

# 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).

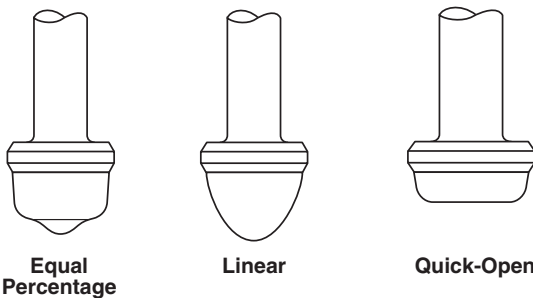
The three most common types of flow characteristics are *equal percentage*, *linear*, and *quick-open*. The ideal curves for these three flow characteristics are shown in Fig. 2.4. However, the inherent characteristic of these curves can be affected by the body style and design, and piping factors.



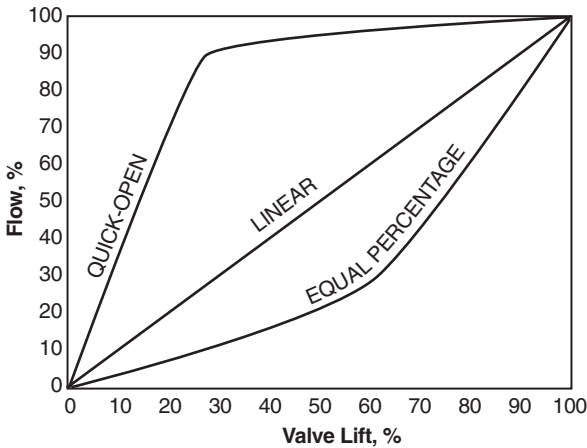
**Figure 2.2** Characterizable cages. (Courtesy of Fisher Controls International, Inc.)

### 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-percentage



**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

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

where  $Q$  = flow rate

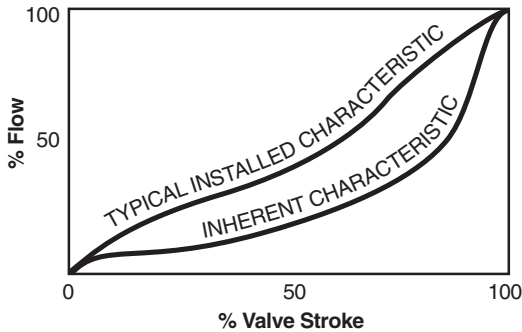
$L$  = valve travel

$e$  = 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

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

where  $Q$  = flow rate

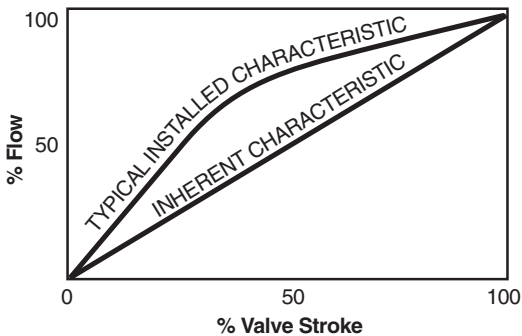
$L$  = valve travel

$k$  = 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

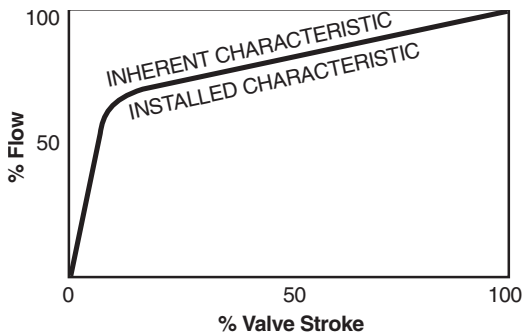


**Figure 2.6** Typical inherent and installed linear flow characteristics. (Courtesy of Valtek International)

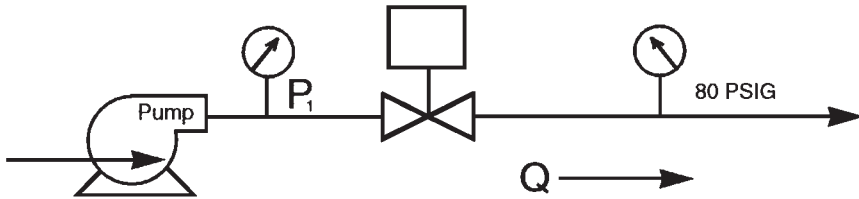
quick-open characteristic, the inherent and installed characteristics are similar.

