must have to control the process. These flow data are included in Table 2.1.

Using the definitions of both equal-percentage and linear characteristics, the installed characteristics can be plotted on a graph, using the data from Table 2.1, which is found in Fig. 2.10. This figure graphically illustrates the effect the installation has on the inherent flow characteristic. The linear characteristic moves away from the ideal linear line toward the quick-open characteristic. On the other hand, the equal-percentage characteristic moves toward the ideal linear line. In this example, either characteristic would provide good throttling control.

2.2.7 Flow Characteristic Example B (with Piping Effects)

For illustration purposes, Example A was simplified with a constant downstream pressure and a pressure drop only affected by the pump

Table 2.1 Flow Rate, C_v, and Pump Pressure (Without Piping Losses)†

Q Flow *gpm*	P_1 Pump Discharge Pressure *psig*	ΔP Across Valve *psi*	C_v Required	Percent of Valve Maximum C_v
50	170	90	5.2	11
100	150	70	12	27
150	125	45	22	49
200	100	20	45*	100

†*Data courtesy of Valtek International.*

*Maximum C_v.

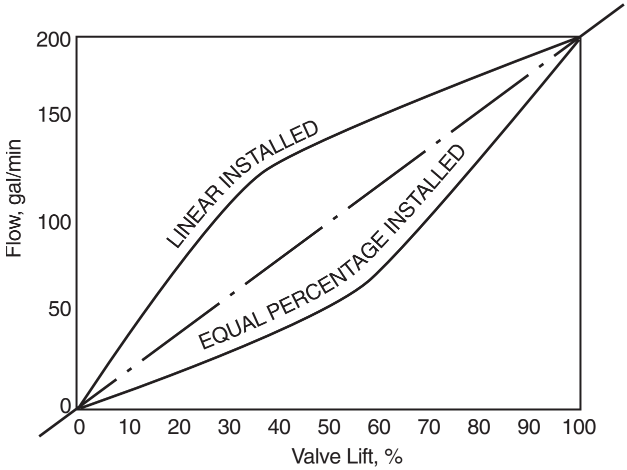


Figure 2.10 Installed linear and equal-percentage flow characteristics (without piping losses). (*Courtesy of Valtek International*)

characteristic. In Example B, the application is modified using a restriction downstream from the valve, as shown in Fig. 2.11. Note that the constant downstream pressure (80 psi or 5.5 bar) must be held constant after passing through the restriction.

Because of the restriction, the pressure drop must be distributed between the valve and the restriction (R). For this example, a 4-psi (0.3-bar) pressure drop across the valve is required at a flow rate of 200 gal/min (757 liters/min). Using the C_v equation, the maximum C_v for the valve is

$$C_v = Q\sqrt{\frac{G_f}{\Delta P}} = 200\sqrt{\frac{1}{4}} = 100$$

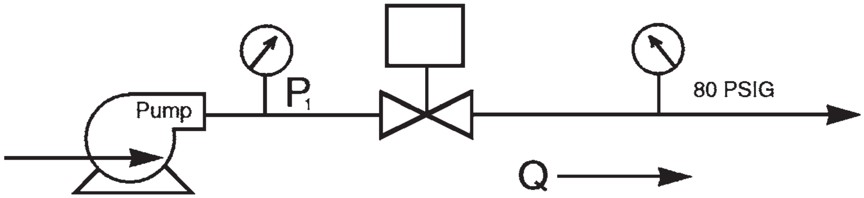


Figure 2.11 Typical flow schematic showing piping losses. (*Courtesy of Valtek International*)

Table 2.2 Flow Rate, C_v, and Pump Pressure (Without Piping Losses)†

Q Flow *gpm*	P_1 Pump Discharge Pressure *psig*	ΔP_R Across Restriction	ΔP Across Valve	C_v Required	Percent of Required Maximum Valve C_v
50	170	1	89	5	5
100	150	4	66	12	12
150	125	9	36	25	25
100	100	16	4	100*	100

†*Data courtesy of Valtek International.*
*Maximum C_v.

According to the square-root law ($Q = R\sqrt{\Delta P}$), the pressure drop across the valve's restriction will vary somewhat. Thus, using the pump characteristic, the available pressure drop across the valve can be estimated, which is shown in Table 2.2. Figure 2.12 shows the installed linear and installed equal-percentage characteristics from the data in

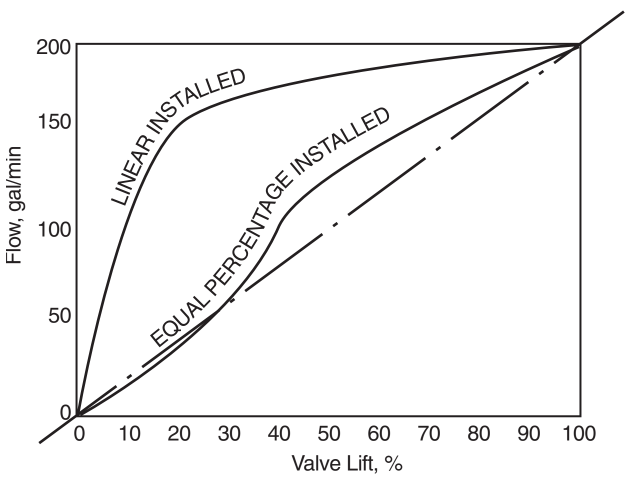


Figure 2.12 Installed linear and equal-percentage flow characteristics (with piping losses). (*Courtesy of Valtek International*)

Table 2.2. Note that the piping losses from the restriction have modified the installed equal-percentage characteristic to an inherent linear characteristic. In turn, the installed linear characteristic has become an inherent quick-open characteristic. Because of this effect of the piping losses, the use of a linear characteristic would create a highly sensitive system with a very small change in lift at the beginning of the stroke. On the other hand, using an equal-percentage characteristic would produce a more constant sensitivity throughout the entire stroke.

2.2.8 Choosing the Correct Flow Characteristic

When throttling valves are selected, a choice must be made between linear and equal-percentage characteristics. Two general rules apply that will simplify this choice. First, if most of the pressure drop is taken

Table 2.3 Recommended Flow Characteristics for Liquid Level Systems*

Constant Valve Pressure Drop	Recommended Inherent Flow 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 Δ P with increasing load: ΔP at maximum load < 200% of minimum load ΔP	Linear
Increasing Δ P with increasing load: ΔP at maximum load > 200% of minimum load ΔP	Quick Open

**Data courtesy of Valtek International.*

through the valve and the upstream pressure is constant, a linear characteristic will provide the best control. However, such systems are rare, especially considering the complexities of today's process systems. A linear characteristic is also recommended when a variable-head flowmeter is installed in the system. Second, if the piping and downstream equipment provide significant resistance to the system, the equal-percentage characteristic should be chosen. This is usually the case with most process systems today, where a majority of all throttling valves have equal-percentage characteristics. The equal-percentage characteristic is also used for applications of high pressure drops with low flows and low pressure drops with high flows. When the valve is oversized as a precaution because limited data are available, the equal-percentage characteristic will provide the greatest range of control. Tables 2.3, 2.4, 2.5, and 2.6 provide more specific recommendations,

Table 2.4 Recommended Flow Characteristics for Pressure Control Systems*

Application	Recommended Inherent Flow Characteristic
Liquid process	Equal Percentage
Gas process, small volume, less than 10 feet (3 meters) 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 feet 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

**Data courtesy of Valtek International.*

Table 2.5 Recommended Flow Characteristics for Flow Control Processes†

Flow Measurement Signal to Controller	Location of Valve in Relation to Measuring Element	Wide Range of Flow Set Point	Small Range of Flow with Large ΔP Change at Valve with Increasing Load
Proportion to flow	Series	Linear	Equal Percentage
	By-pass*	Linear	Equal Percentage
Proportion to Flow Squared	Series	Linear	Equal Percentage
	By-pass*	Equal Percentage	Equal Percentage

†*Data courtesy of Valtek International.*

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

Table 2.6 Recommended Flow Characteristics for Miscellaneous Systems*

Application	Recommended Inherited Flow Characteristic
Three-way valves and two-way valves used as three-way valves (If characterized positioners are used, they must be calibrated by the valve manufacturer)	Linear
Gas compressor recycle control valve	Linear
Constant pressure drop service	Linear
Temperature control where control valve $\Delta P > 50\%$ of System ΔP	Equal Percentage
pH control where control valve $\Delta P < 50\%$ of system ΔP	Equal Percentage
pH control where control valve $\Delta P > 50\%$ of system ΔP	Linear

**Data courtesy of Valtek International.*

depending on whether the system is for liquid level, pressure control, flow control, or another type of system, respectively.

For the most part, today's control instrumentation can make satisfactory signal adjustments to the throttling valve despite the flow characteristic. However, if manual control is ever required, having the correct flow characteristic allows such changes to be made easily.

2.2.9 Rangeability

Related to flow control and flow characteristics is the term *rangeability,* which is defined as the ratio of maximum to minimum flow that can be acted upon by a control valve after receiving a signal from a controller. Today's control valve applications often require a degree of *high rangeability,* which requires a valve to control flow from large to small flows. The rangeability of a control valve is affected by three factors.

The first factor is the valve's geometry (for example, the geometry of the plug and seat in globe valves), which has an inherent rangeability due to the design and configuration of the body and the regulating element. Sometimes the configuration can be modified, improving the rangeability as long as the valve's sensitivity is not affected. *Sensitivity* is defined as the specific change in flow area opening produced by a given change in the regulating element when compared to the previous position. In dealing with small flows when the regulating element is nearly closed, such as when a plug or a disk is close to the seat, oversensitivity can be a problem due to the small clearances involved.

The second factor, seat leakage, can also affect rangeability. Excessive seat leakage can cause instability as the valve lifts off the seat, especially with screwed-in seats that are not lapped, as opposed to floating clamped-in seats that are held in place by a retainer or cage.

Rangeability is also affected by the valve's actuation or actuator, which is the third factor. Some actuators are much more stiff at near-closure than others. For example, when a pneumatic spring diaphragm actuator is specified, a throttling valve is seldom accurate within the 5 percent of the valve closing. This is due primarily to the effects of the positioning spring, hysteresis, changing area of the diaphragm (as the actuator changes position), and the pressure drop itself. On the other hand, spring cylinder actuators use supply air pressure on both sides of a piston, which can provide control within less than 1 percent of valve lift, as well as a stiffness factor up to 10 times that of a comparable diaphragm actuator. Thus, a throttling valve equipped with a spring cylinder actuator would have a higher rangeability than the same valve with a diaphragm actuator.

Taking into account the effects of the valve geometry and the actuator, rangeability can be calculated in a simple manner. For example, if a valve is not accurate at less than 5 percent of stroke, then the rangeability is 20:1 (100 percent divided by 5 percent). As a common rule for common throttling valves, V-notched ball valves usually have the highest rangeability (up to 200:1), followed by eccentric plug valves (100:1), globe valves (50:1), and butterfly valves (20:1). Usually, the valves with the highest rangeability are those with the low sensitivity as the regulating element is nearly closed, but increases in sensitivity as the valve opens. Because the equal-percentage flow characteristic promotes increased sensitivity as the valve opens, it is usually chosen for most throttling applications. The term *clearance flow* is used to designate any flow that occurs between the lower end of the valve's rangeability and the actual closed position.

ISA Standard S75.11 ("Inherent Flow Characteristic and Rangeability of Control Valves") establishes guidelines for rangeability, sensitivity, and limits of deviation.

2.3 Shutoff Requirements

2.3.1 Shutoff Standards

Industry standards have been established for the control valve industry regarding the amount of permissible leakage of the process fluid through a valve's seat or seal. Usually this standard is applied to throttling valves, but may be applicable to other types of valves also. Specifically, ANSI Standard 70-2-1976 (reaffirmed in 1982) provides the outline for six classifications of shutoff.

2.3.2 Shutoff Classifications

Shutoff classifications are determined by a percentage of a test fluid (usually water or air) that passes through the valve, as part of the valve's rated capacity. This must take into account the predetermined pressure, temperature, and time limits. Shutoff classifications range from ANSI Class I, where the valve does not require tight shutoff, to ANSI Class VI, where shutoff must be complete or nearly bubble-tight. The following briefly describes each shutoff classification and maximum leakage rates for each.

The ANSI Class I shutoff is an open classification that does not require a test, while allowing for a specified agreement between the user and the valve manufacturer as to the required leakage. The ANSI Class II shutoff is 0.5 percent of the rated valve capacity and is associated with double-ported seats or pressure-balanced trims where metal piston rings and metal-to-metal seat surfaces are used. The ANSI Class III shutoff is 0.1 percent of rated valve capacity and is associated with the same types of valves listed in Class II, but is used for applications that require improved shutoff.

The ANSI Class IV shutoff is the industry standard for single-seated valves with metal-to-metal seating surfaces, which calls for a maximum permissible seat leakage of 0.01 percent of rated valve capacity. To achieve this higher classification with metal-to-metal seating surfaces, the load applied to the surfaces from the manual operator or actuator must reach certain levels. Table 2.7 provides a listing of typical required seat loads for Classes IV, V, and VI with metal and soft seating surfaces.

Both ANSI Classes V and VI were developed for throttling valves where shutoff is a primary focus. The ANSI Class V shutoff is defined as 0.0005 cm^2/min per inch of orifice diameter per pounds-per-square-inch (psi) differential. Class V is unique in that it is the only classification where the allowable seat leakage is allowed to vary according to the orifice diameter and the differential pressure (pressure drop). This classification is necessary for those applications where a throttling or control valve is used as a blocking valve that is required to stay closed for lengthy periods against a high pressure drop. It is applied to single-seat valves with either metal or soft seating surfaces or with pressure-balanced trim that requires extraordinary seat tightness.

The ANSI Class VI shutoff is commonly referred to as *bubble-tight shutoff* and is associated with metal-to-elastomer soft seating surfaces (such as using an elastomer insert in the seat ring or the plug head)—although with extremely high seating loads (as shown in Table 2.7), it is possible to achieve Class VI shutoff with a metal-to-metal seat. Class VI is independent of the pressure differential, but it does take into account milliliter per minute of leakage versus the seat orifice diameter. That means that valves with large seat diameters applied to a service with a low pressure drop can have a lesser leakage requirement than Class V. Figure 2.13 shows this relationship between Classes V and VI, taking into account the pressure differential for Class V and the lack of pressure differential for Class VI.

2.4 Body End Connections

2.4.1 Introduction to End Connections

A number of different end connections are available that allow the valve to be joined to the system's piping. In most cases, the valve's end

Table 2.7 Typical Seat Loads vs ANSI Classification for Shutoffs*

Seat Surface	ANSI Shutoff Classification	Valve Sizes	Required Seat Load (linear seating force)
Metal	Class IV	0.5 to 4-inch DN 15 to 100	50 pounds/inch 60 joules
Metal	Class IV	6-inch and above DN 150 and above	75 pounds/inch 91 joules
Metal	1% of Class IV	0.5 to 4-inch DN 15 to 100	100 pounds/inch 121 joules
Metal	1% of Class IV	6-inch and above DN 150 and above	150 pounds/inch 181 joules
Metal	Class V	0.5 to 4-inch DN 15 to 100	250 pounds/inch 303 joules
Metal	Class V	6 to 10-inch DN 150 to 250	400 pounds/inch 484 joules
Metal	Class VI	0.5 to 4-inch DN 15 to 100	250 pounds/inch 303 joules
Metal	Class VI	6 to 10-inch DN 150 to 250	400 pounds/inch 484 joules
Soft	Class V	0.5 to 4-inch DN 15 to 100	50 pounds/inch 60 joules
Soft	Class V	6-inch and above DN 150 and above	100 pounds/inch 121 joules
Soft	Class VI	0.5 to 4-inch DN 15 to 100	50 pounds/inch 60 joules
Soft	Class VI	6-inch and above DN 150 and above	100 pounds/inch 121 joules

**Data courtesy of Valtek International.*

connection is designed or specified to match the piping connection. In an ideal situation, end connections and materials between the valve and the piping would be identical; however, this is not always the case.

The general rule is that smaller-sized valves—smaller than 2-in (DN 50) valves—can use threaded connections (Fig. 2.14), while larger

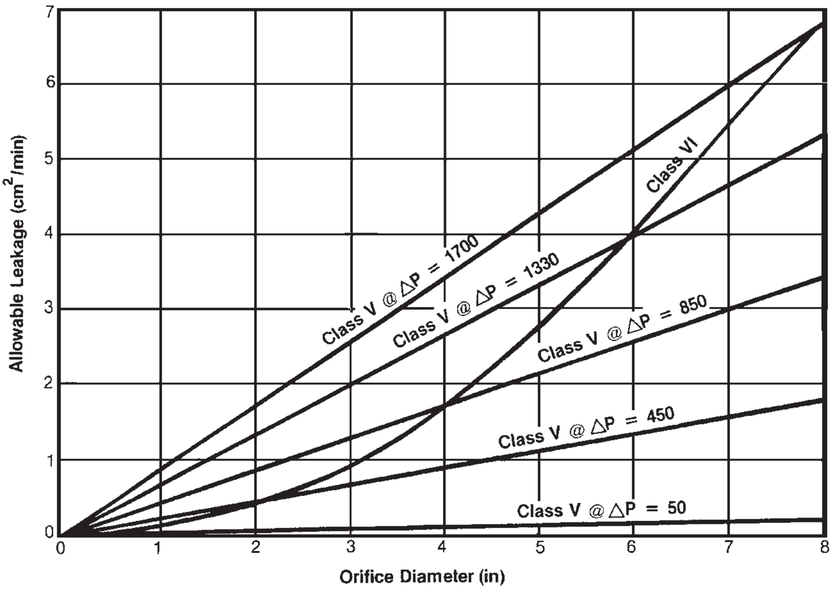


Figure 2.13 ANSI Class V and VI allowable leakage. (*Courtesy of Valtek International*)

sizes—2-in (DN 50) and larger—use flanged connections (Fig. 2.15). The refining industry uses such a standard, since it is very conscious of fugitive-emission mandates against leakage. Some process systems where fugitive emissions or process leakage is not a problem (such as water systems) will use threaded connections in sizes up to 4 in (DN 100).

Most process system applications require both ends of the valve to have identical connections. On some applications, such as vent and

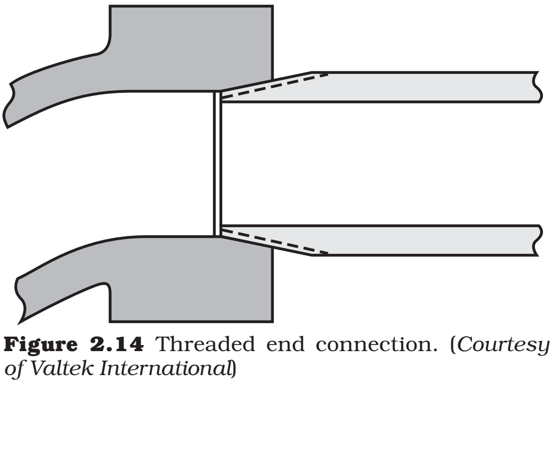


Figure 2.14 Threaded end connection. (*Courtesy of Valtek International*)

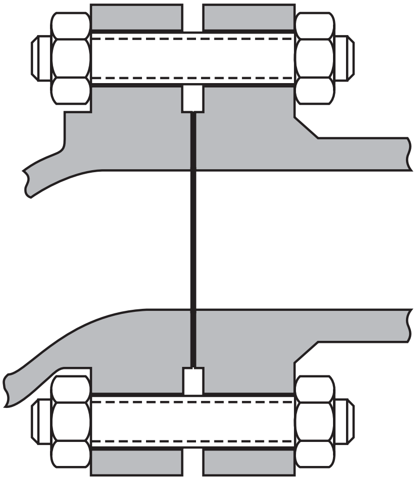


Figure 2.15 Integral flange end connection. (*Courtesy of Valtek International*)

drain valves, one end may require one type of connection on the upstream port and a different connection on the downstream port.

2.4.2 Threaded End Connections

As noted above, threaded connections are used in smaller sizes—1.5 in (DN 40) and smaller. The standard end connection for valves smaller than 1 in (DN 25) is a threaded connection. If leakage is not a concern, threaded connections can be used in sizes up to 4 in (DN 100).

The valve's end connection is designed with a female National Pipe Thread (NPT), which mates with the piping that uses a male NPT thread. Because of the leakage and pressure limitations of threaded ends, they are only rated up through ANSI Class 600. Also, threaded ends should not be used with corrosive processes, since the threads can either fail or become inseparable.

A National Pipe Thread is the most commonplace thread joint. One exception is for fire management systems, which require the use of the National Hose Thread (NHT), which matches connections used by fire departments. Another thread occasionally seen in a process system is the ordinary ¾-in Garden Hose Thread (GHT). Threads can be either cut or molded in place, especially when precision moldings are used. The molded threads do not have sharp edges (which are produced by machining), but are more rounded at the peak of the thread.

When used in smaller sizes, threaded connections are easy to install since the valve is smaller and lightweight. This is important because the pipe and valve must be rotated to make the connection. In some cases, the pipe fitting will require piping tape or compound to ensure a tight seal.

Because the threaded design requires little machining and is commonplace among most valve manufacturers, it is the least expensive to specify.

2.4.3 Flanged End Connections

Flanges are commonly required on valves larger than 2 in (DN 50). Flanges are easier to install than threaded connections, because the valve's face is matched up with piping and bolted together without any rotation of the pipe or valve. Flanges can be applied in most temperatures, from absolute zero to 1500°F (815°C). As the temperature increases, some limitations are placed on high pressures.

Force generated by the flange bolting, coupled with the gasket between the flanges, is used to seal the connection. Flanges are built to the ANSI Standard B16.5 (or API 6A or similar standards), which addresses design criteria for the flat face, the height and diameter of the raised face, standard hole patterns, and the necessary dimensions for even rare joints, such as tongue and groove, and male and female designs. Flanges are rated according to the type of service, material requirement, maximum service temperature, and pressure. Although the main advantage of flanges is that the valve can be removed easily from the line, flanges are subject to thermal distortion and shock. If temperature cycles vary significantly, then a welded connection should be considered as an alternative.

Two types of flange designs exist. *Integral flanges*, as the name implies, are an integral part of the body. With integral flanges, the flange hole pattern is either machined or cast into the body casting. Integral flanges are commonplace since they are standard with many valve manufacturers and have been used from the earliest designs. On the other hand, *separable flanges* have been a relatively new addition to end-connection design. Separable flanges are individual flanges that slide over the hub ends of the body and are held in place by half-rings.

Integral flanges can be provided with a *flat face* (Fig. 2.16), which allows full contact between the two matching flanges and the flange gasket. Flat-face flanges are commonplace with low-pressure applications as well as brass and cast-iron valves. Because the flanges are in complete contact with each other, this design minimizes flange stresses

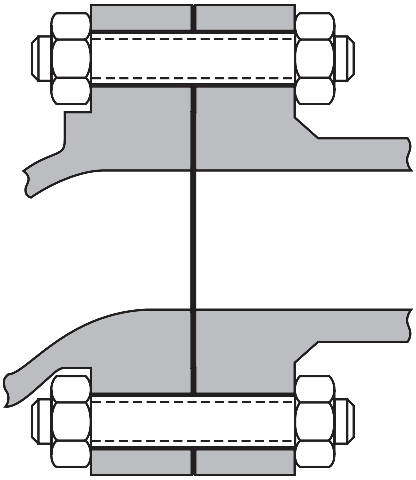


Figure 2.16 Flat-face end connection. (*Courtesy of Valtek International*)

as well as possible bending of the flange as the bolting is tightened. However, the flange faces must be completely flat to create an equal seal through the entire flange. When flat-faced flanges are specified, larger-diameter gaskets (same as the flange) are used to provide the seal.

Another common flange face is *raised face* (refer to Fig. 2.15), which is a circular area that physically separates the two flanges. The raised face is only a slight step. The inside diameter of the raised face is identical to the inside diameter of the pipe–valve port, while the outside diameter is smaller than the bolt circle. ANSI standards call for this raised face to be 0.06 in (1.5 mm) below ANSI Class 600 (PN 100) and 0.25 in (6 mm) in sizes above ANSI Class 600 (PN 100). The raised face separates the flanges themselves, preventing any incidental flange-to-flange contact that may result in decreased gasket sealing pressure, although some flange stress may be created when the bolting is tightened. This raised face may be serrated with concentric circular grooves when using simple sheet gaskets or may have a smoother surface if spiral-wound gaskets are used. The raised face is finished with a series of concentric circular grooves, which are designed to keep the gasket in place (preventing blow-out) and to provide a better seal. This type of flange is specified on ANSI Class 250 iron valves and all steel valves. It is recommended in pressures through 6000 psi (400 bar) and in temperatures to 1500°F (815°C).

The *ring-type joint* (also known as RTJ) is a modification of the raised-face design (Fig. 2.17). A U-shaped groove is cut into the face,

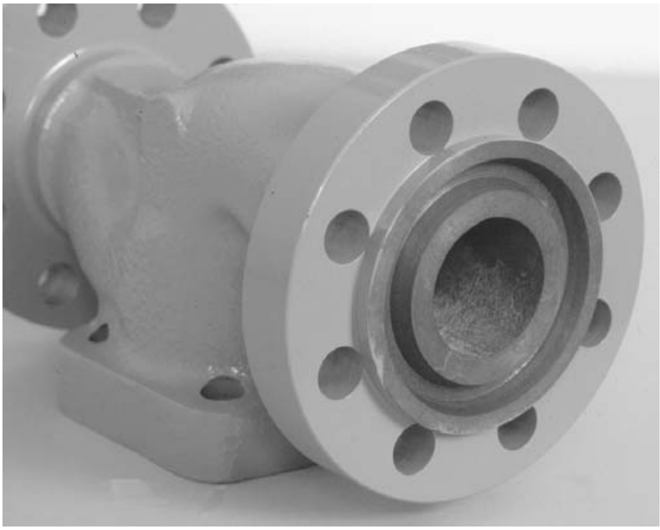


Figure 2.17 Ring-type joint end connection. (*Courtesy of Valtek International*)

which is concentric with the valve port. A soft metal gasket (commonly Monel or iron, but any soft metal can be specified) is then inserted in this groove, which is wedged in place as the flanges are tightened. RTJ flanges are specified for high-pressure applications—up to 15,000 psi (1000 bar)—although not with high-temperature applications.

As mentioned earlier, separable flanges (Fig. 2.18) are now accepted as an inexpensive, versatile alternative to integral flanges. Because the flange is not wetted by the process, it can be produced from simple carbon steel and be painted for atmospheric protection, which lowers the cost of a valve that requires a stainless-steel or alloy body. The separable flange is designed to slide over the body hub. To fasten the flange in place, two half-rings are inserted in a groove in the body, which act as mechanical stops. When the flange bolting is tightened, the flanges lock against the rings, holding the valve body securely in place. Although carbon steel is the most common (and inexpensive) material for separable flanges, stainless steel flanges are necessary for high-temperature–high-pressure applications.

One important advantage of separable flanges over integral flanges is their range of motion when dealing with misaligned pipe flanges. If the flange of an upstream pipe is fixed in place and is not exactly aligned with the flange of the downstream pipe, the misalignment will prevent the installation of a valve with integral flanges—unless the flange and

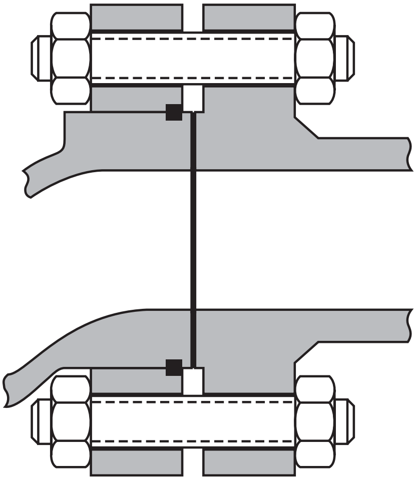


Figure 2.18 Separable flange end connection. (*Courtesy of Valtek International*)

valve hole patterns are modified to align the holes. On the other hand, with separable flanges, the flange on either end of the valve can be rotated slightly to compensate for the misalignment. This ability to modify the alignment of the flanges also allows the valve to be rotated and fixed in a different position (especially if a space conflict exists).

Separable flanges can be designed to be interchangeable among low-pressure classes. They are rated to ANSI Classes 150–600 (PN 16–PN 100) in sizes of 4 in (DN 100) and smaller. With ANSI Classes 150–300 (PN 16–PN 40), flanges are available in 6- and 8-in sizes (DN 150 and DN 200). Separate flanges can also be used with ANSI Class 150 (PN 16) in sizes larger than 10 in (DN 250).

Although the separable flange design is less expensive and more versatile, one drawback is that if the flange bolting is not properly tightened, the valve could rotate accidentally because of gravitation forces or excessive line vibration—especially if the valve has a heavy actuator or other top-works. Following installation, this problem may be remedied by using tack welds to keep the flange or body from rotating.

2.4.4 Welded End Connections

When zero leakage is required—for environmental, safety, sanitary, or efficiency reasons—the piping can be welded to the valve, providing one-piece construction. Many users insist that high-pressure applications—ANSI Class 900 (PN 160) and higher—require a permanent end

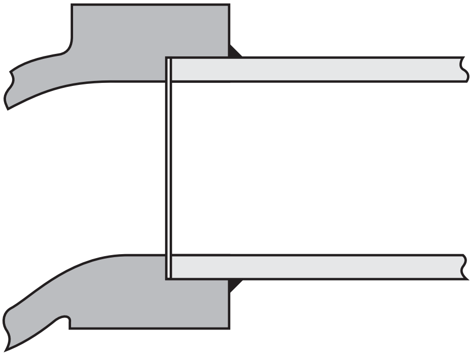


Figure 2.19 Socketweld end connection. (*Courtesy of Valtek International*)

connection, especially if they involve high temperatures. Nearly all steam and water services in the power industry call for welded connections. The two most common welded connections are socketweld and buttweld connections.

The *socketweld connection* (Fig. 2.19) is specified in high-pressure–high-temperature fluids in sizes 2 in (DN 50) and smaller. The socketweld design for a valve involves boring into the valve's body end to a predetermined depth (according to ANSI Standard B16.11). The piping is then mated or inserted into the bore, and a weld is then applied between the pipe outside diameter and the face of the body. The welding standard for socketweld connections is the piping welding specifications according to the local or ANSI codes (B31.1 or B31.3).

For larger valve sizes 3 in (DN 80) and larger, a *buttweld connection* (Fig. 2.20) is specified for high-pressure–high-temperature applica-

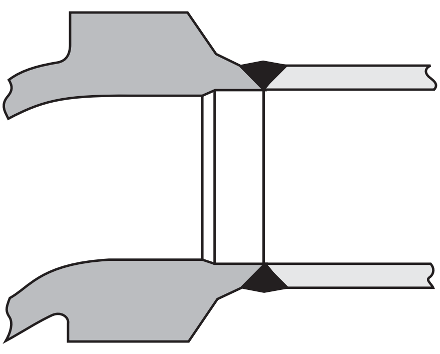


Figure 2.20 Buttweld end connection. (*Courtesy of Valtek International*)

tions. Buttweld ends involve a lip that butts up against a similar lip on the pipe. Following the lip, both the pipe and valve use a single- or double-angle bevel to create a V-shaped butt joint that is filled with a full penetration weld. Some smaller industrial valves may incorporate a J-bevel or U-bevel in the design. These joints are harder to manufacture, but easier to inspect with radiology. Most buttweld ends are specified according to ANSI Standard B16.25, which calls for a 37.5° angle for wall thicknesses up to $\frac{7}{8}$ in (22 mm). If the wall thickness exceeds $\frac{7}{8}$ in, a compound buttweld of 37.5° and 10° is specified.

The user may also designate a special buttweld design according to individual specifications. For example, power applications sometimes require the use of a backing ring, which must be incorporated into the buttweld specifications. Backing rings are inserted to ensure proper alignment of the pipe and valve.

When considering socketweld and buttweld connections, material compatibility between the valve and piping must be a consideration to ensure proper welding and mating of the valve to the piping. Since carbon alloys or high-chrome steel have a tendency to air-harden, they should be avoided (or be heat-treated.)

2.4.5 Other End Connections

Nonmetallic valves, of which plastic is the most common, are equipped with other types of end connections. Small plastic valves can be manufactured with *union end connections,* which are used to join the plastic valve to plastic piping. Each end of the valve retains an external nut that can be threaded onto the pipe to make a solid connection. Small plastic or metal valves used in vacuum service can be equipped with an O-ring joint.

Valves made from polyvinylchloride (PVC) and chlorinated polyvinylchloride (CPVC) use a male–female socket arrangement, similar to the socketweld design, except that a solvent cement is used to fuse the two pieces together. Another method used to bond plastic piping and valves is heat fusion, in which an outside heating source melts the plastic and allows the two parts to fuse together.

Iron valves can be connected to piping using a clamp coupling that fits into special grooves cut into the ends of the valve and pipe. Stainless-steel sanitary valves may use special clamp joints, which allow the system to be disassembled regularly for cleaning (Fig. 2.21).

Some rotary valves have *flangeless* connections, where the valve body—which by its rotary design has a short face-to-face—is placed between two pipe flanges, which are then bolted together. This config-

Figure 2.21 Sanitary end connection. (*Courtesy of Jordan Valve*)

uration allows the valve to be bolted securely between the flanges and uses a simple flat gasket. The outside diameter of the body hub matches the outside diameter of the raised face on the pipe end. Some consideration should be given to thermal expansion, as the longer bolting can lengthen or shorten accordingly, causing leakage or crushing the gasket, respectively. Thermal effects can be modified by using a flexible gasket that can control the compression. However, this design is only recommended when there are no fire-safe considerations. During a fire, thermal expansion can cause the bolting to expand, causing process leakage that may feed the fire.

2.5 Pressure Classes

2.5.1 Introduction to Pressure Classes

A valve is designed to handle a certain range of internal pressure up to a certain point, which is called the valve's *pressure rating*. The higher

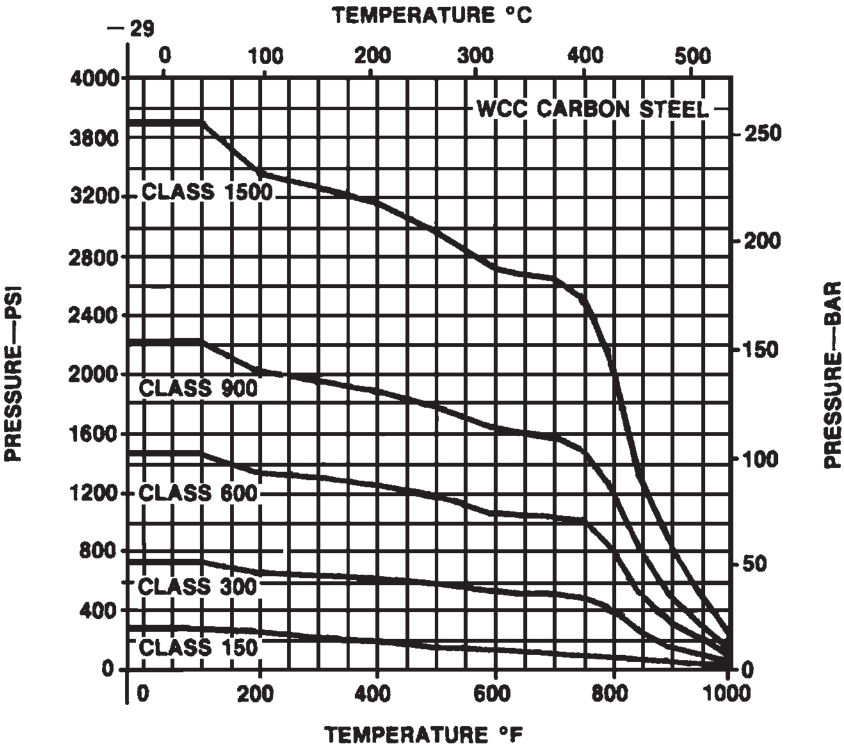


Figure 2.22 Pressure–temperature ratings for carbon steel. (*Courtesy of Fisher Controls International, Inc.*)

the pressure rating for a valve, the thicker the wall thickness must be so that the valve body subassembly will not rupture. The pressure rating is affected by the temperature of the service also: the higher the process temperature, the less pressure can be handled by the body subassembly, as shown in Fig. 2.22. ANSI Standard B16.34 is used to determine the pressure–temperature relationship, as well as applicable wall thickness and end connections.

An understanding of common pressure class ratings and pressure ratings is important, especially since a valve's pressure class can be designated as a standard class, a special class, or an intermediate class.

2.5.2 Standard Classification

The most common pressure class standard is ANSI B16.34, which specifies six *standard classes:* Class 150, 300, 600, 900, 1500, or 2500. (See Table

1.2 for nominal pressure designations.) These classes apply to valves with NPT threaded, flanged, socketweld, and buttweld end connections.

2.5.3 Special Classification

Special class ratings are available when nondestructive examination requirements are met for valves with buttweld end connections. ANSI Standard B16.34 allows buttweld valves to be upgraded to ANSI Special Classes 15, 300, 600, 900, 1500, 2500, and 4500.

2.5.4 Intermediate Classification

This ANSI standard also permits the use of *intermediate ratings* for valves with buttweld end connections, such as an ANSI Intermediate Class 3300. Using this class requires additional engineering time, but does allow a special service valve to be reduced in size, weight, and cost. For example, a carbon-steel valve is required for a 300°F (150°C) service at 6500 psi (450 bar). Normally, if using a conventional standard or special pressure class, the valve would require an ANSI Special Class 4500 pressure rating, which would increase the size, weight, and cost of the valve. However, if the ANSI Intermediate Class 3300 is chosen, a smaller valve could then be used. One point should be remembered, however. Unless the valve manufacturer has engineered this intermediate class, special design and casting patterns will be required, which may increase the cost of the valve. This added cost of new engineering should be weighed against the cost of the larger, existing valve design.

The ANSI intermediate classification can also be used to designate pressure classes larger than ANSI Special Class 4500, although one should not confuse a 6600 psi (450 bar) pressure rating for ANSI Intermediate Class 6600, which has a maximum pressure of 13,200 psi or 910 bar.

2.6 Face-to-Face Criteria

2.6.1 Introduction to Face-to-Face

The dimension between one pipe mating surface of the valve to the surface on the opposite end is called the *face-to-face* dimension. This physical dimension is always determined by the surface-to-surface measurement regardless of the type of end connection (threaded, flanged, or welded).

Most valves' face-to-face is determined by the industry standards, although some custom designs, such as Y-body valves, are determined by the manufacturer or restricted by the limitations of the design. In some cases, the user's process system layout may determine a special face-to-face. For example, some valves designed for the power industry come with buttweld end connections that are designed with custom face-to-faces.

A question often arises about the ring-joint end connection, where the sealing surface is the end of the ring and not the surface of the valve end. In this case, the face-to-face dimension is still considered to be the valve's face surfaces.

2.6.2 Common Face-to-Face Standards

Several standards for face-to-face valves are commonly used throughout the process industry, as outlined in Table 2.8. These standards have been set by the following organizations: American National Standards Organization (ANSI), Instrument Society of America (ISA), American Society of Mechanical Engineers (ASME), British Standards Institute (BSI), and Manufacturers Standardization Society of Valves and Fittings Industry (MSS).

2.7 Body Material Selection

2.7.1 Introduction to Body Materials

Normal practice calls for the control-valve user to specify the body material, especially with special service or severe service valves. Many general service valves are specified with commonly found materials, such as carbon or stainless steels. In most cases, the required body material is the same as the pipe material—which most likely is carbon steel, stainless steel, or chrome–molybdenum steel (commonly called *chrome-moly*).

Carbon steel is probably the most common material specified for valves. Overall, it is the ideal material for noncorrosive fluids. Carbon steel is also widely used for steam and condensate services. It does exceptionally well in high temperatures: up to 800°F (425°C) in continuous service, or even up to 1000°F (535°C) in noncontinuous service. Carbon steel is readily available in most common general service valves and generally inexpensive, especially when compared to other commonly used metals.

Stainless steel is very corrosion resistant, extremely strong, and is commonly specified for high-temperature applications—temperatures

Table 2.8 Common Face-to-Face Standards

Standard	Valve Type	Pressure Rating
ANSI/ISA S75.03	Globe valves	150 - 600 (valve is interchangeable between Class 150, 300 and 600)
ANSI/ISA S75.04	Flanged globe valves	125, 150, 250, 300, 600
ANSI/ISA S75.04	Flangeless globe valves	150, 300, 600
ANSI/ISA S75.08	Flanged clamp or pinch valves	All classes
ANSI/ISA S75.12	Socketweld and threaded end globe valves	150, 300, 600, 900, 1500, 2500
ANSI/ISA S75.14	Buttweld globe valves	4500
ANSI/ISA S75.15	Buttweld globe valves	150, 300, 600, 900, 1500, 2500
ANSI B16.10	Iron (ferrous), gate, plug, globe valves	All classes
BS 2080	Steel valves used in the petroleum, petrochemical and associated industries	All classes
MSS SP-67	Butterfly valves	All classes
MSS SP-88	Diaphragm valves	All classes
MSS SP-42	Stainless steel valves (gate, globe, angle and check)	All classes

of 1000°F (535°C) and higher. Its cost is somewhat higher than carbon steel, although less than other steel alloys.

Chrome–molybdenum steel is a good material that falls between the characteristics of carbon steel and stainless steel. It can handle higher pressures and temperatures than carbon steel, making it ideal for high-pressure steam or flashing condensate applications. Its strength surpasses carbon steel and is nearly equal to that of stainless steel. However, chrome–molybdenum steel is not as corrosion resistant as stainless steel.

Special alloys are specified for special service or severe service valves. For example, Hastelloy B and C or titanium may be selected to avoid fluid incompatibility, such as a highly acidic fluid. In another case, a Monel or bronze body may be selected for a pure oxygen service, where having a nonsparking material is critical for safety reasons.

Table 2.9 lists a number of common valve materials and their temperature limits. Valve bodies are manufactured from castings, forgings, or barstock, or can be fabricated from piping tees and flanges. Castings are the least expensive choice because of the process and the higher volumes run by the manufacturer. Forgings are required for special materials and/or higher-pressure ratings, such as ANSI Classes 1500 (PN 250), 2500 (PN 400), or 4500 (PN 700). Barstock bodies are required for critical deliveries where a cast or forged body is not readily available, or when structural integrity is essential. Fabricated bodies are required for large angle valves.

As a general rule, bonnets or bonnet caps (which are used to seal the upper portion of the body subassembly) are made from the same material as the body, although most are manufactured from barstock instead of castings. One exception to this rule is a low-pressure chrome–molybdenum valve, which often requires a stainless-steel bonnet as the standard for sizes 6 in (DN 150) and smaller.

2.7.2 Material Selection Standards

Since several parts of a valve are exposed to pressure, process fluid, corrosion, and other effects of the service, those parts are required by regulation to be manufactured from approved metals. These parts are usually specified as the body, bonnet, bonnet bolting, plug, ball, disk, wedge, and/or drainage plug. Although a plug stem or rotary shaft extends from the pressure vessel, they are not considered to be pressure-retaining parts by the leading quality- and safety-related organizations.

The American National Standards Institute publishes specific pressure and temperature limits for specified materials (Standard B16.34). This standard should be reviewed before any material is selected to ensure that it will fall within the correct pressure–temperature limits. Materials used in the construction of pressure-retaining parts are designated by codes formalized by the American Society for Testing and Materials (ASTM). ASTM provides specifications for materials as they are subjected to that organization's standard testing procedures, as well as acceptance criteria. ASTM codes are critical in that they ensure that a material is duplicated time and time again according to correct specification, regardless of the manufacturer. If the material is produced according to specification, its properties will be able to with-

Table 2.9 Temperature Limits for Body Materials†

Material	Upper Limit (°F)	Upper Limit (°C)	Lower Limit (°F)	Lower Limit (°C)
Cast Iron	410	210	-20	-5
Ductile Iron	650	345	-20	-5
*Carbon Steel (Grade WCB)	1000	535	-20	-5
Carbon Steel (Grade LCB)	650	345	-50	-10
Carbon Moly	850	455	-20	-5
1-1/4 Cr - 1/2 Mo (Grade WC6)	1000	535	-20	-5
2-1/4 Cr - 1/2 Mo (Grade WC9)	1050	565	-20	-5
5 Cr - 1/2 Mo (Grade C5)	1100	595	-20	-5
9 Cr - 1 Mo (Grade C12)	1100	595	-20	-5
Type 304 (Grade CF 8)	1500	815	-425	-220
Type 347 (Grade CF8C)	1500	815	-425	-220
Type 316 (Grade CF8M)	1500	815	-425	-220
3-1/2 Ni (Grade LC3)	650	345	-150	-65
Aluminum	400	205	-325	-160
Bronze	550	285	-325	-160
Inconel 600	1200	650	-325	-160
Monel 400	900	480	-325	-160
Hastelloy B	700	370	-325	-160
Hastelloy C	1000	535	-325	-160
Titanium	600	315	N/A	N/A
Nickel	500	260	-325	-160
Alloy 20	300	150	-50	-10

†*Courtesy of Valtek International.*

*The carbon phase of carbon steel may be converted to graphite upon long exposure to temperatures above 775°F (415°C). Check applicable codes for maximum temperature ratings of various materials. Other specific data available in ANSI B16.34.

stand or handle the application it was designed for, such as corrosive fluids, severe temperatures, or high pressures.

ASTM codes are not intended to cover all known materials but do cover all common materials used in known applications. Since a number of new materials are being introduced annually, ASTM has procedures that allow new materials to be submitted for acceptance, and sometimes even allowed to be used before being formally accepted by ASTM, as long as the procedures are followed exactly. Table 2.10 provides applicable body and bonnet material standards (ANSI Standards B16.34 and B16.24) for castings, forgings, and barstock.

Another organization associated with the manufacture and performance of pressure-retaining parts is the American Society of Mechanical Engineers (ASME), which oversees and publishes *The Boiler and Pressure Vessel Code.* Section II of that code covers material selection for equipment that is under pressure, which includes valves. A comparison of the materials outlined in Sec. II with ASTM-specified materials shows that nearly all are covered by both standards. The materials listed in Sec. II carry the same numerical designation as ASME, although ASTM uses a specification prefix "S" before the num-

Table 2.10 Common ASTM Materials for Bodies and Bonnets*

Body Type	Material	Body Standard	Bonnet Standard
Castings	Stainless Steel	A351-CF8M	A479-316
	Carbon Steel	A216-WCB	A675-70
	Chrome-moly	A217-WC6	A479-316
		A217-WC9	A479-316
		A217-C5	A479-316
Forgings	Stainless Steel	A743-CF8M	A479-316
	Carbon Steel	A105	A675-70
	Chrome-moly	A182-F11	A479-316
		A182-F22	A479-316
		A182-F5a	A479-316
Barstock	Stainless Steel	A182-F316	A479-316
		A479-316	A479-316
	Carbon Steel	A675-70	A675-70
	Chrome-moly	See Forgings	See Forgings

**Data courtesy of Valtek International.*

ber. ASME also regulates procedures for welding, heat-treating, and preheating. Another organization, the American Welding Society (AWS), oversees procedures and regulations for welding rod and wire.

2.8 Gasket Selection

2.8.1 Introduction to Gaskets

A gasket is a malleable material, which can be either soft or hard, that is inserted between two parts to prevent leakage between that joint. It is designed to be placed in a predetermined space in a joint between the two parts. This space may be a counterbore, groove, or retainer plate (Fig. 2.23). Pressure is applied by bolting or using a clamp to compress the gasket firmly in place. As a general rule, to avoid damage to parts and to seal properly, gaskets must be softer in composition than the materials of the parts themselves.

Gaskets are made from all different types of materials, depending on the temperature, pressure, or fluid characteristics of the process. Some are designed to be resilient or self-energizing to allow for variations in temperature or pressure, which may require the gasket to expand or condense accordingly. Other gaskets, when used in more constant or severe service conditions, are made with harder materials (such as soft metals) that provide a strong seal, but are not self-energizing and once compressed may not be used again.

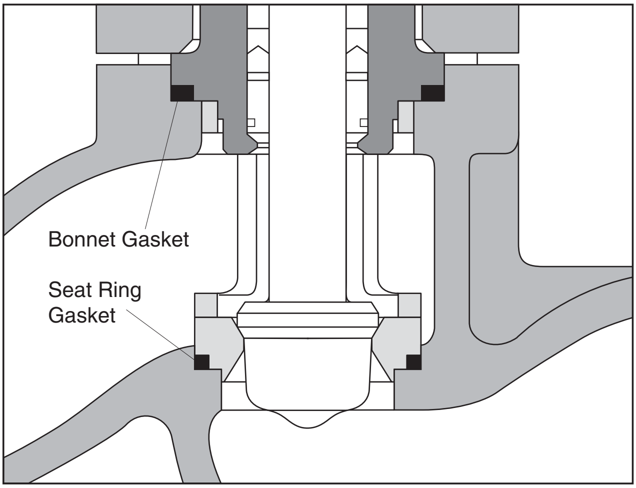


Figure 2.23 Gasket placement in typical globe valve design. (*Courtesy of Valtek International*)

Gaskets are used in valves for three major purposes. First, as mentioned earlier, gaskets prevent leakage around the closure mechanism. For example, a gasket is used to seal the joint between the body and seat in a linear valve to prevent leakage from the upstream side of the valve to the downstream side. Without the gasket, the fluid would leak past the seated plug. Second, gaskets are used to prevent leakage of fluid to atmosphere. For example, split-body and top-entry valves are designed with gaskets at the disassembly joints. Third, gaskets are used to allow the function of internal mechanisms that depend on separate fluid chambers, such as pressure-balanced trim.

Obviously, the ability for gaskets to function correctly is dependent on the correct seating load, which can vary widely according to the style of gasket, free height, wall thickness, material, and groove (or step) depth. Usually the valve manufacturer provides a torque specification for the associated bolting to ensure the proper seating load for the gasket. A common problem with such torque requirements is that if a torque wrench is not readily available, the risk may exist for a technician to overtighten the bolting, thus crushing the structure gasket, which can actually create a leak path. On the other hand, some valve designs prevent gasket crushing by using a metal-to-metal fit between the two mating parts, which ensures the proper gasket seating compression without a torque wrench. When the two parts are tightened so that they achieve a metal-to-metal connection, the height of the step and the gasket compression are assured. When the metal-to-metal connection is achieved, it can easily be felt through the wrench.

Gaskets come in a number of different styles, the most common being flat gaskets, spiral-wound gaskets, metal O-ring gaskets, metal C-ring gaskets, metal spring-energized gaskets, and metal U-ring gaskets. In some applications, the gaskets are coated with a rubber or plastic material to improve the self-energizing ability of the gasket or the corrosion resistance of the gasket. Some metal O-rings can be plated to improve the corrosion resistance.

To seal adequately, the gasket surfaces of the step or groove must be sufficiently smooth and flat. Ideally, surfaces should be finished to between 125 and 500 μin RMS (root mean squared) (between 3.2 and 12.5 μm).

Common specifications for these gasket styles are found in Table 2.11.

2.8.2 Flat Gaskets

Of the different types of gaskets, the most simple and inexpensive are *flat gaskets,* which as the name describes are gaskets that are machined with a simple outside diameter, inside diameter, and a certain height

Table 2.11 Typical Gasket Specifications†

Type	Gasket Material	Maximum Temperature (°F/°C)	Minimum Temperature (°F/°C)	Maximum Pressure (psi/bars)
Flat	Virgin PTFE	350/175	-200/-130	6000 - 1000 psi 415 - 70 bar
Flat	Reinforced PTFE	450/230	-200/-130	6000 - 500 psi 415 - 35 bar
Flat	CTFE	200/95	-423/-250	6000 - 500 psi 415 - 35 bar
Flat	FEP	400/205	-423/-250	6000 - 500 psi 415 - 35 bar
Spiral-wound	AFG*	1500/815	-20/-30	6250 psi 430 bar
Spiral-wound	304 SS/Asbestos	750/400	-20/-30	6250 psi 430 bar
Spiral-wound	316 SS/Asbestos	1000/540	-20/-30	6250 psi 430 bar
Spiral-wound	316 SS/PTFE	350/176	-200/-130	6000 - 500 psi 415 - 35 bar
Spiral-wound	316 SS/Graphite	1500/815	-423/-250	6250 psi 430 bar
Hollow O-ring	Inconel X-750	1500/815	-20/-30	15,000 psi 1035 bar

†*Data courtesy of Valtek International*
*Asbestos-free gasket.

(Fig. 2.24). For the most part, these gaskets adapt easily to any irregularities in metal surfaces of the joint due to its elasticity or plastic deformation.

Flat gaskets are best used for general service applications without severe temperature or pressure considerations. Flat gaskets can be made from industrial plastics, such as polytetrafluoroethylene (PTFE) or chlorotrifluoroethylene (CTFE), or soft metals, such as aluminum, copper, silver, soft iron, lead, or brass. Some metal flat gaskets are applied to high-temperature service, such as nickel [1400°F (760°C)], Monel [1500°F (815°C)], or Inconel [2000°F (1095°C)].

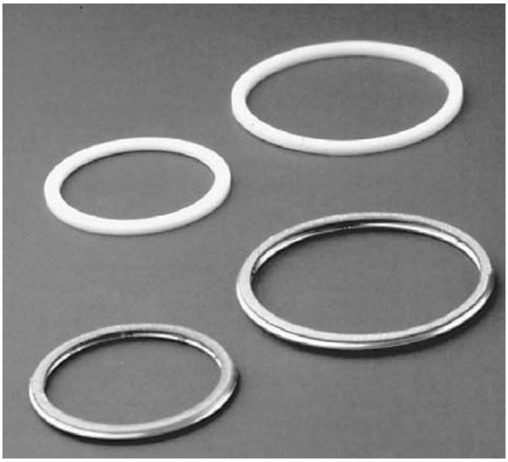


Figure 2.24 Flat (above) and spiral-wound (below) gaskets. (*Courtesy of Valtek International*)

2.8.3 Spiral-Wound Gaskets

Spiral-wound gaskets are all-purpose, medium-priced gaskets that consist of alternate layers of metallic and nonmetallic materials wound together (Fig. 2.24). The metal strip winding is normally V-shaped and is set on edge with the filler material sandwiched between windings. Spiral-wound gaskets combine the elastic properties of flat gaskets with the inclusion of soft metal windings, which adds strength to prevent possible gasket blow-out high-pressure–high-temperature applications. The strength of spiral-wound gaskets can be varied by the materials specified. The strength is also determined by the number of windings: the higher the number of windings, the greater the pressure load handled by the gasket. When spiral-wound gaskets are compressed, the metal layers are crushed, providing an effective seal even with uneven gasket surfaces. However, because the metal strips are deformed during compression, spiral-wound gaskets can never be reused.

As a general rule, spiral-wound gaskets should never be used with soft-seat or soft-seal designs, where the closing device, such as a plug or disk, seats against a nonmetallic surface. The force needed to compress the spiral-wound gasket is partially transmitted through the soft-seat (or seal) insert, which is more compressible than the gasket. Therefore, the soft insert is likely to extrude before the spiral-wound gasket is fully compressed. Unfortunately, the outcome is usually a damaged soft insert or a valve that leaks.

In the past, a common filler material for high-temperature spiral-wound gaskets has been asbestos paper. However, due to the controversial health and legal aspects of this material, many valve manufacturers—especially those in North America—do not offer it as a standard option. In its place, newer (and safer) filler materials have been developed or used, such as a ceramic fiber paper. Gaskets with this new filler have been known by the generic term *asbestos-free gaskets* (AFG), which can be substituted for gaskets with asbestos filler in most high-temperature applications. Their ability to seal at high temperatures is very similar to a spiral-wound gasket that contains graphite. Safety controversies and legal issues aside, asbestos gaskets are occasionally specified by users, especially by those in the power generation industry. As noted earlier, because asbestos spiral-wound gaskets are used primarily for high-temperature applications, they are typically installed in stainless-steel, carbon steel, and chrome-moly valves. Besides asbestos, common filler materials include polytetrafluoroethylene, graphite, mica, or ceramic paper.

Graphite spiral-wound gaskets are used for high-pressure–high-temperature applications associated with valves in severe service. Either 316 stainless steel or Inconel can be used for the metal windings, depending on the process fluid.

Spiral-wound gaskets can be also custom-made depending on the process fluid and its interaction with the metal windings or filler. In addition to those noted earlier, windings can be made from the following materials: 304, 315, 347, or 321 stainless steels, Monel, nickel, titanium, Alloy 20, Inconel, carbon steel, Hastelloy B, Hastelloy C-276, phosphor bronze, copper, gold, or platinum.

2.8.4 Metal O-Ring Gaskets

For exceptional severe service, *metal O-rings* are very versatile and can be applied in a wide range of services. Instead of a flat gasket design, some metal gaskets are designed as a metal O-ring, which is a tube that is circular in nature with the ends welded together. Although most are circular in shape (Fig. 2.25), they can also be formed in custom nonround or irregular shapes. Like most specialized parts, metal O-rings are more expensive than flat or spiral-wound gaskets. The hollow nature of the metal O-ring gasket allows the gasket to be compressed as the bolting or clamp is tightened, providing a reliable seal especially with high-temperature–high-pressure applications. They are especially effective in applications that involve reversing pressures.

Figure 2.25 Metal O-ring. (*Courtesy of Advanced Products Company*)

The inside volume of the rings can be pressurized for certain high-temperature–low-pressure applications.

A chief advantage of using metal O-rings is their ability to conform to the mating gasket surfaces despite any minor variations in flatness or concentricity. Like spiral-wound gaskets, once a metal O-ring has been compressed it cannot be reused but must be replaced every time disassembly takes place.

2.8.5 Metal C-Ring Gaskets

Metal C-ring gaskets are characterized by their unique shape, which is C shaped with the slot facing the inside diameter (Fig. 2.26) and the pressure side of the system. This shape allows the gasket to be self-energizing. Although more expensive than most gaskets, metal C-ring gaskets are ideal for applications that require low seating loads and high spring-back. Typically they are used for low-vacuum or low-pressure systems.

2.8.6 Metal Spring-Energized Rings

Similar in some respects to metal C-ring gaskets, *metal spring-energized rings* include metal springs inside C-ring gaskets (Fig. 2.27), combining the two elements for a highly energized seal. Such gaskets required higher seating loads but provide a more consistent seal because of the greater load and increased spring-back. Generally expensive, metal spring-energized rings are specified only when the service conditions vary widely. Because critical dimensions, such as those associated with the joint, can change in a varying service, the metal spring-energized ring design allows the gasket to expand or contract during changes in temperature or pressure, while maintaining the seal.

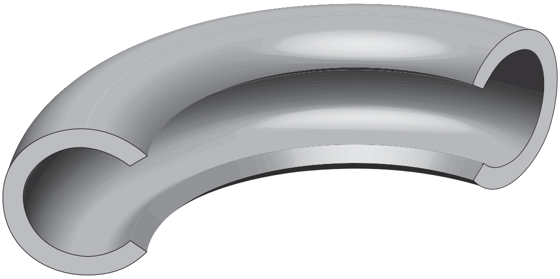


Figure 2.26 Metal C-ring. (*Courtesy of Advanced Products Company*)

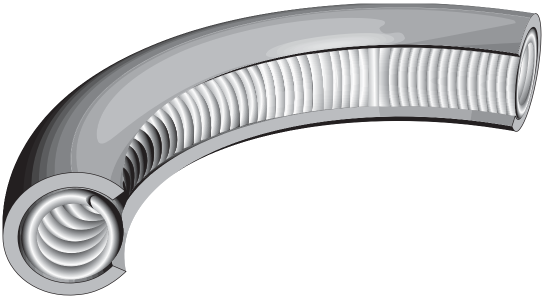


Figure 2.27 Metal spring-energized ring. (*Courtesy of Advanced Products Company*)

2.8.7 Metal U-Ring Gaskets

Metal U-ring gaskets are designed for high-pressure (up to 12,000 psi or 828 bar working pressure) and high-temperature (up to 1600°F or 871°C) applications where reliability is an important consideration. V-shaped by design (Fig. 2.28) the inside of the U faces the pressure side or faces away when used with a vacuum, using the pressure (or vacuum) to assist with function of the gasket. Because the flared ends of the gasket must keep in constant contact with the top and bottom surfaces, those surfaces must have minimal variation in flatness and must be completely parallel.

2.9 Packing Selection

2.9.1 Introduction to Packing

Any soft material encased in a bonnet (linear and some quarter-turn rotary designs) or in a body (butterfly- and some ball-valve designs)

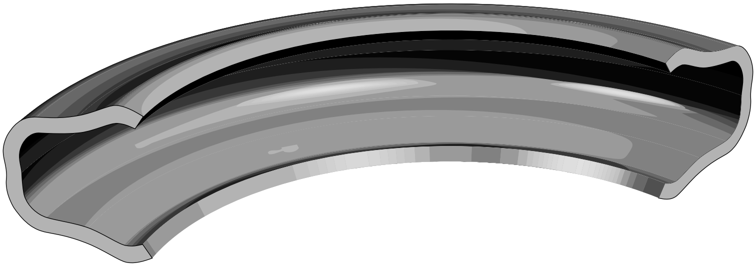


Figure 2.28 Metal U-ring. (*Courtesy of Advanced Products Company*)

used to seal a valve closure element's stem or shaft is called the *packing.* The packing is normally held in place by a packing follower or by guides, with compression supplied by the gland flange. The *packing follower* is a metal ring used to retain the packing inside the bonnet or bonnet cap, as well as compress the packing in a uniform manner. Packing followers are found in manual on–off or low-performance throttling valve designs. *Guides* are used with throttling valves to keep the stem or shaft of the closure element in correct alignment with the valve body, although the upper guide can also act as a packing follower, keeping the packing in place and transferring any force from the gland flange to the packing. The *gland flange* is a thick oblong or rectangular part that is connected to the body with bolting and straddles the guide or packing follower with the stem or shaft extending through a hole in the gland flange. When the bolting is tightened, the gland flange—through the packing follower or upper guide—transfers an axial load to the packing, compressing the packing until a seal is created against the stem or shaft and the inside of the bonnet bore. The *bonnet bore* is a term used to describe the recessed area of the bonnet or body that holds the packing. The configuration of the packing, guides, spacers, etc., is called the *packing box.*

Packing comes in a series of rings: preformed, square, or braided. *Preformed packing* is produced in a particular shape by the packing manufacturer, such as a V-ring configuration. *Square packing,* as the name indicates, is square-shaped and is formed in a solid (unbroken) ring. *Braided packing* is woven strands of a particular elastomeric material, which is manufactured similarly to rope and cut to size.

Individual rings can be grouped together, which is the case with rotary valves (Fig. 2.29), or they can be separated into upper and lower packing sets (Fig. 2.30), which is commonplace with linear valves. The difference between rotary motion—which is circular in nature—and

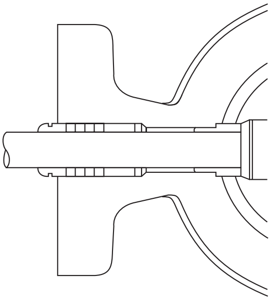


Figure 2.29 Rotary packing box design. (*Courtesy of Valtek International*)

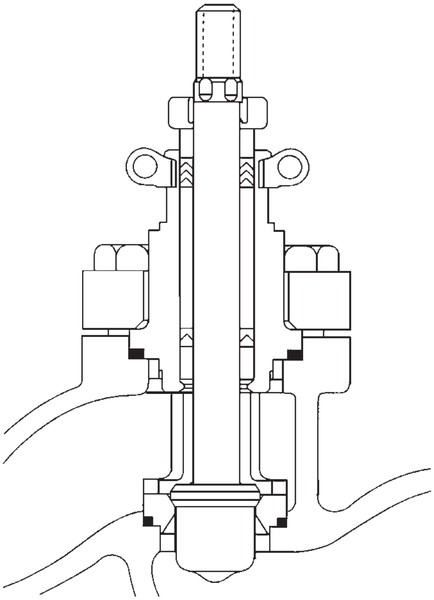


Figure 2.30 Linear packing box design. (*Courtesy of Valtek International*)

linear motion accounts for the two different designs. Because the linear motion of the plug stem involves pulling some of the medium up into the packing box, a *lower packing set* is necessary to wipe the stem free of the fluid or any particulates in the fluid stream. In other words, the lower packing set is sacrificed to the fluid conditions to allow for proper sealing in the upper portion of the packing box. The *upper pack-*

ing set is normally placed far enough away from the contaminated portion of the plug stem to avoid exposure to the fluid medium, allowing the upper set to seal properly. Because of their circular motion, rotary valves do not require a bottom packing set to wipe the fluid.

In addition, some designs provide an allowance above the packing box to allow the use of live loading. *Live loading* is a mechanical device used to apply constant force to the packing to compensate for *packing consolidation,* which is a reduction in the packing's volume due to wear, cold flow, plastic deformation, or extrusion. In most cases, when consolidation occurs, the packing box will begin to leak and the gland flange bolting must be tightened further to seal the leakage. Using a series of disk springs, live loading avoids the need to constantly retighten the packing when consolidation occurs. With the advent of strict fugitive-emission standards, live loading is becoming a popular option. (Chapter 9 provides a more detailed discussion about live loading and fugitive-emission standards.)

Depending on their material, packings produce a unique deformation when compression is applied. Because all packing materials have some degree of fluid tendencies, the axial load that is applied can result in a wide range of radial loads. Ideally, when axial load is applied, the radial load should be at its greatest in the middle of the packing set where the maximum seal occurs. Of all packing materials, soft packing materials—such as polytetrafluoroethylene packings—provide this ideal situation, as shown in Fig. 2.31.

On the other hand, harder packings—such as graphite packings—are unique in that maximum radial force provides a seal closer to the packing guides rather than in the middle of the packing. This occurs because of the high friction between the packing and the stem causes an upward axial force that is inverse to the downward force of the guide. This can be corrected by separating the graphite packing from the guide itself.

Because any variations in the surface of the stem or shaft or the packing box wall can be a potential leak path highly polished surfaces are preferred for nearly all packings. Typically, stems and shafts are polished to between 8 and 4 RMS and bonnet walls between 32 and 16 μin RMS.

Stem and shaft alignment are also critical elements of the packing box's ability to seal. If the stem flexes (inherent to smaller diameter stems) or is not concentric from inadequate guiding, the radial compression of the packing will be unequal, causing a leak path. With rotary valves, often the torque involving certain closure elements (such as a butterfly disk or an eccentric plug) can slightly misalign the

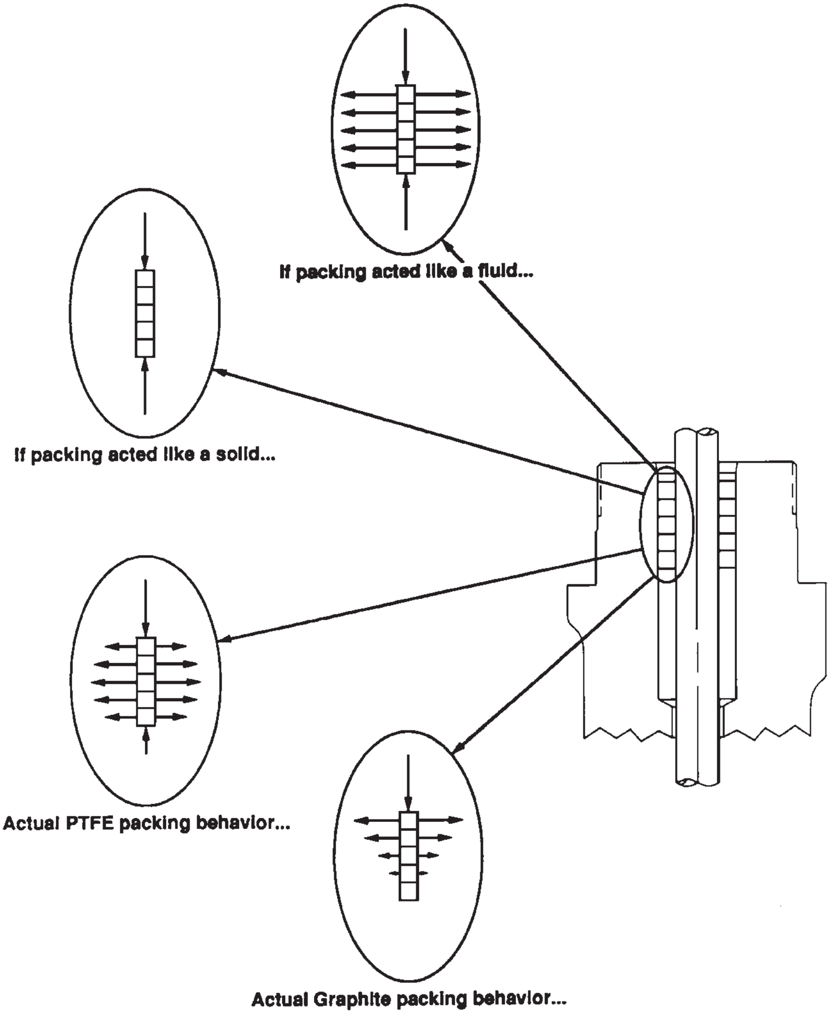


Figure 2.31 Axial pressure effects on packing. (Courtesy of Fisher Controls International, Inc.)

shaft, causing a leak path. Obviously, the type and close tolerance of guiding are critical to maintaining the concentricity of both the stem and shaft.

2.9.2 Packing Configurations

The packing box in the bonnet or body should be designed to permit a wide variety of packing configurations. A common configuration is the V-ring design (Fig. 2.32), which uses a series of V-shaped rings designed with "feather" edges and thus provides for an excellent self-adjusting seal with minimal stem or shaft friction. The user should note that this design requires the upper packing set to seal and the lower packing set to wipe the stem. The two packing sets are separated by a packing spacer. This design requires an extremely smooth bonnet or body bore—upwards to 4 μin RMS. Leakage can occur if the stem, shaft, or bore is scratched, scored, or otherwise damaged. The twin V-ring configuration is similar to the basic V-ring design, except that the lower packing set has more V-rings (Fig. 2.33), allowing for both the upper and lower packing sets to have equal numbers of rings. In theory, some users prefer twin V-ring configurations with the idea that "if a few are good, then many are better." While this configuration may be right for better wiping of the plug stem (allowing a number of rings to be sacrificed instead of a couple), it is less likely to seal. More axial load from the gland flange must be applied to compress the additional rings, which makes sealing more difficult. In addition, twin V-ring seals are harder to remain leak-free over long periods of time. Other

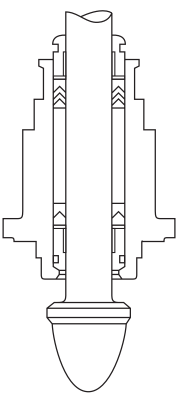


Figure 2.32 Standard V-ring packing configuration. (*Courtesy of Valtek International*)

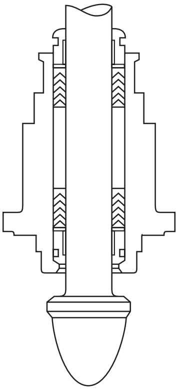


Figure 2.33 Twin V-ring packing configuration. (*Courtesy of Valtek International*)

users employ a twin V-ring configuration with a lantern ring, which is a special spacer with holes and an undercut outer diameter in the middle of the spacer. One purpose of the undercut region of the lantern ring is to allow room for a leak to freely circulate. A sniffing device can then be connected to center region of the packing box to warn of lower packing failure and the potential for future upper packing leakage if the leak migrates past the upper set of packing. Lantern rings also permit the circulation of lubrication that may be injected into the packing box. Figure 2.34 shows a typical twin V-ring–lantern-ring configuration.

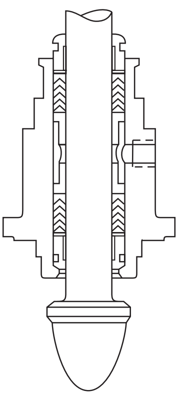


Figure 2.34 Twin V-ring packing or lantern-ring configuration. (*Courtesy of Valtek International*)

Square and braided packing can also be used for standard and twin packing configurations (Fig. 2.35). In the case of the application of square graphite packing, oftentimes a special lubricator is used with twin packing configuration (Fig. 2.36) to allow for the injection of lubrication into the graphite packing. Lubrication keeps the graphite soft and pliable while providing for smooth stem travel. Combinations of square and braided packing are used with a graphite packing configuration, which is normally applied in high-temperature services. Because die-formed solid graphite rings are extremely abrasive and create high stem friction, only one or two are used in the upper packing set. However, two solid graphite rings will not adequately seal the

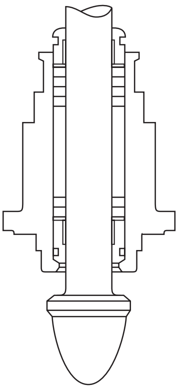


Figure 2.35 Standard square packing configuration. (*Courtesy of Valtek International*)

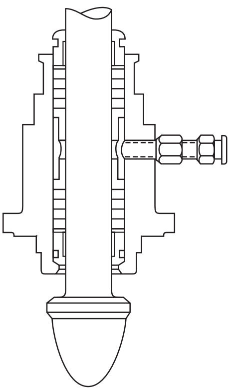


Figure 2.36 Twin square packing or lubricator configuration. (*Courtesy of Valtek International*)

packing box; therefore, braided graphite rings—which are softer—are used to complete the seal. A braided ring is commonly used for the bottom wiper set. Both standard and twin configurations are possible with square and braided packing (Figs. 2.37 and 2.38).

When the process fluid is at vacuum pressure or below atmospheric pressure, a special packing configuration is required. Because of their superior sealing ability, V-rings are used in a vacuum seal configuration (Fig. 2.39). If the process is always under a vacuum, the V-rings of both the upper and lower set of packing are inverted with the chevron facing away from the closure element. If the process pressure varies from vacuum to positive pressure at different times, a twin V-ring

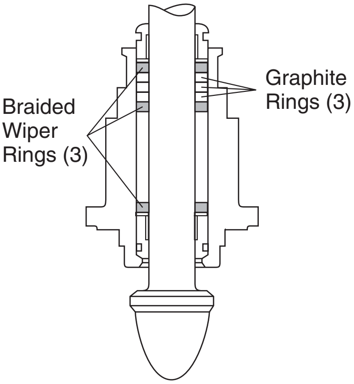


Figure 2.37 Standard graphite packing configuration. (*Courtesy of Valtek International*)

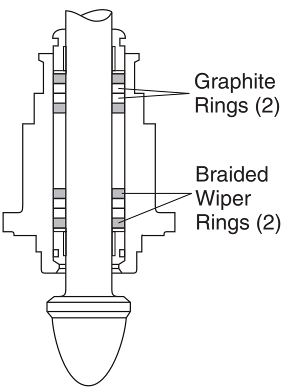


Figure 2.38 Twin graphite packing configuration. (*Courtesy of Valtek International*)

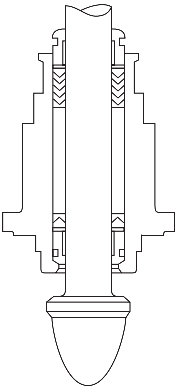


Figure 2.39 Vacuum-seal V-ring packing configuration. (*Courtesy of Valtek International*)

packing (Fig. 2.40) should be used, with the upper packing set inverted, while the lower set remains in a normal configuration. Occasionally a vacuum seal is necessary inside the packing box, which is independent of the process pressure. In this case, the twin V-ring packing configuration will permit this application. A purge may also be included to create and monitor the vacuum.

With the advent of strict fugitive-emission monitors, several configurations using special packing materials have been designed, which are detailed in Chap. 9.

2.9.3 Packing-Material Considerations

Because of the wide variety of valve applications, packing materials must be able to withstand a wide range of temperature changes, as well as withstand contact with the fluid medium, and to generate minimal stem or shaft friction. Packing materials designed for extreme temperatures must sacrifice performance in other ways. For example, graphite is a popular packing for high temperatures, but it is more difficult to achieve a seal without increasing the stem or shaft friction to the point of inhibiting performance.

As a general rule, packing materials are relatively inexpensive for general services and become increasingly more costly for services with higher temperatures and pressures or with corrosive fluids. The ideal packing material is one that operates within the temperature and pressure ranges of the service, creates minimal stem or shaft friction, holds a seal with very little material, and withstands extrusion. *Extrusion*

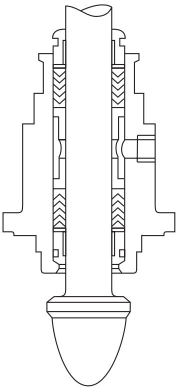


Figure 2.40 Vacuum-seal twin V-ring packing or lantern-ring configuration. (*Courtesy of Valtek International*)

occurs when overcompression of the packing box forces the packing material (especially softer packing materials) to deform and find a path to escape, in most cases up the stem or down into the body (Fig. 2.41).

A *backing ring* (sometimes called an *antiextrusion ring*) is a close tolerance ring made from a harder, less pliable material and is inserted at the top of the packing box to transfer the axial force from the gland flange bolting to the packing. However, the backing ring must also be soft enough to form a seal with the packing. In most cases, backing rings are installed on both sides of the polytetrafluoroethylene packing and provide an exact fit between the ring and the packing box wall as well as the stem or shaft. This exact fit is critical to preventing the cold flow from extruding past the backing ring. If the ring is too large, it will provide additional friction against the metal surfaces of the valve, as well as prevent the full axial force to be transferred to the packing. If the ring is too small, it will allow extrusion to occur. The backing ring should also be made from a material that allows it to retain its shape even if thermal cycling or high compression rates are required.

2.9.4 Polytetrafluoroethylene Packing

Virgin polytetrafluoroethylene packing (the compound abbreviated as PTFE) is a common and inexpensive packing material and is typically used in the V-ring design. With the combination of PTFE's elasticity and the pressure-energized design of the V-ring, little compression is required to create a long-lasting seal. Its smooth surfaces allow for

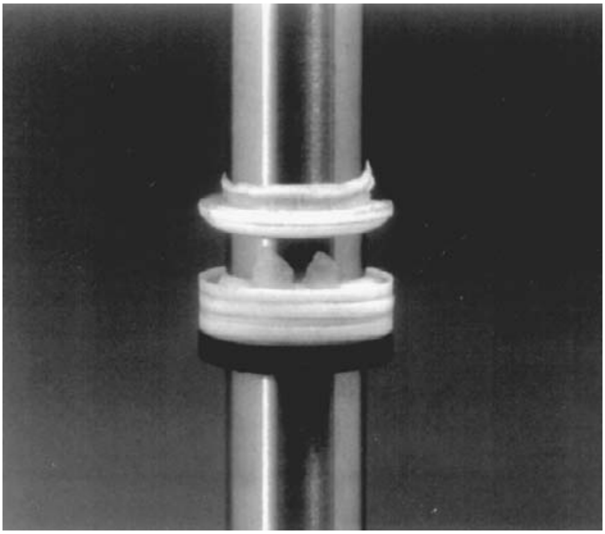


Figure 2.41 An example of packing extrusion. (*Courtesy of Fisher Controls International, Inc.*)

smooth stroking and minimal break-out force, which is the force necessary to begin the valve lift or stroke. PTFE provides very little friction; therefore, wear or erosion is usually not a concern. Because it is inert to many process fluids, it can be used in a number of general services. PTFE is also available in a braided packing.

One major drawback to PTFE is its limited temperature range. Because its thermal expansion is 10 times the thermal expansion of steel, PTFE is especially vulnerable to thermal cycling, which can result in packing loss and shorter life. As PTFE is heated by the process, it expands throughout all available space, which may lead to extrusion. As the temperature drops, the packing returns to its original volume, minus the amount lost to extrusion. Because of this loss, less force is exerted against the bonnet wall or the stem or shaft and leakage can occur. Sometimes only one temperature cycle can cause leakage. When thermal cycling is present, live loading (Sec. 9.9.5) is often recommended to allow for continued sealing of the PTFE; however, eventually through a number of thermal cycles, the packing volume will be so reduced that the force provided by live loading will be inadequate to seal the packing.

Because PTFE is very fluid, another disadvantage is its tendency to consolidate over a period of time. This long-term consolidation is called *cold flow,* and occurs when the packing is compressed several times or if live loading is used. This can occur even if minimal compression force is applied. As a result, the packing can eventually extrude out of the packing area and will not have the material volume to respond to further compression. At that point, the only option is to replace the packing. When PTFE cold flows, backing or antiextrusion rings can be installed that will slow the process. Another option is to reduce the packing compression force, which may lead to an increased chance of leakage. Another drawback to PTFE is that it is not suitable for nuclear-certified valves since radiation can quickly deteriorate the material.

Filled polytetrafluoroethylene is similar to virgin PTFE, although it includes some glass or carbon in its content, which provides for a more rigid V-ring that is less likely to consolidate. However, with less elasticity than virgin PTFE, its ability to seal requires greater force and is not as reliable. Occasionally, the user will alternate rings of virgin PTFE and filled PTFE, combining the benefits of both materials.

2.9.5 Asbestos and Asbestos-Free Packings

In the past, asbestos packing has been used as an effective high-temperature packing with good sealability. However, user interest and installation of asbestos packing have waned with recent litigation as well as with health and safety concerns. Asbestos has hook-shaped fibers that can be ingested into the lungs. Some studies indicate that such ingestion can lead to respiratory illnesses. For this reason, asbestos is not specified for use in North America. However, in some industrial areas outside of North America, asbestos is still in common use. In response to the North American process industry's move away from asbestos, a replacement packing called *asbestos-free packing* (abbreviated *AFP*) was developed. AFP uses a number of substitutes for asbestos (such as ceramic fiber paper) and is normally found in high-temperature applications.

2.9.6 Graphite Packing

As a substitute for asbestos, *graphite packing* and other carbon-based packings are specified for high-temperature applications. Generally considered to be one of the more expensive packings, graphite packings can be produced either in die-formed rings or braided rings.

Die-formed rings are produced from graphite ribbon, which is wound and then compressed in a die according to the specified pressure.

This pressure to form the rings is less than the force required to compress the rings to seal the packing box. Thus, the graphite rings reach their designed density (approximately 90 to 100 lb/ft^3 or 1440 to 1600 kg/m^3) not at formation but when installed under compression. Figure 2.42 shows the relationship between graphite density and the compression stress. This compression is not permanent, because die-formed rings are resilient to a certain extent, although not even close to the resiliency of PTFE.

Braided graphite is produced by winding small strands of graphite together, which makes it quite pliable as compared to die-formed graphite. When used as a sealing packing, it forms to the stem or shaft

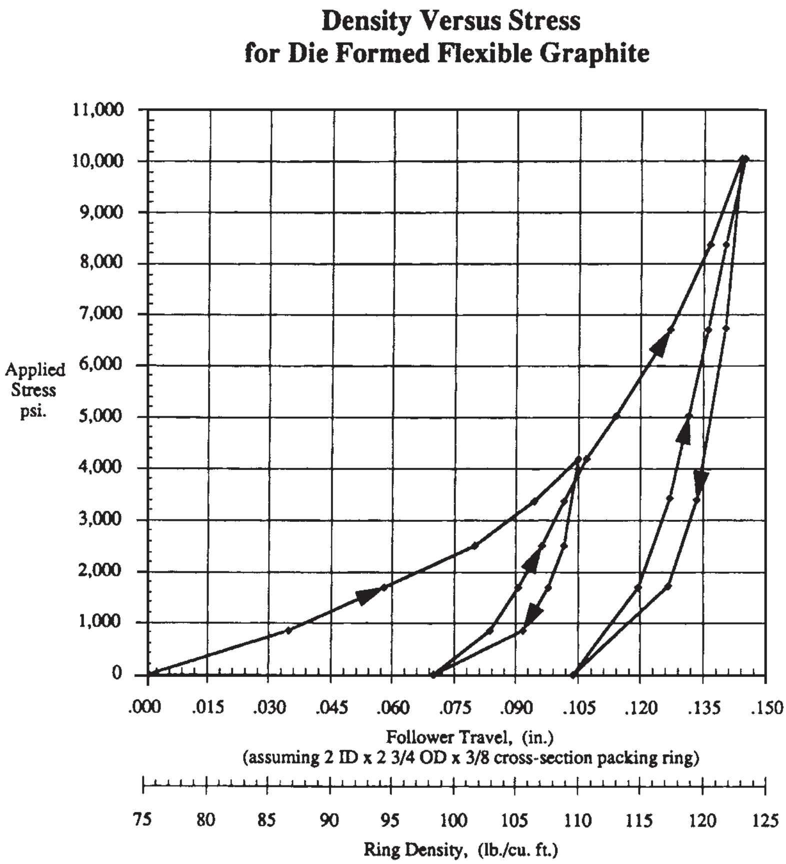


Figure 2.42 Graphite density changes according to compression stress. (*Courtesy of Fisher Controls International, Inc.*)

so well that the resulting stem friction impedes free movement of the stem or shaft. Because of this problem, braided graphite is not used to seal, but rather to act as an antiextrusion ring on both sides of the die-formed rings. However, this may cause a problem when high compression is needed, since the braided graphite has a tendency to grab the stem or shaft and not transfer the load to the die-formed rings. Hence, higher friction results from the braided graphite, yet leakage may occur because insufficient load is reaching the primary seal, which is the die-formed rings. Another problem associated with braided graphite is that it has a tendency to break down when compression exceeds 4000 psi (275 bar).

Graphite packing comes in high-density and low-density composites. For the most part, high-density graphite is much more durable and holds a seal longer but creates extremely high friction, which can lead to premature wear of the stem or shaft. It may also impede the stem or shaft from stroking freely. On the other hand, low-density graphite is softer and allows for smoother stroking, but must be retightened more often.

Graphite offers a number of advantages. Overall, graphite remains stable through a wide range of thermal cycles. Because its thermal expansion is nearly identical to steel, it does not extrude or lose a seal during thermal cycling. Graphite is fire-safe, which is important to chemical and petroleum refining industries where fire migration is a concern. It is also impervious to radiation and therefore is often recommended for nuclear service. It can be used with a wide range of process fluids without a chemical reaction, with the exception of strong oxidizers. Because graphite is bonded using compression alone, it does not have binding materials that can deteriorate when exposed to extreme temperature or certain chemicals.

The chief drawback to using graphite is that when fully compressed to provide an effective seal, it has a tendency to stick to the stem or shaft—resulting in jerky valve motion and premature wear of moving parts. Because a linear valve's plug stem may not stay in constant contact with the graphite, wear is much slower when compared to rotary valves. With rotary valves, the portion of the stem that makes contact with the packing remains constant, providing no relief to the shaft from the friction and creating faster wear. In some cases, when high compression is required, the graphite can cause the shaft to gall, which leads to packing damage and eventual leakage.

Another major problem with graphite packing is that it is extremely fragile and can be broken easily by mishandling. In addition, if graphite rings are overcompressed, they can be crushed and can lose all ability to seal as the smaller bits of graphite begin to extrude. This is a particular problem, since high compression is required to handle some fluid pressures, as well as to deform the graphite to fill any gaps or voids between the packing and the packing box wall, stem, or shaft. If an accurate torque wrench is not available, the temptation to overcompress the packing exists.

When high compression is required, another problem may occur if the compression is not completely uniform. The stem or shaft may become misaligned and create a new leak path. For this reason, use of larger stem or shaft diameters may avoid any type of flexure or off-center movement that is inherent to smaller diameter stems or shafts.

2.9.7 Perfluoroelastomer Packing

Another packing recently developed for eliminating fugitive emissions is *perfluoroelastomer packing* (compound abbreviated as *PFE*). PFE has a better temperature range than PTFE (Table 2.12) and resists chemical attack. A very versatile packing, PFE's only drawback is its cost, which is very expensive.

2.9.8 Temperature and Pressure Limits for Packing

By virtue of its close proximity to the process, the packing material can be affected by the fluid's temperature and pressure. Obviously, as the temperature increases, softer packing materials will become more fluid and are more apt to extrude out of the packing box. High pressures also can cause extrusion. Therefore, the combination of high temperatures and high pressures can greatly accelerate extrusion.

Table 2.12 provides a comparison of temperature limits for various packing materials, both standard length and extended length bonnets. The temperature limit for extended bonnets is always higher since they are longer and are designed to place the packing farther away from the temperature of the fluid. Figures 2.43–2.45 provide pressure and temperature limits for common packings.

2.9.9 Packing Lubricants

Packing boxes are often provided with a lantern ring and a tap to allow the injection of lubricant to help minimize stem friction. A num-

Table 2.12 Temperature Limitations for Common Packing Materials*†

Packing Material	Valve Rating (ANSI/*PN*)	Standard Length Bonnets	Extended Length Bonnets
PTFE	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 450° F -30 to 230°C -20 to 450°F -30 to 230°C	-150 to 600°F -100 to 315°C -150 to 700°F -100 to 370°C
Braided PTFE	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 500°F -30 to 260°C -20 to 500°F -30 to 260°C	-150 to 600°F -100 to 315°C -150 to 700°F -100 to 370°C
Glass-filled PTFE	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 500°F -30 to 260°C -20 to 700°F -30 to 260°C	-150 to 600°F -100 to 315°C -150 to 700°F -100 to 370°C
Asbestos-free Packing	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 750°F -30 to 400°C -20 to 800°F -30 to 425°C	-20 to 1200°F -30 to 650°C -20 to 1200°F -30 to 650°C
Graphite	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 750°F -30 to 400°C -20 to 800°F -30 to 425°C	-20 to 1500°F -30 to 815°C -20 to 1500°F -30 to 815°C
PFE	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 450°F -30 to 230°C -20 to 450°F -30 to 230°C	-20 to 600°F -30 to 315°C -20 to 700°F -30 to 370°C
PFE (with back-up rings)	150 to 600 *16 to 100* 900 to 2500 *160 to 400*	-20 to 550°F -30 to 290°C -20 to 550°F -30 to 290°C	-20 to 700°F -30 to 370°C -20 to 800°F -30 to 425°C

**Data Courtesy of Valtek International.*

†NOTES:

(1) ANSI B16.34 specifies acceptable pressure/temperature limits for pressure retaining materials.

(2) Appropriate body and bonnet materials must be used.

(3) Graphite packings should not be used above 800°F (424°C) in oxidizing service such as air.

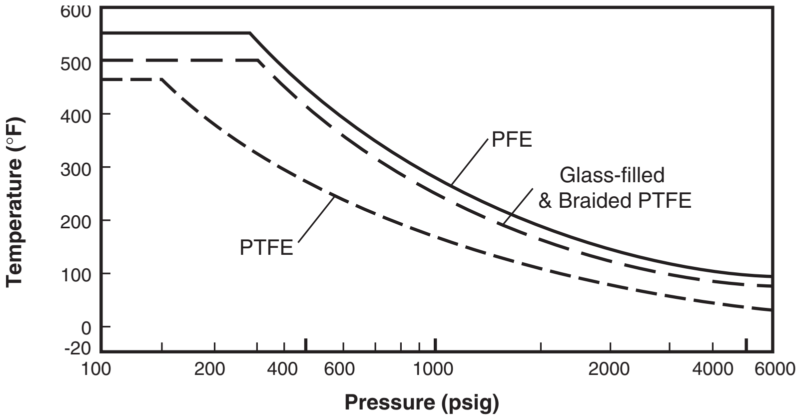


Figure 2.43 Maximum temperature and pressures for packing contained in standard bonnets. (*Courtesy of Valtek International*)

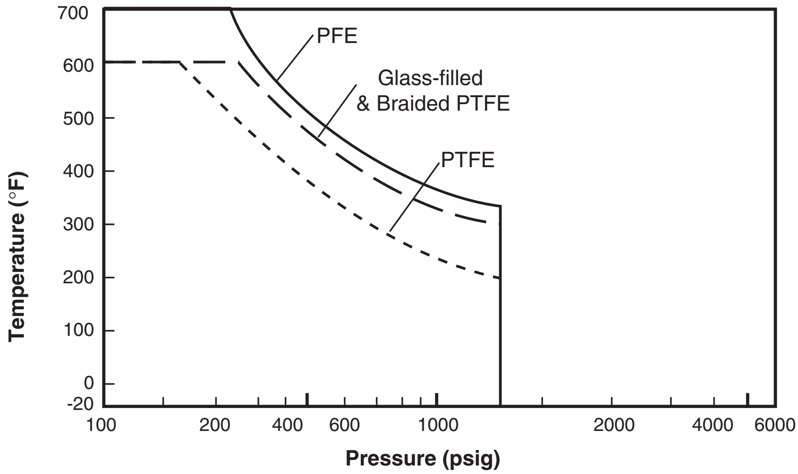


Figure 2.44 Maximum temperature and pressures for packing contained in extended bonnets, ANSI Classes 150, 300, and 600. (*Courtesy of Valtek International*)

ber of effective lubricants exist today. The best lubricant is one that reduces stem or shaft friction without increasing the chance of packing box leakage. The lubricant should not react with the process fluid nor attract dirt or other particulate matter, and it must maintain its characteristics during severe temperatures.

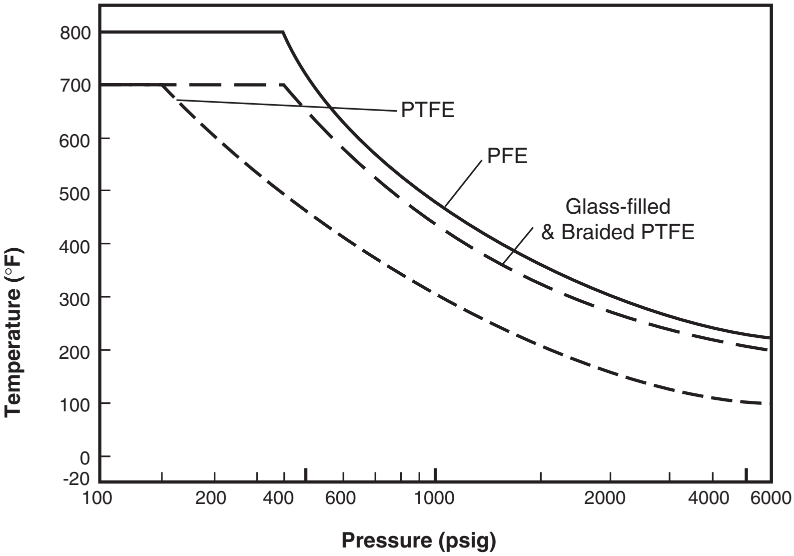


Figure 2.45 Maximum temperature and pressures for packing contained in extended bonnets, ANSI Classes 900, 1500, and 2500. (*Courtesy of Valtek International*)

The most common stem lubricant is silicone grease, which works well in temperatures up to 500°F (260°C), although it may oxidize at temperatures higher than 500°F and create a leak. In most designs, a lubricator is mounted directly to the bonnet. Turning the screw on the lubricator forces the lubricant into the packing box. An isolating valve is required for high-pressure applications to minimize the chance of pressure escaping through the lubricator. With some materials—such as graphite—lubrication is easily absorbed, making the material much more pliable and improving the sealability.

Lubrication has some limitations. Lubrication is not recommended for oxygen service or other flammable services where a petroleum-based lubricant could react with the fluid. When the packing is under high compression, the injection of additional lubricant may be difficult, if not impossible. Therefore, disassembly of the packing box is required. Forcing injection into high-compression packing boxes can sometimes damage the packing and cause leak paths.

3
Manual Valves

3.1 Introduction to Manual Valves

3.1.1 Definition of Manual Valves

By definition, *manual valves* are those valves that operate through a manual operator (such as a handwheel or handlever), which are primarily used to stop and start flow (block or on–off valves), although some designs can be used for basic throttling.

The best manual valves for on–off service are those that allow flow to move straight through the body, with a full-area closure element that presents little or no pressure drop. Usually if a manual valve is used to start and stop flow, as an on–off valve, and the manual operator is placed in a midstroke position, partial flow is possible as a throttling valve. However, some on–off designs in a midstroke position are not conducive to smooth flow conditions and may even cause turbulence and cavitation. Even though a manual on–off valve is being used in a throttling situation, it is not considered a control valve because it is not part of a process loop, which requires some type of self-actuation as well as input from a controlling device to a valve and position feedback. Throttling manual valves used to control flow are those that offer a definite flow characteristic—inherent or otherwise—between the area of the seat opening and the stroke of the closure element.