### 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)

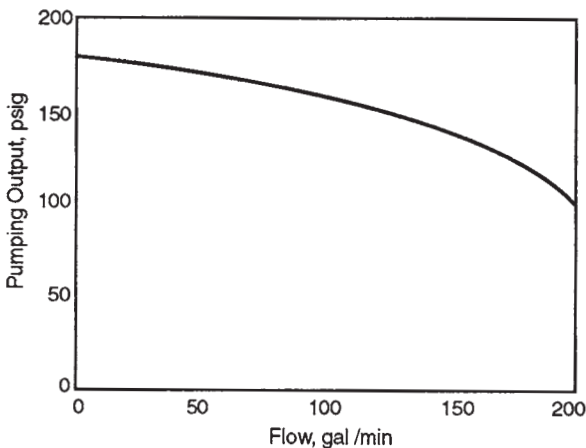


**Figure 2.8** Typical flow schematic showing no piping losses. (Courtesy of Valtek International)

### 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)

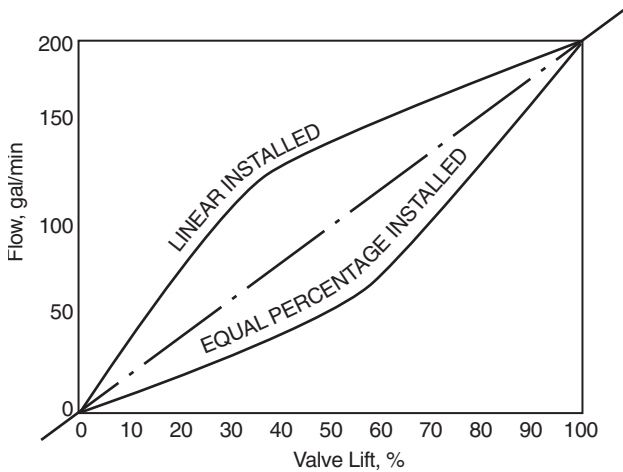
For illustration purposes, Example A was simplified with a constant downstream pressure and a pressure drop only affected by the pump

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

Q Flow <i>gpm</i>	P <sub>1</sub> Pump Discharge Pressure <i>psig</i>	ΔP Across Valve <i>psi</i>	C <sub>v</sub> Required	Percent of Valve Maximum C <sub>v</sub>
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  $C_v$ .

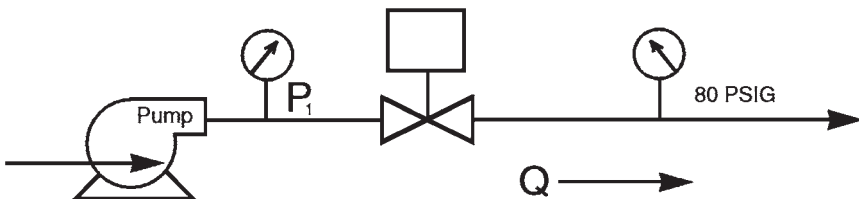


**Figure 2.10** Installed linear and equal-percentage flow characteristics (without piping losses). (Courtesy of Valtek International)

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)<sup>†</sup>

Q Flow gpm	P <sub>1</sub> Pump Discharge Pressure psig	$\Delta P_R$ Across Restriction	$\Delta P$ 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

<sup>†</sup>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

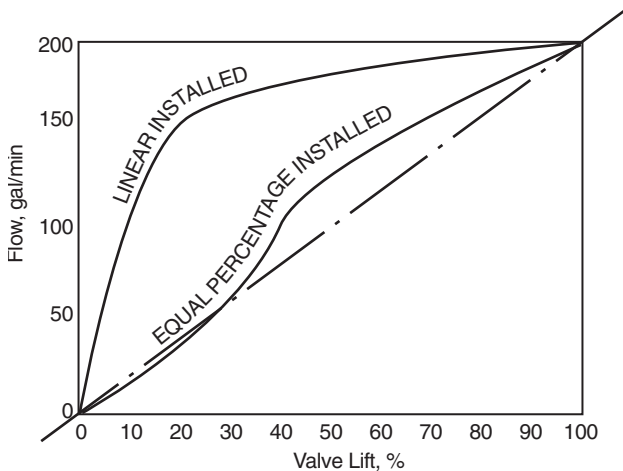
**Figure 2.12** Installed linear and equal-percentage flow characteristics (with piping losses). (Courtesy of Valtek International)

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 $\Delta P$	Linear
Decreasing $\Delta P$ with increasing load: $\Delta P$ at maximum load > 20% of minimum load $\Delta P$	Linear
Decreasing $\Delta P$ with increasing load: $\Delta P$ at maximum load < 20% of minimum load $\Delta P$	Equal Percentage
Increasing $\Delta P$ with increasing load: $\Delta P$ at maximum load < 200% of minimum load $\Delta P$	Linear
Increasing $\Delta P$ with increasing load: $\Delta P$ at maximum load > 200% of minimum load $\Delta P$	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 $\Delta P$ with increasing load, $\Delta P$ at maximum load > 20% of minimum load $\Delta P$	Linear
Gas process, large volume, decreasing $\Delta P$ with increasing load, $\Delta P$ at maximum load < 20% of minimum load $\Delta P$	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 $\Delta P$ 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 $\Delta P$	Equal Percentage
pH control where control valve $\Delta P < 50\%$ of system $\Delta P$	Equal Percentage
pH control where control valve $\Delta P > 50\%$ of system $\Delta P$	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 associ-



ated 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  $0.0005 \text{ cm}^2/\text{min}$  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