Besides on–off and throttling functions, manual valves are also used to divert or combine flow through a three- or four-way design configuration.

3.1.2 Classifications of Manual Valves

Manual valves are usually classified into four types, depending on their design and use. The first classification type of manual valves is *rotating valves,* which includes those manual-valve designs that use a quarter-turn rotation of the closure element. Rotating manual valves have a flow path directly through the body and closure element without any right-angle turns. The most common designs in the rotating-manual-valve family are plug, ball, and butterfly valves. They are most commonly used for on–off, full-flow services. In some applications they can be used for throttling control, as well as diversion and combination service. Overall, because rotating valves are inexpensive and versatile, they are the most common type of manual valve used in the process industry today. As a general rule, rotating valves—except butterfly valves—perform well in less-than-clean services, because the rotation of the closure element has a tendency to sever particulates when closing.

The second classification is *stopper valves,* which are defined as those manual-valve designs that use a linear-motion, circular closure element perpendicular to the centerline of the piping. These manual valves use a globe body to direct the flow through a right-angle turn under or above the closure element. If the valve uses an angle body, the flow continues from that right angle. If the valve has a straight-through body design, another right-angle turn is necessary after the closure element for the flow to be redirected in the same direction as the inlet. The two most common designs in the classification are the globe and piston manual valves. Because of the right-angle turns in these valves, stopper valves take more of a pressure drop than other designs. Therefore, among manual valves, they are the most frequently used throttling control and diversion applications, although they are often used for simple on–off service. Because of the stopper design, particulates can trap solids between the closure element and the seat, causing leakage; therefore, stopper valves are preferred for cleaner services.

The third classification is *sliding valves,* which are described as those manual valves that use a flat perpendicular closure element that intersects the flow. Like rotating valves and unlike stopper valves, sliding valves have a body with straight-through flow. Like stopper valves, the closure element—which is a flat element reaching from wall to wall—slides down from its full-open position (which is out of the fluid stream) into the flow stream, acting as a barrier wall. Both gate and piston valves are considered to be sliding valves. The sliding-seal design is best used for on–off service, although it can roughly control flow services where exact positioning is not required. Because the sliding valve seats at the bottom of the valve body, particulates can prevent full seating; therefore, sliding valves are usually used in nonslurry services.

The fourth classification is *flexible valves*, which are defined as valves with an elastomeric closure element and a body that allows straight-through flow. Overall, the design is similar to a sliding-valve design, although the closure element pushes against a highly flexible elastomeric or rubber insert until it meets against the bottom of the body or the other side of an elastomeric inset, literally pinching the flow closed. Both pinch and diaphragm valves are considered to be flexible manual valves. They are typically used in on–off services where tight shutoff (ANSI Class IV) is important or with slurries or other particulate-laden services.

3.2 Manual Plug Valves

3.2.1 Introduction to Manual Plug Valves

By definition, a *plug valve* is a quarter-turn manual valve that uses a cylindrical or tapered plug to permit or prevent straight-through flow through the body (Fig. 3.1). The plug has a straight-through opening. With a full-port design, this opening is the same as the area of the inlet and outlet ports of the valve.

Plug valves can be applied to both on–off and throttling services. Plug valves were initially designed to replace gate valves, since plug valves by virtue of their quarter-turn action can open and close more easily against flow than a comparable gate valve. For this reason, some plug-valve designs are built to face-to-face specifications used for gate valves.

Plug valves are commonly applied to low-pressure–low-temperature services, although some higher-pressure–higher-temperature designs exist. The design also permits for easy lining of the body with such materials as polytetrafluoroethylene (PTFE) for use with corrosive chemical services. They are also ideal for on–off, moderate throttling, and diverting applications. They are applied in liquid and gas, nonabrasive slurry, vacuum, food-processing, and pharmaceutical services. Abrasive and sticky fluids can be handled with special designs.

Depending upon the required end connection, plug valves are commonly found in sizes up to 18 in (DN 450) and in the lower-pressure classes [ANSI Classes 150 and 300 (PN 16 and 40)].

3.2.2 Manual-Plug-Valve Design

The most common plug-valve design allows for straight-through, two-way service (inlet and outlet), with the closure element in the middle

Figure 3.1 Nonlubricated, PTFE-sleeved quarter-turn plug valve. (*Courtesy of The Duriron Company, Valve Division*)

of the body. The closure element, which is a plug and a sleeve, is accessible through top-entry access in the body and is sealed by a *bonnet cap* (sometimes called a *top cap*). Most plug-valve bodies are equipped with integral flanges, but screwed ends are also common. Three-way bodies are also commonplace, with a third port typically at a right angle from the inlet. With the three-way design, the closure element is used to divert or combine the flow, depending on the installation of the valve as well as the position of the plug. Figure 3.2 shows six such three-way flow arrangements.

The face-to-face standard for plug valves is normally associated with ANSI Standard B16.10, with designations for both long and short patterns. However, many manufacturers have elected to use the face-to-face dimensions provided for gate valves. Not only does this standard better fit the design criteria of the plug valve, but it also allows quarter-turn plug valves to replace gate valves in existing process services.

The plug may be cylindrical in shape, which does present some problems in providing a solid seal between the body wall and the plug. The seal is important so that excessive leakage around the out-

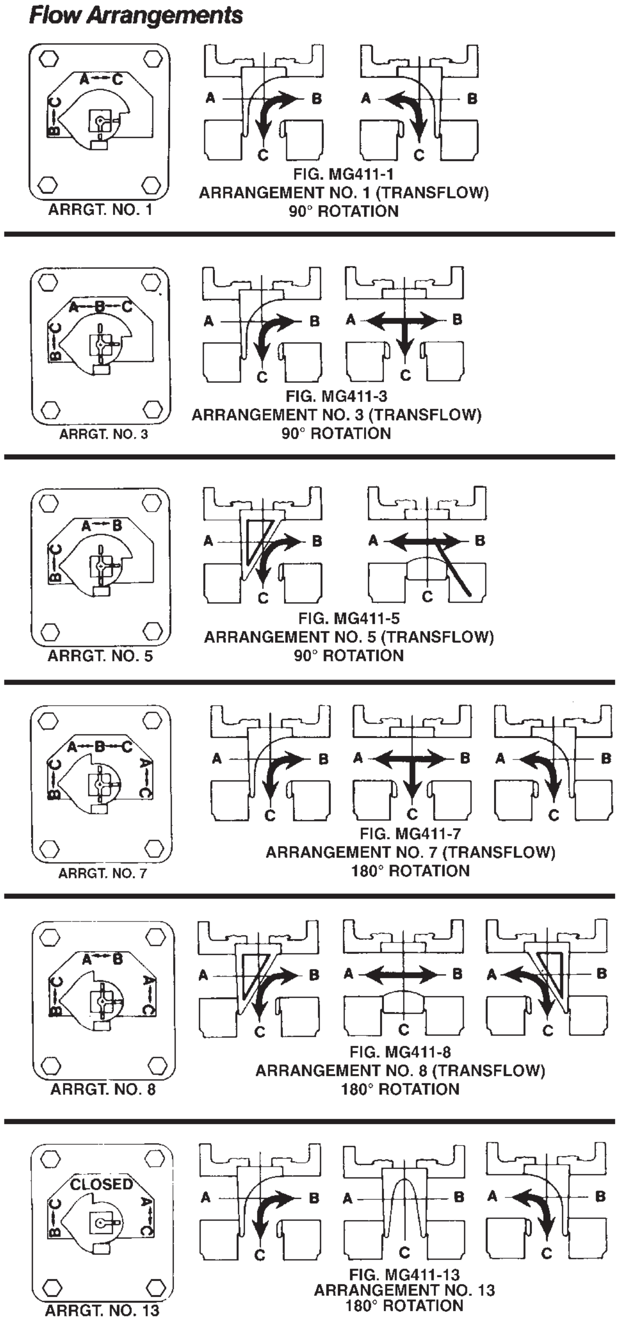


Figure 3.2 Three-way flow arrangements for quarter-turn plug valves. (*Courtesy of The Duriron Company, Valve Division*)

side diameter of the closure element does not occur. It also provides a seal for the top-works of the valve. To provide an adequate seal, three methods are commonly used: a cylindrical sleeve between the plug and the body, a series of O-rings between the plug and the body, and the injection of a malleable sealant. With the cylindrical sleeve, tightening the top-works applies compression to the sleeve against the plug. The force-fit with the O-rings provides an adequate seal also. However, the sealant design poses an inherent maintenance problem with the gradual erosion of the sealant after the valve has been stroked several times. In some high-temperature applications, the sealant may need to be reinjected after each stroke of the valve.

One of the best methods of sealing the plug and the body is to use a tapered plug, which is wedged into the plastic or other nonmetallic sleeve (again refer to Fig. 3.1). As the bonnet cap is tightened, the axial force provided by the tightening of the bonnet cap pushes the tapered plug into the softer sleeve, which provides a tight seal. The sleeve's inside diameter has a smooth surface to help seal the flow against the outside surface of the plug, while the outside surface has a series of ribs to help the sleeve hold its position in the body. The sleeve is typically manufactured from a semirigid elastomer, such as PTFE or other plastic. Because a metal surface slides with minimal friction on a plastic surface, the tapered plug is manufactured from stainless steel or carbon steel with a hard chrome surface.

Plugs can be designed with the flow port in a variety of flow areas, shapes, and functions. A common port design allows for maximum flow area, providing minimal pressure drop. The plug shape can also be characterizable (see Sec. 2.2) for throttling applications. Some cylindrical plugs have full-area ports with the same shape as the flow passages, which allow the passage of a cleaning pig. Self-cleaning ports that prevent particulate clogging or buildup are also available from certain plug-valve manufacturers. Other plugs have multihole designs to prevent or minimize the damage of cavitation (see Sec. 9.2). With cylindrical plugs the shape of the flow port is typically rectangular or round-bored, while tapered plugs are typically triangular. With throttling services, a V-shaped port is used to allow an equal-percentage flow characteristic. The ports of plugs used for three-way services are typically round with vanes contained on the inside diameter of the plug to channel flow, depending on the orientation of the plug in the body.

As previously indicated, a number of sealing designs are used to prevent the fluid from leaking through the closure element: lubricants, O-rings, and sleeves. Overall, the most common method used today is the sleeve and tapered-plug arrangement, which provides not only a good seal through the closure element but also works in conjunction with the top-work's sealing mechanism to prevent atmospheric leakage through the top-works. With most plug valves used in lower-pressure and lower-temperature service, the primary seal to the top-works is the sleeve itself, which seals between the body and the sleeve as well as between the sleeve and the plug. The top-works are further sealed with a metal thrust collar and an elastomeric diaphragm arrangement, which seals around the plug stem. The diaphragm has a spring action that helps provide constant thrust to the plug, keeping it fully seated. The outside portion of the diaphragm also acts as a gasket, sealing the gap between the body and the bonnet cap. Some plug valves—especially those used in higher temperatures and higher pressures—use packing boxes, which effectively seal the stem, but require a gland-flange arrangement to apply compression to the packing. When packing is used, a diaphragm is often not necessary, but a gasket between the body and bonnet cap is required instead.

For some corrosive chemical services (such as hydrochloric acid, sulfuric acid, waste acids, or acid brine), plug-valve bodies are completely lined with PTFE, as well as a similar coating on the plug (Fig. 3.3).

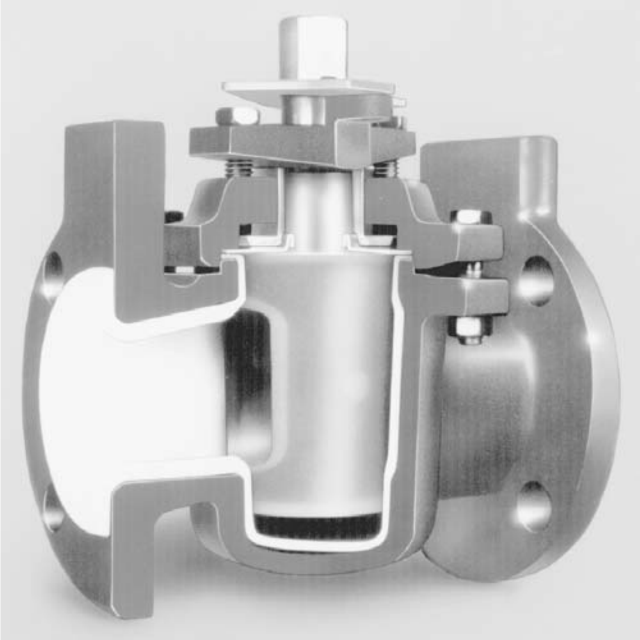


Figure 3.3 Lined quarter-turn plug valve. (*Courtesy of The Duriron Company, Valve Division*)

Other similar linings include PVDF (polyvinylidene fluoride), PVDC (polyvinylidene chloride), polyethylene, and polypropylene. Lined plug valves may have a double seal at the stem to prevent leakage to the atmosphere as well as a corrosion-resistant coating on the exterior surface of the body itself to protect the valve against process drippings. Although lined valves may be more expensive than normal plug valves, they are considerably less expensive than requesting corrosion-resistant metals. As with most corrosion-resistant materials, the lining is completely inert and impermeable. The one disadvantage of a lined valve is that the plastic-on-plastic seal provides a higher breakout torque than a metal-on-plastic seal.

To allow for the correct quarter-turn motion without over- or understroking of a plug valve, a stop-collar arrangement is used. The stop collar is designed so that it fits over the flats at the top of the plug and thus turns with the plug stem. A portion of the stop collar is designed with a quarter-turn path, which intersects a fixed key on the bonnet cap, gland flange, etc. As the plug stem is moved, the fixed key keeps the stop collar and the plug from moving outside the quarter-turn range.

3.2.3 Manual-Plug-Valve Operation

When the opening in the plug is in line with the inlet and outlet ports, flow continues uninhibited through the valve, taking a pressure drop through the reduced area of the plug port—although with a full-area cylindrical plug the pressure drop is minimal.

When the hand operator is turned to the full quarter-turn position (90°), the plug's opening is turned perpendicular to the flow stream, with the edges of the plug rotating through the sealing device (sleeve, lubricant, etc.). When the full quarter-turn rotation is reached, the port is completely perpendicular to the flow stream, creating complete shutoff. In throttling situations, where the plug is placed in a midturn position, the plug takes a double pressure drop. The inlet port's flow area is reduced by the turning of the plug away from the full-port position, taking a pressure drop at that point. The flow then moves into the full-port area inside the plug, where a pressure recovery takes place, followed by another restriction at the outlet port. Leakage is prevented through the seat by the compression of the plug against the sleeve or other sealing mechanism, while the packing or the collar–diaphragm assembly prevents leakage through the stem.

With three-way valve arrangements requiring diverting flow, flow enters at the inlet and moves through the plug, which channels the flow to one of the other two outlets. When the plug is moved 90°, the flow is channeled to the other outlet. At a midway position, flow may be equally diverted to both outlets. With combining flow, flow is directed from two inlets to a single outlet. In order for some of these arrangements to occur, the plug must be turned by half-turn (180°) instead of the typical quarter-turn action.

With larger plug-valve sizes [3 in (DN 80) or larger], the torque required for seal breakout may become somewhat excessive. This is caused by the larger contact surface between the plug and sealing device as well as any adverse operating conditions, such as a high process pressure, temperature extreme, corrosion deposits, etc. In this case, handlevers are typically replaced with geared handwheels, which reduce the torque requirement significantly. Table 3.1 shows the turning torque requirements for a typical plug valve for both handlevers and gear-operated handwheels. (The user should note that these numbers are torque values for turning the plug and do not indicate the higher breakout torque.)

3.3 Manual Ball Valves

3.3.1 Introduction to Manual Ball Valves

Related in design to the plug valve, the *manual ball valve* is a quarter-turn, straight-through flow valve that uses a round closure element with matching rounded elastomeric seats that permit uniform seating stress. The ball has a flow-through port and is seated on both sides. A common manual-ball-valve design is shown in Fig. 3.4. Because the design of manual ball valves are somewhat different than its automated cousin, the ball control valve, the designs associated with the ball control valve are covered in Chap. 4.

Manual ball valves are best used for on–off service, as well as moderate throttling situations that require minimal accuracy. In static high-pressure-drop throttling situations, where the ball's inlet port would be offset from the seal for a long period of time without moving, the velocity may cause the seal to cold flow into the port, creating some interference between the port edge of the ball and the deformed elastomer. This situation can be rectified when the manual ball valve is automated, so that the ball moves more frequently in response to a changing position signal. Ball valves are used in both liquid and gas services, although the service must be nonabrasive in nature. They can also be used in vacuum and cryogenic services.

Table 3.1 Average Run Torques for Manual Plug Valves*

Valve Size	Turning Torque at Plug Stem	Turning Torque with Gear-operator
0.5-inch DN 15	3.0 ft-lbs 4.0 joules	
0.75-inch DN 20	3.0 ft-lbs 4.0 joules	
1.0-inch DN 25	7.0 ft-lbs 9.4 joules	
1.5-inch DN 40	8.0 ft-lbs 10.8 joules	
2.0-inch DN 50	13.0 ft-lbs 17.5 joules	
3.0-inch DN 80	19.0 ft-lbs 25.6 joules	
4.0-inch DN 100	54.0 ft-lbs 72.9 joules	5.0 ft-lbs 6.7 joules
6.0-inch DN 150	140.0 ft-lbs 189.0 joules	8.0 ft-lbs 10.8 joules
8.0-inch DN 200	306.0 ft-lbs 413.0 joules	16.0 ft-lbs 21.6 joules
10-inch DN 250	580.0 ft-lbs 783.0 joules	35.0 ft-lbs 47.3 joules
12-inch DN 300	610.0 ft-lbs 827.0 joules	16.0 ft-lbs 21.6 joules
14-inch DN 350	610.0 ft-lbs 827.0 joules	16.0 ft-lbs 21.6 joules
16-inch DN 400	1170.0 ft-lbs 1587.0 joules	18.0 ft-lbs 24.4 joules
18-inch DN 450	1170.0 ft-lbs 1587.0 joules	18.0 ft-lbs 24.4 joules

**Data courtesy of Durco Valve.*

Manual Valves

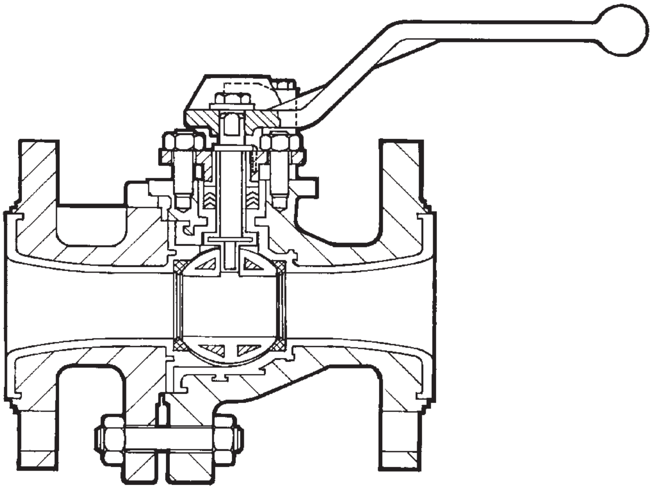


Figure 3.4 Split-body, full-port quarter-turn ball valve. (*Courtesy of Atomac/The Duriron Company*)

Because of the wiping rotary motion of ball valves, they are ideal for slurries or processes with particulates, since the ball port has a tendency to separate or shear the particulates upon closing. Occasionally, lengthy thin particulates can foul or wrap around a ball, causing a high-maintenance situation.

When ball valves are applied in highly corrosive chemical services—such as hydrochloric acid, sulfuric acid, waste acid, or acid brine—the wetted surfaces of the body and ball are completely lined with polytetrafluoroethylene, which is inert and impermeable.

Manual ball valves are typically found in sizes up to 12 in (DN 300) and in lower-pressure classes of ANSI Classes 150 through 600.

3.3.2 Manual-Ball-Valve Design

The ball-valve body features a straight-through style, allowing uninhibited flow with minimal pressure drop. A number of body configurations are available, although the most common are the split body (again refer to Fig. 3.4), solid body with side entry (Fig. 3.5), or solid body with top entry (Fig. 3.6). The defining factor for determining the body design is the complexity of installing the ball inside the body. While the split body offers the easiest disassembly and reassembly, it may present problems with an additional joint that can be affected by piping stresses as well as

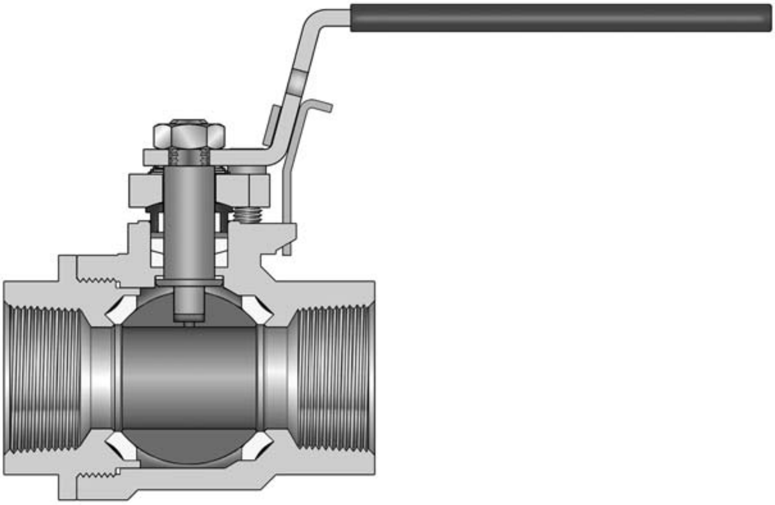


Figure 3.5 Side-entry, full-port quarter-turn ball valve. (*Courtesy of Velan Valve Corporation*)

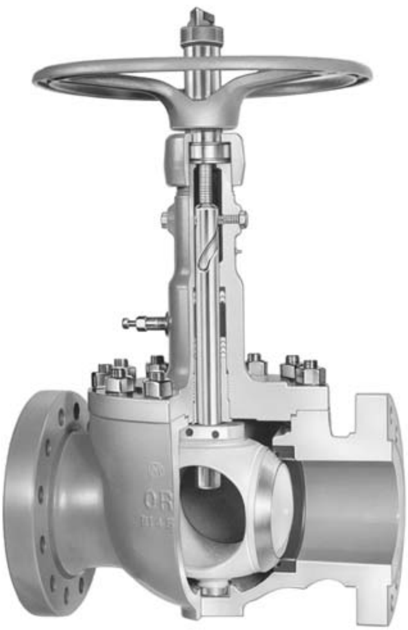


Figure 3.6 Top-entry, full-port, single-seat with tilt-action quarter-turn ball valve. (*Courtesy of Orbit Valve Company*)

another potential leak path. Face-to-face dimensions for ball valves are established by ANSI Standard B16.10, although with some pressure classifications or special designs manufacturers may use the gate valve face-to-face standard. Face-to-face dimensions are usually specified according to a short pattern (ANSI Class 150) or long pattern for higher-pressure classifications. The most common end connection used with manual ball valves is the integral-flange design.

The ball itself can be either round or tapered, depending on the internal seat design. The flow-through port is a reduced area from the body port, approximately 75 percent of the valve's full area. Full-area ports are also available when minimal pressure drop is needed, such as with on–off service, or when a pig is used to scrape the inside diameter of the pipe and a narrow flow restriction in the line would prevent this. Unlike the one-piece plug of plug valves where the stem is an integral part of the plug, the ball is separate from the stem in manual ball valves. A key slot is machined or cast into the top of the ball, into which a key machined into the bottom portion of the stem fits.

Although a ball's port is normally produced in a round flow passage, with either full or reduced area, characterizable balls are also available (Fig. 3.7) with the inlet port of the ball shaped to provide the correct flow-to-position relationship for that flow characteristic. C-shaped balls are also available for eliminating dead spots (Fig. 3.8).

When two round seats are fixed on the upstream and downstream side of the ball, this is commonly called *double seating.* The two seats are designed to conform with the ball's sealing surface. With moderate pressure drops and elastomeric seating materials, bubble-tight shutoff is possible with double-seated ball valves. Several other seating arrangements are utilized with ball valves. One of the most common arrangements is the *floating ball,* in which the ball is not fixed to the stem and is allowed some freedom of movement through the key slot. With the floating ball, the upstream fluid pressure assists the seal by pushing the ball back against the rear or downstream seat. Another seating arrangement involves a *floating seat,* in which the ball is fixed (called a *trunnion-mounted ball*) at two pivot points, and the process pressure pushes the upstream seat against the ball's sealing surface. The seat can also be prestressed during assembly, using seats that have a spring action. This design applies continuous pressure against a trunnion-mounted ball after the ball is installed, while the top-works apply a load to the entire closure element.

Most seats are made from PTFE, which provides excellent bubble-tight sealing and a temperature range that covers most general services. Buna-N and nylon materials are also specified, but may be limited

Figure 3.7 Characterized ball for throttling applications. (*Courtesy of Atomac/The Duriron Company*)

in pressure ranges and process compatibility. For higher temperatures, metal seats and carbon-based materials are specified, although higher leakage rates are common.

With ball-valve design, the stem is usually sealed by packing rings, with a packing follower and gland flange applying compression. With split bodies and solid bodies with side entry, the stem is installed through the body and the packing installed above the body. Because of the keyed slot, the ball can be turned so that the key and the slot are parallel with the flow passage, allowing the ball to enter from the side and the stem to intersect with the stem key.

With top-entry ball valves that use trunnion-mounted balls and spring-loaded seats, the ball has either an integral or separate lower post that is seated in the bottom of the body. The seats are placed on both sides of the ball and the entire assembly is placed in the body. The top-works—consisting of a bonnet cap, packing box, gland flange, and separate stem—are installed above the ball. When the bonnet-cap bolting is tightened, the resulting compression energizes the seats. The joint between the bonnet cap and the body is sealed using a gasket.

Figure 3.8 C-ball for eliminating dead spots. (*Courtesy of Atomac/The Duriron Company*)

In addition to PTFE, linings can be produced from PVDF, PVDC, polyethylene, and polypropylene. Because of the corrosive nature of the service, lined ball valves are painted with a corrosion-resistant coating on the exterior surface of the body. Although lined valves may be more expensive than normal plug valves, they are considerably less expensive than requesting corrosion-resistant metals. The one disadvantage of lined valves is that the plastic-on-plastic seal provides a higher breakout torque than the metal-on-elastomer seal.

To ensure quarter-turn motion without over- or understroking the valve, a stop-collar arrangement is used. The stop collar is designed to allow only a 90° travel of the wrench or handlever.

3.3.3 Manual-Ball-Valve Operation

With normal service, when the port opening of the ball is in line with the inlet and outlet ports, flow continues uninterrupted through the valve, undergoing a minimal pressure drop if a full-port ball is used.

Obviously, the pressure drop increases with the use of a reduced-port ball. When the hand operator is placed parallel to the pipeline, the flow passages of the ball are in-line with the flow passages of the body, allowing for full flow through the closure element. As the hand operator is turned to the closed position, the ball's opening begins to move perpendicular to the flow stream with the edges of the port rotating through the seat. When the full quarter-turn is reached, the port is completely perpendicular to the flow stream, blocking the flow.

In throttling applications, where the ball is placed in a midturn position, the flow experiences a double pressure drop through the valve, similar to a plug valve. The inlet port's flow area is reduced by the turning of the plug away from the full-port position, taking a pressure drop at that point. The flow then moves into the full-port area inside the plug, where a pressure recovery takes place, followed by another restriction at the outlet port.

When a characterizable ball is used to provide specific flow to position, as the ball is rotated from closed to open through the seat, a specific amount of port opening is exposed to the flow at a certain position, until 100 percent flow is reached at the full-open position.

3.4 Manual Butterfly Valves

3.4.1 Introduction to Manual Butterfly Valves

The manual butterfly valve is a quarter-turn (0° to 90°) rotary-motion valve that uses a round disk as the closure element. When in the full-open position, the disk is parallel to the piping and extends into the pipe itself.

Manual butterfly valves are classified into two groups. *Concentric butterfly valves* are used in on–off block applications, with a simple disk in line with the center of the valve body. Generally, concentric valves are made from cast iron or another inexpensive metal and are lined with rubber or a polymer. For throttling services, *eccentric butterfly valves* are designed with a disk that is offset from the center of the valve body. When butterfly valves are automated, eccentric butterfly valves are preferred since the disk does not make contact with the seat until closing, which prevents premature wear of the seat with the continual positioning associated with automated throttling. In most designs, simple concentric butterfly valves are used for strict on–off service and even when used in throttling applications do not lend themselves as well to automatic control as those butterfly designs designed specifically for throttling control. Because the initial development was for blocking service, concentric butterfly valves have poor rangeability and inadequate control close to the seat, while throttling butterfly valves have design modifications to allow for better flow control through the entire stroke.

Butterfly valves have a naturally high pressure-recovery factor, which is used to predict the pressure recovery occurring between the vena contracta and the outlet of the valve. The butterfly valve's ability to recover from the pressure drop is influenced by the geometry of the wafer-style body, the maximum flow capacity of the valve, and the service's ability to cavitate or choke. Overall, because of the high-pressure recovery, butterfly valves work exceptionally well with low-pressure-drop applications.

The largest drawback to using a butterfly valve is that its service is limited to low pressure drops because of its high-pressure recovery. Although flashing is not associated with butterfly valves, cavitation and choked flow easily occur with high pressure drops. While some special anticavitation devices have been engineered to deal with cavitation, users normally prefer to deal with cavitation with other valve styles that allow the introduction of internal anticavitation devices.

Butterfly valves are used for on–off and flow-control applications. Common service applications include both common liquids and gases, as well as vacuum, granular and powder, slurry, food-processing, and pharmaceutical services.

The sizes of butterfly valves are limited to 2 in (DN 50) and larger because of the limitations of the rotary design. Because of the side loads applied to the disk, the maximum size that a high-performance butterfly can reach is 36 in (DN 900). Manual designs are limited to ANSI Class 150 (PN 16), although some manufacturers offer ANSI Classes 300 and 600.

3.4.2 Manual-Butterfly-Valve Design

When compared to plug and ball valves, butterfly-valve bodies have a very narrow face-to-face. The faces of the butterfly valve body are serrated to allow the use of flange gaskets for installation in the pipeline. In many cases, this allows the body to be installed between two pipe flanges using a *through-bolt connection.* Through-bolting is only permissible with certain bolt lengths, since thermal expansion of the process itself or an external fire may cause leakage. The butterfly body can be offered in one of two styles. The *wafer body* (Fig. 3.9), sometimes called the *flangeless body,* is a flat body that has a minimal face-to-face, which is equal to

Figure 3.9 Butterfly wafer-style body. (*Courtesy of The Duriron Company, Valve Division*)

twice the required wall thickness plus the width of the packing box. Within this dimension, the disk in the closed position and the seat must fit within the flow portion of the body. Wafer-style bodies are more commonly applied in the smaller sizes, 12 in (DN 300) and less.

The *flanged body* (Fig. 3.10) is used with larger butterfly valves [14 in (DN 350) and larger] that have larger face-to-face dimensions, which are more apt to leak from thermal expansion. Generally, flanged bodies are used with high-temperature or fire-sensitive applications where potential thermal expansion is expected. The flanged style has integral flanges on the body that match the standard piping flanges with internal room between the flanges for studs and nuts.

As shown in Fig. 3.11, the *lug-body style* has one integral flange with an identical hole pattern to the piping flanges. Each hole is tapped from opposite direction, meeting in the center of the hole. This arrangement allows the body to be placed between two flanges. A stud is then inserted through the piping flange and threaded into the valve's integral flange. After the stud is securely threaded into the integral flange, a nut is used to secure the entire flanged connection.

Figure 3.10 Flanged butterfly body. (*Courtesy of Vanessa/Keystone Valves and Controls, Inc.*)

Lug bodies are used for applications in which the risks of straight-through bolting cannot be taken, such as with thermal expansion, when smaller valve size designs cannot permit two integral flanges.

The inside diameter of the butterfly valve is close to the inside diameter of the pipe, which permits higher flow rates, as well as straight-through flow. The closure element of the butterfly valve is called the *disk,* of which the outside diameter fits the inside diameter of the seat. The disk is described as a round, flattened element that is attached to the rotating shaft with tapered pins or a similar connection. As the shaft rotates, the disk is closed at the 0° position and is wide open at the 90° position. When the shaft is attached to the disk at the exact centerline of the disk, it is known as a *concentric disk* (Fig. 3.12). With a concentric disk, where the middle of the disk and the shaft are exactly centered in the valve, a portion of the disk always remains in contact with the seat regardless of the position. At 0° open, the seating surfaces are in full contact with each other. In any other position, the seating surfaces touch at two points where the edges of the disk touch the seat. Because of this constant contact, the concentric disk–seat design has a greater tendency

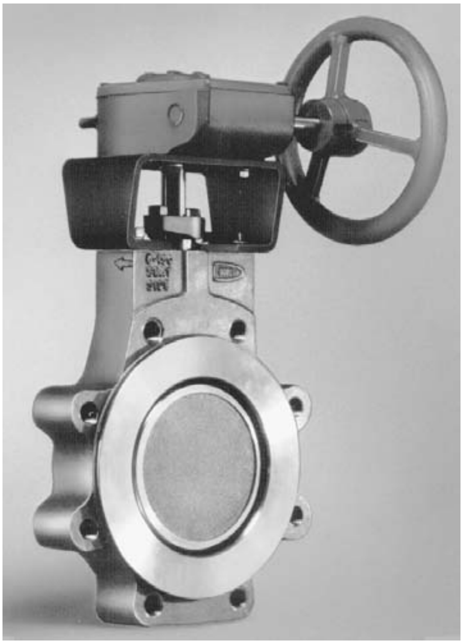


Figure 3.11 Butterfly lug-style body. (*Courtesy of The Duriron Company, Valve Division*)

for wear, especially with any type of throttling application. During throttling, a butterfly valve may be required to handle a small range of motion in midstroke, causing wear at the points of contact. Although the wear will not be evident during throttling, it will allow leakage at those two points when the valve is closed.

To overcome this problem of constant contact between the seating surfaces, butterfly-valve manufacturers developed the *eccentric cammed disk* design (Fig. 3.13). This design allows for the disk and seat to be in full contact upon closure, but when the valve opens the disk lifts off the seat, avoiding any unnecessary contact. Such designs allow for the center of the shaft and disk to be slightly offset down and away from the center of the valve, as shown in Fig. 3.14. When the valve opens, the disk lifts out of the seat and away from the seating surfaces, enough to avoid constant contact. If a manual butterfly valve is operated often, the eccentric cammed disk–seat closure element is preferred because of the minimal wear to the seat.

The seat fits around the entire inside diameter of the body's flow area and is installed at one end of the body. If a polymer is used for the

Figure 3.12 Slit-body, lined butterfly body. (*Courtesy of The Duriron Company, Valve Division*)

seat, it is called a *soft seat.* If a flexible metal is used, it is called a *metal seat.* The seat is installed in the end of the body and is held in place by a *seat retainer,* using screws or a snap-fit to keep the seat and retainer in place. After the seat and seat retainer are in place, the face of the retainer lines up with the face of the body—although some seat–retainer designs protrude slightly from the body face, allowing some final gasket compression when the body is installed in the line.

The shaft is supported by close-fitting guides, sometimes called *bearings,* on both sides of the disk, which are installed in the shaft bore, preventing movement of the shaft and disk. Also, thrust washers are often placed on both sides of the disk, between the disk and the body, to keep the disk firmly centered with the seat.

Some concentric valve bodies are lined with rubber or elastomer. This lining has two purposes: First, it protects the metal body from the process, especially if the service is corrosive or has particulates (like

Figure 3.13 Eccentric and cammed butterfly-valve design. (*Courtesy of The Duriron Company, Valve Division*)

sand) that would erode metal surfaces. Second, the lining also acts as the soft seat when the disk is in the closed position.

The rubber or elastomer lining is held in place in one of three ways: First, it can be retained in place by the flanged piping connections. Second, it can also be held in place with a tongue and groove configuration, where the rubber lining is U shaped and the body has a T machined into the inside diameter, allowing the two pieces to interlock. The third arrangement is a split-body design with a liner sandwiched between two body halves and bolted together. All three designs allow for easy removal of the lining after it becomes worn. Rubber- or elastomer-lined valves are designed with metal disks that can also be coated with a similar material. When closed, the rubber-on-rubber seat makes for a very tight shutoff in low-pressure-drop applications and mild temperatures.

With eccentric butterfly valves, a number of different resilient seat designs are used to handle higher pressures and temperatures. Some

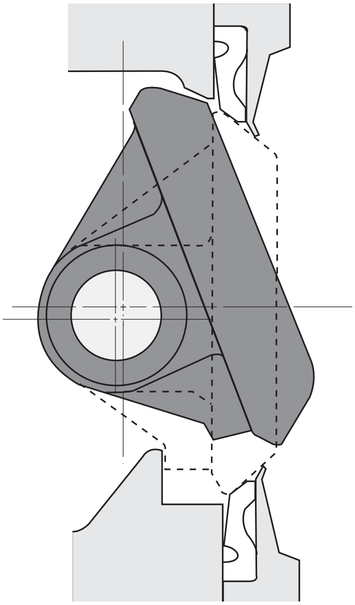


Figure 3.14 Eccentric and cammed disk rotation. (*Courtesy of Valtek International*)

seat designs use the *Poisson effect,* which refers to a concept that if a soft metal, O-ring, or elastomer is placed in a sealing situation with a greater pressure on one side, the softer seat material will deform with the pressure. When deformation takes place, the pressure pushes the material against the surface to be sealed (Fig. 3.15). With the Poisson effect, the greater the pressure, the greater the seal.

Another common resilient seat design utilizes the *mechanical preload effect,* which allows the disk's seating surface to slightly interfere with the inside diameter of the seat. As the disk moves into the seat, the seat physically deforms because of the pressure applied by the disk, causing the polymer to seal against the metal surface (Fig. 3.16). When soft seats are used, a gasket is not required to prevent leakage between the body and the retainer because the seat also acts as a gasket.

Metal seats are applied to high temperatures (above 400°F or 205°C). Metal seats can be integral to the seat retainer with a gasket placed in the space where a soft seat is normally inserted. In some designs, both a soft and metal seat can be used in tandem, allowing the metal seat to be a backup in case of the failure of the soft seat (Fig. 3.17). When butterfly valves are specified for fire-safe applications, the tandem seat is preferred.

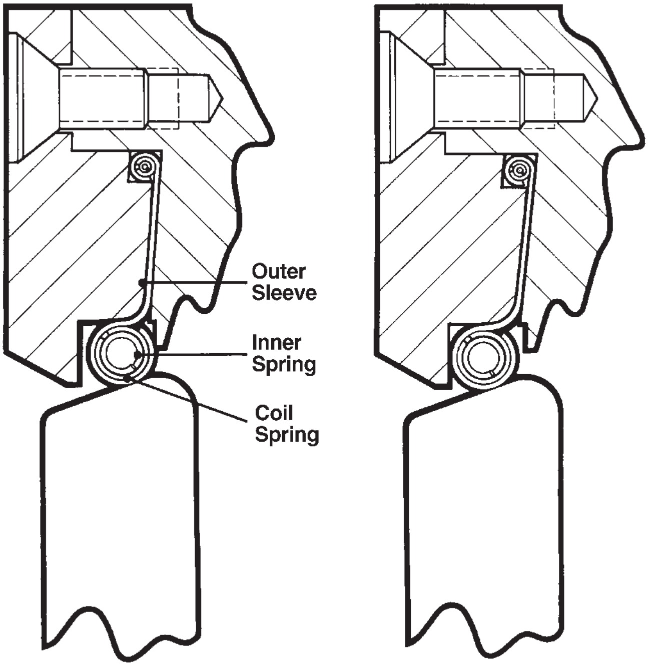


Figure 3.15 Butterfly metal seat assisted by process pressure (Poisson effect). (*Courtesy of The Duriron Company, Valve Division*)

The body contains the packing box, which is similar to other packing boxes used in plug and ball valves. The packing box features a polished bore and is deep enough to accommodate several packing rings. Normally all that is required is the packing and a packing follower. A gland flange and bolting are used to compress the packing. The shaft bore through the body is usually machined from both ends. A plug or flange cover can be used to cover the bore opening opposite the packing box. On the packing box side of the body, mounting holes are provided allowing the handlever or gear operator to be mounted.

The designs of common rotary handlevers, gear operators, and actuation systems are detailed in Chap. 5.

3.4.3 Manual-Butterfly-Valve Operation

In butterfly valves, the fluid moves from the inlet to the outlet, with the only obstruction to the flow being the disk itself. Unlike gate- or globe-valve designs, where the closure element moves out of the flow stream, the

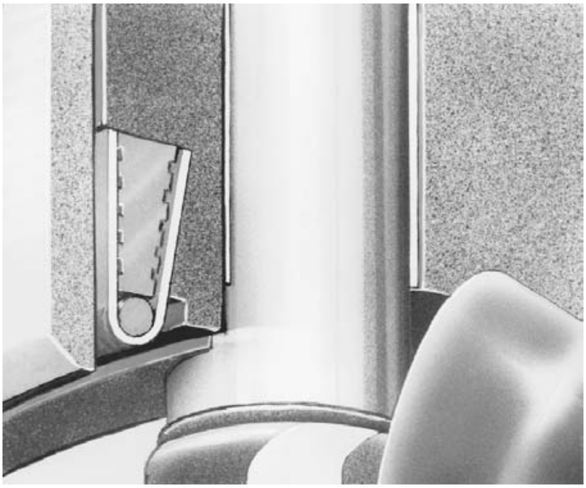


Figure 3.16 Butterfly soft seat assisted by mechanical preloading. (*Courtesy of The Duriron Company, Valve Division*)

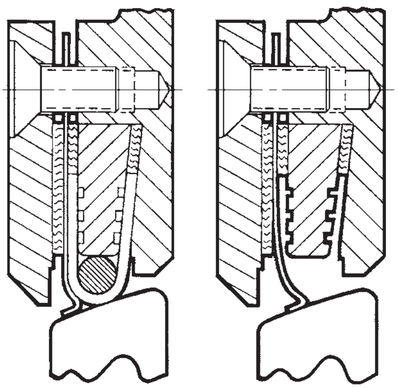


Figure 3.17 Combined metal and soft seat used for fire-safe applications. (*Courtesy of The Duriron Company, Valve Division*)

butterfly disk is located in the middle of the flow stream, creating some turbulence to the flow, even in the open position. This turbulence occurs when the flow reaches the disk and is temporarily divided into two flow streams. As the flow rejoins after the disk, turbulent eddies are created. To offset a potential problem, the disk is designed with gradual angles, as well as smooth and rounded surfaces. These design modifications allow the flow to move past the disk without creating substantial turbulence.

In closing the valve, as the manual operator is turned in a rotary motion, the shaft can turn anywhere between 0° (full-closed) and 90° (full-open). In throttling situations, as the disk closes by approaching the seat, the full fluid pressure and velocity act upon the full area of the face or back side of the disk, depending on the flow direction. Generally, the major drawback of butterfly valves is that control stability is difficult when the disk is nearing the seat. Because the rangeability of butterfly valves is quite low (20:1), the final 5 percent of the stroke (to closure) is not available because of this instability.

As the disk makes contact with the seat, some deformation is intended to take place. Such deformation allows the resilience of the elastomer or the flexible metal strip with metal seats to mold against the seating surface of the disk and create a seal.

As the valve opens, the rotary motion of the shaft causes the disk to move away from the seating surfaces. Because of the mechanical and pressure forces acting on the disk in the closed position, a certain amount of breakout torque must be generated by the manual operator to force the disk to open. The butterfly valves with the greatest requirement for breakout torque are those designs that require a great deal of operator thrust to close and seal the valve. This is why some manufacturers utilize fluid pressure to assist with the seal—in effect, less breakout torque is required.

As the valve continues to open, the disk is in a near-balanced state. As one side resists the fluid forces, the other side is assisted by the fluid forces. If both sides of the disk were identical, the disk could achieve a balanced state. However, both sides of the disk are not identical—usually the shaft is located on one side, while the other side is more flat. This creates a slight off-balance situation. Therefore, the flow direction has a tendency to either push a disk open or pull it closed. When the shaft portion of the disk is facing the outlet side, the process flow tends to open the valve. When the shaft portion is facing the inlet side, the flow tends to close the valve.

Because of the design limitations of the butterfly disk, a particular flow characteristic cannot be easily designed into a butterfly valve, unlike the trim of a globe valve. Therefore, the user must use the inherent flow characteristic of the butterfly valve, which is parabolic in nature.

3.5 Manual Globe Valves

3.5.1 Introduction to Manual Globe Valves

As shown in Fig. 3.18, a *manual globe valve* is a linear-motion valve characterized by a body with a longer face-to-face that accommodates

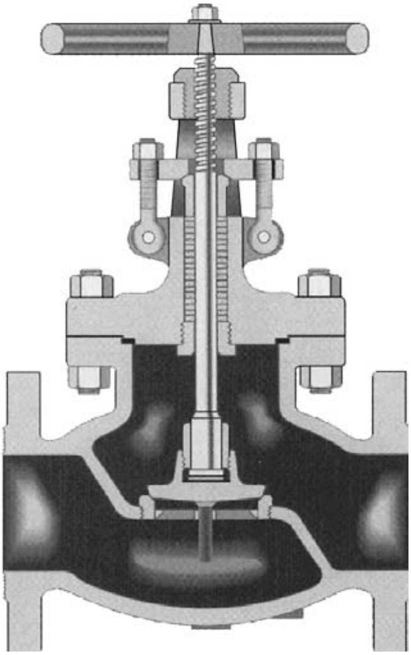


Figure 3.18 Manually operated globe valve. (*Courtesy of Pacific Valves, a unit of the Crane Valve Group*)

flow passages sufficiently long enough to ensure smooth flow through the valve without any sharp turns. It is used for both on–off and throttling applications. The most common closure element is the *single-seat design,* which operates in linear fashion and is found in the middle of the body. The single-seat design uses the plug–seat arrangement: a linear-motion plug moves into a seat to permit low flows or closure, or moves away from the seat to permit higher flows. By virtue of its design, a globe valve is not limited to an inherent flow characteristic like some quarter-turn valves. A particular flow characteristic can be designed into the shape of the plug.

Manually operated globe valves are somewhat more versatile in application than other manual valves, although the overall cost and size factors are higher. Manual globe valves can be applied in both gas and liquid services, although the service should be relatively clean to avoid particulates from being caught in the seat and creating unwanted leakage. Common manual-globe-valve applications include on–off and flow control, frequent stroking, vacuum, and wide temperature extremes. Although the globe body design can handle high-pressure classes (up to ANSI Class 2500 or PN 400), manual globe valves are usually applied to lower-pressure applications because of the thrust limitations of the hand operator. High-pressure applications will require the use of a gear operator. Using the largest available hand operators, a manual handwheel is limited to 9,000 to 13,000 lb (40 to 60 kN), although some designs—such as a nonrotating plug and a stem nut supported by a roller bearing—may surpass this limit. Globe valves can be designed to handle higher-pressure classes by increasing the wall thickness of the body and using heavier-duty flanges, bolting, and internal parts. Manual globe valves are found in sizes from 0.5 to 48 in (DN 6 to 1200).

The majority of globe-valve designs feature top-entry to the trim (the plug and seat). This design permits easier servicing of the internal parts by disassembling the bonnet flange and bonnet-flange bolting and removing the top-works, bonnet, and plug as one assembly. Unlike rotary-motion manual valves, globe-valve bodies with top-entry access can remain in-line while internal maintenance takes place. Because of top-entry access, globe valves are preferred in the power industry where steam applications require the welding of the valve into the pipeline.

The largest drawbacks to the globe valve are that it can weigh considerably more than a comparable rotary valve and is much more costly. Sizewise, it is not as compact as a rotary valve.

3.5.2 Manual-Globe-Valve Design

The globe-style body is the main pressure-retaining portion of the valve and houses the closure element. The flow passages in a globe valve are designed with smooth, rounded walls with no sharp corners or edges, thus providing a smooth process flow without creating unusual turbulence or noise. The flow passages themselves must be of constant area to avoid creating any additional pressure losses and higher velocities. With two widely spaced end connections, globe-valve bodies are adaptable to nearly every type of end connection, although the face-to-face is to too long to accommodate a flangeless design (bolting the body between two pipe flanges, which is commonplace with a rotary valve). With globe valves, mismatched end connections are also acceptable.

The globe valve's trim is more than just a closure element (because a throttling valve does more than just open or close), but rather it is a regulating element that allows the valve to vary the flow rate against the position of the valve according to the flow characteristic, which may be equal percentage, linear, or quick-open (see Sec. 2.2). Typically, this trim consists two key parts: the *plug,* which is the male portion of the regulating element, and the *seat ring,* which is the female portion. The portion of the plug that seats into the seat ring is called the *plug head,* and the portion that extends up through the top of the globe valve is called the *plug stem.* The plug stem may be threaded at the top of the stem to allow for interaction with the handwheel mechanism. The chief advantage of the single-seated trim design is its tight shutoff possibilities—in some cases better than 0.01 percent of the maximum flow of the valve. This occurs because the force of the manual operator is applied directly to the seating surface.

Two sizes of trim can be used in manual globe valves. *Full trim* is the most common and refers to the area of the seat ring that can pass the maximum amount of flow in that particular size of globe valve. On the other hand, *reduced trim* is used when the valve is expected to throttle a smaller amount of flow than that size is rated for. If full trim is used, the valve must throttle close to the seat, as well as in small increments—which is difficult to achieve with a hand operator. The preferred method, then, is to use a smaller seat diameter with a matching plug, which is called reduced trim.

The *bonnet* is a major element of the valve's top-works and acts as a pressure-retaining part, providing a cap or cover for the body. Once mounted on the body, it is sealed by bonnet or body gaskets. It also seals the plug stem with a packing box—a series of packing rings, followers or guides, packing spacers, and antiextrusion rings that prevent or minimize process leakage to atmosphere. Mounted above the packing box is the gland flange, which is bolted to the top of the bonnet. When the gland-flange bolting is tightened, the packing is compressed and seals the stem as well as the bonnet bore.

Keeping the plug head in alignment with the seat ring is important for tight shutoff. To maintain this alignment, one of two types of guiding mechanisms is used: double-top stem guiding or seat guiding. *Double-top stem guiding* uses two close-fitting guides at both ends of the packing box to keep the plug concentric with the seat ring (Fig. 3.19). These guides can be made entirely from a metal compatible with the plug to avoid galling and can include a hard elastomer or graphite liner. The ideal arrangement is for the two guides to be located as far apart as possible to avoid any lateral movement caused by the process fluid acting on the plug head. The guides, bonnet bore, and actuator stem must all be held to close tolerances to maintain a fit that will allow smooth linear motion without binding or slop.

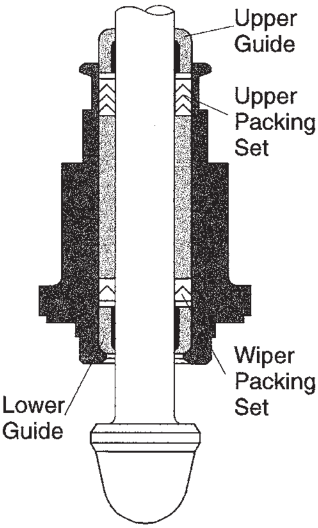


Figure 3.19 Double-top stem guiding. (*Courtesy of Valtek International*)

The other common type of guiding in manual globe valves is the *seat-guiding* design, where the plug stem is supported by one upper guide (which also acts as a packing follower). As a second guiding surface, the outer diameter of an extension of the plug head guides inside the seat (Fig. 3.20). This means that the lower guiding surface remains inside the flow stream, so therefore the process must be relatively clean. The lower portion of the plug head has openings that allow the flow to move through the plug head to the seat during opening. By varying the size and shape of these openings, reduced flow and flow characteristics can be introduced. Because the length between the upper guide and the lower guide are at a maximum length, lateral plug movement due to process flow is not an issue and the tolerances required for this type of guiding are not required to be as close as double top-stem guiding. This design minimizes any chance of vibration of the plug in service. When the plug and seat are made from identical materials, galling may occur during long-term or frequent operation. High temperatures may also lead to thermal expansion and binding.

The metal seat surface of the plug is designed to mate with the metal seating surface seat ring, using angles that slightly differ. Normally the plug has a steeper seating angle than the seat ring. This angular mismatch assures a narrow point of contact, allowing the full axial force of

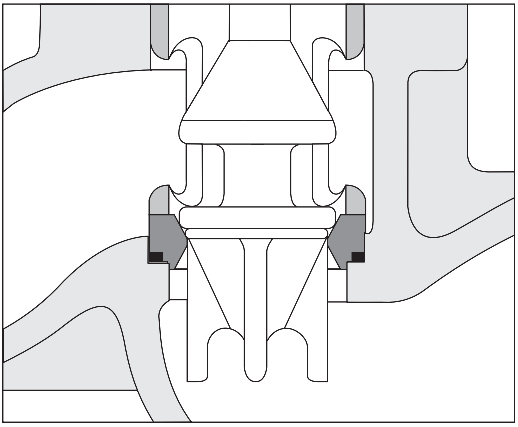


Figure 3.20 Seat-guiding design. (*Courtesy of Valtek International*)

the operator to be transferred to a small portion of the seat only, assuring the tightest shutoff possible for metal-to-metal contact. In most designs, the seat ring for manual globe valves is threaded into the body. This sometimes requires a tool to turn the seat ring into a body with limited space. With threaded seat rings, exact alignment between the seating surfaces of the plug head and seat ring must require *lapping*—a process where an abrasive compound is placed on the seat surface. The plug is then seated and turned until a full contact is achieved. Although simple in concept, threaded seats have some disadvantages. First, in corrosive or severe services the threads can become corroded, making disassembly difficult. Second, alignment between the plug and seat ring require the additional step of lapping to achieve the required shutoff. And third, in situations where vibration is present and the seat ring is not held in place by the plug in the closed position, the seat ring may eventually loosen, allowing leakage through the seat gasket and/or misalignment of the seating surfaces.

Some globe-valve applications require bubble-tight shutoff, which cannot be attained with a metal-to-metal seal. To accomplish this, a soft elastomer can be inserted in the seat ring. In this case, the seat ring is a two-part design with the elastomer sandwiched between the two halves (Fig. 3.21). The metal plug surface pressing against the seat ring's soft seat surface can achieve bubble-tight shutoff if the plug and seat-ring surfaces are concentric. Some manufacturers also insert the elastomer into the plug, which achieves the same effect (Fig. 3.22).

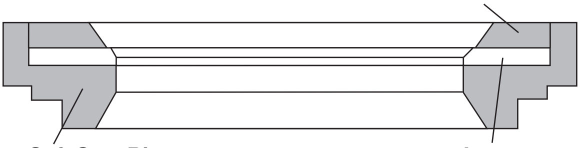


Figure 3.21 Soft-seat design. (*Courtesy of Valtek International*)

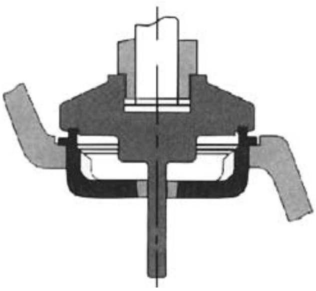


Figure 3.22 Soft-plug design. (*Courtesy of Pacific Valve Group, a unit of the Crane Valve Group*)

3.5.3 Manual-Globe-Valve Operation

Most manual globe valves use a T-style body, allowing the valve to be installed in a straight pipe. Flow enters through the inlet port to the center of the valve where the trim is located. At this point, the flow makes a 90° turn to flow through the seat, followed by another 90° turn to exit the valve.

The flow direction of globe valves is defined by the manufacturer or the application, although in most manual applications, flow direction is almost always under the plug. Seating the plug against the flow provides constant resistance but not enough to be insurmountable. With under-the-plug flow, the valve is relatively easy to close as long as the fluid pressure and flow rate are low to moderate. In addition, under-the-plug flow provides easy opening by the flow pushing against the bottom of the plug.

Manual-globe-valve trim can be modified to allow for equal-percentage, linear, or quick-open flow characteristics. As explained in detail in Sec. 2.2, flow characteristics determine the flow rate (expressed in flow coefficient or C_v) expected at a certain valve position. Therefore, with a certain flow characteristic, the user can roughly determine the flow rate by the linear position of the manual handwheel. If the plug head is in a throttling position (between full-open or full-closed), because of the pressure drop the flow moves toward the flow opening in the seat. In a throttling position, the plug head extends somewhat into the seat ring, providing only so much flow in that particular position for a given flow characteristic. As the plug retracts further away from the seat, more flow is provided. If the plug extends further into the seat, less flow is allowed. As the flow moves through the seat, fluid pressure decreases as velocity increases. After the fluid enters the lower portion of the globe body, the flow area expands again, the pressure recovers, and the velocity decreases.

As the flow enters the seat or plug area of the valve, an important design consideration is the gallery area of the body. In ideal situations the flow should freely circulate around the plug and seat, allowing flow to enter the seat from every possible direction. If the gallery is narrow in any one area (for example, in the back side of the plug), velocities can increase, causing noise, erosion, or downstream turbulence. In addition, unequal forces acting on the plug head can cause slight flexing of the plug head if it is not guided by the seat.

When the globe valve closes, the axial force of the manual handwheel is transferred to the plug. The plug's seating surface is forced against the slightly mismatched angle of the seat ring, not allowing any flow to pass through the closure element. In the full-open position, the entire seating area is open to the flow.

Process flow is retained inside the body and bonnet by the static seals of the gaskets in the end connections [if flanges or ring-type joint (RTJ) end connections are used]. Flow seeking to escape through the sliding stem of the plug is prevented by the packing's dynamic seal in the bonnet's packing box. Depending on the shutoff requirements of the user, flow may or may not be leaking through the regulating element itself.

3.6 Manual Gate Valves

3.6.1 Introduction to Manual Gate Valves

A *gate valve* is a linear-motion manual valve that uses a typically flat closure element perpendicular to the process flow, which slides into the flow stream to provide shutoff. Overall, the simplicity of the gate-valve design and its application to a large number of general, low-pressure-drop services makes it one of the most common valves in use today. It can be applied to both liquid and gas services, although it is mostly used in liquid services. The gate valve was designed primarily for on–off service, where the valve is operated infrequently. For the most part, it can be used in either liquid or gas services. It is especially designed for slurries with entrained solids, granules, and powders and cryogenic and vacuum services. As an on–off block valve, it can be designed for full-area flow to minimize the pressure drop and allow the passage of a pipe-cleaning pig. When compared to other types of manual valves, the gate valve is relatively inexpensive as well as easy to maintain and disassemble. When used with a metal seat, a gate valve is inherently fire-safe and is often specified for fire-safe service.

Gate valves do have some limitations. Gate valves do not handle throttling applications well because they provide inadequate control characteristics. Therefore, they are most commonly applied in simple on–off services as a block valve. They also have difficulty opening or closing against extremely high pressure drops. Tight shutoff is not easily attained in some applications. In addition, they can become fouled with those processes that have entrained solids. Because they are known for lengthy strokes, they take longer to open than other manual valves.

As a general rule, gate valves are divided into one of two designs: parallel and wedge-shaped. The *parallel-gate valve* (Fig. 3.23) uses a flat disk gate as the closure element that fits between two parallel seats—an upstream seat and a downstream seat. To achieve the required shutoff, either the seats or the disk gate are free-floating, allowing the upstream pressure to seal the seat and disk against any unwanted seat leakage. In some designs, the seat is spring-energized by an elastomer that applies constant pressure to the disk gate's seating surface. For the most part, the application of parallel-gate valves is limited to low pressure drops and low pressures, and where tight shutoff is not an important prerequisite.

Some variations of the parallel-gate valve have been designed for specific applications. The *knife-gate valve* (Fig. 3.24) has a sharp edge on the bottom of the gate to shear particulates or other entrained solids as well as to separate slurries. The *through-conduit gate valve* (Fig. 3.25) has a rectangular closure element with a circular opening equal to the full-area flow passageway of the gate valve. By lowering or raising the element, the opening is exposed to the flow or the barrier shuts off the flow, respectively. The through-conduit gate-valve design allows the seating surfaces of the gate to be in contact with the gate at all times. With a full-area opening, it also allows the use of a pig to scour the inside diameter of the line.

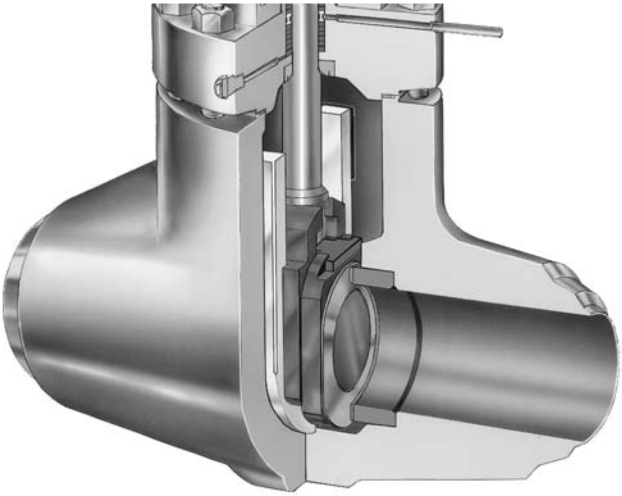


Figure 3.23 Parallel-gate valve. (*Courtesy of Velan Valve Corporation*)

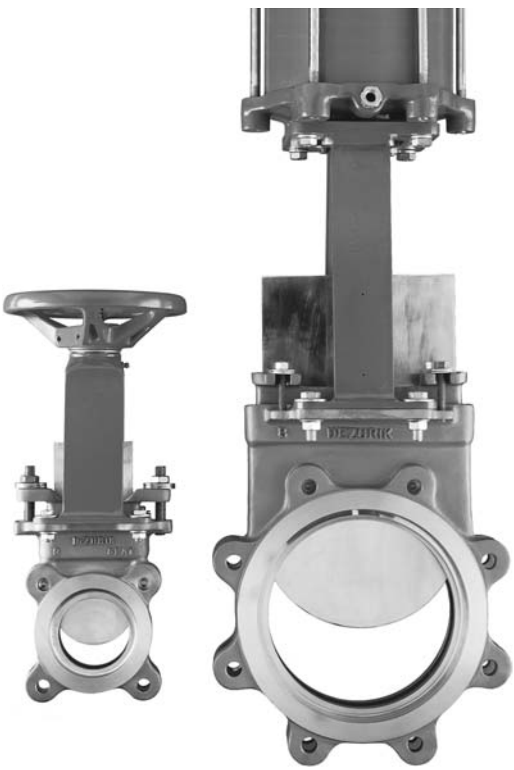


Figure 3.24 Bidirectional knife-gate valve. (*Courtesy of DeZURIK, a unit of General Signal*)

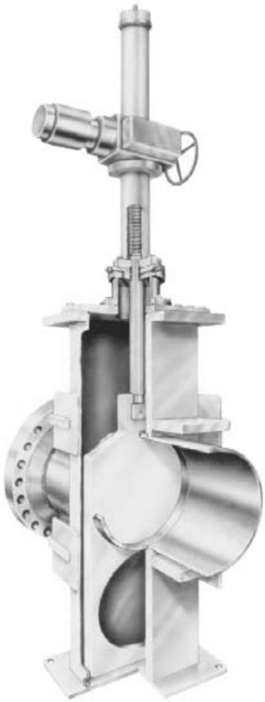


Figure 3.25 Through-conduit valve. (*Courtesy of Daniel Valve Company, a division of Daniel Industries, Inc.*)

The second classification of gate valves, the *wedge-shaped gate valve* (Fig. 3.26), uses two inclined seats and a slightly mismatched inclined gate that allows for tight shutoff, even against higher pressures. The inclined seats are designed 5° to 10° from the vertical plane, while the inclined gate can be designed with a close, but not exact angle. When the seat and gate angles are slightly mismatched, either the seat or gate is designed with some free movement to allow the seating surfaces to conform with each other as the manual actuator force is applied. This can be accomplished through either a floating seat and a solid gate or by a *flexible* or a *split-wedge gate* that provides flexure (or "give") of the gate seating surfaces (Fig. 3.27). Also, pressure-energized elastomer inserts can be installed on a solid gate to provide a tight seal (Fig. 3.28).

Gate valves are commonly found in sizes of 2 through 12 in (DN 50 through DN 300) in ANSI Class 150 (PN 16), although larger sizes are sometimes custom designed.

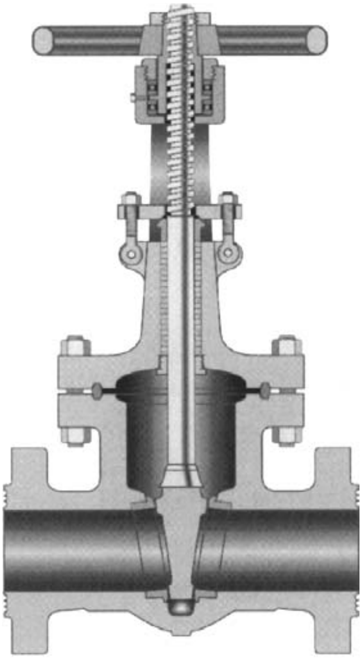


Figure 3.26 Wedge gate valve. (*Courtesy of Pacific Valves, a unit of the Crane Valve Group*)

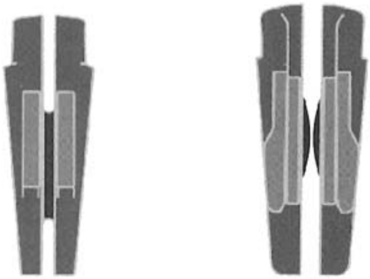


Figure 3.27 Flexible and two-piece split wedges. (*Courtesy of Pacific Valves, a unit of the Crane Valve Group*)

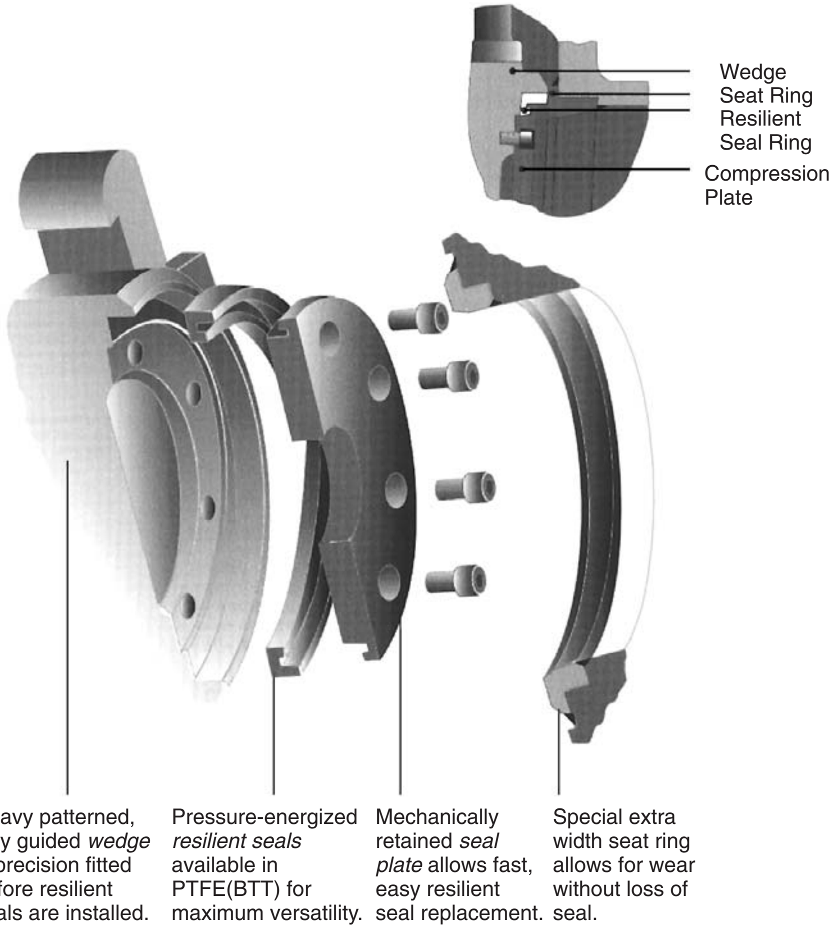


Figure 3.28 Pressure-energized wedge design (with soft seats). (*Courtesy of Pacific Valves, a unit of the Crane Valve Group*)

3.6.2 Manual-Gate-Valve Design

The gate is attached to the manual operator through the *gate stem,* which may be either fixed (rising stem) to the gate or threaded (nonrising stem) to the gate. The fixed-gate stem does not turn with the manual operator but stays stationary with the gate (Fig. 3.29). As the handwheel is turned, the threads (which are located above the packing box) retract the gate from the flow stream, causing the threaded portion of the stem to rise above the handwheel. With a threaded gate stem, the stem is threaded to the gate itself. Turning the handwheel threads the stem into the gate,

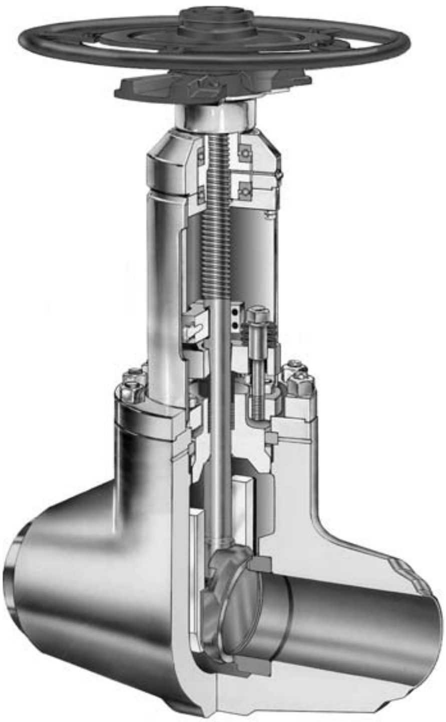


Figure 3.29 Rising-stem gate-valve design. (*Courtesy of Velan Valve Corporation*)

causing the gate to lift out of the flow stream (Fig. 3.30). The gate stem is not integral to the gate but rather uses a T-shaped collar that fits into a T-shaped slot in the gate. The T-slot is parallel to the flow stream, but may also be perpendicular to the flow stream in certain designs.

With both parallel- and wedge-shaped gate valves, the screw-driven manual operator lowers or raises the gate either into the flow stream or out of the flow stream. Most designs call for the gate to rise above the flow stream into a cavity created by the bonnet cap, although some designs allow the gate to be extended into a lower body cavity.

The body itself is a straight-through design with a special face-to-face for gate valves (ANSI Standard B16.34). In most cases, the body is designed with flanged end connection, although buttweld, socketweld, and screwed ends are sometimes offered. With simple wedge-gate valves, an integral seating surface may be machined directly into the body. Separable wedge seats are installed through the valve's top-entry opening. The upstream and downstream parallel-gate seats are a

Figure 3.30 Nonrising-stem gate-valve design. (*Courtesy of American Flow Control, a division of American Cast Iron Pipe*)

floating design and are held in place between the body and gate, which also acts as guides for the gate.

With wedge gates, guiding takes place with slot–rib combinations between the body wall and the gate. In some cases, a rib fits a matching slot in the gate, or a slot in the body fits a matching slot in the gate. Although the slots are machined, the ribs are of a cast finish, providing only simple positioning (enough to place the gate and seats into position) after which the force of the operator seals the seating surfaces together and prevents any leakage between the body and gate during guiding.

The bonnet cap not only provides top-entry to the gate, but also encloses the packing box, which seals the gate stem to prevent process leakage. A gland flange is used to apply compression to the packing.

3.6.3 Manual-Gate-Valve Operation

In the open position, as flow moves into the inlet of the valve, it continues through the flow-through globe body with minimal, if any, pressure drop occurring. This happens because most gate valves have full-area seats and are used for simple on–off blocking applications. Any pressure drop that occurs is due to the geometry of the seats, body guides, or cavities. In the open position, wedge gates and parallel gates are normally located above the seat in the upper body cavity, away from the flow stream. With conduit parallel gates, when the gate is in the open position, the flow opening of the gate is exposed to the full flow.

When the valve begins to close, the rotation of the manual operator turns the threads of the gate stem against either the operator itself (rising stem) or into the gate (nonrising stem). In either case, the gate begins its downward travel into the flow stream. Because gate valves operate in low-pressure or low-pressure-drop applications, the introduction of the gate into the flow stream is met with only moderate resistance.

As the gate valve closes, a parallel gate begins to seal the flow as the upstream pressure builds. In the parallel-gate design, the upstream pressure acts upon the floating seat, pushing the seat against the seating surface of the gate and providing the necessary seal. In the wedge-gate design, when the wedge gate reaches the seat, the thrust applied by the manual operator pushes the gate into the seat. As noted in Sec. 3.6.2, the wedge gate and the seats have some resilience as well as mismatched angles between seating surfaces. As additional thrust is applied, the wedge gate is pushed harder into the seats, providing tighter shutoff.

While the parallel valve requires minimum thrust to close, upon opening it must overcome a greater breakout force because of the upstream pressure pushing against the floating seat, especially if the valve has been in the closed position for some time. Once the flow begins to move through the seat and velocity builds, the upstream pressure is reduced and the gate slides easily to the full-open position without much resistance from the flow.

On the other hand, with a wedge valve, less breakout force is required due to the mismatched seating surfaces and wedge gate, which have a tendency to repel each other upon opening. This action is also enhanced by the natural resilience of the wedge gate. As the operator thrust is reversed and the valve begins to open, the gate and seats separate easily without hindrance or assistance by the flow.

With most applications, unless the flow rate is minimal, keeping the gate in a throttling position (midstroke) results in flutter of the gate as well as vibration and unnecessary wear. Because gate valves create additional flow turbulence in a midstroke position, they are not normally specified for throttling applications.

3.7 Manual Pinch Valves

3.7.1 Introduction to Manual Pinch Valves

A *pinch valve* is any valve with a flexible elastomer body that can be pushed together—or "pinched"—through a mechanism or through fluid pressure (Fig. 3.31). In most cases, the elastomeric body is simply a complete liner that lines the entire inside flow passage as well as the flanges. The liner keeps all moving parts outside of the flow stream; therefore these nonwetted parts can be made from less expensive materials, such as carbon steel. Because the fluid is completely contained inside the liner, the valve has the added benefit of not requiring a packing box or gaskets.

In pinch valves, when the liner seals, the sealing area is large as opposed to a single sealing point with most valves. Because of this characteristic, large objects or particulates can be trapped in the sealed area of the valve, yet the seal can be maintained. For this reason, pinch valves are ideal for particle-entrained fluids or slurries, such as processed food, sand-entrained water systems, sewage treatment, unprocessed water, granular flows, etc. Because of the resilience associated with elastomeric liners, the liner wall effectively resists abrasion

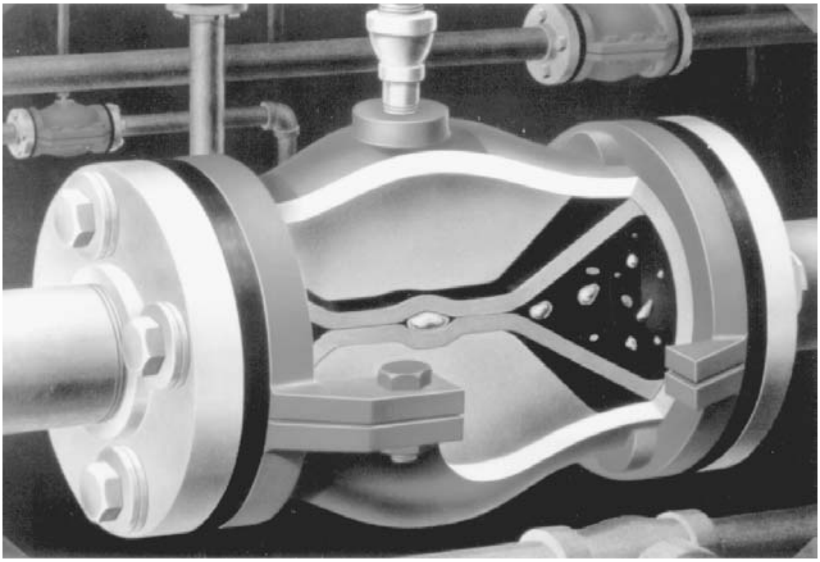


Figure 3.31 Pinch valve closing against entrapped solids. (*Courtesy of Red Valve Company, Inc.*)

damage that results from the passage of solid matter. Also, depending upon the material selection of the liner, pinch valves do exceptionally well with corrosive fluids that may attack metal surfaces.

The main limitation of pinch valves is that they are used in lower-pressure applications, because of the pressure and temperature limits of the elastomer liner. Since common liner materials (polytetrafluoroethylene, Neoprene, Buna-N, and Viton) are also associated with rubber hoses, rubber-hose pressure ratings are used for pinch valves in lieu of common valve pressure ratings. Although elastomeric pressure ratings are typically low, these limits can be increased by using specialized liners or body designs. For example, the pressure limits can be increased by using a rubber liner that has a metal mesh woven into the rubber or by injecting an outside fluid (under pressure) around the liner to offset the fluid pressure.

Another limitation is that if the pressures inside the process system move toward vacuum or if a high pressure drop is experienced, the liner can collapse with a valve in the open position unless the liner is attached physically to the closure mechanism. Pinch valves also work poorly in pulsating flows, where the liner expands and contracts constantly, causing premature failure. When these valves are used in liquid service, the liquid must have some fluid movement to allow for the displacement of fluid by the large sealing area associated with the liner. Otherwise, the incompressible nature of liquids can place additional strain on the liner and cause it to burst.

With straight-through or uninhibited flow, pinch valves have little or no pressure drop and are ideal for on–off service. Because they are commonly used in lower-pressure services, they can be throttled quite easily and provide good flow control at the last 50 percent of the stroke. This is because the smooth walls and resilience of the liner do not provide a significant pressure drop until at least 50 percent of the stroke has been achieved. Therefore, some pinch valves made for throttling service are designed for maximum opening at 50 percent to avoid using the ineffective half of the full stroke.

With services that are extremely erosive (especially with sharp particulates), the recommended practice is not to throttle the valve close to shutoff since the particulates can etch the liner, causing grooves that can potentially tear. Another positive aspect of the liner is that the smooth walls and gentle turns of the fluid produce minimal turbulence and line vibration. The resilient liner also achieves bubble-tight shutoff easily.

Most pinch valves are operated through the injection of air pressure or another fluid or by manual operators. They can also be automated and used as a control valve. Pinch valves are commonly found in sizes of 2 to 12 in (DN 50 to DN 300) in ANSI Class 150 (PN 16).

3.7.2 Manual-Pinch-Valve Design

Two designs are prevalent in pinch valves: the open body and enclosed body. The *open-body pinch valve* has no metal body casing and relies upon a skeletal metal structure. This skeletal structure consists of two cross-bars fastened to metal flange supports. The metal flange supports are designed in halves, allowing the rubber liner to be placed between the halves during assembly (Fig. 3.32). Top and bottom supports are used to connect the cross-bars or flange support halves into one structure unit. The top support is also threaded to accept the threaded handwheel stem. This stem has a free-moving connection to a moving closure bar, called the *compressor,* which is located directly above the liner. When the handwheel is turned, the compressor is lowered, squeezing the liner against the bottom support.

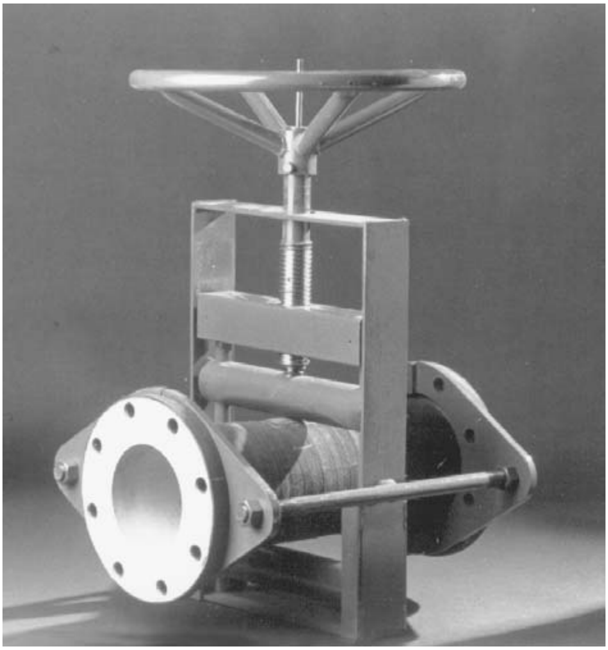


Figure 3.32 Open-body pinch valve. (*Courtesy of Red Valve Company, Inc.*)

The open-body design is fairly simple, does not require expensive metal castings, and allows for easy inspection of the liner for bulges, leaks, tears, or other failures. A primary disadvantage of this design is that the liner is exposed to the adverse effects of the outside environment, which may shorten the life of the liner.

The *enclosed-body pinch valve* has the appearance of most flow-through globe valves (Figs. 3.33 and 3.34), although the body is not actually a body but rather a protective casing for the liner. The closure mechanism is similar in design to the open-body pinch valve, except that the compressor is totally enclosed inside the body above the liner. The body can be designed with an integral bar cast into the bottom of the casing, perpendicular to the flow stream, which acts as the static closure bar. Other designs do not have this integral cast bar, called a *weir,* using the full compression of the liner against the bottom of the

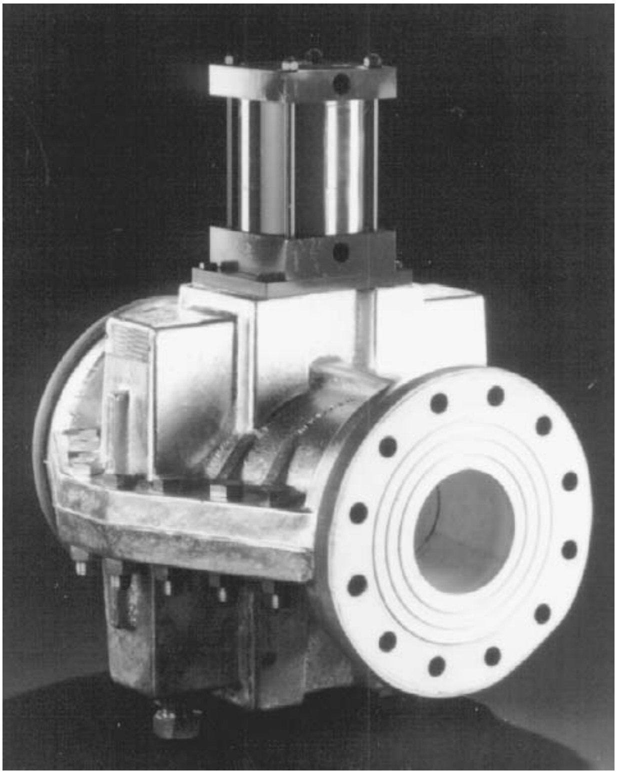


Figure 3.33 Enclosed-body pinch valve. (*Courtesy of Red Valve Company, Inc.*)

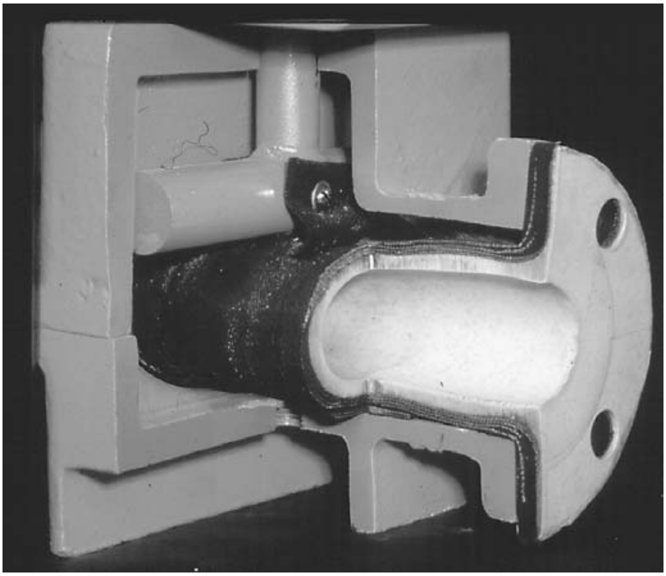


Figure 3.34 Internal view of enclosed-body pinch valve. (*Courtesy of Red Valve Company, Inc.*)

casing to shut off the valve. To allow for each assembly of the liner, the casing is split along the axis of the flow passage and bolted together. A drain can be included in the bottom half of the body as a tell-tale indicator that the liner has failed.

The advantage of using the enclosed-body pinch valve is that an outside fluid or pressure can be introduced through a tapped connection into the casing, assisting the liner in staying open or closed. For example, if the process involves a vacuum, the internal casing area outside the liner in the casing can be depressurized to vacuum. This prevents the liner from collapsing when open. In some applications, additional air pressure is introduced into the casing to assist closing.

Manual handwheel operators are simplified in pinch valves because packing boxes are not required. A threaded bonnet and threaded stem (connected to the handwheel) are used to adjust the height of the compressor when operating the valve.

Another common design of pinch valves is the *pressure-assisted pinch valve,* which uses an outside fluid pressure only to close the valve (instead of a manual operator). This design (Fig. 3.35) uses a casing similar to an enclosed-body pinch valve, except the closure mechanism and operator are missing. Fluid is introduced to the inside of the casing (but outside the

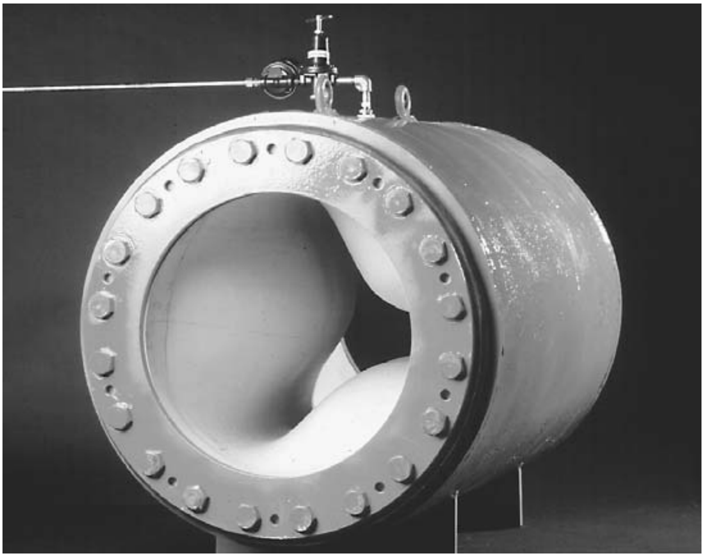


Figure 3.35 Pressure-assisted pinch valve. (*Courtesy of Red Valve Company, Inc.*)

liner) through tapped connections. When the pressure of the introduced fluid overcomes the process-fluid pressure, the liner closes and remains closed until either the system pressure increases or the introduced fluid pressure decreases. This design is very inexpensive, although it is limited to on–off service only. Throttling service is difficult because changes to the downstream pressure will automatically change the position of the valve, requiring the introduced fluid pressure to be reset.

3.7.3 Manual-Pinch-Valve Operation

Generally, pinch-valve operation is quite simple. Turning the handwheel lowers the compressor and moves the upper wall of the liner toward the static lower wall, which is supported by the bottom of the casing or the bottom bracket. In throttling situations, the manual operator is turned until the required flow is achieved and is then left in that position. In on–off situations, the manual operator is turned until the closure mechanism presses the upper wall of the liner against the lower wall, which is supported by either a static lower bar or the bottom of the casing. As more thrust is applied by the manual operator, the two surfaces seal more tightly. When the pinch valve opens, the turning action of the manual operator is reversed, raising the compressor and allowing the liner to open as it moves toward its natural relaxed position. As the opening increases, the pressure of the process pushes the liner against the closure mechanism, which widens the flow area more as the closure mechanism is raised. Eventually, at the full-open position, the liner will have reached its full area capacity.

With pressure-assisted pinch valves, fluid pressure is introduced above and below the body liner. When the introduced pressure is greater than the pressure of the process fluid, the liner begins to collapse. As the introduced pressure builds, the liner begins to collapse, restricting the flow until the liner totally collapses and forms a seal between the upper and lower walls. When the introduced pressure is relieved or if the process pressure builds, the forces reverse and the liner walls separate, opening the pinch valve.

3.8 Manual Diaphragm Valves

3.8.1 Introduction to Manual Diaphragm Valves

Related to the pinch valve, the *diaphragm valve* uses an elastomeric diaphragm instead of a liner in the body to separate the flow stream from the closure element (Fig. 3.36). When compressed, the diaphragm is pushed against the bottom of the body to provide bubble-tight shutoff.

The advantage of a diaphragm valve is similar to a pinch valve. The closure element is not wetted by the process and therefore can be made from less expensive materials in corrosive processes. The flow stream is straight-through or nearly straight-through, providing a minimal pressure drop, which makes it ideal for on–off service, as well as avoiding the creation of turbulent flow. Diaphragm valves can also be used for throttling service. However, maintaining a throttling position close to the bottom of the valve body can sometimes result in erosion as the particulates can cut grooves into the diaphragm and the bottom of the body. Because the diaphragm is contained in a pressure-retaining body, a diaphragm valve is able to handle somewhat higher pressures than a pinch valve, although the overall pressure and temperature ratings are dependent upon the flexibility of the material or reinforcement of the diaphragm. The design of the body flow passageway (such as the addition of a weir) has a bearing on the amount of flexibility of the diaphragm. Another advantage of the diaphragm valve is that if the diaphragm fails, the body can contain the fluid leak better than a pinch valve casing.

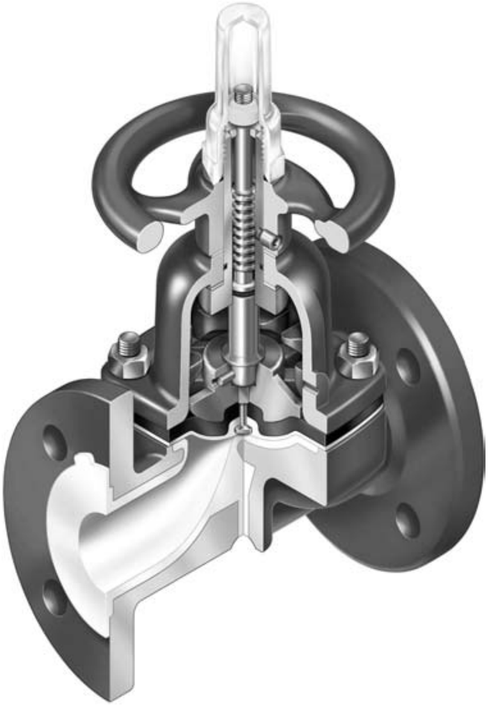


Figure 3.36 Diaphragm valve. (*Courtesy of ITT Engineered Valves*)

Diaphragm valves have an application similar to pinch valves. The resilience of the diaphragm allows it to seal around particulates in the fluid, making it ideal for service with slurries, processed food, or solid-entrained fluids.

When compared to the pinch valve, the primary disadvantage of the diaphragm valve is that the body can cost more than a pinch-valve casing because the body material must be compatible with the process fluid. Also, while the resilience of the diaphragm has a tendency to resist erosion damage from the process, the body can erode, making shutoff more difficult.

Depending on the design, diaphragm valves are available in larger sizes than pinch valves, typically up to 14 in (DN 350), although some special designs can reach up to 20 in (DN 500). Because of the pressure limitations of the liner, diaphragm valves are nearly always rated at ANSI Class 150 (PN 16).

3.8.2 Manual-Diaphragm-Valve Design

Two designs are typically associated with diaphragm valves: the straight-through design and the weir-type design. The *weir-type diaphragm valve* has the same construction as the straight-through design except for the body and diaphragm. As shown in Fig. 3.37, the body has a raised lip that raises up to meet the diaphragm, allowing the use of a smaller diaphragm. This body design is self-draining, which makes it ideal for food-processing applications. Since the diaphragm can be made from heavier materials, the body can also be used with high-pressure services, which are not as flexible and do not allow for a long stroke. Heavier, reinforced diaphragms also allow the weir-style design to be used for vacuum services. On the other hand, the *straight-through diaphragm valve* has a body in which the bottom wall is nearly parallel with the fluid stream, allowing the flow to move uninhibited through the valve with no major obstructions (Fig. 3.38). The flexibility of the diaphragm allows it to reach the bottom of the valve body. Above the diaphragm is the compressor, a round part shaped much like the body's flow passage, which is connected to the hand-

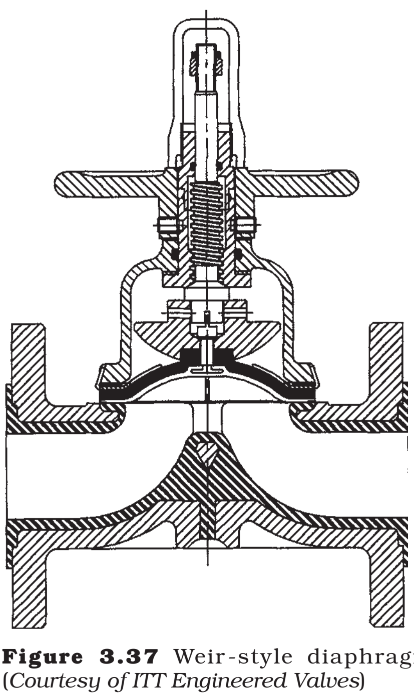


Figure 3.37 Weir-style diaphragm valve. (*Courtesy of ITT Engineered Valves*)

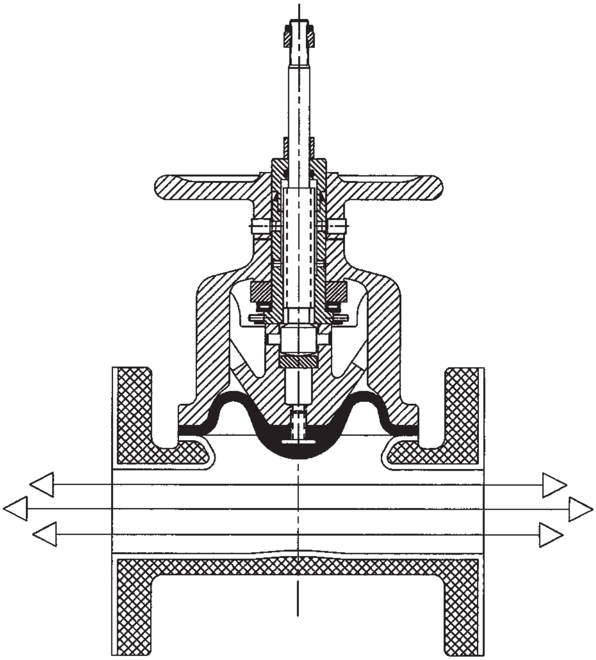


Figure 3.38 Straight-through diaphragm valve. (*Courtesy of ITT Engineered Valves*)

wheel stem. The diaphragm is attached to the bottom of the compressor to ensure that the diaphragm is lifted out of the flow stream during full-open. The compressor, the nonwetted portion of the valve, and the handwheel mechanism are contained by the bonnet cap, which is bolted to the body. The diaphragm itself is used as the gasket between the body and bonnet cap and prevents leakage to the atmosphere.

3.8.3 Manual-Diaphragm-Valve Operation

Manual-diaphragm-valve operation is very similar to the operation of a pinch valve. Turning the handwheel lowers the compressor, which begins to move the diaphragm toward the bottom wall of the body. In throttling situations, the manual operator is turned until the required flow is achieved and is then left in that position. In on–off situations, the manual operator is turned until the compressor pushes the diaphragm against the bottom wall of the body. As more thrust is applied by the manual operator, the two surfaces seal tighter until maximum compression is achieved. When the diaphragm valve opens, the turning action of the manual operator is reversed, raising the compressor and allowing the diaphragm to separate from the bottom body wall. As the opening increases, the pressure of the process keeps the liner pushed against the compressor, widening the flow area as the closure mechanism is raised. Eventually, at the full-open position, the compressor is fully retracted inside the bonnet cap and the diaphragm is out of the flow stream. At this point, the valve is at its full-area capacity.

Generally, diaphragm valves offer an inherent equal-percentage flow characteristic, which tends to move toward linear when installed (Sec. 2.2.5).

4
Control Valves

4.1 Introduction to Control Valves

4.1.1 Definition of Control Valves

Over the years, some confusion has existed between the definitions of a throttling valve and a control valve. Some use the words interchangeably because they both have a similar purpose: to regulate the flow anywhere from full-open to full-closed. For the most part, a *throttling valve* is any valve whose closure element has a dual purpose of not only opening or blocking the flow but also moving to any position along the stroke of the valve, thus regulating the process flow, temperature, or pressure. Using the term *closure element* is not adequate in describing this portion of the throttling valve; thus, for purposes of differentiation, the term *regulating element* is used to describe any portion of the valve that allows for throttling control. A throttling valve is designed to take a pressure drop in order to reduce line pressure, flow, or temperature. The interior passageways of a throttling valve are designed to handle pressure differential, while on–off valves are designed to allow straight-through flow without allowing a significant pressure drop. Because the purpose of the throttling valve is to provide reduced flow to the process, rangeability is a critical issue. The valve's trim size is almost always smaller than the size of the pipeline or flow passages of the valve. Using a full-size valve in a similarly sized pipe will provide poor controllability by not utilizing the entire stroke of the valve. Throttling valves must have some type of mechanical device that uses power supplied by a human being, spring, air pressure, or hydraulic fluid to assist with this positioning. Some manually operated on–off valves can be used or adapted for throttling service. Pressure regulators are also considered throttling valves, since they vary in the position of the regulating element to maintain a constant pressure downstream.

By definition, a *control valve* (also known as an *automatic control valve*) is a throttling valve, but is almost always equipped with some sort of actuator or actuation system that is designed to work within a control loop. As discussed in Sec. 1.2.5, the control valve is the final control element of a process loop (consisting of a sensing device, controller, and final control element). This involvement with the control loop is what distinguishes control valves from other throttling valves. Manually operated valves and pressure regulators can stand alone in a throttling application, while a control valve cannot, hence the difference: a control valve is a throttling valve, but not all throttling valves are control valves. In some cases, a manually operated valve can be converted to a control valve with the addition of an actuation system and can be installed in a control loop—thus in the pure sense of the definition it becomes a control valve.

Control valves are seen as two main subassemblies: the body subassembly and the actuator (or actuation system). This chapter will concentrate on the operation, design, installation, and maintenance of body subassemblies, while Chap. 5 will detail actuators and actuation systems.

Generally, control valves are divided into four types: globe, butterfly, ball, and eccentric plug valves. Variations of these four types have resulted in dozens of different available designs, the most common of which will be covered in this chapter. Each design has specific applications, features, advantages, and disadvantages. Although some control valves have a wider application than others, no control valve is perfect for all services, and each design should be examined to find the best solution at minimal cost.

4.2 Globe Control Valves

4.2.1 Introduction to Globe Control Valves

Of all control valves, the linear-motion (also called rising-stem) globe valve is the most common, due in part to its design simplicity, versatility of application, ease of maintenance, and ability to handle a wide range of pressures and temperatures. The globe valve is the most commonly found control valve in the process industry, although demand is not as great with the advent of high-performance rotary valves, which offer lower cost and smaller packages, size for size. Sizes range from 0.5 to 42 in (DN 12 through DN 1000) in lower-pressure classes (up through ANSI Class 600 or PN 100); from 1 to 24 in in ANSI Classes 900 to 2500 (PN 160 through PN 400); and from 1 to 12 in in ANSI Class 4500 (PN 700).

By definition, a *globe valve* is a linear-motion valve characterized by a globe-style body with a long face-to-face dimension that accommodates smooth, rounded flow passages. The most common regulating element is the *single-seat design,* which operates in linear fashion and is found in the middle of the body. The single-seat design uses the plug–seat arrangement, where a plug moves into a seat to permit low flows or away from the seat to permit higher flows. The alternative to the single-seat arrangement is the double seat, which will be discussed in detail in Sec. 4.2.4.

The advantages of globe control valves are many—hence their overall popularity. Generally, globe valves are quite versatile and can be used in a wide variety of services. The same valve can be used in dozens of different applications as long as the pressure and temperature limits are not exceeded, and the process does not require special alloys to combat corrosion. This versatility allows for reduction in spare parts inventory and maintenance training. Their simple linear-motion design permits a wider range of modifications than other valve styles. Because of the linear motion, the force generated by the actuator or actuation system is transferred directly to the regulating element; therefore, a minimal amount of the energy to the regulating element is lost. On the other hand, rotary valves lose some transfer energy and accuracy because of the dead band (amount of input change needed to observe shaft movement) associated with linear- to rotary-motion linkage. For this reason, globe valves are capable of high performance and are used in applications where such performance is mandatory.

A major advantage to using globe control valves is their ability to withstand process extremes. They are designed to work in extremely high pressure drops, handling pressure differentials of thousands of pounds of pressure (or hundreds of kilograms per centimeter squared). Globe valves can be designed to handle higher pressure classes by increasing the wall thickness of the body and using heavier-duty flanges, bolting, and internal parts. Severe temperatures can be handled with extension modifications to the bonnet or the body, keeping the top-works (actuator, positioner, supply lines or tubing, and accessories) away from the process temperature.

An important advantage of a globe control valve is that it can have a flow characteristic designed into the trim or the regulating element itself—unlike butterfly valves whose design only allows for an inherent characteristic.

Most globe control valves with single seats have top-entry to the trim (plug, seat, and cage or retainer). This allows easy entry into the valve to service the trim by removing the bonnet flange and bonnet-flange bolting and removing the top-works, bonnet, and plug as one assembly. Unlike rotary valves, globe valves can remain in the line during internal maintenance. For this reason, globe valves are preferred in the power industry where steam applications require the welding of the valve into the pipeline.

As mentioned earlier, the main disadvantages of globe valves are that, size for size, they are larger, heavier, and more expensive than rotary valves. They present seismic problems because of their greater height—a problem where an earthquake or process vibration could cause the top-works to place stress on the body subassembly or line.

Another disadvantage is that globe valves are restricted by the significant stem forces required by the throttling process. Globe valves with pneumatic actuators are restricted to sizes smaller than 24 in (DN 600), or 36 in (DN 900) with a hydraulic or electrohydraulic actuator. With higher-pressure classes, the bulk of the globe-valve body assembly, as well as the stem forces, decreases the size availability even more. When large flows must be regulated beyond the size capabilities of a globe valve, users sometimes divide the flow between two smaller pipelines, preferring smaller valves. In some cases, butterfly or eccentric disk rotary valves are used instead.

4.2.2 Globe-Control-Valve Design

In describing the design elements of a globe valve, the *globe body* is the main pressure-retaining portion of the globe valve, which has matching end connections to the piping and also encloses the trim (Fig. 4.1). The flow passages in a globe valve are designed with smooth, rounded walls without any sharp corners or edges, thus providing a smooth process flow without creating unusual turbulence or noise. The flow passages themselves must be of constant area to avoid creating any additional pressure losses and higher velocities. Globe-valve bodies are adaptable to nearly every type of end connection, except the flangeless design. Obviously with a long face-to-face dimension, the long bolting required between two pipe flanges would be susceptible to thermal expansion during temperature cycles.

The *single-seated trim* is more than just a closure element, because a throttling valve does more than just open or close; rather, it is a regulat-

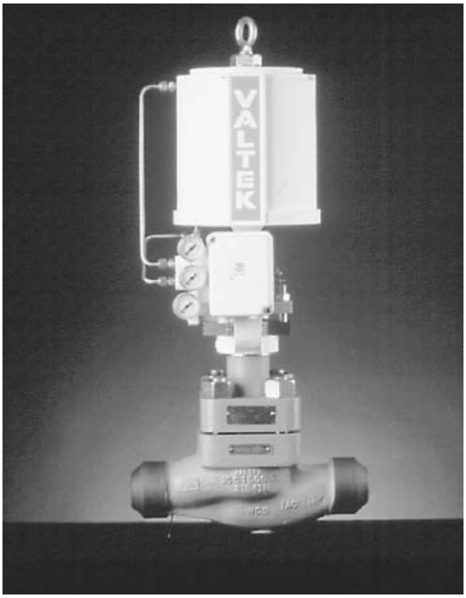


Figure 4.1 Globe-style control valve. (*Courtesy of Valtek International*)

ing element that allows the valve to vary the flow rate with respect to the position of the valve according to the flow characteristic, which may be equal percentage, linear, or quick open (Sec. 2.2). Typically, the trim consists of three parts: the *plug,* which is the dynamic portion of the regulating element; the *seat ring,* which is the static portion; and the *seat retainer* or *cage.* The portion of the plug that seats into the seat ring is called the *plug head,* and the portion that extends up through the top of the globe body subassembly is called the *plug stem.* The plug stem is threaded to the *actuator stem,* allowing a solid connection without any play or movement. The actuator stem is assembled to an actuator piston or diaphragm plate, which transfers pneumatic or hydraulic force to the regulating element. The basic advantage of the single-seated trim design is that it allows the tightest shutoff possible, usually better than 0.01 percent of the maximum flow or C_v of the valve. This is because the actuation force can be applied directly to one seating surface. The greater the actuation force, the greater the shutoff of the valve.

Two sizes of trim can be used in globe valves. *Full trim* refers to the area of the seat ring that can pass the maximum amount of flow in that particular size of globe valve. On the other hand, *reduced trim* is used when the globe valve is expected to throttle a smaller amount of flow than that size is rated for. If full trim is used, the valve would have to throttle close to the seat as well as in small increments—which is difficult for some actuators. The preferred method, then, is to use a seat ring with a smaller seat area—with a matching plug—which is defined as *reduced trim.* Most manufacturers offer four or five sizes of reduced trim for each size of valve.

The *bonnet* is an important pressure-retaining part that has two purposes. First, it provides a static cap or cover for the body, sealed by bonnet or body gaskets. Second, it seals the plug stem with a packing box—a series of packing rings, followers or guides, packing spacers, and antiextrusion rings that prevent or minimize process leakage to atmosphere. Mounted above the packing box is the gland flange, which is bolted to the top of the bonnet. When the gland-flange bolting is tightened, the packing is compressed and seals the stem as well as the bonnet bore.

Guiding the plug head in relation to the seat ring is accomplished by two types of guiding: double-top stem guiding or caged guiding. *Double-top stem guiding* uses two close-fitting guides at both ends of the packing box to keep the plug concentric with the seat ring (see Fig. 4.2). These guides can be made entirely from a compatible, dissimilar

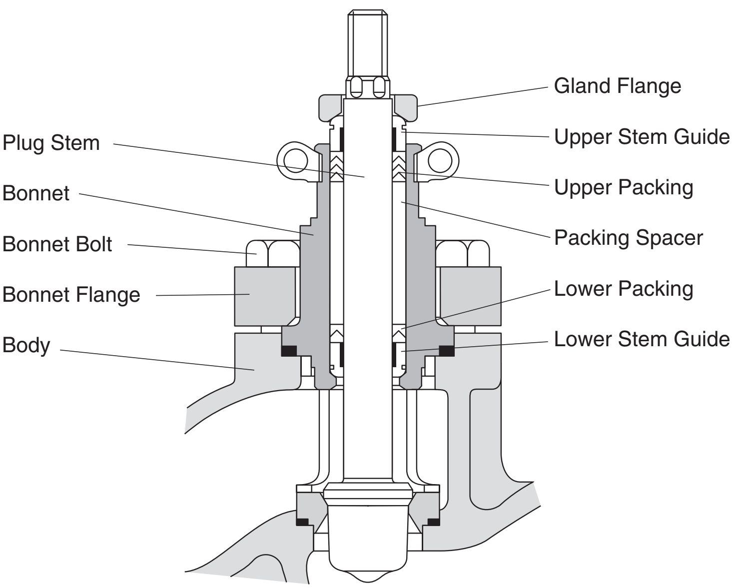


Figure 4.2 Double-top stem guiding in a globe valve. (*Courtesy of Valtek International*)

metal with the plug to avoid galling or can include a hard elastomer or graphite liner. The key element of double-top stem guiding is that the guides must be widely separated to avoid any lateral movement from the process fluid acting on the plug head, which is exposed to the forces of the process stream. The guides—as well as the bonnet bore and the actuator stem—must be held to close tolerances to maintain a fit that will allow a smooth linear motion without binding or slop. To avoid lateral movement as the process impinges on the plug head, some plugs have large-diameter stems to resist flexing. However, when compared to smaller-diameter stems, larger plug stems do have an increased circumference, which increases the sealing surface and the possibility of seal leakage as well as packing friction. However, the stem-friction problem is easily rectified by using higher thrust actuators, such as piston cylinder actuators, which can easily handle the increased stem friction.

The second type of guiding configuration is *caged guiding.* With the cage-guided design (Fig. 4.3), the upper guide is placed at the top of the packing box and the lower guiding surface is placed inside the flow stream, using the outside diameter of the plug head to guide within the inside diameter of the cage. Because the distance between

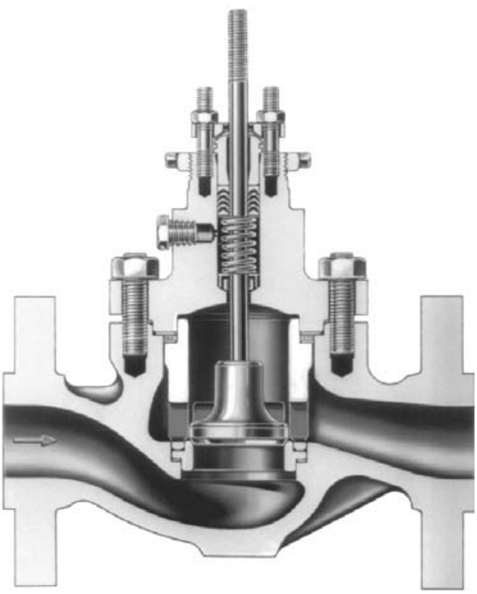


Figure 4.3 Caged-guided trim in a globe valve. (*Courtesy of Fisher Controls International, Inc.*)

the upper guide and the lower guide is at a maximum length, lateral plug movement due to process flow is not an issue and the tolerances required for this type of guiding are not required to be as close as double top-stem guiding. This also permits the use of smaller-diameter plug stems, providing a smaller sealing surface and decreased stem friction (which is necessary when lower-thrust diaphragm actuators are used). Caged guiding also minimizes any change of vibration of the plug in service and helps support the weight of the plug head. Because this guiding surface is in the flow stream, the process must be relatively free from particulates, or binding or scoring may occur. In some situations, identical or similar materials between the plug head and the cage may gall during prolonged operation. High temperatures may also lead to thermal expansion and binding. Galling and temperature problems can be remedied using guiding rings made from an elastomer or nongalling metal, which are installed in grooves machined into the plug head.

Cages are designed with large flow holes (anywhere from two to eight) that allow passage of the flow into or from the seat, depending on the flow direction. They can also be modified to allow a staged pressure drop—reducing the pressure drop and velocities inside the valve to avoid cavitation, flashing, erosion, vibration, or high noise levels. To ensure the alignment of the plug seating surface with the seat-ring seating surface, some designs combine the cage and the seat ring into one part. This one-piece design maintains the concentricity between the inside diameter of the cage and the inside diameter of the seat.

The cage is also used to determine the flow characteristic. The flow holes in the cage are sometimes shaped such that the plug lifts from the seat ring. In this way a certain percentage of the flow hole is opened up, allowing only so much flow at that portion of the stroke. By varying the size and shape of the hole, certain flow characteristics can be generated. Figure 2.2 in Chap. 2 shows a variety of shapes available according to the flow characteristic.

In trim designs that do not feature cages (such as those that use a seat-ring retainer or screwed-in seats, which is discussed later), the plug head can be machined to a particular shape that provides an inherent flow characteristic. Figure 2.3 in Chap. 2 shows how the contour of a plug head can be turned to provide the flow characteristic. In contrast, Fig. 4.4 shows a V-port plug head, which is cylinder shaped with V-shaped grooves machined into the cylinder for a linear characteristic.

With globe valves, the seating surface of the plug is designed to make full contact with the seating surface of the seat ring at the point

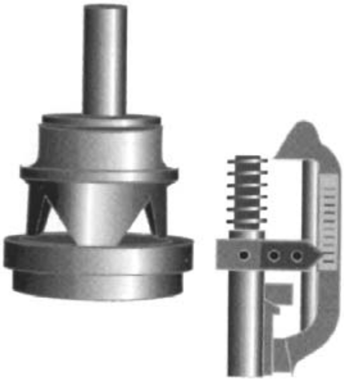


Figure 4.4 V-ported characterized plug. (*Courtesy of Pacific Valves, a unit of the Crane Valve Group*)

of closure. Although some early valve designs used identical angles, current designs use angles that slightly differ, with the plug at a steeper angle than the seat ring. This slight mismatch ensures a narrow point of contact, allowing the full axial force of the plug to be transferred to the seat, ensuring the tightest shutoff possible for metal-to-metal contact (normally ANSI Class II shutoff is standard, although Class IV shutoff can be achieved with high-thrust cylinder actuators). Even with ANSI Class IV shutoff, metal-to-metal seats can never completely shut off the flow, as the classification allows a small amount of process leakage.

The seat ring is fixed in the body, while the gap between the seat ring and the body is sealed by a gasket. The seat ring can be fixed in the body by one of two arrangements. First, a common method of fixing the seat ring is through a retained arrangement. The seat ring is inserted into a slightly larger diameter machined into the body and held in place by a part between the bonnet and the seat ring, called the *seat retainer.* If the retainer is used to guide the plug head, it is called a cage, but it can serve the dual purpose of retaining the seat ring. If the diameter machined into the body is wide enough, the seat ring will have some play, allowing lateral movement, which can lead to a quick, easy method of correct plug and seat-ring alignment. During assembly, and before the bonnet-flange bolting is completely tightened, a signal can be sent to the actuator to seat the plug in the seat, providing the correct alignment between the matching seat surfaces of the two parts. After the plug and seat ring are aligned, the bonnet-flange bolting is tightened and the subsequent force is transferred through the retainer or cage to secure the location of the seat ring with the plug head. If the seat ring does not have this self-adjustment feature, its seating surface must be lapped with the seating surface of the plug head. *Lapping* is the process in which an abrasive compound is placed on the seat-ring seat surface and the plug is seated and turned until a full contact is achieved. The retained seat ring is also known for easy disassembly, especially in corrosion-prone applications, since it just lifts out of the body once the bonnet and seat retainer or cage are removed. The only disadvantage to retained seat rings is that they work best when a high-thrust actuator is used, since high seating force is needed to ensure a good seat-ring gasket seal.

The second method of securing the seat ring is the threaded arrangement in which the seat ring is threaded into the body. This process normally requires a special tool from the manufacturer to turn the seat ring into the body. The major advantage of this design is that no other part is needed to retain the seat ring, providing a simplified trim arrangement, as well as no cage or seat retainer to restrict the flow. With three-way or double-seated valves, the use of seat retainers or cages is not possible from a design standpoint, and the only alternative is to use threaded seats. Threaded seat rings are widely used with cryogenic applications in which the top of the body must be elongated to provide a fluid barrier between the process and the packing box and top-works.

The disadvantages of threaded seats are threefold. First, and most evident, the threads can become corroded, making disassembly difficult, if not impossible in some long-term situations. Second, alignment between the plug and seat ring will require the extra step of lapping to achieve the required shutoff. And third, in situations in which vibration is present and the seat ring is not held in place by the plug in the closed position, the seat ring may eventually loosen and allow leakage and misalignment. Overall, the disadvantages of the threaded seat ring far outweigh the advantages; therefore many newer single-seat designs use the retained arrangement. When a seat retainer or cage is not possible or preferred and the application is too corrosive to allow a threaded seat ring, a split-body arrangement is a practical substitute.

Some globe-valve applications require bubble-tight shutoff (ANSI Class VI), which cannot be attained with a metal-to-metal seal. To accomplish this, a soft elastomer can be inserted in the seat ring. In most designs, the seat ring is made from two parts with the elastomer sandwiched between the two, as shown in Fig. 4.5. The combination of the metal plug surface pressing against the seat ring's soft seat surface can achieve bubble-tight shutoff if the plug and seat-ring surfaces are

Figure 4.5 Exploded view of soft-seat design. (*Courtesy of Valtek International*)

concentric. Some manufacturers also insert the elastomer in the plug, which achieves the same effect (Fig. 3.22, Chap. 3).

4.2.3 Globe-Control-Valve Operation

The most common globe valve uses a T-style body, which allows the valve to be installed in a straight pipe with the top-works or actuator perpendicular to the line and will be used to explain the basic operation of a globe valve. Flow enters through the inlet port to the center of the valve where the trim is located. At this point, the flow must make a 90° turn to flow through the seat, followed by another 90° turn before exiting the valve through the outlet port.

The flow direction of globe valves is defined by the manufacturer and in many applications is critical to the valve's operation. With standard single-seated globe valves using inlet and outlet ports, the two choices are flow-under-the-plug and flow-over-the-plug. With manually operated globe valves, flow is almost always under the plug. The plug closing against the flow provides constant resistance, but not enough to be insurmountable, and is relatively easy to close as long as the fluid pressure and flow rate are low to moderate. Flow-under-the-plug provides for easy opening, as the fluid pushes against the bottom of the plug. However, flow direction is an important consideration with control valves equipped with diaphragm actuators, which are not capable of high thrusts. If the flow is over the plug and the process involves high pressures, the diaphragm actuator is not usually stiff enough to prevent the plug from slamming into the seat ring when throttling is close to the seat. Also, the actuator must pull the plug out of the seat against the full upstream pressure, which may be difficult in a high-pressure application. Therefore, lower-thrust actuators demand flow-under-the-plug, allowing the full thrust to close against the upward force of the fluid pressure. Another situation in which flow-under-the-plug is an issue is with fail-open applications, where the service requires the valve to remain open during a signal or power failure. Even if an actuator with a fail-safe spring is rendered inoperable during a fire, the flow-under-the-plug design will ensure continued flow as the flow pushes the plug away from the seat.

Inversely, flow-over-the-plug is important in fail-closed situations, where the service requires the valve to shut during a loss of signal or power. If the actuator fails and the fail-safe spring also fails, the flow acts on the top of the plug to push it into the seat. Obviously, with flow-over-the-plug situations, throttling close to the seat presents a problem if the actuator does not have sufficient stiffness (the ability to hold a position despite process forces). The actuator must have enough thrust to pull the plug out of the seat against the fluid's upstream pressure—which increases to its maximum value in a nonflow state. As the issues of stiffness and thrust are considered, in a majority of situations where the flow must be over the plug, piston cylinder actuators are preferred over diaphragm actuators.

As alluded to earlier in Sec. 4.2.2, the globe-valve trim can be modified to allow for equal-percentage, linear, or quick-open flow characteristics. As explained in detail in Sec. 2.2, flow characteristics determine the expected flow rate (expressed in flow coefficient or C_v) at a certain valve position. Therefore, with a particular flow characteristic, the user can determine the flow rate at a given instrument signal. As the flow reaches the trim, and if the trim is in a throttling position, the flow is directed to a restriction. This restriction may be created by the exposed portion of a hole in a cage, which is based upon the linear position of the plug. It may also be created by the portion of the V-shaped slot of a V-port plug that is exposed above the seat ring. Also, the restriction may be created by the amount of the seat that is open to the flow when the area of a contoured plug is filling a portion of the seat area. When a pressure-drop situation occurs (the downstream pressure is lower than the upstream pressure), the flow moves from the inlet through the seat to the outlet. As the flow moves through the seat, line pressure decreases as velocity increases. After the fluid enters the lower portion of the globe body, the area expands, the pressure recovers to a certain extent, the velocity decreases, and flow continues through the outlet port and downstream from the valve. As the flow enters the trim area of the valve, an important consideration is the gallery area of the body surrounding the trim. In ideal situations the flow should freely circulate around the trim, allowing flow to enter the trim from every possible direction. With cages and retainers, flow should enter equally from every hole to provide equal forces to act on the plug head. If the gallery is narrow in any one area (for example, in the back side of the cage), velocities can increase, causing noise, erosion, or downstream turbulence. In addition, unequal forces acting on the plug head can cause slight flexing of the plug head if it is not supported by a cage.

When the globe control valve closes, the axial force from the actuator is transferred to the plug and its seating surface makes contact against the slightly mismatched angle of the seat ring. As full contact is made, the valve is closed, allowing minimal or no flow to pass through the trim according to the ANSI leakage classification. If the axial force is applied in the opposite direction, the plug lifts and, in the full-open position, the entire seating area is open to the flow as well as the holes of the cage or retainer.

Because the process flow is under pressure and the environment outside the valve is at atmospheric pressure, the flow seeks to escape through the gaps in the valve. This leakage is prevented by the static seal of the gaskets in the end connections (if flanges or RTJ end connections are used) and the bonnet gaskets. Flow seeking to escape through the sliding stem of the plug is prevented by the packing's dynamic seal in the bonnet's packing box. In closed positions, flow may escape through the seat but is prevented by the static seal between the seat ring and the body.

4.2.4 Globe-Control-Valve Trim Variations

With special service requirements, globe control valves can use a number of specialized trims for unique flow requirements. Some applications require extremely low flow coefficients, with C_vs anywhere down to 0.000001. Because of these extremely low flows, these designs are found only in smaller valve sizes (less than 2 in or DN 50). The

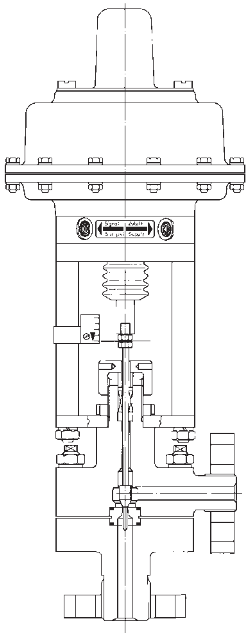


Figure 4.6 Low-flow control valve with needle trim. (*Courtesy of Kammer Valves*)

plug head is shaped very narrowly, earning the designation *needle-valve trim* because of its needlelike appearance (Fig. 4.6). Because even the smallest variations in diameter can have a wide impact on the overall flow coefficient and flow rate, needle plugs are machined using special micromachining procedures (using technologies developed by the watchmaking industry). These precise trims require the flow characteristic to be machined into the plug head contour. Needle-valve trim requires a very precise method of adjustment of the distance between the seat and plug-seating surfaces. A very fine thread (twice the magnitude of a normal plug thread) is normally required, allowing a very minute amount of linear adjustment per turn.

Pressure-balanced trim is defined as a special trim modification that allows the upstream pressure to act on both sides of the plug head, significantly reducing the off-balance forces and operator thrust needed to close the valve. It is sometimes used to replace normal trim arrangements when the valve must close against a large seat diameter coupled with high-pressure process forces or high-pressure drops. Because the regulating element must overcome these forces, exceptional actuator force from a high-thrust actuator or a larger lower-thrust actuator must be used to close the valve. In other applications, a standard valve may need a smaller actuator size to fit into a tight space. In this case, pressure-balanced trim reduces the valve's need for a larger standard actuator by reducing the off-balanced area of the trim. Pressure-balanced trim is common with valves in larger sizes [size 12 in (DN 300) and higher] in which a large amount of flow is passing through a large seat and where the cost of a larger actuator would be greater than the cost of the pressure-balanced trim.

Pressure-balanced trim requires a special plug and *sleeve,* which is similar in many respects to a cage. These parts allow the upstream pressure to act on both sides of the plug, as shown in Fig. 4.7. The sleeve's inside diameter is slightly larger than the inside diameter of the seat ring. The plug requires a smaller plug stem to minimize the off-balance area, and is equipped with metal piston rings, O-rings, or polymer rings that, when installed inside the sleeve, create a pressure chamber above the plug. One or two holes are machined through the plug head, allowing the fluid pressure to act on both sides of the plug. In effect, this results in a net force equal to the pressure multiplied by the off-balance area.

With high inlet pressures and a large seat area, a high actuator force is required to close the valve. With standard trim (unbalanced plug),

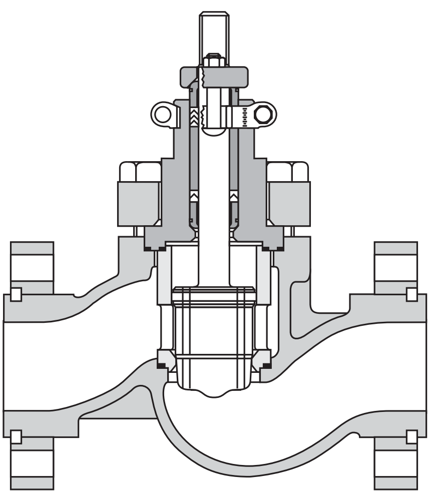


Figure 4.7 Globe-body subassembly with pressure-balanced trim. (*Courtesy of Valtek International*)

the force necessary to close the valve is the total *off-balance area*, which is written as

$$F_{OBA} = P_1(A_S - A_{stem}) - P_2(A_S)$$

where F_{OBA} = actuator force required to overcome the off-balance area
P_1 = upstream pressure
P_2 = downstream pressure
A_S = area of the inside diameter of the seat
A_{stem} = area of the outside diameter of the plug stem

However, with pressure-balanced trim and its counter-balanced design, the off-balance area is far less, which requires less actuator force, as written in the following equation:

$$F_{OBA} = P_1(A_{sleeve} - A_{stem}) - P_2(A_S)$$

where A_{sleeve} = area of the inside diameter of the sleeve

With pressure-balanced trim, the larger the off-balance area (slight as it may be), the greater the shutoff. For example, in smaller globe-valve sizes (0.5 through 3 in or DM 12 through DN 80), the off-balance area is slight and an ANSI Class II shutoff is usually the standard—ANSI Class II calls for a maximum leakage rate of 0.5 percent of rated valve capacity. On the other hand, for sizes of 4 in (DM 100) and larger, the off-balance area of the trim increases and ANSI Class III shutoff is possible—ANSI Class III calls for a maximum leakage rate of 0.1 percent of rated valve capacity.

With standard unbalanced trim, the direction of the flow assists with the motion of failure (flow-over-the-plug is used for fail-closed and flow-under-the-plug is used for fail-open cases). With pressure-balanced trim, however, the opposite occurs. Flow direction is under the plug for fail-closed situations and over the plug for fail-open situations. The actuator force required to fail-open or fail-closed is related to the off-balance area. Hence, for flow-over-the-plug and fail-closed situations, this off-balance area is equal to the sleeve area minus the seat-ring area. The spring must be able to overcome this off-balance area, which can be written as

$$F_{open} = P_1(A_{sleeve} - A_{seat})$$

where F_{open} = spring force required to fail-open

With flow-over-the-plug and fail-closed applications, the off-balance area is equal to the sleeve area minus the plug stem area, as indicated in the following equation:

$$F_{\text{closed}} = P_1(A_{\text{sleeve}} - A_{\text{stem}} - A_{\text{seat}})$$

where F_{closed} = spring force required to fail-closed

In standard services, the major advantage of using pressure-balanced trim is that smaller or less powerful actuators can be used. Another advantage is that high-pressure drops or higher process pressures can be handled without resorting to expensive, large nonstandard actuators. In some instances, use of pressure-balanced trim is the only method by which some applications can be handled because an actuator with extremely high thrust may not be available for the required valve size or may not fit in the available space.

On the other hand, pressure-balanced trim has four major disadvantages: First, because pressure-balance trim only works with a sliding seal between the plug and the sleeve, the fluid must be relatively clean and free from particulates; otherwise, the seals can be damaged and cause leakage or galling between the plug and sleeve. Second, because of the balanced nature of the plug, coupled with the lower thrust of a smaller actuator, leakage rates through the seat are not as good as with unbalanced trim—ANSI Class II is normal. Third, pressure-balanced trim is more costly initially than standard trim, although the use of a smaller actuator may offset that cost or even make the overall cost more attractive. And fourth, because of the seal within the process flow, the trim may require a shorter servicing cycle, especially if the process has entrained particulates.

Double-ported trim is a special trim design used to fill the same purpose as pressure-balanced trim: to reduce the effect of the process forces on the plug, thereby lowering the thrust requirement and allowing the use of smaller actuators. Flow is directed by the inlet port to the body gallery and the trim, which features two seats and a single plug that features two plug heads, one above the other (Fig. 4.8). In air-to-open (fail-closed) applications, the plug–seat combination at the top of the gallery is a flow-under-the-plug design, while the plug–seat combination at the bottom is a flow-over-the-plug design. In air-to-close (fail-open) applications, the opposite design is used. The plug–seat arrangement at the top is flow over the plug and that at the bottom is flow-under-the-plug.

Upon opening, the net forces working on these two seats nearly cancel each other out. The fluid pressure is pushing the upper plug head out of the seat, while the lower plug head is pulling out against the fluid pressure. Upon closing the opposite occurs. The upper plug head

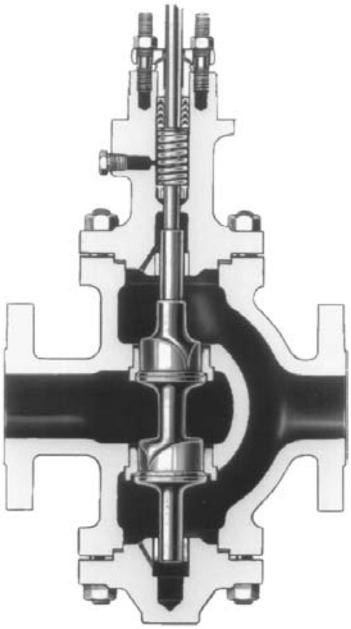


Figure 4.8 Double-ported globe-body assembly. (*Courtesy of Fisher Controls International, Inc.*)

pushes against the flow, while the lower plug head is assisted by the flow. Although in principle double-seated valves are close to pressure-balanced valves, in reality they are somewhere between pressure balanced and unbalanced. This is because the fluid is acting against the plug contour with one seat and the top of a plug head (usually a flat surface) with the other seat, creating a dynamic imbalance. With double-seated valves, flow characteristics are nearly always determined by the contour of the plug head. Guiding is accomplished with upper and lower guides. The upper guide is placed above the upper seat, while the lower guide is located in the lower body region with a lower body cap for access and assembly. This arrangement also allows for easy reversal of the stroke direction (air-to-open to air-to-close, or vice versa). The body can be inverted, with the bonnet and the lower body cap retaining their previous positions.

Double-ported trim can also be used with three-way valves for diverting, combining, or dividing flows. In the case of diverting flow, the plugs are offset, meaning that one of the two plug heads is always seated, while the other is in the full-open position. As the valve moves from one end of the stroke to the other, the opposite occurs: the previously closed plug head moves to the full-open position and the previously open plug head moves to full-closed. To divide flow between the two outlets, this same arrangement can be used, except that the stroke remains in the middle as if throttling, allowing both seats to be open to some extent and flow to move down both outlets. For combining flows, the flow direction of the valve is reversed, allowing for two inlet ports and a single outlet port. Using a double-seated valve for three-way service means that a lower guide surface as part of the body is not possible, since that area is used as a port. In these cases, the plug head is designed to guide in the seats, using notches in the plug head to achieve flow control.

Double-ported trim does have drawbacks: First, the alignment of the plug and the seat is critical in T-line valve styles (one inlet and one outlet), and if one plug head is out of alignment, one may fully seat, while the other will be slightly off the seat, allowing leakage through that seat. Because of the extreme difficulty of aligning the two seats to provide equal shutoff, allowable leakage is 0.5 percent of the rated flow of the valve. Thermal expansions can also cause the distance between the seats to widen, leading to increased leakage. The second drawback is that the design requires screwed-in seat rings, which are prone to corrosion and must be lapped to ensure tight shutoff.

Another trim variation is *sanitary trim,* which is required for those valves used in the food and beverage industry. Such valves require stainless-steel construction of all wetted parts and are specified with angle-style bodies, which allow the downstream port to be 90° from the inlet port. In other words, the flow is directed straight down from the seat ring. With sanitary applications, pockets of fluid cannot be allowed to stand or pool; otherwise contamination or bacterial growth can result. When the system is flushed by water or steam, the self-draining allows for the system to quickly dry and be readied for another type of process fluid or for the system to remain dormant.

Sanitary-trim design (Fig. 4.9) allows the valve to self-drain when the system is depressurized or if the valve is closed, allowing the outlet side to drain. To avoid pockets of trapped fluid, sanitary trim has very few flat areas and no walled pockets. In some designs, the seating surface is machined into the body to avoid a gap between a seat ring and the body. The plug head is tapered on its top side until it reaches the plug stem. Because sanitary services must have tight shutoff, the plug head is fitted with an elastomeric insert to provide bubble-tightness. Because of possible pooling areas, pressure-balanced trim is never an option with sanitary services. Most sanitary valves also

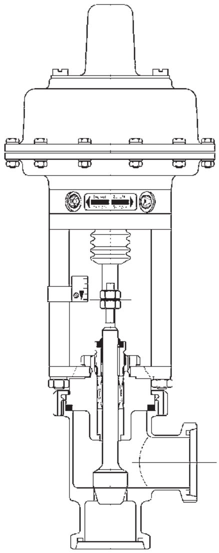


Figure 4.9 Sanitary-trim control valve. (*Courtesy of Kammer Valves*)

require stainless-steel actuators to avoid any sort of oxidation in the clean environment.

4.2.5 Globe-Control-Valve Body Variations

Globe valves are considered to be one of the most versatile valve designs because the body can be varied in numerous ways to allow for different piping configurations or functions. The most common single-seated globe body style is the flow-through design (or sometimes called the *T-style body*), which is shown in Fig. 4.10. Basically, this body style allows the valve to be installed in a straight piping configuration, with the rising-stem action perpendicular to the centerline of the piping. Unlike most quarter-turn valves or gate valves where the flow moves straight through the body relatively unimpeded, the flow-through design brings the flow through two right-angle turns, allowing for a significant pressure drop, which is essential for some applications. As the flow moves through the inlet port, the flow passage shifts up (or down, depending on the flow direction) approximately 30° until the flow reaches the gallery of the body, bringing the flow above (or below) the seat, which is usually on the piping centerline. At that

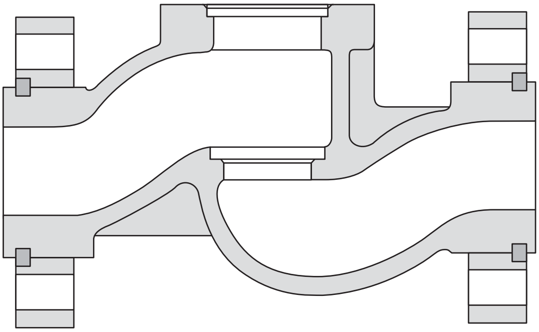


Figure 4.10 Globe body with top-entry to the trim and separable flanges. (*Courtesy of Valtek International*)

point, in flow-over-the-plug situations, the flow enters the gallery area that surrounds the trim. The flow then turns 120° to flow through the seat. At this point, the flow is perpendicular to the piping centerline. As the flow exits the seat, it turns 120° again by the flow passage, shifting up (or down) until the flow meets the outlet port and moves out into the downstream piping.

Globe flow-through bodies can be modified with a elongated body chamber above the regulating element (Fig. 4.11) for cryogenic applications. The upper chamber of this body style allows for a small amount of liquefied gas to vaporize between the process and the packing, acting as a vapor barrier—the pressure from the vaporization actually prevents any further liquid from entering the chamber.

An alternative single-seated body style, somewhat related to the flow-through style, is the *angle-body* style (Fig. 4.12). Instead of the two ports being in-line with the straight piping configuration, one port is turned 90° from the other port (or at a right angle) to match piping that requires such a turn. The port that is perpendicular to the rising stem is called the *side port,* and the port that is in-line with the rising stem is called the *bottom port*. Valves with an angle-style body are used in a number of applications. First, angle valves are sometimes used in cavitating services where the imploding bubbles are channeled directly into the center of the downstream piping. Depending upon the severity of the cavitation, the bubbles may not directly impact a metal wall (such is the case with the bottom of the globe straight-through body). Rather, they implode harmlessly in the middle of the pipe. If the control valve is part of a piping system that discharges into a tank, an angle valve can be used so that any cavitating liquid can flow into the large vessel, where it

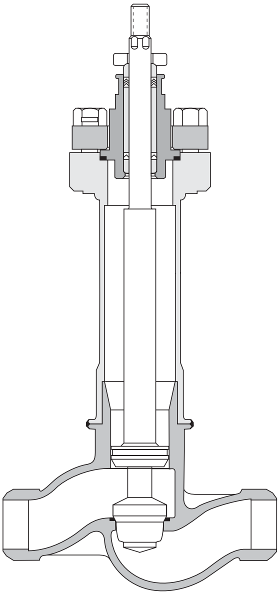


Figure 4.11 Elongated globe body for cryogenic service. (*Courtesy of Valtek International*)

will not affect any nearby metal surfaces. An angle valve also allows the use of a *Venturi seat ring* (Fig. 4.13), which is an extended seat ring that can protect the sides of the bottom port and downstream piping from adverse process effects, such as abrasion or erosion. Also, because of the right-angle turn in the body design, angle valves can be installed in services that have a natural upward flow, such as in crude oil or natural gas applications or boiler services. A special kind of

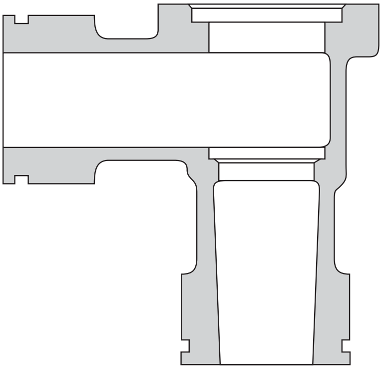


Figure 4.12 Angle body with top-entry to the trim and separable flange hubs. (*Courtesy of Valtek International*)

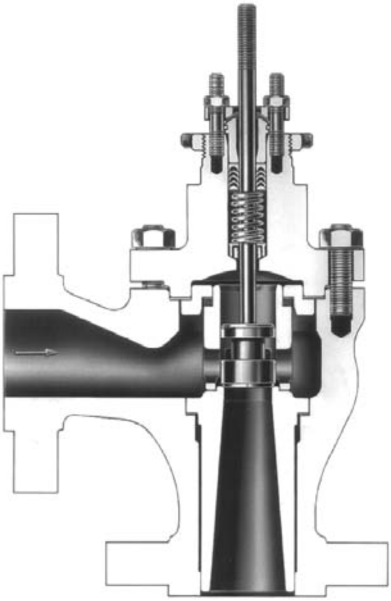


Figure 4.13 Venturi seat ring design. (*Courtesy of Fisher Controls International, Inc.*)

angle valve, called a *choke valve,* is used for most wellhead applications. Many mining applications involve gas services that have particulate matter such as sand or dirt, which have a tendency to erode—a process similar to sandblasting. Modified-sweep-style angle valves (Fig. 4.14), with trim made from ceramic for durability, allow the particulates to be channeled down a pipe without directly impinging on any body walls. Also, angle valves allow for easy draining, since no pockets exist that allow the fluid to pool.

One disadvantage of using an angle valve is that turbulent flow created by the regulating element can channel the turbulence directly into the downstream piping, creating more vibration and noise than would be created using a flow-through body. The downstream side of the flow-through body is quite stiff, handling some of the flow's energy conversion in an unyielding vessel before the flow proceeds into downstream piping. Angle valves also have a higher pressure recovery than other types of globe valves, resulting in a lower σ value (the cavitation index, Sec. 9.2), which means an increased chance of cavitation.

A variation of the globe straight-through style is the *expanded-outlet* style, which is basically a straight-through design except that the end

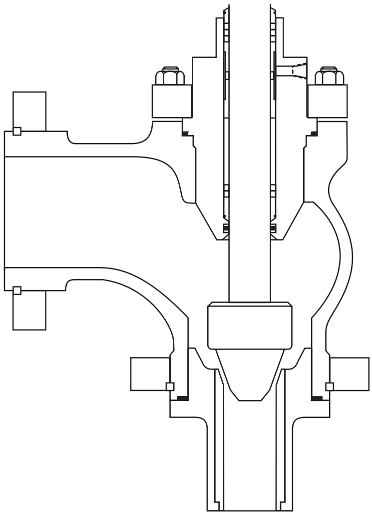


Figure 4.14 Sweep-angle body subassembly. (*Courtesy of Valtek International*)

connections are a larger pipe size than the trim is designed for. For example, a 4 × 2-in expanded outlet valve would have 4-in end connections (for mounting to a 4-in pipe), but would have the full-area trim for a 2-in valve. Expanded-outlet valves are used to lower the cost of welding or installing piping increasers to the valve body. The expanded-outlet body's face-to-face is also shorter than a normal globe straight-through valve with increasers, which may be important in piping systems with limited space. This style is also a cost-saving measure when a larger valve size is required with reduced trim. The smaller trim size may also act as a reduced trim—although technically it is considered a full-area trim for the smaller valve size.

Another variation of the globe straight-through style is the *offset body* style, which provides for straight-through flow except that the inlet and outlet ports are parallel and not in-line with each other (Fig. 4.15). The seat is placed in a center position between the two piping centerlines. Offset valves are used for unique piping configurations because the flow passages do not shift up or down to bring the flow above and below the seat. Unlike the T-style globe body, less pressure drop occurs with the offset body.

The *split-body* style involves a body made of two separate parts: the upper body half and the lower body half (Fig. 4.16). These two body parts connect at the center of the valve body with the seat ring sand-

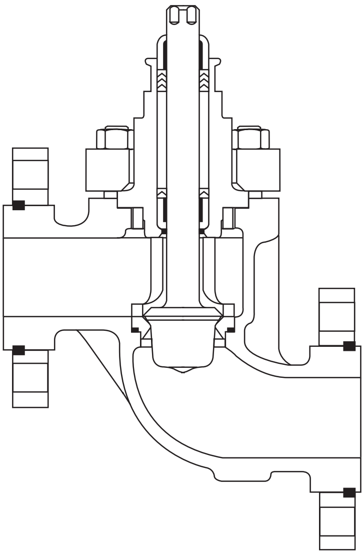


Figure 4.15 Offset globe-body subassembly. (*Courtesy of Valtek International*)

wiched between the two body parts. Body bolting is used to secure the two body halves together. Two gaskets are used on both sides of the seat ring to ensure pressure retention. The bonnet can be integrally connected to the upper body half. This is preferred, since a good design should minimize potential leak paths—having a separate bonnet would add another potential leak path. Using a split-body design offers several advantages. First, the seat ring is retained in place without a seat retainer or cage to center or hold the seat ring in place, in effect, combining the advantages of both retained and threaded seat rings. If the application is such that the plug and seat ring must be inspected or replaced often, such as in chemical services that are highly corrosive, the simplicity of construction and disassembly permits frequent inspections. The split-body design also reduces the trim by one part, which may be a factor if the valve body is made from an exotic alloy. It also avoids any flow difficulties associated with a cage or retainer, such as galling or noise. Second, the seat ring can be removed with minimal disassembly, although the lower body half would need to be removed entirely from the line. And third, in some designs, the two body halves can be disassembled and turned 90° in either direction to provide a right-angle valve, perpendicular to the rising stem, as opposed to a true angle valve where the lower port is

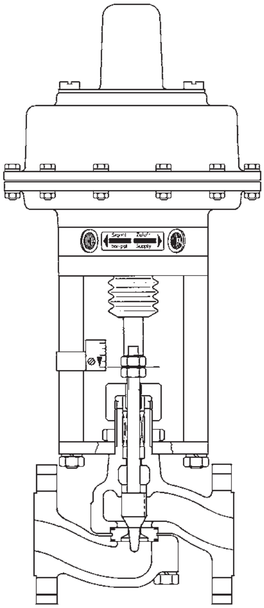


Figure 4.16 Split-body control valve. (*Courtesy of Kammer Valves*)

in-line with the rising stem. With a split body, the actuator or manual handwheel could remain upright. With a true angle valve, the actuator would be on its side. The split-body valve has some limitations. For example, it is usually only specified with flanged end connections. It cannot be used in steam or other high-temperature services where buttweld or socketweld end connections are required for welding the valve into the line, since the body could not be disassembled to access the seat ring. If process leakage occurs at the body connection, the body bolting is located where fluid could cause corrosion, making disassembly difficult.

Another unique body style is the *Y-body* style, which is a body where the rising stem is inclined 45° (or sometimes 60°) from the axis of the inlet and outlet ports, which are in-line with the piping (Fig. 4.17). Y-body valves are the best type of globe control valve for passing the largest C_v possible with minimal pressure drop—short of using a globe body with an integral seat and an oversized plug. Also, because the body avoids the right-angle turns and the plug pulls nearly out of the flow stream, less turbulence is generated through the body, which may reduce noise. Y-body valves are also commonly applied in piping sys-

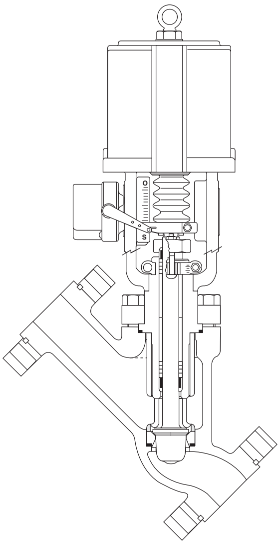


Figure 4.17 Y-body control valve. (*Courtesy of Valtek International*)

tems with piping set at 45°, allowing the valve body to be in-line with the piping, while the top-works is vertical to the ground. This allows easier maintenance and better operation. Because the body, when placed at a 45° angle, has little if no pockets for a fluid pool, the Y body is often applied in self-draining applications.

A *three-way body* style has three ports: two ports in-line with the piping centerline and one port in-line with the rising stem. This design uses a plug head featuring an upper and lower seating surface and two matching seats (Fig. 4.18). Depending on the position of the plug or the orientation of the piping, the process flow can be diverting, splitting, or mixing. With diverting flow, the flow enters a side port and, if the plug is fully extended into the lower seat, the flow is diverted out the opposite side port. If the plug is fully retracted into the upper seat, the flow is diverted through the bottom port. When the plug remains in a throttling position between the two seats, flow is diverted to both the side and bottom ports for when the flow needs to be split. Combining two separate flows can be accomplished with the same body style, except that the opposite side port and the bottom port both receive the upstream process flow. When the plug is placed in midposition, both processes flow together and combine before exiting the side port.

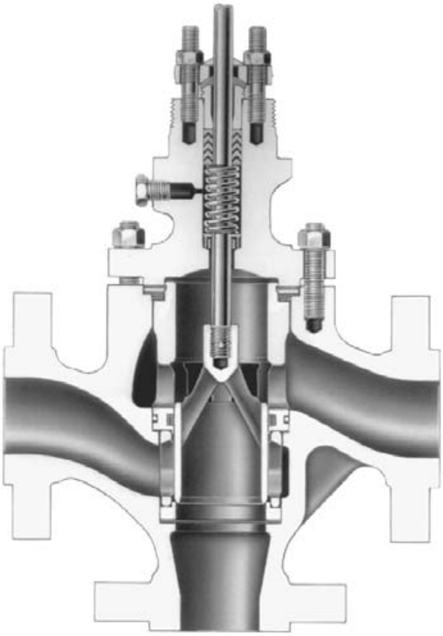


Figure 4.18 Three-way body subassembly with integral three-port body and pressure-balanced trim. (*Courtesy of Fisher Controls International, Inc.*)

Another optional design with three-way valves involves the use of a *three-way adapter* with a conventional globe straight-through body (Fig. 4.19). The adapter consists of an upper-body extension that is mounted above the body where the bonnet normally sits. An upper seat ring is sandwiched between the body and the adapter. The adapter is equipped with a side port, which can be mounted in any one of four quadrants if the end connection can be used without interfering with another port. One exception is flanged end connections, which can only be possible at right angles since the flanges would interfere with the in-line piping or other flanged connections. The bonnet sits above the adapter and a special three-way, dual-seating plug is used to divert, mix, or separate process flow. The obvious advantage to this type of design is that a valve can be converted to three-way service without a new body—only a new adapter, upper seat ring, and plug are required. The disadvantage is that an additional possible leak path is added to the body subassembly.

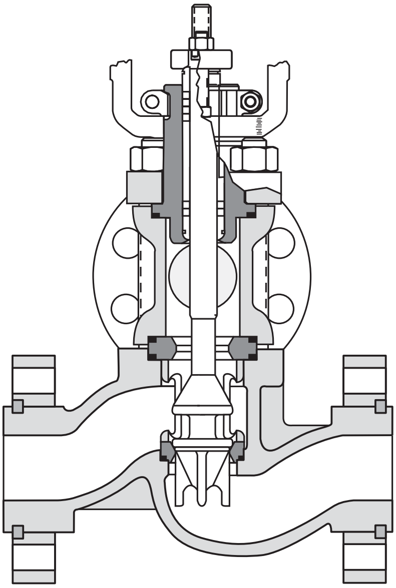


Figure 4.19 Three-way body subassembly with three-way adapter. (*Courtesy of Valtek International*)

4.3 Butterfly Control Valves

4.3.1 Introduction to Butterfly Control Valves

Although the butterfly valve has been in existence since the 1930s, it was used mainly as an on–off block valve until the past two decades, when it began to be used for throttling services. In the late 1970s, design advancements were made to the butterfly valve that not only made it more applicable for throttling service, but also made it preferred over globe valves in some applications. Such butterfly control valves are differentiated from their on–off block cousins by the name *high-performance butterfly valves.* In simple terms, the high-performance butterfly control valve is a quarter-turn (0° to 90°) rotary-motion valve that uses a rotating round disk as a regulating element. Typically, butterfly control valves are available in sizes 2 through 8 in (DN 50 through DN 200) from ANSI Classes 150 to 600 (PN 16 through PN 100); 10 and 12 in (DN 250 and DN 300) in ANSI Classes 150 and 300 (PN 16 and PN 40); and 14 through 36 in (DN 350–900) in ANSI Class 150 (PN 16).

When fully open, the disk actually extends into the pipe itself, which makes butterfly valves distinct from other valve designs. Butterfly-valve bodies have very narrow face-to-face dimensions compared to other types of valves, allowing the body to be installed between two pipe flanges without any special end connections. This type of arrangement is called a *through-bolt connection* and is only permissible with certain bolt lengths. If the bolt length is too long, the bolting may be subject to thermal expansion of the process or during an external fire, causing leakage.

Initially, butterfly control valves were designed as automatic on–off block valves. However, with recent improvements to rotary-valve actuators and body subassemblies, they can now be used in throttling services with the addition of an actuator or an actuation system. As detailed in Sec. 3.4, the family of butterfly valves is classified into two groups. *Concentric butterfly valves* are normally used in on–off block applications, with a simple disk in-line with the center of the valve body. Generally, concentric valves are made from cast iron or another inexpensive metal and are lined with rubber or polymer. Because of their lower performance, they are normally equipped with manual operators. In some applications, the manual operators are replaced with an actuation system for throttling service. In most applications, however, simple concentric butterfly valves are used strictly for on–off service. Even when used in throttling applications, they do not lend themselves as well to automatic control as other butterfly designs specifically designed for throttling control. This is because the initial development was for blocking service. Concentric butterfly valves have poor rangeability, while throttling-specific butterfly valves have design modifications to allow for better flow control through the entire stroke.

Eccentric butterfly valves are valves designed specifically for high-performance throttling services, using a disk that is offset from the center of the valve body. The majority of butterfly valves used as control valves feature the eccentric design. For the most part, eccentric butterfly valves are specified in common valve materials, such as carbon, stainless, or alloy steels. When equipped with actuators and positioners, they are much more precise than concentric butterfly valves that have been automated.

Compared to other types of throttling valves, eccentric butterfly valves are one of the fastest growing types of control valves today for a number of reasons. Because of the increased dead band associated with the mechanical conversion of linear motion to rotary motion, globe valves are more precise in high-pressure-drop applications than butterfly valves. However, the control provided by today's butterfly valves is more than adequate for many low-pressure-drop applications and other standard services.

When compared to globe control valves, butterfly control valves are much smaller and lighter in weight because the butterfly valve's body subassembly weight can be anywhere from 40 to 80 percent of a comparable valve and less than half the mass of the globe body subassembly. In addition, smaller actuators can often be used with butterfly valves since the weight of the regulating element is not a critical factor in factoring the necessary actuator force. The difference in regulating-element weight between butterfly and globe control valves becomes much more evident as sizes become larger, as shown in Table 4.1. This means that butterfly valves are preferred in applications where limited space or weight is a consideration.

Table 4.1 Weight Comparisons between Globe and Butterfly Valves*

Valve Size	**Flanged Globe Valve** *Standard Valve with Actuator, ANSI Class 150*	**Flangeless Butterfly Valve** *Standard Valve with Actuator, ANSI Class 150*	Percent Reduction
2-inch DN 50	75 pounds 34 kilograms	40 pounds 18 kilograms	47%
3-inch DN 80	160 pounds 73 kilograms	46 pounds 21 kilograms	71%
4-inch DN 100	240 pounds 109 kilograms	52 pounds 24 kilograms	78%
6-inch DN 150	360 pounds 163 kilograms	96 pounds 44 kilograms	73%
8-inch DN 200	590 pounds 268 kilograms	110 pounds 50 kilograms	81%
10-inch DN 250	1050 pounds 477 kilograms	267 pounds 121 kilograms	75%

**Data courtesy of Valtek International.*

Another major benefit of using a butterfly control valve is that, size for size, it has a larger flow coefficient, producing a greater flow than comparable globe valves. Because the shaft of the butterfly valve moves in a rotary motion instead of a linear motion, the frictional forces are far less than a linear-motion valve, requiring less thrust and permitting a smaller actuator. A butterfly valve has a naturally high pressure-recovery factor (Sec. 7.2.9). This factor is used to predict the pressure recovery occurring between the vena contracta and the outlet of the valve. The butterfly valve's ability to recover from the pressure drop is influenced by the geometry of the wafer-style body, the maximum flow capacity of the valve, and the service's ability to cavitate or choke. Overall, because of the high-pressure recovery, a butterfly valve works exceptionally well with low-pressure-drop applications.

The largest drawback to using a butterfly valve is that its service is usually limited to low-pressure drops because of its high pressure recovery. Although flashing is normally not associated with a butterfly-valve design, cavitation and choked flow occur easily with a butterfly valve installed in an application with a high-pressure drop. Although some special anticavitation devices have been engineered to deal with cavitation, users prefer to deal with cavitation in a globe valve because of its design versatility in allowing the inclusion of an anticavitation device. Another disadvantage is that a butterfly valve has a poor-to-fair rangeability of 20 to 1 because of the difficulty the disk has in holding a position close to the seat. The process pressure applied to the butterfly disk creates a significant side load, which can only be remedied by using a larger-diameter shaft. Another drawback to the butterfly control valve is the increased hysteresis and dead band associated with the mechanical transfer of linear action from the actuator to the rotary motion needed for the regulating element. Valve manufacturers have utilized splined shafts or other secure linkages to minimize this problem, although a globe valve avoids this problem altogether with its direct linear motion. The sizes of butterfly valves are also limited to 2 in (DN 50) and larger because of the limitations of the rotary regulating element. Because of the side loads applied to the disk, the maximum size that a high-performance butterfly can reach is 36 in (DN 900).

4.3.2 Butterfly-Control-Valve Design

The butterfly body typically involves one of two styles. The *wafer body* (sometimes called the *flangeless body*) is a flat body that has a minimal face-to-face, which is equal to double the required wall thickness plus the width of the packing box (Fig. 4.20). Within this dimension, the disk in the closed position and the seat must fit within the flow portion of the body. Because the wafer-style body has a minimal face-to-face, straight-through bolting using the two flanged piping connections is possible without fear of thermal expansion causing leakage. Wafer-style bodies are more commonly applied in the smaller sizes, 12 in (DN 300) and less. The other body style is the *flanged body,* which is used with larger butterfly valves [14 in (DN 350) and larger] that require a longer face-to-face (Fig. 4.21) when a higher degree of thermal expansion is expected or when the regulating element cannot fit within the wafer-style body. The flanged style has integral flanges on the body that match the standard piping flanges.

As shown in Fig. 4.22, another body style is the *lug-style body,* in which the butterfly body has one integral flange that has an identical hole pattern to the piping flanges. Each hole is tapped from each direction, meeting in the center of the hole. This arrangement allows the body to be placed between two flanges. Studs are then inserted through the piping flange and threaded into the valve's integral flange. After the stud is securely threaded into the integral body flange, a nut is threaded to the stud to secure the piping flange to the body. Lug bodies are used in

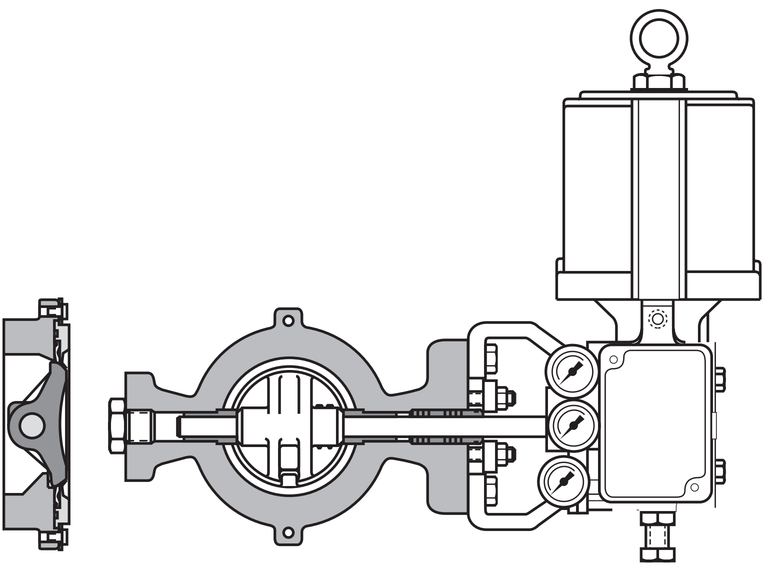


Figure 4.20 Flangeless butterfly control valve (wafer style). (*Courtesy of Valtek International*)

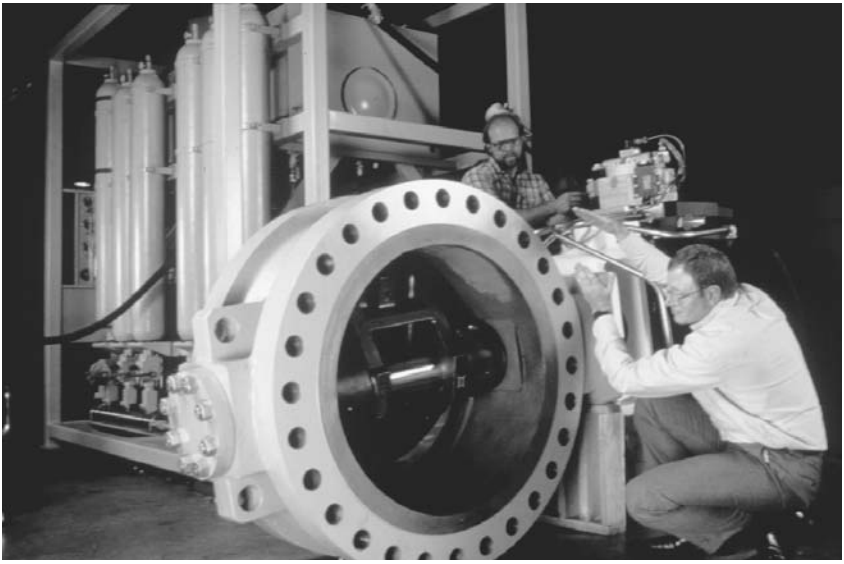


Figure 4.21 Flanged butterfly control valve. (*Courtesy of Valtek International*)

applications in which the risks of straight-through bolting cannot be taken—such as with thermal expansion—in smaller valve sizes that do not permit the use of two integral flanges.

The faces of the butterfly-valve body are often serrated to fix and secure the location of the flange gaskets between the pipeline and the valve. The inside diameter of the butterfly valve is close in size to the inside diameter of the pipe, which permits higher flow rates as well as straight-through flow. Perpendicular to the flow area of the valve is the shaft bore, which is drilled from both sides. Drilling from one side through the entire body is extremely difficult without the wandering associated with using a long drill bit.

The regulating element of the butterfly valve is the called the *disk,* which rotates into the *seat.* The disk is described as a round, flattened element that is attached (usually by tapered pins) to the rotating shaft. As the shaft rotates, the disk is closed at the 0° position and wide open at the 90° position. As explained earlier in Chap. 3, if the shaft is attached to the disk at the exact centerline of the disk, it is known as a *concentric disk.* When the disk is offset both vertically and horizontally (refer to Fig. 3.14), it is referred to as an *eccentric cammed disk.*

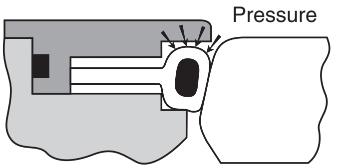
The disk is designed to minimize interruption of the flow as the process fluid moves through the valve. Slight angles and rounded sur-

Figure 4.22 Lug-style butterfly control valve. (*Courtesy of Automax, Inc. and The Duriron Company, Valve Division*)

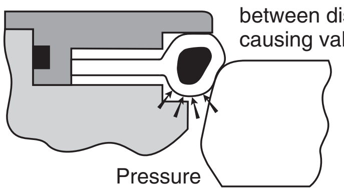
faces are characteristic of a common disk design. When closed, the flat side (facing the seat) is called the *face,* while the opposite side is called the *back side.* The face is often designed slightly concave so that maximum flow can be achieved in the open-flow position. On the backside, sometimes a *disk-stop* is provided that matches up with a similar stop inside the body's flow area. This stop prevents the valve from overstroking. Overstroking can cause the disk to drive through the seat, irreparably damaging the seat. The circumference of the seat wraps around the entire inside diameter of the body's flow area and is installed at one end of the body. If a polymer is used for the seat, it is called a *soft seat.* When a flexible metal is used as the seating surface, it is called a *metal seat.* The seat is installed in the end of the body and is held in place by a *seat retainer,* using screws or a snap-fit to keep the seat and retainer in place. After the seat and seat retainer are in place, the face of the retainer usually lines up with the face of the body. In some designs, the seat–retainer design protrudes slightly from the body face, allowing some gasket compression when the body is installed in the line.

The disk is attached to the shaft with the use of one or more tapered pins. The shaft is supported by close-fitting *guides* (sometimes called *bearings*) on both sides of the disk, which are installed in the shaft bore to prevent lateral movement of the shaft and disk that can cause misalignment. Thrust washers may also be placed on both sides of the disk, between the disk and the body, to keep the disk firmly centered with the seat.

A number of different resilient seat designs exist for eccentric butterfly control valves, which are designed to handle higher pressures and temperatures—most of which operate by similar principles. One of the most common soft-seat designs is the seat that utilizes the *Poisson effect,* which states that if an O-ring or an elastomer is placed in a seating situation with a greater pressure on one side, the soft material will deform away from the pressure. In other words, deformation takes place when the pressure pushes the softer material against the surfaces to be seated (Fig. 4.23). With the Poisson effect, the greater the upstream pressure compared to the downstream pressure, the greater the seal. Because of their flexibility, O-rings encased in a polymer work exceptionally well with the Poisson effect. Related to the Poisson effect is the *jam-lever* or *toggle effect,* which uses a hinged elastomer that is designed to be thinner in the midsection than at the outside or inside diameter. This design permits the outside diameter of seat to flex and seal against metal surfaces when process pressure is applied (Fig. 4.24). A third resilient seat design uses the *mechanical preload effect,* which calls for the inside diameter of the seat to slightly interfere with the outside diameter of the disk. As the disk approaches the seat to close, it makes contact with the seat. As the disk moves further into the seat, the seat physically deforms because of the pressure applied by the disk, causing the polymer to seat against metal surfaces. In some cases, a manufacturer may use both the mechanical preload and Poisson effects to achieve the correct shutoff (Fig. 4.25). When a soft seat is used, it also has a secondary purpose, acting as a gasket between the body and the retainer. Metal seats are typically applied to high temperatures (above 400°F or 205°C). Metal seats are integral to the seat retainer—with a gasket placed where a soft seat is normally inserted (Fig. 4.26). In some designs, both a soft and metal seat can be used in tandem, allowing the metal seat to be a backup in case of failure of the soft seat (Fig. 4.27). When butterfly valves are specified for fire-safe applications, the tandem seat is



Poisson Effect with Pressure Upstream



Poisson Effect with Pressure Downstream

Figure 4.23 Poisson effect on a butterfly seal for both upstream and downstream pressures. (*Courtesy of Valtek International*)

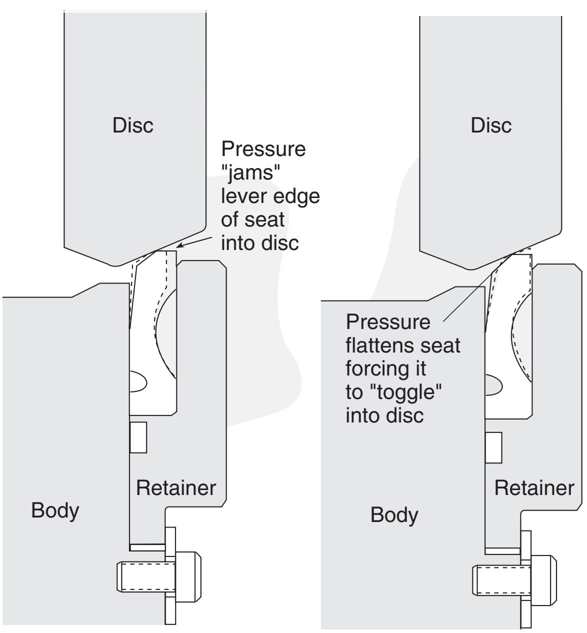


Figure 4.24 Jam lever or toggle effect on the butterfly seal. (*Courtesy of Valtek International*)

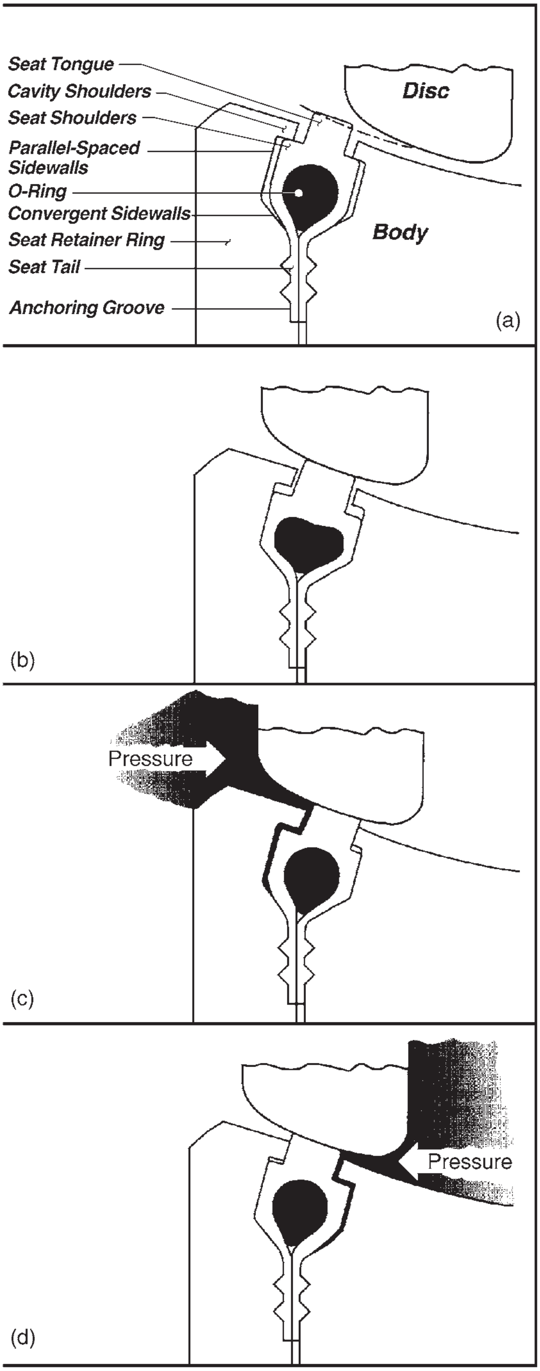


Figure 4.25 Butterfly seal using both mechanical preloading and the Poisson effect. (*a*) Basic seal design, (*b*) preloading effect on the seat caused by disk seating (with minimal pressure effects), (*c* and *d*) Poisson effect on the seat caused by increased upstream or downstream pressures. (*Courtesy of Flowseal, a unit of the Crane Valve Group*)

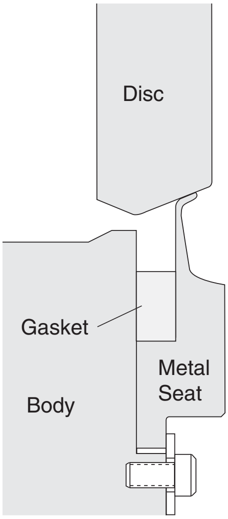


Figure 4.26 Butterfly metal seat design. (*Courtesy of Valtek International*)

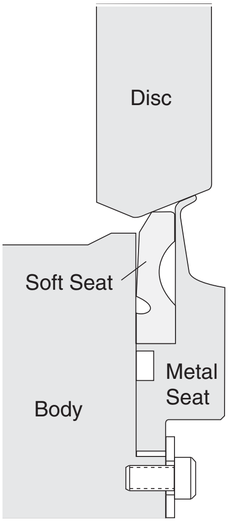


Figure 4.27 Butterfly dual soft- and metal-seat design. (*Courtesy of Valtek International*)

installed. In pure throttling applications, where the valve is intended to remain in midstroke at all times and never close, the valve can be built without a seat as a cost-saving measure.

A butterfly valve's packing box is similar in some regards to the globe valve's packing box. The packing box has characteristics similar to all packing boxes: a polished bore and a depth to accommodate various packing designs. One major difference, however, is that a butterfly valve does not require a lower set of packing. Because of the rotary-motion design, the stem rotates and never changes linear position. In other words, the packing always remains in contact with the same region of the stem. Since the stem never moves its linear position, a "wiper" packing set is not necessary. All that is required is an optional spacer, the packing, and a packing follower. An upper guide or bearing is not needed at the open end of a butterfly-valve packing box as the shaft has its own guides on each side of the disk. The shaft can also be guided by a bearing in the actuator's transfer case. A gland flange and packing follower are used to compress the packing.

Because the shaft bore is normally machined from both ends, a plug or flange cover can be used to cover the bore opening opposite the packing box. To retain the body pressure, a gasket or O-ring is required. If a threaded plug end is used, it should not come in contact with the shaft, since the quarter-turn action of the shaft could possibly rotate the end plug, causing process leakage to atmosphere.

On the packing box side of the body, mounting holes are provided allowing the transfer case to be mounted. The *transfer case* contains the linear-motion to rotary-motion mechanism that allows a linear-motion actuator to be used with a quarter-turn valve. The end of the shaft that fits into the transfer case is either splined or milled with several flats to allow for attachment of the linkage. The designs of common rotary actuators, actuation systems, and handwheels are detailed in Chap. 5.

4.3.3 Butterfly-Control-Valve Operation

As the process fluid enters the butterfly body, it moves in a straight direction through the flow passage. The only obstruction to the flow is the disk itself. In the open position, the gradual angles and smooth, rounded surfaces of the disk allow the flow to continue past the regulating element without creating substantial turbulence. However, some turbulence should always be expected because the disk is located in the middle of the flow stream. In closing the valve, as the signal is received by the actuator or actuation system, the force is transferred to rotary motion, turning the shaft in a *quarter-turn motion,* which is defined anywhere between 0° (full-closed) and 90° (full-open). As the disk approaches the seat, the full pressure and velocity of the process fluid are acting on the full area of the face or back side of the disk (depending on the flow direction), which makes stability difficult. This instability may be compounded when diaphragm actuators are used, since they do not generate high thrust to begin with. Because the rangeability of butterfly valves is so poor (20 to 1), the final 5 percent of the stroke (to closure) is not available to the user. As the disk makes contact with the seat, some deformation takes place, allowing the resilient elastomer or flexible metal strip to mold against the seating surface of the disk.

To open the valve, the signal causes the disk to move away from the seating surfaces. Because of the mechanical and pressure forces acting on the disk in the closed position, a certain amount of rotary-motion force, called *breakout torque,* must be generated by the actuator or handwheel to allow the disk to open. The designs with the greatest requirement for breakout torque are those designs that require a great deal of actuator thrust to close and seat the valve. Therefore the greater the actuator force for closure, the greater the breakout torque. When fluid pressure is utilized to assist with the seat, less actuator force is required and thus less breakout torque.

In principle, the opening disk is nearly in a balanced state, since one side is pushing against the fluid forces, while the other side is pulling with the fluid forces. However, because both sides of the disk are not identical—the shaft is connected on one side, while the opposite side is more flat—flow direction has a tendency to either push a disk open or pull it closed. In most cases, when the shaft portion of the disk is facing the outlet (downstream), the process flow tends to open the valve. On the other hand, when the shaft portion is facing the inlet side (upstream), the flow tends to close the valve. The failure mechanism of the actuator must complement the flow direction, so that the proper failure mode will occur.

With concentric disk–seat arrangements (the center of the disk and the shaft are exactly centered in the valve), a portion of the disk always remains in contact with the seat in any position. At 0° open, the seating surfaces are in full contact with each other. In any other position, the seating surfaces touch at two points where the edges of the disk touch the seat. Because of this constant contact, the concentric disk–seat design has a greater tendency for wear, especially with automated control applications. During throttling, a butterfly valve may be required to handle a small range of motion in midstroke, causing wear at those two points of contact. Although the wear will not be evident during throttling, it will eventually allow leakage at those two points when the valve is closed. To overcome this problem of constant contact between the seating surfaces, some butterfly-valve manufactures prefer to use the eccentric cammed disk–seat configuration, which allows for the disk and seat to be in full contact upon closure, but when the valve is open the disk and seat are no longer in contact. Such designs allow for the center of the shaft (and disk) to be slightly offset down and away from the center of the valve. When the valve opens, the disk lifts out of the seat and slightly away from the seating surfaces—enough to avoid constant contact.

Because of the design limitations of the disk and seat arrangement, a flow characteristic is not easily designed into the body subassembly, unlike the trim of a globe valve. Thus, a butterfly valve must use its inherent flow characteristic, which is parabolic in nature. To achieve a flow characteristic, an actuator with a cammed positioner must be used to provide a modified flow characteristic.

A feature unique to high-performance valves is the ability to mount the valve on either side of the pipeline so that the shaft orientation (shaft upstream or shaft downstream) and the failure mode (fail-open and fail-closed) can operate in tandem with the air-failure action of the actuator. Figure 4.28 shows the four common orientations [(1) fail-closed, shaft upstream, air-to-open; (2) fail-open, shaft upstream, air-to-close; (3) fail-open, shaft downstream, air-to-close; and (4) fail-closed, shaft downstream, air-to-close].

4.4 Ball Control Valves

4.4.1 Introduction to Ball Control Valves

Similar in many respects to the butterfly control valve, ball valves have been used for throttling service for the past two decades. As control valves, they have been adapted from the automation of simple on–off valves to automatic control valves designed specifically to accurately control the process. Improved sealing devices and highly accurate machining of the balls have provided tight shutoff as well as characterizable control. For the most part, they are used in services that require high rangeability. Ball control valves typically handle a rangeabilty of 300 to 1, notably higher than butterfly control valves that offer 20 to 1. Such high rangeability is permitted by the basic design of the regulating element, which allows the ball to turn into the flow without any significant side loads that are typical of a butterfly disk or a globe-valve plug.

Ball control valves are also well suited for slurry applications or those processes with fibrous content (such as wood pulp). The rotary

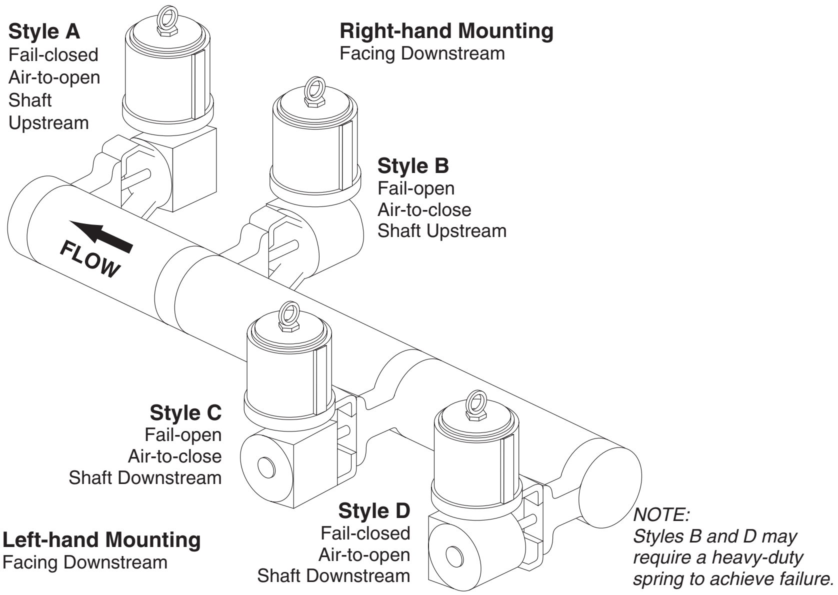


Figure 4.28 Rotary actuator mounting orientations. (*Courtesy of Valtek International*)

action of the ball provides a shearing action against the seal, which allows for clean separation of the process during closure. The same process would clog or bind in a butterfly or globe control valve (which uses a regulating element or trim directly in the path of the process flow). Similar to the butterfly-valve design that features straight-through flow, a ball valve can be installed in a vertical pipeline (Fig. 4.29) to avoid the settling or straining of fibrous or particulate matter. A globe valve, on the other hand, allows heavier portions of the process to settle at the bottom of the globe body (horizontal line installations) or in the body gallery (vertical line installations).

Tight shutoff is a characteristic of ball control valves, since the ball remains in continual contact with its seal. With soft seals, ball control valves can achieve ANSI Class VI shutoff (bubble-tight) but have a limited temperature range. For higher-temperature ranges, metal seals are used although they permit greater leakage rates (ANSI Class IV). Ball valves are also capable of higher flow capacity than globe valves, and even butterfly valves where the presence of the disk in the flow stream can restrict the flow capacity. Because the flow capacity of a typical ball valve can be two to three times greater than that offered by

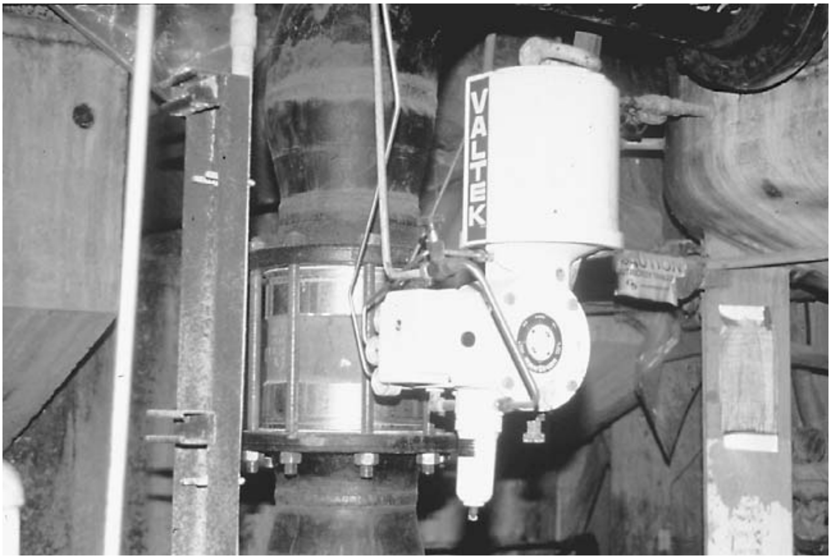


Figure 4.29 Ball control valve mounted in a vertical line. (*Courtesy of Valtek International*)

a comparably sized globe valve, a smaller-sized ball valve can be used, which may be a significant economic consideration. Table 4.2 shows a comparison of flow capacity between globe (both T and Y styles), butterfly and ball valves.

One major disadvantage of ball control valves is that as the valve throttles the geometry changes dramatically, providing lower pressure differentials, higher pressure drops, and an increasing chance of cavitation, although the straight-through flow style of ball valves provides a minimal pressure drop. Therefore if the service conditions are likely to result in cavitation, larger-sized ball valves may be required to provide higher differentials and to prevent a high-pressure drop from developing—defeating one of the purposes of ball valves, which is to use a smaller-sized valve with a large C_v. Using a larger ball valve also means that a good portion of the valve stroke will not be available for control purpose, utilizing the portion of the stroke closest to the closed position.

Two basic ball-valve designs are used today: the *full-port ball valve* and *characterizable-ball valve*. Similar in design to a manually operated on–off block ball valve, a full-port ball valve uses a spherical ball as the regulating element, characterized by a hole that is bored to the same inside diameter as the pipeline (Fig. 4.30). When the full-port ball

Table 4.2 C_v Comparisons Globe vs. Ball Valves*

Valve Size	Globe Valve (T-body style, flow-over-the-plug, full area trim, 100 percent open)	Ball Valve (Wafer-style, shaft downstream)	Percentage Increase
2-inch DN 50	46	104	126%
3-inch DN 80	104	275	164%
4-inch DN 100	179	445	149%
6-inch DN 150	355	844	138%
8-inch DN 200	606	1338	121%
10-inch DN 250	897	3180	255%
12-inch DN 300	1310	4150	217%

**Data courtesy of Valtek International.*

valve is wide open, the flow continues unimpeded through this hole. Therefore, the flow does not impinge on a regulating element or trim, creating little (if any) pressure drop as well as minimal process turbulence. Although best utilized for on–off services, a full-port valve is rarely used for a pure throttling service because the sharp edges associated with the ball's bore may create noise, cavitation, erosion, and an increased pressure drop. Although a full-bore ball valve is often associated with on–off services, it is also applied where a pig or cleaning rod is used to clean out the interior of the pipeline. (This requires using a valve with straight-through flow that does not have a regulating

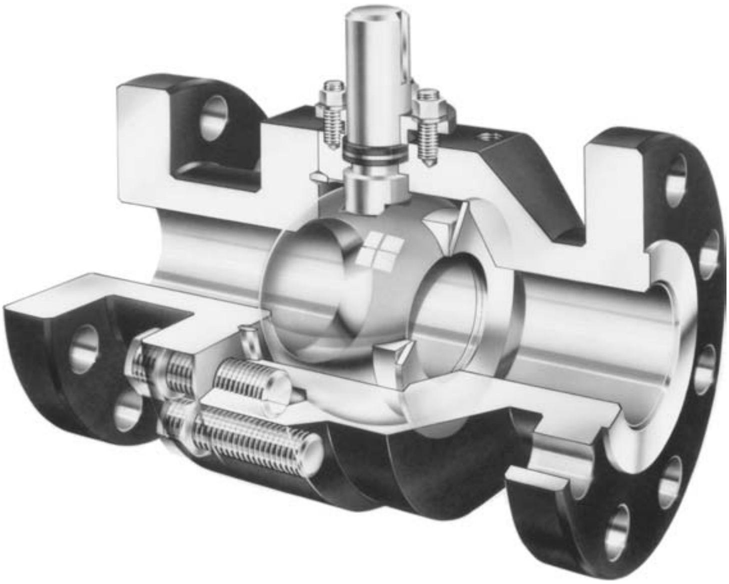


Figure 4.30 Full-port ball valve with floating seal. (*Courtesy of Vanessa/Keystone Valves and Controls, Inc.*)

element in the flow stream.) Because of the design limitations of full-port ball, a flow characteristic cannot be designed into the ball. The machining of orifice shapes other than circular is exceptionally difficult and expensive. The inherent flow characteristic associated with full-port valves is close to the equal-percentage characteristic, and any flow characteristic modifications must be made with a positioner cam.

The characterizable-ball valve (Fig. 4.31) does not use a spherical ball. Instead, it uses a hollow segment of a sphere that, when full-open, is turned out of the path of the process flow. This allows reasonably smooth flow through the valve body, although the contours of the body and geometry of the characterized ball will take a small pressure drop and may create some turbulence. However, as the valve moves to a midstroke throttling position, the characterized ball moves into the flow path. The flow characteristic is cut into the ball with either a V-notch or a parabolic curve to provide the necessary flow per position. As the valve continues through the quarter-turn motion, this notch or curve becomes progressively smaller until the entire surface of the ball is exposed to the flow area, providing a full-closed position. The V-notch provides an inherent linear flow characteristic, which can

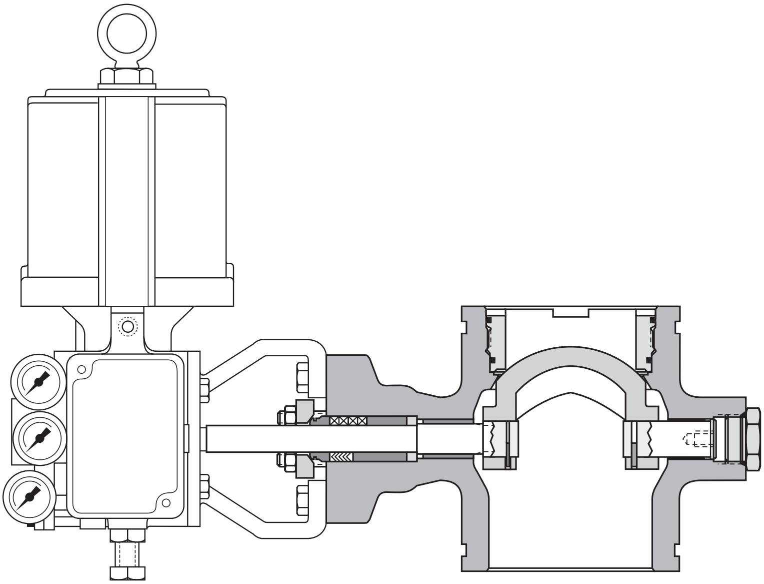


Figure 4.31 Characterizable-ball control valve. (*Courtesy of Valtek International*)

become close to the equal-percentage characteristic when installed. The parabolic notch can be modified to meet specific flow requirements.

Ball control valves are typically found in sizes 1 through 12 in (DN 25 through DN 300) in pressure classes up through ANSI Class 600 (PN 100).

4.4.2 Ball-Control-Valve Design

Outside of the regulating element, ball control valves are similar in many regards to butterfly control valves: quarter-turn motion, rotary-action actuators, and packing boxes without wiper (lower) packing.

As described in Sec. 4.4.1, two basic ball-valve styles exist: the full-port ball valve and characterizable-ball valve. The regulating element of the full-port body subassembly features a spherical ball that is supported by one of two methods. The first is a *floating-seal* design (Fig. 4.30), similar to most manual ball-valve designs, where two full contact seals are placed on both the inlet and outlet ports, in which the ball is fully supported by these two seals without coming in direct contact with the body. The ball is connected to the shaft using a slip fit or other comparable connection. This connection must be extremely tight to avoid any mechanical hysteresis, especially in light of the continuous seal friction evident in this design. The basic advantage of this design is that a blind end bore is not required to support the nonshaft end of the ball. The disadvantage is that the sphere must have extremely tight tolerances to ensure constant contact at both seals. These seals are designed for more rigorous, heavy-duty service since they must both seal the flow and support the ball. Because this design is dependent upon the support of the seals, it is specified for general services featuring moderate pressures and temperatures.

The characterizable ball is typically *segmented,* meaning that only a portion of the sphere is used instead of an entire sphere. The segmented ball includes only enough of the sphere to entirely close off the flow area plus enough ball surface to provide a seal. A segmented ball is normally *trunnion-mounted* (Fig. 4.32). With trunnion mounting, the ball is supported by both the shaft and the side opposite the shaft using another shaft or post, which can be separate or integral to the ball. Because support is not handled by a seal, trunnion-mounted balls are normally designed with one seal (although two-seal designs are available), which provides less friction between the ball and seal. Trunnion-mounted designs are best for more severe services where higher pressures and temperatures are involved.

Ball valves can be provided with either soft or metal seals. With soft seals, the elastomer seal is provided with a metal or hard-elastomer backup ring to apply continual pressure to the sealing surface, act as a backup in case the elastomer fails, and to provide additional wiping of sealing surfaces. With highly corrosive or nonsparking services—such as an oxygen application—metal backup rings are prohibited in favor of hard elastomers. If a metal seal is required because of temperature extremes, care must be taken to provide complementary metals so that galling or scoring does not take place. Metal seals require heat treatment and/or coating of the ball.

The style of the body determines how the seals are held in place in relation to the ball. With one-piece bodies, the ball is installed followed by the seal, which is held in place by a retainer. Most retainers are threaded into the body, allowing for minute adjustments of the retainer to increase or decrease the compression of the seal against the ball. This design balances the integrity of the seal versus increased ball–seal friction. Ideally, the retainer should not encompass the entire gasket region surface of the body face but should share it with the body. If the

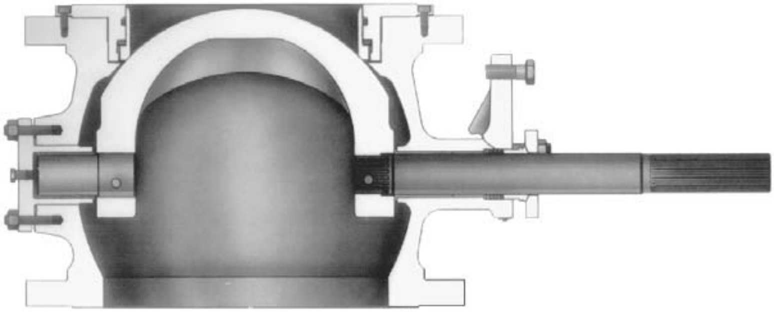


Figure 4.32 Trunnion-mounted segmented-ball valve. (*Courtesy of Fisher Controls International, Inc.*)

retainer does handle the entire seal, its compression of the seal will be affected by the piping forces. With uneven piping forces, they can create an uneven seal. To ensure uniform seal tightness, shims of varying width are often used between the retainer and the seal.

A few ball-valve bodies use two-piece designs in which the body is divided in half (much like a split-body globe valve), allowing for easier assembly and the use of a floating ball. The major drawback to using the two-piece design is that piping forces or process temperature can alter the seal tightness. As with all split-body designs another potential leak path is created at the joint between the two body halves.

Because the body's face-to-face is dependent upon the design of the body subassembly, that dimension varies from manufacturer to manufacturer. No overall standards have been established that all manufacturers adhere to, as opposed to ANSI/ISA Standard S75.15 or ANSI/ISA Standard S75.16 for globe-style valves. Because the ball-valve face-to-face is larger than the thin wafer-style body of the butterfly valve, yet smaller than the globe body, its body can be installed between piping flanges in some applications. When high temperatures or thermal cycling are present, the longer bolting between the piping flanges can result in lost compression through thermal expansion and cause leakage. Also, even if temperatures are moderate, the bolting associated with larger valves [8 in (DN 200) or larger] can stretch over time and cause leakage. For those applications in which a flangeless design is not practical, ball valves are also available with integral flanges or separable flanges. Integral flanges offer solid, one-piece structure integrity, while separate flanges offer lower cost (with alloy bodies) as well as easier installation when piping does not match up with the valve flanges.

The packing box is nearly identical to that found in butterfly control valves. Similar to other packing boxes, the bore is polished and deep enough to accommodate a wide variety of packing designs. As is the case with butterfly valves, the rotary quarter-turn action of the ball valve does not require a lower set of packing to wipe the shaft of any process. A typical packing box will include the packing set, an optional spacer and a packing follower (which is used to transfer the force of the gland flange to the packing). Unlike globe valves, an upper guide or bearing is not needed at the open end of a ball-valve packing box as the shaft is normally guided on each side of the ball. In some automated rotary-motion valves, the shaft is also guided by a bearing in the actuator's transfer case.