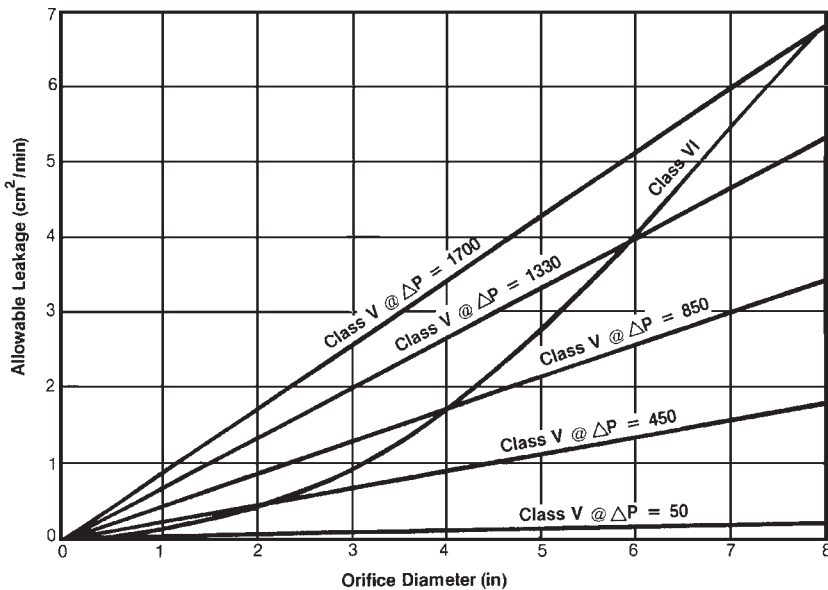
**Table 2.7** Typical Seat Loads vs ANSI Classification for Shutoffs\*

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.

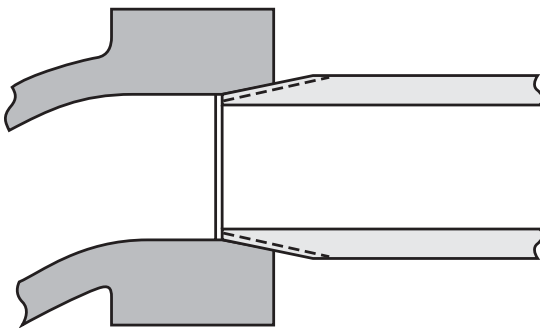
The general rule is that smaller-sized valves—smaller than 2-in (DN 50) valves—can use threaded connections (Fig. 2.14), while larger



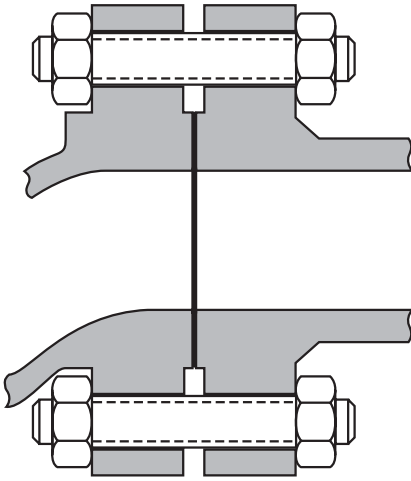
**Figure 2.13** ANSI Class V and VI allowable leakage. (Courtesy of Valtek International)

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  $\frac{3}{4}$ -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.

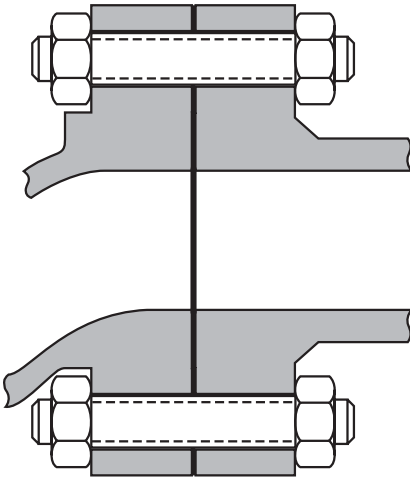
### 2.4.3 Flanged End Connections

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,

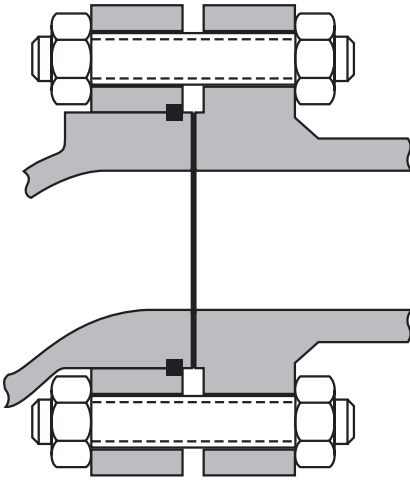


**Figure 2.17** Ring-type joint end connection. (Courtesy of Valtek International)

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).

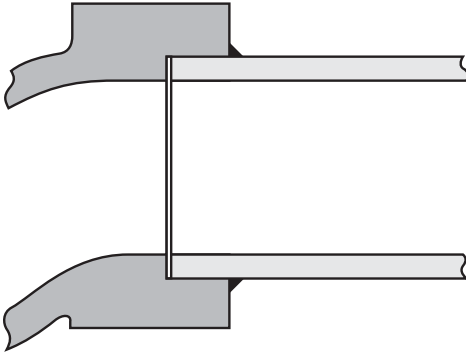
Separable flanges can be designed to be interchangeable among low-pressure classes. They are rated to ANSI Classes 150–600 (PN 16–PN 100) in sizes of 4 in (DN 100) and smaller. With ANSI Classes 150–300 (PN 16–PN 40), flanges are available in 6- and 8-in sizes (DN 150 and DN 200). Separate flanges can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



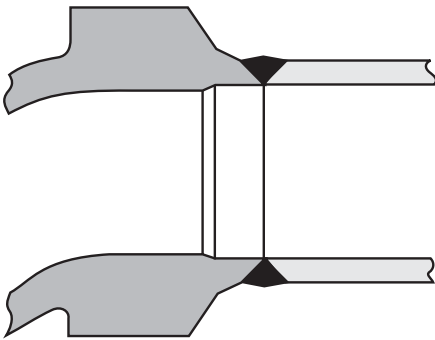


**Figure 2.19** Socketweld end connection.  
(Courtesy of Valtek International)

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  $37.5^\circ$  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  $37.5^\circ$  and  $10^\circ$  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-



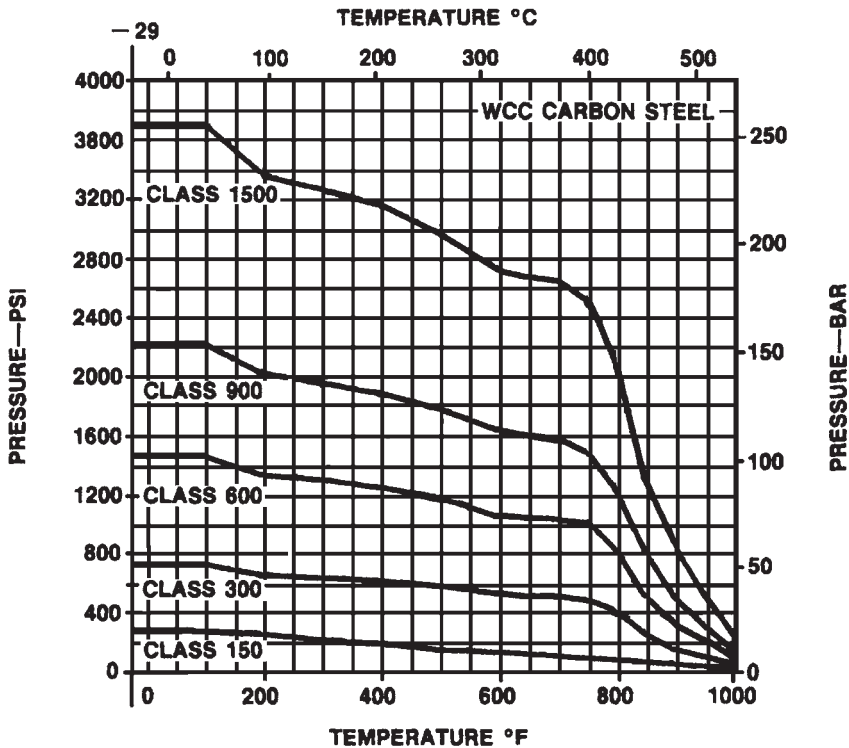
**Figure 2.21** Sanitary end connection. (Courtesy of Jordan Valve)

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 butt-weld 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.