For machining simplicity of the trunnion-mounted design, the shaft bore is machined from both ends of the body, and a plug or flange cover (plus a gasket or O-ring) can be used to cover the bore opening opposite the packing box. If a threaded plug is used, it should not come in contact with the shaft, since the quarter-turn action of the shaft could unthread the plug, causing process leakage to atmosphere. Mounting holes are provided on the packing-box side of the body, allowing the transfer case of the actuator to be mounted. As with all automated rotary valves, the transfer case contains the linear-motion to rotary-motion mechanism that allows a linear-motion actuator to be used with a quarter-turn valve. The end of the shaft that fits into the transfer case is either splined or milled with several flats to allow for attachment of the linkage. The designs of common rotary actuators, actuation systems, and handwheels are detailed in Chap. 5.

4.4.3 Ball-Control-Valve Operation

As with all rotary-action valves, the ball valve strokes through a quarter-turn motion, with 0° as full-closed and 90° as full-open. The actuator can be built to provide this rotary motion, as is the case with a manual handlever, or can transfer linear motion to rotary action using a linear actuator design with a transfer case.

When full-open, a full-port valve has minimal pressure loss and recovery as the flow moves through the valve. This is because the flow passageway is essentially the same diameter as the pipe inside diameter, and no restrictions, other than some geometrical variations at the orifices, are present to restrict the flow. The operation of throttling full-port valves should be understood as a two-stage pressure drop process. Because of the length of the bore through the ball, full-port valves have two orifices, one on the upstream side and the other on the downstream side. As the valve moves to a midstroke position, the flow moves through the first narrowed orifice, creating a pressure drop, and moves into the larger flow bore inside the ball where the pressure recovers to a certain extent. The flow then moves to the second orifice, where another pressure drop occurs, followed by another pressure recovery. This two-step process is beneficial in that lower process velocities are created by the dual pressure drops, which is important with slurry applications. The flow rate of a full-port valve is determined by the decreasing flow area of the ball's hole as the valve moves through the quarter-turn motion, providing an inherent equal-percentage characteristic with a true circular opening. As the area of the flow passageway diminishes as the valve approaches closure, the sliding action of the ball against the seal creates a scissorslike shearing action. This action is ideal for slurries where long entrained fibers or particulates can be sheared off and separated at closing. On the other hand, globe-valve trim and butterfly disks do not have this shearing action and can only attempt to separate the fibers by pinching them between seating or sealing surfaces. In many cases, the fibers stay intact and do not allow for a complete seal, creating unplanned leakage.

At the full-closed position, the entire face of the ball is fully exposed to the flow, as the flow hole is now perpendicular to the flow, preventing it from continuing past the ball.

With the characterized segmented-ball design, only one pressure drop is taken through the valve—at the orifice where the seal and ball come in contact with each other. When the segmented ball is in the full-open position, the flow is restricted by the shape of the flow passageway. In essence, this creates a better throttling situation, since a pressure drop is taken through the reduction of flow area. As the segmented ball moves through the quarter-turn action, the shape of the V-notch or parabolic port changes with the stroke, providing the flow characteristic. Like the full-port design, the sliding seal of the characterizable ball provides a shearing action for separating slurries easily.

4.5 Eccentric Plug Control Valves

4.5.1 Introduction to Eccentric Plug Control Valves

One control valve design that is growing in demand is the *eccentric plug valve* (sometimes called *eccentric rotating plug valve*), which combines many of the positive aspects of the globe, butterfly, and ball valves. In simple terms, the eccentric plug valve is a rotary valve that uses an offset plug to swing into a seat to close the valve, much like an eccentric butterfly valve. However, the eccentric movement of the plug swings out of the flow path, similar to a segmented-ball valve. Overall this design provides minimal breakout torque, as well as tight shutoff without excessive actuator force. Figure 4.33 shows the internal construction of an eccentric plug valve.

Eccentric plug valves can typically handle pressure drops from 1450 psi (100 bar). The eccentric motion also avoids water-hammer effects and the poor rangeability inherent with butterfly valves. Unlike a ball valve where the ball is in constant contact with the seal, the plug lifts off the seat upon opening. Seat contact and partwear only occur when the valve is closed (Fig. 4.34)—a feature similar to globe-valve trim. Because the plug swings out of the flow area—as does a segmented-ball valve—it allows for greater flow capacity and avoids erosion from the process.

With the stability of the plug design, eccentric plug valves provide exceptional stability, providing rangeabilty of greater than 100:1, compared to 50:1 for globe valves and 20:1 for butterfly valves. Only the ball control valve has better rangeability (up to 400:1). Because the shaft and plug do not directly intersect the flow, the flow capacity is slightly less than ball valves but is better than most high-performance globe and butterfly valves. Its design permits a reasonable pressure drop to

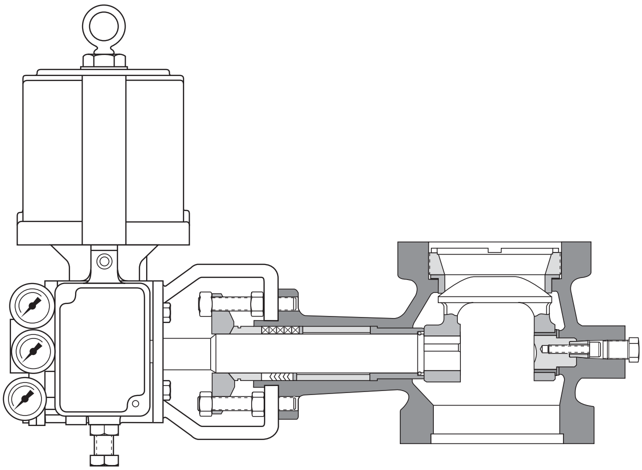


Figure 4.33 Eccentric plug valve. (*Courtesy of Sereg/Valtek International*)

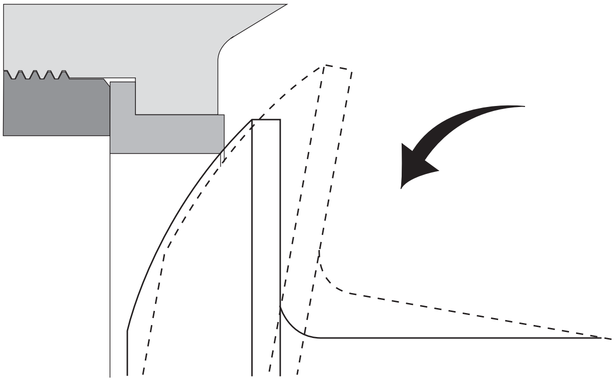


Figure 4.34 Seating path of eccentric plug design. (*Courtesy of Sereg/Valtek International*)

be taken across the valve. Eccentric plug valves are best applied in applications with moderate pressure drops. In normal applications, the eccentric plug valve operates equally well in either flow-to-close or flow-to-open applications. The design of the plug permits the flow direction to assist with the closure or opening of the valve. As the eccentric plug valve opens, the flow characteristic is an inherent linear characteristic. With the regulating element outside on the outside boundaries of the flow, very little process turbulence is created.

Eccentric plug valves are typically available in sizes from 1 in (DN 25) to 12 in (DN 300), in ANSI Classes up through Class 600 (PN 100), and handle temperatures typically from −150°F (−100°C) to 800°F (430°C).

4.5.2 Eccentric-Plug-Control-Valve Design

The body design of an eccentric plug valve is very similar to a characterizable segmented ball valve in many aspects. The valve body and packing box are similar in shape and function, although the shaft alignment with the seal is different. With a ball valve, the centerline of the shaft is aligned exactly with the seal so that the ball is always in direct contact with the seal, whereas the shaft of an eccentric plug valve is slightly offset from the seat. This offset keeps the rotating plug away from any seating surfaces until closure occurs. Overall, this is similar in concept to the offset of an eccentric and cammed disk in high-performance butterfly valves. With fail-closed situations, the offset design positions the plug correctly upon failure, reducing the actuator failure spring requirements.

Although a segmented ball and an eccentric plug look similar at first appearance, each is designed differently. Where the ball is spherical in design, the plug is designed more like the plug head of a globe valve that is attached at a right angle with the shaft. The contour of the face of the rotary plug is similar to a modified quick-open plug contour in a globe valve, although the major difference is that the contour of the eccentric plug is also the seating surface. The seat construction is similar to the seat retainer in a ball valve, which can be threaded in place. Newer designs use a two-piece construction featuring a floating, self-centering seat with a threaded seat retainer that, when tightened, fixes the seat in place. On the other hand, one-piece seats have difficulty achieving tight shutoff because of the possibility of misalignment between the plug and seat. Seats can be either metal (providing ANSI Class IV shutoff) or provided with a soft seat elastomer (providing ANSI Class VI shutoff).

One design attribute of the eccentric plug valve that is similar to globe valves is its ability to provide reduced trims by simply changing the seat to one with a smaller opening. Because the eccentric plug has one large seating surface, it can be used with a variety of smaller seats, providing a reduced trim option that is not normally available in other rotary valves.

Eccentric plug valves utilize straight-through bolting or flanged end connections.

4.5.3 Eccentric-Plug-Control-Valve Operation

The eccentric ball valve strokes through a quarter-turn motion, with 0° at full-closed and 60° to 80° at full-open. Maximum rotation (80°) is preferred because it provides increased controllability and resolution. When less than full rotation is required, some actuators have limit-stops that can prevent the full motion.

When the valve is in the full-open position, the plug is located nearly perpendicular to the seat (Fig. 4.35) and parallel to the flow. As the flow moves through the body, it is restricted by the diameter of the seat and geometric shape of the plug, taking a reasonable pressure drop.

In fail-open applications, the flow assists the opening of the plug since the shaft is downstream from the flow and the plug swings with the flow until it is perpendicular to the seat. The process flows through

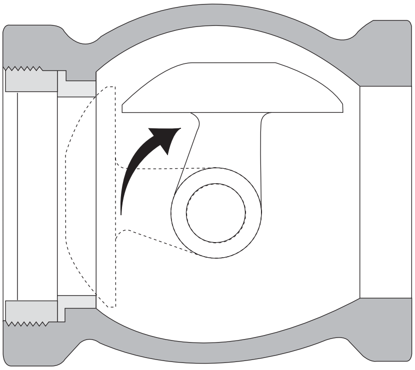


Figure 4.35 Eccentric plug in the open position. (*Courtesy of Sereg/Valtek International*)

the seat, taking a small pressure drop, and then slightly recovers inside the body. The majority of the flow moves through the center of the valve body and the horseshoe-shaped opening of the plug, encountering minimal flow resistance. As the flow exits the valve body, the pressure recovery is completed. As the valve begins to close, the plug moves against the flow, restricting the flow by degrees until the plug is approaching the closed position. At that point, the offset shaft aligns the plug exactly with the seat, seating surfaces meet, and the valve closes.

In fail-closed applications, the shaft is upstream from the flow and the plug must open against the flow, moving perpendicular to the seat. Flow moves through the body and the plug opening to the seat, taking a small pressure drop at the plug opening and a larger pressure drop at the seat, with pressure recovering in the downstream piping. As the valve fails, the direction that the plug swings to close is the same as the flow, using the flow pressure to assist with the closure. A feature unique to automatic rotary valves in general is the ability to mount the valve on either side of the pipeline so that the shaft orientation (shaft upstream or shaft downstream) and the failure mode (fail-open and fail-closed) can operate in tandem with the air-failure action of the actuator. Figure 4.28 is a good reference illustration for showing the four common orientations (fail-closed, shaft upstream, and air-to-open; fail-open, shaft upstream, and air-to-close; fail-open, shaft downstream, and air-to-close; and fail-closed, shaft downstream, and air-to-close).

5 Manual Operators and Actuators

5.1 Introduction to Manual Operators and Actuators

5.1.1 Purpose of Manual Operators and Actuators

With most valves, some mechanical device or external system must be devised to open or close the valve, or to change the position of the valve if it is to be used in throttling service. Manual operators, actuators, and actuation systems are those mechanisms that are installed on valves to allow this action to take place.

5.1.2 Definition of Manual Operators

A *manual operator* is any device that requires the presence of a human being to provide the energy to operate the valve, as well as to determine the proper action (open, closed, or a throttling position). Manual operators require some type of a mechanical device that allows the human being to easily transfer muscle strength to mechanical force inside the valve, usually through a handwheel or lever that provides mechanical leverage. Since the beginning of process industry, manual operators have been in use and are very commonplace, although over the past three decades, their use has declined somewhat in favor of automatic control actuators. The reason is simply the cost as well as imperfections of the human operator. A human being must be dispatched to the valve with a manual operator and complete the action on the valve. With simple on–off control, this action may be adequate. However, with the accuracy required in today's process systems, the human operator may not be fast enough to reach a valve—or stroke it—when an action is required. With throttling situations, a human operator can only guess at an approximate position of the valve's closure element, which may not be exact enough for a critical service. Even an extra half turn of a handwheel may create too much or too little flow, pressure, or temperature for some applications, especially with some inherent or installed flow characteristics. In addition to the slowness and inaccuracies of human beings, some applications have high internal forces that manual operators cannot overcome because of the physical limitations of the human being, even with extraordinarily long levers or wide handwheels. Also, in business terms, human beings are expensive. The days are over when runners on bicycles were dispatched from the control room to turn handwheels. Nearly all plants today are looking for technology to replace human beings, not only because of the human resource cost, but also for the greater accuracy, efficiency, and productivity associated with higher technology.

5.1.3 Definition of Actuators and Actuation Systems

Automatic control of valves requires an *actuator,* which is defined as any device mounted on a valve that, in response to a signal, automatically moves the valve to the required position using an outside power source. The addition of an actuator to a throttling valve, which has the ability to adjust to a signal, is called a *control valve.* Some say that by the pure definition of actuator, a manual operator is an actuator. However, when most people associated with valves discuss the term actuator, they are referring to a power-actuated operator using an outside signal and power source rather than a human being. Typical classifications of actuators include pneumatic actuators (diaphragm, piston cylinder, vane, etc.), electronic motor actuators, and electrohydraulic actuators. *Actuation systems* are special actuators that are commonly mounted on manually operated valves and can be used in either on–off or throttling applications.

Actuators are critical elements in the *control loop,* which consists of a sensing device, controller, and an actuator mounted on a valve. With a control loop, a sensing device in the process system—such as a temperature sensor or a flow meter—is installed downstream from the control valve and is set to measure a particular variable in the process. The sensor reports its finding to a controller, which compares the actual data against the predetermined value required by the process. If the measured value is different from the predetermined value, the controller sends a correction signal to control valve's actuator. This signal can be sent using one of three methods: increasing or decreasing air pressure, varying electric voltage, or increasing or decreasing hydraulic pressure. The actuator receives this signal and moves accordingly to vary the position of the closure element until the controller determines that the measured value is equal to the predetermined value. At that point, the signal increase or decrease stops, and the actuator—and subsequently, the closure element—holds its position.

Not only must the actuator have the ability to adjust to a changing signal, but it must also have enough power to overcome the internal forces of the process, the effects of gravity, and friction in the valve itself. The majority of applications requiring actuators today require the use of compressed air, with nine out of ten actuators pneumatically driven. Air is by far the preferred power medium, since it is relatively cheap and is available in nearly all plants. In addition, it does not contaminate the environment and can be regulated easily. Typical plant compressed air supply is generally between 60 and 150 psi (between 4 and 10 bar), which is sufficient to run a large portion of the pneumatic actuators available today. When a valve must overcome exceptionally high pressures or when the valve must stroke quickly, bottled nitrogen is often used, allowing pressures up to 2200 psi (150 bar). Not only does a bottle allow for high pressures of nitrogen, it also relatively moisture-free and extremely free of particulates and other foreign material. In general, the disadvantage of air-driven actuators is that, because of the compressibility of gases, some exactness is lost through that medium.

Other power sources can include electrical (both ac and dc power) as well as hydraulics (and to a far lesser extent, steam). Although electromechanical and electrohydraulic actuators are more expensive than pneumatic actuators, they do have the advantages of extremely good accuracy and the ability to operate in environments experiencing low temperatures (where typical air lines can freeze from condensed water) or when high thrusts are required.

If a signal is sent separately from the power supply, pneumatic or electric signals are the industry preference. Prior to 1980, the majority of actuators received pneumatic signals. These signals were typically 3 to 15 psi (0.2 to 1 bar), although 3 to 9 psi (0.2 to 0.6 bar) and 9 to 15 psi (0.6 to 1 bar) were also commonplace. However, with the arrival of the precise control associated with electropneumatic and digital control systems, the pendulum has swung in favor of the electric signals (4 to 20 mA or 10 to 50 mA).

Actuators are described as either single or double acting. A *single-acting actuator* uses a design in which the power source is applied to only one side of an *actuator barrier* (piston, diaphragm, vane, etc.) and the opposite side is not opposed by the power sources. A spring may be added to the opposite side to counteract the single action. A related term is the *direct-acting actuator,* which refers to a design in which the power source is applied to extend the stem. On the other hand, a *reverse-acting actuator* refers to an actuator where the power source causes the actuator stem to retract. *Double-acting actuator* is a term used for actuators that have power supplied to both sides of an actuator barrier. By varying the pressure on either side of the actuator barrier, the barrier moves up or down. Pneumatic double-acting actuators nearly always require the use of a positioner to provide the varying power to the chambers above and below the barrier.

An actuator is normally a separate subassembly from the body, meaning it can be removed from the body for servicing without disassembly of the body subassembly. On the other hand, the body can be serviced without disassembly of the actuator.

5.2 Manual Operators

5.2.1 Introduction to Manual Operators

As discussed in Sec. 5.1, manual operators require the strength and positioning ability of a human being in order to operate the valve. Generally, manual operators are associated with the operation of on–off applications, as well as simple throttling applications not requiring undue accuracy or immediate feedback. The majority of the valves described as manual valves in Chap. 3 uses manual operators.

The advantage of a manual operator lies in its mechanical simplicity—minimal moving parts and no sealed chambers to leak or fail. A human being moves one part (such as a handwheel or a lever) and the valve is opened, closed, or placed in a midstroke position. Design simplicity also means that troubleshooting, maintenance, and disassembly are easier. The disadvantage of manual operators is slow response, since response depends upon a human being operating the manual operator—which in some cases may take some time. For example, a linear handwheel may require 30 or more revolutions to close a valve with a 4-in stroke. And, because a human being must be dispatched to a manually operated valve, the travel time to the valve makes for even slower response. Also, if the valve requires throttling (a midstroke) position, the position of the valve depends upon the judgment of the operator, which may vary widely. In some applications this may not be a problem, but as systems have become more exact over the years, finding the right throttling point has become much more difficult with a manual operator.

5.2.2 Manual Operator Design

Generally, manual operators are divided into two categories: linear motion and rotary motion. Linear-motion manual handwheels use a threaded connection between a fixed-position part of the handwheel assembly, such as a yoke or housing, and a dynamic part (usually a handwheel stem). Multiple turns of a hand-held part mechanism—in most cases, a handwheel—cause linear movement of the dynamic stem, which is connected to a linear-motion closure element.

One of the more common designs is shown in Fig. 5.1, which shows an independent linear handwheel operator that is mounted directly to a body subassembly and is not an integral part of the valve. The actuator uses a yoke to support the handwheel mechanism and to attach the operator to the valve. The connection to the body is made with an inside diameter of the lower portion of the yoke, called the *spud*. The yoke's spud fits over the bonnet and is secured with a yoke nut or other clamping device. The closure device's stem—such as a plug stem, compressor stem, or gate stem—is threaded to the bottom of the handwheel stem. The upper portion of the yoke houses the handwheel nut, which turns with the handwheel. Some designs allow the handwheel and nut to be one integral part, while others make them separate because of material considerations. When the handwheel is separate, a key or locking bolt is used to secure the handwheel to the handwheel nut. The handwheel nut is retained in position, allowing rotational movement, and is internally drilled and tapped to receive the handwheel stem. The matching external threads of the handwheel stem are threaded into the handwheel nut, allowing for several threads to be engaged at any given position. Generally, ACME threads are used for manual operators. To avoid problems with constant contact between similar metals, which can lead to galling, the handwheel stem and handwheel nut are made from dissimilar materials. The most common combination is brass or bronze for the nut and stainless steel for the stem. As the handwheel is turned, the retained handwheel nut turns the engaged threads of the handwheel stem, extending or retracting the stem, depending on which direction the handwheel was

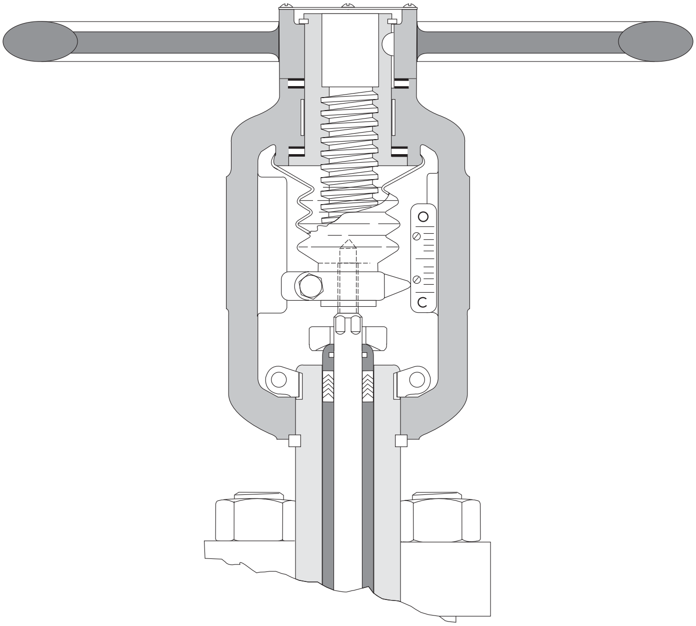


Figure 5.1. Independent linear handwheel operator. (*Courtesy of Valtek International*)

turned. The extension or retraction of the stem then operates the linear motion of the closure element. In some larger designs or high-pressure applications, rollers or races are placed between the handwheel and the upper portion of the yoke to minimize friction between mating parts, providing easier turning of the handwheel.

The chief advantage of the independent operator is that the valve does not need to be disassembled to service the operator. The disadvantage is that the overall valve has a greater height than other designs.

The other common linear manual-operator design is the dependent linear handwheel operator, which has the handwheel mechanism built directly into the bonnet cap of the valve, as shown in Fig. 5.2. In this case, instead of a yoke, the bonnet cap retains the handwheel nut. The one-piece stem has dual duty of operating both the closure element and the handwheel. The obvious advantage of this design over the independent operator is that the height of the valve is far lower. The disadvantage is that operator problems require some valve disassembly.

Linear operators are also divided into two design categories: the rising-stem and nonrising-stem designs. The *rising-stem* design uses a

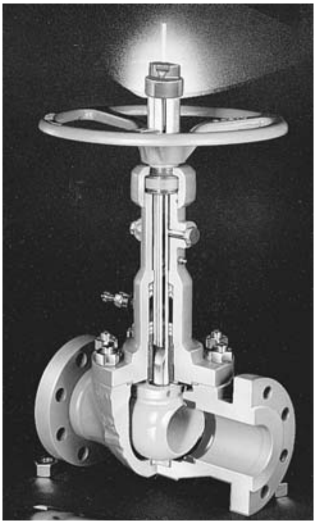


Figure 5.2. Integral linear handwheel operator. (*Courtesy of Orbit Valve Company*)

handwheel nut to retract the handwheel stem. As the handwheel nut is turned, the handwheel stem rises above the handwheel. A majority of manual linear-motion valves use rising-stem operators. On the other hand, the *nonrising-stem* design is typically used with dependent operators. The handwheel turns the retained and threaded stem, which engages the closure element (such as a wedge gate). As the handwheel is turned, the stem turns with it. The closure element is designed to be fixed by guiding so that it cannot rotate; therefore the closure element has a tendency to rise or lower with the stem rotation.

As noted earlier, the most common way of handling a linear manual operator is through a handwheel. Handwheels come in all different surface finishes, from smooth to rough, depending on the work conditions and the type of construction. Many are spoked to save weight, although some petroleum and refining applications require solid handwheels to ensure that they stay intact during a fire. Spoked handwheels have the added advantage of greater security, by allowing a locking mechanism to be placed on the operator to prevent accidental or intentional tampering with the valve's position. Another common handwheel design is the chain wheel. A *chain wheel* is a handwheel with teeth or grooves to accommodate a circular length of chain, allowing for the user to operate an out-of-reach valve.

Rotary-motion manual operators are used with quarter-turn valves, such as plug, ball, and butterfly valves. The most efficient method to turn a quarter-turn closure element is through a right-angle extension of the stem, which allows for better leverage. The two most common types of rotary-motion manual operators are the *handle* and the *wrench.* Many technicians refer to the two terms interchangeably, but a difference does exist. Handles are bolted to the stem of the closure element (Fig. 5.3) and are commonplace with smaller sizes in the lower-pressure classes. Handles are specified with soft-seated ball valves in sizes up to 6 in (DN 150) and butterfly valves in sizes up to 8 in (DN 200). On the other hand, wrenches are not permanently secured to the stem and can be moved from valve to valve (Fig. 5.4), allowing for the operator to place the valve in a particular position and leave it alone without fear of accidental or intentional tampering. Wrenches are normally equipped with plug valves up through 4 in (DN 100) with sleeved plugs and 6 in (DN 150) with lubricated plugs. In some ball and butterfly manual-valve designs, the handle is integral to the stem, but the most common and inexpensive design is a separable handle in which the handle (or wrench) has an opening that is cut to the shape of the

Figure 5.3. Quarter-turn handle mounted on lined ball valve. (*Courtesy of Atomac/The Duriron Company, Valve Division*)

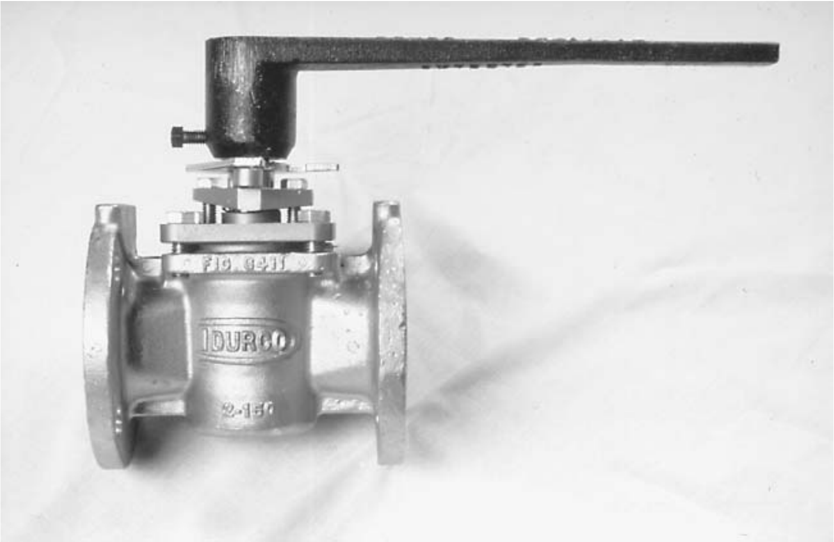


Figure 5.4. Quarter-turn wrench mounted on plug valve. (*Courtesy of The Duriron Company, Valve Division*)

plug stem. A square stem allows for the positioning of the handle or wrench in any one of the four quadrants, while a two-sided flatted stem allows for positioning in one of two positions, front and back. Handles are secured to the stem using a bolt and locking washer.

Handles and wrenches are usually made from ductile iron, although stamped stainless-steel plate is used also. A plastic or rubber grip is placed on the end for comfortable turning. Most manufacturers supply a standard length that handles most applications within the pressure or temperature range of the valve, although longer lengths are sometimes offered to allow for easier operation. Longer lengths, however, may cause problems where space is restricted, not allowing the full quarter-turn motion.

Below the wrench is a *collar-stop* that is used to limit the motion of the closure element to a 90° (or quarter-turn) range. Turning the wrench moves the stem, which in turn moves the plug, ball, or disk, until the collar stops the travel. When the travel is stopped, the closure element should either be in its full-open or full-closed position.

Because of the large forces that can act upon a disk in some applications, butterfly valves may require handlevers for manual operation. A *handlever* is a two-piece, spring-loaded operator that can be positioned in a number of preset slots (Fig. 5.5). The handlever has a fixed upper

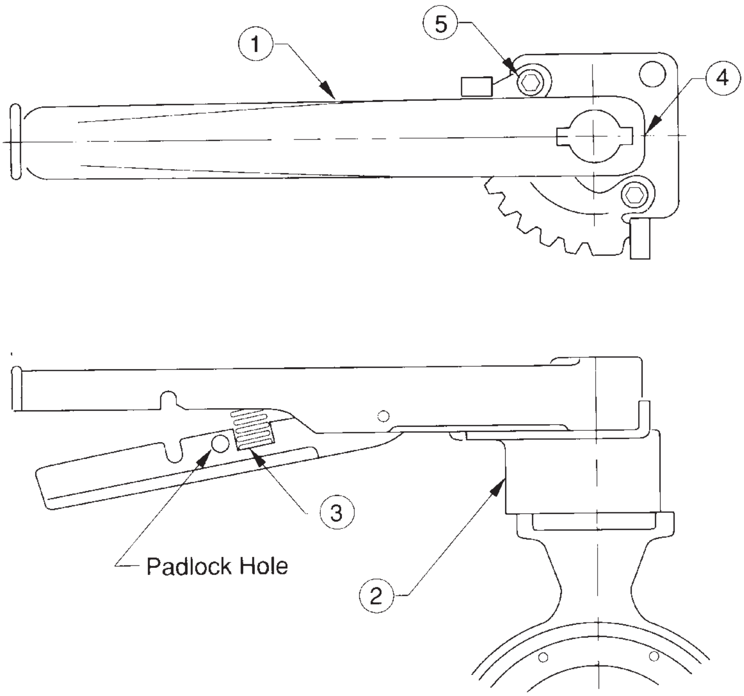


Figure 5.5. Quarter-turn lever operator. Numbered parts are as follows: (1) lever, (2) rachet plate, (3) spring, (4) set screw, (5) socket head cap screw. (*Courtesy of Flowseal, a unit of the Crane Valve Group*)

lever and a movable lower lever. In the static position, the spring loading of the lower lever allows it to seat in one of multiple slots in the collar. By squeezing the upper and lower levers, the lower lever disengages the slot, allowing rotational movement to another desired slot. When the handlever it released, the lower lever seats into the slot, locking the valve in that particular position. The range of slots can vary according to the number of positions required. A typical handlever has a minimum of three positions, full-open, full-closed, and midstroke position, although any number of positions can be planned for as long as room exists for the desired number of slots in the collar.

In larger linear and rotary valves, or in higher-pressure classes, the use of conventional handwheels, handles, and wrenches is not desirable. The circumference of the handwheel or length of the wrench or handle would be so long to handle the leverage that the arc and the

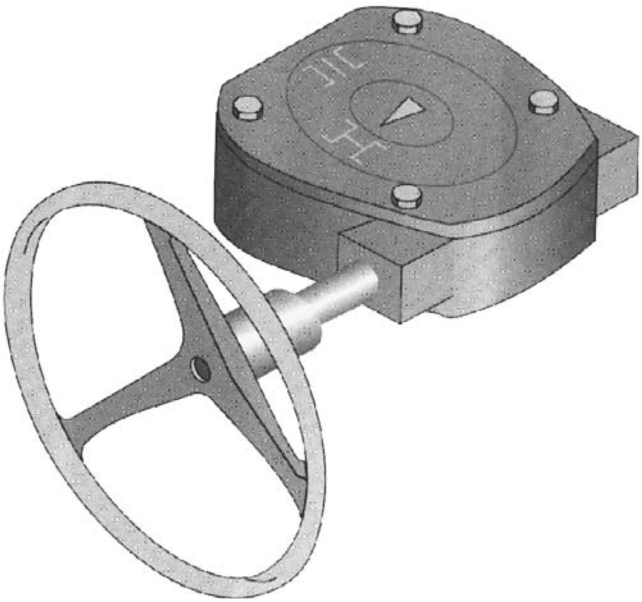


Figure 5.6. Quarter-turn worm-gear operator. (*Courtesy of Flowseal, a unit of the Crane Valve Group*)

weight of the operator would be impractical. In this case, gear operators are used. As shown in Figs. 5.6 and 5.7, *gear operators* (sometimes called *gearboxes*) use gearing to translate handwheel torque into high-output thrust, which is necessary to overcome the greater thrust requirements of larger flows or higher pressures. Linear-motion gearboxes use spur or beveled gearing, while rotary-motion gearboxes use rack-and-pinion or worm gearing. Gear operators use gears with ratios anywhere between 7:1 and 3:1. Both handwheels and cranks are used to turn the gears. The gearing is protected by the gearbox, which not only protects nearby personnel from the turning gears but also minimizes contact with atmospheric or outside conditions. Gear operators are normally bolted onto the bonnet or bonnet cap of linear-motion and some quarter-turn valves and bolted onto the body of butterfly and some ball valves. With linear-motion valves, the stem is threaded directly to the operator stem. With rotary-motion valves, the shaft end may be splined or squared and may intersect with the internal opening of a gear inside the gearbox. When a valve is installed in the line, its position may be difficult to determine without some type of positioner indicator. Most operators have a position indicator consisting of an

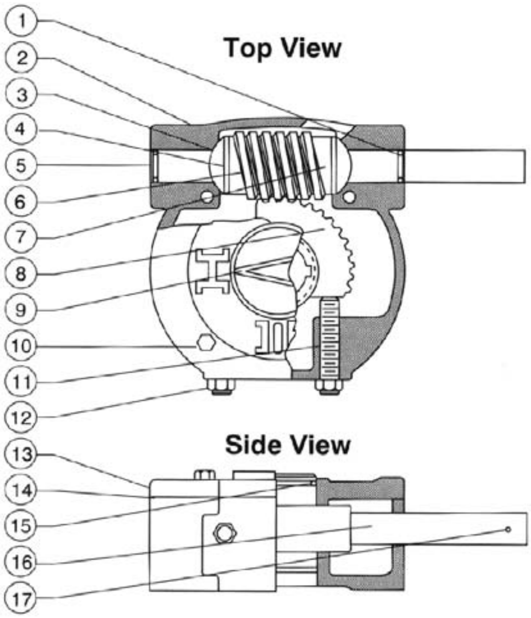


Figure 5.7. Internal view of quarter-turn worm-gear operator. Numbered parts are as follows: (1) seal-input shaft, (2) housing, (3) bearing, (4) washer, (5) plug, (6) worm gear, (7) worm pin, (8) gear segment, (9) indicator cap, (10) cover bolt, (11) stop adjustable screw, (12) hex nut, (13) cover, (14) gasket cover, (15) O-ring, (16) worm shaft, (17) roll pin. (*Courtesy of Flowseal, a unit of the Crane Valve Group*)

arrow and a matching position plate, which shows the position of the valve.

5.3 Pneumatic Actuators

5.3.1 Introduction to Pneumatic Actuators

The most commonly applied actuator is the pneumatically driven actuator, because the power source—compressed air—is relatively inexpensive when compared to a human resource or electrical or hydraulic power sources. For that reason, approximately 90 percent of all actuators in service today are driven by compressed air. When compared to the cost of electromechanical and electrohydraulic actuators, pneumatic actuators are relatively inexpensive as well as easy to understand and maintain. Most are available as standard off-the-shelf products in a number of predetermined sizes corresponding to maximum thrust. Only in special services are special-engineered actuators produced, such as those applications requiring exceptionally long strokes, high stroking speeds, or severe temperatures. From a maintenance standpoint, pneumatic actuators are more easily serviced and calibrated than other types of actuators. Some pneumatic actuators are designed to be *field-reversible,* meaning that they can be converted from air-to-extend to air-to-retract (or visa versa) in the field without special tools or maintenance procedures. Although not as powerful as hydraulic actuators, pneumatic actuators can generate substantial thrust to handle a majority of applications, including high-pressure and high-pressure-drop situations. While air lines are not easy to install, the cost is less than installing electrical conduit and electrical lines as well as hydraulic hoses. Pneumatic actuators also bleed compressed air to atmosphere, which is environmentally safe, when compared to hydraulics. When pneumatic positioners are used with a pneumatic actuator, they are ideal for use in explosive and flammable environments since they do not depend upon electrical signals or power, which could potentially spark a fire if not explosion-proof or intrinsically safe.

The chief disadvantage of pneumatic actuators is that some response and stiffness are lost because of the compressibility of gases—especially with pneumatic actuators that use elastomers with large areas, such as diaphragms. This is not a factor, however, in the majority of applications that do not require a high degree of stiffness or response. With larger actuators, speed is an issue since the volume of the actuator must be filled with compressed air and/or bled to atmosphere to move. For this reason, larger actuators take longer to stroke from full retraction to full extension than smaller actuators, as shown in Table 5.1. Also, pneumatic actuators must be close to an air supply and are dependent upon the continued operation of a compressor unless a separate backup system or volume tank arrangement is installed. Although some designs are better than others, pneumatic actuators do have limits on the amount of thrust available, making some designs unlikely choices for high-pressure applications in large line sizes. Low thrust is commonly associated with diaphragm actuators since the diaphragms can only handle so much air pressure without failing, thus limiting their thrust capabilities.

Table 5.1. Actuator Stroking Times*

Actuator Size *(piston area in inches²/cm²)*	Seconds to Maximum Stroke *(0.25-inch/6 mm tubing)*	Stroke Length *(inches/cm)*	Seconds Per Inch (2.5 cm) of Stroke
25/161	1.2	1.5/3.8	0.8
50/323	3.5	3/7.6	1.2
100/645	9.6	4/10.2	2.4
200/1290	20.8	4/10.2	5.2
300/1936	31.3	4/10.2	7.8

**Data courtesy of Valtek International.* Data based upon cylinder actuator design.

Figure 5.8. Single-acting diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

5.3.2 Pneumatic Actuator Design

The most commonly applied pneumatic actuator over the past 40 years has been the *diaphragm actuator* (Fig. 5.8). Most diaphragm actuators are designed for linear motion, although some rotary-motion designs exist. By definition, a typical diaphragm actuator is a single-acting actuator that provides air pressure to one side of an elastomeric barrier (called the *diaphragm*) to extend or retract the actuator stem, which is connected to the closure element. The diaphragm is sandwiched between upper and lower casings, either of which can be used to hold air pressure, depending on the style of the actuator.

In the single-acting design, the air chamber on one side of the diaphragm is opposed on the other side of the diaphragm by an internal spring, called the *range spring,* that allows the actuator to move in the opposite direction when the air pressure in the chamber is lessened. The range spring also acts as a fail-safe mechanism, allowing the actuator to return to either an open or closed position when the air supply to the actuator is interrupted. Depending on the configuration, the spring is installed next to the diaphragm or the diaphragm plate. The actuator stem is connected to the diaphragm plate and is supported through the top of the yoke with the assistance of a guide. As the diaphragm moves with increasing air pressure, the plate moves in a corresponding manner. That linear motion is directly transferred to the actuator stem, which moves the closure element in the valve. A *yoke* attaches the actuator to the valve body to show the position of the actuator and valve, to support the actuator stem, and to make the actuator-stem to valve-stem connection. It also provides a convenient place to attach accessories. With diaphragm actuators, the most common connection between the body and the actuator is a threaded yoke nut. A clamp is used to prevent the accidental rotation of the actuator stem with the valve stem. The clamp can also be equipped with a pointer that can indicate actuator or valve position.

With conventional single-acting diaphragm actuators, the air signal from the controller to the actuator has a dual role. First, it provides a positioning signal. Second, it provides the power to generate the thrust necessary to overcome the process forces, friction, gravity, the weight of the closure element, and the opposing force generated by the range spring.

Diaphragm actuators have both direct-acting and reverse-acting designs. With the direct-acting design (Fig. 5.9), air pressure is sent to the actuator, which extends the actuator stem and allows the valve to close. This also means that the actuator will retract its stem upon loss of air, allowing the valve to open and remain open. With the reverse-acting design (Fig. 5.10), as the air pressure is sent to the actuator, the

Figure 5.9. Direct-acting diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

stem retracts and the valve opens. If the supply or signal air pressure is interrupted, the actuator moves to the extended position, allowing the valve to close.

With the direct-acting design, air is introduced to the upper casing located above the diaphragm. Beneath the diaphragm are the diaphragm plate and the range spring. The range spring bottoms out in the bottom of the yoke, allowing the upper end of the spring to push against the diaphragm plate and subsequently the diaphragm. In this relaxed (or failure) position, the diaphragm is pushed into the area of the upper casing. As air is introduced into the upper casing and pressure builds, the diaphragm and plate push against the spring. As the signal pressure increases, the air pressure overcomes the opposing forces and the diaphragm and plate move downward. This movement allows the actuator stem to extend and the valve to move toward the closed position. Eventually as the full signal air pressure is reached and the resulting air pressure is introduced into the chamber, the diaphragm and plate reach their full travel. On the other side of the plate, the range spring is nearly fully compressed. At this point, the stem is at its full extension and the valve is closed at the full pressure end of the signal.

Figure 5.10. Reverse-acting diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

As the signal is lessened, resulting in lower air pressure in the chamber, the counterforce of the range spring begins to take effect, and the actuator moves to its relaxed state and the valve is opened.

With the reverse-acting design, the lower chamber is used to provide the air pressure to retract the actuator stem, while a reverse-acting spring is used to provide the counterforce, as well as the failure mode. The upper casing is static and only needs to retain the diaphragm and to vent displaced air volume to atmosphere. With this configuration, the lower casing is pressure retaining and requires an air connection to inject air into that chamber. The diaphragm plate is installed above the diaphragm. The range spring, which is still located below the diaphragm, is seated below the lower casing and is not in direct contact with the diaphragm and plate assembly. Instead, the range spring is seated on a retainer on the lower portion of the actuator stem. Because the range spring bottoms out (or in this case, tops out) at the bottom of the lower casing, as the actuator stem retracts with air to the lower chamber, the spring's resistance increases proportionately. As the actuator stem retracts, the valve begins to open. When the air signal is at the high end of the range, the actuator stem is fully retracted, and the range spring is almost completely compressed. When the signal changes and moves to the lower end of the range, the air pressure to the lower chamber is lessened. At that point, the range spring's counterforce begins to push the actuator stem to the relaxed (extended) state until the full extension is reached and the valve closes.

When positioners are used to improve the overall response of the actuator, three-way positioners can be installed that supply or exhaust air pressure to only one side of the diaphragm. Three-way positioners can be mounted on the actuator's yoke leg or can be integrally mounted inside the actuator, as shown in Fig. 5.11.

Diaphragm actuators are produced in several sizes, with a different diaphragm area for each size as well as several range-spring options. Each size has a given range of thrust that is available to overcome process forces, frictional forces, gravitational forces, and the range spring. Therefore, the actuator size has less to do with the process' line size than the service conditions. Whether the valve is used primarily for on–off service or throttling service has some bearing on the actuator size. With diaphragm actuators, the instrument signal can vary

Figure 5.11. Diaphragm actuator with integral three-way positioner. (*Courtesy of Kammer Valves*)

widely to accommodate power considerations. Although 3 to 15 psi (0.2 to 1.0 bar) is considered standard, diaphragm actuators can have signal ranges as high as 3 to 27 psi (0.2 to 1.9 bar) or 6 to 30 psi (0.4 to 2.1 bar). Diaphragm actuators are sized according to the square inches of the diaphragm. For example, a size 125 diaphragm actuator has a diaphragm of 125 square inches (in^2).

The chief advantage of diaphragm actuators is that they are relatively inexpensive to produce and are commonly seen through the entire process industry. Although limited in high-thrust requirements, they are well suited to a good portion of applications in lower-pressure ranges, where thrust requirements are not so demanding. The basic single-acting design and method of operation are simple to understand. Because the positioning signal is also conveniently used to power the actuator, the expense of a positioner and tubing is not necessary. Without a positioner, an involved calibration process and the potential for mechanical difficulties associated with that device are not necessary. The lack of positioner also means that less moving parts, such as a positioner-to-actuator linkage, are involved that may cause potential maintenance problems. When used with linear-motion valves, the entire movement of the actuator stem is transferred directly to the valve's closure element. Because no tight dynamic seals, such as O-rings, are involved with the diaphragm, no breakout force is necessary during positioning, providing immediate and accurate response. Generally, diaphragm actuators are ideal for those applications in which precise positioning and immediate response are important and in which medium to low thrust is acceptable to overcome the process and valve forces.

Several disadvantages of the design should be noted. Because the diaphragm is relatively large, the subsequently large casing may present weight and height problems, especially when mounted on smaller valve sizes. This can cause problems with stress at the connection point between a small valve and an oversized actuator. Because of the restrictions in the elasticity of the diaphragm, its stem travel is limited. Strokes are somewhat short, when compared to other types of actuators. This poses a problem with special severe service trims in which a long stroke is necessary to provide a particular flow characteristic or provide a greater flow capacity through a stack or other trim device. Most diaphragm actuators have strokes of 2 in (5.1 cm) or less, although 4-in (10.2-cm) strokes are possible in some special designs. The largest drawbacks are the thrust and air-pressure restrictions of the diaphragm itself. Because the amount of force produced by the diaphragm actuator is proportional to the size of the actuator, the physical size required for high thrusts is limited by the size of the diaphragm. Most diaphragms are rated for operation in the 20- to 30-psi (1.4- to 2.1-bar) range, therefore limiting the amount of air pressure acting on the diaphragm. For example, a size 125 diaphragm actuator operating with 30 psi (2.1 bar) air pressure can produce a maximum of 3750 lb of thrust (1700 kgf). For that reason, the only way to increase the thrust is to increase the size of the diaphragm, which results in a larger actuator and air chamber. In turn, this larger volume produces slower actuator speed and decreases overall response. The air-pressure limitations of the diaphragm also require the use of air regulators because the air pressure supplied by most plant compressors is between 80 and 125 psi (between 5.5 and 8.6 bar). If diaphragms could handle such high air pressures, the thrust capabilities of the example above would increase dramatically to 10,000 lb (4400 kgf) of thrust. Unfortunately, no diaphragm material has been developed that can provide such strength yet provide the required resilience to move through the full stroke. The thrust limitations of a diaphragm actuator can be overcome by using it with valve designs that can balance the process flow conditions, such as double-seated valves or pressure-balanced trim. Although the cost of such valve bodies may be higher than unbalanced designs, the cost may be negated by the smaller actuator.

Generally, diaphragm actuators—because of the limitations of the diaphragm—do not provide exceptional stiffness and therefore have problems with fluctuations in the process flow. They also experience problems when throttling close to the seat, not having enough power to prevent the closure element from being pulled into the seat. The stiffness value of a diaphragm actuator is usually constant throughout the entire stroke. When the closure element is close to the seat, a sudden change or fluctuation in the process flow can cause the valve to slam shut, causing water-hammer effects.

From a maintenance standpoint, the life of diaphragm actuators is somewhat limited by the life of the diaphragm. If the diaphragm develops even a minor failure, the actuator is inoperable. Since the two casings are bolted together with numerous bolts, disassembly can be somewhat laborious and time consuming. Diaphragm actuators are not field-reversible, because different parts are required for the direct- and reserve-acting designs. Diaphragm actuators have about one-third more parts than other types of pneumatic actuators, which increases their cost somewhat.

Although the diaphragm actuator is the most common pneumatic actuator, the piston cylinder actuator (Fig. 5.12) is gaining widespread acceptance, especially as processes become more advanced and

Figure 5.12. Piston cylinder actuator. (*Courtesy of Valtek International*)

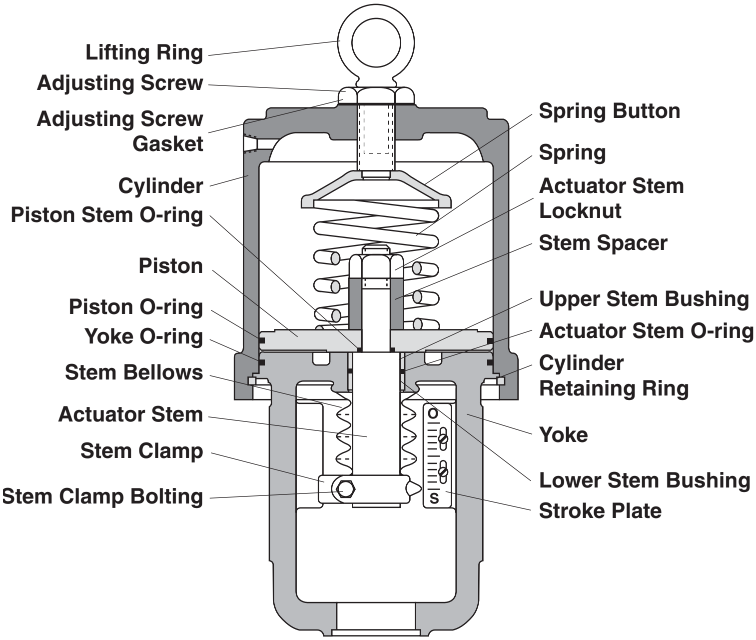


Figure 5.13. Internal view of piston cylinder actuator. (*Courtesy of Valtek International*)

demanding. As shown in Fig. 5.13, the *piston cylinder actuator* uses a sliding sealed plate (called the *piston*) inside a pressure-retaining cylinder to provide double-acting operation. With the double-acting design, air is supplied to both sides of the piston by a positioner. As with all double-acting actuators, a positioner must be used to take the pneumatic or electric signal from the controller and send air to one side of the piston while bleeding the opposite side until the correction position is reached. An opposing range spring is not necessary with the piston cylinder actuator, although a spring may be included inside the cylinder to act as a fail-safe mechanism. More information about the use and operation of positioners is found in Sec. 5.6.

Like diaphragm actuator designs, piston cylinder actuators can be used with either linear or rotary valves. Linear designs are the most efficient since the entire movement of the actuator stem is transferred directly to the valve stem. On the other hand, the rotary design must use some type of linear- to rotary-motion linkage. This can create some hysteresis and dead band because of the lost motion caused by the use of linkages or slotted levers.

The design of the linear cylinder actuator involves a cast yoke, which is used to make the connection to the valve body. It also provides room for the connection between the valve's stem and the actuator stem, attaches the cylinder mechanism to the valve, supports the actuator stem, and allows the installation of the positioner and other accessories. The cylinder can be made from either aluminum (for weight and machining considerations) or steel, based on the application. Fire-sensitive applications prefer the higher melting point of steel over aluminum. The inside of the cylinder is machined to a polished finish to allow for a good seal. The piston itself is a flat disk that is machined nearly to the inside diameter of the cylinder. An O-ring (or similar elastomer seal) fits inside a groove along the sealing edge of the piston. When the O-ring and piston are installed inside the cylinder, the cylinder wall is lubricated to allow a strong, sliding seal. If a fail-safe spring is required, it can be installed either above or below the piston. Unlike the diaphragm actuator that requires a different range spring for different opposing forces, the piston cylinder actuator spring is only needed for fail-safe operation. Therefore, only one heavy-duty spring is needed to cover most applications with the thrust requirements of that actuator size. For extremely high-pressure-drop-applications, a nested spring configuration (one spring inside another) can be used, as shown in Fig. 5.14. Spring compression is applied by the introduction of an adjusting bolt, which compresses the spring to the required return force. Adjusting bolts of different lengths can be used to vary the spring compression. The cylinder is installed above the yoke with either a snap-ring arrangement or bolting.

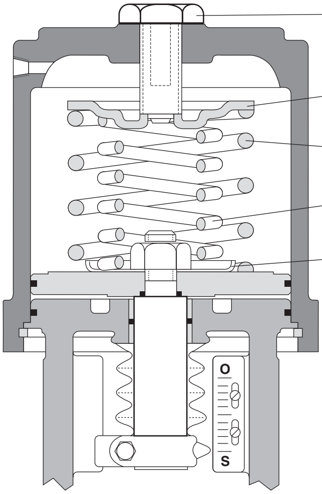


Figure 5.14. Piston cylinder actuator with dual springs. (*Courtesy of Valtek International*)

The actuator stem is attached to the piston and is supported by the top of the yoke with guides. It is sealed from the lower chamber with an O-ring. With piston cylinder actuators, the most common connection between the body and the actuator is a two-piece yoke clamp (Fig. 5.15). This permits a tight connection without larger threads to contend with, which can be a problem with atmospheric corrosion. A clamp is used to prevent the accidental rotation of the actuator stem with the valve stem. The clamp can also be equipped with a pointer to indicate actuator or valve position.

Most rotary designs use some type of linkage to transfer linear motion to rotary action. Figure 5.16 shows one common design in which a splined lever is attached to the valve's shaft and has a pivot point on the actuator stem to minimize hysteresis. Such a design requires a sliding seal to allow for the rocking motion of the piston, which will rock slightly as the actuator stem rotates with the travel of the lever. As shown in Fig. 5.17, another common rotary piston cylinder design uses a slotted lever that intersects a pinned actuator stem. This design avoids the rocking piston and its requirement for a sliding seal, although it does have potential for some slight hysteresis and dead band because of the slotted-lever design. With this design, the heavy-duty return spring is placed in a separate housing, opposite the cylinder.

Piston cylinder actuators are reversible, meaning that the same actuator can be modified for either air-to-close (actuator stem extends) or air-

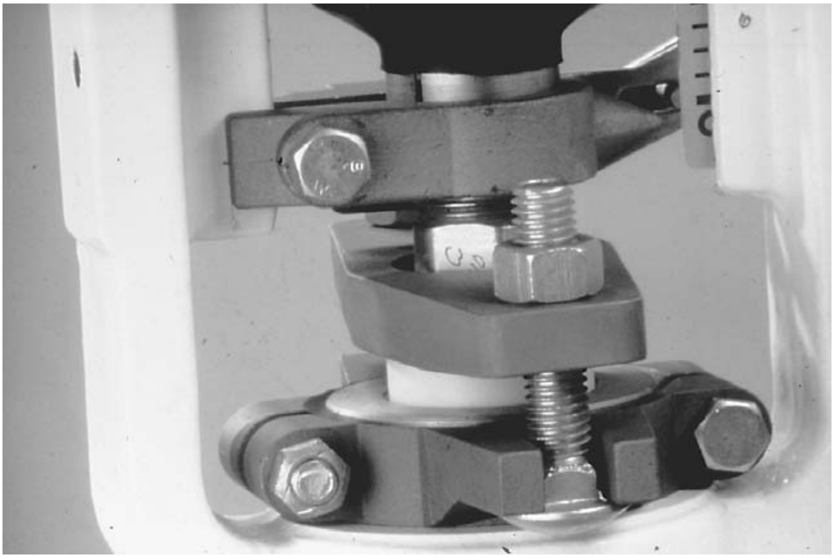


Figure 5.15. Two-piece yoke clamp connection between yoke and bonnet. (*Courtesy of Valtek International*)

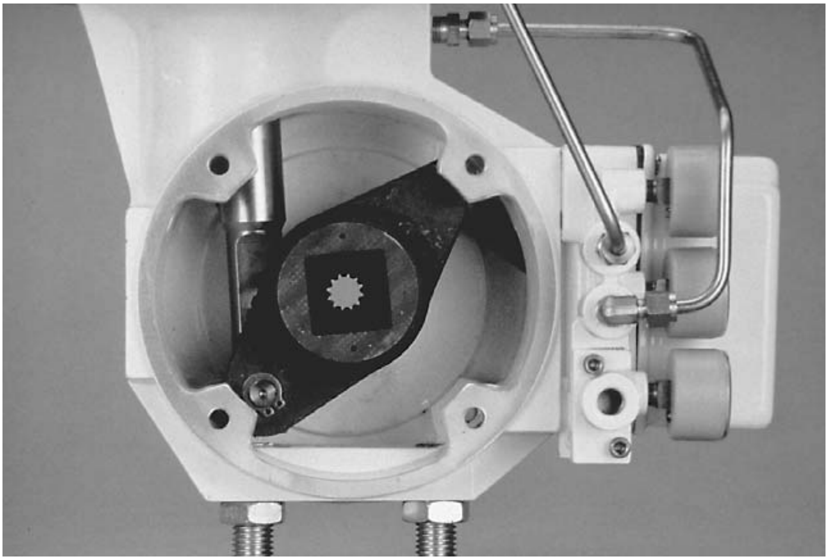


Figure 5.16. Splined clamp connection between rotary actuator and shaft. (*Courtesy of Valtek International*)

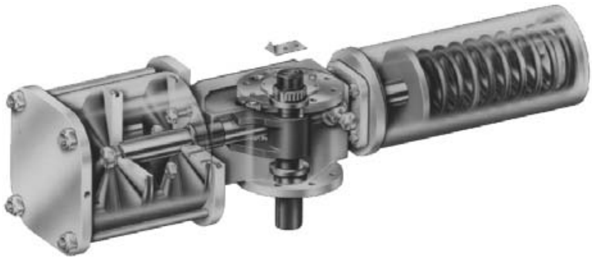


Figure 5.17. Slotted-lever and pinned actuator-stem connection between rotary actuator and shaft. (*Courtesy of Automax, Inc.*)

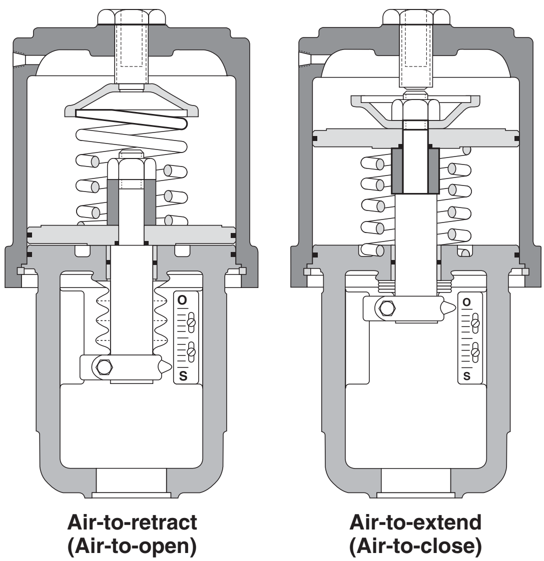


Figure 5.18. Air-to-retract and air-to-extend configurations for piston cylinder actuators. (*Courtesy of Valtek International*)

to-open (actuator stem retracts), as shown in Fig. 5.18. With air-to-close designs, the spring is placed below the piston and is held in place by a ringed groove in the top of the yoke.

The operation of piston cylinder actuators is quite simple. As an air-to-close signal is sent from the controller to the positioner, the positioner sends air to the cylinder's upper chamber above the piston, while the positioner bleeds a comparable amount of air from the lower chamber below the piston. The changing pressures in these two chambers cause the piston to move downward. Subsequently the actuator stem moves downward, as does the valve stem. As the signal changes to "open," the air pressure in the lower chamber builds, while the air pressure in the upper chamber is bled off, allowing the piston to move upward. Therefore the valve's closure element opens. If the signal or power supply is lost, the piston is assisted by the fail-safe spring and moves to its relaxed position. In air-to-close configurations, the relaxed state is with the stem retracted. In air-to-open configurations, the relaxed state is with the stem extended.

The primary advantage of cylinder actuators is the higher thrust capability, size for size, over comparable diaphragm actuators. Because the cylinder actuator with a positioner does not need to use air supply as a signal, the plant's full air-supply pressure can be used to power the actuator. The piston with its sliding O-ring seal is much more capable of handling greater air pressure than the diaphragm. To demonstrate the significance of this difference, a piston cylinder actuator with a piston of 25 in^2 (161 cm^2) used with an 80-psi (5.5-bar) air supply is capable of producing 2000 lb of thrust (910 kgf). Assuming a 6- to 30-psi (0.4- to 2.1-bar) range, a comparable diaphragm actuator would only generate 750 lb (340 kgf) of thrust using the 30-psi (2.1-bar) air supply. A far larger diaphragm actuator would be needed to provide the same thrust requirement as the piston cylinder actuator.

Piston actuators, which have smaller chambers to fill with higher pressures of air, have faster stroking speeds than diaphragm actuators, which must fill larger chambers with lower pressures of air. For example, a size 25 piston cylinder actuator can stroke 1.5 in (3.8 cm) in less than 1 s, while a diaphragm actuator takes over 2 s to stroke the same distance.

Generally, cylinder actuators can be operated with air supplies as high as 150 psi (10.3 bar) or as low as 30 psi (2.1 bar). A side benefit to a piston cylinder actuator handling up to 150-psi (10.3-bar) plant air is that air regulators are not required. For diaphragm actuators such regulators are necessary since they cannot handle plant air normally beyond 40 psi (2.8 bar).

Placing air pressure on both sides of the piston also permits greater actuator stiffness, meaning that the actuator can hold a position without being influenced by fluctuation of the process flow. This is especially important with globe or butterfly valves when the plug or disk is being throttled close to the seat and the "bathtub stopper effect" (Sec. 9.6) can take place. Single-acting actuators have difficulty with the bathtub stopper effect because the range spring (which provides the counterforce) may not be strong enough to prevent it from happening. Stiffness of piston cylinder actuators can be calculated by using the following equation:

$$K = \frac{kPA^2}{v}$$

where K = stiffness
k = ratio of specific heat
P = supply pressure
A^2 = piston area
v = cylinder volume under the piston

To illustrate how drastic the stiffness rates vary between piston cylinder actuators and diaphragm actuators, a comparison can be made using a piston cylinder actuator with a 25-in^2 (161-cm^2) piston, which is typical for a 2-in (DN 50) globe valve. With a supply pressure of 100 psi (6.9 bar) and a 0.75-in (1.9-cm) stroke, the stiffness value at midstroke would be 9333 lb/in (1667 kg/cm). In comparison, a diaphragm actuator with a 46-in^2 diaphragm (296 cm^2), which is required for a 2-in valve, only has a stiffness value of 920 lb/in (164 kg/cm). In addition, as the closure element approaches the closed position with a very close throttling position, the reduced volume in the bottom of the cylinder provides for increased and exceptional stiffness. With the 25-in actuator example used earlier, if the plug in a globe valve is 0.125 in (0.3 cm) away from the seat, the piston is only 0.375 in (1 cm) away from the top of the yoke. That would yield over 18,000 lb/in (3214 kg/cm) of stiffness. For that reason, piston cylinder actuators are preferred when process fluctuations occur or if throttling close to the seat is required by the application.

As a general rule, piston cylinder actuators are much more compact, being smaller in height and weight, than diaphragm actuators—an important consideration with installation, maintenance, and seismic requirements. Of course, the size difference is highly accentuated when larger-diaphragm actuators are needed to generate higher thrusts. A height comparison of comparable actuators is shown in Fig. 5.19.

Another consideration is the length of the stroke. With spring cylinder actuators, the stroke is only limited by the height of the cylinder, permitting longer strokes that diaphragm actuators, which are restricted by the resilience limitations of a diaphragm.

Due to the accuracy associated with the positioner, piston cylinder actuators generally perform better than diaphragm actuators, with virtually no hysteresis, highly accurate signal response, and excellent linearity.

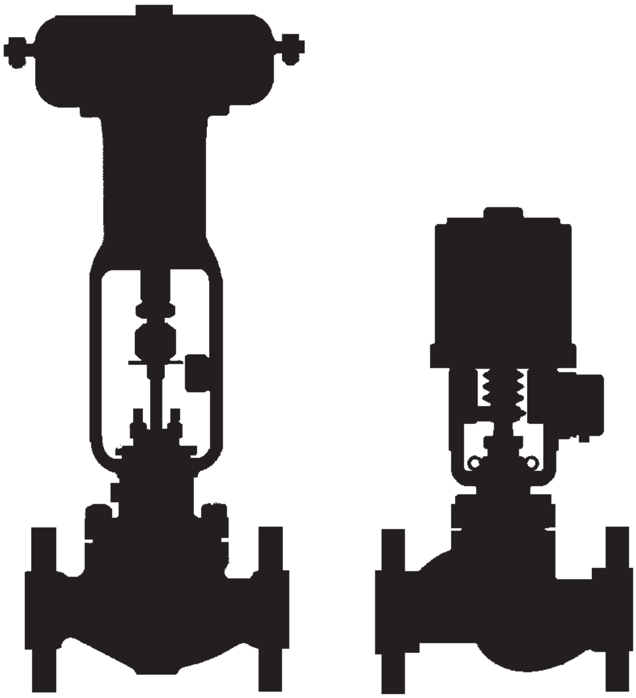


Figure 5.19. Height comparison between comparable diaphragm (left) and piston (right) cylinder actuators. (*Courtesy of Valtek International*)

Piston cylinder actuators have some drawbacks. First, if the actuator remains in a static position for some time, some breakout force may be necessary to move the piston when a signal is eventually sent. When considering the added thrust and response associated with piston cylinder actuators, this breakout torque may not be noticeable. The requirement of a positioner does add expense to the actuator—although with less parts, the actuator itself is less expensive than a diaphragm actuator. A positioner also requires calibration. As discussed in Sec. 5.6, positioners can present problems with exposed linkage and fouled air passages.

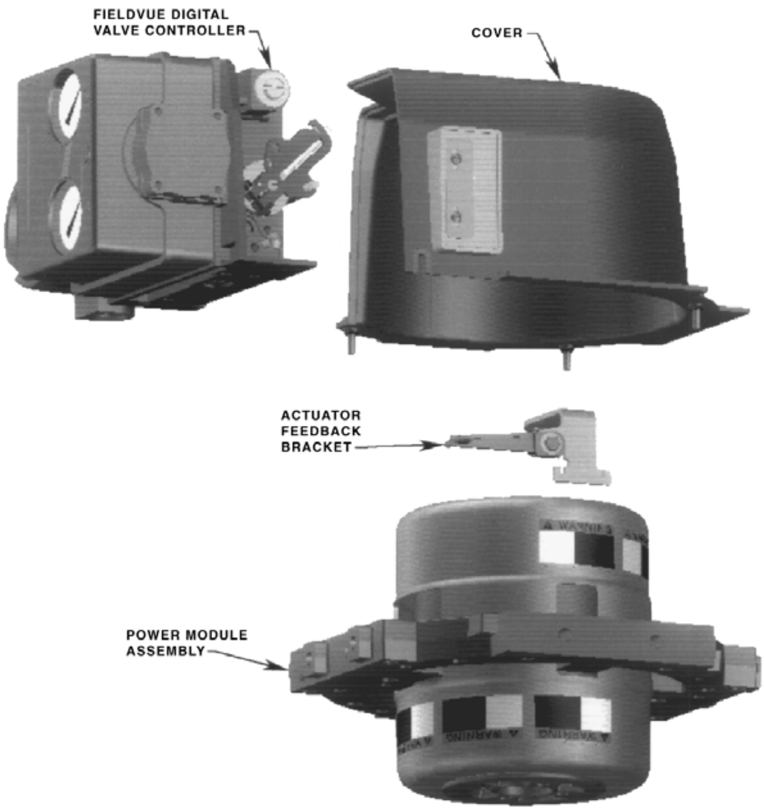
A recent modification of the piston cylinder actuator, a similar design that features a canister assembly and an integral positioner, is shown in Fig. 5.20. Instead of using a dynamic piston, the piston is static and the chambers are dynamic. As shown in Fig. 5.21, the entire

Figure 5.20. Piston cylinder actuator with canister assembly and integral positioner. (*Courtesy of Fisher Controls International, Inc.*)

canister assembly is held in place by the upper and lower casings. As the upper chamber moves, the integral positioner (which is encased in the upper casing) has a follower arm that can receive position feedback by the top of the chamber. Instead of tubing, special air chambers channel air to either the lower or upper chamber.

This design provides a low-profile, compact actuator without the problems associated with external linkage between the actuator and the positioner. With internal air passages, tubing is eliminated—reducing the possibility of damaged tubing or leaking connections. The only disadvantage of this design is that the canister assembly is not designed to be disassembled. The need for a spare part involves the entire assembly, which is far more costly than replacing typical soft goods.

Another commonly applied pneumatic actuator is the *rack-and-pinion actuator*, which is used to effectively transfer the linear motion of piston cylinder actuators to rotary action. Rack-and-pinion actuators are used extensively for actuating quarter-turn valves (ball, plug, and butterfly valves). As shown in Fig. 5.22, two pistons are placed on each end of a one-piece housing, typically extruded aluminum or stainless steel. Each



5.21. Internal view of piston cylinder actuator with canister assembly and integral positioner. (*Courtesy of Fisher Controls International, Inc.*)

piston is connected to a *rack*, a series linear teeth, that move in a linear motion with the piston. In most cases, the rack is an integral part of the piston itself. Sandwiched between the two racks is the *pinion*, which is a shaft equipped with linear teeth. The shaft is connected directly to the valve stem. With direct-acting rack-and-pinion actuators, as air is applied to the two outer pressure chambers, the pistons move toward the inner chamber, exhausted to atmosphere. As shown in Fig. 5.23, when the two pistons move toward each other, the attached racks move in opposite directions, allowing the rack teeth to drive the teeth of the pinion in a counterclockwise rotational manner. As shown in Fig. 5.24, when increasing air pressure is directed to the inner chamber and the outer chambers are exhausted, the pistons move away from each other and the pinion is driven in a clockwise direction.

Figure 5.22. Double-acting rack-and-pinion rotary actuator. (*Courtesy of Automax, Inc.*)

Rack-and-pinion actuators can be equipped with internal springs to allow the actuator to achieve a failure mode (fail-clockwise, fail-counter-clockwise) when the air supply or signal is lost. They are also field-reversible by removing the end caps and rotating the pistons 180°. Rack-and-pinion actuators can also be provided with travel stops to allow for precise adjustment of the open and closed positions of the valve.

Overall, rack-and-pinion actuators are ideal for automating manually operated rotary valves: They are compact, allow for field reversibility, provide adequate torque for most standard operations, and are easy to maintain and to understand.

Another common, inexpensive double-acting actuator is the *vane actuator,* which uses a pie-shaped pressure-retaining housing and a rectangular piston, called the *vane,* to seal between the two pressure chambers (Fig. 5.25). As with rack-and-pinion actuators, vane actuators are commonly used with quarter-turn valve applications.

The housing is divided into two halves and is pie-shaped to allow the vane to move the 90° required for quarter-turn operation. The vane is pinned to the actuator shaft, avoiding excessive hysteresis and dead band. The vane seals the two pressure chambers with an O-ring. Generally the design does not permit the inclusion of a spring. Instead, a pneumatic fail-safe system is often used in place of the spring. The

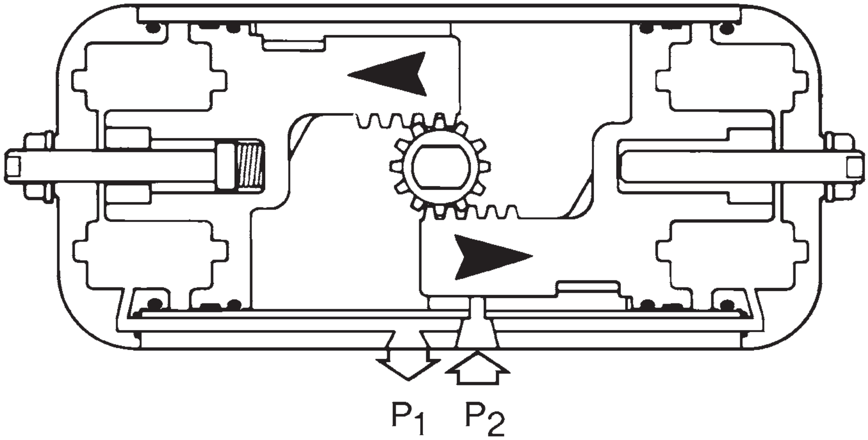


Figure 5.23. Counterclockwise action of rack-and-pinion actuator. P_1 = upstream pressure; P_2 = downstream pressure. (*Courtesy of Automax, Inc.*)

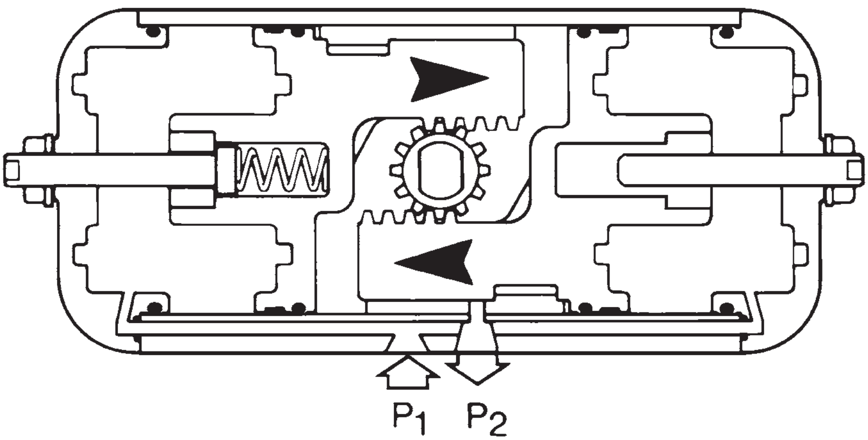


Figure 5.24. Clockwise action of rack-and-pinion actuator. P_1 = upstream pressure; P_2 = downstream pressure. (*Courtesy of Automax, Inc.*)

double-acting design requires the use of a positioner for throttling applications; each pressure chamber has an air connection for increasing or exhausting air pressure.

The operation of the vane actuator can be reversed by simply removing the actuator from the valve and installing it upside down (since both ends of the actuator have universal mounting). Limit-stops can be included on both ends of the housing to limit the motion of the vane.

Figure 5.25. Vane rotary actuator. (*Courtesy of Xomox/Fisher Controls International, Inc.*)

The advantages of the vane actuator are its simple design with few moving parts, no hysteresis, low cost, minimal weight, and compact size. The chief disadvantage of the vane actuator is that it only generates relatively low torque values when compared to other designs; therefore, vane actuators are commonly applied to low-pressure applications. In addition, the two-piece housing with a joint down the middle provides a possible leak path between air chambers.

5.4 Nonpneumatic Actuators

5.4.1 Electric Actuators

Electric motors installed on process valves were one of the first types of actuators used in the process industry. Such electric actuators have been used since the 1920s, although the designs have improved dramatically since those early days, especially in terms of performance, reliability, and size. In basic terms, the electric actuator consists of a reversible electric motor, control box, gearbox, limit switches, and other controls (such as a potentiometer to show valve position).

The chief applications for electric actuators are in the power and nuclear power industries, where high-pressure water systems require smooth, stable, and slow valve stroking.

The main advantages of electric actuators are the high degree of stability and constant thrust available to the user. In general, the thrust capability of the electric actuator is dependent on the size of the electric motor and the gearing involved. The largest electric actuators are capable of producing torque values as high as 500,000 lb (225,000 kgf) of linear thrust. The only other comparable actuator with such thrust capabilities is the electrohydraulic actuator, although the electric actuator is much less costly.

Stiffness is far better with electric actuators, because no compressibility of air is involved with the electric actuator. One additional benefit of an electric actuator is that it always fails in place upon loss of electrical power, whereas a pneumatic actuator requires a complex fail-in-place system. Since fluids (such as air or hydraulics) are not required to power the actuator, leaks and tubing costs are not factors.

The disadvantage of electric actuators is their relative expensive cost when compared to the more commonly applied pneumatic actuators. Also, they are much more complex—involving an electric motor, electrical controls, and a gearbox—therefore much more can go wrong. An electric motor is not conducive to flammable atmospheres unless stringent explosion-proof requirements are met. When high amounts of torque or thrust are required for a particular valve application, an electric actuator can be quite large and heavy, making it more difficult to remove from the valve. Depending upon the gear ratios involved and the pressures involved with the process, an electric actuator can be quite slow, when compared to electrohydraulic actuators or even pneumatic actuators. It can also generate heat, which may be an issue in enclosed spaces. If the torque or limit switches are not set correctly, the force of the actuator can easily destroy the regulating element of the valve.

Based on the thrust requirements, electric actuators are available in compact, self-contained packages (Fig. 5.26), as well as larger units with direct-drive handwheels (Fig. 5.27). As shown in Figs. 5.28 and 5.29, the basic design of the electric actuator consists of the electric motor, the gearbox or gearing, the electrical controls, limit or torque switches, and the positioning device. By design, electric motors are more efficient at their maximum speed; therefore, most electric actuators use some type of mechanical device, such as a hammer blow yoke nut, to engage the load after the motor has achieved its full speed. This is especially important since the largest amount of thrust or torque is

Figure 5.26. Compact electric actuator. (*Courtesy of Kammer Valves*)

Figure 5.27. Electric actuator with direct-drive handwheel. (*Courtesy of Rotork Controls Inc.*)

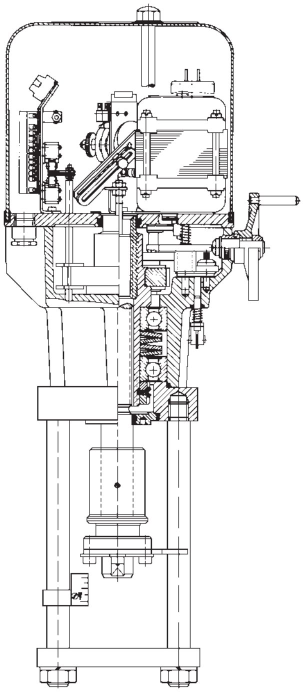


Figure 5.28. Internal view of compact electric actuator. (*Courtesy of Kammer Valves*)

required at the opening or closing of the valve. For the actuator to operate in both directions, the motor must be reversible to open and close the valve. For efficiency reasons, electric motors operate best at high revolutions per minute (1000 to 3600 r/min). Therefore, gearing is used to reduce the stroking speed for use with valves. The gearbox uses worm gearing to make the reduction and is totally encased in an oil bath for maximum life of the gears.

Because of the exceptional stiffness and torque associated with electric actuators, the valve can overstroke if the actuator is not adjusted correctly—and possibly damage or destroy the regulating element or limit the stroke of the valve. To avoid overtravel, limit switches are used to shut off the motor when the open or closed position is reached. Torque switches can also be used to shut off the motor when the torque resistance increases as the closed or open positions are reached. The added benefit of the torque switch is that if an object is caught in the regulating element or if the valve is binding, the actuator will shut off rather than apply thrust to reach the closed position and further damage the valve.

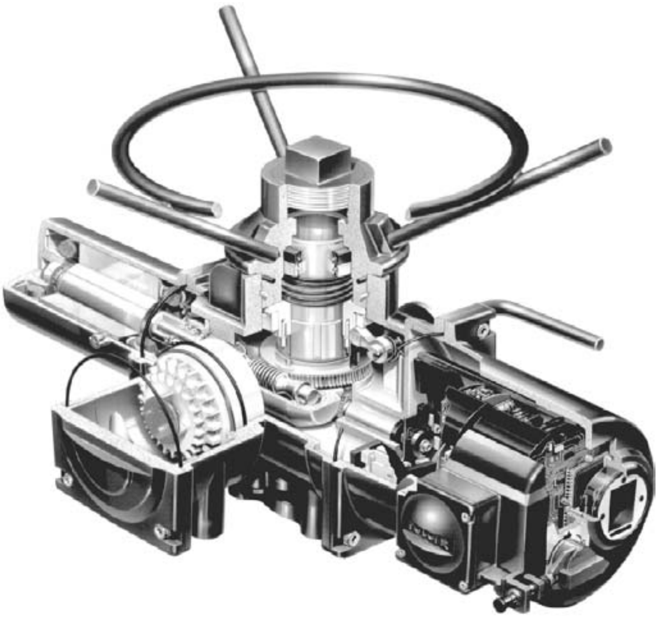


Figure 5.29. Internal view of electric actuator with direct-drive handwheel. (*Courtesy of Rotork Controls Inc.*)

Ideally, torque switches are best used with valves that have floating seats (such as ball or wedge gate valves), while limit switches are best used with valves with fixed seats (such as globe or butterfly valves).

The electrical controls can be accessed on the valve itself or controlled at a remote location using extended electrical lines. Either handlevers or buttons are provided to operate the electric motor. With the handlever, turning the lever clockwise extends the actuator stem, while counterclockwise retracts the stem. Placing the handlever in the middle position shuts off the motor and maintains that particular valve position. With button controls, three buttons are used in the normal configuration: one to extend the actuator stem, one to retract, and another to stop the motor. Red and green lights are used to show the user if the valve is in the open position (usually green) or closed position (usually red). When the motor is in operation, both lights are on.

Electric actuators, in smaller sizes, operate using 110 to 120 V ac, 60-Hz, single-phase power, drawing anywhere between 3 and 30 A. Larger electric actuators use 220 to 240 V, three-phase, 50- or 60-Hz power supply—or 125 or 250 V dc. This may require drawing up to 300 A. Exceptionally large actuators may require even greater voltage (up to 480 V ac).

When manual operation or manual override is needed, most electric actuators allow for the electric motor to be disengaged. A declutchable handwheel can then be used to position the valve manually. Because of the complex electrical and mechanical nature of electric actuators, most calibration adjustments and recommended servicing are made at the manufacturer's factory or an authorized service center.

5.4.2 Hydraulic and Electrohydraulic Actuators

When exceptional stiffness and high thrust are required—as well as fast stroking speeds—hydraulic and electrohydraulic actuators are specified. *Hydraulic actuators* use hydraulic fluid above and below a piston to position the valve. Hydraulic pressure can be supplied by an external plant hydraulic system (Fig. 5.30). Its design is similar to a cylinder actuator, with a cylinder and a piston acting as a divider between the two chambers. Hydraulic actuators do not have a failure spring, so providing a failure action requires a series of tripping systems, which are very complex and require special engineering. On the other hand, an *electrohydraulic actuator* uses a hydraulic actuator—rather than use an external hydraulic system, it has a self-contained

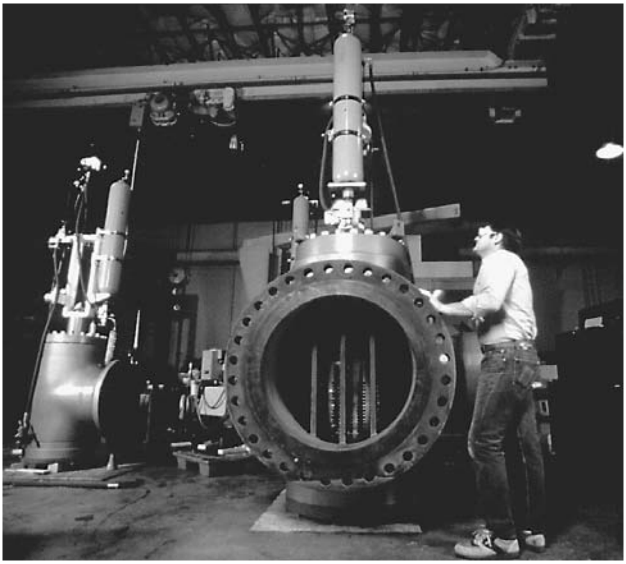


Figure 5.30. Hydraulic actuator mounted on a severe service valve. (*Courtesy of Valtek International*)

hydraulic source that is a physical part of the actuator. An electrical signal feeds to an internal pump, which uses hydraulic fluid from a reservoir to feed hydraulic fluid above or below the piston.

The advantage of using hydraulic and electrohydraulic actuators is that they are exceptionally stiff because of the incompressibility of liquids. This is important with those throttling applications that can be unstable when the regulating element is close to the seat. In some cases, these actuators are used in valves with traditionally poor rangeability, such as butterfly valves. When specially engineered, they can be designed to have exceptionally fast stroking speeds, sometimes closing long strokes in under a second—which makes them ideal for safety management systems. The chief disadvantages of hydraulic and electrohydraulic actuators are that they are expensive, large and bulky, highly complex, and require special engineering.

5.5 Actuator Performance

5.5.1 Performance Nomenclature

A number of technical terms are used to describe the performance capabilities of an actuator.

Hysteresis is a common term used to describe the amount of position error that occurs when the same position is approached from opposite directions. *Repeatability* is similar to hysteresis, although it records the maximum variation of position when the same position is approached from the same direction. Typically hysteresis and repeatability readings can be anywhere between 0.25 and 2.00 percent of the full stroke of the actuator. *Response level* is the maximum amount of input change required to create a change in valve-stem position (in one direction only). Typically response levels can be anywhere between 0.1 to 1.0 percent of full stroke. *Dead band* is a term used to describe the maximum amount of input that is required to create a reversal in the movement of the actuator stem. Typical dead-band measurements can fall between 0.1 and 1.0 percent of the full stroke. *Resolution* describes the smallest change possible in a valve-stem position. Typical resolution is between 0.1 and 1.0 percent full stroke.

Steady-state air consumption applies to actuators with positioners in which the positioner consumes a certain amount of air pressure to maintain a required position. Depending on the positioner design, typical steady-state air consumption can vary anywhere between 0.2 and 0.4 SCFM (standard cubic feet per minute) (between 1.6 and 3.2 cm^3/min) at 60 psi (4.1 bar). *Supply-pressure effect* describes the change of the actuator stem's position for a 10-psi (0.7-bar) pressure change in the supply [for example, if a 50-psi (3.5-bar) supply is increased suddenly to a 60-psi (4.1-bar) supply]. Typical supply-pressure effects can vary anywhere between 0.05 and 0.1 percent of the full stroke of the actuator. *Open-loop gain* is the ratio of the imbalance that occurs when an instrument signal change is made and the actuator stem is locked up. Typical open-loop gains can be anywhere between 550:1 to 300:1 at 60-psi (4.1-bar) supply. *Stroking speed* is defined as the amount of time, in seconds, that an actuator requires to move from the fully retracted to the fully extended position. Stroking speed depends on the length of the stroke, the volume of the pressure chambers, the air supply, and internal resistance of the actuator itself.

Frequency response is a response to a system or device to a constant-amplitude sinusoidal input signal. In other words, it is a measurement of how fast a system can keep up with a changing input signal. When frequency response is calculated, the output amplitude and phase shifts are recorded at a number of frequencies. They are then recorded as a function of input signal frequency. *Independent linearity* is the maximum amount that an actuator stem will deviate from a true straight linear line. Typical linearity can vary anywhere between ±1.0 and ±2.0 percent.

Maximum flow capacity is the volume of air pressure that can flow into an actuator during a particular time period. This is recorded in standard cubic feet per minute (SCFM) or in cubic centimeters per minute.

5.6 Positioners

5.6.1 Introduction to Positioners

By definition, a *positioner* is a device attached to an actuator that receives an electronic or pneumatic signal from a controller and compares that signal to the actuator's position. If the signal and the actuator position differ, the positioner sends the necessary power—usually through compressed air—to move the actuator until the correct position is reached. Positioners are found in one of two designs. *Three-way positioners* (Fig. 5.31) send and exhaust air to only one side of a single-acting actuator that is opposed by a range spring. *Four-way positioners* (Fig. 5.32) send and exhaust air to both sides of the an actuator, which is required for double-acting actuators. A four-way positioner can be used as a three-way positioner by plugging one of the positioner-to-actuator air-supply lines on the positioner itself.

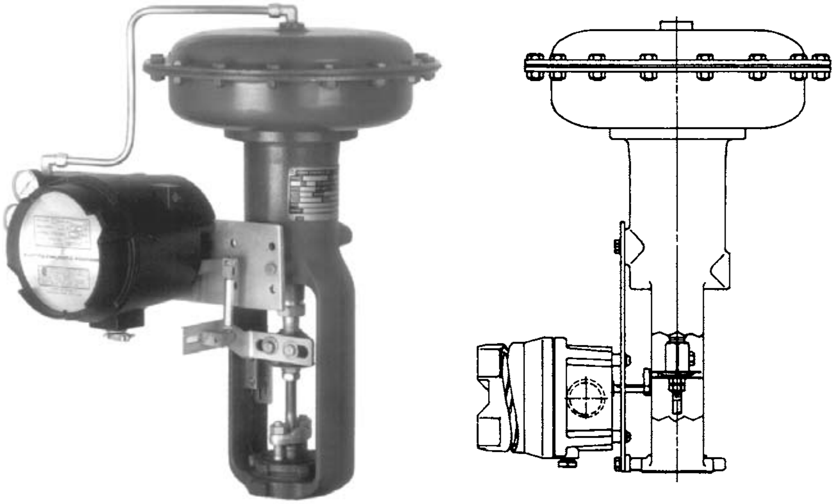


Figure 5.31. Three-way electropneumatic positioner mounted on a diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

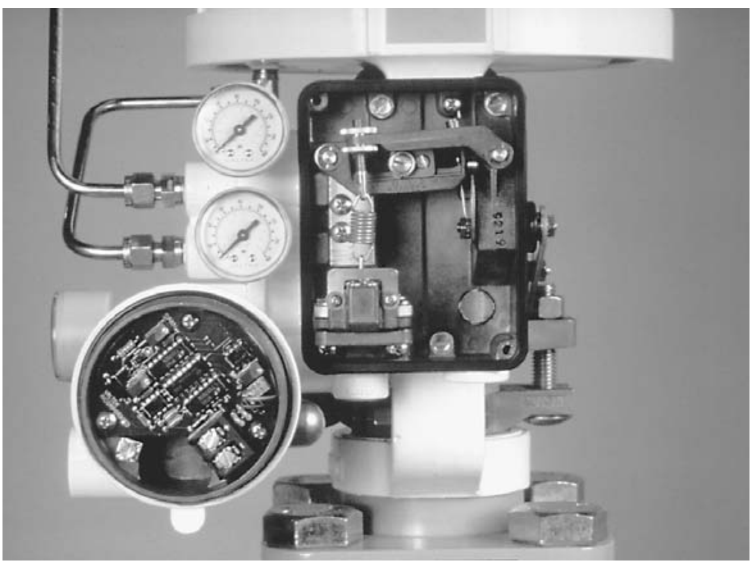


Figure 5.32. Four-way electropneumatic positioner mounted on a piston cylinder actuator (without covers). (*Courtesy of Valtek International*)

When a position signal is sent from a controller, positioners can receive either electronic signals with ranges of 4 to 20 mA and 10 to 50 mA or pneumatic signals with ranges of 3 to 15 psi (0.2 to 1.0 bar) or 6 to 30 psi (0.4 to 2.1 bar). The term *range* is used to show the region between the lower and upper signal limits. A *span* is defined as the difference between the lower and upper limits of the signal. For example, for a range of 3 to 15 psi (0.2 to 1.0 bar), the span is 12 psi (0.8 bar). Internal feedback springs (sometimes called *range springs*) are used inside the positioner to help determine the correct span. *Split range* is the term used to indicate a partial use of a range, such as a 3- to 9-psi (0.2- to 0.6-bar) signal or a 12- to 20-mA signal. In some designs, a split range can be achieved by adjusting a zero or range adjustment on the positioner, while in others a new range spring is required.

As the use of distributive control systems has increased in the past decade, so has the need for electropneumatic (I/P) positioners to handle the milliampere-current control signals. I/P positioners are capable of converting the milliampere signal to an equivalent pneumatic signal, which can then operate the pilot valve of the positioner.

5.6.2 Positioner Operation

Positioning is based on balancing the force between the incoming signal from the controller and the actuator positioner. In other words, the positioner works to balance two forces: first, the force proportional to the incoming instrument signal, and second, the force proportional to the actuator's stem position. As shown in Fig. 5.33, an incoming instrument signal is received by the positioner. If this signal is a milliampere signal, a conversion to a pneumatic signal must take place through the use of a *transducer.* The transducer consists of a feedback loop of a pressure sensor, electromagnetic pressure modulator, and necessary electronics. The pressure modulator consists of a flapper that can open or close an air nozzle. The flapper itself moves when attracted by an electromagnet. As the signal moves the electromagnet, the flapper moves accordingly, creating a proportional air signal to the positioner. The transducer can also include a small air regulator to assist in providing the proper air pressure for the pneumatic signal. If the positioner accepts a pneumatic signal, that signal is sent directly to the positioner.

As the pneumatic signal changes, the air pressure inside the instrument signal capsule also changes, causing a repositioning of the pilot valve. As the pilot valve opens, air is supplied or exhausted to one side of the actuator (three- and four-way positioners). In four-way

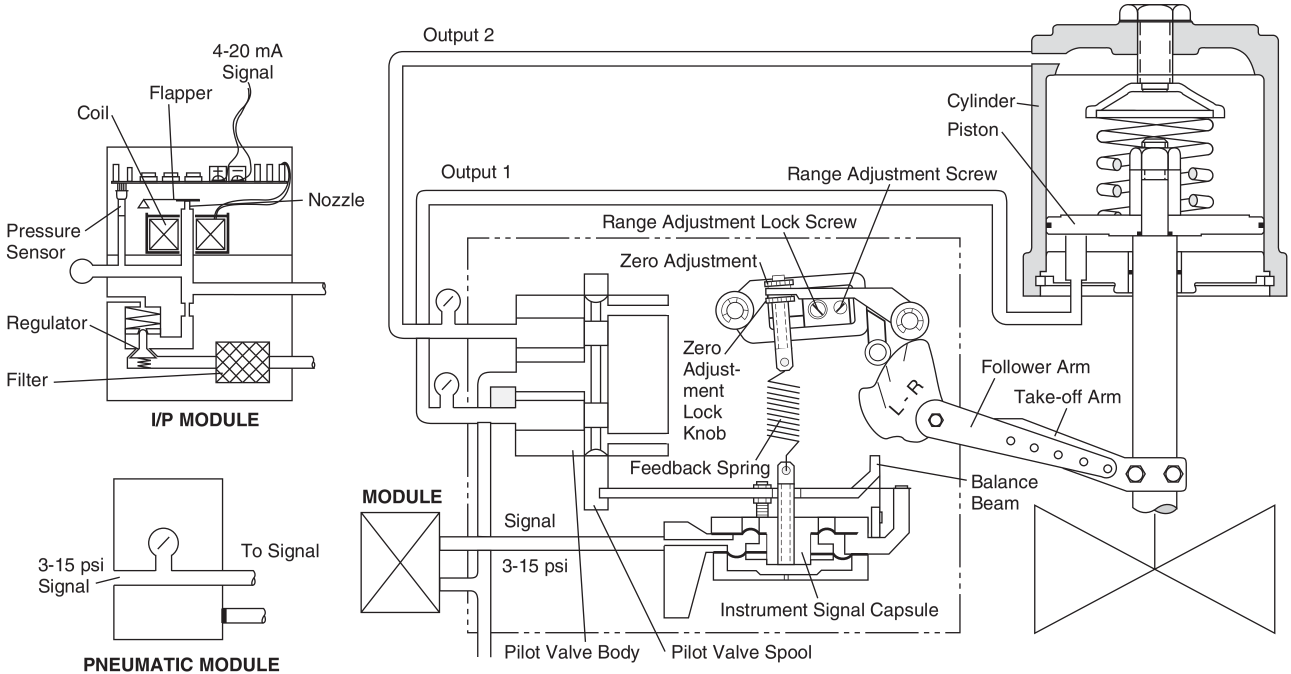


Figure 5.33. Positioner schematic for linear actuator (air-to-retract). (*Courtesy of Valtek International*)

positioners working with double-acting actuators, the opposite action occurs on the opposing side. If air is increased on one side, the other side must exhaust.

The change in air pressures to the upper and lower chambers of the actuator causes the actuator stem to move either upward or downward. The motion of the actuator stem is transmitted to the positioner through some type of internal or external linkage or lever. As this feedback motion is received by the positioner, the stretch and force of the feedback spring are increased or decreased, which changes the counterforce to the instrument signal capsule. At this point, when the correct actuator position is achieved, the instrument signal capsule and pilot valve return to their state of equilibrium and the air flow to the actuator discontinues.

With valves that only have inherent flow characteristics, such as a butterfly valve, a characterizable cam (Fig. 5.34) can be used with the positioner to provide a modified flow characteristic.

5.6.3 Positioner Calibration

Positioners normally come from the factory calibrated to the requirements of the actuator and valve application; however, shipping and handling may cause the calibration to shift. Prior to service, the positioner should be connected to the signal and supply lines and should then be operated. If significant inaccuracy occurs, the positioner calibration should be examined. The two most common adjustments with positioners are the zero and the span. The zero adjustment is used to vary the point where the actuator begins its stroke, normally 3 psi (0.2 bar) or 4 mA for most common applications. After the zero has been calibrated, the span adjustment is used to increase or decrease the span from the zero point, normally 12 psi (0.8 bar) for a 3- to 15-psi (0.2- to 1.0-bar) pneumatic signal or 16 mA for a 4- to 20-mA electronic signal. Some span adjustments allow for certain split ranges without changing the feedback spring. For example, a 3- to 15-psi (0.2- to 1.0-bar) feedback spring may allow the span to be adjusted to a 3- to 9-psi (0.2- to 0.6-bar) or a 9- to 15-psi (0.6- to 1.0-bar) split range. After the span adjustment has been made, the user should return to the zero point to ensure that it stayed true during the span adjustment. Locking nuts or other locking devices are installed to prevent the calibration from shifting during service.