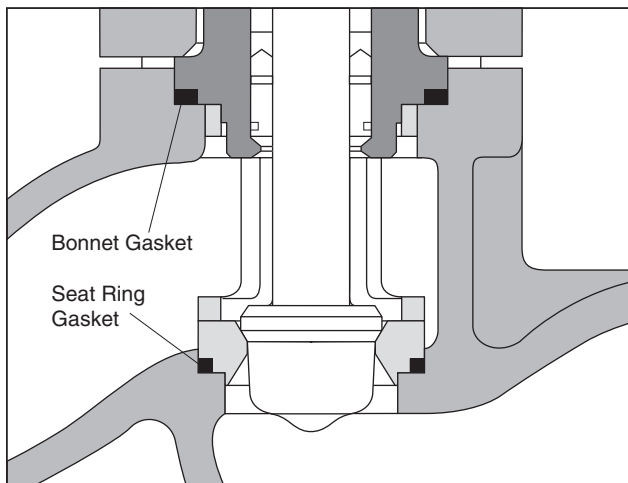
ber. ASME also regulates procedures for welding, heat-treating, and preheating. Another organization, the American Welding Society (AWS), oversees procedures and regulations for welding rod and wire.

## 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  $\mu\text{in RMS}$  (root mean squared) (between 3.2 and 12.5  $\mu\text{m}$ ).

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.

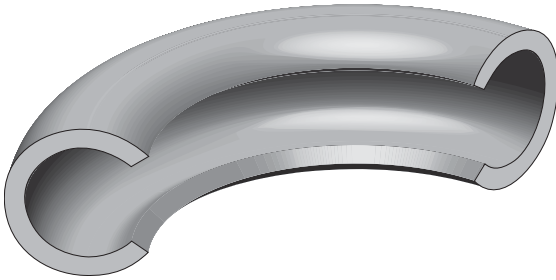
### **2.8.5 Metal C-Ring Gaskets**

*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.

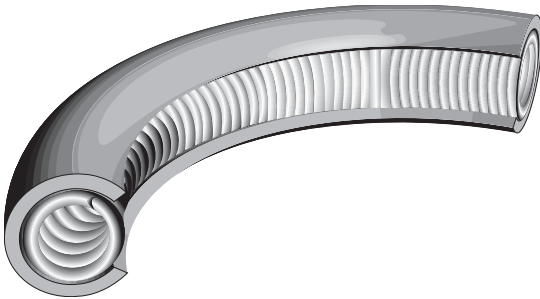
### **2.8.6 Metal Spring-Energized Rings**

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)

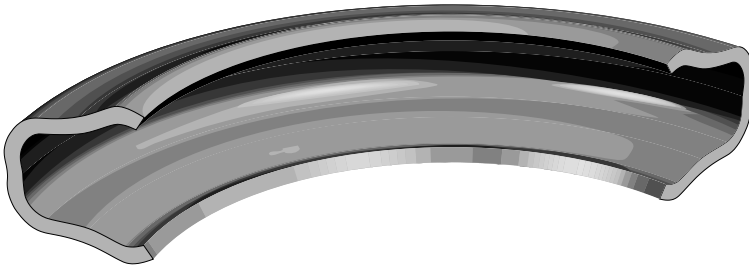
### 2.8.7 Metal U-Ring Gaskets

*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)

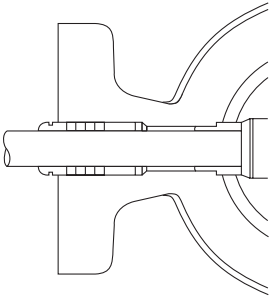


**Figure 2.28** Metal U-ring. (Courtesy of Advanced Products Company)

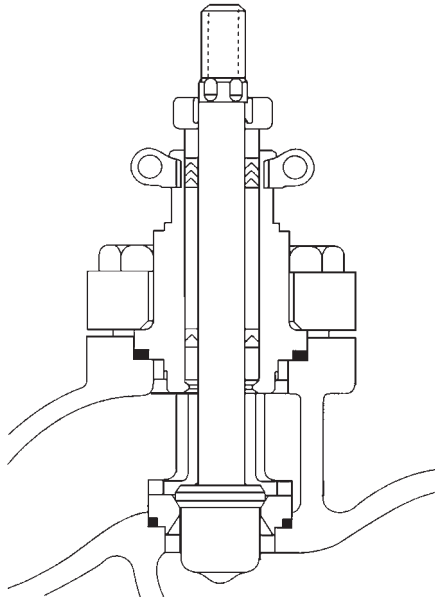
used to seal a valve closure element's stem or shaft is called the *packing*. The packing is normally held in place by a packing follower or by guides, with compression supplied by the gland flange. The *packing follower* is a metal ring used to retain the packing inside the bonnet or bonnet cap, as 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.