The zero and span adjustments, as well as a number of split ranges available, depend on the type of the feedback spring being used. Significant range changes, such as changing from a 3- to 15-psi (0.2- to

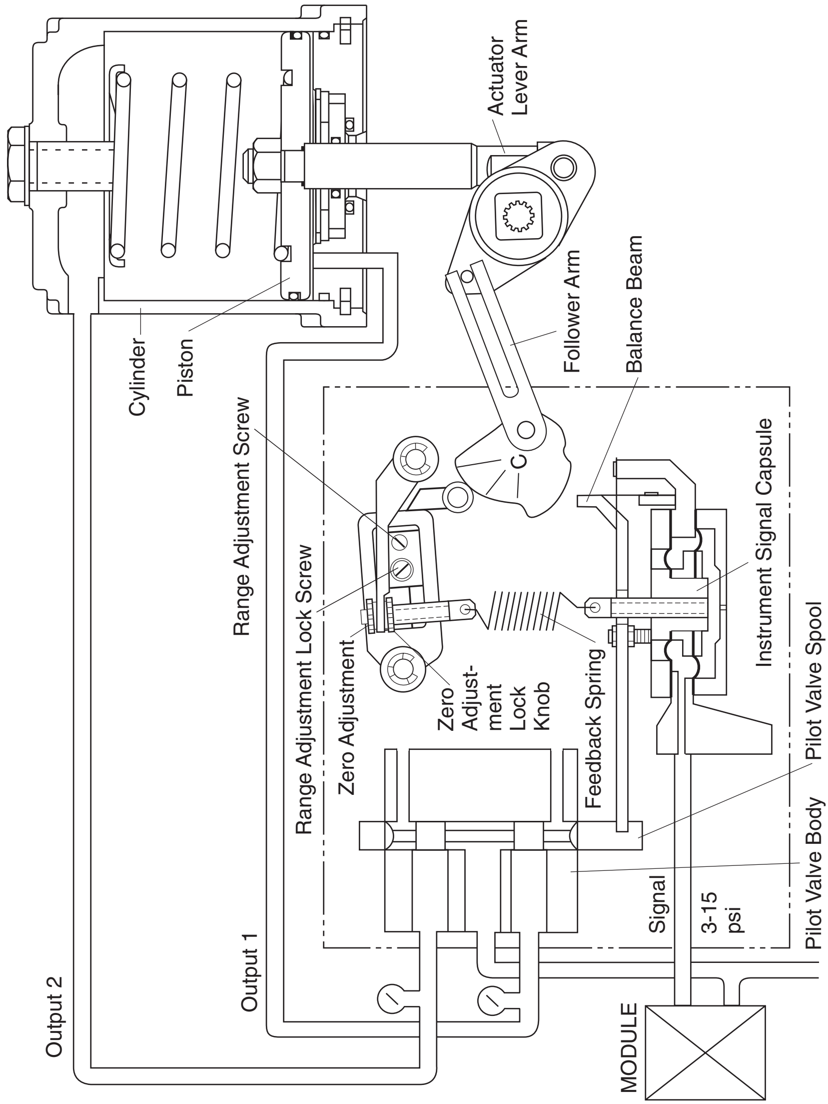


Figure 5.34. Positioner schematic for rotary actuator (air-to-retract). (*Courtesy of Valtek International*)

1.0-bar) range to a 6- to 30-psi (0.4- to 2.1-bar) range would require a new feedback spring.

5.7 Auxiliary Handwheels

5.7.1 Introduction to Auxiliary Handwheels

Occasionally manual operation of an actuated valve is preferred or required; therefore, an *auxiliary handwheel* is attached to the actuator to allow for manual operation of the actuated valve in case of an emergency or when a major power interruption or failure occurs. Not only do auxiliary handwheels allow for manual operations, but some designs can be set in a position so that the handwheel acts as a stop to limit the stroke of the valve.

If an auxiliary handwheel is used while the actuator is still under signal, a three-way bypass valve is installed before the actuator or positioner to shut off the air supply and bleed or neutralize the pressure chamber(s). To prevent accidental or intentional manual operation, some manufacturers provide a locking bar that can be placed around a leg of the handwheel and locked. If this feature is not provided, a simple chain and lock can prevent movement of the handwheel.

5.7.2 Auxiliary-Handwheel Designs

Designs of auxiliary handwheels vary widely. Designs are sometimes based upon the linear or rotary motion of the actuator and/or valve. Some are an integral part of the actuator, while others are an addition to the existing actuator design, following minor modification for attachment. Auxiliary handwheels can be mounted above the actuator (called *top-mounted handwheels*) or on the side of the actuator (called *side-mounted handwheels*).

The most common auxiliary-handwheel design for linear actuators is the *continuously connected handwheel,* which is an assembly attached to the actuator stem with a neutral range that accommodates the full stroke of the actuator without interference from the handwheel. When the handwheel is turned, the handwheel nut (or similar device) moves out of a neutral range and engages either an upper or lower stop. As the handwheel continues to turn, the handwheel nut pushes against the stop, causing the actuator stem to move in that direction. The advantage of the continuously connected design is that it does not require a declutching mechanism to engage or disengage the handwheel in order to operate

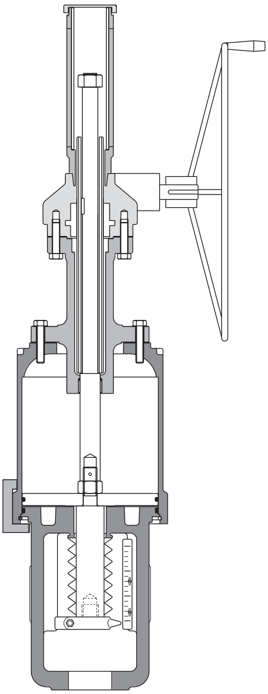


Figure 5.35. Top-mounted continuously connected handwheel mounted on a linear actuator. (*Courtesy of Valtek International*)

the actuator. In addition, when the handwheel is left in a non-neutral position, it can act as a limit-stop for that direction. Continuously connected handwheels that are integral to the actuator can be either top- or side-mounted designs (Figs. 5.35 and 5.36).

Side-mounted continuously connected handwheels can also be designed as a separate unit, which is then added to an existing actuator with slight modifications (Figs. 5.37 and 5.38), such as a special yoke. The attachment of the handwheel to the actuator stem is made external to the cylinder or diaphragm case. Therefore, the chief advantage of this design is that the handwheel can be used to lock the stem position, allowing for disassembly of the cylinder or diaphragm casing for maintenance while the valve remains in operation.

Another common auxiliary-handwheel design for linear actuators is the *push-only handwheel,* which is commonly seen with both diaphragm and piston cylinder actuators (Figs. 5.39 and 5.40). This design is top-

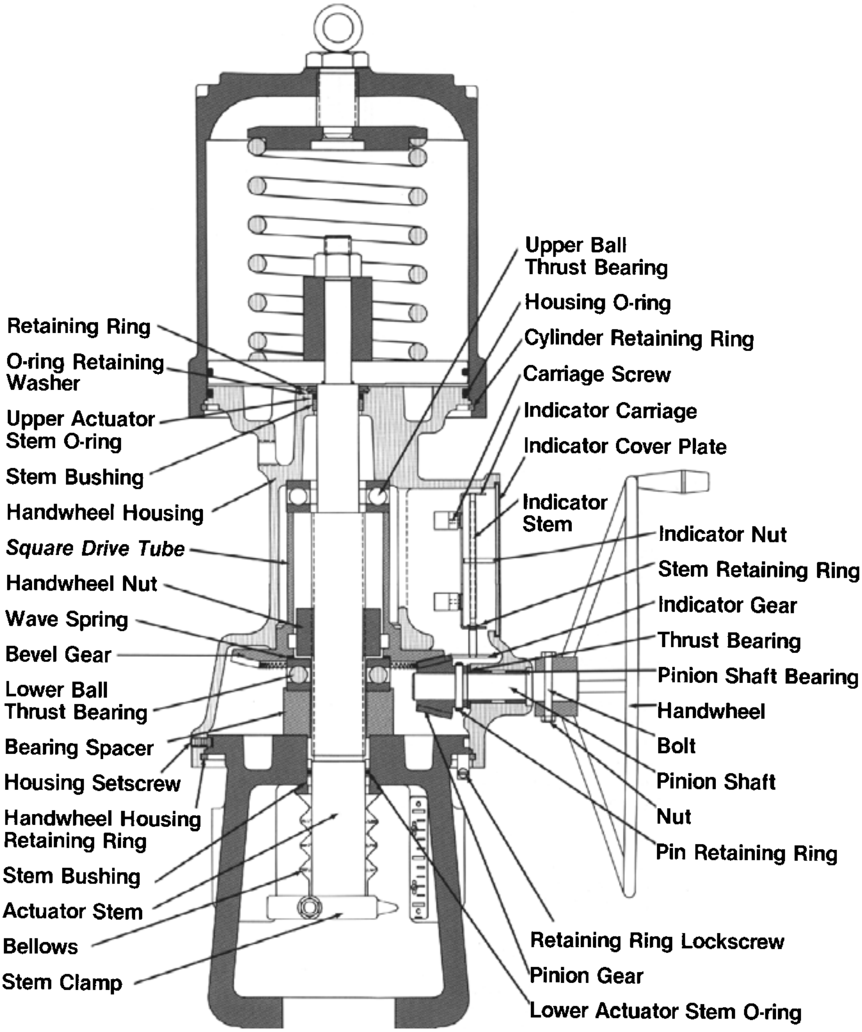


Figure 5.36. Side-mounted continuously connected handwheel mounted on a linear actuator. (*Courtesy of Valtek International*)

mounted and very simple in concept. When the handwheel is turned, the handwheel stem—which is threaded to the top of the actuator—lowers until the handwheel stem makes contact with the piston or diaphragm plate and pushes it until the valve is closed or reaches a midstroke point. The push-only design requires a spring on the opposite

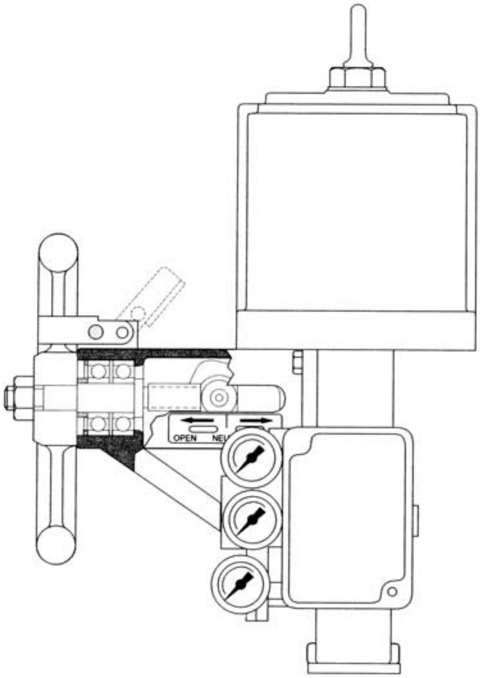


Figure 5.37. Internal view of auxiliary side-mounted handwheel mounted on a piston cylinder actuator. (*Courtesy of Valtek International*)

Figure 5.38. Auxiliary side-mounted handwheel mounted on a diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

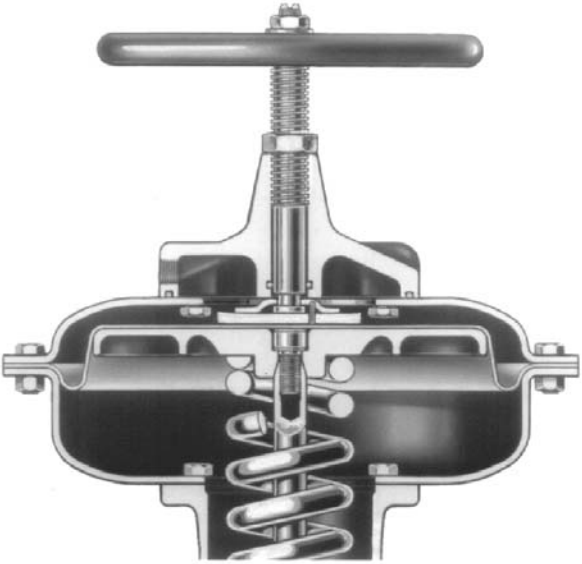


Figure 5.39. Top-mounted handwheel mounted on a direct-acting diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

side of the piston or diaphragm to ensure a counterforce. Not only can the handwheel be used to close or throttle the valve, but it can also be used as an upper limit-stop. A modified design is available for reverse-acting diaphragm actuators (Fig. 5.41).

Rotary-motion valves can also be equipped with auxiliary handwheels (Fig. 5.42), although the rotation of the shaft does not normally permit the continuously connected design. Instead, a declutchable handwheel is used that allows the user to engage or disengage the handwheel from making a positive connection with the shaft. The main problem with the declutchable handwheel is that forces on the handwheel during operation make it difficult to disengage. Also, the user must be careful to remember to disengage the auxiliary handwheel after use, since automatic operation of the actuator and valve will turn the handwheel, creating potential safety and eventual maintenance problems.

5.8 External Failure Systems

5.8.1 Introduction to External Failure Systems

In some situations, the conditions of a service are greater than the capability of an actuator's fail-safe spring. In other applications, an

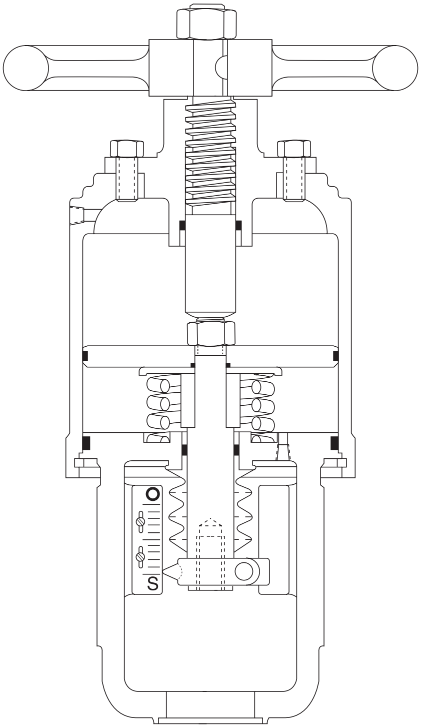


Figure 5.40. Top-mounted push-only handwheel mounted on a piston cylinder actuator. (*Courtesy of Valtek International*)

actuator with a heavy-duty spring may not be practical, either mechanically or economically. In these cases, an external failure system (called an *air spring*) may be added to a pneumatic actuator. An air spring is a self-contained, pressurized system that has enough pneumatic power to force the closure element to move to a particular position when the actuator's power supply is interrupted. In most cases, this failure action is to close the valve, although some applications exist that require a fail-open action. The volume of air required for this action can sometimes be provided by the actuator, or in other cases, by an external volume tank.

Occasionally the design of the valve will permit a smaller air spring. For example, with globe valves, a flow-over-the-plug design allows the plug to remain in the seated position because of the process forces; therefore the air spring needs to generate only enough force to move

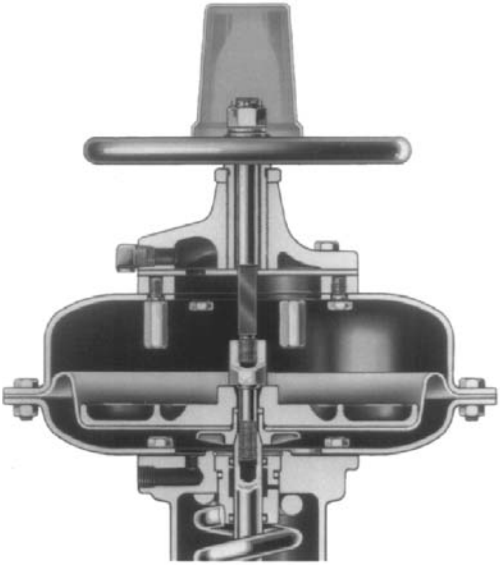


Figure 5.41. Top-mounted handwheel mounted on a reverse-acting diaphragm actuator. (*Courtesy of Fisher Controls International, Inc.*)

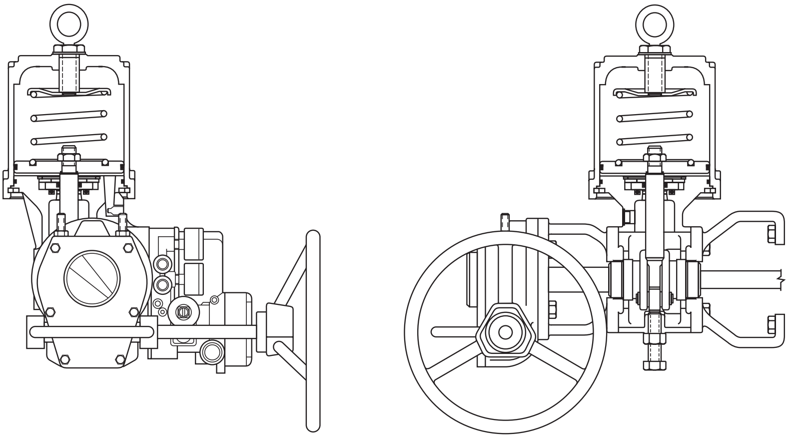


Figure 5.42. Auxiliary rotary declutchable handwheel mounted on a rotary actuator (two views). (*Courtesy of Valtek International*)

the plug to the seated position. If the valve is a flow-under-the-plug design, the air spring must not only have the capability of seating the valve, but also maintaining that position, which may require a larger volume of air and larger external volume tanks. Obviously, if the air spring is designed to open the valve upon failure, a flow-under-the-plug design would help that situation. The point to remember is that sometimes modifying the design of the valve itself can sometimes overcome the need for a huge external failure system.

Occasionally, the application will require that the valve remain in its last position upon loss of power, which requires a different failure system configuration. In that case, the design of the valve has no bearing on the size of the failure system, because the system must be able to handle any throttling position between full-open and full-closed.

5.8.2 Air Springs Using Cylinder Volume

For applications where the service conditions are moderate in nature, yet the failure spring cannot overcome the process, an air spring can be applied, using the air volume from the actuator. This system (air spring using cylinder volume) requires the use of a positioner. As shown in the two schematics for fail-closed and fail-open in Figs. 5.43 and 5.44, the air spring uses a three-way switching valve and an airset. The positioner acts as a three-way positioner, providing air to only one side of the actuator. The airset is used to supply a constant air pressure on the opposite side of the actuator. It is preset at the factory to provide the necessary pressure to overcome the unbalanced forces for that particular application while still allowing the actuator to stroke normally. The three-way switching valve is used to monitor the air supply and is preset at a level close to the expected air supply—yet low enough to avoid problems with normal swings of the supply pressure. When the air supply fails or decreases below a certain preset point, the constant-pressure side of the actuator drives the actuator to its failure position. When the air supply is restored to normal levels, the three-way switching valve opens to allow normal operation of the actuator.

When air springs using cylinder volume are required, the set pressure must be calculated, using the following equation:

$$P_{A1}V_{C1} = P_{A2}V_{C2}$$

where P_{A1} = initial air pressure (absolute)
P_{A2} = final air pressure (absolute)

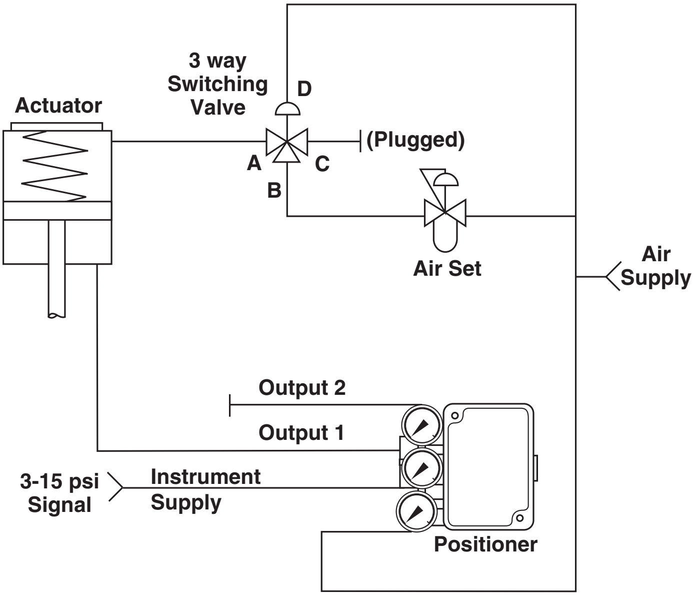


Figure 5.43. Signal-to-open (fail-closed) air spring using cylinder volume schematic. (*Courtesy of Valtek International*)

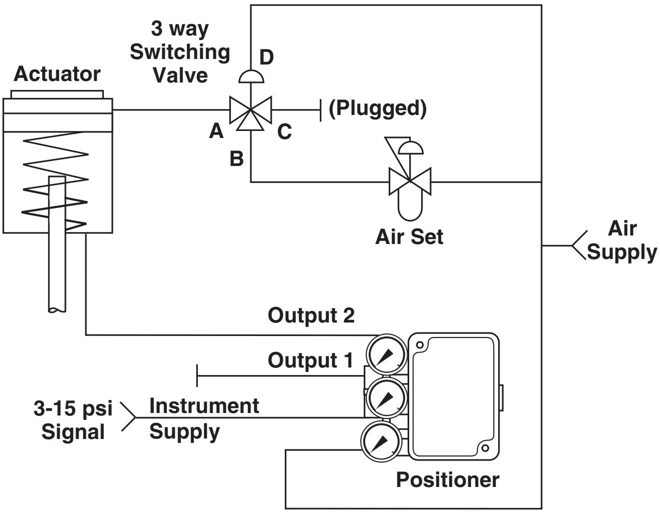


Figure 5.44. Signal-to-close (fail-open) air spring using cylinder volume schematic. (*Courtesy of Valtek International*)

V_{C1} = initial volume of the actuator's pressure chamber
V_{C2} = final volume of the actuator's pressure chamber

The user must then evaluate the worst-case scenario for the required actuator force (F_A) and the actuator's piston or diaphragm area (A), which can be obtained from the manufacturer. After the force and the area are known, the following equation is used to determine the final air pressure required in the actuator (P_{A2}) for the proper failure operation:

$$P_{A2} = \frac{F_A}{A} + 14.7$$

where F_A = required actuator force
A = area of the piston or diaphragm (square inches)

To determine the switching valve setpoint (also known as the initial actuator pressure), the following equation should be used:

$$P_{SVS} = \frac{P_{A2}V_M}{V_M - AS} - 14.7$$

where P_{SVS} = switching valve setpoint (psig)
V_M = maximum volume of the actuator side that requires air to move actuator to failed position (in^3)
S = length of valve stroke (inches)

If the switching valve setpoint (P_{SVS}) exceeds 80 percent of the air supply pressure, then the air volume of the actuator is not capable of handling the failure mode and an external volume tank must be used.

5.8.3 Air Springs Using a Volume Tank

When the air volume inside an actuator is not large enough to drive the actuator to its failure position, an external volume tank is provided with the valve to supply the necessary air volume. The typical air spring using a volume tank system involves an external volume tank, a three-way switching valve, two pilot-operated three-way lock-up valves, and a check valve. A four-way positioner is necessary for this arrangement, which acts to supply air to both sides of the actuator. The purpose of the check valve is to maintain the air pressure inside the volume tank if the air supply should fail.

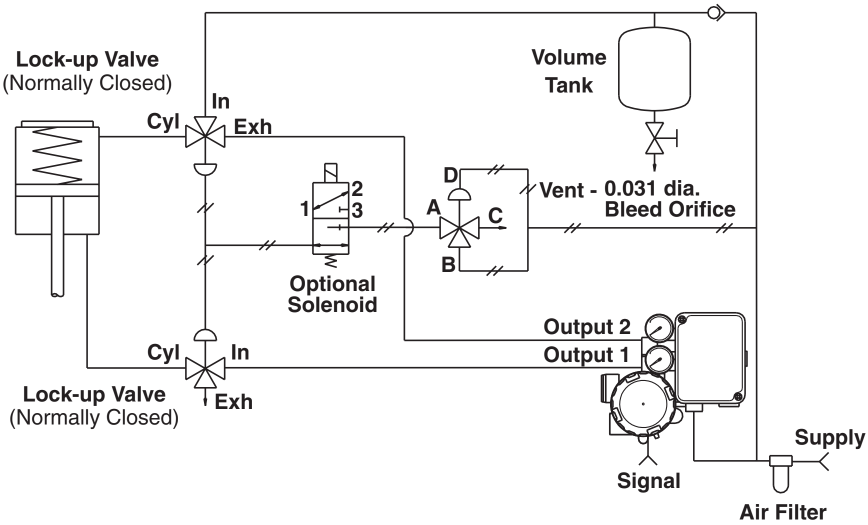


Figure 5.45. Signal-to-open (fail-closed) external volume tank schematic. (*Courtesy of Valtek International*)

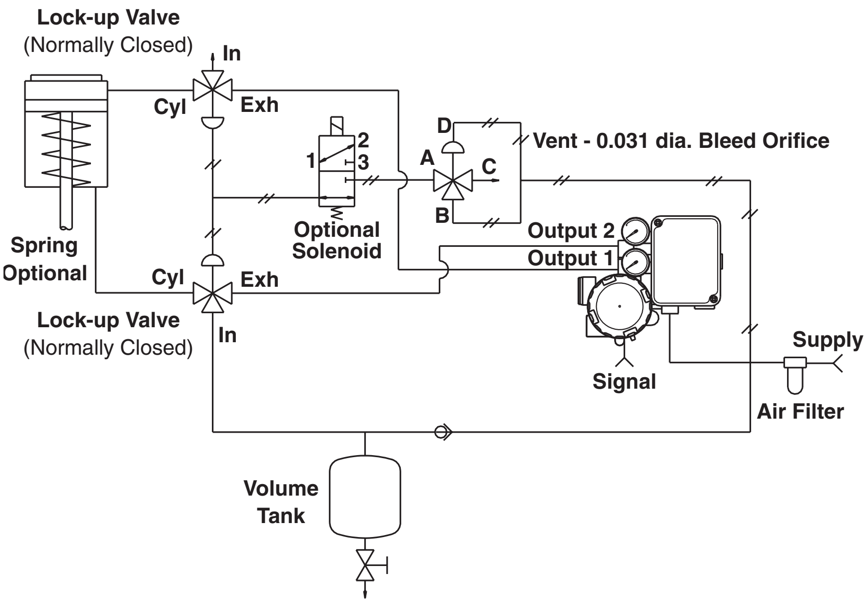


Figure 5.46. Signal-to-close (fail-open) external volume tank schematic. (*Courtesy of Valtek International*)

As shown in the two schematics for fail-closed and fail-open cases in Figs. 5.45 and 5.46, the three-way switching valve monitors the air supply and is preset to a level close to the expected air supply yet low enough to avoid problems with normal swings of the supply pressure. During normal operation, the lock-up valves allow air to flow normally between the positioner and the actuator. When the air supply decreases or falls below the preset value, the pressure from the pilot of the three-way switching valve causes the two lock-up valves to be released. One lock-up valve channels air from the volume tank to one side of the actuator, while the other lock-up valve exhausts the other side of the actuator to atmosphere. Air from the volume tank drives the actuator to its failure position. Unless air leakage is occurring through the tubing, connections, lock-up valve, or check valve between the volume tank and the actuator, the actuator should maintain its position indefinitely. The seal between the two sides of the actuator must also be leak-free.

If the tank volume must be calculated, the following equation should be used:

$$P_{A1}V_{T1} = P_{A2}V_{T2}$$

where V_{T1} = initial volume of the external volume tank
V_{T2} = final volume of the external volume tank

The user must then evaluate the worst-case scenario for the required actuator force (F_A), and the actuator's piston or diaphragm area (A), which can be obtained from the manufacturer. After the force and the area are known, the following equation should be used to determine the final air pressure required in the actuator (P_{A2}) for the proper failure operation:

$$P_{A2} = \frac{F_A}{A} + 14.7$$

To determine the switching valve setpoint (also known as the initial actuator pressure), the following equation should be used:

$$P_{SVS} = \frac{P_{A2}V_M}{V_M - AS} - 14.7$$

If the initial pressure exceeds 80 percent of the air supply pressure, an external volume tank must be used. The following calculations help determine the correct size of volume tank.

Fail-closed actuators:

$$V_T = \frac{P_{A2}V_M}{P_{SVS} + 14.7 - P_{A2}}$$

Fail-open actuators:

$$V_T = \frac{P_{A2}V_M A}{P_{SVS} + 14.7 - P_{A2}}$$

where V_T = volume of the external volume tank (cubic inches)

5.8.4 Lock-Up Systems

Some applications require that the valve remain in place on loss of power supply. In these situations, a *lock-up system* is used. As shown in Fig. 5.47, the typical lock-up system requires a three-way switching valve, two pilot-operated three-way lock-up valves, and a four-way positioner. The three-way switching valve monitors the air supply and is preset to a level close to the expected air supply yet low enough to avoid problems with normal swings of the supply pressure. During normal operation, the lock-up valves allow air to flow normally between the positioner and the actuator.

When the air supply decreases or falls below the preset value, the pilot pressure from the three-way switching valve to the lock-up valves is released, causing both lock-up valves to close. This traps the existing air pressure on both sides of the actuator. The exhaust ports of the two lock-up valves must be plugged; otherwise, the existing air to the actuator bleeds out, creating an unstable condition.

5.9 Common Accessories

5.9.1 Introduction to Accessories

Some special actuation systems or actuators require fast stroking speeds, signal conversions from one medium to another, position transmissions, etc. In these applications, *accessories* are included with the actuator to help perform these special functions. Ideally, accessories are mounted directly onto the valve to ensure that the user is aware of the location of the device—although sometimes the accessory is not mounted directly onto the valve and the user must determine the location of the device.

Accessories may be produced directly by the valve manufacturer; however, in most cases they are produced by a separate manufacturer and purchased by the valve manufacturer. Rather than recreate the

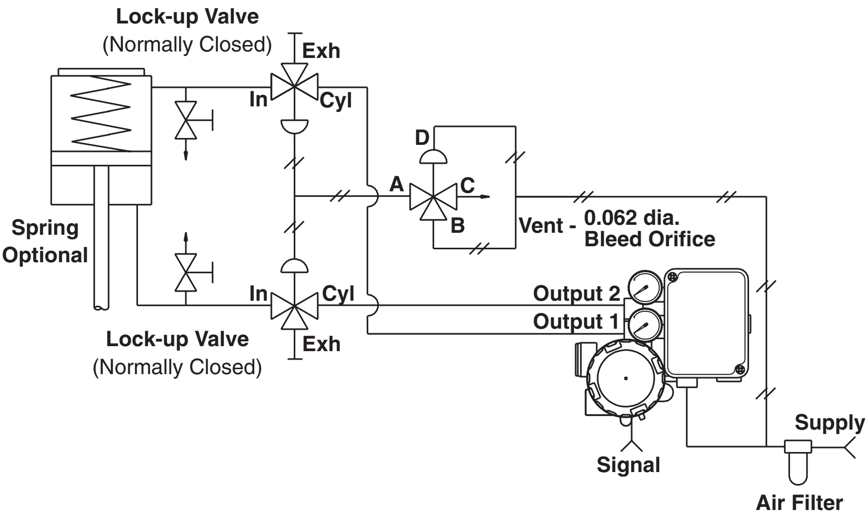


Figure 5.47. Signal-to-open, fail-in-place lock-up system schematic. (*Courtesy of Valtek International*)

original vendor instructions, valve manufacturers normally include them with the valve shipment. These instructions are either attached to the accessory or included with the valve's or actuator's instructions. Keeping this literature for both the valve or actuator and accessories is important, since it details installation and servicing instructions. If instructions about the accessory are not included in the shipment, the valve manufacturer should be contacted.

5.9.2 Filters

One of the most basic accessories for actuators, whether pneumatic or hydraulic, is the filter. The *filter* is designed to screen the power supply medium of impurities or other foreign fluids or objects that may contaminate an actuation system, positioner, or other accessory. As shown in Fig. 5.48, filters are installed between the source of the power supply and the actuator or positioner. Generally, the filter is mounted immediately upstream from the accessory to ensure that the fluid is screened just prior to entering the actuator or positioner. Most are nipple- or bracket-mounted to the actuator or positioner. Filters have either a filter cartridge that has minute openings or a series of screens (screen openings are typically 5 μm in diameter). These filters or

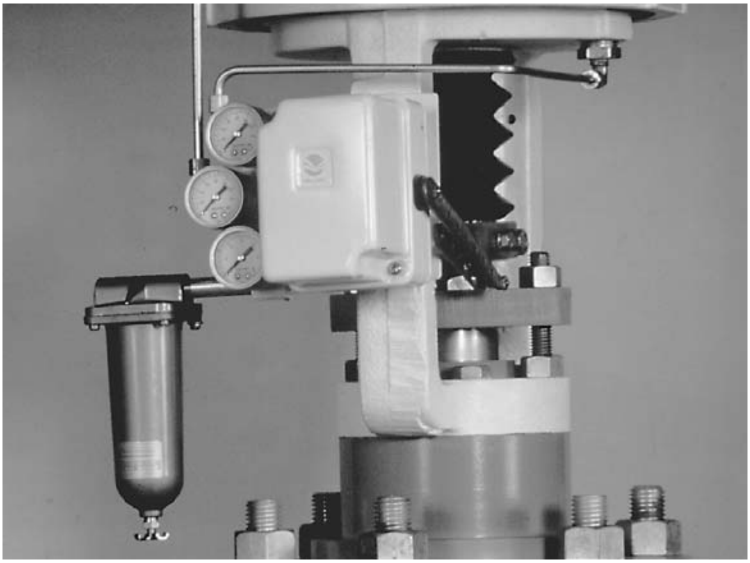


Figure 5.48. Air filter installed before a pneumatic positioner. (*Courtesy of Valtek International*)

screens trap any particles of a larger diameter that can clog the inside small passages of a positioner, foul a metal moving part (such as a piston), or damage an elastomer (such as an actuator stem O-ring).

Because compressed air, especially in humid environments, has a tendency to produce water condensation, air filters have a drip well and a drain valve to allow for draining of any water. Water can foul the passageways in a positioner or cause bacterial growth that can lead to erratic performance. In single-acting valves without an air filter, the pressure chamber can fill with water, causing slow operation or eventually no actuation at all. Through the air pressure of the system itself, the drain can also be used to remove oil and large particulates, which may be present in the air line.

5.9.3 Pressure Regulators

A *pressure regulator* (also known as an *airset*) is used to regulate or limit the air supply to the actuator. A typical pressure regulator is shown in Fig. 5.49. While many plants provide air pressures between 60 and 80 psi, some actuators cannot operate at such pressures without an internal failure. As discussed in Sec. 5.3, single-acting actuators are limited to the lower range of air pressure (usually limited to 40 psi or 2.8 bar) and require the installation of pressure regulators.

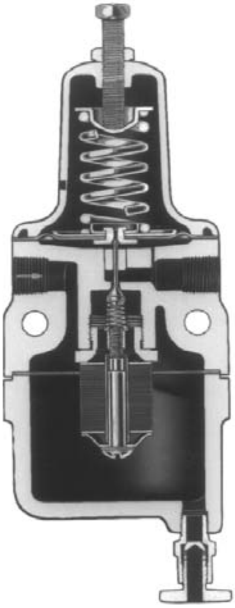


Figure 5.49. Pressure regulator, including air filter and moisture trap. (*Courtesy of Fisher Controls International, Inc.*)

A common problem found in plants that use both single- and double-acting actuators is that some technicians, as a routine procedure, install pressure regulators on all valves regardless of the style—thereby limiting the pressure to all actuators. However, some actuators, such as piston cylinder actuators, actually operate better at higher pressures, providing greater thrust, faster stroking speeds, better stiffness, etc. In addition, placing a pressure regulator on an actuator unnecessarily can lead to possible misadjustments or add one more device that can possibly fail. Manufacturers commonly provide a sticker or tag on the actuator, notifying the user as to the pressure limits of the actuator. The general rule is to install pressure regulators only on those actuators that can only perform with lower air pressures.

5.9.4 Limit Switches

When an electrical signal must be sent indicating an open, closed, or midstroke position of an actuator or valve, an electrical switching device—called a *limit switch*—is used. Limit switches are normally used to sound alarms or operate signal lights, electric relays, or small solenoid valves. A typical signal-at-open and signal-at-closed limit-switch design is shown in Fig. 5.50, while a cammed limit switch is shown in Fig. 5.51. Limit switches are mounted directly to the actuator or rotary-transfer case and use energized arms to make a connection with the moving stem or shaft through a stop plate or similar device. Limit switches come in two basic styles: a single-pole–double-throw style that allows one signal to be sent to one receiver, and a double-pole–double-throw style that allows for two signals to be sent to two receivers. Cammed limit switches are capable of operating anywhere between two and six switches with one unit. Both ac and dc voltage models are available.

5.9.5 Proximity Switches

When a mechanical connection between the limit switch and the stem or shaft is not desirable, a proximity switch is used. A *proximity switch* is a limit switch that use a magnetic sensor instead of a mechanical arm. The switch's sensor is placed close to the stem or shaft, and a metal protrusion is used to trigger the switch when it approaches the sensor.

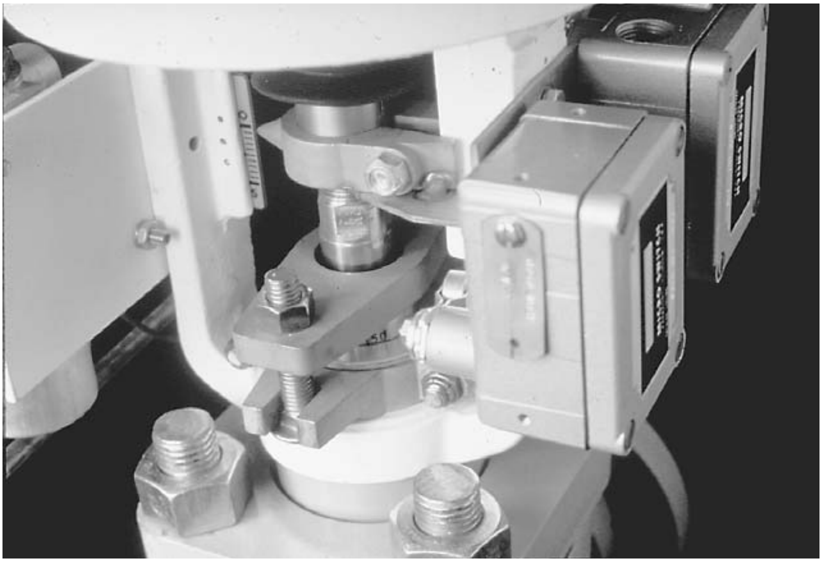


Figure 5.50. Signal-at-open and signal-at-closed limit switch schematic. (*Courtesy of Valtek International*)

Figure 5.51. Cammed limit switch. (*Courtesy of Fisher Controls International, Inc.*)

5.9.6 Position Transmitters

A *position transmitter* is a device that provides a continuous signal indicating the position of the valve or actuator, allowing for signal indication, monitoring actuator performance, logging data, or controlling associated instrumentation or equipment. A potentiometer inside the position transmitter is directly linked to the actuator stem or rotary linkage through an energized arm or linkage (Fig. 5.52). Separate zero and span adjustments are provided, allowing for special modifications, such as monitoring only a critical portion of an actuator stroke. Position transmitters can also be designed with up to four limit switches. Most position transmitters operate off of a two-wire loop, using a 4- to 20-mA dc power supply, and can be made explosion-proof with a special housing.

From a performance standpoint, position transmitters typically provide linearity and hysteresis of between ±1 and ±2 percent of full scale and repeatability between ±0.25 and ±1 percent of full scale.

5.9.7 Flow Boosters

Flow boosters are used to increase the stroking speed of larger pneumatic actuators. Because of their increased volumes, large actuators have difficulty making fast and immediate stokes. Overall, flow boosters respond quickly to sizable changes in the input signal while allowing

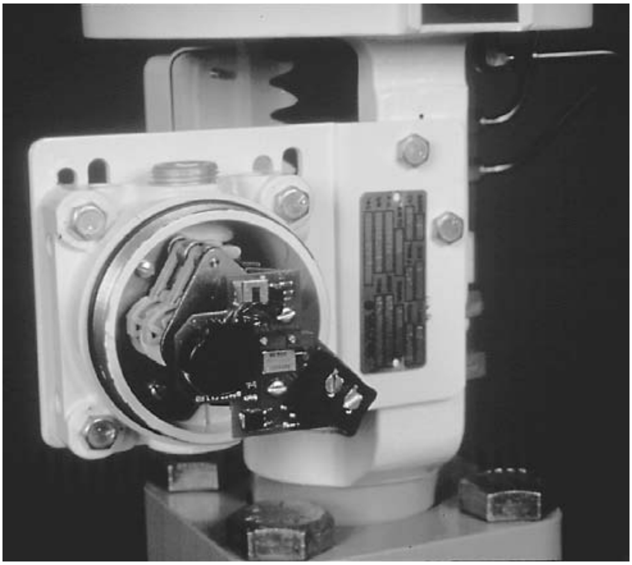


Figure 5.52. Position transmitter (without cover). (*Courtesy of Valtek International*)

Figure 5.53. Flow boosters mounted to double-acting actuator. (*Courtesy of Valtek International*)

for smooth response when the actuator receives small signal changes. A common flow-booster arrangement is shown in Fig. 5.53.

Flow boosters are typically used with positioners with the flow booster being mounted between the positioner and actuator. The flow booster is tubed to the air supply, allowing for the full air pressure to be used to stroke the actuator in the event a larger signal increase or decrease is given. The flow booster utilizes the full air supply only if a large signal is received; otherwise, the normal air flow from the positioner moves through the booster unaided. The air flow is preset using a bypass valve inside the booster. However, when a larger signal is received, the booster inlet or exhaust port opens. If the booster inlet opens, full air supply is sent unregulated to the desired air chamber. At the same time, another booster's exhaust port opens, allowing the opposite air chamber to vent. Both boosters remain in these positions until the pressure differential reaches the dead-band limits of the bypass valve in the booster. When the bypass valve opens, the supply inlet or exhaust ports close and the flow boosters return to normal operation.

To illustrate the advantage of using flow boosters, the following example is provided. A standard 50-in^2 (322-cm^2) actuator requires nearly 4 s to stroke 3 in (7.6 cm), using 0.25-in (0.6-cm) tubing and a 80-psi (5.5-bar) air supply. With flow boosters, this same actuator can stroke in under 1 s. In larger actuators, the example is even more dramatic. A 300-in^2 (1935-cm^2) actuator with a 4-in (10.2-cm) stroke, using 0.375-in (1-cm) tubing and a 80-psi (5.5-bar) air supply, requires over 30 s to stroke. However, with the aid of flow boosters, the stroking time is decreased to under 3 s.

Figures 5.54 and 5.55 show flow-booster schematics for both signal-to-open and signal-to-close arrangements. For exceptional situations, two flow boosters can be installed on each side of the actuator—as long as both flow boosters are connected parallel to the cylinder port, positioner output tubing, and the air supply.

5.9.8 Solenoids

A *solenoid* is an electrical control device that receives an electrical signal (usually a 4- to 20-mA signal) and, in response, channels air supply directly to the actuator. Two types of solenoids, three-way and four-way, are commonly used with actuators and positioners. *Three-way solenoids* are sometimes used to operate single-acting actuators, such as diaphragm actuators, since they are designed to only send air to one air chamber in the actuator. With double-acting actuators, three-way solenoids are used to interrupt or override an instrument signal to a pneumatic positioner.

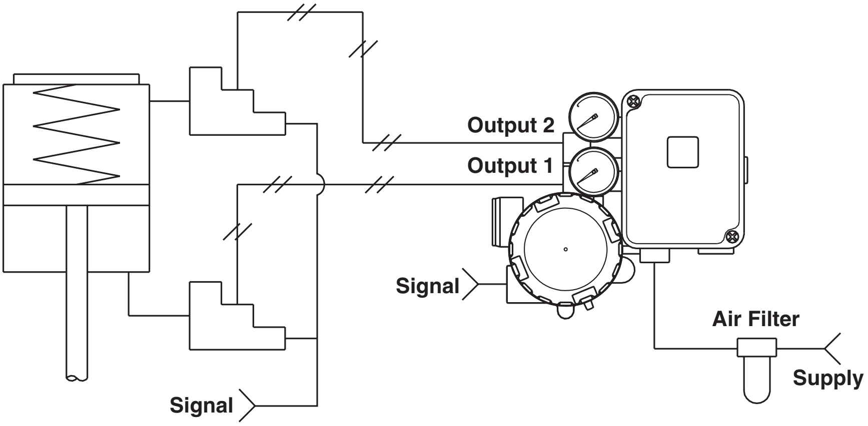


Figure 5.54. Signal-to-open, fail-closed flow-booster schematic. (*Courtesy of Valtek International*)

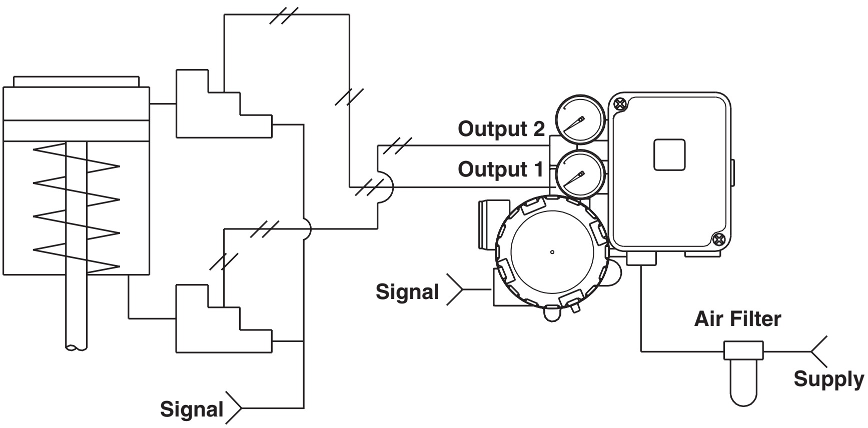


Figure 5.55. Signal-to-close, fail-open flow-booster schematic. (*Courtesy of Valtek International*)

Four-way solenoids are used in lieu of positioners to provide on–off operation of double-acting actuators, providing a positive two-direction action. As shown in Figs. 5.56 and 5.57 (showing both closed and open actions), upon deenergization the four-way solenoids send full air supply to one side of the actuator, while exhausting the other side to atmosphere.

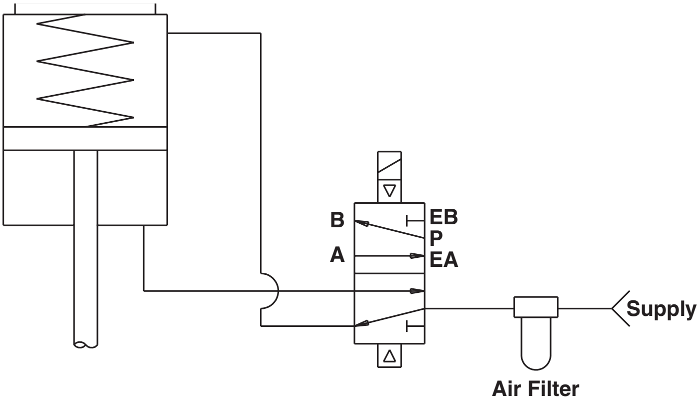


Figure 5.56. Deenergized-to-close, fail-closed four-way solenoid schematic. (*Courtesy of Valtek International*)

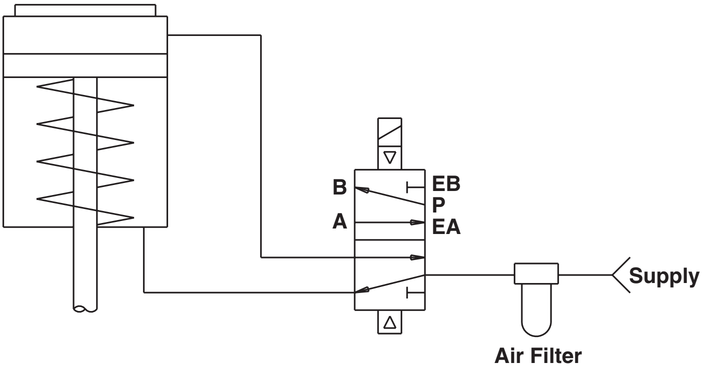


Figure 5.57. Deenergized-to-open, fail-open four-way solenoid schematic. (*Courtesy of Valtek International*)

5.9.9 Quick Exhaust Valves

Quick exhaust valves are pressure-sensitive venting devices that are used with double-acting actuators in on–off applications where positioners are not required. When triggered, quick exhaust valves almost instantaneously vent one side of the double-acting actuator to atmosphere, allowing the valve to move to the full-closed or full-open position. Quick exhaust valves are installed between the air supply and the actuator. As long as a normal air supply is provided to the actuator, normal operation continues. However, when the air supply fails or is interrupted, the quick exhaust valve reacts to the significant differential pressure. An internal diaphragm diverts the exhaust flow coming from the actuator through an enlarged orifice, allowing the internal pressure of the actuator to vent much more quickly. A needle valve must be installed parallel to the quick exhaust valve so that the trip point of the quick exhaust valve can be adjusted, allowing it to react only to large signal demands.

Quick exhaust valves are especially helpful with on–off applications, where exceptional stroking speeds are required in both directions (see Figs. 5.58 and 5.59). Another common application for quick exhaust valves is when a double-acting actuator with a positioner must provide a fast stroke in one direction (as shown in Figs. 5.60 and 5.61).

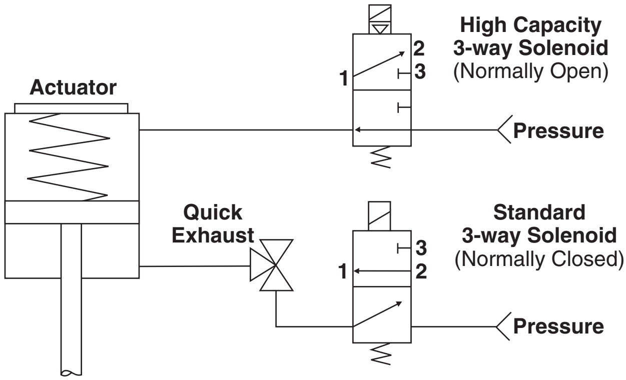


Figure 5.58. Fast-closing, fail-closed on–off system with quick exhaust schematic. (*Courtesy of Valtek International*)

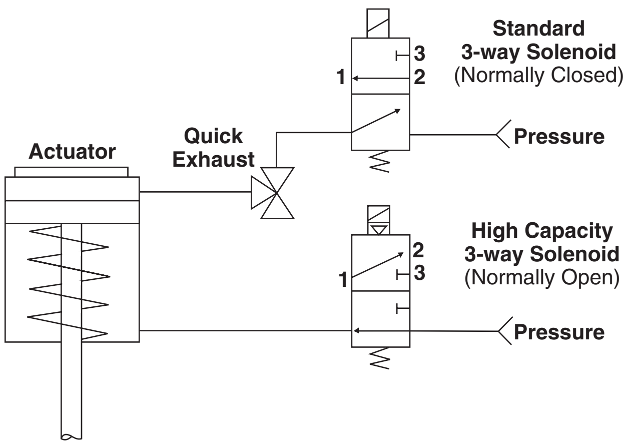


Figure 5.59. Fast-opening, fail-open on–off system with quick exhaust schematic. (*Courtesy of Valtek International*)

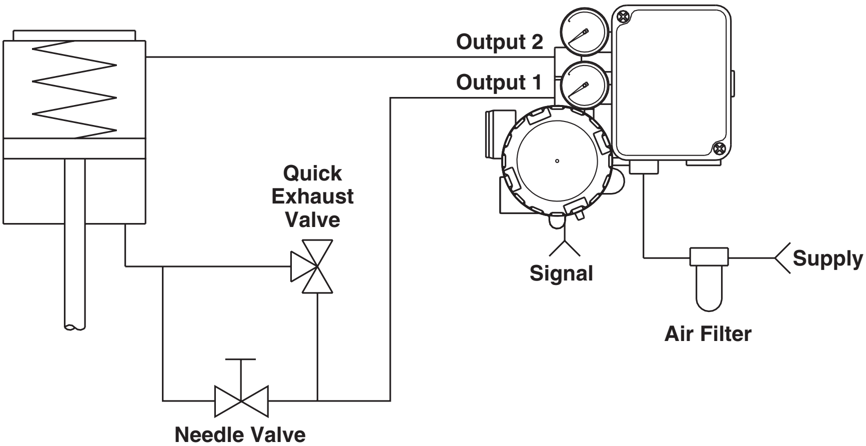


Figure 5.60. Signal-to-open, fail-closed positioner with quick exhaust schematic. (*Courtesy of Valtek International*)

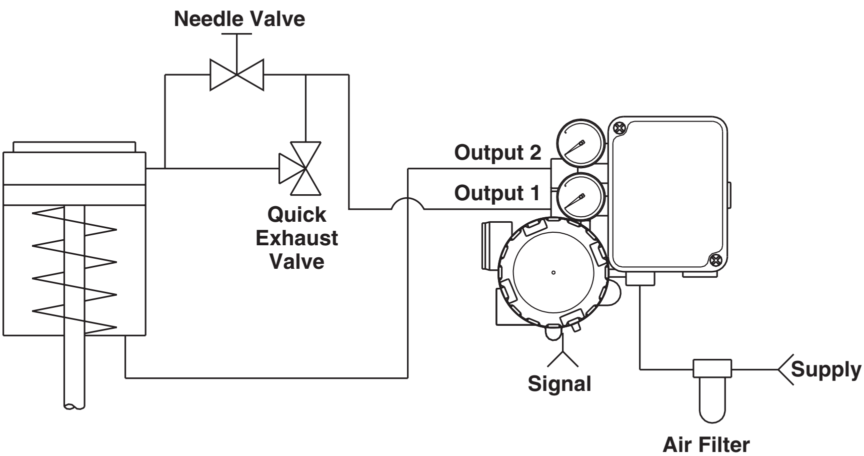


Figure 5.61. Signal-to-close, fail-open positioner with quick exhaust schematic. (*Courtesy of Valtek International*)

5.9.10 Speed Control Valves

Speed control valves are used to limit the stroking speed of an actuator by restricting the amount of air flow to or from the actuator. These small valves can be mounted between the tubing and the actuator and are available in sizes that match common tubing sizes. They can only be used in one direction; therefore, if stroking speeds must be controlled in

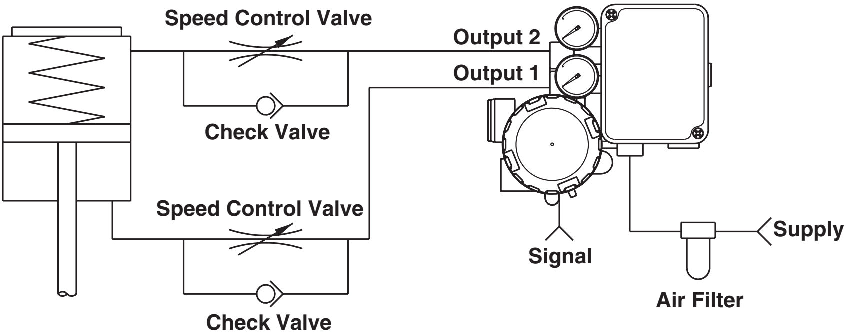


Figure 5.62. Signal-to-open, fail-closed speed control system schematic. (*Courtesy of Valtek International*)

both directions, two speed control valves must be used (one in each direction). A typical application using speed control valves is found in Fig. 5.62.

5.9.11 Safety Relief Valves

When volume tanks are used or if high-pressure actuators must be used to handle the service conditions, some local codes require the installation of safety relief valves on these high-pressure vessels as protection against overpressurization. By definition, *safety relief valves* are designed to open to atmosphere when a particular pressure is exceeded. Because of the differing codes for local governing bodies, valve manufacturers normally defer to the user to install safety relief valves.

5.9.12 Transducers

Transducers are devices that convert an electrical signal to a pneumatic signal, which may be required to operate a positioner with a pneumatic actuator. Transducers have become more commonplace as the popularity of I/P signals has increased with newer control systems, and existing positioners must be converted from pneumatic to electrical signals. The most common transducer is one that converts a 4- to 20-mA signal to a 3- to 15-psi (0.2- to 1.0-bar) pneumatic signal. The pneumatic output signal coming from the transducer normally follows a linear characteristic. Transducers can be mounted directly on the actuator or installed separately, if vibration is a problem (Fig. 5.63).

Figure 5.63. Transducer separately mounted from actuator. (*Courtesy of Valtek International*)

6 Smart Valves and Positioners

6.1 Process Control

6.1.1 Introduction to Process Control

Until recently, the majority of valves and actuators were used as part of analog systems. Today, as the process industry enters a new millennium, the face of process control is changing such that smart technology is quickly overtaking those antiquated analog systems, which were once so prevalent. Smart final control elements—such as intelligent systems mounted on valves or digital positioners used with actuators—have fewer or no moving parts to fail, and the performance associated with digital communications is far and away better than the 4- to 20-mA signal found with I/P analog systems. Plus, today's smart final control elements offer a whole host of new functionalities once thought futuristic—such as automatic loop tuning, self-diagnostics, information processing, planned maintenance, and warning/alarm management.

To understand the terminology and abilities of smart products, a number of common instrumentation and control principles and terms must be generally understood.

6.1.2 Controllers and Distributive Control Systems

A wide majority of control systems that link process sensors and final control elements, such as control valves and actuators, use controllers or distributive control systems to provide intelligence in the control loop. A *controller* is a microprocessor that receives input from a process sensor—such as a pressure or temperature sensor or flow meter—and compares that signal against a predetermined value. After the comparison is made, it sends a correcting signal to a final control element until the predetermined value is reached. A common controller seen in today's systems has a three-way mode that allows for loop tuning—in other words, the adjustment by the user of the proportional, integral, and derivative settings, which is commonly called *PID control.* With PID control, these three settings can be adjusted to optimize the control loop or to provide certain control loop characteristics. For example, variations between the set-point and process variable can be automatically corrected or the system speed can be increased to improve system response.

Related to a controller, but on a much larger scale, is the *distributive control system* (or DCS). The DCS is a central microprocessor designed to receive data from a number of devices and control the feedback to several final control elements. With a DCS, all wiring for the input devices and final control elements lead to one central area, usually in a control room where the DCS is located.

6.1.3 Analog Process Control Systems

The analog process control system has had a long history—beginning in the mid-1970s—as the industry standard. However, by the year 2000, analog process control systems generally had been replaced by the digital process control system as the industry standard. Regardless of this shift, a sizeable number of process control plants still rely on this technology, and its operation should be understood.

With a conventional analog system, the process sensing device transmits a 4- to 20-mA signal to a controller or DCS. The signal is sent through a dedicated line, which is typically a shielded two-wire line. Because the controller or DCS is simply a process computer that utilizes digital signals, the analog information coming from the field must be converted to a digital signal for the controller or DCS to use. This is accomplished through an analog input/output interface card, which converts the analog signal to a digital signal for the microprocessor to use, as shown in Fig. 6.1. If the information received from the transmitted signal is different from the value needed by the process, the controller or DCS sends a correcting signal to the final control element, which can be a control valve. Once again, because of the analog communication lines involved, the controller or DCS will send a digital signal, which is then converted to an analog signal and transmitted across a dedicated analog line to the control valve. The control valve responds by moving its position until the correct process value is achieved.

Analog devices—such as a flow meter, a limit switch, or a positioner— are used to generate process information or react to feedback from the controller and create an analog signal through mechanical means. For example, a limit switch depends on the mechanical movement of the shaft to make contact with the lever arm of the limit switch, which causes the contacts of the switch to meet and send the analog signal.

The main advantage of an analog process control system is that, because of the analog input/output interface, any analog device—whether it is a flow meter or control valve—can communicate with the controller or DCS, making equipment interchangeability easy. A secondary advantage is that the analog system has general acceptance around the world. Instrumentation people are familiar with it and the majority of process devices still use it.

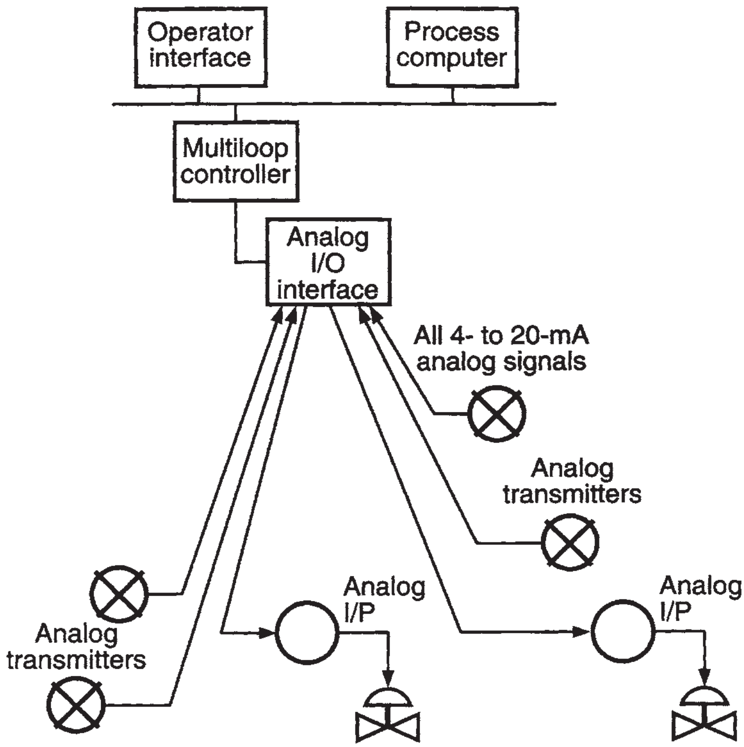


Figure 6.1 Analog process communication network. (Courtesy of Fisher Controls International, Inc.)

Analog systems have a number of disadvantages, as well. First, they must have dedicated lines—or in other words, one line per device. If two devices are placed on one 4- to 20-mA line, the signals are apt to interact adversely with one another and confuse the controller or DCS. Of course, electrical lines can be influenced adversely by magnetic fields and radio frequencies. In addition, wires can be damaged or broken. Analog devices must have moving parts to create the analog signal, which can wear, fail, or hang up. Also, because analog devices have mechanical adjustments, calibration can wander or drift from the necessary settings, especially where vibration occurs.

6.1.4 HART Field Communication Protocol Development

Control valves installed in analog process control systems have benefited by development of the HART® field communication protocol and a wide assortment of HART field instruments. The acronym HART stands for *highway addressable remote transducer* and began as the brainchild of the instrument manufacturer Rosemount in the late 1980s. Rosemount opened up the protocol to other developers and a user group was formed in 1990. In 1993, this user group evolved into the HART Communication Foundation, which was established to support the application of HART protocol to the process industry.

Because of the existence of hundreds of plants with conventional analog process control systems, HART protocol has allowed the use of digital technology within their 4- to 20-mA wired infrastructures, allowing digital communication with HART-designed control valves and other HART devices.

By design, HART protocol preserves the 4- to 20-mA signal, while allowing two-way digital communication to work within the 4- to 20-mA line without disrupting the original purpose of the signal line. HART protocol is developed around a slave/master environment, where the control valve and positioner (with digital capabilities) or other smart device (referred to as the *slave*) only communicates when communicated to by the *master*, which may be a personal computer or handheld communicator. (A typical handheld communicator is depicted in Figure 6.2.)

Typically, HART protocol (when used in conjunction with a smart valve or digital positioner) provides greater functionality or improved performance over conventional 4- to 20-mA devices. In addition, it allows for other information-gathering and performance-based functions, such as calibration, diagnostics, setting device parameters, and data storage.

Figure 6.2 Two-way communications link with a digital positioner, using a HART Handheld Communicator. ***(Courtesy of Fisher Contorls International)***

6.1.5 Digital Process Control Systems

Because of the disadvantages of the analog process control system, coupled with the recent advent of microprocessor-based controllers, distributive control systems, and fieldbus communications, the demand for digital communication has grown significantly throughout the 1990s and into the new millennium. A digital process control system not only utilizes the digital communications associated with the controller or DCS, but also uses the same digital communications with the process sensors and final control elements. This eliminates the analog-to-digital interface conversion as well as some of the mechanical parts and motion associated with analog devices. It greatly improves product reliability, with a minimal amount of moving parts to fail or wires to break. It also ensures that exact information is received by the controller and that the final control element follows the feedback perfectly. With digital systems, hysteresis, repeatability, and other control problems are minimal when compared to analog systems. Although physical lines are usually still required between the controller or DCS, as well as the process sensor and final control element, digital communications allow a number of devices to use a single line. This is because each device can have an electrical signature that would allow it to identify itself to the DCS or controller without signal interference.

Digital communications are dependent on a standardized communication all-digital language, called *fieldbus*. With a standardized fieldbus, field devices not only communicate with the controller or DCS, but also with other field devices. The fieldbus also provides a reasonable power supply to run the complex functions of smart equipment.

With a digital system, analog input/output interfaces are replaced by a fieldbus digital interface, which can receive a number of signals from multiple devices connected to one digital line. The main advantage of digital communications is that the signals sent by any device are easily identified through an instrument signature and can be separated from competing signals. This allows the DCS to sort the information according to one device and send feedback input to another device, all on one line, as shown in Figure 6.3.

The most obvious advantage of a digital system is the improved accuracy and response of the system. With digital communications, no portion of a signal is lost. The lack of moving parts or linkages means better performance, less maintenance and recalibration, and lower spare part inventories. Once a full digital communication link is in place, interchangeability between all devices is possible, which was one of the benefits of the analog system. If PID control is included with the system, the digital system will allow for automatic loop-tuning, improving the performance of the control loop. Information about the performance of equipment can lead to equipment and process diagnostics, which assists with planned maintenance and eliminates maintenance surprises. With fieldbus technology, power is available within the communication lines to run the digital equipment, eliminating outside power sources.

6.1.6 Fieldbus Standardization

Up until the late 1990s, the problem with a standardized fieldbus was the lack of a general agreement by the process industry as to which communication language would best serve its global needs. Early fieldbus developers each produced a different communication language. However, for a user to have full digital communication, all the smart devices must operate off of the same fieldbus; this limits the options of the user since several fieldbus standards were proposed by

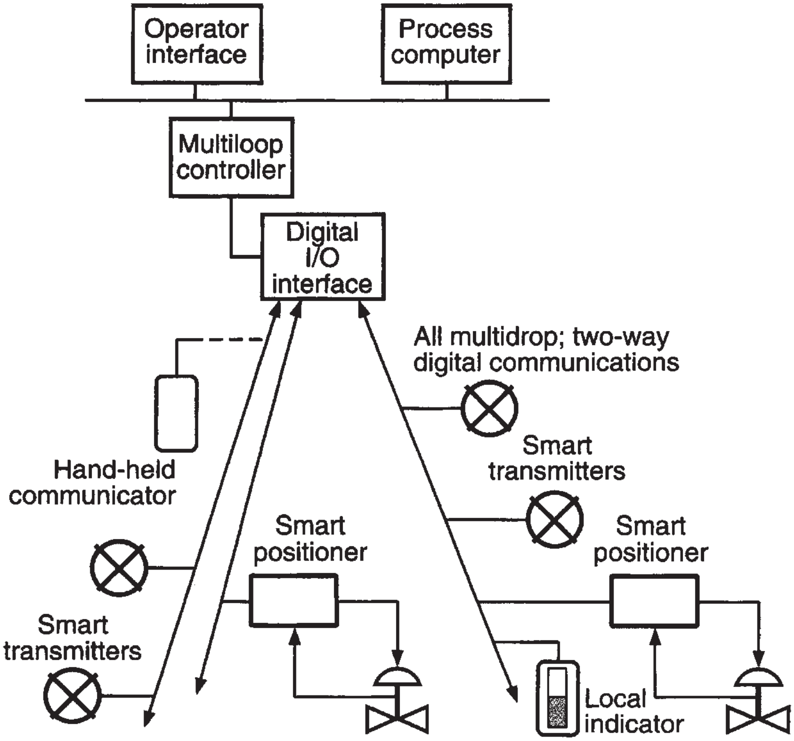


Figure 6.3 Digital fieldbus communications network. (Courtesy of Fisher Controls International, Inc.)

different competing sources. This debate is finally being resolved among those promoting various fieldbus languages, with the Foundation fieldbus technology (developed by the Fieldbus Foundation) seen as the primary leader. With the emergence of a true standardized fieldbus, all smart process equipment and its associated software can then be developed to use the same language, creating a true digital relationship among all digital devices.

Established in 1994, the Fieldbus Foundation brings together over 100 developers and manufacturers of digital process control products. Working together under the Fieldbus Foundation umbrella, these companies have supported a global fieldbus protocol with contributions and product development, although the Foundation fieldbus technology is not owned by any one company. The overwhelming volume of support for this field has driven its acceptance by the process industry, and standardization to the Foundation fieldbus is expected in the near future. Foundation fieldbus is an interoperable system based on the seven-layer communications model established by the International Standards Organization's Open System Interconnect (OSI/ISO), as shown in Table 6.1. Its specifications are also compatible with the ISA's SP50 standards project and the International Electrotechnical Committee (IEC).

6.1.7 Development of Smart Valves

Prior to the 1990s, control valve design and functionality had remained relatively stable. While some new control valve designs, such as the eccentric plug valve and the spring cylinder actuator, were deemed "advances," it was widely accepted that the primary role of the control valve as a final control element had not changed in over 30 years.

The idea of integrating digital communications and intelligence with control valves first surfaced in the early 1990s with early prototypes being heralded by industry experts as the future of process control. However, these early prototypes were designed primarily as self-sustaining devices because the process industry did not have access to a uniform fieldbus standard to link and utilize these valves with the DCS.

However, over the next 10 years (with the aggressive development of fieldbus communications, wireless Internet technology, condition monitoring, Ethernet systems, and field communication devises), the control valve industry has been re-energized by a spurt of digital product development. In 2004, an entire subindustry of digital prod-

Table 6.1. OSI 7-Layer Model

OSI layer	Function
1. Physical	Transmitting raw bit stream through existing mechanical and electrical connections
2. Data link	Establishing data packet structure, bus arbitration, error detection, and framing
3. Network	Routing of all packets and resolving of all network addresses
4. Transport	Providing transparent message transfer, which is independent of the network
5. Session	Providing connection management services for all applications
6. Presentation	Converting data from applications between the local format and the network
7. Application	Providing network-capable applications

ucts and software for control valves has been realized and has driven growth for the entire control valve industry.

6.1.8 Role of Smart Valves

The term *smart valves* has been applied to those control valves that use on-board microprocessors or digital positioners to communicate with either analog or digital systems. As final control elements, control valves must have the ability to communicate digitally with the controller or DCS, as well as to interact with other digital field instruments, to take advantage of the positive aspects of digital communications. As a minimum, this requires a digital positioner.

This development of smart products, however, was slowed initially by the lack of a standardized fieldbus—although a number of smart valves and positioners available today have been developed so that they can handle a number of proposed fieldbus versions (each requiring unique software and hardware versions). Today's smart valves vary widely according to the capabilities of microprocessor and design. For example, intelligent systems provide complete single-loop control when placed on a valve—which requires process sensors, a controller, and a digital positioner.

This allows for a wide range of functions, from process control to data acquisition to self-diagnostics. In addition, with some smart valve designs, PID control can be added to automatically loop tune the process so that it is more efficient. On the other hand, a digital positioner has the microprocessor included with the positioner and is used only to assist the valve with its ability to act as the final control element. Overall, both smart valves and positioners can provide various levels of valve self-diagnostics and management of safety systems, such as a controlled shutdown.

Many existing plants today remain wired with analog lines, each attached to an individual input device or final control element. To replace these analog lines with digital lines is time-consuming and expensive; thus this conversion has been somewhat delayed due to economic concerns. For this reason, an open protocol has been developed. The open protocol allows smart products to utilize existing analog lines for both communication and power needs. This means that those smart devices that use the existing 4- to 20-mA lines must use the worst case scenario—4 mA—as the main power source. The problem with such low power is that the device can only have a limited amount of electronics and, therefore, the smart capabilities of such devices are limited. As mentioned earlier, one advantage of a fieldbus is that the power could be increased to expand the capabilities of smart devices in general.

Smart valves are primarily linear-motion valves, with globe valves being the primary focus, although some rotary-motion designs have been modified to smart service. An advantage of using smart products with a rotary valve—which has an inherent flow characteristic—is that a modified flow characteristic can be custom programmed, providing better flow control for the user. Also, a smart valve can correct the problems associated with a positioner's linear-to-rotary motion, which does not produce a true linear signal because of the swing arc of the positioner take-off arm.

6.2 Intelligent Systems for Control Valves

6.2.1 Introduction to Intelligent Systems

As discussed earlier, the most sophisticated smart valve is a control valve that is equipped with an intelligent system with process sensors. The *intelligent system* is a microprocessor-based controller that is capable of providing local process control, diagnostics, and safety management. Process input to the intelligent system comes through process sensors mounted on the body, as shown in Fig. 6.4. The system also has internal sensors to monitor valve stem position and pressures on both sides of the pneumatic actuator.

Placing a controller and process sensors on a control valve allows for *single-loop control*—defined simply as an input sensor sending information to the controller, which sends a correcting signal to a final control element until the correct value is achieved. By monitoring the upstream pressure, downstream pressure, temperature, and the stem position, the intelligent system can calculate the flow rate for the valve and compare that against the predetermined setpoint—and make any necessary position adjustments to provide the correct flow rate. The intelligent system can be configured to handle single-loop control for the pressure differential, upstream pressure, downstream pressure, temperature, flow rate, stem position, or another auxiliary process loop. Because the intelligent system can be programmed to handle local control and measurement of the process, the DCS can be used to handle more demanding control situations elsewhere in the plant or to provide an overall process supervising function. With its local controller, the intelligent system is then capable of monitoring and creat-

Figure 6.4 Intelligent control system with an integral digital positioner mounted on a globe control valve. (*Courtesy of Valtek International*)

ing a record of the upstream and downstream pressure, differential pressure, process temperature, and the flow rate. The controller of intelligent systems can be equipped with PID control that uses a value from an external transmitter or internal process parameters as the control variable. This allows the process to be tuned for more efficient process control in a number of wide-ranging applications.

Intelligent systems can be used in either analog or digital systems with digital or conventional analog positioners (Fig. 6.5). They can respond to PID operation with a 4- to 20-mA analog signal, a digital signal, or through a preprogrammed set-point. Intelligent systems sometimes require the use of a personal computer or the DCS to set the tuning and operating parameters of the smart valve—although some of the newer versions come equipped with an on-board keypad, which allows for direct operation.

The user communicates with the intelligent system through a number of operator interfaces: DCS input/output interface card, hand station and recorder, or personal computer. When a personal computer is used to communicate with the intelligent system, interface software (provided by the manufacturer) must be installed.

Figure 6.5 Intelligent control system combined with an analog positioner mounted on a globe control valve. (*Courtesy of Valtek International*)

The close proximity of the process sensors and control valve to the controller greatly reduces the dead time or lag time, significantly increasing the response to process changes. When a digital positioner is included in the intelligent system, the problems associated with hysteresis, linearity, and repeatability are greatly reduced. The intelligent system has the capability of collecting and issuing flow and process data to the DCS, which provides the user with a current engineering analysis of the process. Remote sensors can also be tied to the intelligent system for improved control of the other parameters of the process without having to channel the data through the DCS.

An important side benefit of an intelligent system is that line penetrations are reduced significantly—an important consideration in this age when fugitive emissions are a critical concern. Because the process sensors are installed on the valve itself, the single-point installation of the valve eliminates separate line penetrations for the flow meters as well as the temperature and pressure sensors. Therefore, instead of having four or five line penetrations as part of the control loop, only one (the smart valve) exists, which eliminates a number of potential leak paths as well as decreasing EPA (or other governing body) reporting functions.

Intelligent systems allow for valve and process self-diagnostics through their ability to record a signature of the valve or process. When the valve is first installed, a signature can be taken of the valve's initial start-up performance or of the process itself by plotting the flow against certain travel characteristics. As the valve continues in operation, periodic monitoring of the valve's and system's performance can be compared against the initial start-up signature. When this performance begins to falter through normal wear or through an unexpected failure, the intelligent system can warn the user of pending or existing problems, allowing for preventative maintenance or corrective action to take place before a major system or valve failure. For example, the system can take a signature of the leakage through the seat in a closed position (by monitoring the downstream pressure). Over time the intelligent system can compare the initial signature against the current body leakage signature. If the current reading exceeds the ANSI leakage class (a preset condition) due to a damaged or worn closure or regulating element, the system can warn the user that servicing of the closure element is needed. By monitoring the upper and lower pressure chambers of the actuator, intelligent systems can also evaluate a loss of packing compression and actuator seals or recognize jerky stem travel, which may point to a problem with the closure or regulating element. If an analog positioner is used with the system, hysteresis, repeatability, and linearity can be monitored.

Since a process signature is possible, the system's overall performance, which can be affected by associated upstream or downstream equipment, can be monitored and evaluated as well. For example, if an upstream pump begins to slow, the upstream pressure will decrease and fall below acceptable limits at a certain point. When the intelligent system finds the pressure dropping below the preset value, it can alert the user, who can then schedule the necessary valve or actuator maintenance.

Safety management is another use for intelligent systems, since they are capable of programmable settings that can notify the user when process limits are violated by a system upset. In addition, the systems can be used to monitor and analyze the process during start-up and shutdown, warning of any sudden departures from the normal service conditions. Multiple failure modes can be programmed into the intelligent system, which will provide a different mode for a variety of failures: loss of air supply or power, process failure, loss of command signal, etc.

Data logging is another advantage of intelligent systems, as they have the ability to record process conditions through user-specified intervals. For example, some intelligent systems are capable of recording up to 300 lines of process conditions at intervals anywhere between a second to three hours apart. This data log is normally provided so that the user can evaluate the process, looking for any abnormalities or upsets.

The wide range of benefits of an intelligent system is often reflected in the price of the intelligent system, which may produce some "sticker shock" to those accustomed only to the cost associated with other actuator accessories. However, the user should look at the larger picture: The intelligent system takes the place of a controller, individual pressure and temperature sensors, a flow meter, limit switches, tubing and wiring, etc. Taken together, the cost of an intelligent system mounted directly on a control valve is less than the sum of the individual pieces of equipment. The only evident problem with an intelligent system is that it requires a separate 24-V dc power supply to run the electronics, which may require some additional wiring and a conversion box if only standard ac power is available.

A simplification of the intelligent system is to install the system to the actuator without including the process sensors in the body (using existing sensors already installed in the system)—in essence, creating a very powerful digital positioner. This allows the intelligent system to function with many of the advantages discussed earlier, but without the on-board single-loop control. The advantage is that the cost is less, yet offers many of the smart technology benefits associated with the full intelligent system.

6.2.2 Intelligent System Design

Shown in Fig. 6.6 is a schematic of a typical intelligent system. Power is supplied by a separate 24-V dc source as well as a compressed air source. Pressure sensors are mounted directly to the body on the upstream and downstream sides of the closure element. The location of the pressure sensors on the body is critical to ensuring proper pressure readings without being affected by an increase of velocity as the flow moves through the closure or regulating element or any other narrowed section of the body. The temperature sensor is placed between the pressure sensors and as close to the closure or regulating element as necessary to determine the best process temperature reading. The wiring for the sensors is tubed directly to the intelligent system. Pneumatic lines feed air from the digital positioner (in this case, the digital positioner is part of the intelligent system) to the upper and lower chambers of the actuator.

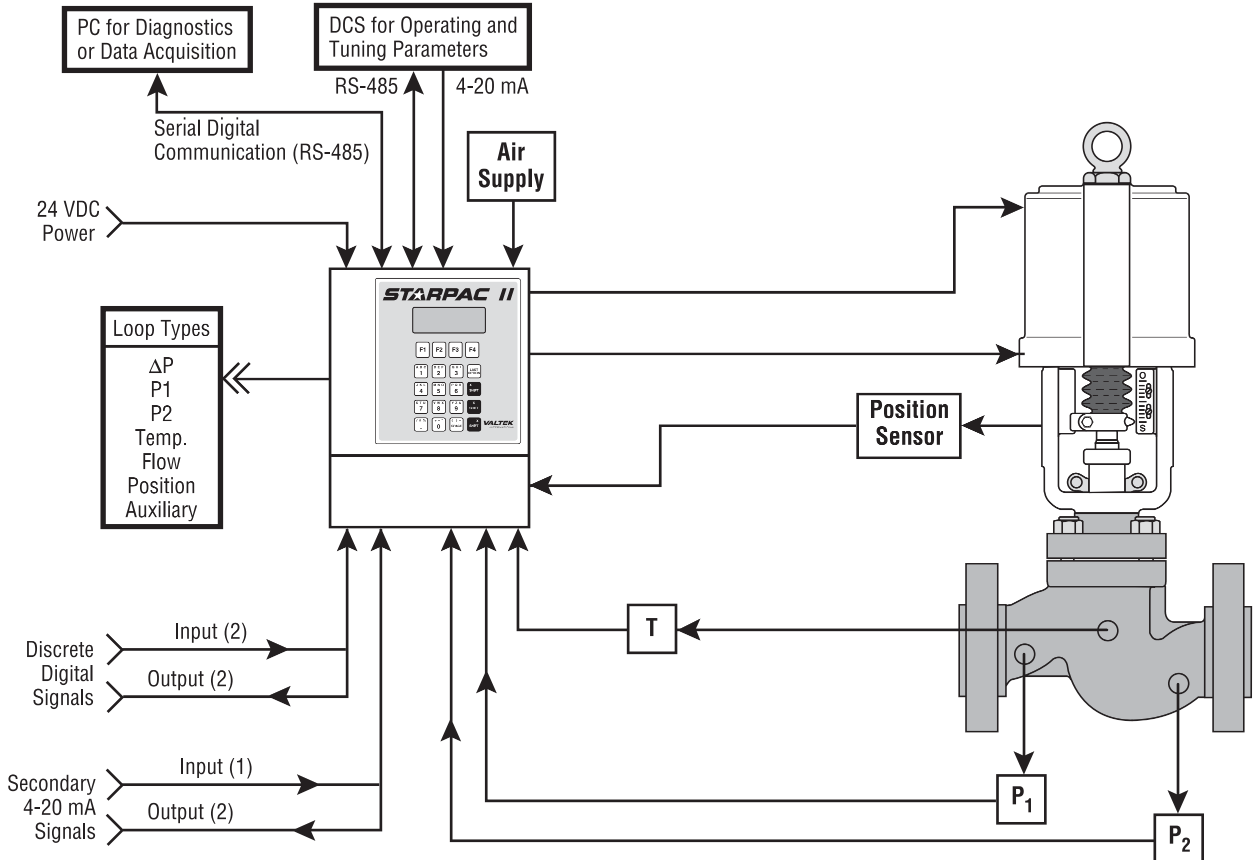


Figure 6.6 Schematic showing an intelligent control system with an internal digital posiioner. (*Courtesy of Valtek International*)

Operating or tuning input, as well as data acquisition, takes place through either the supervisory DCS or through a personal computer via a serial digital communication line, which is a designated electrical signal, such as RS-485. A separate 4- to 20-mA line is linked from the DCS to the intelligent system for any standalone command signals. With the single-loop control associated with an intelligent system, this line is often not necessary but is available if needed.

Input and output lines are provided for discrete digital signals that act as switches, allowing the user to toggle between manual and automatic operation of the intelligent system or for other custom configurations. The secondary 4- to 20-mA signal inputs are used for any auxiliary input, such as from a remote flow meter to control downstream pressure. The secondary 4- to 20-mA outputs are used to communicate with another supervisory device, such as another controller.

As noted earlier, intelligent systems can be used with standalone positioners. A schematic of an intelligent system with an analog positioner is shown in Fig. 6.7.

6.3 Digital Positioners

6.3.1 Introduction to Digital Positioners

Following the introduction of the intelligent system for control valves, a logical step was to move toward *digital positioners,* which are devices that use a microprocessor to position the pneumatic actuator and to monitor and record certain data (Fig. 6.8).

Digital positioners do not provide single-loop control as intelligent systems do; therefore, they must be installed in a more conventional process loop, with a controller and process sensors. Although they are not equal to intelligent systems, digital positioners can perform some of the same functions. For example, a digital positioner can measure and transmit actuator stem position, providing alarm signals (similar to limit switches) when a certain position is reached or exceeded and eliminating any requirement for an independent position transmitter. PID control and tuning are also possible.

Because the pressures to the actuator are monitored, changes in actuator operation pressures can allow self-diagnostics of the actuator and certain aspects of the valve, such as changes in packing compression or a binding closure element. As with all smart devices, digital positioners have an electronic signature that allows for remote identification. The positioner can be characterized and calibrated remotely through input

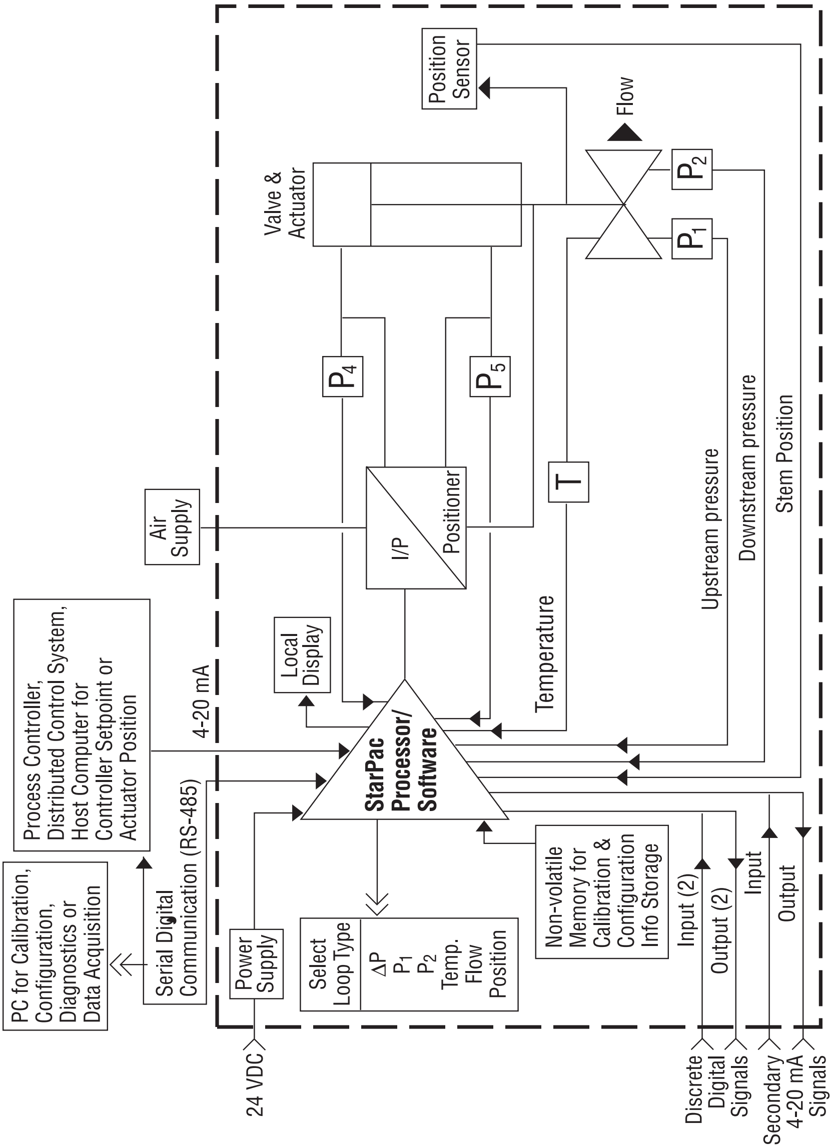


Figure 6.7 Schematic showing an intelligent control system with separate electro-pneumatic positioner. (*Courtesy of Valtek International*)

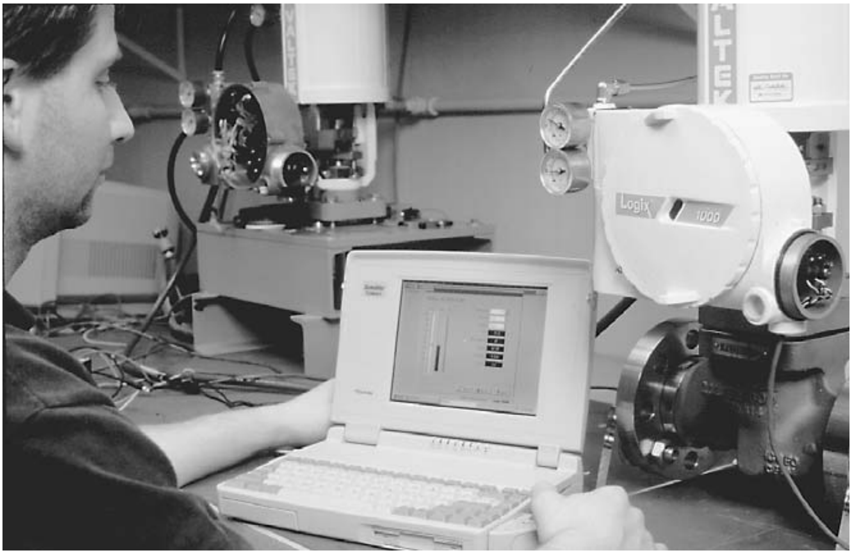


Figure 6.8 Computer interface with a digital positioner. (*Courtesy of Valtek International*)

from the DCS or a personal computer. No characterizable cam is required to modify an inherent valve characteristic; instead, the electronics can be used to provide a modified or customized flow characteristic.

As stated earlier, digital positioners have far lower hysteresis and better repeatability and linearity than analog positioners. However, because digital positioners still have some moving parts–such as a spool valve and a linear-to-rotary linkage at the actuator stem— some hysteresis, repeatability, and linearity problems can exist. The advantage to using smart electronics is that such errors can be zeroed out, allowing the positioner to take such problems into account. Both an advantage and disadvantage of the digital positioner is its reliance upon two-wire 4- to 20-mA signal and power sources. The obvious advantage is that an analog positioner receiving an electrical signal could be replaced with a digital positioner. The disadvantage is that only 4 mA is available to run the positioner, which limits the amount of electronics that can be run through the power source.

6.3.2 Digital Positioners Design and Operation

A typical digital positioner schematic is shown in Fig. 6.9. The command 4- to 20-mA signal provides the power source to the electronics.

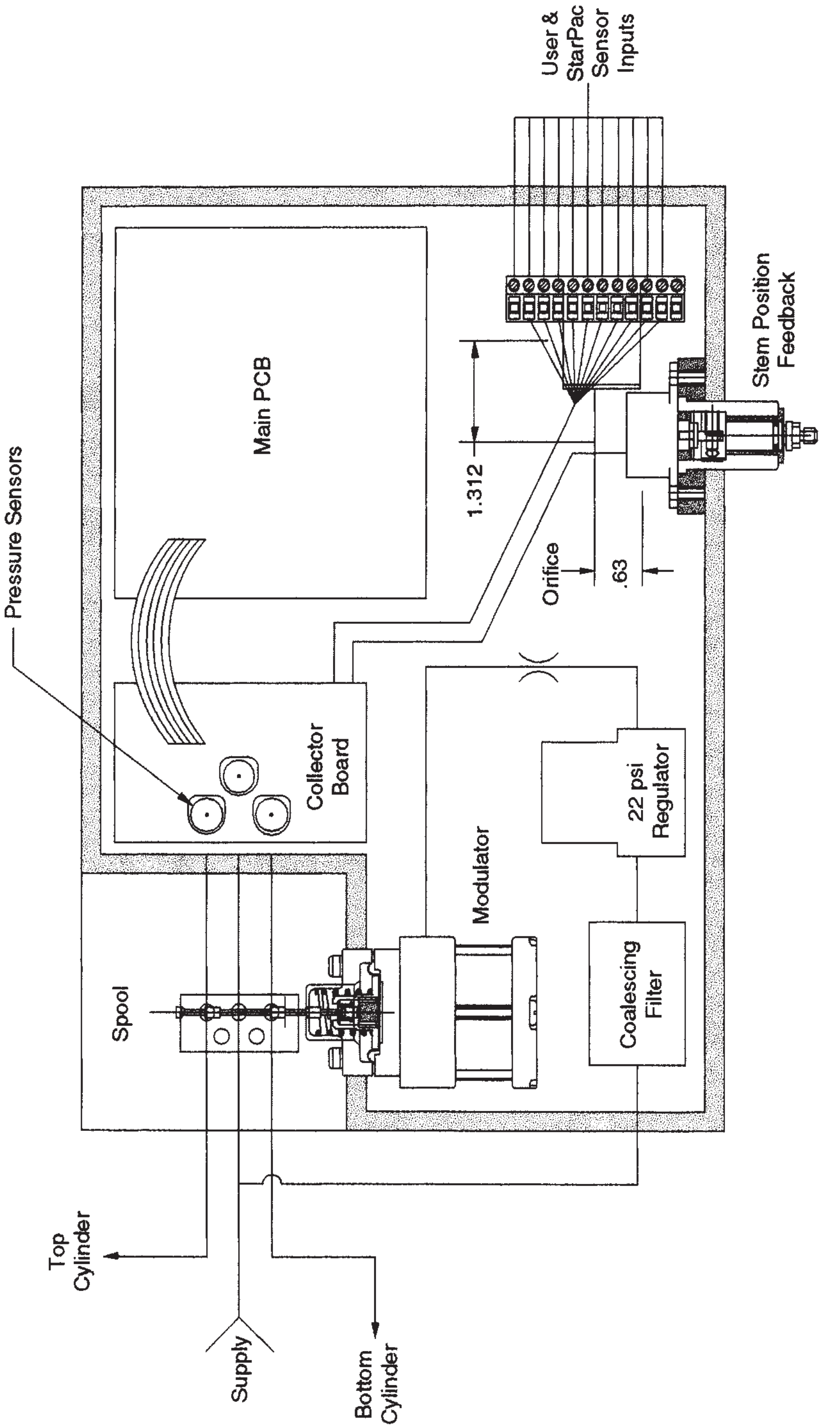


Figure 6.9 Digital positioner schematic. (*Courtesy of Valtek International*)

Compressed air is also required to provide the power to the pneumatic actuator. The actuator's feedback position is provided by a special take-off arm that provides a mechanical-to-electronic function: The linear motion of the actuator stem turns a rotating potentiometer, which provides position feedback to the positioner's electronics and compares that feedback to the signal. If a discrepancy occurs either through a changing signal or through an incorrect actuator position, a correcting electronic signal is sent to a pressure modulator. The pressure modulator then positions an inner spool, which sends air to one side of the actuator and exhausts the other side. This action moves the position of the actuator and continues until the correct position is reached. At this point, the feedback is equal to the signal and the pressure modulator places the inner spool in a holding position.

A key element in the correct operation of digital positioners is the placement of pressure sensors in the electronics that can monitor the air pressure sent to the actuator. This information is important in recording an initial signature for the actuator's function, as well as providing future signatures that can be used for self-diagnostics.

7 Valve Sizing

7.1 Introduction to Valve Sizing

7.1.1 The Importance of Correct Valve Sizing

Generally, valve sizing is based on the standard thermodynamic laws of fluid flow. The application of these laws is affected by the particular function of the valve plus the type and severity of the service. Simple on–off block valves are expected to pass nearly 100 percent of the flow without a significant pressure drop, since they are not expected to control the flow other than to shut it off. On the other hand, throttling services are expected to produce a certain amount of flow at certain positions of opening and take a particular pressure drop. Therefore, the science of valve sizing is almost always directed toward sizing throttling valves.

With manually operated on–off block valves, the valve is often expected to pass full flow. If the valve's internal flow passage or closure element is sized smaller than the upstream piping, flow will be restricted from that point forward. This will cause the valve to take a pressure drop and pass less flow, defeating the major purpose of the on–off valve. If the on–off block valve is sized larger than the upstream piping, installation costs are more expensive (since increasers are required). The larger valve is also more expensive. On the other hand, throttling valves, which are intended to take a pressure drop and to reduce the flow, may have a seat that is significantly less in diameter than the upstream port. Determining the flow through this diameter is the science behind valve sizing. If a throttling valve is sized too small, the maximum amount of flow through the valve will be limited and will inhibit the function of the system. If a throttling valve is sized too large, the user must bear the added cost of installing a larger valve. Another major disadvantage is that the entire flow control may be accomplished in the first half of the stroke, meaning that a minor change in position may cause a large change in flow. In addition, because regulation occurs in the first half of the stroke, flow control is extremely difficult when the regulating element is operating close to the seat. The ideal situation is for the throttling valve to utilize the full range of the stroke while producing the desired flow characteristic and maximum flow output.

Throttling valves are rarely undersized because of the number of safety factors built into the user's service conditions and the manufacturer's sizing criteria. Because of these safety factors, a large number of throttling valves actually end up being oversized. This happens because the user provides a set of service conditions that are usually the maximum conditions of the service (temperature, pressure, flow rate, etc.). The manufacturer then adds its own safety factors into the sizing equations. The valve manufacturer does this to avoid the error of undersizing, which is less forgiving than oversizing. Although not ideal, an oversized valve is still workable.

7.1.2 Valve-Sizing Criteria for Manual Valves

The basic function of manual on–off block valves is quite simple: to pass full flow while the valve is open or to shut off or divert the full flow when closed. Therefore, the valve size can sometimes be determined simply by the size of the piping, which has already been sized by the system engineers. Manual-valve manufacturers often provide sizing charts that indicate the relationship between the flow-rate requirement (Q) are the minimum and maximum valve size that can pass the given flow rate.

An important choice in manual-valve sizing is whether the valve should be full bore or reduced bore. In many cases this is more a function of the valve's purpose to pass full flow or to take a slight pressure drop. If the valve is installed in an application that must allow the passage of a pig to clean or scour the pipeline, the valve chosen must be full bore, since the pig is the same size as the inside diameter of the pipe. Another application calling for full-bore manual valves is one installed in slurries or services with entrained materials or particulates. If the valve has a reduced bore, these particulates or slurries have a tendency to settle and become trapped at the narrowed constriction. A full-bore valve has no such restriction, allowing for free passage of the foreign material without collection. Full-bore manual valves are also chosen for services with high velocities, for which a restriction would increase the chance of erosion as well as increase the velocity further.

The service conditions generally required for correct manual-valve sizing are maximum and minimum temperatures, pressures, flow rates, and specific volume (steam applications). Not only are the extremes important, but also the average operating conditions are important. The specific volume is normally provided to the user by commonly published steam tables, which show the specific volume in cubic feet per pound. Most steam tables provide the data in *pounds per square inch absolute* (psia), which does not take atmospheric pressure (14.7 psi or 1.03 bar) into account. On the other hand, *pounds per square inch gage* (psig) accounts for this adjustment for the atmospheric pressure. The metric equivalent for psig is *barg.*

7.1.3 Valve-Sizing Criteria for Check Valves

The most critical element of check-valve sizing is that a sufficient pressure drop and minimum flow exist for the check valve to open. Without a pressure drop, the closure element will not open and the valve will remain closed, which is what happens when a pump fails to maintain a proper flow or flow reverses.

The minimum pressure drop required for check valves to open is typically 1 psi (0.07 bar). This minimum pressure drop is needed to maintain the open position of the closure element without failing. If the pressure drop falls to less than 1 psi, the closure element will float back and forth, which is commonly called “flutter.” As the disk moves toward the seat, the opening narrows and pressure rebuilds, which causes the disk to open higher. This low-pressure drop situation will cause this cycle to repeat until the pressure drop is increased, causing wear of the moving parts and shortening the life of the check valve. The maximum pressure drop is approximately 10 psi (0.7 bar), depending on the size of the check valve. Higher pressure drops lead to severe erosion of the check valve’s closure element.

Check-valve manufacturers provide the cracking pressure of their check valves. The cracking pressure is the minimum pressure required to open the check valve and is a fixed number associated with the style and size of the check valve. It can vary anywhere from 0.1 to 0.5 psi (0.01 to 0.03 bar). Generally, cracking pressures are of little concern unless the pressures in the process are extremely low or the pressure drop is small (less than 1 psi). However, the cracking pressure can be important if the valve is installed in a vertical line, where the check valve must open against gravitational forces in addition to the process pressure. Smaller lines have higher cracking pressures than larger lines. This is because the larger the line, the larger the process force must be against the component's mass in the check valve.

Unless the flow experiences a wide range of flow during the service, check valves are sized for minimum flow, which in turn determines the valve size. This is done using manufacturer's sizing charts. If the size provided for the minimum flow is equivalent to or greater than the pipeline size, the pipeline size should be used for the valve size. For example, if the manufacturer's literature calls for a 4-in check valve, yet the pipe size is 3-in line, a 3-in check valve should be satisfactory. The larger, oversized valve will not benefit the flow rate yet is more expensive and would require the installation of increasers. If the suggested valve size for the minimum flow is smaller than the pipeline, reducers must be installed and the smaller-sized check valve installed.

The user should ensure that the flow rates are within the parameters of the check-valve design. High flow rates can increase the frequency of vortices and currents, which will increase the pressure drop across the valve as well as cause valve wear. Insufficient flow will cause the valve to flutter. The flow must be sufficient to overcome the closed position of the check valve—whether it be gravity, weight of the closure element, line orientation, or spring force.

As a general rule, the maximum liquid flow velocity for check valves is 11 ft/s (3.4 m/s). The minimum liquid flow velocity is normally 6 to 7 ft/s (1.8 to 2.1 m/s), although some designs (such as a double-disk check valve) can operate at 3 ft/s (0.9 m/s).

7.1.4 Valve-Sizing Criteria for Throttling Valves

Throttling valves require a systematic method of determining the required flow through the valve, as well as the size of the valve body, the body style, and materials that can accommodate (or tolerate) the process conditions, the correct pressure rating, and the proper installed flow characteristic. The industry standard for determining the flow capacity of a throttling valve is ANSI/ISA Standard S75.01, which contains the equations required to predict the flow of incompressible (liquid) and compressible (gas) process fluids. Because of the compressibility issues between liquids and gases, equations have been formulated for each and are included in this chapter.

Proper selection of the valve is based on the service conditions of the process. For correct sizing, the following conditions are needed: the upstream pressure; the maximum and minimum temperatures; the type of process fluid; the flow rate that is based upon the maximum flow rate, the average flow rate, and the minimum flow rate; vapor pressure; pipeline size, schedule, and material; the maximum, average, and minimum pressure drop; specific gravity of the fluid; and the critical pressure.

7.2 Valve-Sizing Nomenclature

7.2.1 Upstream and Downstream Pressures

In process systems, most valves are designed to either pass or restrict the flow to some extent. In order for the process to flow in a particular direction through a valve, the upstream and downstream pressures must be different; otherwise, the pressure would be equal and no flow would occur. By definition, the *upstream pressure* is the pressure reading taken before the valve, while the *downstream pressure* is the pressure reading taken after the valve.

7.2.2 Pressure Drop

The resulting difference between the upstream and downstream pressures is called the *pressure drop* (or the *pressure differential*). The pressure drop allows for the flow of fluid through the process system from the upstream side of the valve to the downstream side. In theory, the greater the pressure drop, the greater the flow through the valve.

7.2.3 Flow Capacity

The most commonly applied sizing coefficient is known as the *valve coefficient* (C_v), which is defined as one U.S. gallon (3.8 liters) of 60°F (16°C) water that flows through a valve with 1.0 psi (0.07 bar) of pressure drop. This general equation is written several ways, but two of the most common methods are

$$C_v = Q\sqrt{\frac{S_g}{\Delta P}}$$

or

$$C_v = Q\backslash\sqrt{\frac{\Delta P}{S_g}}$$

where C_v = required flow coefficient for the valve
Q = flow rate (in gal/min)
S_g = specific gravity of the fluid
ΔP = pressure drop (psi)

When calculated properly, C_v determines the correct trim size (or area of the valve's restriction) that will allow the valve to pass the required flow while allowing stable control of the process throughout the stroke of the valve.

7.2.4 Actual Pressure Drop

Another term for pressure drop, *actual pressure drop* (ΔP), is defined as the difference between the upstream (inlet) and downstream (outlet) pressures. When the choked and actual pressure drops are compared and the actual pressure drop is smaller, it is used in the C_v sizing equation.

7.2.5 Choked Pressure Drop

As the C_v equation is examined, the assumption is made that if the pressure drop is increased, the flow should increase proportionately. A point exists, however, where further increases in the pressure drop will not change the valve's flow rate. This is what is commonly called *choked flow.*

As illustrated in Fig. 7.1, with liquid applications having a constant upstream pressure, the flow rate Q is related to the square root of the pressure drop with a proportional and constant C_v. When the valve begins to choke, the flow-rate curve falls away from the linear relationship. Because of the choked condition, the flow rate will reach a maximum condition due to the existence of cavitation in liquids or sonic velocity with gases.

Depending on the valve style, this departure from the linear relationship will occur at different regions of the line, with some being more gradual and others more abrupt. For example, globe-style valves tend to handle higher pressure drops without choking, as opposed to rotary valves, which tend to choke and cavitate at smaller pressure drops.

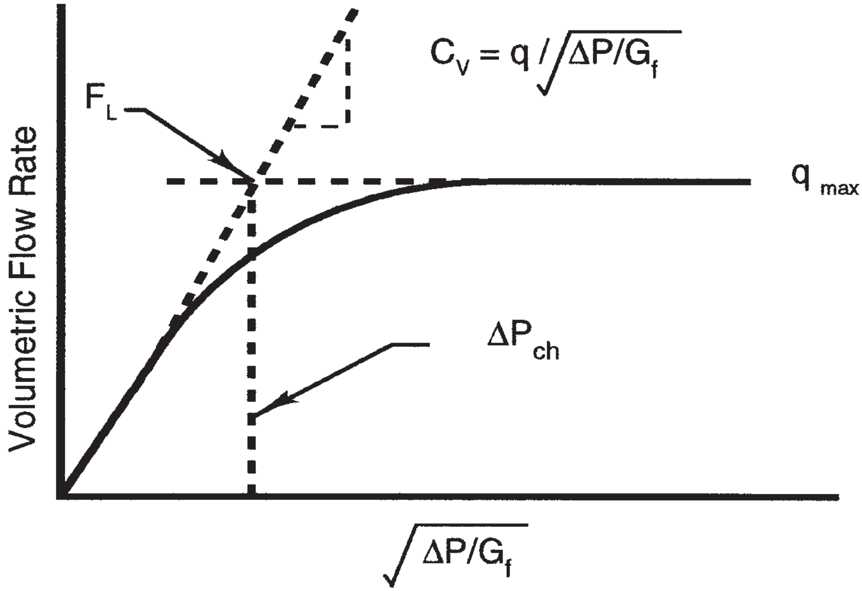


Figure 7.1. Maximum flow rate occurring due to choked conditions. (*Courtesy of Valtek International*)

For simplicity, the term *choked pressure drop* ΔP_{choked} is used to show the theoretical point where choked flow occurs, intersecting the linear lines of the constant C_v and the maximum flow rate Q_{max}. This point is known as the liquid pressure-recovery factor F_L, which is discussed in more detail in this section. The ANSI/ISA sizing equations for liquids use F_L to calculate the theoretical point where choked flow occurs (ΔP_{max}) so that the valve can be sized without the difficulty of the process being choked.

For gas applications, the terminal pressure-drop ratio x_T is used to describe the choked pressure drop for a particular valve.

7.2.6 Allowable Pressure Drop

The *allowable pressure drop* ΔP_a is chosen from the smaller of the actual pressure drop or the choked pressure drop and is used in the determination of the correct C_v. When determining the C_v of a liquid application, the following must be considered to see if the allowable pressure drop should be used: first, if the inlet pressure P_1 is fairly close to the vapor pressure; second, if the outlet pressure P_2 is fairly close to the vapor pressure; and third, if the actual pressure drop is fairly large when compared to the inlet pressure P_1. If any one of the above three conditions exists, the user should calculate the allowable pressure drop and compare it against the actual pressure drop, using the smaller value.

7.2.7 Incipient and Advanced Cavitation

With liquid applications, when the fluid passes through the narrowest point of the valve (vena contracta), the pressure decreases inversely as the velocity increases. If the pressure drops below the vapor pressure for that particular fluid, vapor bubbles begin to form. As the fluid moves into a larger area of the vessel or downstream piping, the pressure recovers to a certain extent. This increases the pressure above the vapor pressure, causing the vapor bubbles to collapse or implode. This two step-process—creation of the vapor bubbles and their subsequent implosion—is called *cavitation* and is a leading cause of valve damage in the form of erosion of metal surfaces.

As the pressure drops, the point where vapor bubbles begin to form is called *incipient cavitation.* The pressure level where cavitation is occurring at its maximum level is called *advanced cavitation.* During advanced cavitation, the flow is choked and cannot increase, which affects the flow capacity of the valve as well as its function. The point where advanced cavitation occurs can be predicted. To do this, the pressure drop must be determined, using the liquid cavitation factor F_i. A detailed discussion about the causes and effects of cavitation is found in Sec. 9.2.

7.2.8 Flashing Issues

When the downstream pressure does not recover above the vapor pressure, the vapor bubbles remain in the fluid and travel downstream from the valve, creating a mixture of liquid and gas. This is called *flashing.* Problems typically associated with flashing are higher velocities and erosion of valve components. Section 9.3 provides a more detailed discussion about flashing and its effects.

7.2.9 Liquid Pressure-Recovery Factor

A critical element in liquid sizing is the *liquid pressure-recovery factor* F_L, which predicts the effect the geometry of a valve's body will have on the maximum capacity of that valve. F_L is used to predict the amount of pressure recovery occurring between the vena contracta and the outlet of the body.

The liquid pressure-recovery factor is determined by the manufacturer through flow testing that particular valve style. F_L factors can vary significantly depending on the internal design of the valve. Valves from the same basic design (for example, butterfly valves) may have varying F_L factors depending on the unique internal designs of the manufacturer. Generally, rotary valves, especially ball and butterfly valves, allow for a high recovery of the fluid following the vena contracta. Therefore, they tend to cavitate and choke at smaller pressure drops than globe valves. For the most part, globe valves have better F_L factors and are able to handle severe services when compared to rotary valves.

7.2.10 Liquid Critical-Pressure Ratio Factor

The *liquid critical-pressure ratio factor* F_F is important to liquid sizing because it predicts the theoretical pressure at the vena contracta, when the maximum effective pressure drop (or in other words, the choked pressure drop) occurs across the valve.

7.2.11 Choked Flow

With liquid services, the presence of cavitation or flashing expands the specific volume of the fluid. The volume increases at a faster rate than if the flow increased due to the pressure differential. At this point, the valve cannot pass any additional flow, even if the downstream pressure is lowered.

With gas and vapor services, choked flow occurs when the velocity of the fluid achieves sonic levels (Mach 1 or greater) at any point in the valve body or downstream piping. Following the basic laws of mass and energy, as the pressure decreases in the valve to pass through restrictions, velocity increases inversely. As the pressure lowers, the specific volume of the fluid increases to the point where a sonic velocity is achieved.

Because of the velocity limitation [Mach 1 for gases and 50 ft/s (12.7 m/s) for liquids], the flow rate is limited to that which is permitted by the sonic velocity through the vena contracta or the downstream piping.

7.2.12 Velocity

As a general rule, smaller valve sizes are better equipped to handle higher velocities than larger-sized valves, although the actual sizes vary according to the valve style. For liquid services, the general guideline for maximum velocity at the valve outlet is 50 ft/s (12.7 m/s), while gas services are generally restricted to Mach 1.0. When cavitation or flashing is present, creating a higher velocity associated with the liquid–gas mixture, the maximum velocity is usually restricted to 500 ft/s (127 m/s). Some exceptions exist, however, for liquids. In services where temperatures are close to saturation point, the velocity must be less—approximately 30 ft/s (7.6 m/s). This lower velocity prevents the fluid from dropping below the vapor pressure, which will lead to the formation of vapor bubbles. The rule of 30 ft/s (7.6 m/s) is also applicable to those valve applications that must have a full flow rate with minimal pressure drop. A valve in which the pressure drop falls below the vapor pressure and advanced cavitation is occurring should be restricted to 30 ft/s (7.6 m/s) to minimize the cavitation damage that would spread from the valve into the downstream piping. Ideally, the user would try to restrict the pressure recovery and allow the subsequent cavitation damage to be contained in the body and not downstream into the piping. In essence, the valve body is sacrificed and the piping is saved.

7.2.13 Reynolds-Number Factor

Some processes are characterized by nonturbulent flow conditions in which laminar flow exists (such as oils). Laminar fluids have high viscosity, operate in lower velocities, or require a flow capacity requirement that is extremely small. The *Reynolds-number factor* F_R is used to correct the C_v equation for these flow factors. In most cases, if the viscosity is fairly low (for example, less than SAE 10 motor oil), the Reynolds-number factor is insignificant.

7.2.14 Piping-Geometry Factor

The flow capacity of a valve may be affected by nonstandard piping configurations, such as the use of increasers or reducers, which must be corrected in the C_v equation using the *piping-geometry factor* F_P.

Standardized C_v testing is conducted by the valve manufacturer with straight piping that is the same line size as the valve. The use of piping that is larger or smaller than the valve, or the close proximity of piping elbows, can decrease these values and must be considered during sizing.

7.2.15 Expansion Factor

With gas services, the specific weight of the fluid varies as the gas moves from the upstream piping and through the valve to the vena contracta. The *expansion factor Y* is used to compensate for the effects of this change in the specific weight of the gas. The expansion factor is important in that it takes into account the changes in the cross-sectional area of the vena contracta as the pressure drop changes in that region.

7.2.16 Ratio of Specific Heats Factor

Because the C_v equation for gases is based upon air, some adjustment must be made for other gases. The *ratio of specific heats factor* F_K is used to adjust the C_v equation to the individual characteristics of these gases.

7.2.17 Terminal Pressure-Drop Ratio

With gases, the point where the valve is choked (which means that increasing the pressure drop though lowering the downstream pressure cannot increase the flow of the valve) is predicted by the *terminal pressure-drop ratio* x_T. Similar in many respects to the liquid pressure-recovery factor F_L, the terminal pressure-drop ratio is affected by the geometry of the valve's body and varies according to valve style and individual size.

7.2.18 Compressibility Factor

Because the density of gases varies according to the temperature and pressure of the fluid, the fluid's compressibility must be included in the C_v equation. Therefore the *compressibility factor Z* is included in the equation and is a function of the temperature and pressure.

7.3 Body Sizing of Liquid-Service Control Valves

7.3.1 Basic Liquid Sizing Equation

The liquid C_v sizing equation is a general-purpose equation for most liquid applications, using the actual pressure drop (upstream pressure minus downstream pressure) to calculate the flow capacity. For non-laminar liquids,

$$C_v = \frac{Q}{F_P}\sqrt{\frac{S_g}{\Delta P_a}}$$

where C_v = valve-sizing coefficient
F_P = piping geometry function
Q = flow rate (gal/min)
S_g = specific gravity (at flowing temperature)
ΔP_a = allowable pressure drop across the valve (psi)

For sizing purposes, the liquid C_v equation can be determined step by step by following Secs. 7.3.2 through 7.3.14.

7.3.2 Actual-Pressure-Drop Calculation

Before the allowable pressure drop is determined, the actual pressure drop should be determined by using the following equation:

$$\Delta P = P_1 - P_2$$

where ΔP = actual pressure drop (psi)
P_1 = upstream pressure (at valve inlet, psia)
P_2 = downstream pressure (at valve outlet, psia)

7.3.3 Choked Flow, Cavitation, and Flashing Determination

The choked flow point must be predicted using the following equation:

$$\Delta P_{\text{choked}} = F_L^2(P_1 - F_F P_V)$$

where ΔP_{choked} = choked pressure drop
F_L = liquid pressure-recovery factor
F_F = liquid critical-pressure-ratio factor
P_V = vapor pressure of the liquid (at inlet temperature, psia)

The liquid pressure-recovery factor F_L is usually provided by the manufacturer. Table 7.1 provides typical F_L values for throttling linear globe and rotary valves.

Table 7.1 Typical F_L Factors*,†

Valve Style	Flow Direction	Trim Area	F_L
Linear globe	Over seat	Full area	0.85
	Over seat	Reduced area	0.80
	Under seat	Full area	0.90
	Under seat	Reduced area	0.90
Butterfly	60° open	Full area	0.76
	90° open	Full area	0.56
Ball	60° open	Full area	0.78
	90° open	Full area	0.66

**Data courtesy of Valtek International.*

†*Note:* All values provided are full-open.

Continuing with the ΔP_{choked} equation, the liquid critical-pressure ratio factor F_F is determined by using the following equation:

$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}}$$

where P_C = critical pressure of the liquid (psia)

Critical pressures for common gases and liquids are found in Table 7.2.

If the calculation for the choked pressure drop ΔP_{choked} is a smaller value than the actual pressure drop ΔP, the ΔP_{choked} value should be used for the actual pressure drop ΔP_a in the C_v equation. To determine at what pressure drop advanced cavitation begins, the following equation should be used:

$$\Delta P_{\text{cavitation}} = F_i^2(P_1 - P_V)$$

Table 7.2 Critical Pressures for Common Process Fluids

Liquid	Critical Pressure *psia/bar*
Ammonia	1636.1/112.8
Argon	707.0/48.8
Benzene	710.0/49.0
Butane	551.2/38.0
Carbon Dioxide	1070.2/73.8
Carbon Monoxide	507.1/35.0
Chlorine	1117.2/77.0
Dowtherm A	547.0/37.7
Ethane	708.5/48.8
Ethylene	730.5/50.3
Fuel Oil	330.0/22.8
Gasoline	410.0/28.3
Helium	32.9/2.3
Hydrogen	188.1/13.0
Hydrogen Chloride	1205.4/83.1
Isobutane	529.2/36.5
Isobutylene	529.2/36.5
Kerosene	350.0/24.1
Methane	667.3/46.0
Nitrogen	492.4/33.9
Nitrous Oxide	1051.1/72.5
Oxygen	732.0/50.5
Phosgene	823.2/56.8
Propane	615.9/42.5
Propylene	670.3/46.2
Refrigerant 11	639.4/44.1
Refrigerant 12	598.2/41.2
Refrigerant 22	749.7/51.7
Seawater	3200.0/220.7
Water	3208.2/221.2

Table 7.3 Typical F_i Factors*,†

Valve Style	Flow Direction	Trim Area	F_i
Linear globe	Over seat	Full area	0.75
	Over seat	Reduced area	0.72
	Under seat	Full area	0.81
	Under seat	Reduced area	0.81
Butterfly	60° open	Full area	0.65
	90° open	Full area	0.49
Ball	60° open	Full area	0.65
	90° open	Full area	0.44

**Data courtesy of Valtek International.*

†*Note:* All values provided at full open.

where $\Delta P_{\text{cavitation}}$ = pressure drop with advanced cavitation
F_i = liquid cavitation factor

Typical liquid cavitation factors for common valve styles are found in Table 7.3.

7.3.4 Specific-Gravity Determination

The value for the fluid's specific gravity S_g should be determined using the operating temperature and a reference table for specific-gravity data.

7.3.5 Approximate-Flow-Coefficient Calculation

Using the values calculated to this point, the approximate flow capacity should be calculated, using the C_v sizing equation for liquids from

Sec. 7.3.1. For this calculation, the assumption should be made that the piping-geometry factor F_P is 1.0. When the valve is not operating in a laminar flow—due to high viscosity, low velocity, or low flow—the effects of nonturbulent flow can be ignored.

7.3.6 Approximate Body Size Selection

Using the manufacturer's C_v tables, the smallest-sized body that can accommodate the calculated C_v should be selected. Typical C_v data are found in Fig. 7.2.

7.3.7 Reynolds-Number-Factor Calculation

The following equation can be used to determine the Reynolds-number factor:

$$Re_V = \frac{N_4 F_d Q}{\nu\sqrt{F_L C_v}}\left(\frac{F_L{}^2 C_v{}^2}{N_2 d^4} + 1\right)^{0.25}$$

where Re_V = valve Reynolds number
N_4 = 17,300 (when Q is in gal/min and d in inches)
F_d = valve style modifier (see Table 7.4)
ν = kinematic viscosity (centistokes, μ/S_g)
C_v = valve flow coefficient (from Sec. 7.3.1)
N_2 = 890 (when d is in inches)
d = valve inlet diameter (inches)

Valve Type: Mark One, Unbalanced
Body Rating: Class 900-1500
Trim Characteristics: Quick Open
Flow Direction: Flow Over

SIZE	TRIM NO	STROKE	F_L	PERCENT OPEN 100	90	80	70	60	50	40	30	20	10	SEAT AREA
1.00	.81	.75	.87	9.0	8.9	8.9	8.7	8.6	8.5	7.5	5.7	3.5	1.9	.52
1.50	1.25	1.00	.85	24	24	24	24	24	21	18	13	8.7	4.9	1.23
2.00	1.62	1.50	.87	41	41	40	40	39	39	34	26	15	8.1	2.06
3.00	2.00	2.00	.86	106	105	105	104	104	94	81	62	39	21	5.41
4.00	3.50	2.50	.87	185	185	183	181	178	162	139	105	68	37	9.62
6.00	5.00	3.00	.85	382	382	381	380	355	317	270	210	140	75	19.63

Note: *All Cv values are shown at 100% open. For each vavle size below, the full area values are shaded. Reduced trim values follow, in descending order.*

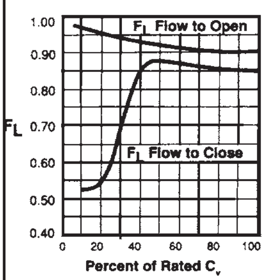
Figure 7.2. Typical manufacturer's C_v data. (*Courtesy of Valtek International*)

Table 7.4 Valve Recovery Coefficient and Incipient Cavitation Factors*

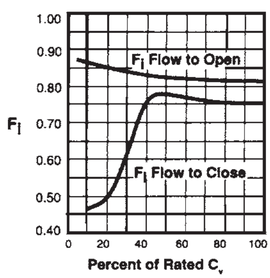
Valve Type	Flow Direction	Trim Size	F_L	F_i	x_T	F_d
Globe	Over Seat	Full Area	0.85	0.75	.70	1.0
	Over Seat	Reduced Area	0.80	0.72	.70	1.0
	Under Seat	Full Area	0.90	0.81	.75	1.0
	Under Seat	Reduced Area	0.90	0.81	.75	1.0
Valdisk Rotary Disc	60° Open	Full	0.76	0.65	.36	.71
	90° Open	Full	0.56	0.49	.26	.71
ShearStream Rotary Ball	60° Open	Full	0.78	0.65	.51	1.0
	90° Open	Full	0.66	0.44	.30	1.0
CavControl	Over Seat	All	0.92	0.90	N/A	$.2/\sqrt{d}$
MegaStream	Under Seat	All	~1.0	N/A	~1.0	$(n_s/25d)^{2/3}$**
ChannelStream	Over Seat	All	~1.0	0.87 to 0.999	N/A	.040*
Tiger-Tooth	Under Seat	All	~1.0	0.84 to 0.999	~1.0	.035*

* Typical ** n_s = number of stages

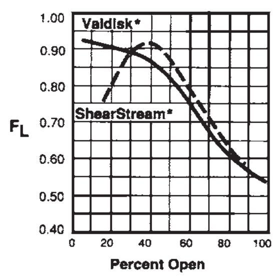
NOTE: Values are given for full-open valves. See charts below for part-stroke values



Globe Valve F_L Values



Globe Valve F_i Values



Rotary Disc Valve F_L Values

Courtesy of Valtek International.

If the valve Reynolds number (Re_V) is equal to or greater than 40,000 ($Re_V \geq 40{,}000$), 1.0 should be used for the Reynolds-number factor F_R. The following equation is used to find F_R, if the valve Reynolds number is less than 40,000 ($Re_V \leq 40{,}000$):

$$F_R = 1.044 - 0.358\left(\frac{C_{vS}}{C_{vT}}\right)^{0.655}$$

through use of these equations:

$$C_{vS} = \frac{1}{F_S}\left(\frac{Q\mu}{N_S \Delta P}\right)^{0.667}$$

$$F_S = \frac{F_d^{\,0.667}}{F_L^{\,0.333}}\left(\frac{F_L^{\,2} C_v^{\,2}}{N_2 d^4} + 1\right)^{0.167}$$

where C_{vS} = laminar flow C_v
C_{vT} = turbulent flow C_v (the C_v is used from the liquid C_v equation in Sec. 7.3.1)
F_S = laminar or streamline flow factor
N_2 = 890 (when d is in inches)
N_4 = 17,300 (when Q is in gal/min and d is in inches)
N_S = 47 (when Q is in gal/min and ΔP is in psi)
μ = absolute viscosity (centipoise)

7.3.8 Flow-Coefficient Recalculation

Flow is considered to be laminar when the Reynolds-number factor F_R is less than 0.48 ($F_R < 0.48$). That means that the C_v is the same as the C_{vS}, which is determined from the equation in Sec. 7.3.7.

If F_R is larger than 0.98 ($F_R > 0.98$), the flow is determined to be turbulent and assumed to be equal to 1.0 ($F_R = 1.0$). At this point, the C_v is determined from the standard C_v liquid sizing equation found in Sec. 7.3.1. The piping-geometry factor F_P is not required in this situation and should not be figured into the C_v equation.

If F_R falls between 0.48 and 0.98, the flow is determined to be in a transitional stage, which is calculated using the following equation:

$$C_v = \frac{Q}{F_R}\sqrt{\frac{S_g}{\Delta P}}$$

where F_R = Reynolds-number factor
S_g = specific gravity (at flowing temperature)

7.3.9 Piping-Geometry-Factor Calculation

The inside diameter of the piping is required to determine the piping-geometry factor F_P. In the event that the pipe size is not provided or known, the body size determined from Sec. 7.3.6 should be used to determine the pipe size. Tables 7.5 and 7.6 can be used to find the piping-geometry factors. Table 7.7 provides F_P for valves with reducers (or increasers) on both the inlet and outlet of the valve. Table 7.8 provides F_P for a valve with the reducer (or increaser) on the valve outlet only. The maximum effective pressure drop (defined as ΔP_{choked}) can be affected by the use of increasers and reducers.

Table 7.5 Piping-Geometry Factors for Valves with Reducers and Increasers on Both Ends*,†

C_v/d^2	d / D				
	0.50	0.60	0.70	0.80	0.90
4	0.99	0.99	1.00	1.00	1.00
6	0.98	0.99	0.99	1.00	1.00
8	0.97	0.98	0.99	0.99	1.00
10	0.96	0.97	0.98	0.99	1.00
12	0.94	0.95	0.97	0.98	1.00
14	0.92	0.94	0.96	0.98	0.99
16	0.90	0.92	0.95	0.97	0.99
18	0.87	0.90	0.94	0.97	0.99
20	0.85	0.89	0.92	0.96	0.99
25	0.79	0.84	0.89	0.94	0.98
30	0.73	0.79	0.85	0.91	0.97
35	0.68	0.74	0.81	0.89	0.96
40	0.63	0.69	0.77	0.86	0.95

**Courtesy of Valtek International.*

†*Note:* The maximum effective pressure drop (ΔP choked) may be affected by the use of reducers and increasers. This is especially true of butterfly valves.

7.3.10 Final-Flow-Coefficient Calculation

After the piping-geometry factor F_P is determined, it should be applied to the liquid C_v equation (Sec. 7.3.1) and the final C_v calculated.

7.3.11 Valve Exit-Velocity Calculation

As discussed in Sec. 7.2.12, the general rule for velocities in liquids is that the velocity should be limited to 50 ft/s (15.2 m/s), although this may vary according to the size of the valve—smaller valves can handle

Table 7.6 Piping-Geometry Factors for Valves with Reducers and Increasers on Outlet Only*,†

C_v / d^2	d / D				
	0.50	0.60	0.70	0.80	0.90
4	1.00	1.00	1.00	1.00	1.00
6	1.01	1.01	1.01	1.01	1.01
8	1.01	1.02	1.02	1.02	1.01
10	1.02	1.03	1.03	1.03	1.02
12	1.03	1.04	1.04	1.04	1.03
14	1.04	1.05	1.06	1.05	1.04
16	1.06	1.07	1.08	1.07	1.05
18	1.08	1.10	1.11	1.10	1.06
20	1.10	1.12	1.12	1.12	1.08
25	1.17	1.22	1.24	1.22	1.13
30	1.27	1.37	1.42	1.37	1.20
35	1.44	1.65	1.79	1.65	1.32
40	1.75	2.41	3.14	2.41	1.50

**Courtesy of Valtek International.*

†*Note:* d = valve port inside diameter in inches; D = internal diameter of the piping in inches.

higher velocities, while larger valves handle lower velocities. To calculate the exit velocities from the valve, the following equation is used:

$$V = \frac{0.321Q}{A_V}$$

where V = velocity (ft/s)

A_V = flow area of valve body port (square inches) from Table 7.9

If the exit velocity exceeds the acceptable velocity for that given application, a larger valve size may be chosen to prevent damage from erosion. If a larger body size is chosen, the piping-geometry factor F_P will have to change, requiring a new C_v calculation.

Table 7.7 Piping-Geometry Factors, with Reducers or Increasers on Both Inlet and Outlet of Valve*,†

C_v/d^2	0.50 d/D	0.60 d/D	0.70 d/D	0.80 d/D	0.90 d/D
4	0.99	0.99	1.00	1.00	1.00
6	0.98	0.99	0.99	1.00	1.00
8	0.97	0.98	0.99	0.99	1.00
10	0.96	0.97	0.98	0.99	1.00
12	0.94	0.95	0.97	0.98	1.00
14	0.92	0.94	0.96	0.98	0.99
16	0.90	0.92	0.95	0.97	0.99
18	0.87	0.90	0.94	0.97	0.99
20	0.85	0.89	0.92	0.96	0.99
25	0.79	0.84	0.89	0.94	0.98
30	0.73	0.79	0.85	0.91	0.97
35	0.68	0.74	0.81	0.89	0.96
40	0.63	0.69	0.77	0.86	0.95

**Data courtesy of Valtek International.*
†*Note:* d = inside diameter of valve port (inches); D = inside diameter of piping (inches).

7.3.12 Trim-Size Selection

Control-valve manufacturers provide tables that outline the C_vs for a certain valve style, flow direction, body pressure rating, flow characteristic, size of the valve seat or the seal, and length of stroke. Some charts may be broken down to percentages of opening, since some throttling services may not utilize the entire stroke.

Using the manufacturer's C_v table based upon the correct criteria (body size, flow characteristic, flow direction, etc.), the correct size of the valve opening (of the seat or the seal) should be chosen. This open-

Table 7.8 Piping-Geometry Factors with Reducer or Increaser on Outlet of Valve*,†

C_v/d^2	0.50 d/D	0.60 d/D	0.70 d/D	0.80 d/D	0.90 d/D
4	1.00	1.00	1.00	1.00	1.00
6	1.01	1.01	1.01	1.01	1.01
8	1.01	1.02	1.02	1.02	1.01
10	1.02	1.03	1.03	1.03	1.02
12	1.03	1.04	1.04	1.04	1.03
14	1.04	1.05	1.06	1.05	1.04
16	1.06	1.07	1.08	1.07	1.05
18	1.08	1.10	1.11	1.10	1.06
20	1.10	1.12	1.12	1.12	1.08
25	1.17	1.22	1.24	1.22	1.13
30	1.27	1.37	1.42	1.37	1.20
35	1.44	1.65	1.79	1.65	1.32
40	1.75	2.41	3.14	2.41	1.50

**Data courtesy of Valtek International.*

†*Note:* d = inside diameter of valve port (inches); D = inside diameter of piping (inches).

ing and its dimension are often called the *trim number.* A globe-style valve will have a number of trim-number options, including one that is a *full-area trim number,* the largest sized diameter opening for that particular size. The valve may also have several *reduced-area trim numbers,* which are progressively smaller in diameter and allow smaller C_vs in the same body size.

7.3.13 Flashing-Velocity Calculation

As described in Sec. 7.2.8, if the valve outlet pressure is lower than the vapor pressure, the vapor bubbles that are formed remain in a gaseous state, providing a downstream flow that has a combined liquid–gas mixture. This results in increased velocity and difficult control situations. Since the application is found to be flashing, certain measures

Table 7.9 Valve Port Areas*,†

Valve Size (inches)	Valve Outlet Area, A_v (Square Inches)						
	Class 150	Class 300	Class 600	Class 900	Class 1500	Class 2500	Class 4500
1/2	0.20	0.20	0.20	0.20	0.20	0.15	0.11
3/4	0.44	0.44	0.44	0.37	0.37	0.25	0.20
1	0.79	0.79	0.79	0.61	0.61	0.44	0.37
1 1/2	1.77	1.77	1.77	1.50	1.50	0.99	0.79
2	3.14	3.14	3.14	2.78	2.78	1.77	1.23
3	7.07	7.07	7.07	6.51	5.94	3.98	2.78
4	12.57	12.57	12.57	11.82	10.29	6.51	3.98
6	28.27	28.27	28.27	25.97	22.73	15.07	10.29
8	50.27	50.27	48.77	44.18	38.48	25.97	19.63
10	78.54	78.54	74.66	69.10	60.13	41.28	28.27
12	113.10	113.10	108.43	97.12	84.62	58.36	41.28
14	137.89	137.89	130.29	117.86	101.71	70.88	50.27
16	182.65	182.65	170.87	153.94	132.73	92.80	63.62
18	233.70	226.98	213.82	194.83	167.87	117.86	84.46
20	291.04	283.53	261.59	240.53	210.73	143.14	101.53
24	424.56	415.48	380.13	346.36	302.33	207.39	143.14
30	671.96	660.52	588.35	541.19	476.06	325.89	
36	962.11	907.92	855.30				
42	1320.25	1194.59					

**Data courtesy of Valtek International.*

†*Note:* To find approximate fluid velocity in the pipe, use the equation $V_P = V_V A_V / A_P$, where V_P = velocity in pipe, A_V = valve outlet area. V_V = velocity in valve outlet, and A_P = pipe area.

To find equivalent diameters of the valve or pipe inside diameter use $d = \sqrt{4A_V/\pi}$, $D = \sqrt{4A_P/\pi}$.

must be taken to prevent undue damage and premature wear to the valve, such as using special trims or hardened materials. Flashing applications must be limited to a velocity of 500 ft/s (152 m/s), unless special modifications are made to the valve-body design to accommodate the increased volume and velocity. Either of the following equations can be used to calculate flashing velocity, depending on the flow-rate measurement (lb/h or gal/min):

$$V = \frac{0.040}{A_V} w \left[\left(1 - \frac{x}{100\%} \right) V_{f2} + \frac{x}{100\%} V_{g2} \right]$$

$$V = \frac{20}{A_V} Q \left[\left(1 - \frac{x}{100\%} \right) V_{f2} + \frac{x}{100\%} V_{g2} \right]$$

where V = velocity (ft/s)
w = liquid flow rate (lb/h)
V_{f2} = saturated liquid specific volume (ft^3/lb at outlet pressure P_2)
V_{g2} = saturated vapor specific volume (ft^3/lb at outlet pressure P_2)
x = percentage of liquid mass flashed to vapor (Sec. 7.3.14)

7.3.14 Percentage of Flashing Calculation

To calculate the percentage of the liquid flashing into gas, the user should have access to steam tables, which provides a listing of enthalpies and specific volumes. To make this calculation, the following equation should be used:

$$x = \left(\frac{h_{f1} - h_{f2}}{h_{fg2}}\right) \times 100\%$$

where h_{f1} = enthalpy of saturated liquid at inlet temperature
h_{f2} = enthalpy of saturated liquid at outlet pressure
h_{fg2} = enthalpy of evaporation at outlet pressure

7.3.15 Liquid Sizing Example A

For this example, the following service conditions are given in Imperial units:

Liquid	Water
Critical pressure P_C	3206.2 psia
Temperature	250°F
Upstream pressure P_1	314.7 psia
Downstream pressure P_2	104.7 psia
Flow rate Q	500 gal/min
Vapor pressure P_V	30 psia
Specific gravity S_g	0.94
Kinematic viscosity v	0.014 cS
Pipeline size	4 in (ANSI Class 600)
Valve	Globe, flow-to-open
Flow characteristic	Equal percentage

The actual pressure drop ΔP is calculated using the C_v equation for liquids (Sec. 7.3.1):

$$\Delta P = P_1 - P_2 = 314.7 \text{ psia} - 104.7 \text{ psia} = 210 \text{ psi}$$

Choked flow can be checked by finding the liquid pressure-recovery factor F_L from Table 7.4, which is 0.90. Then, the liquid critical-pressure ratio factor (F_F) is calculated by using the equation found in Sec. 7.3.3.

$$F_F = 0.96 - 0.28\sqrt{\frac{P_V}{P_C}} = 0.96 - 0.28\sqrt{\frac{30}{3206.2}} = 0.93$$

After determining F_L and F_F, these numbers are used in the choked pressure drop (ΔP_{choked}) equation from Sec. 7.3.3:

$$\Delta P_{\text{choked}} = F_L^2\,(P_1 - F_F P_V) = (0.90)^2[314.7 - (0.93)(30)] = 232 \text{ psi}$$

A comparison should be made between the actual pressure drop ΔP of 210 psi and the choked pressure drop ΔP_{choked} of 232 psi. Since the actual pressure drop is smaller than the choked pressure drop, the actual pressure drop will be used to size the valve.

By using the equation in Sec. 7.3.3, the advent of incipient cavitation should be checked:

$$\Delta P_{\text{cavitation}} = F_L^2(P_1 - P_V) = (0.81)^2(314.7 - 30) = 187 \text{ psi}$$

In this example, the actual pressure drop (ΔP) of 210 psi is greater than the pressure drop associated with incipient cavitation ($\Delta P_{\text{cavitation}}$) of 187 psi. This can be interpreted to mean that, although cavitation is occurring in the service, the cavitation is not causing the flow to choke. In this case, the user should begin considering methods to deter the cavitation damage, such as special trims or hardened materials. With a specific gravity of 0.94 and assuming the piping-geometry factor F_P is 1.0 (Sec. 7.3.9), the C_v should be calculated using the original liquid sizing equation (Sec. 7.3.1):

$$C_v = \frac{Q}{F_P}\sqrt{\frac{S_g}{\Delta P_a}} = \frac{500}{1}\sqrt{\frac{0.94}{210}} = 33.4$$

The required valve is a globe valve with flow-under-the-plug trim design, equal-percentage flow characteristic, and ANSI Class 600 pressure class. The manufacturer's C_v tables should be examined to determine the smallest valve available that would allow the flow of $33.4C_v$ through the flow area of the seat or seal. In this case, the assumption is made that, according to the charts, a 2-in valve body would be the smallest size with a trim number available to pass the required C_v.

At this point, the Reynolds-number factor F_R is calculated by using the equation from Sec. 7.3.7:

$$\mathrm{Re}_V = \frac{N_4 F_d Q}{v\sqrt{F_L C_v}}\left(\frac{F_L{}^2 C_v{}^2}{N_2 d^4} + 1\right)^{0.25}$$

$$= \frac{(17{,}300)(1)(500)}{(0.014)\sqrt{(0.90)(33.4)}}\left(\frac{(0.\,90)^2(33.4)^2}{(890)(2)^2} + 1\right)^{0.25}$$

$$= 114 \times 10^6$$

Because the Reynolds-number factor F_R is significantly larger than 40,000 (114×10^6 versus 40,000), the calculated C_v remains 33.4 and is used in further calculations. With a 2-in body tentatively chosen for this application and a 4-in pipeline, the calculation of the piping-geometry factor F_P is made using Table 7.5 with the following numbers:

$$\frac{d}{D} = \frac{2}{4} = 0.5$$

and

$$\frac{C_v}{d^2} = \frac{33.4}{2^2} = 8.35$$

According to the table, the piping-geometry factor (F_P) should be 0.97. Now, the F_P of 0.97 can be inserted into the C_v equation to determine the final C_v:

$$C_v = \frac{Q}{F_P}\sqrt{\frac{S_g}{\Delta P_a}} = \frac{500}{0.97}\sqrt{\frac{0.94}{210}} = 34.5$$

Using Table 7.9, for a 2-in valve in ANSI Class 600 service, the valve outlet area A_V is 3.14 in^2. Using this number and a flow rate of 500 gal/min (1892 liters/m), the velocity through the valve can be calculated as

$$V = \frac{0.321Q}{A_V} = \frac{0.321(500)}{3.14} = 51 \text{ ft/s (130 m/s)}$$

The velocity of 51 ft/s exceeds the limit of 50 ft/s for liquids. Since the service is cavitating, damage will most likely occur to the valve body. At this point, the only option to lower the velocity is to chose the next larger valve size, a 3-in body with reduced trim. Using a 3-in body and an A_V of 7.07, the velocity is significantly lowered to acceptable levels:

$$V = \frac{0.321Q}{A_V} = \frac{0.321(500)}{7.07} = 23 \text{ ft/s (5.8 m/s)}$$

Despite the lower velocity with the 3-in body, cavitation remains a concern and some material or design action should be taken to prevent damage. Another option that may reduce the cost of a larger valve would be to use an expanded outlet body—for example, a 2 × 4-in expanded outlet valve (since the piping is 4 in). Because of the velocity issue, which required the changing of the valve size to 3 in, the C_v equation will need to be recalculated using a new piping-geometry factor F_P:

$$\frac{d}{D} = \frac{3}{4} = 0.75$$

and

$$\frac{C_v}{d^2} = \frac{33.4}{3^2} = 3.71$$

With a piping-geometry factor F_P of 1.00 (interpolated from Table 7.5), the revised C_v for a 3-in body is

$$C_v = \frac{Q}{F_P}\frac{S_g}{\Delta P_a} = \frac{500}{1}\frac{0.94}{210} = 33.4$$

7.3.16 Liquid Sizing Example A (with Flashing)

For this example, the same service conditions as the previous example are provided, except that the temperature is increased by 100°F from 250 to 350°F. Using the saturated steam temperatures in the steam tables, the saturation pressure for water at 350°F is 134.5 psia. Because the saturation pressure (134.5 psia) is significantly higher than the downstream pressure of the valve (104.7 psia), the service is flashing. Because of the flashing, the percent flash x must be calculated:

$$x = \left(\frac{h_{f1} - h_{f2}}{h_{fg2}}\right) \times 100\% = \left(\frac{321.8 - 302.3}{886.4}\right) \times 100\% = 2.2\%$$

where h_{f1} = 321.8 Btu/lb at 350°F (from the saturation temperature table)
h_{f2} = 302.3 Btu/lb at 105 psia (from the saturation pressure table)
h_{fg2} = 886.4 Btu/lb at 105 psia (from the saturation pressure table)

The equation from Sec. 7.3.13 must then be used to determine the velocity from a 3-in valve:

$$V = \frac{20}{A_V} Q\left[\left(1 - \frac{x}{100\%}\right)V_{f2} + \frac{x}{100\%}\, V_{g2}\right]$$

$$V = \frac{(20)(500)}{7.07}\left[\left(1 - \frac{2.2\%}{100\%}\right)0.0178 + \frac{2.2\%}{100\%}\, 4.234\right] = 156 \text{ ft/s}$$

where V_{f2} = 0.0178 ft^3/lb at 105 psia (from the saturation pressure table)
V_{g2} = 4.324 ft^3/lb at 105 psia (from the saturation pressure table)

From Sec. 7.2.12, the maximum velocity for flashing services is 500 ft/s. The calculated velocity of this service is 156 ft/s, which is far below the maximum level. Once again, however, the presence of flashing should be considered by selecting hardened materials or special trim features.

7.3.17 Liquid Sizing Example B

In this second liquid example, the following service conditions are provided in Imperial units:

Liquid	Ammonia
Critical pressure P_C	1638.2 psia

Temperature	20°F
Upstream pressure P_1	149.7 psia
Downstream pressure P_2	64 psia
Flow rate Q	850 gal/min
Vapor pressure P_V	465.6 psia
Specific gravity S_g	0.65
Kinematic viscosity ν	0.02 cS
Pipeline size	3 in (ANSI Class 600)
Valve	Globe, flow-to-close
Flow characteristic	Linear

The actual pressure drop ΔP is calculated as follows:

$$\Delta P_a = P_1 - P_2 = 149.7 \text{ psia} - 64.7 \text{ psia} = 85 \text{ psi}$$

Choked flow is checked by determining the liquid pressure-recovery factor F_L from Table 7.4, which is 0.85. The liquid critical-pressure ratio factor F_F can then be calculated by using the following equation found in Sec. 7.3.3.

$$F_F = 0.96 - 0.28 \frac{P_V}{P_C} = 0.96 - 0.28 \sqrt{\frac{45.6}{1638.2}} = 0.91$$

After determining that F_L (= 0.85) and F_F (= 0.91), these numbers are inserted in the choked-pressured-drop ΔP_{choked} equation from Sec. 7.3.3:

$$\Delta P_{\text{choked}} = F_L^2(P_1 - F_F P_V) = (0.85)^2[149.7 - (0.91)(45.6)] = 78.2 \text{ psi}$$

In comparing the actual pressure drop ΔP of 85.0 psi and the choked pressure drop ΔP_{choked} of 78.2 psi, the choked pressure drop is smaller than the actual pressure drop. Therefore, the smaller of the two numbers—the choked pressure drop—is used to size the valve. Because the valve is choked, the service is also cavitating. Therefore, checking for incipient cavitation $\Delta P_{\text{cavitation}}$ is not necessary. In this case, the user should plan to use special anticavitation trim inside the valve as well as hardened materials to avoid the erosion of metal parts associated with cavitation.

With a specific gravity of 0.65 and assuming a piping-geometry factor F_P of 1.0 (Sec. 7.3.9), a preliminary C_v can be calculated using the original liquid sizing equation (Sec. 7.3.1):

$$C_v = \frac{Q}{F_P}\sqrt{\frac{S_g}{\Delta P_a}} = \frac{850}{1}\sqrt{\frac{0.65}{78.2}} = 77.5$$

From the conditions of this example, the preferred valve is a globe valve with flow-over-the-plug trim design, a linear flow characteristic, and ANSI Class 600 pressure classification. The manufacturer's C_v tables can then be examined to estimate the smallest valve available that would allow the flow of $77.5C_v$ through the flow area of the seat. In this case, the assumption is made that the manufacturer's C_v tables show that a 3-in valve body would be the smallest size with a trim number that would pass the required C_v.

Because the flow is cavitating, it is turbulent when exiting the valve. Because of the turbulent flow, the Reynolds-number factor F_R is assumed to be $F_R = 1.0$ and no further calculations are required.

Since a 3-in body was chosen initially for this application and the pipeline is determined to be a 3-in line, the piping-geometry factor F_P will be 1.0 (no reducers or increasers are required). Because $F_P = 1.0$, the C_v calculation made earlier does not change because of the piping geometry and remains at 77.5.

Using Table 7.9, for a 3-in valve in ANSI Class 600 service, the valve outlet area A_V is 7.07 in^2. Using this number and a flow rate of 850 gal/min, the velocity through the valve can be calculated as

$$V = \frac{0.321Q}{A_V} = \frac{0.321(850)}{7.07} = 39 \text{ ft/s}$$

The velocity of 39 ft/s is below the limit of 50 ft/s for liquids. Therefore, a 3-in body is acceptable for this application, although the cavitating service will need to be dealt with through modifications to the valve, such as special trim or hardened materials.

7.4 Body Sizing of Gas-Service Control Valves

7.4.1 Basic Gas Sizing Equations

The basic difference between liquid sizing and gas sizing deals with the compressibility of gases. Because of their compressibility, gases have a tendency to expand as the pressure drop occurs through the vena contracta. In turn, this lowers the specific weight of the gas. This changing specific weight must be taken into account during the sizing process using a special factor called the *expansion factor Y.*

Depending on the given service conditions or variables, one of four gas sizing equations is used. The numerical constants included in each equations deal with unit conversion factors.

$$w = 63.3 F_P C_v Y \sqrt{x P_1 \gamma_1}$$

$$Q = 1360 F_P C_v P_1 Y \sqrt{\frac{x}{G_g T_1 Z}}$$

$$w = 19.3 F_P C_v P_1 Y \sqrt{\frac{x M_w}{T_1 Z}}$$

$$Q = 7320 F_P C_v P_1 Y \sqrt{\frac{x}{M_W T Z}}$$

where w = gas flow rate (lb/h)
F_P = piping-geometry factor
C_v = valve sizing coefficient
Y = expansion factor
x = pressure-drop ratio
γ_1 = specific weight at inlet service conditions (lb/ft^3)
Q = gas flow (scfh)
G_g = specific gravity or gas relative to air at standard conditions
T_1 = absolute upstream pressure (°R = °F + 460)
Z = compressibility factor
M_W = molecular weight
P_1 = upstream absolute pressure (psia)

One of the four gas sizing equations should be selected based on the available data for the given service conditions.

7.4.2 Choked-Flow Determination

The terminal pressure-drop ratio x_T is determined by taking the appropriate value from Table 7.10. The ratio of specific heats factor F_K can be calculated by using the following equation:

$$F_K = \frac{k}{140}$$

Table 7.10 Typical x_T Factors*,†

Valve Style	Flow Direction	Trim Area	X_T
Linear globe	Over seat	Full area	0.70
	Over seat	Reduced area	0.70
	Under seat	Full area	0.75
	Under seat	Reduced area	0.75
Butterfly	60° open	Full area	0.36
	90° open	Full area	0.26
Ball	60° open	Full area	0.51
	90° open	Full area	0.30

**Data courtesy of Valtek International.*
†*Note:* All values provided at full-open.

where F_K = ratio of specific heats factor
k = ratio of specific heats

The ratio k of specific heats can be found for common gases in Table 7.11, which is provided for quick reference.

The ratio x of actual pressure drop to absolute inlet pressure is determined by using the following equation:

$$x = \frac{\Delta P_a}{P_1}$$

where x = ratio of actual pressure drop to absolute inlet pressure
ΔP = actual pressure drop (psi)
P_1 = upstream pressure (at inlet, psia)
P_2 = downstream pressure (at outlet, psia)

If the value for x is less than the value for $F_K x_T$, choked flow is not occurring. Inversely, when x reaches or exceeds the value of $F_K x_T$, the

Table 7.11 Physical Data for Common Gas Services

Gas	Molecular Weight (M_w)	Critical Temperature (°R*)	Critical Pressure (psia/bar)	Ratio of Specific Heats (k)
Air	28.97	227°	492/33.9	1.40
Ammonia	17.00	730°	1636/112.8	1.31
Argon	39.95	271°	707/48.8	1.67
Carbon Dioxide	44.01	547°	1070/73.8	1.29
Carbon Monoxide	28.01	239°	507/35.0	1.40
Ethane	30.07	549°	709/48.9	1.19
Ethylene	28.10	508°	731/50.4	1.24
Helium	4.00	9°	33/2.3	1.66
Hydrogen	2.02	59°	188/13.0	1.40
Methane	16.04	343°	667/46.0	1.31
Natural Gas	16.04	343°	667/46.0	1.31
Nitrogen	28.00	227°	492/33.9	1.40
Oxygen	32.00	278°	732/50.5	1.40
Propane	44.10	665°	616/42.5	1.31
Steam	18.02	1165°	3208/221.2	1.33

*°R = °F + 460.

flow is choked. If the flow is choked, the value $F_K x_T$ should be used instead of x, if x is used in the chosen gas sizing equation.

7.4.3 Expansion-Factor Calculation

Because of the compressibility of gases, the expansion factor Y must be determined by using the following equation. If choked flow is occurring, the value $F_K x_T$ should be used instead of x.

$$Y = 1 - \frac{x}{3F_K x_T}$$

where Y = expansion factor
x_T = terminal pressure-drop ratio

7.4.4 Compressibility-Factor Determination

The compressibility factor Z is determined by calculating the reduced-pressure value P_r and the reduced-temperature value T_r:

$$P_r = \frac{P_1}{P_c}$$

where P_r = reduced pressure

$$T_r = \frac{T_1}{T_c}$$

where T_r = reduced temperature
T_1 = absolute upstream temperature
T_C = absolute critical temperature

Once the reduced pressure P_r and reduced temperature T_r are known, the compressibility factor Z can be determined with either Fig. 7.3 or 7.4.

7.4.5 Flow-Coefficient Calculation

Using the factors determined to this point, a preliminary C_v is calculated by using the applicable gas sizing C_v equation. For this equation, the piping-geometry factor F_P should be assumed to be 1.0.

7.4.6 Approximate Body-Size Selection

Using the manufacturer's C_v tables, the smallest sized body is selected that can accommodate the calculated preliminary C_v.

7.4.7 Piping-Geometry-Factor Calculation

When the pipeline size has not been determined or is unknown, for calculation purposes the body size that was determined from Sec. 7.4.6 is used as pipeline size. The inside diameter of the piping is required

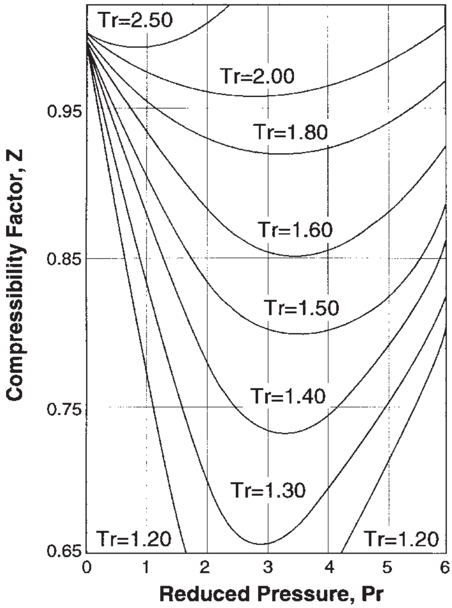


Figure 7.3. Compressibility factors, reduced pressures 0 to 6. (*Courtesy of Valtek International*)

to determine the piping-geometry factor F_P. Tables 7.5 and 7.6 are used to find the piping-geometry factors. Table 7.5 provides F_P for valves with reducers (or increasers) on both the inlet and outlet of the valve. Table 7.6 provides F_P for a valve with the reducer (or increaser) on the valve outlet only.

7.4.8 Final-Flow-Coefficient Calculation

Using the piping-geometry factor F_P, the final C_V is calculated, using one of the four equations provided. Usually, the C_V will be close to the preliminary C_V chosen earlier. Therefore, the body size will most like stay the same, unless high velocities are present.

7.4.9 Valve Exit Mach-Number Calculation

With the flow coefficient known, as well as the body size, the exit velocity of the gas from the valve is determined in Mach numbers. The

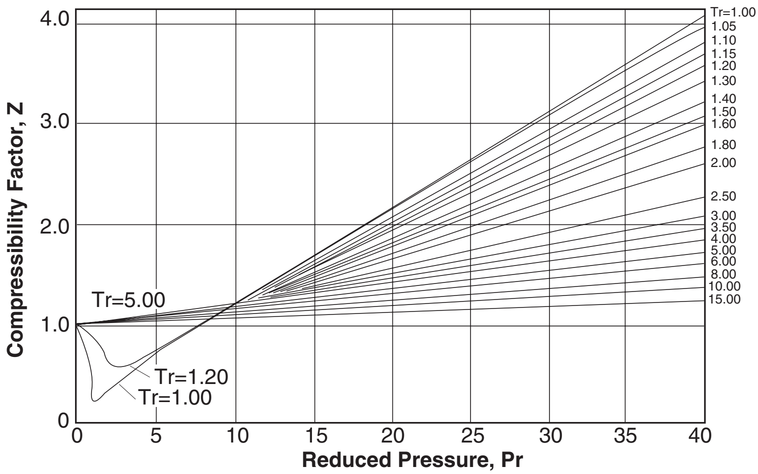


Figure 7.4. Compressibility factors, reduced pressures 0 to 40. (*Courtesy of Valtek International*)

following two equations are used for calculating velocities in gas services:

$$M_{\text{gas}} = \frac{Q_a}{5574 A_V \sqrt{\dfrac{kT}{M_W}}}$$

$$M_{\text{gas}} = \frac{Q_a}{1036 A_V \sqrt{\dfrac{kT}{G_g}}}$$

where M_{gas} = Mach number for gas service
Q_a = actual flow rate (cfh instead of scfh)
A_V = applicable flow area of body port (square inches) from Table 7.9
k = ratio of specific heats
T = absolute temperature (°R or °F + 460)
M_W = molecular weight
G_g = specific gravity at standard conditions relative to air

The following velocity equation is used for air service:

$$M_{\text{air}} = \frac{Q_a}{1225 A_V \sqrt{T}}$$

where M_{air} = Mach number for air service

To convert scfh to cfh, the following equation is used:

$$\frac{P_a Q_a}{T_a} =$$

where P_a = actual operating pressure
Q_a = actual volume flow rate (cfh)
T_a = actual temperature (°R or °F + 460)
P_S = standard pressure (14.7 psi)
Q = standard volume flow rate (scfh)
T_s = standard temperature (520°R)

The following velocity equation is used for steam service:

$$M_{\text{steam}} = \frac{wv}{1514 A_V \sqrt{T}}$$

where M_{steam} = Mach number for air service
w = mass flow rate (lb/h)
v = specific volume at flow conditions (ft^3/lb)

Once the exit velocity has been calculated and is found to exceed Mach 0.5, the possibility of excessive vibration and noise will become evident because of the turbulence caused in the valve. The velocity limit for valves is near Mach 1. If noise is occurring in the valve and a special antinoise trim is used in the valve, the velocity is normally limited to Mach 0.33. If the high velocity exceeds the Mach-0.5 limit for noise generation, a larger valve body will need to be chosen. If the velocity approaches Mach 1.0 in this situation, a larger body size should also be chosen.

7.4.10 Trim-Size Selection

Valve manufacturers provide tables that outline the flow coefficients for a certain valve style, flow direction, body-pressure rating, flow characteristic, size of the valve seat (either full or reduced area) or the seal, and stroke. Some charts may be broken down to percentages of opening, since some throttling services may not utilize the entire stroke. Depending on the style of the valve, a trim number is offered with a predetermined flow area that allows the passage of flow equal to the C_v maximum.

7.4.11 Gas Sizing Example A

For this example, the following service conditions and equipment requirements are given in Imperial units:

Gas	Steam
Temperature	450°F
Upstream pressure P_1	140.0 psia
Downstream pressure P_2	50.0 psia
Flow rate Q	10,000 lb/h
Critical pressure P_C	3206.2 psia
Critical temperature T_C	705.5°F
Molecular weight M_W	18.03
Specific volume	10.41
Ratio k of specific heats	1.33
Pipeline size	2 in (ANSI Class 600)
Valve	Globe, flow-to-open
Flow characteristic	Equal percentage

Of the four C_V equations given for gas sizing (Sec. 7.4.1), the following equation is appropriate for the provided service conditions:

$$w = 19.3F_P C_v P_1 Y \sqrt{\frac{xM_W}{T_1 Z}}$$

From Table 7.4, the pressure-drop ratio x_T for a globe valve with flow-to-open action is 0.75. The user should check for choked flow by calculating the ratio of specific heats factor F_K:

$$F_K = \frac{k}{1.40} = \frac{1.33}{1.40} = 0.95$$

The ratio of actual pressure drop to absolute inlet pressure x is now calculated with the following equation:

$$x = \frac{\Delta P}{P_1} = \frac{140-50}{140} = 0.64$$

The value $F_K x_T$ can then be calculated as

$$(0.95)(0.75) = 0.71$$

Because the ratio of actual pressure drop to absolute inlet pressure x is less than the combined value $F_K x_T$, choked flow is not occurring and the value x is used with the remaining calculations.

The expansion factor Y is now calculated using the following equation:

$$Y = 1 - \frac{x}{3(F_K x_T)} = 1 - \frac{0.64}{3(0.71)} = 0.70$$

The compressibility factor Z can be determined by using the equations for the reduced-pressure factor P_r and the reduced-temperature factor T_r.

$$P_r = \frac{P_1}{P_C} = \frac{140}{3208.2} = 0.04$$

$$T_r = \frac{T_1}{T_C} = \frac{450 + 460}{705.5 + 460} = 0.78$$

With the aid of these two numbers and Fig. 7.3, the compressibility factor Z is found to be 1.0. Assuming that the piping-geometry factor F_p is 1.0, the appropriate C_v equation should be used to calculate a preliminary C_v:

$$w = 19.3 F_p C_v P_1 Y \sqrt{\frac{x M_W}{T_1 Z}} \qquad \text{or} \qquad C_v = \frac{w}{19.3 F_p P_1 Y} \sqrt{\frac{T_1 Z}{x M_W}}$$

$$C_v = \frac{10{,}000}{(19.3)(140)(0.70)} \sqrt{\frac{(910)(1.0)}{(0.64)(18.02)}} = 47.0$$

From the manufacturer's C_v tables, the smallest valve body should be chosen that will pass the required C_v of 47. For assumption purposes, a 2-in valve is the smallest size that will accommodate a C_v of 47. Because the 2-in body is the same size as the pipeline size, the piping-geometry factor F_p is 1.0 and C_v remains the same. In this case, the preliminary C_v becomes the final C_v.

At this point, the exit velocity should be calculated to ensure that it is within the velocity limits of Mach 0.5 for noise or Mach 1.0 for maximum velocity. The valve outlet of a 2-in valve is 3.14 (from Table 7.9). From the steam tables, v is found to be 10.41 ft^3/lb and T is 414°F.

Therefore, the following velocity equation should be used for steam service:

$$M_{\text{steam}} = \frac{wv}{1514\ A_V\sqrt{T}} = \frac{(10{,}000)(10.41)}{(1515)(3.14)\sqrt{414 + 460}} = 0.74$$

Because Mach 0.74 is greater than the noise limit of Mach 0.5, the turbulence will most likely create noise in the valve, and preventative measures may be necessary, such as special trim, insulation, or isolation of the valve. Because the velocity did not exceed the limit of Mach 1.0, a larger valve size is not necessary and the final C_v remains the same.

7.4.12 Gas Sizing Example B

For the second gas example, the following service conditions and equipment requirements are provided in Imperial units:

Gas	Natural gas
Temperature	65°F
Upstream pressure P_1	1314.7 psia
Downstream pressure P_2	99.7 psia
Flow rate Q	2,000,000 scfh
Critical pressure P_C	672.9 psia
Critical temperature T_C	342.8°F
Molecular weight M_W	16.04
Ratio k of specific heats	1.31
Pipeline size	Unspecified (ANSI Class 600)
Valve	Globe, flow-to-open
Flow characteristic	Linear

Of the four C_v equations given for gas sizing from Sec. 7.4.1, the following equation is best for the provided service conditions:

$$Q = 7320\ F_P C_v P_1 Y \sqrt{\frac{x}{M_W T}}$$

Referring to Table 7.4, the pressure-drop ratio x_T for a globe valve with flow-to-open action is 0.75. A choked-flow condition should be checked first by calculating the ratio of specific heats factor F_K:

$$F_K = \frac{k}{1.40} = \frac{1.31}{140} = 0.94$$

The ratio x of actual pressure drop to absolute inlet pressure is determined by using the following equation:

$$x = \frac{\Delta P}{P_1} = \frac{1314.7 - 99.7}{1314.7} = 0.92$$

The value $F_K x_T$ can then be calculated as follows:

$$(0.94)(0.75) = 0.70$$

Because the combined value $F_K x_T$ is less than the ratio of actual pressure drop to absolute inlet pressure x, choked flow is occurring and $F_K x_T$ is used with the remaining calculations. The expansion factor Y is now calculated using the following equation:

$$Y = 1 - \frac{x}{3(F_K x_T)} = 1 - \frac{0.70}{3(0.70)} = 0.67$$

Before the compressibility factor Z can be determined, the reduced-pressure factor P_r and the reduced-temperature factor T_r must be calculated with the following equations:

$$P_r = \frac{P_1}{P_C} = \frac{1314.7}{667.4} = 1.97$$

$$T_r = T\frac{1}{T_C} = \frac{65 + 460}{342.8} = 1.53$$

Using the P_r and the T_r factors with Fig. 7.3, the compressibility factor Z is found to be approximately 0.86. With the assumption that the piping-geometry factor F_P is 1.0 and that x is now replaced by the combined value $F_K x_T$, the chosen C_v equation is used to calculate a preliminary C_v:

$$Q = 7320 F_P C_v P_1 Y \sqrt{\frac{F_K x_T}{M_W T Z}} \qquad \text{or} \qquad C_v = \frac{Q}{7320 F_P P_1 Y} \sqrt{\frac{M_W T Z}{F_K x T}}$$

$$C_v = \frac{2{,}000{,}000}{(7320)(1.0)(1314.7)(0.667)} \sqrt{\frac{(16.04)\,(525)(0.86)}{0.70}} = 3\,2$$

From the manufacturer's C_v tables, the user should find the smallest valve body that will pass the required C_v of 32. For this example, a 1.5-in valve is assumed to be the smallest size that will accommodate the preliminary C_v of 32. Because the pipeline size is unspecified, the user must assume that the piping-geometry factor F_p is 1.0 and the final C_v remains the same as the preliminary C_v.

The exit velocity is now calculated to ensure that the 1.5-in body will handle the velocity limit of Mach 1.0. If the velocity exceeds Mach 0.5, noise will most likely be generated. From Table 7.9, the valve outlet area A_V of a 1.5-in body is 1.77. Since the fluid is natural gas, the following velocity equation for gas service is used after converting scfh to cfh (Sec. 7.4.9):

$$M_{\text{gas}} = \frac{Q_a}{5574A_V\sqrt{\dfrac{kT}{M_W}}} = \frac{297{,}720}{(5574)(1.77)\sqrt{\dfrac{(1.31)(65+460)}{16.04}}} = \text{Mach } 4.61$$

Because a Mach number exceeding sonic velocity (Mach 1.0) at the outlet of the valve is not possible, a larger valve size must be chosen to lower the velocity to below Mach 1.0.

The chosen valve would ideally handle a velocity of Mach 0.5 or less. To find the correct valve size to handle the process at Mach 0.5, the velocity equation should be used—except the user should solve for the unknown factor, which is the valve outlet area A_V:

$$A_V = \frac{Q_a}{5574M_{gas}\sqrt{\dfrac{kT}{M_W}}} = \frac{297{,}720}{(5574)(0.5)\sqrt{\dfrac{(1.31)(65+460}{16.04}}} = 16.3 \text{ in}^2$$

The valve outlet area A_V can then be used to solve the size of the valve:

$$A_V = pd^2 \qquad \text{or} \qquad d = \sqrt{\frac{4A_V}{\pi}} = \sqrt{\frac{(4)(16.3)}{3.14}} = 4.6 \text{ in}$$

Because a 4-in valve would be too small and a 5-in valve does not exist, a 6-in valve is necessary. This valve will need a reduced trim to accommodate a C_v of 32.

8 Actuator Sizing

8.1 Actuator-Sizing Criteria

8.1.1 Introduction to Actuator Sizing

With the automation of process systems, the use of actuators on throttling valves and actuation systems on manual on-off valves has increased dramatically. Generally, actuator sizing is a complex science, involving a number of factors that must be considered to match the correct actuator with the valve. For the valve to open, close, and/or throttle against process forces, proper actuator selection and sizing are critical.

Some users equate valve-body size with the actuator size; for example, a false assumption can be made that a 3-in valve always uses a certain size actuator, whose standard actuator yoke connection matches the valve connection. If all process service conditions and valve designs were equal, this might be possible. However, processes vary widely in terms of pressures, pressure drops, temperatures, shutoff requirements, etc. Valves vary according to motion (linear and rotary), packing friction, balancing (nonbalanced versus pressure-balanced), etc. Because of all the variables between the process and the valve, one valve size may have a number of actuator size options. For this reason, the user cannot simply place any spare actuator on a valve and expect it to work correctly-the actuator will most likely be undersized or oversized for that valve and the process. If the actuator is undersized, the major problem is that it will not be able to overcome the process and valve frictional forces. If the actuator is slightly undersized, it will struggle to overcome the forces working against it, providing sluggish and erratic stroking, as well as possibly not meeting the shutoff requirement. In addition, if the actuator is not stiff enough to hold its position close to the seat or seal, the “bathtub stopper” effect will take place and the closure element will slam into the seat or seal, causing a water-hammer effect. If the actuator is extensively undersized, it will not be able to open or close or throttle correctly.

If the actuator is oversized, the main disadvantage is that the actuator cost is higher. In addition, the oversized actuator is heavier and taller, which may create seismic, space, or maintenance concerns. From a performance standpoint, the larger actuator may be more sluggish in terms of speed and response. Larger actuators also produce greater thrust, which may damage the internal parts of the valve if the process forces are not present to counter that thrust. For this reason, oversized actuators require the use of a pressure regulator, which may create additional problems of incorrect settings and even slower response.

Generally, actuators have a tendency to be oversized because of the buildup of safety factors that the user and manufacturer add to the design process to ensure adequate "worst-case scenario" protection. If the calculations show a certain actuator size to be marginally or slightly undersized for a given process and valve, most users tend to move to the next larger size. However, because of the safety factors already built into the sizing process, the smaller size may function just as well, if not better, with that process.

8.1.2 Basic Actuator-Sizing Criteria

Actuator-sizing methods vary from manufacturer to manufacturer, depending on the basic design; however, several basic concepts are central to any actuator sizing. First and most importantly, the actuator must have the thrust to overcome the process forces that are operating inside the valve—in particular, the upstream and downstream pressures. In some services, a valve is working in an unbalanced situation where the upstream pressure is working against one side of the closure element, and the downstream pressure is working against the opposite side. These forces can be significant and will require a larger actuator as the force increases. Other valves permit a pressure-balanced design in which the upstream pressure is allowed to act on both sides of the closure element. This allows a minimal amount of process force to act against the element, permitting a smaller actuator.

The actuator must also provide enough force to overcome the process pressures in order to close the closure element, as well as to maintain the shutoff requirements indefinitely, according to the seat leakage classification (Sec. 2.3). The tighter the shutoff requirement, the greater the force must be provided by the actuator. If tight shutoff is not a main consideration, or if the valve is expected to throttle and close rarely, a lower shutoff classification may suffice that will allow the use of a smaller actuator.

The actuator must also overcome any frictional forces between the valve's stem and packing box. This friction can vary from a number of factors: number of rings, packing material, linear versus rotary motion, and packing compression requirements.

The final factor that may create a need for additional force is the design criteria of the valve itself. For example, a linear globe valve may be designed with pressure-balanced trim. Although the process forces are minimized, the seals of the pressure-balanced plug will increase the frictional forces, as well as add to the weight of the plug. In extremely large valves, the weight of the closure or regulating element (especially with globe-style plugs) must be taken into consideration.

Therefore, the forces that must be considered to determine the size and subsequent thrust of the actuator are written as

$$F_{\text{total}} = F_{\text{process}} + F_{\text{seat}} + F_{\text{packing}} + F_{\text{miscellaneous}}$$

where F_{total} = total force (or actuator thrust) required to open, close, or throttle valve

F_{process} = force to overcome unbalanced process pressure

F_{packing} = force required to overcome packing friction

F_{seat} = force to provide correct seat load

$F_{\text{miscellaneous}}$ = force to overcome special design factors, weight, etc.

Another design criteria is the speed requirement of the actuator. In some cases, such as applications in which the process or personnel safety is a concern, the user may want the valve to close in a short time, such as less than a second, as opposed to several seconds. However, excessively fast actuator speed can present multiple problems, including water-hammer effects and position overshoot. Pneumatic actuators are subject to a number of factors that affect air capacity, such as pressure fluctuations, piping and tubing bends, filters, etc. For these reasons, high-speed actuation systems are normally hydraulic or electrohydraulic designs.

8.1.3 Free Air

Because the majority of actuators or actuation systems are pneumatically driven, certain principles concerning air compressibility and volume changes occurring with pressure changes must be understood. The specifications for pneumatically driven equipment, including actuators, are provided using the term *free air*. By definition, free air is the flow or volume rate at standard atmospheric temperature [70°F (21°C)] and pressure [14.7 psia (1 bar)]. Using free air avoids any misunderstanding regarding changes in volume. Typically, absolute pressure is designated as *psia*, gauge pressure as *psig*, and differential pressure as simply psi. For most equipment, the free-air flow rate is expressed in standard cubic feet per minute (scfm).

Because air volume can vary according to changes in pressure, the amount of free air contained in a vessel can be written as

$$V_1 = \mathrm{V}_2 \frac{P_2}{P_1}$$

where V_1 = free-air volume (standard cubic feet)
V_2 = vessel volume
P_1 = atmospheric pressure (14.7 psia)
P_2 = absolute vessel pressure (psia)

8.1.4 Supply Flow Rates

For pneumatically driven actuators, determining the correct air supply rate to the actuator is critical to ensure that enough air will be available to operate the actuator and provide the thrust necessary for the application. The relationship between flow rate and pressure drop is demonstrated by the following equations:

$$\Delta P = \frac{L\,Q_2}{kC_R d^{5.31}} \quad \text{or} \quad Q = \sqrt{\frac{\Delta P k C_R d^{5.31}}{L}}$$

where ΔP = pressure drop (psi)
L = length of tubing or piping (ft)
Q = standard air flow rate (scfm)
k = constant of 35,120
C_R = ratio of line pressure (psia) to atmospheric pressure (14.7 psia)
d = inside diameter of piping or tubing (inches) from Tables 8.1 and 8.2

For example, if a given actuator operates best at 80 psi and must have 4.3 scfm to operate at the required speed, the following parameters apply:

Line pressure	85 psia
Length *L of* tubing	100 ft
Tubing size	0.25 in

Table 8.1 Piping Values for d and $d^{5.31}$*,†

Factor	0.25 in.	0.375 in.	0.5 in.	0.75 in.	1.0 in
d	0.364	0.493	0.622	0.824	1.049
$d^{5.31}$	0.0047	0.0234	0.0804	0.3577	1.2892

*All dimensions are outside diameter.
†*Data courtesy of Automax, Inc.*

The pressure drop ΔP is 5 psi (85 – 80 psi) and the ratio C_R of line pressure to atmospheric pressure is 6.78, which is shown as:

$$C_R = \frac{P_1 + 14.7}{14.7} = \frac{99.7}{14.7} = 6.78$$

Using the calculations above, the flow rate for 0.25-in tubing (from Table 8.2, d = 0.204) can be calculated as follows:

$$Q = \sqrt{\frac{\Delta P k C_R d^{5.31}}{L}} = \sqrt{\frac{(5)(35{,}120)(6.78)(0.204)^{5.31}}{100}} = 1.6 \text{ scfm}$$

Because 1.6 scfm is less than the 4.3 scfm required for the speed requirement, a larger tube size must be chosen. A 0.375-in tube would produce 5.7 scfm, which is more than adequate:

$$Q = \sqrt{\frac{\Delta P k C_R d^{5.31}}{L}} = \sqrt{\frac{(5)(35{,}120)(6.78)(0.329)^{5.31}}{100}} = 5.7 \text{ scfm}$$

Table 8.2 Tubing Values for d and $d^{5.31}$*,†

Factor	0.25 in.	0.375 in.	0.5 in.	0.75 in.	1.0 in
d	0.204	0.329	0.430	0.555	0.680
$d^{5.31}$	0.0002	0.0027	0.0113	0.0439	0.129

*All dimensions are outside diameter.
†*Data courtesy of Automax, Inc.*

8.1.5 Air Usage and Consumption

The user must ensure that the air-supply capacity can meet the needs of all the pneumatic operators involved with a typical process system. This means that the compressor must be sized according to the air requirements of the actuators, which requires knowledge of the air usage and consumption. Correct calculations of the air usage and consumption allow for a more accurate prediction of air requirements and proper sizing of the compressor. In those cases where the air requirements exceed the capacity of the compressor or if the compressor is undersized, the pressure will not be adequate. Overall, this results in sluggish response or not enough thrust to operate the valve.

The term *air usage* refers to the amount of air used by a pneumatic actuator to stroke the valve. After the valve is stroked, the air usage stops until the valve is stroked again. The term *air consumption* refers to those pneumatic instruments that bleed air constantly, such as is the case with positioners. For spring-return (single-acting) diaphragm actuators, no air is used on the spring side of the diaphragm. Therefore, when the actuator is fully stroked, the air usage is the amount of the actuator's free-air volume at the pressure given. For example, using the free-air equation from Sec. 8.1.3, the assumption is made that a single-acting actuator has a volume of 2.1 ft^3 at 60 psi of air supply and will stroke six times per hour. The usage per cycle in standard cubic feet is

$$V_1 = V_2 \frac{P_2}{P_1} = 2\left(\frac{60 + 14.7}{14.7}\right) = 10.2 \text{ scf}$$

The usage per hour (standard cubic feet per hour) involving six strokes per hour can then be calculated:

$$10.2 \text{ scf} \times 6 = 61.2 \text{ scfh}$$

Double-acting actuators use air on both sides of the diaphragm or piston, depending on the design. Ideally, the air volume on both sides would be equal, but this is not the case because one side has less volume due to the actuator stem, travel stop, or fail-safe spring. In this case, the total volumes of the two sides are calculated separately and added together to present air volume per cycle. For example, a rack-and-pinion actuator has 500 in^3 on one side and 300 in^3 on the opposite side. It will be stroked 12 times an hour with 80 psi of air supply.

Using the conversion factor, cubic inches are converted to cubic feet for the first side:

$$\text{scf} = \frac{\text{in}^3}{1728} = \frac{500}{1728} = 0.29 \text{ scf}$$

The opposite side is converted likewise:

$$\text{scf} = \frac{\text{in}^3}{1728} = \frac{300}{1728} = 0.17 \text{ scf}$$

The combined air volume for the actuator is then 0.46 scf (0.29 + 0.17) and the air usage per cycle is calculated as

$$V_1 = V_2 \frac{P_2}{P_1} = 0.46\left(\frac{80 + 14.7}{14.7}\right) = 2.96 \text{ scf}$$

The usage per hour (standard cubic feet per hour) involving 12 strokes per hour can then be calculated:

$$2.96 \text{ scf} \times 12 = 35.52 \text{ scfh}$$

If a positioner is used with a single-acting actuator, the air usage can vary considerably since the actuator is throttling between the open and closed positions. Depending on the position movement, which can be large or small, the air usage is directly proportional to the movement. The air usage for an actuator with a positioner can be determined by the following equation:

$$\text{scfh} = \frac{V}{A}\,[P_S(M_2 - M_1) + 0.4PM_1)]N$$

where V = actuator volume (ft^3)
P_S = supply pressure (psia)
M_1 = starting position (fraction of stroke)
M_2 = finished position (fraction of stroke)
A = atmospheric pressure (14.7 psia)
N = number of strokes per hour

For example, a single-acting actuator with a positioner has an air volume of 500 in^3 and is stroked between 10 and 50 percent open. It is required to stroke eight times an hour using 60 psi of air supply. This would be calculated as

$$\text{scfh} = \frac{V}{A}\,[P_S(M_2 - M_1) + 0.4\,PM_1]N$$

$$= \frac{\frac{500}{1728}}{14.7}\,\{[(60 + 14.7)(0.5 - 0.1)] + [0.4(60 + 14.7)(0.1)]\}8$$

$$= 5.26 \text{ scfh}$$

This calculation provides only the air usage. Because some positioners bleed continually, they provide air consumption, which must be figured into the total air requirement when sizing the compressor. If a positioner is used with a double-acting actuator, as most are, the above equation is modified slightly:

$$\text{scfh} = \frac{V}{A}\,\{[2P_S(M_2 - M_1)] + [0.4P(1 - M_2 - M_1)]\}N$$

For example, a double-acting actuator with a positioner has an air volume of 300 in^3 and is stroked between 20 and 70 percent open. It is required to stroke 12 times an hour using 80 psi of air supply. This would be calculated as

$$\text{scfh} = \frac{V}{A}\,([2P(M_2 - M_1)] + [0.4\,P(1 - M_2 - M_1)]\}N$$

$$= \frac{\frac{300}{1728}}{14.7}\,([2(80 + 14.7)(0.7 - 0.2)] + [0.4(80 + 14.7)(1 - 0.7 - 0.2)]\}12$$

$$= 11.82 \text{ scfh}$$

Once again, the user should remember that any bleeding of air from the positioner (air consumption) must also be added to the air usage calculation.

8.2 Sizing Pneumatic Actuators

8.2.1 Actuator Force Calculation for Linear Valves

To determine what size of actuator is required for a linear-motion valve, such as a globe control valve, the user must examine the force that the process is applying inside the valve. This force value is known as $F_{process}$. A major factor in determining the process force is calculating the unbalanced area. The *unbalanced area* is defined as the area of the cage (or sleeve) .minus the stem area. The unbalanced area must be greater than the area of the seat. In equation form, it is written as

$$A_{unbalanced} = A_{cage\ or\ sleeve} - A_{stem} > A_{seat}$$

where $A_{unbalanced}$ = unbalanced area
$A_{cage\ or\ sleeve}$ = area of the cage or sleeve*
A_{stem} = area of the plug stem
A_{seat} = area of the seat

Formulas for calculating the process force are based upon the service conditions as well as three design criteria: The first determination is whether the flow assists with the opening or the closing of the valve. The second determination is whether the valve is unbalanced or pressure-balanced (globe or double-ported valves only). And the third determination is whether the flow is under or over the closure or regulating element (assumed to be a globe valve plug). The following formulas apply for the following valve configurations:

Pressure assists opening, unbalanced trim, flow under the plug:

$$F_{process} = (P_1 - P_2)\, A_V + P_2\, A_{stem}$$

Pressure assists opening, unbalanced trim, flow over the plug:

$$F_{process} = (P_1 - P_2)\, A_V - P_1\, A_{stem}$$

Pressure assists opening, balanced trim, flow under the plug:

$$F_{process} = (P_1 - P_2)\, A_{unbalanced} - P_2\, A_{stem}$$

*If the valve does not have a cage or sleeve, the area of the top of the plug is used.

Pressure assists opening, balanced trim, flow over the plug:

$$F_{\text{process}} = (P_1 - P_2)\, A_{\text{unbalanced}} + P_2\, A_{\text{stem}}$$

Pressure assists closing, unbalanced trim, flow under the plug:

$$F_{\text{process}} = -\,[(\text{P}_1 - P_2)A_V + P_1\, A_{\text{stem}}]$$

Pressure assists closing, unbalanced trim, flow over the plug:

$$F_{\text{process}} = -\,[(P_1 - P_2)A_V - P_1\, A_{\text{stem}}]$$

Pressure assists closing, balanced trim, flow under the plug:

$$F_{\text{process}} = -\,[(P_1 - P_2)\, A_{\text{unbalanced}} - P_1\, A_{\text{stem}}]$$

Pressure assists closing, balanced trim, flow over the plug:

$$F_{\text{process}} = -\,[(P_1 - P_2)\, A_{\text{unbalanced}} + P_1\, A_{\text{stem}}]$$

where F_{process} = force to overcome the process pressure unbalance
P_1 = upstream pressure at inlet (psia)
P_2 = downstream pressure at outlet (psia)
A_V = area of the valve port (in^2)
A_{stem} = area of the plug stem (in^2)
$A_{\text{unbalanced}}$ = unbalanced area (in^2)

When the actuator force is used to open the valve, three of the four forces oppose the actuator: the process force, the packing friction force, and any miscellaneous design forces. Because no actuator force is needed for seat loading, that value is not necessary. This can be written as

$$F_{\text{open}} = F_{\text{process}} + F_{\text{packing}} + F_{\text{miscellaneous}}$$

where F_{open} = total force (or actuator thrust) required to open valve
F_{process} = force to overcome the process pressure unbalance
F_{packing} = force required to overcome packing friction
$F_{\text{miscellaneous}}$ = force to overcome special design factors, weight, etc.

If the total force must close the valve, the process force must be a negative number, as demonstrated in the latter four equations. In other words, because the process pressure is assisting the valve to close, the process force actually decreases, rather than increases, the force required by the actuator. The actuator has to produce only enough force to overcome the combined force produced by the packing friction, seat load, and miscellaneous design forces, minus- the process force. In this case, the actuator force requirement may be minimal. This can be written as

$$F_{\text{close}} = F_{\text{seat}} + F_{\text{packing}} + F_{\text{miscellaneous}} - F_{\text{process}}$$

where F_{close} = total force (or actuator thrust) required to close valve
F_{seat} = force required to provide-correct seat load

In applications where the actuator must open and close the valve, both forces for opening and closing, F_{open} and F_{close}, must be calculated. The largest force of the two is then used to determine the size of the actuator.

After the process force has been determined, the next force to be calculated is the load required by the shutoff classification, which uses the following equation:

$$F_{\text{seat}} = F_{\text{class}} C_{\text{port}}$$

where F_{class} = required seat force of shutoff classification (see Table 2.7)
C_{port} = circumference of the valve port

The force required to overcome the packing friction, F_{packing}, is provided by the manufacturer. Packing friction is determined by the diameter of the stem and the packing material, assuming correct compression. Overcompressing the packing will add to the packing friction and the force required to overcome it.

After the cumulative forces are calculated, an actuator can be chosen from the manufacturer based on the thrust capabilities of the actuator. The final requirement is that the correct actuator can be mounted on the valve that has been sized for the service. In some applications involving large oversized actuators required for severe services, the yoke-to-bonnet connection may not be a standard and will require modifications.

With pneumatic actuators, the appropriate size and spring will need to be chosen for the application. Most manufacturers provide actuator tables that include the thrust that the actuator can generate. In addition, the user must chose the desired failure action (fail-open or fail-closed) and yoke-to-bonnet connection. The correct actuator is the smallest actuator that meets the thrust and mounting requirements.

8.2.2 Actuator Force Calculation for Butterfly Valves

A different actuator-sizing criteria must be considered with rotary valves. Critical to rotary actuator sizing is the butterfly valve's torque requirement, in other words, the amount of thrust that the actuator must apply to the shaft to produce a rotational force to operate the valve. In particular, the user must calculate the *seating torque*, which is the torque needed to close the valve against or with the process; the *breakout torque*, which is the torque needed to begin to open the valve; and the *dynamic torque*, which is the torque needed to throttle the valve. When these torque values are known, the correct rotary actuator can be chosen.

The first step in sizing an actuator for a butterfly valve is to determine the orientation of the shaft and the actuator stiffness requirements. Shaft orientation is critical with eccentric butterfly valves. When the shaft is placed on the upstream side of the flow, the process fluid forces the disk into the seal. On the other hand, when the shaft is placed on the downstream side of the flow, the process fluid forces the disk to open. In gas applications, when the butterfly valve is designated to fail-closed, the shaft is generally upstream. If the valve is designated to fail-open, the shaft is downstream. With liquid applications, the disk has a tendency to slam into the seal in fail-closed applications if the actuator is not stiff enough to withstand the process flow. A rotary actuator with insufficient stiffness is likely to cause water-hammer effects; therefore a stiffness calculation must be made by finding the ratio of the maximum pressure drop to the supply pressure:

$$A_S = \frac{P_1 - P_2}{P_S}$$

where A_S = required actuator stiffness
P_1 = upstream pressure at inlet (psia)
P_2 = downstream pressure at outlet (psia)
P_S = Supply pressure

Table 8.3 shows the maximum actuator stiffness values for three sizes of actuators. If the calculated value is larger than the table value, a larger actuator size must be chosen for that size of valve.

For example, a 4-in butterfly valve has an upstream pressure P_1 of 240 psia, and a downstream pressure P_2 of 60 psia, and a supply pressure of 80 psi. The required actuator stiffness ratio is

$$A_S = \frac{P_1 - P_2}{P_S} = \frac{240 - 60}{80} = 2.25$$

Looking at Table 8.3 for 4-in valves, the actuator stiffness is slightly larger than the maximum value for the smallest actuator, size A. Therefore, the next larger size, size B, would be required.

The chosen actuator must also have the necessary force to generate torque for the butterfly valve to close, to open (breakout torque), and

Table 8.3 Actuator Stiffness Factors

Valve Size	Actuator Size A	Actuator Size B	Actuator Size C
2-inch DN 50	4.2		
3-inch DN 80	3.1		
4-inch DN 100	2.0	6.7	
6-inch DN 150		4.3	7.0
8-inch DN 200		2.5	5.9
10-inch DN 250			4.1

to throttle between the open and closed positions. The following equations are used to determine seating and breakout torques:

Shaft downstream, torque required to close the valve:

$$T_{\text{seat}} = -T_P - T_S - T_H - \Delta P_{\max}(C_B + C_O)$$

Shaft downstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + T_H + \Delta P_{\max}(C_B - C_O)$$

Shaft upstream, torque required to close the valve:

$$T_{\text{seat}} = -T_P - T_S - T_H - \Delta P_{\max}(C_B - C_O)$$

Shaft upstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + T_H + \Delta P_{\max}(C_B + C_O)$$

where T_{seat} = seating torque required
T_{breakout} = breakout torque required
T_P = packing torque
T_S = seat torque
T_H = handwheel torque*
$\Delta P_{\max}$ = maximum pressure drop at shutoff
C_B = bearing (or guide) torque factor
C_O = off-balance torque factor

*Handwheel torque is 0 if no handwheel exists.

The packing torque T_P is the torque required to overcome. the rotational friction of the packing on the shaft. The seat torque T_S is the torque required to overcome the friction of the seat on the disk. The bearing torque factor CB indicates the relationship that as the pressure across the valve increases, the force on the bearing increases proportionally. The handwheel torque T_H is the torque required to overcome the friction of an attached handwheel. If a declutchable handwheel is used, this factor is considered only when the handwheel is in gear. The off-balance torque factor C_O shows the relationship of the off-balance forces between the disk and the mechanical connection in the actuator (which converts the actuator force to torque). Because these torques and factors vary according to individual valve designs, they are provided by the manufacturer. If the final torque value is a negative value, this indicates that the butterfly disk will have a tendency to resist closing. Conversely, if the value is positive, the disk will have a tendency to resist opening.

When a high pressure drop is expected at any part of the quarter-turn stroke, the net torque output can vary dramatically throughout the shaft rotation. For this reason, the dynamic torque is calculated at various degrees of opening. When the shaft is downstream, a reversal of torque takes place at approximately 75 percent open, which can lead to control problems with the valve. If this happens, the user has the choice of changing the orientation of the shaft to shaft upstream (if possible), or placing the limit stops on the actuator to prevent rotation beyond 70 percent.

The following equations are used when calculating the dynamic torque for butterfly valves in gas services:

To close the valve:

$$T_D = -T_P - \Delta P_{\text{eff}}(C_{BT})$$

To open the valve:

$$T_D = T_P + \Delta P_{\text{eff}}(C_{BT})$$

where T_D = dynamic torque
T_{P} = packing torque value (from manufacturer)
$\Delta\text{P}_{\text{eff}}$ = ΔP_{actual} at the flowing condition at the degree of opening (limited to the ΔP_{choked})
C_{BT} = bearing or guide torque value (from manufacturer)

For liquid applications, the following equations are used:

To close the valve:

$$T_D = -T_P - \Delta P_{\text{eff}}(C_{BT} - C_D)$$

To open the valve:

$$T_D = T_P - \Delta P_{\text{eff}}(C_{BT} - C_D)$$

where C_D = dynamic torque factor (from Table 8.4)

Table 8.4 Butterfly-Valve Torque Factors*,†,‡

Valve Size (in.)	Dynamic Torque Factor vs. Disc Position (Degrees Open)																	
	10°		20°		30°		40°		50°		60°		70°		80°		90°	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
3	0	0	0	0	0		0	1	0	2	0	2						
4	0	0	0	0	0	0	0	0	0	1	1	2	1	4	0	6	-4	4
6	0	0	0	1	1	1	1	3	2	4	4	9	6	17	0	23	-19	19
8	0	0	0	1	1	2	2	4	3	7	4	12	1	19	-8	28	-23	23
10	0	1	1	2	2	6	5	9	7	17	8	28	4	46	-18	65	-55	55
12	1	2	2	7	5	13	10	20	16	38	18	61	8	101	-40	144	-121	121
14	1	3	3	10	6	18	15	28	22	54	25	85	12	142	-56	201	-168	168
16	2	4	4	12	8	23	19	36	27	67	32	106	15	177	-69	250	-211	211
18	5	10	10	29	19	54	44	83	64	156	73	247	34	411	-161	582	-489	489
20	5	10	10	31	21	58	47	89	68	168	78	264	6	440	-173	623	-524	524
24	7	14	14	42	28	78	64	121	92	227	106	358	50	596	-234	845	-710	710
30	24	49	49	146	97	268	219	414	316	779	365	1230	170	2046	-804	2899	-2436	2436

**A*-shaft downstream; *B* = shaft upstream.

†*Courtesy of Valtek International.*

‡*Note:* When degrees of opening are not known, use highest value of C_d for valve size.