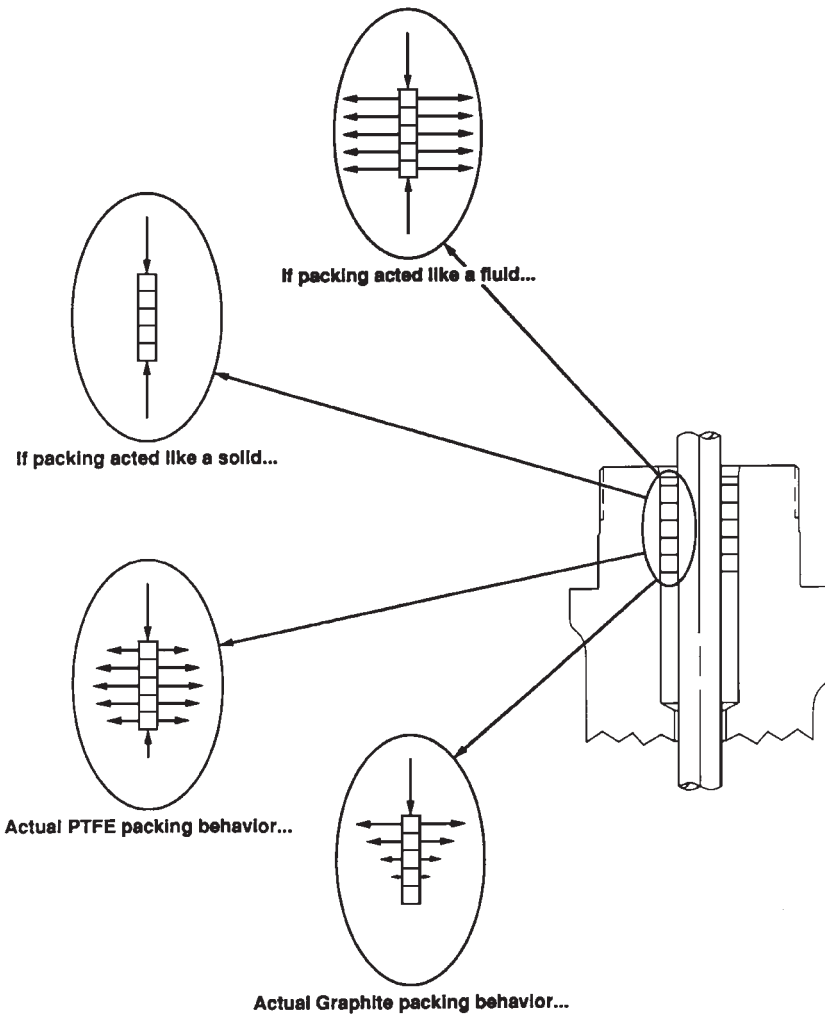
In addition, some designs provide an allowance above the packing box to allow the use of live loading. *Live loading* is a mechanical device used to apply constant force to the packing to compensate for *packing consolidation*, which is a reduction in the packing's volume due to wear, cold flow, plastic 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  $\mu\text{in}$  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

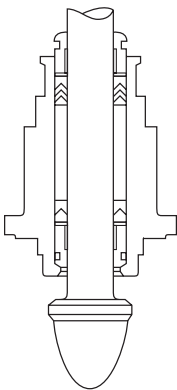


**Figure 2.31** Axial pressure effects on packing. (Courtesy of Fisher Controls International, Inc.)

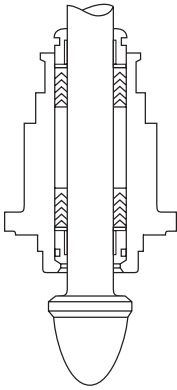
shaft, causing a leak path. Obviously, the type and close tolerance of guiding are critical to maintaining the concentricity of both the stem and shaft.

### 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  $\mu\text{in}$  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

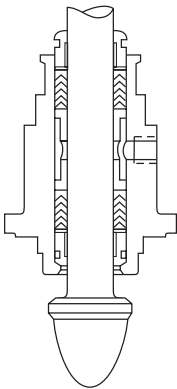


**Figure 2.32** Standard V-ring packing configuration. (Courtesy of Valtek International)



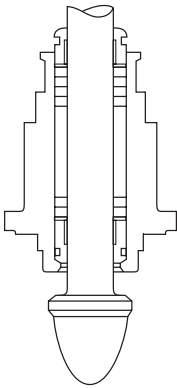
**Figure 2.33** Twin V-ring packing configuration.  
(Courtesy of Valtek International)

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.

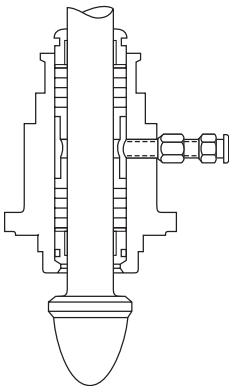


**Figure 2.34** Twin V-ring packing or lantern-ring configuration. (Courtesy of Valtek International)

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)

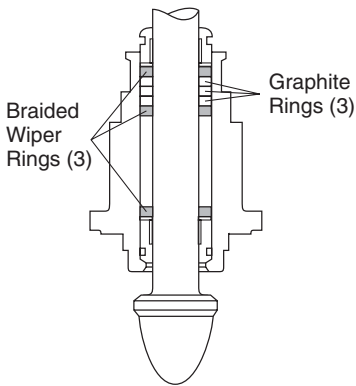


**Figure 2.36** Twin square packing or lubricator configuration. (Courtesy of Valtek International)

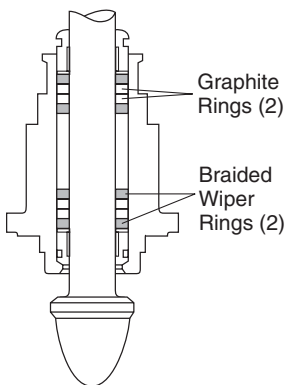


packing box; therefore, braided graphite rings—which are softer—are used to complete the seal. A braided ring is commonly used for the bottom wiper set. Both standard and twin configurations are possible with square and braided packing (Figs. 2.37 and 2.38).

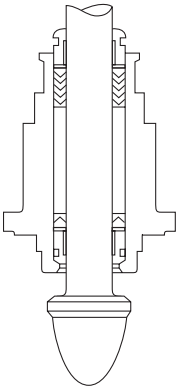
When the process fluid is at vacuum pressure or below atmospheric pressure, a special packing configuration is required. Because of their superior sealing ability, V-rings are used in a vacuum seal configuration (Fig. 2.39). If the process is always under a vacuum, the V-rings of both the upper and lower 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)

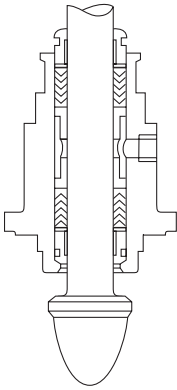
packing (Fig. 2.40) should be used, with the upper packing set inverted, while the lower set remains in a normal configuration. Occasionally a vacuum seal is necessary inside the packing box, which is independent of the process pressure. In this case, the twin V-ring packing configuration will permit 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*



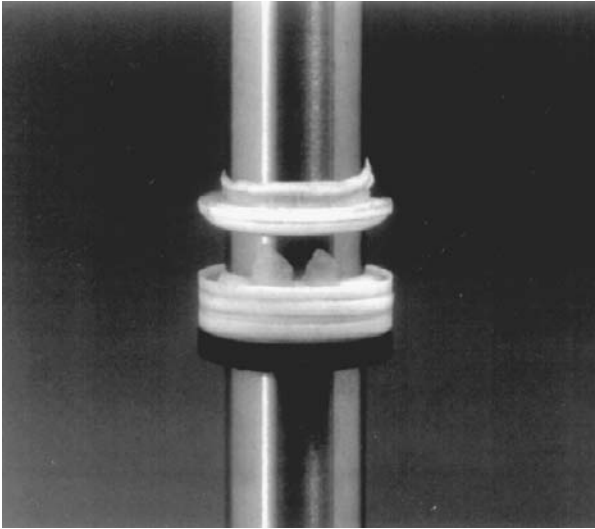
**Figure 2.40** Vacuum-seal twin V-ring packing or lantern-ring configuration. (Courtesy of Valtek International)

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**

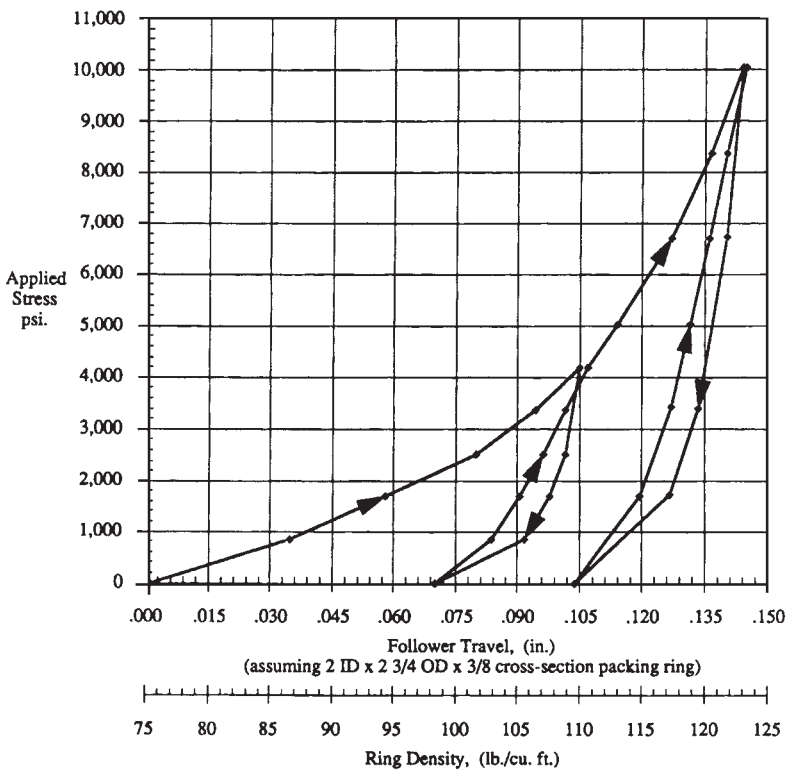
As a substitute for asbestos, *graphite packing* and other carbon-based packings are specified for high-temperature applications. Generally considered to be one of the more expensive packings, graphite packings can be produced either in die-formed rings or braided rings.