If the final number for the dynamic torque value is a negative number, the disk will resist closing with the flow moving the disk toward the open position. If the dynamic torque number is positive, the disk will resist opening—with the flow moving the disk toward the closed position. From the manufacturer's data and given the necessary available air supply, an actuator with sufficient torque can then be selected. This torque must overcome the seating and breakout torques—as well as the dynamic torque, which is required through the entire stroke of the valve. If the actuator's available torque is less than the dynamic torque, a larger actuator size with more torque force should be selected.

Following selection of the actuator, stiffness should again be checked to prevent the disk from slamming into the seat for those applications with the shaft downstream.

Consideration should be given to whether a spring is necessary to move the disk to a particular failure position (fail-open or fail-closed). For fail-closed applications that do not require a high degree of shutoff, the spring must have adequate torque to overcome the dynamic torque. If the valve requires tight shutoff, the spring must generate enough torque to overcome the required seating torque at the closed position. For fail-open applications, the spring must have enough torque to overcome the required breakout torque at the closed position, as well as to overcome any dynamic torque as it moves though the full stroke to the full-open position. If the spring is incapable of producing enough force to overcome the seating or breakout or dynamic torque, a volume tank could be specified to ensure adequate force to move the valve to the correct position upon loss of air supply.

8.2.3 Actuator Force Calculation for Ball Valves

Because of the ball valve's design with the ball moving into the flow stream, as opposed to a disk that is already in the flow stream, the forces acting on the ball valve (and the torques required) are somewhat different. This requires a different set of torque calculations for seating or breakout:

Shaft downstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + \Delta P_{\max}(C_B + C_S) + T_H$$

Shaft upstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + \Delta P_{\max} C_B + (T_S - \Delta P C_S) + T_H$$

where ΔP = actual pressure drop $(P_1 - P_2)$
C_S = seat torque factor

The packing torque T_P is the torque required to overcome the rotational friction of the packing on the shaft. The seat torque T_S is the torque required to overcome the friction of the seat on the disk. The bearing torque factor C_B shows the relationship that as the pressure across the valve increases, the force on the bearing increases proportionately. The handwheel torque T_H is the torque required to overcome the friction of an attached handwheel. If a declutchable handwheel is specified with the actuator, this factor is only considered when the handwheel is in gear. Because these torques and factors vary widely due to design differences, they are usually determined by the manufacturer.

With liquid services, the dynamic torque must also be calculated. As noted in the previous section, dynamic torque is the torque required to overcome the torque on the closure element caused by the fluid dynamic forces on the ball. To calculate dynamic torque, the following equation is used:

$$T_D = T_P + \Delta P_{\text{eff}}(C_D + C_B)$$

where ΔP_{eff} = actual pressure drop across the valve at the flowing condition that occurs when the valve is in the open position (ΔP_{eff} is less than or equal to ΔP_{choked})
C_D = dynamic torque factor (from Table 8.5)
C_B = bearing torque factor (from manufacturer)

Once the seating or breakout and dynamic torques have been calculated, the correct actuator with sufficient torque is then chosen from the manufacturer's tables.

If a spring is required to move the ball to a particular failure position (fail-open or fail-closed), special consideration should be given to sizing the correct spring that can overcome the process forces. For fail-closed applications that do not require a high degree of shutoff, the spring must have adequate torque to overcome the dynamic torque. If the ball valve requires tight shutoff, the spring must generate enough torque to overcome the required seating torque at the closed position. For fail-open applications, the spring must have enough torque to overcome the required breakout torque at the closed position as well as to overcome any dynamic torque as it moves through the full stroke

Table 8.5 Ball-Valve Torque Factors*,†

Valve Size	T_p=Packing Torque (in-lb)					T_s=Seat Torque	C_b=Bearing Torque Factor		C_s=Seat Torque Factor	C_D = Dynamic Torque Factor	
(in.)	(1)	(2)	(3)	(4)	(5)	(in-lb)	(6)	(7)		60°	90°
1	43	228	421	301	57	20	0.06	0.09	0.1	0.25	0.6
1½	50	280	477	350	63	40	0.06	0.09	0.1	0.5	1.0
2	50	280	477	350	63	60	0.06	0.09	0.15	1.0	2.1
3	57	333	533	399	71	150	0.19	0.28	0.42	4.5	8.0
4	57	333	533	399	71	360	0.38	0.58	0.82	10.0	17.0
6	71	438	646	496	92	540	0.97	1.45	1.64	19.5	30.5
8	71	438	646	496	92	670	1.58	2.37	2.62	52.0	75.5
10	104	648	870	691	151	1100	4.38	6.57	4.55	108.0	165.5
12	104	648	870	691	151	1300	5.61	8.41	6.05	191.0	218.5

**Courtesy of Valtek International.*

†(1)PTFE or filled PTFE V-ring packing, (2) grafoil, (3) twin grafoil, (4) asbestos-free packing (AFP), (5) braided PTFE, (6) PTFE lined bearings, (7) metal bearings.

to the full-open position. If the available springs are not capable of producing enough force to overcome the seating, breakout, or dynamic torque, a volume tank should be specified to ensure adequate force to move the valve to the correct position upon loss of air supply.

8.3 Sizing Electromechanical and Electrohydraulic Actuators

8.3.1 Introduction to Actuator Sizing for Electromechanical and Electrohydraulic Actuators

For the most part, electromechanical and electrohydraulic actuators are sized according to the thrust needed to overcome the forces inside the body as shown in the following equation from Sec. 8.1.2:

$$F_{total} = F_{process} + F_{seat} + F_{packing} + F_{miscellaneous}$$

where F_{total} = total force (or actuator thrust) required to open, close, or throttle valve

$F_{process}$ = force to overcome process pressure unbalance

$F_{packing}$ = force required to overcome packing friction

F_{seat} = force to provide correct seat load

$F_{miscellaneous}$ = force to overcome special design factors, weight, etc.

Individual sizing equations to determine actuator size vary widely depending on the design and application of the actuator and are not specifically included in this section.

8.3.2 Special Considerations

Typically, the application engineers for the electromechanical or electrohydraulic manufacturer will size the actuator based upon the process and frictional forces associated with the valve as well as include some additional thrust for safety considerations. With the high level of engineering required for these actuators, the prevailing thought is better too much actuator than not enough. However, in some cases, the accumulation of safety factors over the sizing process can add anywhere from 25 to 50 percent to the total thrust of the actuator. High costs are associated normally with electromechanical and electrohydraulic actuators. Therefore, if sizing formulas show the thrust requirement to be slightly more than a given size, all safety factors should be reconsidered to check for an impractical accumulation. If that is the case, the smaller actuator size can be considered.

Electromechanical and electropneumatic actuators are normally specified for those applications requiring faster stroking speeds or higher performance than provided normally by pneumatic actuators.

From a sizing standpoint, application engineers use specialized sizing equations to determine the stroking speed, frequency response, and level of precision positioning. Because most applications requiring electromechanical and electrohydraulic actuators are special or severe, services, manufacturers have a tendency to size actuators based on flow rate and pressure drop.

9 Common Valve Problems

9.1 High Pressure Drops

9.1.1 Introduction to High Pressure Drops

Flow moves through a valve due to a difference between the upstream and downstream pressures, which is called the *pressure drop* (ΔP) or the *pressure differential*. If the piping size is identical both upstream and downstream from the valve and the velocity is consistent, the valve must reduce the fluid pressure to create flow by way of frictional losses. A portion of the valve's frictional losses can be attributed to friction between the fluid and the valve wall. However, this friction is minimal and is not sufficient to create enough pressure drop for an adequate flow. A more effective way to create a significant frictional loss in the valve is through a restriction within the body. Because many valves are designed to allow a portion of the valve to be more narrow than the piping, they can easily provide this restriction in the fluid stream. Because of the laws of conservation, as the fluid approaches the valve, its velocity increases in order for the full flow to pass through the valve, inversely producing a corresponding decrease in pressure (Fig. 9.1). The inverse relationship between pressure and velocity is shown by Bernoulli's equation, which is

$$\frac{\rho V_1^2}{2g_C} + P_1 = \frac{\rho V_{VC}^2}{2g_C} + P_{VC}$$

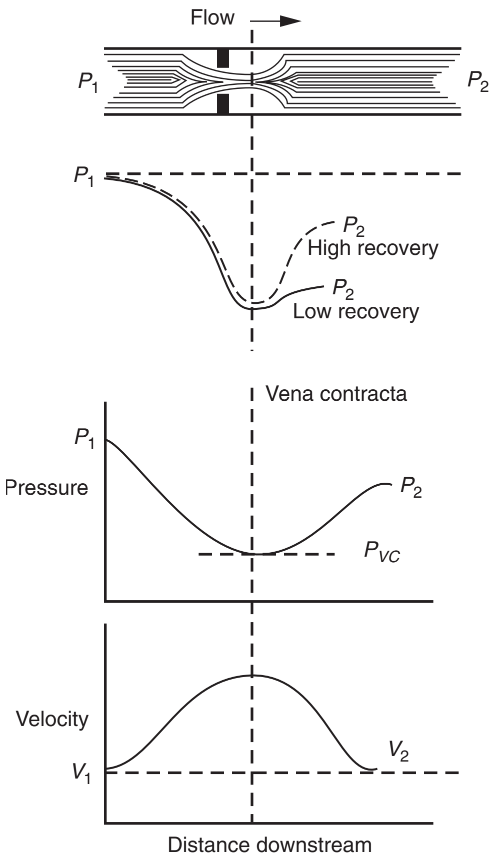


Figure 9.1 Location of vena contracta from point of orifice restriction and pressure and velocity curves. (*Courtesy of Fisher Controls International, Inc.*)

where ρ = density units
V_1 = upstream velocity
g_C = gravitational units conversion
V_{VC} = velocity at vena contracta
P_{VC} = pressure at vena contracta
P_1 = upstream pressure

The highest velocity and lowest pressure occur immediately downstream from the narrowest constriction, which is called the *vena contracta*. Figure 9.2 shows that the vena contracta does not occur at the

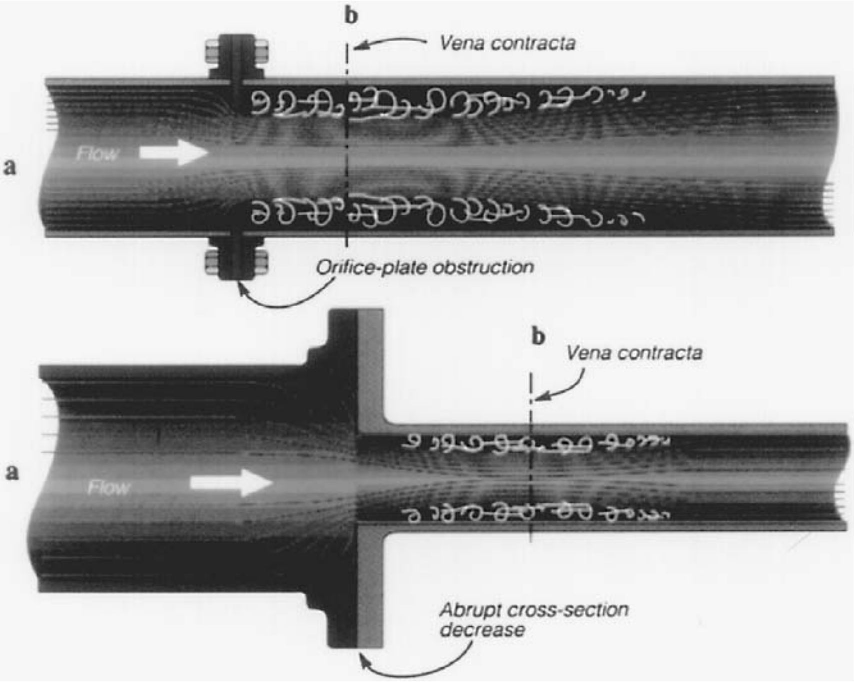


Figure 9.2 Relationship between orifice restriction and turbulence generation. (*Courtesy of Fisher Controls International, Inc.*)

restriction itself but rather downstream some distance from the restriction. This distance may vary according to the pressures involved. At the vena contracta the flow velocity is at a maximum speed, while the flow area of the fluid stream is at its minimum value.

Following the vena contracta, the fluid slows and pressure builds once again, although not to the original upstream pressure. This difference between the upstream and downstream pressures is caused by frictional losses as the fluid passes through the valve, and is called the *permanent pressure drop.* The difference in pressure from the pressure at the vena contracta and the downstream pressure is called the *pressure recovery.* A simplified profile of the permanent pressure drop and pressure recovery is shown in Fig. 9.3.

The flow rate for a valve can be increased by decreasing the downstream pressure. However, in liquid applications the flow can be limited by the pressure drop falling below the vapor pressure of the fluid, which will create imploding bubbles or pockets of gas (called cavitation or flashing, respectively). *Choked flow* occurs when the liquid flow

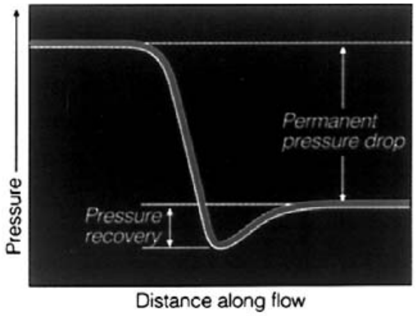


Figure 9.3 Flow curve showing pressure recovery and permanent pressure drop. (*Courtesy of Fisher Controls International, Inc.*)

is saturated by the fluid itself mixed with the gas bubbles or pockets and can no longer be increased by lowering the downstream pressure. In other words, the formation of gas in a liquid crowds the vena contracta, which limits the amount of flow that can pass through the valve. With gases, as the velocity approaches sonic speeds, choked flow also occurs and the valve will not be able to increase flow despite a reduction in downstream pressure.

9.1.2 Effects of High Pressure Drops

As discussed in Sec. 9.1.1, the flow function of the valve is dependent on the existence of a pressure drop, which allows flow movement from the upstream vessel to the downstream vessel or to atmosphere. Because a pressure drop generated by the valve absorbs energy through frictional losses, the ideal pressure drop allows the full flow to pass through the body without excessive velocity, absorbing less energy. However, some process systems, by virtue of their system requirements, may need to take a larger pressure drop through the valve.

A high pressure drop through a valve creates a number of problems, such as cavitation, flashing, choked flow, high noise levels, and vibration. Such problems present a number of immediate consequences: erosion or cavitation damage to the body and trim, malfunction or poor performance of the valve itself, wandering calibration of attached instrumentation, piping fatigue, or hearing damage to nearby workers. In these instances, valves in high-pressure-drop applications require expensive trims, more frequent maintenance, large spare-part inventories, and piping supports. Such measures drive up maintenance and engineering costs.

Although users typically concentrate on the immediate consequences of high pressure drops, the greatest threat that a high pressure drop presents is lost efficiency to the process system. Because high pressure drops absorb a great deal of energy, that energy is lost from the system. In most process systems, energy is added to the system through heat generated by a boiler or through pressure created by a pump. Both methods generate energy in the system, and as more energy is absorbed by the system—including that energy lost by valves with high pressure drops—larger boilers or pumps must be used. Consequently, if the system is designed with few valves with high pressure drops, the system is more efficient and smaller boilers or pumps can be used.

9.2 Cavitation

9.2.1 Introduction to Cavitation

Cavitation is a phenomenon that occurs only in liquid services. It was first discovered as a problem in the early 1900s, when naval engineers noticed that high-speed boat propellers generated vapor bubbles. These bubbles seemed to lessen the speed of the ship, as well as cause physical deterioration to the propeller.

Whenever the atmospheric pressure is equal to the vapor pressure of a liquid, vapor bubbles are created. This is evident when a liquid is heated, and the vapor pressure rises to where it equals the pressure of the atmosphere. At this point, bubbling occurs. This same phenomenon can also occur by decreasing the atmospheric pressure to equal the vapor pressure of the liquid. In liquid process applications, when the fluid accelerates to pass through the narrow restriction at the vena contracta, the pressure may drop below the vapor pressure of the fluid. This causes vapor bubbles to form. As the flow continues past the vena contracta, the velocity decreases as the flow area expands and pressure builds again. The resulting pressure recovery increases the pressure of the fluid above the vapor pressure. This phenomenon is described in Fig. 9.4.

As a vapor bubble is formed in the vena contracta, it travels downstream until the pressure recovery causes the bubble to implode. This two-step process—the formation of the bubble in the vena contracta and its subsequent implosion downstream—is called *cavitation.* In simple terms, cavitation is a phase that is characterized by a liquid–vapor–liquid process, all contained within a small area of the valve and within microseconds. Minor cavitation damage may be considered normal

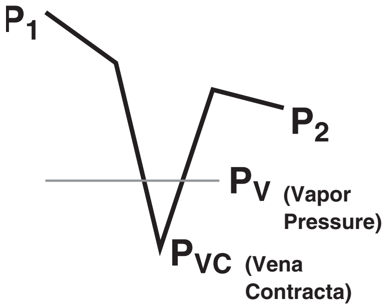


Figure 9.4 Flow curve showing pressure drop falling below the vapor pressure, which results in cavitation. (*Courtesy of Valtek International*)

for some applications, which can be dealt with during routine maintenance. If unnoticed or unattended, severe cavitation can limit the life expectancy of the valve. It can also create excessive seat leakage, distort flow characteristics, or cause the eventual failure of the pressure vessels (valve body, piping, etc.). In some severe high-pressure-drop applications, valve parts can be destroyed within minutes by cavitation.

In general, five conditions must be present to produce cavitation. First, the fluid must remain a liquid both upstream and downstream from the valve. Second, the liquid must not be at a saturated state when it enters the valve or the pressure drop will create a residual vapor downstream from the valve. Third, the pressure drop at the vena contracta must drop below the vapor pressure of the process fluid. Fourth, the outlet pressure must recover at a level above the vapor pressure of the liquid. Fifth, the liquid must contain some entrained gases or impurities, which act as a "host" for the formation of the vapor bubble. This host is sometimes called the *nuclei.* The nuclei are contained in the process fluid as either microscopic particulates or dissolved gases. Since most process fluids contain either particulates or dissolved gases, the chances of forming vapor bubbles are very likely. In theory, if the liquid was completely nuclei-free, some experts believe that cavitation would not occur; however, this would be nearly impossible, especially considering the effects of thermodynamics.

The creation and implosion of the cavitation bubble involve five stages: First, the liquid's pressure drops below the vapor pressure as velocity increases through the valve's restriction. Second, the liquid expands into vapor around a nuclei host, which is either a particulate or an entrained gas. Third, the bubble grows until the flow moves away from the vena contracta and the increasing pressure recovery inhibits the growth of the bubble. Fourth, as the flow moves away from the vena contracta, the area expands—slowing velocity and increasing pressure. This increased pressure collapses or implodes the bubble vapor back to a liquid. Fifth, if the bubble is near a valve surface, the force of the implosion is directed toward the surface wall, causing material fatigue.

The bubbles created by cavitation are much smaller and more powerful than bubbles caused by normal boiling. This release of energy by the imploding bubbles can easily be heard as noise in the valve or in the downstream piping. The noise generated in the early stages of cavitation is described as a popping or cracking noise, while extensive cavitation produces a steady hiss or sizzling noise. Some describe the noise as gravel rolling down the piping. Noise is normally complemented by excessive vibration, which can cause metal or piping fatigue or miscalibration or malfunctioning of sensitive instrumentation. In some cases, the vibration can be minimized by anchoring the valve or piping securely to floors, walls, etc.

The most permanent damage caused by cavitation is the deterioration of the interior of the valve created by the imploding bubbles. As the bubbles expand in the vena contracta, they move into the downstream portion of the valve and then implode as the pressure recovery occurs. If the bubbles are near a metal surface, such as a body wall, they have a tendency to release the implosion energy toward the wall. This phenomenon occurs when unequal pressures are exerted upon the bubble. Since the fluid pressure is less on the side of the cavitation bubble closest to a nearby object, the energy of the implosion is channeled toward that surface (Fig. 9.5). This principle is identical to the implosion of a depth charge in antisubmarine warfare.

With cavitation, the real damage occurs in the second half of the process, when the bubbles implode. This energy burst toward the metal surface can tear away minute pieces of metal, especially if the pressure intensity reaches or surpasses the tensile strength of the valve material. These shock waves have been reported to be as high as 100,000 psi (6900 bar). This initial destruction is magnified since the drag in torn metal surface attracts and holds other imploding bubbles, causing even more cavitation damage. Valve parts damaged by cavitation have a pitted appearance or feel like a sandblasted surface (Fig. 9.6). The appearance of cavitation damage is far different from flashing or erosion damage, which appears smooth. Another possible long-term effect of cavitation is that it may attack a material's coating, film, or

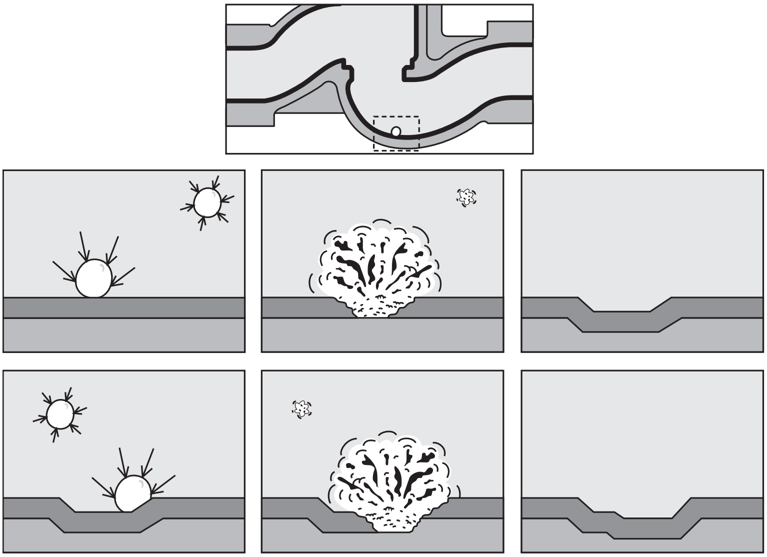


Figure 9.5 Implosion of cavitation bubbles by a valve-body wall. (*Courtesy of Valtek International*)

oxide, which will open up the base material to chemical or corrosion attack.

The hardness of the metal plays a large role in how easily the metal can be torn by the cavitation bubbles. Soft materials, such as aluminum, yield easily to the forces generated by cavitation bubbles and

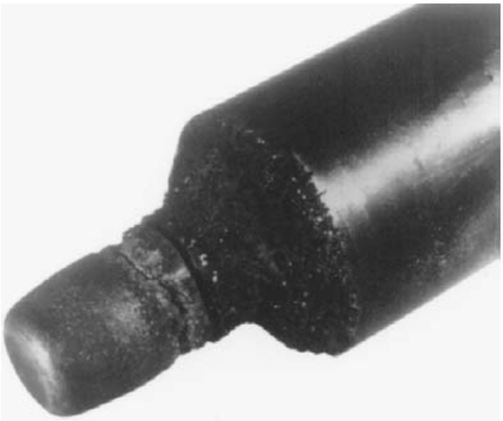


Figure 9.6 Plug damaged by cavitation. (*Courtesy of Fisher Controls International, Inc.*)

tear away quickly. Hardened materials are better able to withstand the effects of cavitation, and only after a period of time will they fatigue and begin to wear. No material can resist cavitation indefinitely. Even the hardest materials will eventually wear away.

Another serious side effect of cavitation is decreased performance in the valve and reduced efficiency in the process system. When cavitation occurs, the valve's ability to convert the entire pressure drop to mass flow rate is diminished. In other words, cavitation can cause less flow through the valve, generating a smaller C_v in actual service than what was originally calculated.

Cavitation can be controlled or eliminated by one of three basic methods: first, by modifying the system; second, by making certain internal body parts out of hard or hardened materials; or third, by installing special devices in the valve that are designed to keep cavitation away from valve surfaces or prevent the formation of the cavitation itself.

9.2.2 Incipient and Choked Cavitation

As the downstream pressure is lowered, creating a larger pressure drop, the advent of cavitation is called *incipient cavitation.* When damage occurs to the vessel, that stage is known as *incipient cavitation damage.* As the flow increases, it will eventually become choked, which is called *choked cavitation.* This linear relationship is shown in Fig. 9.7, which is based on the linear relationship between the flow rate Q and

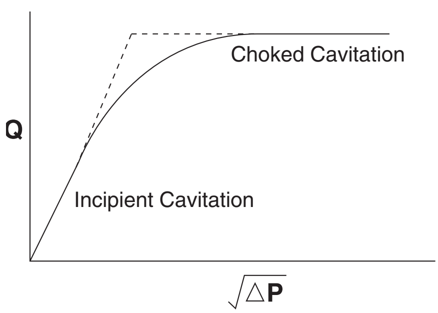


Figure 9.7 Fluid plot of flow vs. $\sqrt{\Delta P}$ and points of incipient and choked cavitation. (*Courtesy of Valtek International*)

the square root of the pressure drop $\sqrt{\Delta P}$. The constant of proportionality of this relationship is based upon the equation

$$Q = C_v \sqrt{\frac{\Delta P}{GS}}$$

where Q = flow rate
C_v = flow coefficient
ΔP = pressure drop
G_S = specific gravity

9.2.3 Cavitation Indices

Over the years, cavitation experts have developed a number of cavitation indices to predict the possibility of cavitation in process equipment, including valves. The ability to predict cavitation is critical to the design and application of the valve. For example, if cavitation exists, the valve can be fitted with special trim to minimize the effects or eliminate cavitation altogether. Certain parts, such as the plug or seat, can be made from hard or hardened materials, or the process system can be changed to minimize the pressure drop through the valve so that cavitation does not form.

For many years, the valve industry used the *flow curve cavitation index* K_C, which shows the effect of cavitation on the linear relationship between the flow rate and the square root of pressure drop. The index K_C is still in use today with some manufacturers and is occasionally used in calculations as

$$K_C = \frac{P_1 - P_2}{P_1 - P_V} = \frac{\Delta P}{P_1 - P_V}$$

where K_C = cavitation index
P_1 = valve inlet pressure
P_2 = valve outlet pressure
P_V = vapor pressure of liquid (at valve inlet and vena contracta)

The cavitation index assumes that a valve may function without cavitation at any pressure drop less than the pressure drop calculated with the index K_C. The basic problem associated with the cavitation index K_C is that it does not take into consideration any prechoked cavitation conditions, which may be just as damaging to the valve. Table 9.1 provides several common K_C values for a number of valve styles.

Table 9.1 Typical K_c Values†

Valve Style	K_c
Butterfly	0.50 K_M*
Ball	0.67 K_M
Rotary Plug	K_M
Globe with Hardened Trim (Cage Characterized)	K_M
Globe (Plug Characterized)	0.85
Globe with special trim	1.0

†*Data courtesy of Fisher Controls International, Inc.*
*K_M is equal to F_L^2 (valve recovery coefficient).

A more useful cavitation index for valves is σ, which was approved in 1995 by the Instrument Society of America. In general terms, σ is a ratio of forces that resist cavitation to forces that promote cavitation and is written as

$$\sigma = \frac{P_2 - P_V}{P_1 - P_2}$$

where σ = cavitation index
P_1 = upstream pressure (measured one pipe diameter upstream from the valve)
P_2 = downstream pressure (measured five pipe diameters downstream from the valve)
P_V = liquid vapor pressure (at flowing temperature)

As σ becomes larger, less cavitation damage is occurring inside the valve. Inversely, as σ becomes smaller, cavitation damage is increasing. If σ is at zero or is a negative number, flashing is occurring. σ is expressed in two forms: *Incipient σ* is the value that indicates when cavitation is beginning. *Choked σ* is the value that indicates when

Table 9.2 Typical σ Values†,‡

Valve Style	Flow Direction	Trim Size	Incipient σ	Choked σ
Globe	over the plug	full area	0.73	0.38
	over the plug	reduced	0.93	0.56
	under the plug	full/reduced	0.52	0.52
Butterfly	60° open	full	1.40	0.73
	90° open	full	3.16	2.19
Ball	60° open	full	1.40	0.64
	90° open	full	5.20	2.19
Globe with special trim	under the plug	full/reduced	0.30 to 0.001	*

†*Data courtesy of Valtek International.*
‡*Note:* For estimation only; sigmas may vary according to particular valve design.
*Choking will not occur when properly applied.

choked flow or full cavitation is occurring. If the calculated σ falls between the incipient σ and choked σ values, some measures should be taken (using special trim, hard materials, or process changes) to avoid cavitation damage in the valve. Both incipient σ and choked σ values are determined through laboratory and field testing by the valve manufacturer. Examples of typical σ values for a given valve style are shown in Table 9.2.

9.2.4 σ Example A

To show an application of incipient σ and choked σ, the following example is used:

Fluid	Water
Temperature	80°F
Vapor pressure P_V	0.5 psia
Upstream pressure P_1	200 psia
Downstream pressure P_2	55 psia
Valve type	Single-seated globe valve, 100 percent open, flow-over-the-plug

The value for σ is

$$\sigma = \frac{P_2 - P_V}{P_1 - P_2} = \frac{55-0.5}{200-55} = 0.38$$

Referring to Table 9.2, incipient σ begins at σ = 0.73 (for a single-seated globe valve that is at 100 percent open with flow under the plug) and the choked σ occurs at σ = 0.38. In this example severe cavitation damage is occurring and the valve is choked and cannot increase flow any further.

9.2.5 σ Example B

Using the same valve in example A, new service conditions are applied to illustrate a cavitating, but nonchoking, situation:

Fluid	Water
Temperature	80°F
Vapor pressure P_V	0.5 psia
Upstream pressure P_1	500 psia
Downstream pressure P_2	200 psia
Valve type	Single-seated globe valve, 100 percent open, flow-over-the-plug

Using the σ index equation for these operating conditions, we find that σ is significantly higher:

$$\sigma = \frac{200 - 0.5}{500 - 200} = 0.67$$

This σ value is above the choked σ value for this valve (which is σ = 0.38) and indicates that the valve is not experiencing choked flow. However, this value is below the incipient σ value, which indicates that the valve is experiencing cavitation and damage may be occurring in the valve.

9.2.6 System Modifications to Prevent Cavitation

To eliminate the formation of cavitation, the answer lies in reducing the pressure from the upstream side to the downstream side, preventing the pressure at the vena contracta from falling below the vapor pressure. When this reduction is made, vapor bubbles are not formed and cavitation is avoided. This normally requires special trim or modifications of the system to provide a series of smaller pressure drops that result in the required downstream pressure. By taking a series of pressure drops, rather than one large drop, the service can be modified so that the pressure will not fall below the vapor pressure (Fig. 9.8).

In some cases, the process system and related service conditions, or the process equipment used in the system, can be modified to minimize the effects of cavitation. Even the type of valve or number of valves used in one system can modify cavitation effects. One system solution is the injection of air into the system. At first this may appear to worsen a bad situation as the addition of air will provide additional nuclei that can play host to vapor bubbles and increase the damage. However, cavitation studies have shown that at a certain point, additional air content to the process stream disrupts the explosive force of the imploding bubbles and can reduce the overall damage. This solution works best with large valves dumping into tanks or when large particulates in the flow stream prevent the use of cavitation-control trim, anticavitation trims, or downstream devices.

The intensity of cavitation can be modified by varying the downstream pressure, if possible. Increasing the downstream pressure may decrease the pressure drop sufficiently to avoid the pressure falling below the vapor pressure, although this will decrease the process flow capacity. Lowering the downstream pressure may not seem to be an option, since a greater pressure drop would create even more vapor

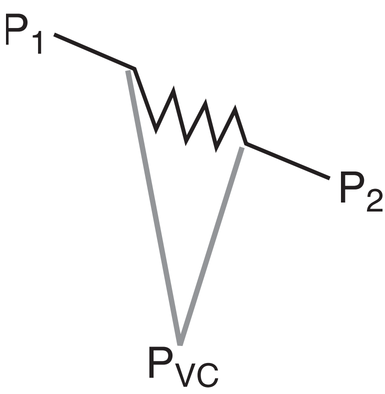


Figure 9.8 Flow curve showing gradual pressure reduction without dropping below vapor pressure. (*Courtesy of Valtek International*)

Figure 9.9 Back-pressure device used with globe valves. (*Courtesy of Fisher Controls International, Inc.*)

bubbles. However, the increased pressure differential provides less cavitation intensity.

A *downstream back-pressure device* is a device that is installed between the valve and the downstream piping to lower the pressure drop taken by the valve while increasing the resistance of the downstream and the downstream pressure. Although a wide variety of back-pressure devices are available today, a typical device is shown in Fig. 9.9. Because back-pressure devices may limit the flow capacity of the valve, a larger valve or different trim reduction may be required. The device must be examined periodically to ensure that it is not wearing out through erosion or minimal cavitation. A worn back-pressure device will ultimately decrease the downstream pressure, increase the pressure drop, and create cavitation. In addition, the user must be careful to use the back-pressure device within the limits of the flow range; otherwise cavitation can occur after the device in the downstream piping. A back-pressure device is commonly used with a rotary valve (Fig. 9.10), which cannot be designed to include an internal anti-cavitation device because of design limitations. Not only does the device perform the function of raising the downstream pressure, but it also controls existing cavitation by allowing it to occur in the small tubes where cavitation intensity is lower and can be absorbed by the tubes themselves. One caution applies when using a downstream cavitation-control device with a rotary valve: As the rotary valve begins to open (less than 30° open), the most severe cavitation intensity may occur in the outlet half of the body before it reaches the downstream device, causing serious damage between the valve and the device.

Figure 9.10 Back-pressure device used with rotary valves (*Courtesy of Fisher Controls International, Inc.*)

Some valve designs can be used to minimize cavitation damage. For example, while a globe-style linear valve exposes the bottom of the valve body to the cavitation, an angle-style linear valve may experience less damage since the flow continues straight down from the vena contracta and is directed into the center of the piping—no valve or piping surface is directly bombarded with vapor bubbles.

As a general rule, the face-to-face dimension of rotary valves—such as butterfly, eccentric plug, and ball valves—is far less than comparably sized globe valves. Therefore, the vena contracta generated by a rotary valve is most likely to occur not in the valve itself, but in the downstream piping. In this case, cavitation might be allowed and a segment of the downstream piping routinely replaced as part of periodic maintenance. Another option is to install two or three valves in lieu of one valve, allowing the pressure drop to be taken over more than one restriction and preventing a large pressure drop from falling below the vapor pressure. This option is more expensive in terms of additional valves, but may still be less expensive than obtaining a specially engineered valve. This solution has one disadvantage, however, that may occur when the first valve opens against a high upstream pressure. For a very short time, the first valve will take the entire pressure drop until the flow reaches the second valve. This may result in cavitation damage to the first valve in some unusual cases. In such an application, installing anticavitation trim in the valve may be a better option.

9.2.7 Materials of Construction

Cavitation easily attacks softer materials, which have a lower tensile strength than harder materials. One of the most common methods of dealing with cavitation is to make the valve out of hard or hardened materials (those materials exceeding a Rockwell hardness of 40). Generally, materials such as chrome–molybdenum and steel alloys (ASTM SA-217 Grade WC9 and C5) are used for the body, while solid alloy hard-facing, a solid alloy overlay with 316 stainless steel or 416 stainless steel, is used on trim parts.

One advantage to using angle-style valves in cavitating service is that one of three options—a hardened seat ring, an extended *Venturi seat ring*, or body liner—can be installed in the downstream portion of the valve. This part can then be replaced periodically after cavitation damage compromises the part. These liners can be made from Alloy 6 or 17-4ph stainless steels.

Because nonmetallic materials, such as PTFE liners or bodies made from plastic, have lower yield values than metal, they are more prone to cavitation damage and are not recommended for cavitating services.

9.2.8 Cavitation-Control Devices

Some valves can be equipped with special trims that will direct the cavitating flow, along with vapor bubbles, away from critical metal surfaces. Since cavitation-control trims are not as highly engineered as trims designed to prevent cavitation, they generally cost less and are simpler in concept.

The design shown in Fig. 9.11 illustrates how this principle works. In flow-over-the-plug applications, a special retainer with specially designed holes is placed inside the valve. As the close-fitting plug lifts out of the seat, the holes in the special retainer are exposed and allow the flow to pass through the seat. In this case, the holes in the retainer are the restrictions and cavitation occurs at that region. Because the holes are directly opposite each other, the cavitating flow from one hole impinges on the opposite hole's flow, thus keeping the cavitation in the center of the retainer. At this point, the only metal surface affected by the cavitation is the bottom of the plug, which can be made from hardened material. Since the middle of the bottom of the plug is flat

Figure 9.11 Laboratory experiment showing diversion of cavitation away from boundary surfaces using cavitation control trim. (*Courtesy of Valtek International*)

and not necessary for shutoff, it can be sacrificed over a period of time. Only when the deterioration reaches the plug's seating surface will the plug need replacement.

Such cavitation-control designs can be engineered with a wide range of C_vs and in either linear or equal-percentage flow characteristics. Because flow must always be over the plug, pressure-balanced trim is necessary in fail-open applications to prevent instability near the seat.

9.2.9 Cavitation-Elimination Devices

Some valves are designed to prevent the formation of cavitation altogether. Although it is a more expensive option, in some applications anticavitation design features are the only choice. Globe-style valves can be designed with special retainers or cages, which use either (or a combination of) a tortuous path, pressure-drop staging, and/or expanded flow areas to decrease the pressure drop through the valve and to prevent cavitation.

A *tortuous-path device* uses a series of holes and/or channels to increase the flow resistance through the trim (Fig. 9.12). This decreases the overall velocity through the valve, thereby reducing the pressure recovery. In addition, a tortuous path creates pockets of high and low

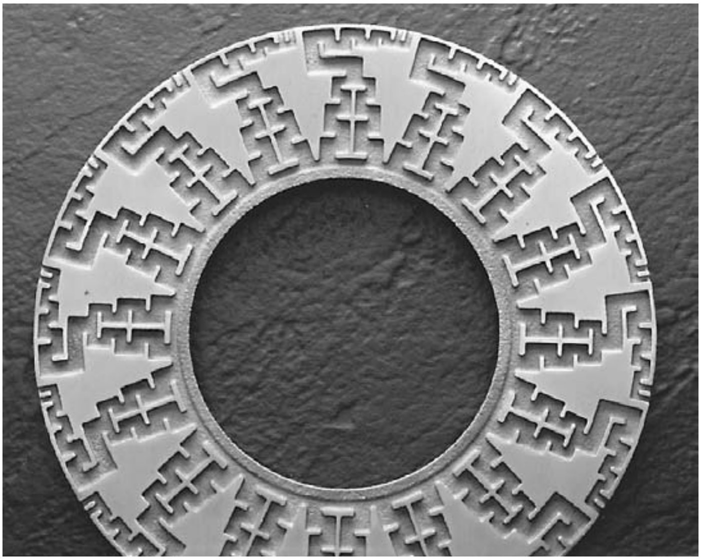


Figure 9.12 Tortuous-path trim for velocity reduction. (*Courtesy of Control Components Inc., an IMI company*)

pressures as the flow moves through the trim, creating substantial frictional losses. To illustrate the effect of frictional loses in this trim, the losses associated with a single 90° piping elbow are equal to 60 ft of straight pipe. The typical tortuous path uses a series of right-angle turns—similar in principle to a 90° elbow—to create frictional losses and lower velocities. Each turn reduces the velocity by one velocity head ($V_H = \rho V^2/2$). This velocity reduction can be calculated by changing the velocity equation as follows:

$$V = \sqrt{2SGV_H} \qquad \text{to} \qquad V = \sqrt{\frac{2SG\,V_H}{N}}$$

where V = required velocity (below sonic or generally below 300 ft/s)
SG = specific gravity
V_H = velocity head
N = number of turns (in series) in each passageway

Determining the number of turns is critical in the design of tortuous-path designs, since they determine the overall velocity-head loss, as well as the diameter of the stack.

Another method of decreasing the pressure drop is by *staged pressure reduction,* in which several smaller restrictions are taken through a trim rather than one large restriction. In effect, this creates a number of small pressure drops in lieu of one large pressure drop (refer again to Fig. 9.8). As the flow moves through the trim, it reaches the first restriction or stage, absorbing a certain amount of energy and taking a small pressure drop. As the flow continues, it provides a lower inlet pressure to the next stage where another pressure drop is taken, and so forth. The net result is that the entire pressure drop is taken over a series of small pressure drops without falling below the vapor pressure at the vena contracta, yet the overall pressure drop remains unchanged. In some cases, for whatever reason, systems pressures may change. This change may exceed the operating parameters of the valve and create cavitation in the valve, even if a staged-pressure-reduction trim is used. In this case, although cavitation is occurring, the anticavitation trim will continue to modify the pressure differential and the cavitation will not be as severe.

Related to the staged-pressure-drop concept is the *expanded flow-area* concept, in which the flow continues through several restrictions in the trim, the flow area is increased at each stage (Fig. 9.13). With compressible fluids, as dictated by the law of conservation of mass flow, the flow area must increase as the fluid pressure and density are reduced. In this concept, the largest portion of the pressure drop is taken at the first restriction, and then succeeding smaller portions of the pressure drop are taken over the following restrictions. When the flow reaches the last restriction, a minimal pressure drop is taken and

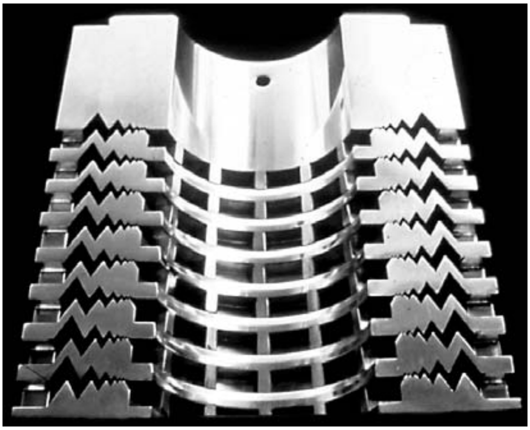


Figure 9.13 Expanding tooth trim for staged pressure reduction. (*Courtesy of Valtek International*)

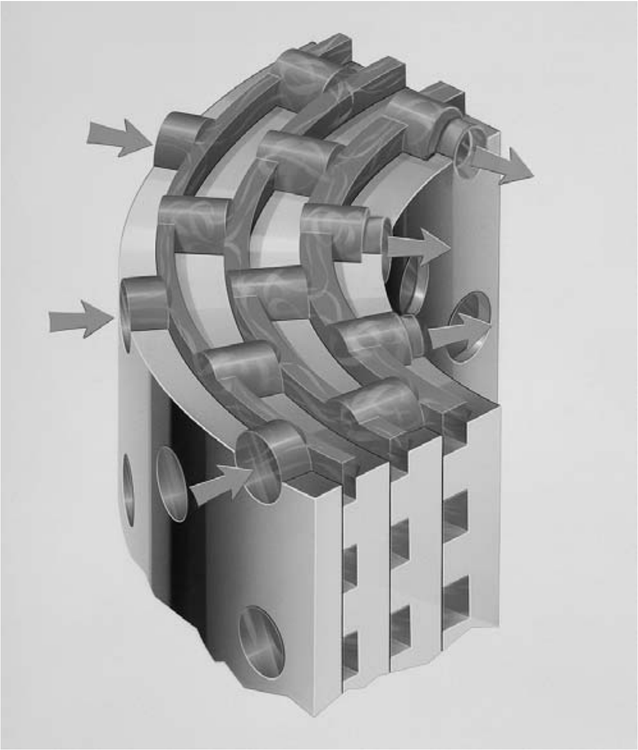


Figure 9.14 Anticavitation trim with multiple pressure-reduction mechanisms. (*Courtesy of Valtek International*)

the pressure recovery at that point is significantly decreased, preventing cavitation from occurring.

Valve manufacturers have developed a variety of sophisticated trims that use one or a combination of these concepts (tortuous path, staged pressure reduction, and expanded flow areas). For example, Fig. 9.14 shows a flow-over-the-plug trim that directs the flow through a series of close-fitting cylinders with each cylinder acting as a stage. The flow must follow a tortuous path as it travels through a series of 90° angles via the narrow channels and drilled holes, increasing the frictional losses. Pressure reduction is staged through the number of cylinders, allowing the pressure to remain above the vapor pressure. In addition, the channels become progressively deeper and the number of holes increase with each stage, providing expanded flow areas.

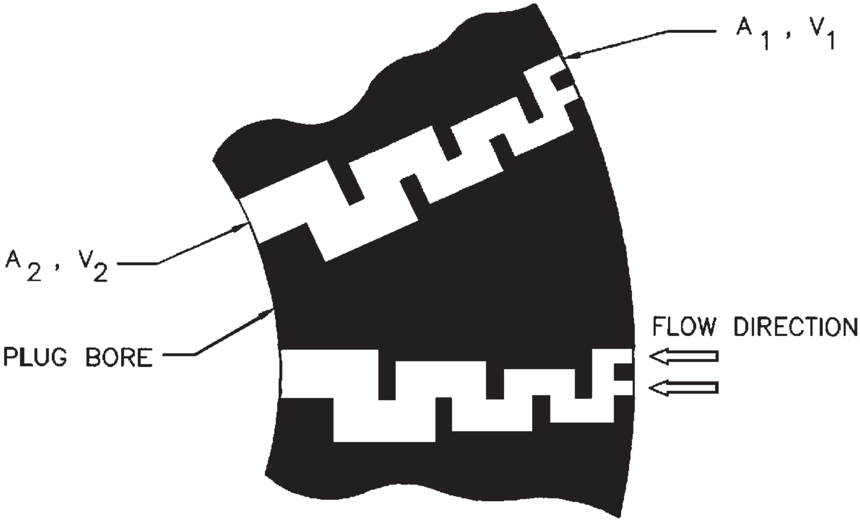


Figure 9.15 Expanding flow area of tortuous-path trim. (*Courtesy of Control Components Inc., an IMI company*)

Another common design that uses these principles is the *expanding tortuous-path trim.* In addition to the velocity control through the right-hand turns, the tortuous pathways can be enlarged, allowing for expanded flow areas (Fig. 9.15). The tortuous path can follow either a horizontal direction with etched disks (Fig. 9.16) or disks made from a punched plate (Fig. 9.17).

Most anticavitation trims follow a linear characteristic, although some designs allow for an equal-percentage characteristic. When the disks or flow areas of the trim are identical throughout the entire stack, the trim follows a linear characteristic. An equal-percentage characteristic is generally obtained by using different disks or passageways that increase the flow as the stroke continues. Another method of modifying an anticavitation linear characteristic is by using a shaped cam in the actuator positioner.

9.2.10 Anticavitation-Trim Sizing

Although methods of sizing a valve with anticavitation trim vary according to different valve manufacturers, the following procedure utilizes σ values and provides a general idea of the steps involved. The first step is to calculate the required C_v for the given application (see

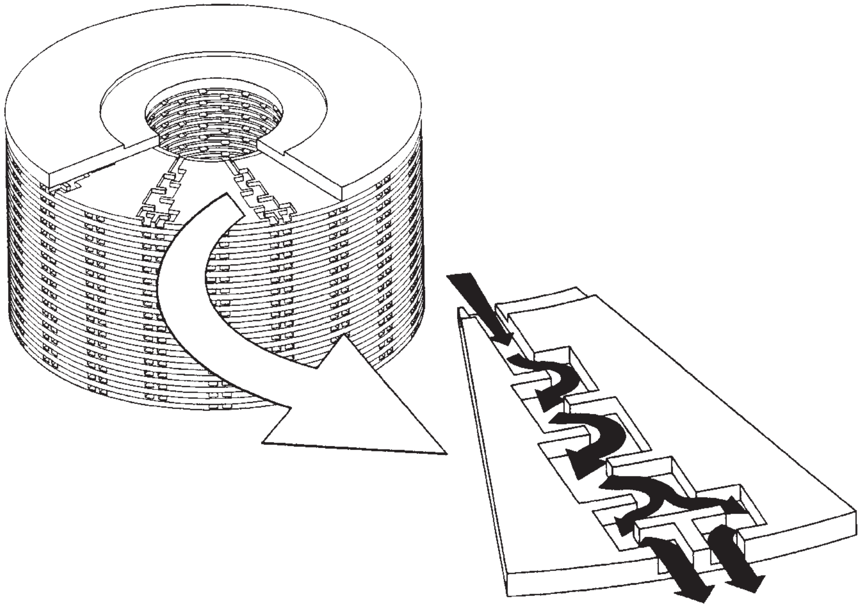


Figure 9.16 Etched tortuous-path trim for horizontal flow. (*Courtesy of Control Components Inc., an IMI company*)

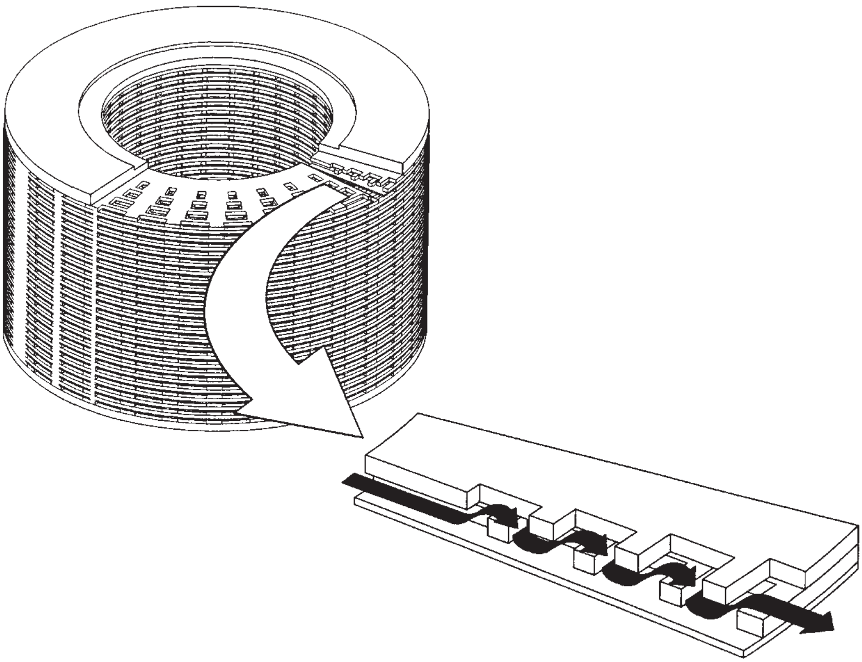


Figure 9.17 Punched tortuous-path trim for vertical and horizontal flows. (*Courtesy of Control Components Inc., an IMI company*)

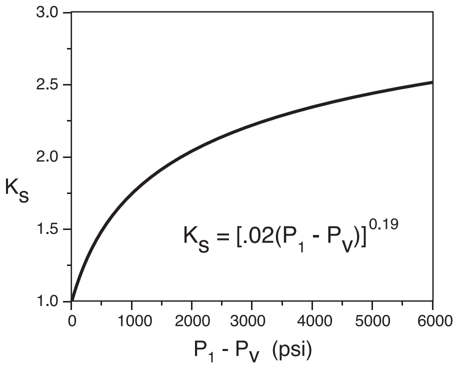


Figure 9.18 Pressure scale for factor K_S (water service), where $K_S = [0.02\ (P_1 - P_V)]^{0.19}$. (*Courtesy of Valtek International*)

Chap. 7). Choked-flow conditions should be considered, and the F_L factor should be adjusted to compensate for the use of the anticavitation trim. In this case, the required C_v for a valve with anticavitation trim will most likely be smaller than a conventional valve. Using the σ formula, the operating σ can be calculated from the flow conditions for the required C_v. The difference between the upstream pressure and the vapor pressure should be calculated $(P_1 - P_V)$. Following this calculation, the K_S factor can be determined by referring to Fig. 9.18 or by using the following calculation:

$$K_S = [0.02(P_1 - P_V)]^{0.19}$$

The service σ can now be calculated for each pressure:

$$\sigma_{\text{service}} = \frac{\sigma}{K_S}$$

The manufacturer of the anticavitation trim provides tables (Table 9.3) that provide C_v and σ values. If the requirement of the service σ is less than the minimum requirement of the calculated σ, then a larger valve with more stages will mostly likely be needed. The velocity of the flow should also be considered to ensure that it does not approach the maximum velocity capacity of the trim.

Common Valve Problems

Table 9.3 Cavitation Trim Sizing Table*

Body Size	Trim No. (Seat Dia.)	Stages	Stroke	C_v	$\sigma_{min.}$	Bore Area
1½	1.38	2	1.50	17	.170	1.77
	1.25	3	1.50	11	.070	1.48
	1.12	4	1.50	6	.020	.99
2	1.38	2	1.50	18	.170	1.77
	1.25	3	1.50	12	.070	1.49
	1.12	4	1.50	7	.020	.99
3	2.50	2	2.50	50	.200	5.41
	2.38	3	2.50	34	.080	4.91
	2.00	4	2.50	20	.025	3.55
	1.62	5	2.50	12	.007	2.41
	1.25	6	2.50	7	.002	1.49
4	3.50	2	3.00	85	.200	10.3
	3.12	3	3.00	54	.080	8.3
	2.75	4	3.00	33	.025	6.51
	2.38	5	3.00	21	.007	4.91
	1.88	6	3.00	12	.002	3.14
6	5.25	2	4.00	175	.200	22.7
	4.75	3	4.00	105	.080	18.7
	4.25	4	4.00	65	.025	15.1
	3.50	5	4.00	40	.007	10.3
	3.00	6	4.00	25	.002	7.67
8	6.50	2	6.00	320	.200	34.5
	6.00	3	6.00	200	.080	29.5
	5.50	4	6.00	130	.025	24.8
	5.00	5	6.00	85	.007	20.6
	4.50	6	6.00	55	.002	16.8
10	8.75	2	7.50	530	.230	61.9
	8.38	3	7.50	350	.090	56.7
	7.88	4	7.50	230	.028	50.3
	7.38	5	7.50	155	.008	44.2
	6.88	6	7.50	105	.002	38.5
12	9.75	2	8.00	640	.230	76.6
	9.00	3	8.00	400	.090	65.4
	8.38	4	8.00	260	.028	56.7
	7.88	5	8.00	180	.008	50.3
	7.38	6	8.00	125	.002	44.2
14	11.00	2	8.00	720	.240	97.2
	10.25	3	8.00	460	.095	84.5
	9.50	4	8.00	300	.030	72.8
	8.75	5	8.00	200	.008	61.9
	8.00	6	8.00	135	.002	51.8

**Courtesy of Valtek International.*

9.2.11 Anticavitation-Trim Sizing Example

The following service conditions apply to this example:

Fluid	Water
Maximum flow	515 gal/min
Inlet temperature	208°F
Inlet pressure	287 psia
Outlet pressure	24 psia
Vapor pressure	13.57
Specific gravity	0.92

Using the flow capacity calculations in Chap. 7, the required C_v is calculated at $C_v = 30$. σ is calculated as follows:

$$\sigma = \frac{P_2 - P_V}{P_1 - P_2} = \frac{287 - 13.57}{287 - 24} = 0.04$$

Using the K_S chart (Fig. 9.18), the K_S is 1.38. Knowing K_S, the allowable σ can be calculated as follows:

$$\sigma_{\text{service}} = \frac{0.04}{1.38} = 0.029$$

Using an anticavitation trim table from the manufacturer (Table 9.3) for an application requiring a C_v of 33 and a σ value of 0.029, the required valve would be a 4-in (DN 100) valve with a four-stage anticavitation trim.

9.2.12 Other Cavitation-Control Solutions

A number of other solutions to cavitation control or elimination exist, such as characterized cages or separation of the valve's seat and the throttling mechanism. In applications where the pressure drop decreases as the plug travel and flow rate increase, a characterizable cage can be used. For example, a typical characterizable cage would have two stages of pressure reduction, the middle portion would have one stage of pressure reduction, and the top portion would have straight-through flow. With this design, cavitation control is provided at the early stages of plug travel, when it is needed most. As the travel continues and the

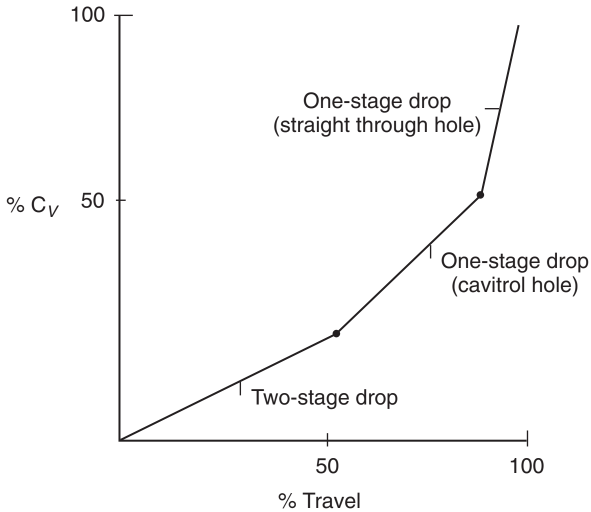


Figure 9.19 Flow curve showing effects of two-stage characterized cage. (*Courtesy of Fisher Controls International, Inc.*)

pressure drop and chance of cavitation decrease, the mechanism allows greater flow with less restriction. Figure 9.19 plots how a characterizable cage works in relation with flow versus travel.

In most flow-over-the-plug applications, the pressure reduction device is located above the seat in the body gallery. However, in some cavitating applications where tight shutoff is important, the body can be designed with the seat separate from the pressure-reduction mechanism. As shown in Fig. 9.20, the seat is located above the anticavitation trim, which is contained in the downstream portion of the valve. The trim area above the seat is designed to take a large flow, hence a lower pressure drop. This design keeps the velocities at a minimum through the seat, which improves the stability of the valve plug close to the seat and makes for easier shutoff.

Traditionally, anticavitation trim is associated with linear throttling valves, although some designs exist for quarter-turn valves. For example, a plug valve can utilize a special plug (Fig. 9.21) to take an additional stage of pressure reduction for those applications where the pressure drop falls just below the vapor pressure. As the plug closes, the grid turns into the flow, taking a pressure drop and preventing cavitation from forming. The grid prevents severe cavitation from forming and channels remaining cavitation away from metal bound-

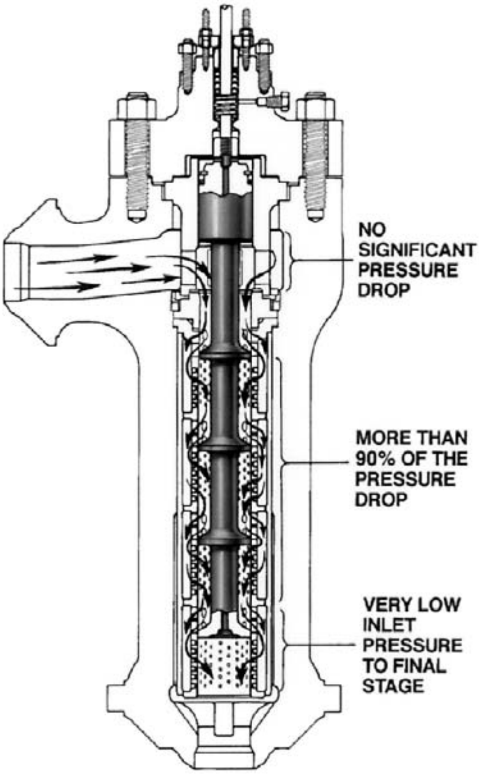


Figure 9.20 Anticavitation trim located downstream from the seating surface. (*Courtesy of Fisher Controls International, Inc.*)

aries. This design also allows large particulates to bounce off the grid and be flushed downstream.

9.3 Flashing

9.3.1 Introduction to Flashing

In liquid applications, when the downstream pressure is equal to or less than the vapor pressure, the vapor bubbles generated at the vena contracta stay intact and do not collapse. This happens because the pressure recovery is high enough for this to happen. As shown in Fig. 9.22, this phenomenon is known as *flashing*. When flashing occurs, the fluid downstream is a mixture of vapor and liquid moving at very

Figure 9.21 Anticavitation plug for quarter-turn plug valves. (*Courtesy of The Duriron Company, Valve Division*)

high velocities, resulting in erosion in the valve and in the downstream piping (Fig. 9.23).

9.3.2 Controlling Flashing

Unfortunately, eliminating flashing completely involves modifying the system itself, in particular the downstream pressure or the vapor pressure. However, not all systems are easily modified and this may not be an option. The location of the valve may be considered—especially if the valve empties the downstream flashing flow into a large vessel, tank, or condenser. Placing the valve closer to the larger vessel will allow the flow to impinge into the larger volume of the vessel and away from any critical surfaces. When flashing occurs, no solution can be designed into the valve, such as is the case with anticavitation or cavitation-control trim, except to offer hardened trim materials.

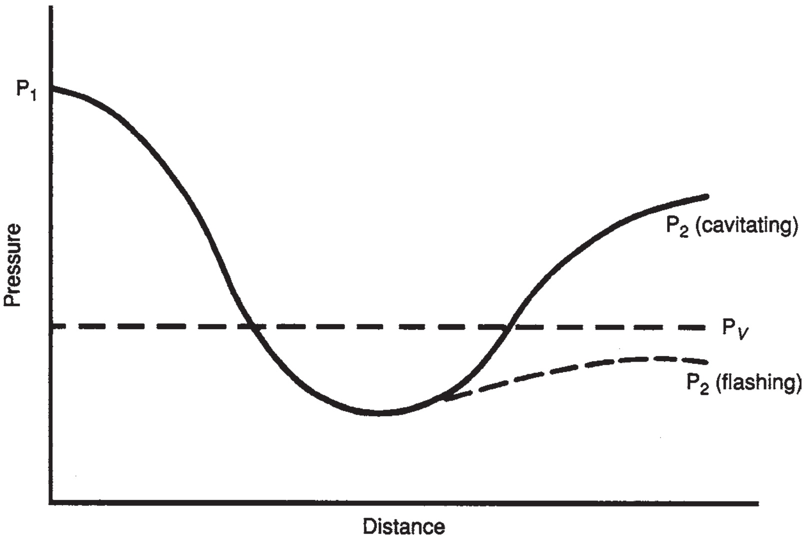


Figure 9.22 Pressure curve showing outlet pressure below the vapor pressure, resulting in flashing. (*Courtesy of Fisher Controls International, Inc.*)

9.4 Choked Flow

9.4.1 Introduction to Choked Flow

Choked flow occurs in gases and vapors when the velocity of a process fluid achieves sonic speeds in the valve or the downstream piping. As the fluid in the valve reaches the valve restriction, the pressure

Figure 9.23 Plug damaged by flashing. (*Courtesy of Fisher Controls International, Inc.*)

decreases and the specific volume increases until sonic velocities are achieved. When choked flow occurs, the flow rate is limited to the amount of flow that can pass through the valve at that point and cannot be increased unless the service conditions are changed.

In liquid applications, the presence of vapor bubbles caused by cavitation or flashing significantly increases the specific volume of the fluid. This increase rises at a faster rate than that generated by the pressure differential. In liquid choked flow conditions, if upstream pressure remains constant, decreasing the downstream pressure will not increase the flow rate. In gas applications, the velocity at any portion of the valve or downstream piping is limited to Mach 1 (sonic speed). Hence, the gaseous flow rate is limited to the flow that is achieved at sonic velocity in the valve's trim or the downstream piping.

As noted in Sec. 7.2, choked flow must be considered when sizing a valve, especially when considering $\Delta P_{\text{allowable}}$ and the valve recovery coefficient K_M.

9.5 High Velocities

9.5.1 Introduction to High Velocities

In general, large pressure differentials create high velocities through a valve and in downstream piping. This in turn creates turbulence and vibration in liquid applications and high noise levels in gas applications. The velocity is inversely related to the pressure losses and gains as the flow moves through the vena contracta (Fig. 9.24). The velocity

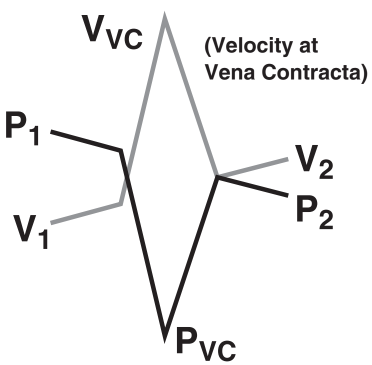


Figure 9.24 Velocity and pressure profiles as flow travels through an orifice restriction. (*Courtesy of Valtek International*)

reaches its maximum peak just slightly after the vena contracta, which is when the pressure is at its lowest point.

9.5.2 Velocity Limits

The following general rules apply to velocities: Liquids should generally not exceed 50 ft/s (15.2 m/s) (or 30 ft/s or 9 m/s in cavitating services). Gases should not exceed sonic speeds (Mach 1.0). And, mixtures of gases and liquids (such as flashing applications) should not exceed 500 ft/s (152 m/s). These general rules can vary, however, according to the size of the valve. For example, smaller-sized valves can normally handle higher velocities, while larger valves only handle lower velocities.

Generally, process liquids that have temperatures close to the saturation point must keep the velocity at or under 30 ft/s (9 m/s) to avoid the fluid pressure from falling below the vapor pressure and creating cavitation. The 30-ft/s rule also applies to cavitating services, where higher velocities result in greater cavitation damage to downstream piping. Lower velocities will also reduce problems associated with flashing and erosion.

9.6 Water-Hammer Effects

9.6.1 Definition of the Water-Hammer Effect

In liquid applications, whenever flow suddenly stops, shock waves of a large magnitude are generated both upstream and downstream. This phenomena is known as the *water-hammer effect.* It is typically caused by a sudden pump shutoff or a valve slamming shut when the closure element is suddenly sucked into the seat ("bathtub stopper effect") as the valve nears shutoff. In control valves, the bathtub stopper effect is caused by a low-thrust actuator that does not have the stiffness to hold a position close to the seat. In some cases, valves with a quick-open or an installed linear flow characteristic can also cause water-hammer effects.

Although water hammer generates considerable noise, the real damage occurs through mechanical failure. Because of the drastic changes from kinetic energy to the static pipe pressure, water hammer has been known to burst piping or damage piping supports as well as damage piping connections. In valves, water hammer can create severe shock through the trim, which can cause trim, gasket, or packing failure.

9.6.2 Water-Hammer Control

With valves, the best defense against water hammer is to prevent any sudden pressure changes to the system. This may involve slowing the closure of the valve itself or providing a greater degree of stiffness as the closure element approaches the seat. To avoid pressure surges, the valve should be closed with a uniform rate of change. In some cases, when a quick-open or installed linear characteristic (which approaches the quick-open characteristic) is used, a change to an equal-percentage characteristic may be required. With control valves that must throttle close to the seat, using an exceptionally stiff actuator—such as a spring cylinder pneumatic actuator or a hydraulic actuator—or a special notch in the stroke collar of a manual quarter-turn operator will minimize or prevent the bathtub stopper effect. Adding some type of surge protection to the piping system can also reduce water hammer. This may be accomplished with a pressure-relief valve or a rubber hose containing a gas, which can be run down the length of the piping. In addition, gas may be injected into the system. Gas injection reduces the density of the fluid and provides some compressibility to handle any unexpected surge.

9.7 High Noise Levels

9.7.1 Introduction to Noise

One of the most noticeable and uncomfortable problems associated with valves is noise. To the human, not only can noise be annoying, but it can also cause permanent hearing loss and unsafe working conditions. Extensive studies have shown that human hearing is damaged by long exposures to high noise levels. Hearing damage is cumulative and irreversible and begins with the loss of high frequencies. As hearing loss continues, lower frequencies are eventually lost, which affects the ability to understand normal speech patterns. When subjected to noise at lower frequencies, the performance of human organs, such as the heart or the liver, can also be affected. In addition, noise and the accompanying vibration can affect the valve's performance and cause fatigue in the valve, piping, and nearby process equipment.

In essence, noise is generated when vibration produces wide variations in atmospheric pressure, which are then transferred to the eardrums as noise. Noise spreads at the speed of sound [which is 1100 ft/s (335 m/s) or 750 mi/h (1200 km/h)]. Noise in valves can be created in a number of different ways; however, the most common cause is

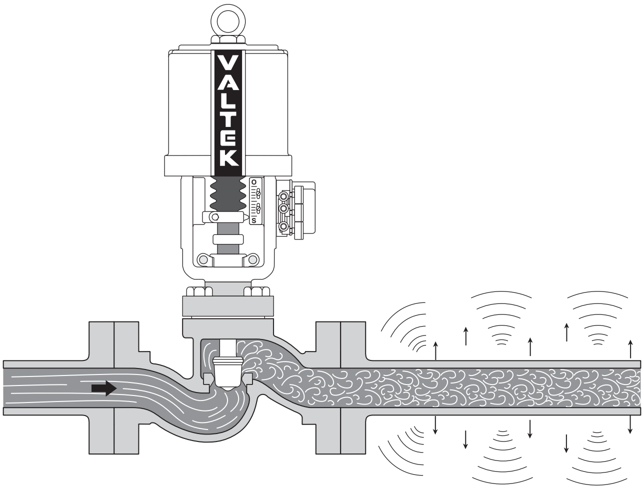


Figure 9.25 Downstream pipeline vibration caused by valve turbulence. (*Courtesy of Valtek International*)

turbulence generated by the geometry of the valve, which is radiated by the downstream piping (Fig. 9.25). In many cases, noise does not radiate from the valve itself, because the body itself is stiff and unyielding.

Process turbulence can create mechanical vibration of the valve or valve components. Such noise is caused by vibrations created by random pressure fluctuations within the body assembly or the fluid impinging on obstacles in the fluid steam, such as the plug, disk, or other closure element. This often causes a rattling noise, as the closure element impacts continually against its guides. Because the frequency level is less than 1500 hertz (Hz), it is normally not annoying to the hearer. However, this rattling of the stem or shaft with the guides can damage critical guiding or seating surfaces. One side benefit of a rattling noise associated with valve parts is that such secondary noise provides a warning signal that turbulence is taking place inside the valve and that corrections may be necessary before failure occurs. Vibration can also be caused by certain valve parts or accessories that resonate at their natural frequency, which is often found in lower noise levels—less than 100 dBA. This type of noise is characterized by a single tone or hum (with a frequency between 3000 and 7000 Hz). Although this noise is not an annoyance, it does produce high levels of stress in the material, which may fatigue the material of the component and cause it to weaken. Noise can also be generated by hydrodynamic and aerodynamic fluid sound. With liquid applications, hydrodynamic noise is caused by the turbulence of the flow, cavitation, flashing, or the high velocities that occur as the flow moves through the vena contracta. Generally, however, the noise generated by the liquid flow does not occur at high levels and can be tolerated by workers. In severe cavitating or flashing applications, noise levels can reach higher levels and must be dealt with by changing the process or installing anticavitation components in the valve.

When cavitation occurs in liquid services, the noise generated by the implosion of the bubbles occurs just slightly downstream from the valve and sounds similar to rocks flowing down the pipe. Overall, this noise is simply irritating and does not reach levels that cause harm. On the other hand, aerodynamic noise is often a problem for nearby workers when dealing with gaseous services. It generates frequencies in the range between 1000 and 8000 Hz, the range that is most sensitive to the human ear. In many cases, gaseous noise levels rise above 100 dBA (decibels for human hearing) and in some extreme cases, above 150 dBA.

In general, the noise level is a function of the velocity of the flow stream. As the pressure profile indicates, when pressure drops at the vena contracta, the velocity increases proportionately. Because of the vena contracta, high noise levels can be generated as velocity increases through the restriction, even though the velocity decreases as low as Mach 0.4 as the flow reaches the downstream side of the valve.

The mechanisms used in cavitation control—tortuous paths, staged pressure drops, and expanding flow areas—can also be applied in order to lower sound levels in gas services. In addition, the mechanism in providing a flow path with sudden expansions and contractions is also used to lower aerodynamic noise.

9.7.2 Sound Pressure Level

Vibrations or atmospheric pressure changes are based upon the number of cycles per second (hertz). A young hearer has a hearing range of 20 to 20,000 cycles per second (20 kilocycles or 20 kHz). The intensity of sound that is heard by a hearer is expressed as in units as *decibels.* In order to understand decibels, the relationship of microbars to 1 Newton per square meter must be understood. One μbar is one-millionth of a normal atmospheric pressure and 10 μbar equal 1 N/m^2. Zero decibel (dB) is defined as 0.00002 N/m^2, which is considered the absolute threshold of hearing for a young hearer. Decibels are applied to three common weighted sound levels; dBA for human hearing, dBB

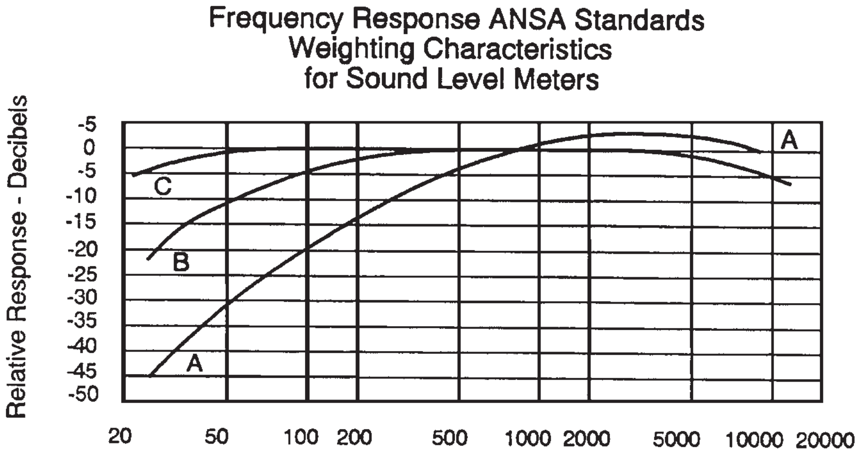


Figure 9.26 Decibel curves for *A*, *B*, and *C* scales. (*Courtesy of Valtek International*)

for an intermediate range, and dBC for equipment (Fig. 9.26). In nearly all cases, dBA is the most commonly applied sound level because it is weighted to account for the sensitivity of human hearing. With the dBA-weighted scale, the loudness of a particular noise at a certain frequency is compared to the loudness given for a 1000-Hz level. In other words, at 1000 Hz, the dBA value is zero. With the 1000-Hz scale, the sound pressure level is equal to the actual dB level. However, if a different hertz level is applied, the noise may sound less loud. For example, with 200 Hz, a sound measured at approximately 120 dB is lower in loudness (110 dB). Or in other words, the correction for dBA at 200 Hz is −10, as shown in Fig. 9.26. Table 9.4 indicates a number of common sounds measured in dBA levels.

Valve noise is calculated as a *sound pressure level,* which is defined as

$$\mathrm{SPL} = 20 \log_{10} \frac{P}{0.0002\ \mu\mathrm{bar}}\ \mathrm{dB}$$

where SPL = sound pressure level
P = root-mean-square sound pressure (N/m^2)

Approximately 90 dB equals one sound pressure level, and this level doubles every 6 dB. Therefore, 96 dB is two times the sound pressure level and 102 dB is four times the sound pressure level. To illustrate the magnitude of this change, the range between 80 and 120 dB is

Table 9.4 Typical dBA Sound Levels*

Sound	dBA
Threshold of hearing	0
Soft whisper at 5 feet (1.3 meters)	10
Average home residence	50
Busy highway	57
Freight train at 100 feet (25.4 meters)	67
Subway train at 20 feet (5.1 meters)	80
Textile weaving plant	83
Electric furnace	90
Pneumatic peen hammer	94
Riveting machine	100
Discomfort threshold	110
Jet take-off at 200 feet (50.8 meters)	123
50 HP siren at 100 feet (25.4 meters)	135
Pain threshold	140

**Courtesy of Valtek International.*

100 times the sound pressure level. Noise radiating from a single point decays at a rate of 6 dB for every doubling of distance. However, if the noise is radiating from a radial line source—such as noise radiating from a pipeline—the noise decays at half that rate or 3 dB for every doubling of distance. Conversely, hard surfaces close to the noise source can increase the noise by reflecting sound. A single hard surface, such as a floor, increases the noise level by 3 dBA. Two hard surfaces, such as a floor and wall, reflect an additional 6 dBA and three hard surfaces (a corner) add 9 dBA. Theoretically, if the noise was enclosed in a completely sealed room with hard surfaces, noise levels would approach infinity—although this is highly unlikely with atmospheric friction. However, the possibility exists that a loud valve installed in a small metal building could easily achieve the pain threshold of 140 dBA.

Sound pressure levels are measured by a sound-level meter, which is normally held 1 m downstream from the valve's outlet and 1 m away from the pipe itself. Because of the effect reflective surfaces can have on the sound pressure levels, the measurement must be taken in a free-field area without any reflective surfaces. In some cases, sound intensity levels may be preferred for measuring or comparing sound intensities. This is calculated as

$$\text{sound intensity level} = 10 \log_{10} \frac{\dfrac{P_S^{\,2}}{\rho C}}{10^{-16}} \text{ dB}$$

where P_S = amplitude of sound pressure
ρ = density
C = sonic velocity

In some cases, two noise sources may be occurring at the same time, which will increase the overall sound pressure level. The energy of the two sources is logarithmically combined as one noise source. Table 9.5 represents a simple method of determining the increase in noise when two noise sources are combined. After sound pressure levels are taken at each source, the difference between the two readings is used to find the correct dB factor, which is then added to the loudest noise source. As Table 9.5 shows, as the difference in the sound pressure level between two sources widens, the overall noise increase lessens. Therefore, the obvious solution is to concentrate on correcting the source with the loudest noise.

9.7.3 Turbulence

To achieve an understanding of how to decrease valve noise, the causes of turbulence must be examined. As the flow moves through the valve, the flow stream is interrupted by the valve geometry, such as the presence of a seat, disk, plug, or a sharp contour of the body. Turbulence causes pressure fluctuations in a variety of ways; however, in simple terms the pressures work against the wall of the downstream piping and cause wall fluctuations, which radiates the noise frequencies to the atmosphere. Figure 9.27 shows the pressure profile of a throttling valve as the upstream pressure is released to atmosphere. The profile shows a wide range of fluctuations in the downstream pressure that can vary by more than 15 psi (1.0 bar). As the upstream pressure decreases, the pressure drop decreases, and the variations of downstream pressure and resultant noise are less. Using the same test data, Fig. 9.28 shows a downstream test plot of the sound pressure

Table 9.5 dB Factors for Two Noise Sources†

Difference in dB between two sources	dB Factor*
0	3.01
1	2.54
2	2.12
3	1.76
4	1.46
5	1.20
6	0.97
7	0.79
8	0.64
9	0.52
10	0.42
11	0.33
12	0.27
13	0.22
14	0.17
15	0.14
16	0.11
17	0.09
18	0.07
19	0.06
20	0.05

†*Data courtesy of Fisher Controls International, Inc.*

*Added to loudest source to provide overall sound pressure level.

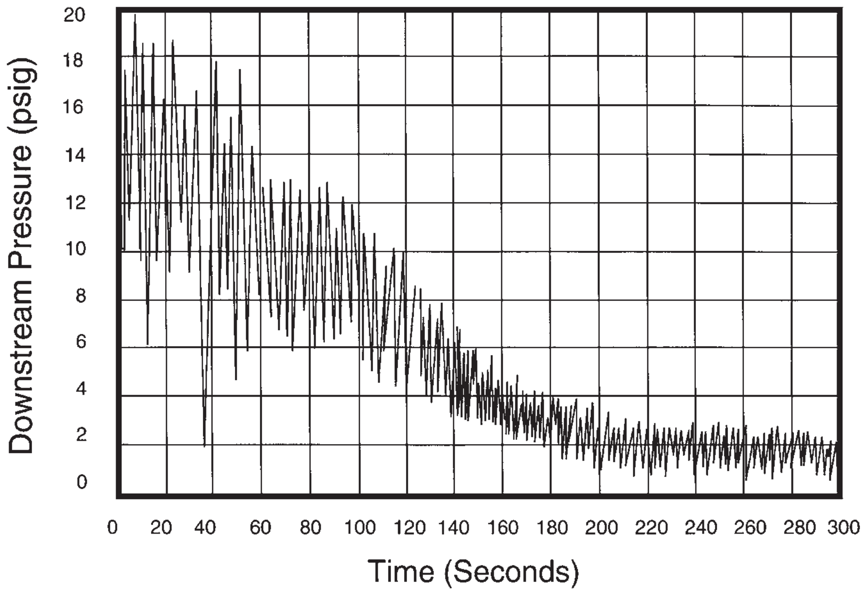


Figure 9.27 Pressure vs. time profile—downstream from valve. (*Courtesy of Valtek International*)

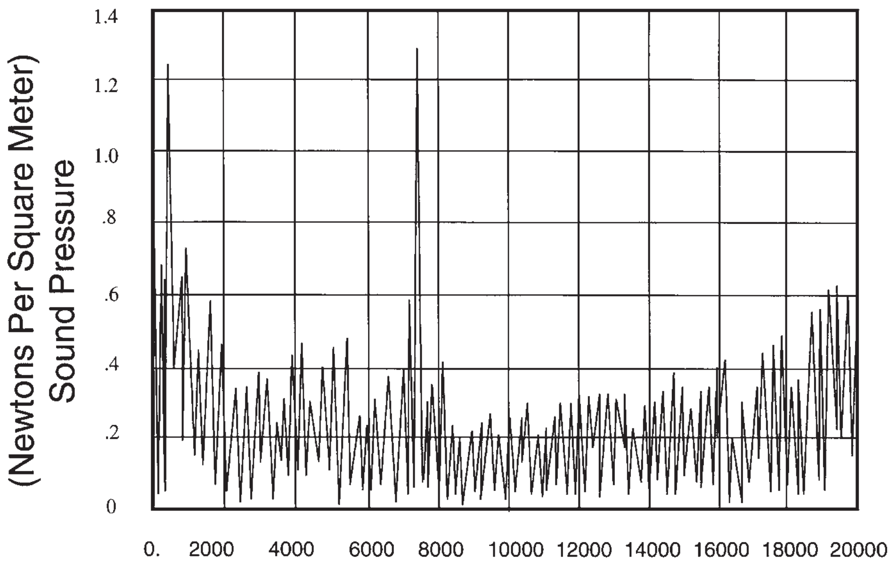


Figure 9.28 Plot of sound pressure level—downstream from valve. (*Courtesy of Valtek International*)

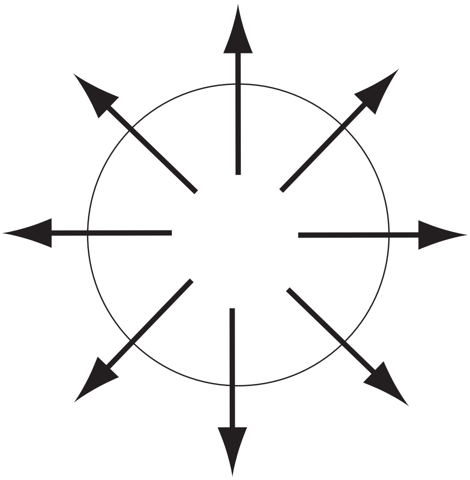


Figure 9.29 Monopole noise source. (*Courtesy of Valtek International*)

level (in hertz). The plot shows one discrete frequency peak occurring at 7500 Hz. Such peaks in that range are commonplace with valves that experience prolonged high noise levels. Although the test data indicate the presence of a wide range of subharmonics, the discrete peak frequency is principally responsible for the valve noise.

Turbulence is designated as one of three classifications: monopole, dipole, and quadrupole. *Monopole turbulence* is often described as an expanding and contracting source of noise (Fig. 9.29). The energy generated by a monopole-turbulent source is directly proportional to the flowing energy of the process fluid times the Mach number of the fluid, or in equation form:

$$\text{(turbulent energy)}\ \alpha\ \text{(flowing energy)} \times \text{(Mach number)}$$

The formula of monopole turbulence indicates that the greater speed of the flow stream will convert to more turbulent energy. Monopole energy can be easily illustrated by using a Hartmann generator (Fig. 9.30). Air flows through the nozzle (d_0) into the bore (d), causing shock waves to form inside the bore and attach to the flat surface at the bottom of the bore. As these shock waves resonate back and forth, they create discrete peak frequencies, resulting in noise that can increase by as much as 24 dBA. The importance of the Hartmann generator can be

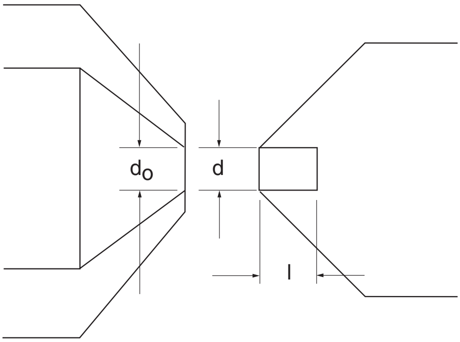


Figure 9.30 Hartmann generator (monopole noise source). (*Courtesy of Valtek International*)

seen, for example, if one envisions the shapes inside a globe valve that is made from barstock. The fluid follows through a small opening (seat ring) to a flat surface in the cavity leading to the outlet port (the bottom of the valve body). In an open situation, flow moves past the seat ring into the flat bottom portion of the body, where shock waves can attach and resonate. The position of the plug plays a large role in how much flow and velocity occur as well as the resulting noise (Fig. 9.31). However, studies have shown that if the seat-ring design is modified to a very narrow surface on the inside diameter, the shock waves

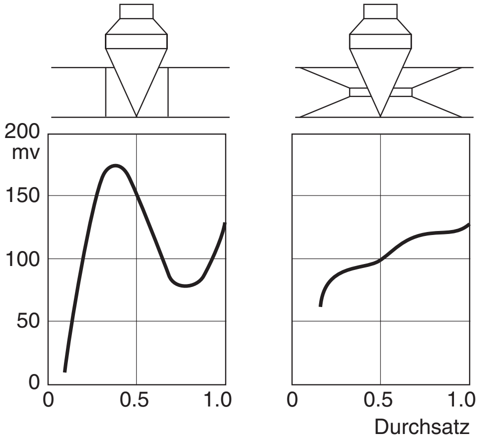


Figure 9.31 Effect of monopole noise with conventional globe valve's closure element. (*Courtesy of Valtek International*)

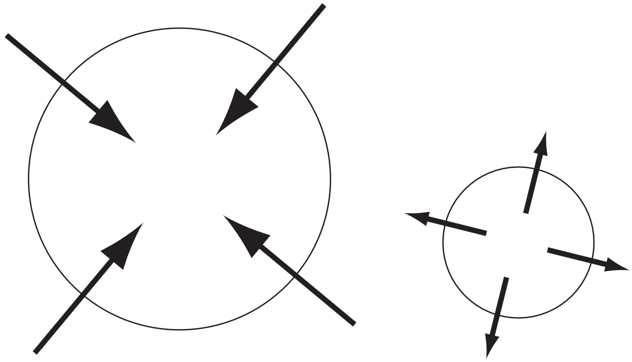


Figure 9.32 Dipole noise source. (*Courtesy of Valtek International*)

seemed to attach themselves to the sharp point and do not resonate in the body cavity.

Dipole turbulence is defined by two energy sources, one contracting in size as the other expands inversely (Fig. 9.32). With dipole turbulence, the energy of the turbulence is proportional to the Mach number cubed or in equation form:

$$(\text{turbulent energy})\ \alpha\ (\text{flowing energy}) \times (\text{Mach number})^3$$

Because of the cubed Mach number, higher velocities are much more critical in dipole turbulence than monopole turbulence. A common example of dipole turbulence is the "singing" telephone line (Fig. 9.33), in which alternate vortices are generated from both the top and bottom of the wire. These alternate vortices produce a discrete frequency, which can vary in pitch as the velocity changes. Dipole turbu-

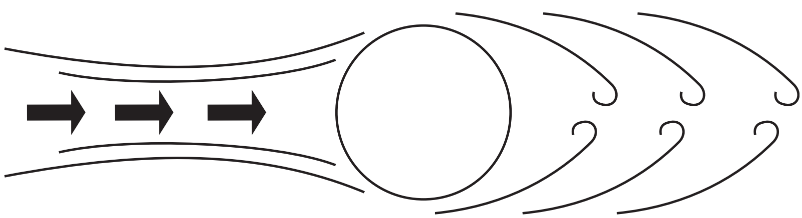


Figure 9.33 Karmen vortex street (dipole noise source). (*Courtesy of Valtek International*)

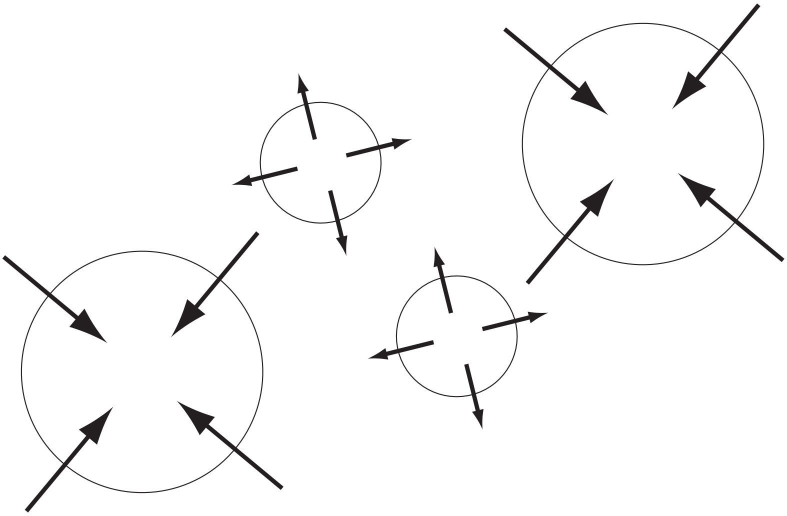


Figure 9.34 Quadrupole noise source. (*Courtesy of Valtek International*)

lence can be created in valves, such as with the sharp edges of a butterfly valve body and the disk. In addition, some trim-hole designs can generate dipole noise—a good example is the large flow characteristic holes designed into cage-guided trim.

Quadrupole turbulence is related to dipole noise; however, it involves two pairs of dipole turbulent energy. Although each pair is in phase (contracting and expanding inversely) with each turbulent source, the two pairs are out of phase with each other (Fig. 9.34). In this case, the turbulent energy varies according to the Mach number to the fifth power, or in equation form:

$$(\text{turbulent energy}) \ \alpha \ (\text{flowing energy}) \times (\text{Mach number})^5$$

Even more than in dipole turbulence, velocity is critical to the formation of quadrupole turbulence. One important difference with quadrupole turbulence is that it involves a number of random peak frequencies rather than one discrete frequency. Nearly all noise radiating from a downstream pipe is related to quadrupole turbulence. As shown in Fig. 9.25, as turbulence generated by a valve travels downstream inside the pipe, the turbulence has a tendency to move to the outer wall while smoother portions of the flow stay in the center of the pipe.

9.7.4 Noise Regulations

A growing number of organizations monitor the amount of noise workers can be safely exposed to. For example, in the United States, the Occupational Safety and Health Act (OSHA) and the Environmental Protection Agency (EPA) both regulate noise as it affects workers and the surrounding community. Initially, the Occupational Safety and Health Act (1970) stipulated that workers could be exposed to no more than 90 dBA for an 8-hour work day. Later, the Walsh Healy Public Contracts Act was enacted to further protect workers. It regulates the exact amount of time workers may work around noise. According to this legislation, the higher the dBA level, the less time workers can spend in that area, as outlined in Table 9.6.

Table 9.6 Permissible Noise Levels*
Walsh Healy Public Contracts Act

Duration Per Day *(hours)*	dBA
0.25 or less	115
0.5	110
1.0	100
2.0	97
4.0	95
6.0	92
8.0	90

**Data courtesy of Valtek International.*

9.7.5 Hydrodynamic Noise Prediction

Similar in some aspects to calculating the advent of cavitation and flashing, the prediction of noise levels in liquid services is based upon a number of common factors, including the pressure drop and flow capacity. In addition, the factors associated with pipe attenuation and distance from hearers are also considered. Using these factors, the following empirical equation can be used to predict hydrodynamic noise:

$$\text{dBA} = DP_S + C_S + R_S + K_S + D_S$$

where dBA = sound pressure level
DP_S = pressure-drop factor
C_S = flow capacity factor
R_S = ratio factor
K_S = pipe attenuation factor
D_S = distance factor

To calculate R_S and DP_S the pressure-drop ratio (DP_F) must be determined, which involves the following equation:

$$DP_F = \frac{\Delta P}{P_1 - P_v}$$

where DP_F = pressure-drop ratio
ΔP = pressure drop
P_1 = upstream pressure
P_v = vapor pressure

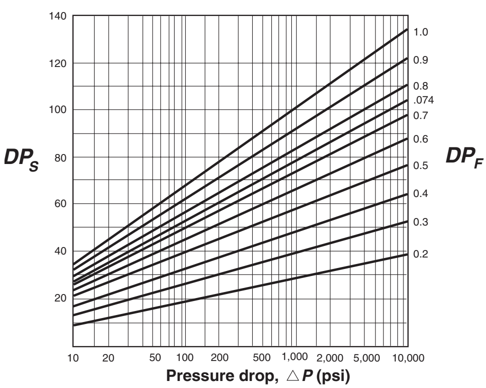


Figure 9.35 Pressure-drop factor. (*Courtesy of Valtek International*)

If DP_F is calculated to be 1 or greater, a flashing situation is occurring in the valve. Because flashing is indicative of a system problem, no modification to the valve will abate flashing and the resultant noise.

Once the pressure-drop ratio DP_F is determined, the pressure-drop factor DP_S can be determined using Fig. 9.35 and the ratio factor R_S can then be found using Fig. 9.36. Figure 9.37 provides a typical representation of the flow-capacity factor C_S. Table 9.7 provides typical distance factors D_S. Pipe attenuation factors K_S are found in Table 9.8.

9.7.6 Hydrodynamic Noise Example

The following service conditions apply for this example:

Fluid	Water
Upstream pressure	300 psig
Downstream pressure	90 psig
Vapor pressure	29.89 psia
Required C_v	34.8
Pipe size	2 in
Pipe schedule	Schedule 40
Distance of hearer	3 ft

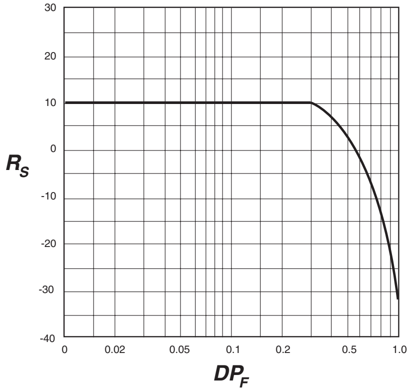


Figure 9.36 Ratio factor. (*Courtesy of Valtek International*)

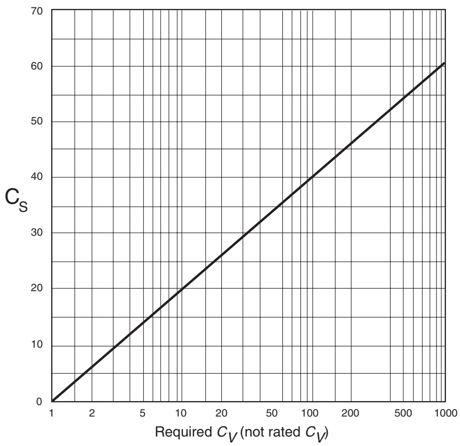


Figure 9.37 Flow-capacity factor. (*Courtesy of Valtek International*)

Table 9.7 Distance Factors*

Distance of hearer from noise source *(feet/meters)*	D_s
3 feet 0.9 meters	0 dBA
6 feet 1.8 meters	-5 dBA
12 feet 3.6 meters	-10 dBA
24 feet 7.2 meters	-15 dBA
48 feet 14.4 meters	-20 dBA
96 feet 28.8 meters	-25 dBA

**Data courtesy of Valtek International.*

Note: Factors are affected by type of noise source, as well as any reflecting surfaces close to the valve.