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<sup>3</sup> or 1440 to 1600 kg/m<sup>3</sup>) 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

### Density Versus Stress for Die Formed Flexible Graphite



**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 over-compress 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-



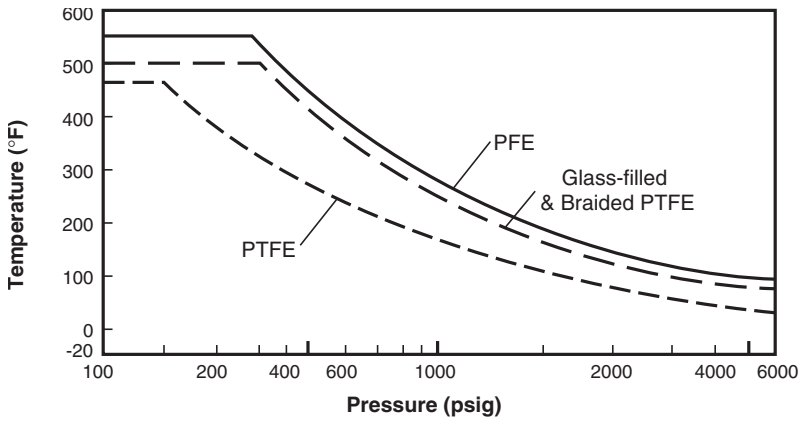
**Table 2.12** Temperature Limitations for Common Packing Materials\*†

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

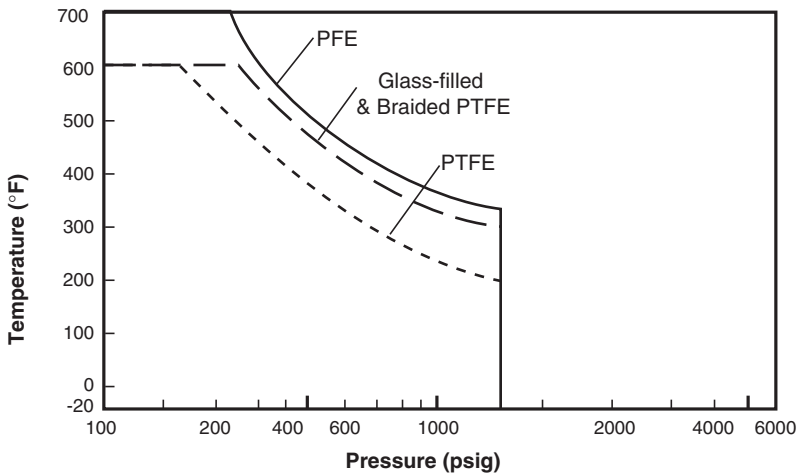
\*Data Courtesy of Valtek International.

†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.

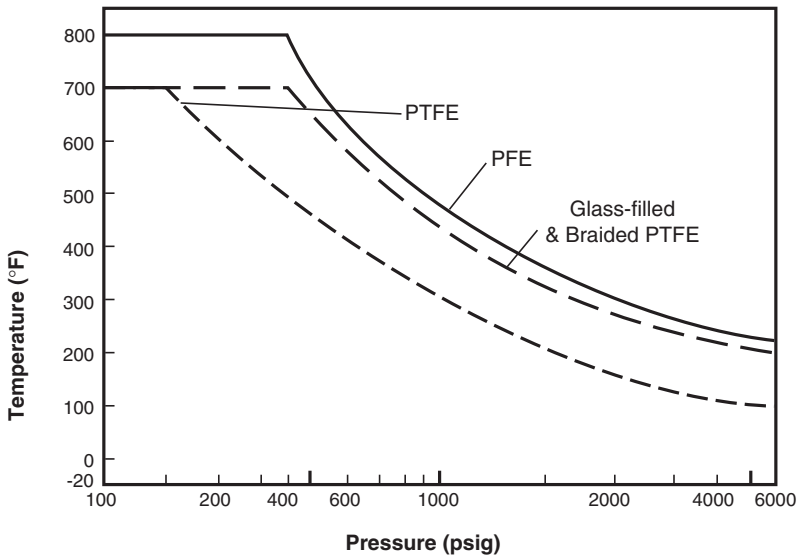


**Figure 2.43** Maximum temperature and pressures for packing contained in standard bonnets. (Courtesy of Valtek International)



**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.