Table 9.8 Pipe Attenuation Factors for Liquids*

	Pipe Schedule												
	10	20	30	40	60	80	100	120	140	160	STD.	XS	XXS
0.5				0		-5				-11	0	-5	-15
0.75				0		-5				-11	0	-5	-15
1.0				0		-6				-12	0	-6	-15
1.5				0		-6				-12	0	-6	-14
2				0		-6				-12	0	-6	-14
3				0		-7				-13	0	-7	-16
4				0		-7		-9		-13	0	-7	-14
6				0		-8		-10		-14	0	-8	
8		4	3	0	-3	-9	-8	-12	-13	-18	0	-9	
10		5	3	0	-5	-9	-9	-13	-14	-19	0	-7	
12		6	2	-1	-6	-10	-11	-14	-15	-20	0	-6	
14	6	3	0	-2	-6	-11	-12	-15	-16	-22	0	-4	
16	6	3	0	-4	-8	-12	-13	-16	-18	-24	0	-4	
18	5	3	-2	-6	-9	-13	-15	-18	-19	-25	0	-4	
20	5	0	-4	-6	-10	-14	-16	-19	-21	-26	0	-4	
24	5	0	-6	-8	-12	-15	-19	-21	-23	-27	0	-4	
30	3	-4	-7	-8		-15				-27	0	-4	
36	3	-4	-7	-9		-15				-27	0	-4	
42		-4	-7			-15							

**Courtesy of Valtek International.*

By using the pressure-drop ratio equation, DP_F is calculated as 0.74:

$$DP_F = \frac{DP}{P_1 - P_V} = \frac{314.7 - 104.7}{314.7 - 29.89} = 0.74$$

From Figs. 9.35 to 9.37 and Tables 9.7 and 9.8, the following factors apply: $DP_S = 60$, $R_S = -10$, $C_S = 31$, $D_S = 0$, and $K_S = 0$. Therefore, the hydrodynamic noise equation can be used to predict the noise from this application:

$$\text{dBA} = DP_S + R_S + C_S + D_S + K_S = 60 + (-10) + 31 + 0 + 0 = 81 \text{ dBA}$$

With a predicted sound pressure level at 81 dB, hearers could safely work in the vicinity of the valve for 8 h per day (as outlined by the Walsh Healy Act).

9.7.7 Aerodynamic Noise Prediction

Because aerodynamic noise is the most irritating type of noise to nearby hearers and communities, predicting the noise level emitted from a valve is critical to the sizing and selection process. The noise prediction for gas services varies from the hydrodynamic noise equation in that factors relating to pressure, temperature, and gas properties must also be considered. The following empirical equation is used:

$$\text{dBA} = V_S + P_S + E_S + T_S + G_S + A_S + D_S$$

where V_S = flow factor
P_S = pressure factor
E_S = pressure ratio factor
T_S = temperature correction factor
G_S = gas property factor
A_S = attenuation factor

The flow factor V_S is determined by using the valve's required C_v, as shown in Fig. 9.38. The pressure factor P_S is found by using the valve's upstream pressure (Fig. 9.39). To determine the pressure ratio factor E_S, the ratio between the upstream and downstream pressures must be calculated (Fig. 9.40). The temperature correction factor T_S is determined by Table 9.9. The gas property factor G_S is found by applying the molecular weight of the gas against Fig. 9.41. The attenuation factor A_S is found for a given pipe size and schedule in Table 9.10. The same distance factor table (Table 9.7) that was used in the hydrodynamic calculations still applies for D_S.

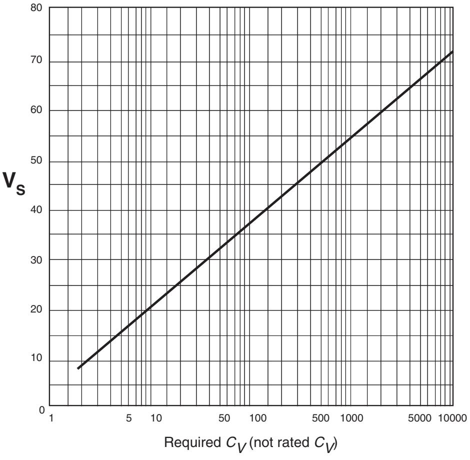


Figure 9.38 Flow factor. (*Courtesy of Valtek International*)

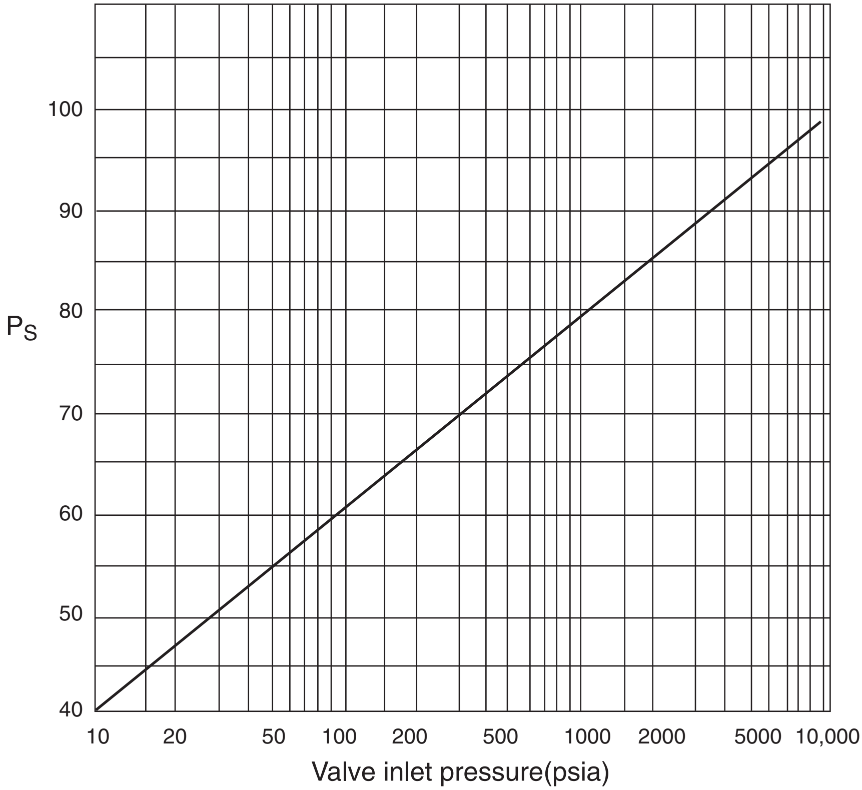


Figure 9.39 Pressure factor. (*Courtesy of Valtek International*)

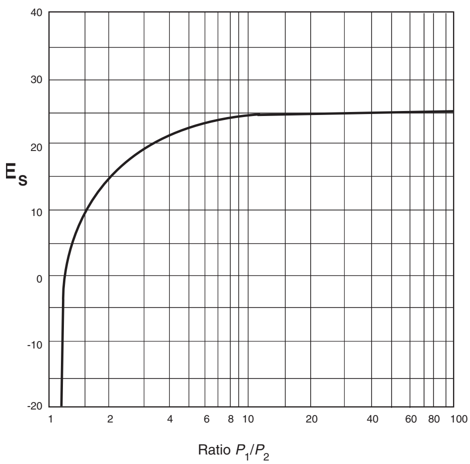


Figure 9.40 Pressure ratio factor. (*Courtesy of Valtek International*)

Table 9.9 Temperature Correction Factors*

Flowing Temperature of Gas	T_s
70° F / 21° C	0.0
100° F / 38° C	0.0
200° F / 93° C	-1.0
300° F / 150° C	-1.5
500° F / 260° C	-2.0
700° F / 370° C	-3.0
1000° F / 540 ° C	-3.5
1200° F / 490° C	-4.0

**Courtesy of Valtek International.*

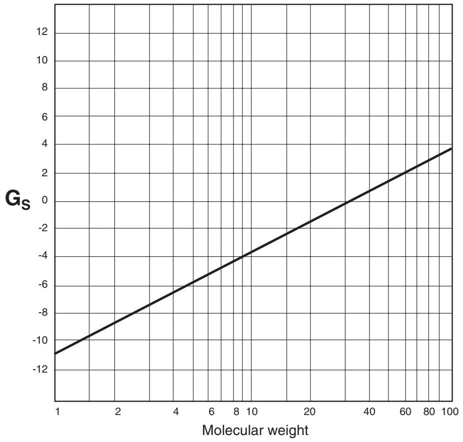


Figure 9.41 Gas property factor. (*Courtesy of Valtek International*)

Table 9.10 Pipe Attenuation Factors for Gases*

	Pipe Schedule												
Pipe Size	10	20	30	40	60	80	100	120	140	160	Std.	XS	XXS
1/2				-8.0		-11.5				-13.1	-8.0	-11.5	-18.5
3/4				-10.0		-13.7				-17.0	-10.0	-13.7	-21.0
1				-11.4		-15.0				-18.8	-11.4	-15.0	-22.2
1 1/2				-13.6		-17.6				-21.6	-13.6	-17.6	-25.5
2				-14.8		-19.2				-24.6	-14.8	-19.2	-27.4
3				-16.6		-20.8				-25.4	-16.8	-20.8	-29.1
4				-18.0		-22.4		-25.7		-28.0	-18.0	-22.4	-30.9
6				-20.0		-25.5		-28.8		-31.8	-20.0	-25.5	-34.1
8		-18.1	-19.4	-21.4	-24.3	-27.0	-29.1	-31.5	-33.1	-34.4	-21.4	-27.0	-34.0
10		-17.6	-20.3	-22.5	-26.6	-28.7	-31.2	-33.2	-35.3	-36.8	-22.5	-26.6	-35.3
12		-18.2	-21.8	-24.5	-28.5	-31.2	-33.8	-36.0	-37.4	-39.3	-24.5	-28.7	-36.0
14	-18.8	-21.6	-24.0	-26.0	-29.9	-32.9	-35.5	-37.7	-39.3	-40.8	-24.0	-28.5	
16	-19.5	-22.4	-24.8	-28.5	-32.0	-35.2	-37.7	-39.8	-41.9	-43.2	-24.8	-28.5	
18	-20.2	-23.1	-27.4	-30.7	-35.4	-37.2	-39.9	-42.1	-43.7	-45.3	-25.4	-29.0	
20	-20.8	-26.1	-29.8	-32.0	-36.0	-39.1	-41.8	-43.8	-45.7	-47.2	-26.1	-29.8	
24	-21.9	-27.1	-32.3	-34.9	-39.3	-42.3	-45.2	-47.3	-48.9	-50.5	-27.1	-29.5	
30	-26.1	-32.2	-35.1	-38.5	-42.7	-45.5	-48.3	-50.5			-26.0	-32.2	
36	-27.2	-33.3	-36.2	-42.0	-45.5	-48.5	-51.2				-26.4	-33.3	
42	-28.7	-37.0	-40.3	-44.5	-48.0	-50.7	-53.7				-26.7	-30.4	
48	-29.8	-39.0	-42.5	-46.5	-50.3	-53.0					-27.0	-30.5	
54	-30.5	-41.0	-44.3	-48.5	-52.2								
60	-31.2	-42.5	-45.7	-50.3	-53.5								

**Courtesy of Valtek International.*

9.7.8 Aerodynamic Noise Example

The following service conditions apply to this example:

Fluid	Steam
Upstream pressure	139.7 psig
Downstream pressure	29.7 psig
Required C_v	46.2
Pipe size	2 in
Pipe schedule	Schedule 40
Distance of hearer	3 ft
Molecular weight	18.02

Using the upstream and downstream pressures, the ratio P_1/P_2 is:

$$\frac{P_1}{P_2} = \frac{139.7}{29.7} = 4.70$$

From Figs. 9.38 to 9.41, and Tables 9.7 and 9.8, the following factors are applied: $V_S = 31$, $P_S = 61$, $E_S = 22.5$, $T_S = -2$, $G_S = -1.0$, $D_S = 0$, and $A_S = -18.0$. With these factors, the aerodynamic noise equation can be used to predict the noise from this application:

$$\begin{aligned} \text{dBA} &= V_S + P_S + E_S + T_S + G_S + A_S + D_S \\ &= 31 + 61 + 22.5 + (-2) + (-1.0) + (-18.0) + 0 \\ &= 93.5 \text{ dBA} \end{aligned}$$

According to the Walsh Healy Act, at 93.5 dB, hearers could remain in the vicinity of the valve for 4 h per day.

9.8 Noise Attenuation

9.8.1 Introduction to Attenuation

Because hydrodynamic noise is often associated with cavitating services, it can be controlled with anticavitation measures. Generally, however, noise is associated with gas applications. This section emphasizes methods to lower noise levels in gaseous applications, although some methods may be applicable to liquid applications also. The process of lowering noise or sound pressure levels is called *attenuation*. Noise pollution is a primary environmental concern, for both plant and community environments. In many cases, the sound pressure levels must be reduced by noise attenuation of the source itself (the valve) or the path (downstream piping). Correcting the offending source is the ideal situation, but this involves sophisticated attenuation devices that reduce sound pressure levels to comfortable levels. Unfortunately, the costs associated with these special attenuation devices are high. Depending on the size of the valve, the cost could increase anywhere from 40 to 200 percent. If material fatigue or diminished performance is not a concern, path attenuation may be a less expensive, easier option, although it is only treating the symptom rather than the root cause.

Valve manufacturers, especially those that offer sizing and selection software programs, routinely predict noise as part of the valve selection process. However, the user should be aware that these predicted sound pressure levels assume that the valve is installed in a completely nonreflective environment and do not consider the additional noise levels associated with walls, floors, and ceilings. For example, a valve installed in a natural-gas pressure-reduction application is predicted to produce 85 dBA, which is within the safety standards of most regulations. However, because the valve is installed in a metal building, which is highly sound-reflective, the sound pressure level rises to 115 dBA. Therefore, the location of the valve should always be considered before determining that noise-attenuation devices or preventative measures are not necessary.

As this section outlines, a great deal of options are available to either reduce or eliminate noise. Some are more expensive than others, while some present additional problems, such as increased maintenance or added potential leak paths. Because of the costs and safety factors involved, the user should examine all options before deciding on installing an expensive antinoise valve.

9.8.2 Valve Attenuation Options

Although many users consider expensive valve trims the only solution to valve noise, a number of less expensive options exist that should be explored prior to specifying a specially engineered valve. The most simple, but overlooked option would be to restrict the access of workers to a high noise level or to provide ear protection while in that area. If equipment damage is not an issue, the main benefit of reducing noise levels is to protect the hearing of nearby workers. If workers do not need access to the affected area, then safety warnings and requirements for ear protection can be mandated and the process left alone.

Changes to the process may also be an option. The velocity may be varied by slightly changing the upstream or downstream pressures. In many cases, a discrete signal, which is within the range of hearers, is often prevented by a slight pressure variation to either side of the valve. The valve's position can also be slightly increased or decreased, allowing a minor change in flow that may disrupt the retention of shock waves on a given surface.

An interesting aspect of noise is that some linear valve styles, such as a globe valve, produce a discrete signal at 30 percent lift, despite the valve size or length of stroke. One way to deal with this phenomenon is to use a special diverting seat ring that has a special lip built into the bottom of the seat ring and breaks up the formation of shock waves.

As discussed in Sec. 9.7, velocities are directly related to turbulence and noise and can be controlled through right-angle turns. Frictional losses associated with 60 ft (18 m) of straight pipe are equal to the frictional forces produced by one 90° elbow, which will slow the velocity. Designing the system with several elbows can produce attenuation. In addition, placing two or more globe valves in series will produce a staged pressure drop and also add two or more right-angle turns per valve.

When a gas process is vented to atmosphere, high noise levels above 100 dBA can be generated. Such noise can be channeled in the opposite direction using shields or shrouds, which may lower the sound pressure level within acceptable limits. Moving the vent to a distant location may be an option in some cases, although the additional piping may be cost prohibitive.

The valve style can have some bearing on the type of noise that is generated. As explained in the preceding section, rotary valves are more apt to produce sharp dipole vortices as the flow travels past the sharp edges of the body. In addition, valve bodies produced from barstock commonly cause monopole noise as the flow moves through the seat and attaches to the flat surface of the outlet port. On the other hand, the conventional casting design of a globe body would avoid sharp edges associated with rotary valves and the flat surfaces of barstock bodies. The user should remember that different valve styles and internal geometries react differently to the same process. As a last resort, trial and error may be required to discover the one valve style that is able to handle the service without producing turbulence and subsequent pressure fluctuations that lead to noise.

When the flow direction is not a critical element of the application, the valve can be installed backwards (inlet port is installed downstream, and the downstream port is installed upstream), so that the flow direction is opposite the normal operation. (For example, a flow-over-the-plug linear valve will become a flow-under-the-plug valve.) When this is done, the fluid will then flow through a different valve geometry, in which monopole, dipole, or quadrupole noise is less likely to be created. For example, changing from flow-over-the-plug to flow-under-the-plug may avoid monopole noise that would be created from the Hartmann generator effect (Sec. 9.7.2). (The process stream flows up through the seat into the upper gallery, where the geometry provides no flat surface perpendicular to the flow where shock waves can form.)

In some cases, modifications can be made to the existing valve trim to attenuate the noise without installing expensive trims or downstream attenuation equipment. As discussed earlier, monopole noise will attach itself to a very narrow landing on the seating surface. Reducing this landing through machining may be possible, as long as the seat's seating surface and overall strength is not affected.

The location of the valve is vitally critical to the amount of noise generated by turbulence. Often noise is generated by turbulence in the valve and is then carried to downstream piping. The noise radiates the pressure fluctuations through the downstream pipe wall to the environment as sound waves. This phenomenon occurs with long, straight sections of thin-walled piping that are more apt to flex. Conversely, piping elbows and other nonlinear piping configurations are stiffer and are not apt to allow wall fluctuations. If a valve is included in a long stretch of piping, the preferred arrangement would be a long length of pipe on the upstream side of the valve and on the downstream side an elbow or a shorter length of pipe. The longer the pipe, the more sound radiation is possible. Piping supports can also be used to stiffen long lengths of piping, preventing the flex of the pipe wall.

In some process services in which the valve discharges fluid into a large vessel, the valve can be located next to the vessel without a long expanse of pipe. This will allow the valve to discharge the fluid into the vessel and the noise to be absorbed in a larger area.

If the valve and downstream piping are located in a room or protective shed with a number of close-by hard reflective surfaces, the sound pressure levels may increase significantly, upwards of 30 to 40 dB. However, by moving the location of the valve to the wall, the downstream side of the pipe can be placed outside of the room. Not only will the noise be eliminated from the room, but the noise radiated to the environment outside of the room will also be less.

Another option is to specify a thicker wall schedule in the downstream piping, which provides greater stiffness. For example, using a schedule 80 pipe instead of a schedule 40 pipe will lower the sound pressure level by approximately 4 dB. Table 9.11 provides a correction factor for noise attenuation for piping that has a heavier wall schedule (assuming schedule 40 pipe wall thickness is standard.)

One of the more common methods of dealing with high sound pressure levels is to absorb the noise with thermal or acoustic insulation, which can be wrapped around the valve or downstream piping. This is the best solution only when high sound pressure levels offer no threat of fatigue to materials or substandard performance of instrumentation. Generally, 1 in (2.5 cm) of normal thermal insulation will provide a reduction in sound pressure level of between 3 and 5 dB. Acoustic insulation is manufactured to absorb more sound energy and can provide a reduction of 8 to 10 dBA per inch of insulation. Depending on the *R* value of the insulation, a 3-in insulation will provide the maximum attenuation anywhere from 15 to 24 dB. (Additional insulation does not attenuate the noise any further.) Table 9.12 outlines typical insulation factors.

One caution should be noted, however. As previously explained, noise levels close to a valve and its immediate downstream piping may be reduced with an elbow, thick schedule pipe, or insulation.

Table 9.11 Pipe-Wall Attenuation*

Pipe Size	Schedule 40	Schedule 80	Schedule 120	Schedule 160
2-inch (DN 50)	0 dBA	- 6 dBA	-8 dBA	-12 dBA
3-inch (DN 75)	0 dBA	-7 dBA	-9 dBA	-13 dBA
4-inch (DN 100)	0 dBA	-7 dBA	-10 dBA	-13 dBA
6-inch (DN 150)	0 dBA	-8 dBA	-12 dBA	-15 dBA
8-inch (DN 200)	0 dBA	-9 dBA	-14 dBA	-18 dBA
10-inch (DN 250)	0 dBA	-10 dBA	-14 dBA	-19 dBA
12-inch (DN 300)	0 dBA	-11 dBA	-16 dBA	-20 dBA

**Data courtesy of Fisher Controls International, Inc.*

However, these methods only protect the hearer in the immediate vicinity of the valve. Since these methods do not attenuate the source of the noise, sound will continue in the downstream piping and may surface at an unprotected point further downstream (Fig. 9.42). At that point, either the noise must be tolerated or additional corrective action must be taken.

9.8.3 Downstream Antinoise Equipment

In some applications, adding an antinoise element immediately downstream from the valve may be effective in attenuating the noise to reasonable levels. In addition, these elements can absorb energy or straighten turbulent flow so that noise is not carried downstream. The cost associated with these supplemental devices is less than or equal to special valve trim. Access to a downstream element is much easier for

Table 9.12 Insulation Factors*

Depth of Insulation	dBA Reduction
1 in / 2.5 cm	-5 dBA
2 in / 5.1 cm	-10 dBA
3 in / 7.6 cm	-15 dBA

Data courtesy of Valtek International.

maintenance purposes than gaining access to special trim. Common antinoise elements include attenuator plates, diffusers, silencers, and external stacks. Because these devices all utilize small holes or flow paths, they are susceptible to plugging if the process fluid contains particulate matter, which may require additional maintenance.

Placed downstream in series with the valve, the *attenuator plate* is a downstream antinoise element (Fig. 9.43) that provides anywhere from single to multiple stages of pressure reduction (Fig. 9.44). Attenuator plates typically reduce the overall sound pressure level by up to 15 dB. Using a pattern of holes, each stage of the attenuator plate has its own

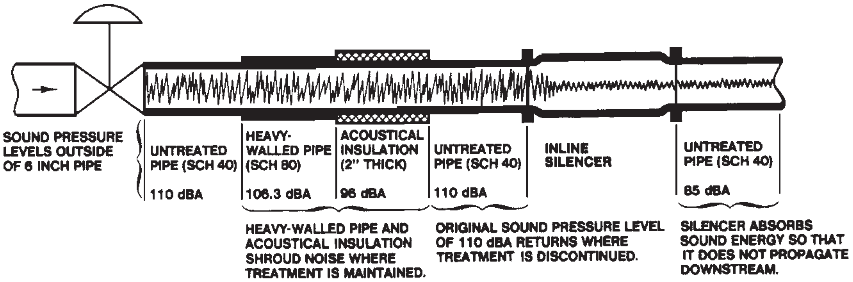


Figure 9.42 Multiple methods of path treatment of noise. (*Courtesy of Fisher Controls International, Inc.*)

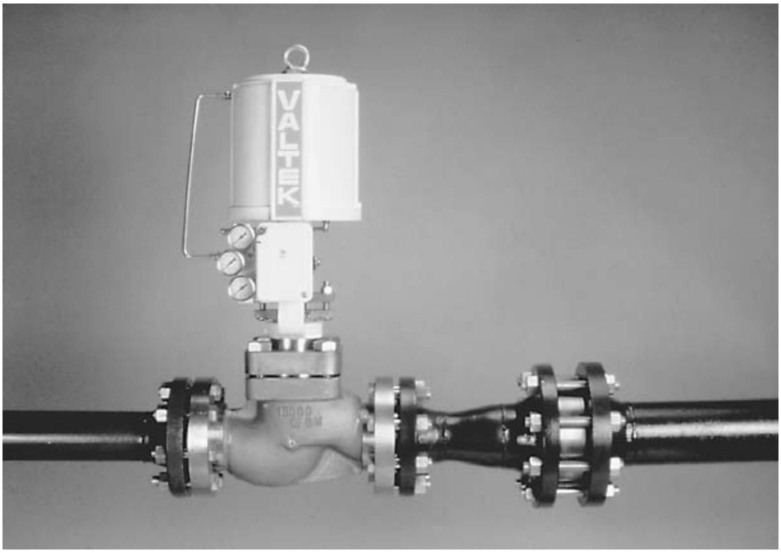


Figure 9.43 Attenuator plate mounted downstream from a globe control valve. (*Courtesy of Valtek International*)

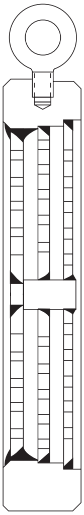


Figure 9.44 Three-stage attenuator plate. (*Courtesy of Valtek International*)

Common Valve Problems

individual flow capacity. As Fig. 9.44 illustrates, each succeeding stage has a larger flow area, which provides the staged pressure reduction and maintains velocities at lower levels. The multiple holes act as a straightening device for the turbulent fluid, providing a series of controlled, smaller fluid streams instead of a large turbulent eddy. Although these smaller fluid streams still have some turbulence, they are more easily dissipated throughout the overall process stream because of their size. Since the area of the plate is limited to the inside diameter of the pipe, as well as the hole pattern, only so much flow can pass through the first stage. The maximum flow capacity through the attenuator plate is achieved with a pressure ratio of 4.5 to 1 (or less). High rangeability is highly unlikely with attenuator plates; therefore, they should be installed only in moderate to low rangeability applications. Because the flow capacity is limited, attenuator plates should be considered only for those applications that can handle such a reduction in flow. In some applications where additional flow is needed, a larger plate can be specified with more flow area, but pipe expanders or reducers must be used to allow the installation of the larger plate in a smaller pipeline. Not only does this raise costs, but it also adds a number of line penetrations that could leak.

For applications that require greater flow than offered by an attenuator plate, a *diffuser* is often specified, which also offers reductions of up to 15 dB. As shown in Fig. 9.45, a diffuser is a long cylinder tube with a closed end that can vary in length according to the flow needed. As with attenuator plates, the diffuser is installed downstream in series with the valve. The diffuser is designed to fit inside the pipeline, allowing for a specific clearance between the inside diameter of the pipe and outside diameter of the diffuser. The diffuser is held in place between the raised face flanges of the valve and pipeline, or it can be welded in place. A diffuser can also be directly bolted or welded to the valve and be used to vent to atmosphere. When venting to atmosphere, a diffuser can be equipped with shrouds to direct the noise away from hearers. Although the diffuser shares the overall pressure drop with the valve, its flow capacity can be expanded by making the diffuser longer and adding more holes. These holes control the sound pressure level by passing the flow through the holes to absorb sound energy and minimize turbulence. The major disadvantage of a diffuser is the maintenance problems associated with the small holes, which can become plugged if the process contains oversized particulates. Because the holes in the diffuser are perpendicular to the piping, they have a tendency to impinge condensates and particulates directly on the piping wall, which may lead to erosion.

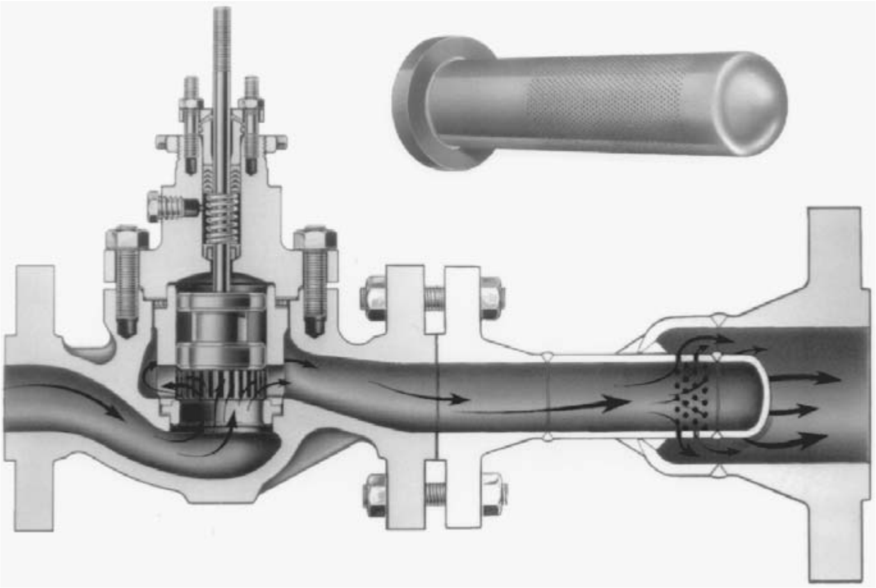


Figure 9.45 Downstream diffuser. (*Courtesy of Fisher Controls International, Inc.*)

A *silencer* is used when reductions of sound pressure levels of more than 15 dB are required, which are beyond the capability of attenuator plates or diffusers. Depending on the design and process service conditions, attenuations as high as 35 dB can be achieved with a silencer. Similar to a diffuser in that it shares the pressure drop with the valve, a silencer also lowers the sound pressure level by absorbing noise. As shown in Fig. 9.46, a common silencer incorporates a series of compartments that use tubes with holes, much like minidiffusers. Acoustic material is used throughout the silencer to absorb sound and process energy. The primary disadvantage of silencers is that they are designed to attenuate a particular frequency. Overall, silencers are good for applications with a constant flow. However, if the application is such that the flow varies routinely, the frequency will also vary and may render the silencer ineffective. While a silencer is less expensive than other antinoise options, it requires some piping modifications, including piping supports. Depending on the application, the size of the silencer can be quite large. This may become a factor where space is limited. Silencers are normally flanged and bolted to the pipeline, although they can also be used to vent to atmosphere.

Some valves use an *external stack* (also known as an *atmospheric resistor*) as a downstream element to reduce noise in venting or blowdown

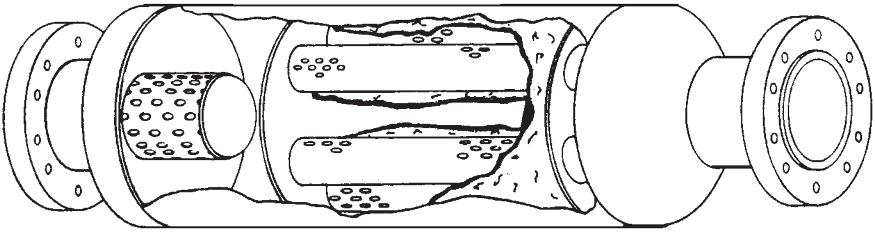


Figure 9.46 In-line silencer. (*Courtesy of Fisher Controls International, Inc.*)

applications (Fig. 9.47). Instead of installing antinoise trim inside the valve, the stack is placed immediately downstream from the valve's outlet port. This design provides several benefits. First, the physical characteristics of the stack can be larger, allowing a greater outside diameter and stack height. This allows greater flow and increased

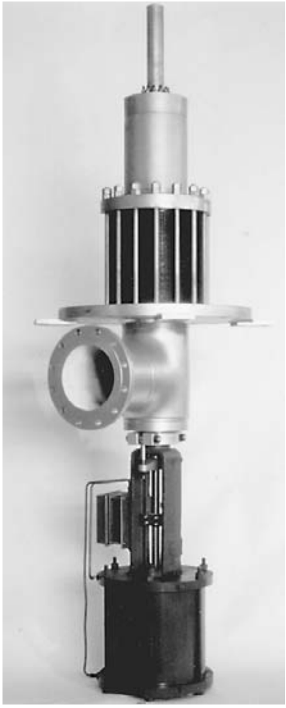


Figure 9.47 External stack mounted on outlet of angle body control valve. (*Courtesy of Control Components Inc., an IMI company*)

attenuation than a stack inside the valve—which is limited by the body's gallery height. Second, the antinoise mechanisms built into the stack—such as expanding flow areas, tortuous paths, etc.—can lower the exit velocity and share the pressure drop with the valve. This design provides greater attenuation while not affecting the overall flow rate.

9.8.4 Downstream Antinoise Equipment Sizing

Sizing for the flow capacity of downstream equipment is based on the number of stages of pressure drop taken. These stages can be taken through one element (such as an attenuator plate) or a number of single-stage elements in series (such as two diffusers). A common equation for attenuation plates follows:

$$C_v = \frac{1}{\sqrt{\left(\frac{1}{C_{v_1}}\right)^2 + \left(\frac{1}{C_{v_2}}\right)^2 + \left(\frac{1}{C_{v_3}}\right)^2 + \left(\frac{1}{C_{v_N}}\right)^2}}$$

where C_v = total flow capacity
C_{v_1} = flow coefficient of the first control element (or first stage)
C_{v_2} = flow coefficient of the second control element (or second stage)
C_{v_3} = flow coefficient of the third control element (or third stage)
C_vN = flow coefficients of any additional control elements

9.8.5 Downstream Antinoise Equipment Sound-Pressure-Level Prediction

Predicting the overall sound pressure level is determined by the following two equations:

$$\mathrm{SPL} = 22 + 12\log_{10}\left(\frac{P_1}{P_O} - 1.05\right) + 10\log_{10}(C_vF_L) + 10\log_{10}(P_1P_2)$$

$$+ 30\log_{10}\left(\frac{t_{40}}{t}\right) - G_S + T_L$$

$$\frac{P_1}{P_O} = \frac{\frac{P_1}{P_2} + Z}{Z}$$

where SPL = sound pressure level
Z = number of elements (or stages)
P_1 = inlet pressure
P_2 = outlet pressure
P_O = outlet pressure for each stage
t_{40} = schedule 40 pipe wall thickness
t = wall thickness for given wall pipe
G_S = gas property correction factor (Table 9.12)
T_L = SPL velocity-correction factor for gas discharges above Mach 0.15 ($T_L = 20 \log_{10} [1/(1.1-M)]$)
M = Mach number of outlet pipe

When two elements are combined in series, up to 3 dB should be added to the total sound pressure level to compensate for having two separate noise sources. Figure 9.48 provides this data. If two noise sources have identical sound pressure levels, the overall intensity will not be equal to that level, but will be greater than either noise source. A 6-dB insertion loss factor should be included in the overall sound pressure level to compensate for a close connection between the element and the valve. *Close connection* is defined as one pipe reducer length. The sound pressure level of venting applications can also be determined. Although the general rule is that spherical radiation of

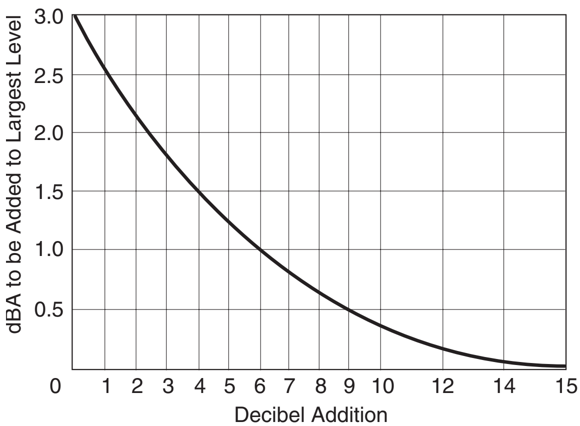


Figure 9.48 dBA addition for two elements installed in series. (*Courtesy of Valtek International*)

noise reduces the sound pressure level by 6 dB for every doubling of distance, noise emitted over long distances can absorb even more sound because of atmospheric absorption, and the attenuating effects of nearby objects and the ground.

The following equation can be used to calculate the sound pressure level for all gaseous venting applications, except steam:

$$\mathrm{SPL}_{\mathrm{intermediate}} = \mathrm{SPL}_{\mathrm{elements}} - 10\log_{10}\left(\frac{(3.2)(10^{-11})\,P_2\,D^2\,T^{0.5}\,R^2}{t^3}\right) - \frac{G_S}{2}$$

where $\mathrm{SPL}_{\mathrm{intermediate}}$ = uncorrected sound pressure level from vent
$\mathrm{SPL}_{\mathrm{elements}}$ = sound pressure level emitted from control elements
D = downstream nominal pipe diameter
P_2 = valve downstream pressure
R = distance from vent
T = absolute temperature

Although this equation is used to find the total sound pressure level emitting from the vent, the sound pressure level can also be lowered by the direction of the noise.

The following equation is used to calculate the sound pressure level for steam-venting applications:

$$\mathrm{SPL}_{\mathrm{intermediate}} = \mathrm{SPL}_{\mathrm{elements}} - 10\log_{10}\left(\frac{(3.2)(10^{-10})\,P_2\,D^2\,R^2\,(1 + 0.00126T_{\mathrm{SN}})^3}{t^3}\right)$$

where T_{SN} = superheated steam temperature

Sound pressure levels are also reduced if the noise radiates in a directional nature rather than spherical. In other words, the farther the noise is pointed away from the hearer, the less noise is heard. This phenomena is called *directivity.* This concept is illustrated in Fig. 9.49 and Table 9.13. With particular venting applications, directivity can occur if the vent is pointed away from workers or nearby communities or if a resistor shroud is used to direct the sound upward (or away from the hearer). In venting applications where directivity occurs, the reduction of the sound pressure level can be determined by the following equation:

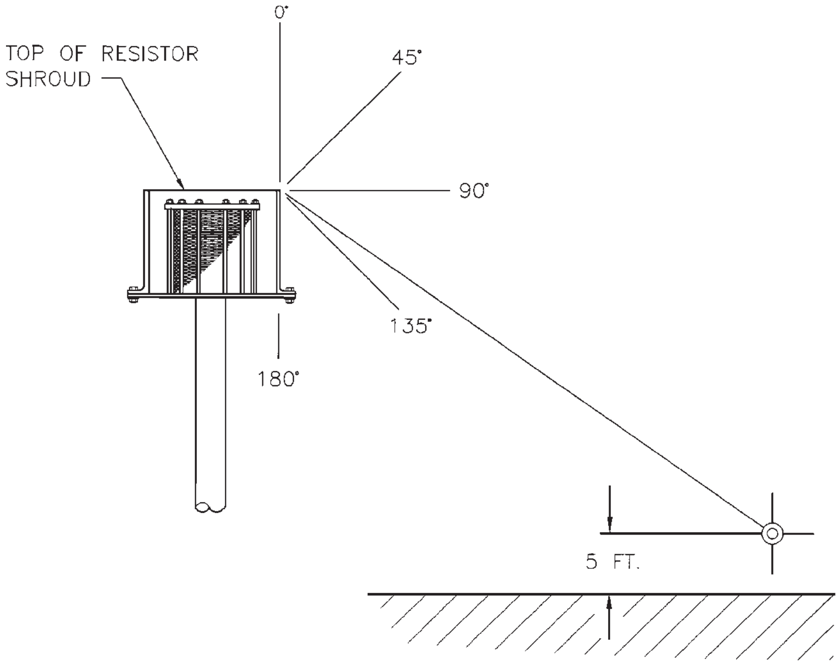


Figure 9.49 Angle location of hearer from noise source as associated with the directivity index. (*Courtesy of Control Components Inc., an IMI company*)

$$SPL_{total} = SPL_{intermediate} + DI$$

where SPL_{total} = total noise emitting from the vent
DI = directivity index (Table 9.13)

Atmospheric noise can be divided into *near field* and *far field*. The near-field noise is the noise that is generated within 3 to 10 ft (1 to 3 m) of the source, while the far-field noise is that generated beyond 10 ft from the source. In far-field situations where sound spreads in a radial, homogeneous pattern, the noise intensity is attenuated by the distance—intensity decreases inversely to the (distance)^2 from the vent. This relationship is found in the following equation:

$$I = \frac{W}{4\pi r^2}$$

Table 9.13 Typical Directivity Index*

Angle Away from Axis of Resister Shroud	dBA Addition or Subtraction
0°	0
20°	+1
40°	+8
60°	+2
80°	-4
100°	-8
120°	-11
140°	-13
160°	-15
180°	-17

**Data courtesy of Control Components, Inc.*

where I = sound intensity (W/m^2)
W = sound power (W)
r = distance from sound source

This calculation applies only to far-field situations, in which large distances are involved and can be affected by a number of different factors, including humidity, wind, presence of trees, etc.

9.8.6 Antinoise Valve Trims

In difficult gaseous applications, noise must be treated at the source rather than treating the symptom with insulation, heavier or nonlinear piping, or ear protection. This means that modifications must be made to the valve to minimize or eliminate the high sound pressure levels and resultant vibration that can fatigue metal or affect the performance of nearby instrumentation. The antinoise trim must reduce the pressure drop, so that the resultant high velocities do not approach sonic levels. The most common approach to this problem is to install special trims in globe valves. In principle, these trims channel the fluid through a series of turns, which affects the velocities and pressures involved. Each turn is typically called a *stage.* Antinoise trims can include anywhere from 1 to 40 or even 50 stages, based on the design. While anticavitation trims are designed to flow over the plug in linear valves, antinoise trims are designed to flow under the plug. This direction allows an expanding flow area in the later stages of the antinoise device, which slows the velocity to subsonic levels.

A number of different antinoise trim devices are in existence, but for the most part they can be categorized into four different styles: slotted, multihole, tortuous path, and expanding teeth. *Slotted trim* use is a single-stage cage or retainer that contains long, narrow slots around the entire diameter (Fig. 9.50). As the fluid passes through the slots, turbu-

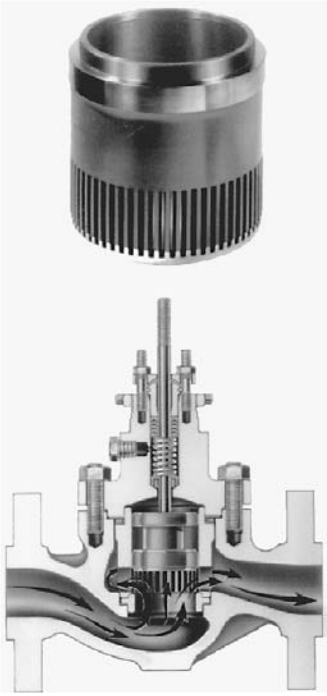


Figure 9.50 Single-stage multiple-slotted cage. (*Courtesy of Fisher Controls International, Inc.*)

lence is broken up into smaller eddies, and the velocity is distributed evenly throughout the gallery of the globe body. This design works best when the pressure drop to inlet pressure ratio ($\Delta P/P_1$) is equal to or less than 0.65 and when the maximum downstream pressure (P_2) is less than half of the fluid's sonic velocity. If the $\Delta P/P_1$ ratio is higher than 0.65, the pressure drop may be handled by adding a second device (such as a downstream element) to share the pressure drop. Slotted cages or retainers offer noise attenuation up to 15 dB and are relatively inexpensive when compared to other antinoise trims. Outlet velocity is limited to below Mach 0.5. Additional dB reduction can be handled by adding an attenuation plate or diffuser downstream to the valve, which can also be cost effective when compared to other antinoise trims.

Multihole trim utilizes a number of cylinders, also known as stages, with drilled or punched holes that control turbulence in the flow stream (Fig. 9.51). This device also has a secondary use as a seat-ring retainer, which allows a clearance between the plug and the inside diameter of the retainer (Fig. 9.52). This device can also be designed as a cage, where the plug guides on the inside diameter (Fig. 9.53). One of

Figure 9.51 Single- and multiple-stage attenuators. (*Courtesy of Valtek International*)

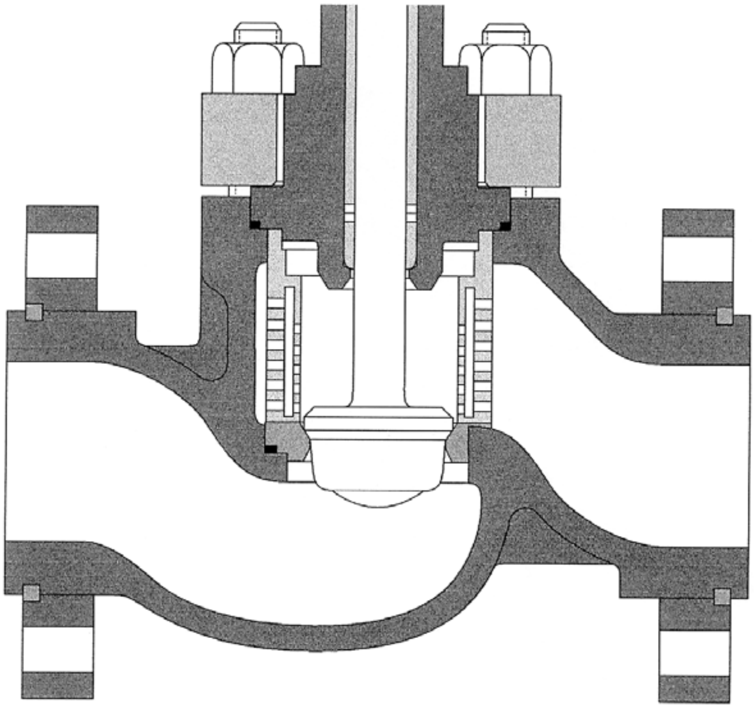


Figure 9.52 Globe valve equipped with two-stage attenuator. (*Courtesy of Valtek International.*)

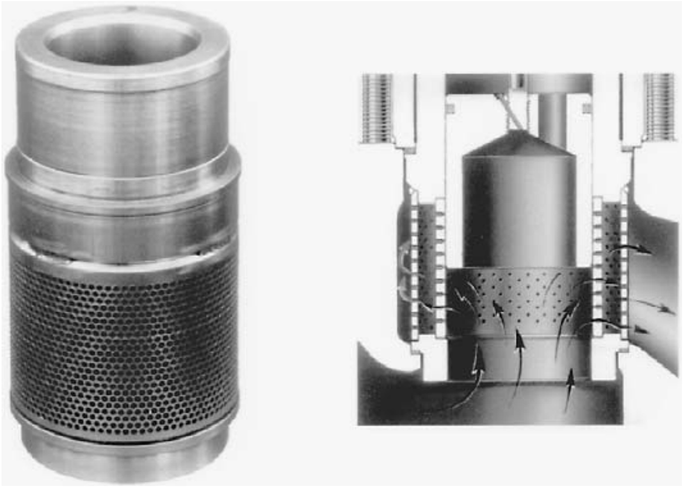


Figure 9.53 Globe valve equipped with two-stage attenuation cage. (*Courtesy of Fisher Controls International, Inc.*)

the key engineering elements of multihole trim is its utilization of sudden expansions and contractions. With flow under the plug, the pressure drop occurs as the flow moves through the seat to the inside diameter (an expansion), through the first stage cylinder (a contraction), through the area between cylinders (an expansion), through the second-stage cylinder (a contraction), and so forth. With this method, a portion of the pressure drop is taken at each stage. As the pressure drop is taken in stages, velocity is maintained at acceptable and reasonable levels of around Mach 0.33. The number of stages, flow areas, and flow-area ratios are determined by the velocity control required to avoid high sound pressure levels. In other words, the greater the control, the more stages and flow area that are required. The only limitations to the number of stages are the inside dimensions of the globe-body gallery and the amount of flow required to pass through the valves.

As the flow moves through the valve's vena contracta, the increased velocity, along with the geometry of the seat, creates turbulence. If untreated, this may create pressure fluctuations and eventually noise as the flow carries down the pipe. With multihole devices, the large turbulent eddy is broken up into smaller eddies. As the flow moves through the entire trim, the resulting small eddies are easily dissipated into the overall flow stream, which is illustrated in Fig. 9.54. The use of

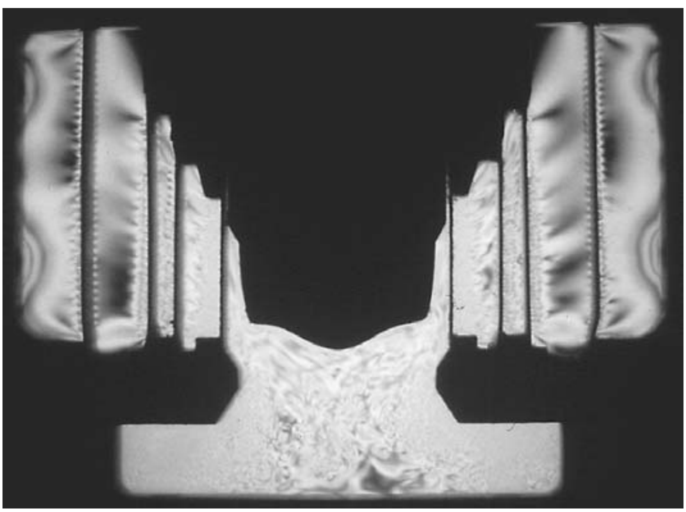


Figure 9.54 Schlieren display showing dissipation of turbulent eddies with attenuation trim. (*Courtesy of Valtek International*)

smaller holes also decreases the noise energy significantly. If one hole in a cage generates 90 dB, studies have shown that two smaller holes (which add up to the total area of the original hole) will generate less noise—in this case 84 dB. This is due, in part, to the principle that the energy generated by noise is proportional to the square of the hole area. Therefore, using one hole instead of two will provide the desired flow, but will also double the sound pressure level. Each succeeding cylinder is designed with more or larger holes. Not only does this provide an increased flow area, but it also handles the increased gas volume that results from the pressure drop. In addition, the materials and overall design of multistage devices are selected to provide maximum acoustic impedance, avoiding any geometry that may create monopole, dipole, or quadrupole noise. This is especially important when the plug is throttled close to the seating surface where noise is most likely to occur.

When used as part of the valve's trim, multihole devices can achieve attenuation of sound pressure levels up to 15 dB for one- and two-stage devices, while multistage devices can achieve up to 30 dB. When high-pressure ratios ($\Delta P/P_1$) are greater than 0.8, the addition of a downstream element (in conjunction with the valve) can divide the pressure drop between the two. However, both should be engineered to produce the same noise level so as to not increase the overall sound pressure level. As discussed in Sec. 9.2, quarter-turn plug valves can be equipped with severe service grids, similar to multihole devices, which take an additional pressure drop and control turbulence.

As detailed extensively in Sec. 9.2.7, a tortuous-path device uses a series of 90° turns etched or machined into a stack of metal disks to slow velocity to acceptable levels. For gas service, this same device can be used, although the flow direction is opposite that of liquid applications. The flow direction moves from the inside diameter of the stack to the outside diameter. The tortuous path becomes wider and/or deeper as it progresses, widening the flow area. Each turn in the tortuous-path device is considered to be one stage. With some mazelike paths, upwards of 40 right-hand turns are possible, achieving the same number of stages and providing extremely high attenuation. Tortuous-path devices typically provide attenuation up to 30 dB.

Like the tortuous-path trim, *expanding-teeth trim* uses a stack of disks. Instead of a tortuous path, however, expanding-teeth trim uses a series of concentric grooves (referred to as *teeth*) that are machined onto both sides (face and backside) of the disk (refer again to Fig. 9.13). Flow arrives from under the plug to the inside diameter where it passes through the wavelike teeth in a radial manner. As shown in that figure, the spacing between the teeth grows significantly larger as the flow moves to the outside diameter, permitting flow expansion, increasing pressure, and decreasing velocity. In addition, as the flow moves over the grooves, the phenomenon of sudden expansions and contractions takes place, which provides staged pressure-drop reduction and increased frictional losses. Figure 9.55 shows where the fluid expansions and contractions take place as the flow moves over the grooves. One advantage of the expanding tooth design over the tortuous-path or multiple-hole trims is that its passages are wider than the beginning of a tortuous path or a hole, which allows for particulates to flow through the stack without clogging the inlet passages. Each groove (or tooth) in the stack is considered to be a stage, and in most cases this trim can have up to seven grooves, providing seven stages of pressure drop. Depending on the number of teeth in the design, this trim can provide up to 30 dB attenuation.

An antinoise trim can often be used in series with a downstream element to attenuate noise to acceptable levels. For example, when noise is close to the threshold of pain (140 dBA) and the valve cannot be removed from a reverberate chamber, such as a metal building, installing antinoise trim may make a significant reduction of up to 30 dB. However, to reduce the sound pressure level down to 85 dBA (allowing employees to work an entire 8-h shift), a downstream element must be installed to reduce the noise by another 15 dBA, which

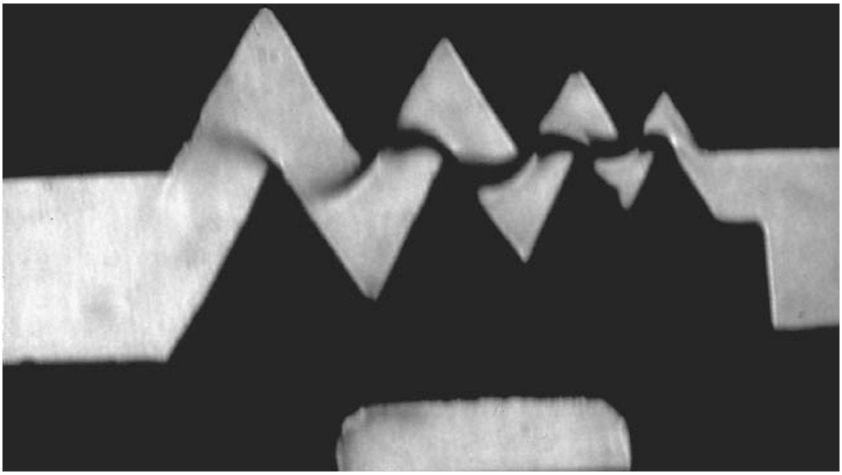


Figure 9.55 Schlieren display showing pressure reduction through sudden expansions and contractions with expanding-teeth trim. (*Courtesy of Valtek International*)

brings the noise level to 95 dBA. Insulation will most likely be needed to bring the level close to the desired 85 dBA. Although this is an extreme application, it does indicate the multiple options necessary to bring extremely high noise down to acceptable levels.

Antinoise trims with multiple stages that provide attenuation up to 30 dB are often the most expensive method of noise control. One- and two-stage devices are often less expensive but only provide attenuation up to 15 dB. The addition of a diffuser, for example, with a two-stage trim may provide the same attenuation as the more expensive multistage trim. When the required noise attenuation is 15 dB or less, in many cases a downstream element will accomplish what a two-stage trim can but at a lower cost. And in some cases, simple modifications to the process system or the orientation of the valve or changing of the pipe configuration may be even better cost-effective options—as long as the noise is only a hearing concern and is not destructive to the equipment.

9.9 Fugitive Emissions

9.9.1 Introduction to Fugitive Emissions

In many industrial regions of the world, increasing levels of environmental pollution have led to enactment of strict antipollution laws, which target emissions from automobiles, home heating systems, and industry. In particular, process industries have been under legislative mandate to reduce or eliminated fugitive emissions from their process systems. These antipollution laws target all devices that penetrate a process line, such as valves, sensors, regulators, flow meters, etc. Although many users see such legislation as costly and labor-consuming, a side benefit to tighter fugitive-emissions control is a more efficient system, with less lost product and greater efficiency. Even if a user is not under legislative mandate to reduce emissions, maintaining a strict antifugitive emissions program can provide greater production savings than the actual cost of the program. A case in point is the power-generation industry that, in the past, has accepted leakage of steam applications as standard operating procedure. Although steam (being water-based) is not a fugitive emission, power plants have discovered that using high-temperature seals prevents significant steam losses, which in turn lowers operating costs. In addition, power plants are operating more in the range of high-pressure superheated steam to improve energy efficiencies, which requires new sealing systems for safety reasons.

9.9.2 Clean Air Legislation

In the United States, the Clean Air Act was amended in 1990 to include some of the strictest laws regarding industrial pollution. In general terms, it mandates lower fugitive emissions from process equipment, including valves. Because most valves in today's chemical plants were installed prior to the new standards, maintenance personnel face a choice of retrofitting existing valves to the new standard or replacing them with new valves equipped with packing-box designs that comply with the Environmental Protection Agency (EPA). The Clean Air Act mandates a 500-parts-per-million (ppm) standard on all valves. As compared to past leakage standards, this new standard is 20 times more stringent. The Clean Air Act lists 189 hazardous materials that must be monitored by the law; 149 of these hazardous materials are volatile organic compounds (VOC), which can be easily monitored using an organic sniffer (Fig. 9.56). The Clean Air Act provides an

Figure 9.56 Organic sniffer used to detect fugitive emissions. (*Courtesy of Valtek International*)

incentive of fewer inspections if the valves are tested below the mandated 500 ppm. On the other hand, process systems with fugitive emissions higher than 500 ppm must increase the number of inspections and/or implement programs designed to improve the quality of the system.

The final phase of the Clean Air Act began in April 1997. A 500-ppm standard applies, but quarterly testing is permitted if less than 2 percent of all valves fail to meet the standard. If the failure rate is higher, monthly testing is mandatory unless a quality-improvement program is instituted. A plant can earn semiannual testing status if less than 1 percent of the valves fail to meet the standard. And finally, if less than 0.5 percent of the valves do not meet the standard, the plant can earn an annual test status. With the number of valves in a typical plant numbering in the hundreds and even thousands, achieving the higher semiannual or annual test status is important in order for the plant to avoid additional paperwork, testing, and maintenance. A graph indicating the program as outlined by the Clean Air Act is shown in Fig. 9.57.

9.9.3 Detection Standards

Clean air legislation calls for field monitoring of all line penetrations. Static seals at the flanges or body gaskets retain their seals for some

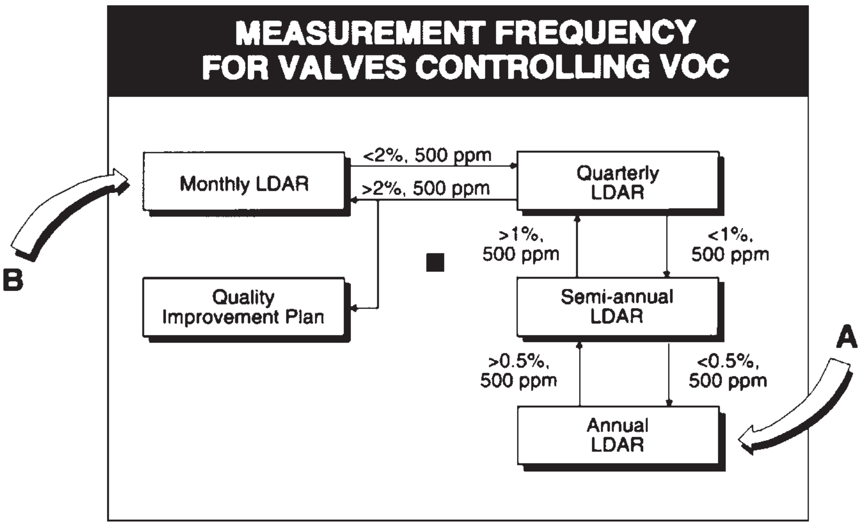


Figure 9.57 Monitoring frequency required by the Clean Air Act (United States). (*Courtesy of Fisher Controls International, Inc.*)

time. However, the sliding seal at the stem or shaft is more apt to leak because packing damage occurs over time due to friction. Because of the potential leak paths, valve packing boxes attract the most attention when fugitive emissions testing is performed.

The leak detection and repair (LDAR) procedure outlines the procedures for inspections and leak repairs. In addition to the LDAR procedure, a related regulation is "Method 21: Determination of Volatile Organic Compound Leaks." In general terms, Method 21 provides leakage definitions, as well as the proper procedures for using an organic sniffer to detect a leak or measure the leakage from the valve's static seals and the dynamic seal at the packing box. With linear-motion valves, the leakage reading is taken where the rising-stem slides out of the bonnet. With rotary valves, the reading is taken where the shaft penetrates the body. Measurements are also taken at all static seals around the body, bonnet, and flange gaskets.

When metal bellows-sealed valves are used (Sec. 9.9.6), they can be equipped with a leak-detection port, which can be used to monitor any fluid leakage between the bellows and the packing box. Although a negative (no emissions) measurement can be taken at the seal, the user can also read a leak-detection gauge for visual verification that the bellows has remained pressurized.

9.9.4 Packing-Box Upgrades

The user may replace an existing valve with one that has EPA-compliant designs. However, before the valve is replaced, its design should be reviewed to determine if the valve packing box can be upgraded to an improved packing or a live-loaded configuration. Overall, upgrades are more cost effective than purchasing a newer design. However, the upgrade may affect valve performance with more stem friction that can create sticking or erratic stroking. Upgrading the valve also means that continual monitoring is required during a period of break-in. Maintenance costs will also increase.

Because the packing box is the valve's primary dynamic seal it usually receives the most attention rather than the static seals (body, bonnet, and flange gaskets). One criteria for the new packing-box design should be its ability to compensate for packing consolidation, which occurs when the packing volume is reduced by wear, cold flow, plastic deformation, or extrusion. When packing is first installed, a certain amount of space can be found between the rings. As the packing is compressed to form a seal, these gaps slowly collapse. As packing loses its seal through friction, more force is applied to once again provide a seal. After several tightenings, all available space between the rings is exhausted. The packing is now one solid block and is incapable of further compression. When continued force is applied to consolidated packing, if the packing is soft and fluid, it may extrude up or down the stem or shaft. A photograph of packing that has extruded is found in Chap. 2 (Fig. 2.41).

Since 1990 when the Clean Air Act was amended, valve manufacturers introduced a number of packing-box designs that comply with the EPA requirements, many of which can be upgraded or retrofitted into existing valves. In nearly all cases, the costs associated with upgrading an existing valve are far less than installing a new valve. The following criteria should be evaluated before determining if a valve can be upgraded to an EPA-compliant packing. First, the user should ensure that the upgrade can be accomplished easily, safely, and economically. In some cases, the valve can remain in the line while the retrofit takes place—although the line should be drained and decontaminated, if necessary, for safety reasons. In some cases, the retrofit procedure may be so complicated that the valve must be sent to the factory or an authorized repair center for the conversion. This may present a problem if the valve is a critical final element of the system or if a replacement valve is not available. Second, the user should ensure that the upgraded packing box will meet the 500-ppm standard without continual packing readjustments. In addition, the packing box should continue to perform under the 500-ppm standard for long periods of time. Third, some consideration should be given to whether the upgraded packing-box design requires new maintenance procedures or installation equipment (which may require additional training for maintenance personnel). The best solution, and the least costly, is an upgrade that permits using the original bonnet, body, stem, or shaft. If live-loading is necessary, space for the fasteners and live-loading mechanism must be available above the bonnet or body. In some cases, an upgrade requires a new bonnet for linear valves or a new body for rotary valves. Unfortunately, the introduction of these expensive new parts often increases costs so much that an overall new valve is the best option.

A careful review of an existing valve's packing-box features should be conducted to reveal upgrade possibilities and the probability of success. Some packing-box designs have features that are better suited for upgrading, while others have features that may result in leakage or premature failure. A number of design features improve the likelihood of success in upgrading packing boxes. Bonnets manufactured from forgings or barstock inherently seal better than bonnets made from castings. Although less expensive, bonnets made from castings may have minuscule cracks or porosity, which sometimes cannot be detected during manufacture and inspection without use of a dye penetrate test. The problem with these minute cracks or porosity is that leakage from these avenues cannot be halted by tightening the packing. Double-top stem guiding is commonly used in linear-motion valves to contain the packing with both the top and bottom guides. This arrangement provides a concentric and constant alignment between the plug stem and the bonnet bore. The lower guide also acts as a barrier against particulates or other impurities, which may affect the integrity of the packing. Double-top stem guiding also avoids the problems inherent to caged-guided trim, which may lead to increased fugitive emissions. The longer distance between the two guiding elements (the upper guide and the cage) allows column loading and stem flex. Plug stems with small diameters can create side loading in the packing box and possible leakage. Because the packing box itself lacks a bottom guide, particulates in the fluid can damage the “wiper” set of packing.

Deep packing boxes are designed to allow for a wider separation of upper and lower guides in the double-top stem guiding design, which provides accurate guiding of the plug head into the seat. Regarding fugitive emissions, a side benefit of a deep packing box is that it allows the upper set of packing to be completely separated from the lower set, which is designed to protect and “wipe” the fluid medium from the plug stem. This wide spacing of the packing sets avoids contact with any part of the plug stem exposed to the flowing medium. Shallow packing-box designs permit the exposed plug stem to contaminate the upper seal. A buildup of process material could also damage the dynamic seal between the stem and packing.

Packing works best with a highly polished plug stem or shaft. A typical plug stem or shaft will be approximately 8 μin root mean squared (rms). On the other hand, a static seal (such as a bonnet bore in a linear valve or a body bore in a rotary valve) would be designed with a surface finish of 32 μin rms.

If the application requiring low fugitive emissions can utilize either a linear or rotary valve, a rotary valve may be the best choice. Because of the circular action of the ball or disk, the seal between the packing and the shaft travels around the shaft circumference instead of linearly up the shaft. This shorter action produces less friction and wear and in the long term promotes packing life. Additionally, consolidation of the packing is far less because the individual rings are stressed in a tangential direction rather than an axial direction.

9.9.5 Live-Loading

Live-loading is often installed to apply a constant packing load without requiring continual retightening of the packing bolting. Live-loading is designed to compensate for packing load losses due to consolidation as well as thermal contraction and expansion. If space exists between the gland flange and the top-works of the valve, live-loading can be retrofitted on most linear and rotary valves. As illustrated in Fig. 9.58, a typical live-loading design uses disk springs above the packing flange to provide a constant load to the packing when properly torqued. The typical disk spring is a metal washer, with the inside diameter formed so that it rises higher than the outside diameter. Two disk springs are placed from inside diameter to inside diameter and stacked with other sets, allowing for a springlike configuration. Disk springs are normally made from corrosion-resistant stainless steel, although Inconel is sometimes used for highly corrosive environments.

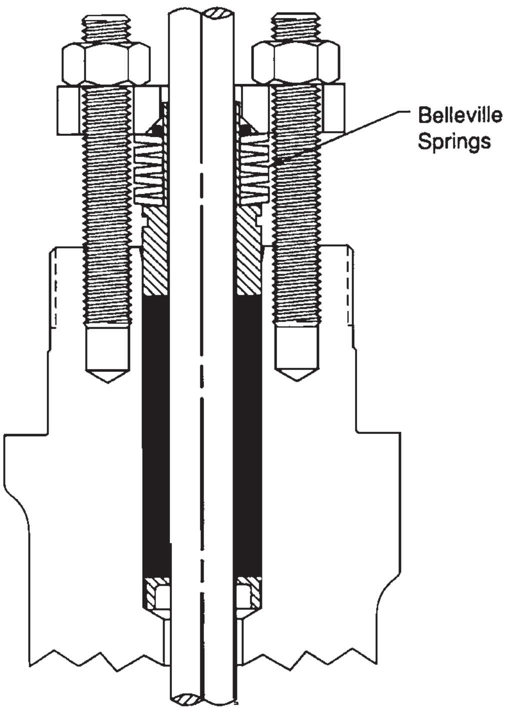


Figure 9.58 Conventional live-loading design with single stack of disk springs (*Courtesy of Fisher Controls International, Inc.*)

In live-loading, the disk springs are compressed by the gland-flange bolting, allowing a certain percentage of possible travel (typically 80 to 85 percent). As the packing volume decreases due to extrusion or friction, the disk spring's action continues to provide a load to the packing without retorquing. This is especially important since most packings can lose at least 0.02 in (0.5 mm) during the early stage of compression. Without live-loading this height loss would result in the relaxation of the packing and eventual leakage, unless the user retightens the packing. The use of live-loading compensates for this first initial loss in height. As packing settles over time, causing the springs to return to their natural position, the spring force will decrease slightly. However, the overall loss is so low that the seal is not normally affected. The amount of force applied by the live-loading can be controlled by the type of disk spring as well as the compression of the disk spring.

In addition to the reduced need for retorquing, live-loading is ideal for applications in which thermal cycling is a problem. With normal packing configurations, if the packing is tightened when the temperature is high, the packing will leak when the temperature lowers. If the packing is tightened when the temperature is low, the stem or shaft may grab or stick due to thermal expansion when the temperature increases.

Live-loading has other disadvantages than the initial cost as well as the acquisition and installation of new parts. With some valves, little or no room exists between the packing box and the top-works of the valve for upgrading to live-loading, although some manufacturers provide special live-loading configurations for limited space applications, as shown in Fig. 9.59. This design uses an upper plate as the gland flange and a lower plate as the packing compressor with stacks of disk springs located on the outside fringes of the two plates.

The torque values provided by the manufacturer to maintain the proper spring compression of the washers may be affected by the condition of the bolting. If the bolting is new and lubricated, the resulting torque value will be much different than if the threads are corroded and nonlubricated. Some packings may not respond to live-loading as well. For example, because of its high density, graphite packing requires a greater load than the manufacturer specifies for normal packings. If the live-loading is placed in a corrosive atmosphere, the disk springs can also lose strength through corrosion or even bond together, restricting free movement of the disk springs.

Some users argue that the use of live-loading actually contributes to early failure of packing through extrusion by applying more force to the packing than is required to achieve an adequate seal. If extrusion

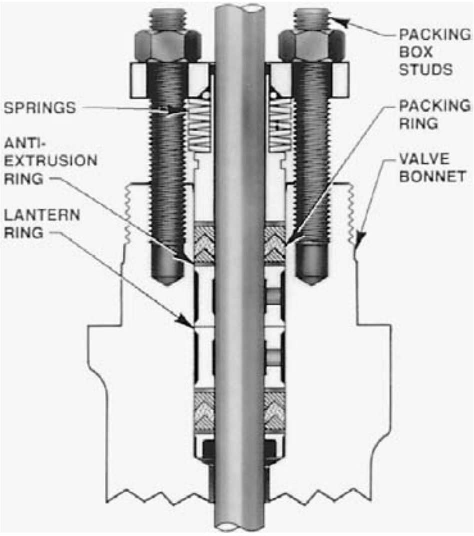


Figure 9.59 Live-loading design used for limited space applications. (*Courtesy of Fisher Controls International, Inc.*)

occurs, after some time the packing box will lose so much packing material that a seal will not be possible. Consequently, if live-loading is desired, antiextrusion rings should be included inside the packing box, especially if a soft packing is used. Too much compression may also be the problem. In that case, a thinner disk spring (which will apply less force) can be specified.

Another argument against live-loading is that, unless the live-loading provides equal amounts of force on the packing, it can cause stem-alignment problems with linear valves. This can occur if tolerance buildup occurs on some disk-spring stacks and not others, causing an unbalanced packing load and slightly affecting the stem alignment (especially with extremely thin stems or shafts which can flex). Such misalignment can affect both the shutoff and packing seal. This may be remedied, however, by using stem guides that have close-fitting guide liners or by using linear valves with oversized stems.

9.9.6 Metal-Bellows Seals

As a safety measure to workers and the general community, hazardous and corrosive applications must not be allowed to leak any fugitive emissions. In some toxic or lethal processes, however, the

Environmental Protection Agency (EPA) can designate a portion of the plant as a nonattainment area where a small amount of fugitive emissions are allowed. If a process is expanded to include more line penetrations, the parameters of the nonattainment area are often not easily expanded by regulations; therefore the user must not introduce new fugitive emissions. In this case, valves that are incapable of leaking are often required.

Linear valves equipped with a standard packing box always present a risk of leakage. When zero leakage is required, a *metal-bellows seal* is usually specified. A typical metal-bellows seal design contains the fluid with a specially formed metal-bellows welded to the stem of the plug. As shown in Fig. 9.60, the bellows is designed to expand or contract with the linear stroke of the valve, while providing a solid, permanent barrier between the fluid medium in the body and any potential leak paths to atmosphere. A metal bellows presents the best solution against fugitive emissions, as long as the body gaskets hold

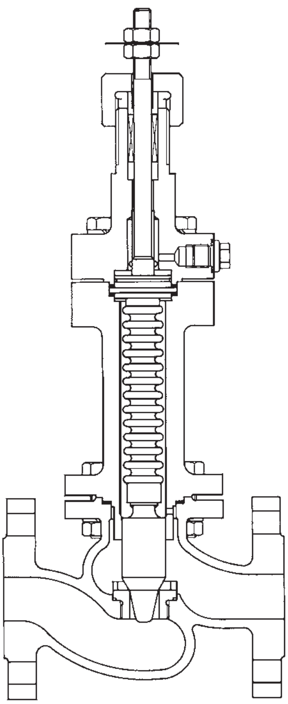


Figure 9.60 Hydroformed bellows with extended bonnet design. (*Courtesy of Kammer Valves*)

their static seals and the metal-bellows seal remain intact. Valves equipped with bellows seals do have some limitations, such as shorter strokes, decreased stroking life, and greater height. Valves with bellows seals can also cost 20 to 40 percent more, although that cost is offset by less monitoring and packing maintenance.

In throttling applications, the bellows is welded to the stem in the middle of the stroke. In the middle of the stroke the bellows is in a "relaxed" state and is equally stretched at the full-open and full-closed positions. This maximizes the life of the bellows. In applications in which a majority of the throttling is done between the 25 and 75 percent range, a bellows life of up to 200,000 strokes is possible. If a full stroke is required (0 to 100 percent), the life drops dramatically—up to 60,000 strokes. On the other hand, in applications in which the valve remains shut (or wide open) for a good portion of the time, the bellows can be welded at different locations in the plug. The bellows stays in the relaxed position for a majority of the service, prolonging its life. A metal-bellows cycle life is expressed as the number of times that the bellows can be stretched to its full limit and then compressed without failure. Because a full cycle involves a complete expansion and contraction, a bellows rated at 10,000 cycles actually translates into 20,000 full valve strokes. Because throttling service may not require a full-open or full-closed position, the bellows may be stretched or compressed less than a full stroke, which will further prolong the bellows life. The bellows life can also be prolonged by changing the tuning setting on the process controller. Process controllers can be so highly tuned that they continually search for the correct signal, sending minute signals to the valve that varies in position with each signal. Although minimal, this continual movement of the valve will shorten the overall life of the bellows. The rated bellows life number is determined by the minimum number of cycles that a bellows can withstand at the maximum operating temperature and pressure. Although a bellows is designed for the operating services, the actual operating conditions are usually less than the maximum temperature and pressure, which further prolongs bellows life. This means that the bellows life can be many more times than expected. Some applications require minimal stroke travel in a service with lower-than-rated service conditions. For example, a bellows rated at 10,000 cycles can provide beyond 100,000 strokes, given the right conditions. Table 9.14 shows how reducing the stroke by half significantly prolongs the life of the bellows.

Bellows life is also dependent on the process pressures that act on the bellows. Bellows can be designed to allow the process fluid to be

Table 9.14 Bellows Cycle Life*

Valve Size	Maximum Bellows Travel	Half Stroke *(cycles)*	Full Stroke *(cycles)*
0.5-inch DN 12	0.56 inches 1.42 cm	1,400,000	150,000
2-inch DN 50	0.84 inches 2.13 cm	1,400,000	150,000
3-inch DN 80	1.12 inches 2.84 cm	700,000	300,000
4-inch DN 100	1.50 inches 3.81 cm	450,000	100,000

**Data courtesy of Fisher Controls International, Inc.*

Note: Data based on single-wall formed bellows, Inconel 625 material, 100°F (39°C), and 150 psig (10.3 bar).

contained in the inside or on the outside of the bellows. However, because a bellows is harder to compress externally than to expand internally, external pressure can double the life of the bellows. A bellows typically handles process pressures from 250 to 550 psi (17.2 to 37.9 bar). It can also be designed with up to four walls, ranging in wall thickness from 0.004 to 0.006 in (0.1 to 0.15 mm)—depending on the pressure and temperature ratings. Multiwall designs provide longer cycle life, because the multiple walls all share the stress of the process pressure instead of a single wall bearing the entire stress of the pressure. Multiple walls also allow for higher pressures over single-wall designs, as shown in Table 9.15.

Although many standard bellows are designed for pressures between 250 and 550 psi (between 17.2 and 37.9 bar), severe service bellows can be designed for pressures up to 3800 psi (262 bar) and temperature ranges from −320 to 1000°F (−195 to 535°C). Both high temperatures and pressures can affect the cycle life of the bellows, as is shown in Fig. 9.61. As a safety measure, bellows-seal valve manufac-

Table 9.15 Pressure Ratings for Single- and Double-Wall Bellows*

Valve Size	Bellows Walls	100° F 38° C	300° F 149° C	500° F 260° C	800° F 427° C
0.5 - 2-inch DN 12 - 50	Single	550 psig 38 bar	497 psig 34 bar	429 psig 30 bar	396 psig 27 bar
0.5 - 2-inch DN 12 - 50	Double	1,000 psig 69 bar	870 psig 60 bar	780 psig 54 bar	720 psig 50 bar
3 - 4-inch DN 80 - 100	Single	346 psig 24 bar	296 psig 20 bar	265 psig 18 bar	245 psig 17 bar
3 - 4-inch DN 80 - 100	Double	625 psig 43 bar	544 psig 37 bar	488 psig 34 bar	450 psig 31 bar

**Data courtesy of Fisher Controls International, Inc.*

turers usually pressure-test each bellows seal at or over the rated service pressure.

Because of corrosion or erosion problems, the bellows is not normally placed in direct contact with the fluid; instead, it is placed just outside the flow stream, usually above the plug. A hole or a number of holes are used to allow the process fluid and pressure to bleed either to the outside or inside of the bellows. One problem that can occur with bellows pressurization is that process fluid leaving the flow stream may enter the area next to the bellows, where it cools and thickens. This can cause maintenance problems or undue bellows fatigue. In this case, external pressurization is preferred (Fig. 9.62), since cleaning the outside surfaces of a bellows during maintenance is much easier. Larger bleed holes can allow more liquid to circulate around the bellows and prevent the fluid from cooling.

Two types of metal bellows are in general use today and each is classified by its method of manufacture. *Welded bellows* (Fig. 9.63), also referred to as *diaphragm bellows*, are fabricated using a series of flat

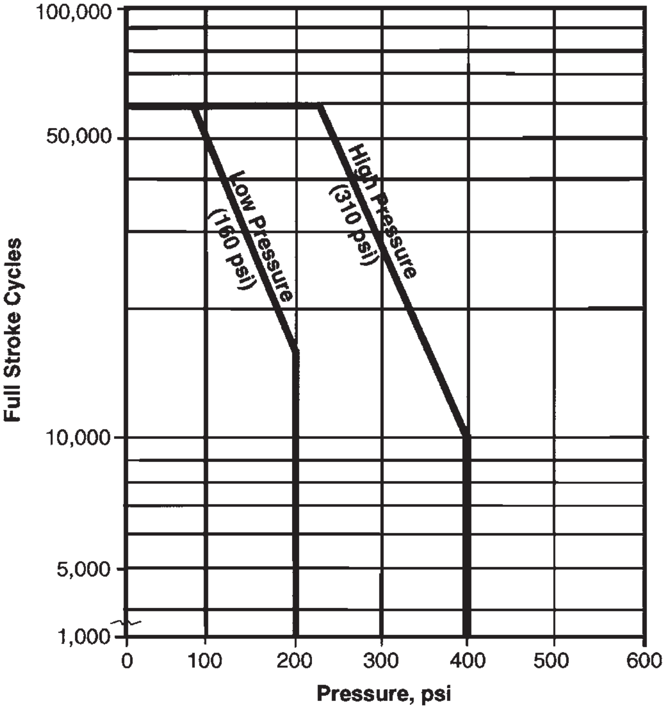


Figure 9.61 Full-stroke cycle life according to pressure. (*Courtesy of Valtek International*)

rings that are joined at the outside diameter and inside diameter by a fillerless tungsten inert gas (TIG) weld, creating a series of uniform convolutions. These convolutions have the general appearance of an accordion. Because welded bellows are made from flat rings, the overall height is quite compact and therefore can be contained in a relatively small area, adding only minimal height to the valve. For those applications requiring a small stroke, bellows can be contained inside the body (Fig. 9.64). This is particularly important where space consideration is critical or where seismic requirements limit the height of the valve's top-works. A primary disadvantage of the welded bellows is the welded edges of each convolution, which are easily stressed during expansion or contraction and are usually the first area to fail. Another problem can occur when particulates or solid matter becomes caught in the tight crevices of the convolutions. When this happens, these solids can create stress points in the convolutions and can cause premature failure. Welded bellows are also susceptible to corrosion

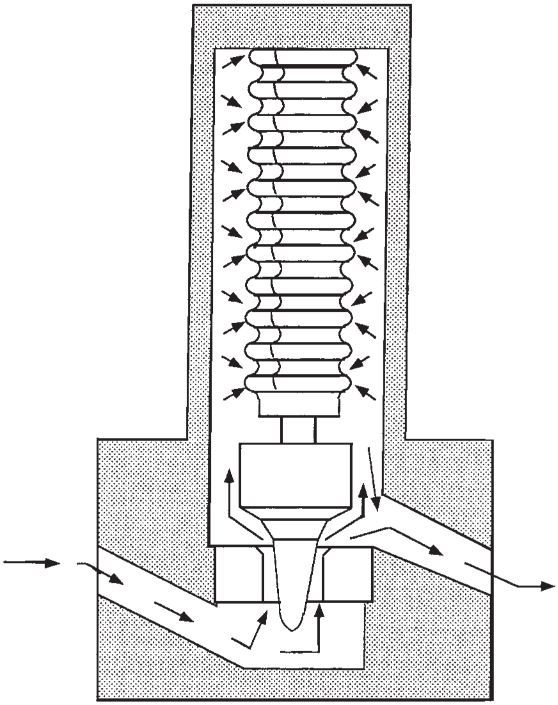


Figure 9.62 External pressurization of bellows. (*Courtesy of Kammer USA*)

because of the thin plate used in manufacture, especially when process fluid is continually trapped in the crevices. In addition, due to the difficulties associated with welding some alloys, material selection is limited. Because of the welded edges of the convolutions, the outside diameter of the bellows may restrict their use with some valve styles.

Hydroformed bellows (again refer to Fig. 9.63) is made from a flat metal sheet, which is rolled and fusion welded for solid construction. This tube is then mechanically or hydraulically pressed to create a series of uniform corrugations. More space is required for a complete corrugation—up to three times longer than a single convolution of a welded bellows. For this reason, hydroformed bellows are much longer than welded bellows for the same stroke length. They are encased inside an extended bonnet and have a greater height than normal valves (refer again to Fig. 9.60). One important advantage of the rolled construction is that process matter does not become entrapped in the folds, as is the case with welded bellows. Generally, formed bellows last longer than welded bellows because of the minimal welding,

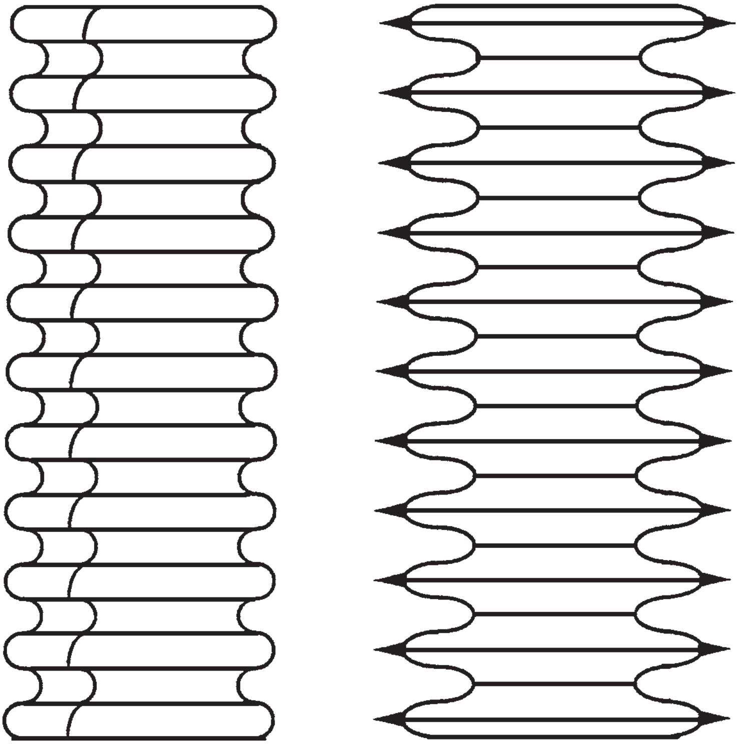


Figure 9.63 Hydroformed (left) and welded bellows. (*Courtesy of Kammer USA*)

the overall strength of the corrugations, and the limited travel of each fold (as compare to welded flat rings). They also handle higher pressures because of their greater strength. The main disadvantage is that formed bellows must be three times longer than welded bellows to handle the same stroke. The longer length may present problems with upgrading if space restrictions or seismic limitations exist.

In most designs, a packing box is placed above the bellows as a backup, in case the bellows ruptures from mechanical failure. To provide a warning of a bellows failure, a "telltale tap" can be installed in the bonnet, which is connected to an alarm system. Although not failproof, a metal-bellows seal provides the most reliable seal against leakage to atmosphere. Bellows can be made from a number of different materials, depending on the application, but 300 series stainless steels, Inconel, or Hastelloy C are standard materials because of their ability to resist stress fatigue and corrosion. Bellows can also be made from titanium, nickel, or Monel.

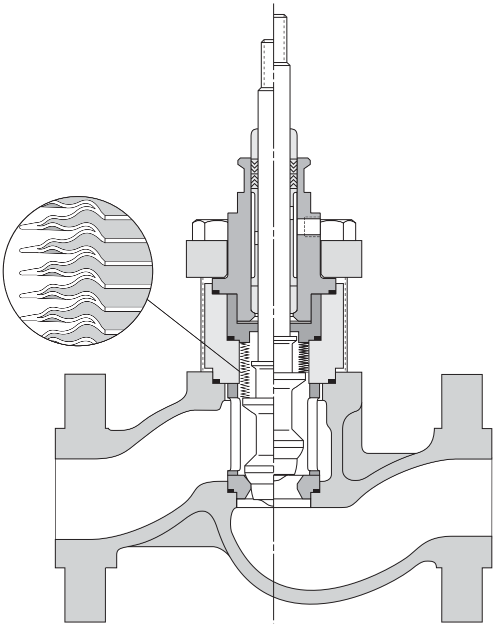


Figure 9.64 Body-contained welded bellows. (*Courtesy of Valtek International*)

A metal-bellows sealed valve may be seal-welded between the bonnet and the body as a precautionary measure with lethal or highly toxic services.

Because bellows seals are highly complex, retrofitting a linear valve is equally complex as well as very costly. In most cases, a new bonnet, plug or bellows assembly, and housing must be acquired, which can cost more than a new metal-bellows sealed valve.

9.9.7 Packing-Box Issues

When valves are initially installed in service, their packing boxes normally meet fugitive-emissions requirements. However, over time with continual operation, the packing will consolidate somewhat and begin to leak, requiring retightening of the gland-flange bolting. Most packing boxes will require retightening over time, until the packing reaches full compression. Further retightening only results in crushing the packing, rendering it useless. Manufacturers often provide suggested torque rates for given packing-box designs. This torque is applied to the gland-flange bolting, which in turn compresses the gland flange against the packing guide, finally resulting in full compression of the packing. Because of the problems associated with exact torque measurements, some designs have been simplified with the packing bolting tightened to just a flat or two past finger-tight. A manufacturer's recommended torque value can be affected by environmental corrosion or a lack of adequate lubrication, which can cause increased thread resistance and a false torque reading. Ideally, correct packing compression can be determined by measuring the packing's height when uncompressed and then applying torque until the manufacturer's ideal packing height is reached. Normally, the manufacturer's recommended packing height requires a 15 to 30 percent compression.

Maintenance technicians will sometimes overtorque the gland-flange bolting, believing that overcompression is better than undercompression. Unfortunately, too much torque can crush the packing, creating even greater leak paths. Because the packing will be compressed against the stem or shaft, high torques will boost the breakout force, causing an uneven (jerky) stroking motion. Due to the severe nature of the process or wide temperature swings, some applications require retightening often. For example, superheated steam applications may require retightening every few days. If this is not done, the packing box may develop a serious leak and be destroyed quickly by the high temperatures and pressures of the superheated steam.

The issue of torque is related directly to balancing leakage rates versus stem friction. As compression is applied to the packing by an axial load, packing deforms radially, pushing against two surfaces: the wall of the packing box and the stem or shaft. With greater compression, the greater the stress will be applied against these surfaces. As the packing deforms against the wall, any voids are closed off, permitting an effective seal. However, more compression also increases stem friction as the inside diameter of the packing grips the plug or shaft stem. This leads to erratic stem movement. Conversely, if the force to the packing is decreased to allow for smoother stroking, the packing may not fully grip the stem or shaft and leakage can occur.

Another factor that plays an important part in packing-box friction is the amount of contact between the packing and the stem or shaft. As the surface area of the packing touching the sliding stem or rotating shaft increases, more friction is produced that must be overcome to produce movement. High levels of friction will require greater force by the actuator, or a longer lever or larger diameter handwheel with manual valves. Some packings are V-shaped, which provides a very narrow point of contact and generates minimal friction. On the other hand, square packing (such as graphite) provides full contact and increased packing friction.

Most valve manufacturers today provide highly polished valve plug stems or shafts to accommodate the dynamic seal between the inside diameter of the packing and the stem or shaft. However, valve stroking or the service conditions of the process itself can deform, pit, or corrode the stem or shaft. Such wear can significantly increase the stem friction, while decreasing valve performance. This problem is often compounded when the seal is leaking and additional torque is applied to the packing to stop the leak. During routine maintenance, the stem or shaft should be carefully examined to ensure a smooth surface finish. If the finish is not smooth, that part should be replaced or repaired if the scratches or pits are not too deep.

9.9.8 Packings Specified for Fugitive-Emission Control

Today's packing materials are well suited to control fugitive emissions and can be adapted for retrofitting. Although no packing material or design is universal, many different packings exist that have broader applications than in years past. Choosing the correct packing is critical to the successful performance of the packing box. The packing should be compatible with the process fluid and service temperature and pressures as well as provide the desired seal between maintenance checks, without excessively high torque of the gland-flange bolting. The proper packing should also withstand consolidation and should minimize the friction on the stem or shaft, avoiding poor stroking performance.

A number of packing materials are commonly applied to anti-fugitive-emission packing boxes. Recently introduced in the past several years, perfluoroelastomer (PFE) packing is generally regarded by valve users as the best packing for complying with fugitive-emission standards. PFE provides an excellent seal with even the most difficult application. It resists degradation and chemical attack and is very resilient and elastic. PFE is rated to handle service temperatures from 20 to 550°F. A special low-temperature PFE has been developed that handles temperatures down to −40°F. As shown in Fig. 9.65, PFE requires a rigid backup V-ring system to support the packing. A property unique to PFE is that it wears well and compensates for any consolidation that takes place—although PFE can consolidate and eventually extrude if not supported by backup rings. Live-loading is not normally required, since PFE has an ability to return to its precompressed position. However, in applications with large thermal gradi-

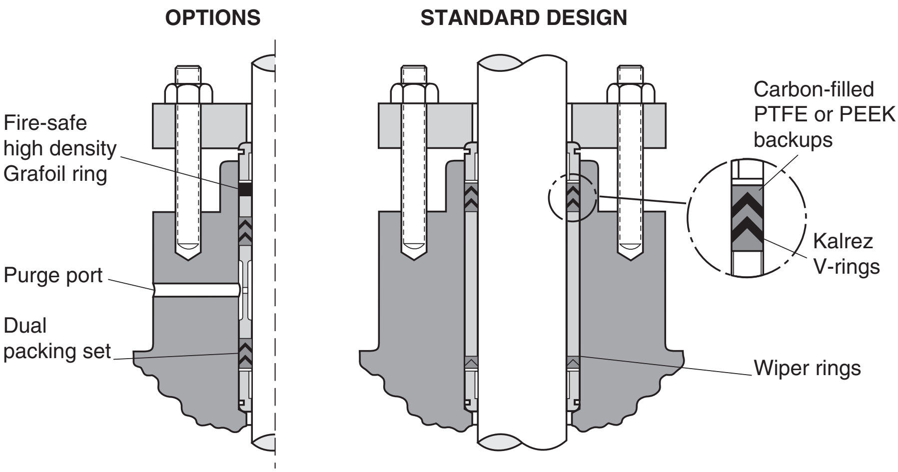


Figure 9.65 PFE backup ring packing configuration. (*Courtesy of Valtek International*)

ents, live-loading should be considered. The chief disadvantage of PFE is its high cost, although the initial cost of the packing is easily offset by reduced maintenance and increased up-time.

One of the most common and least expensive packings, "virgin" polytetrafluoroethylene (PTFE) packing is typically applied in a V-ring design (see Fig. 2.32 in Chap. 2). Virgin PTFE is often chosen because it has numerous advantages. Due to its pressure-energized design, coupled with "feather" edges, little compression is required to create a strong seal. Overall, virgin PTFE has good elasticity, which minimizes packing consolidation while responding well to a live-loading option. Because it is inert to many chemicals, virgin PTFE is found in a wide range of process services. The surfaces of virgin PTFE are extremely smooth; therefore, little breakout force is required to begin stroking the valve. Despite its wide application, virgin PTFE has some disadvantages. Its performance is limited to temperatures between −20 and 350°F. If the packing bolting is overtorqued—providing an excess load on the packing—the voids between the male and female rings can compress and result in consolidation. In addition, the spaces between packing spacers and the plug stem can result in extrusion, although antiextrusion rings or close-fitting spacers can be installed to prevent extrusion. Because of its tendency to cold flow and consolidate over time, virgin PTFE does require retorque on occasion.

The composition of "filled" PTFE contains 15 to 20 percent glass or carbon, which creates a more rigid V-ring design that is less likely to produce consolidation (which is common to virgin PTFE). Because its elasticity is less, filled PTFE does not seal as well as virgin PTFE. It also produces greater friction and is slightly abrasive to the stem or shaft. And, it is more expensive than virgin PTFE. Sometimes, as a compromise between virgin and filled PTFE, rings of both materials are alternated in the packing configuration to provide a good seal with reduced consolidation. Live-loading can also be used with filled PTFE to minimize retorquing.

Graphite and other carbon-based packings are commonly manufactured in die-formed or straight braided carbon-ring sets. As a measure against graphite migration, braided rings are often included in die-formed packing sets. This feature also protects the graphite rings from foreign particles. Braided rings are known to cause additional friction and leakage in high-compression applications. The main advantage of graphite packing is its ability to handle high temperatures (up to 800°F with a standard-length bonnet in an oxidizing environment). Graphite packings are usually offered in low-density or high-density graphite. Low-density graphite seals well and has lower friction, but must be retorqued often. High-density graphite has higher friction and provides a marginal seal but allows for a longer retorque cycle. To convert low-density to high-density packing, the packing can be torqued several times over a period of time. Compared to other packings, graphite packings are more expensive and do not respond well to live-loading systems. Also, the higher friction can affect the performance of the valve, requiring high breakout forces that may result in unstable stem movement. Typically, torque requirements for graphite ring packings can be eight to 10 times higher than those of PTFE or PFE packings. This usually requires the use of a torque wrench to ensure that overcompression does not occur. Overcompression will crush the graphite, causing it to extrude from the packing box.

9.9.9 Other Packing Considerations

Some users believe that if using the standard number of packing rings provides a good seal, using more rings should provide an even better seal against fugitive emissions. If a packing box is exceptionally deep, a user may be tempted to double the number of rings during routine maintenance. However, the use of extra-ring compounds several problems. First, multiple rings maximize the adverse affects of thermal expansion of the packing. Second, they increase stem friction substantially. Third, the manufacturer's recommend torque values will now be incorrect, providing far less compression than required. This may necessitate a trial-and-error approach to determining the correct torque value, which could shorten the life of the packing. Fourth, with more soft packing material in the packing box, unnecessary consolidation and extrusion can take place.

With rotary valves, the closure or regulating element or the actuation unit can apply stresses to the shaft, causing an incorrect center alignment. If a small-diameter stem (linear valves) is used, the force applied by the actuation to the closure element in the seated position can actually flex the stem. Whenever the stem or shaft are off-center with the packing box, a leak path for fugitive emissions can occur on one side. This problem can usually be avoided by using valves that feature oversized stems or shafts. Oversized stems or shafts present a large contact area between the stem or shaft and the packing, which will result in higher friction—although this is not an issue with high-thrust actuation units, such as piston cylinder actuators.