

9

Common Valve Problems

9.1 High Pressure Drops

9.1.1 Introduction to High Pressure Drops

Flow moves through a valve due to a difference between the upstream and downstream pressures, which is called the *pressure drop* (ΔP) or the *pressure differential*. If the piping size is identical both upstream and downstream from the valve and the velocity is consistent, the valve must reduce the fluid pressure to create flow by way of frictional losses. A portion of the valve's frictional losses can be attributed to friction between the fluid and the valve wall. However, this friction is minimal and is not sufficient to create enough pressure drop for an adequate flow. A more effective way to create a significant frictional loss in the valve is through a restriction within the body. Because many valves are designed to allow a portion of the valve to be more narrow than the piping, they can easily provide this restriction in the fluid stream. Because of the laws of conservation, as the fluid approaches the valve, its velocity increases in order for the full flow to pass through the valve, inversely producing a corresponding decrease in pressure (Fig. 9.1). The inverse relationship between pressure and velocity is shown by Bernoulli's equation, which is

$$\frac{\rho V_1^2}{2g_c} + P_1 = \frac{\rho V_{vc}^2}{2g_c} + P_{vc}$$

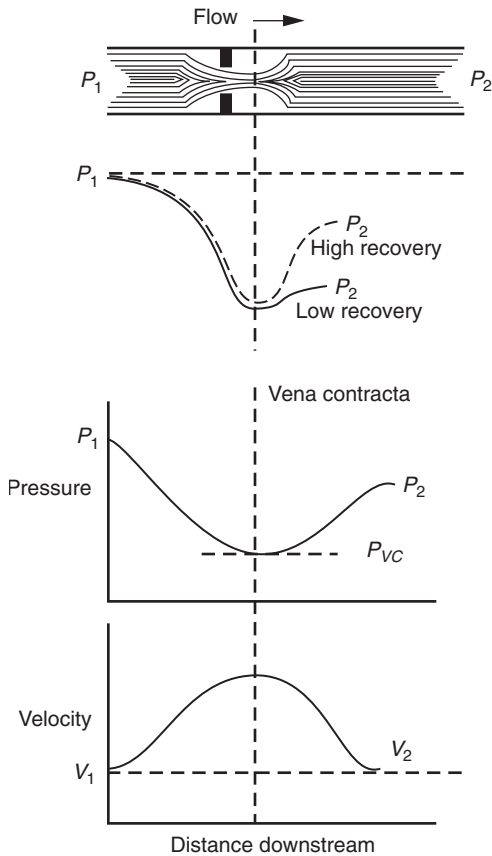


Figure 9.1 Location of vena contracta from point of orifice restriction and pressure and velocity curves. (Courtesy of Fisher Controls International, Inc.)

where ρ = density units

V_1 = upstream velocity

g_C = gravitational units conversion

V_{VC} = velocity at vena contracta

P_{VC} = pressure at vena contracta

P_1 = upstream pressure

The highest velocity and lowest pressure occur immediately downstream from the narrowest constriction, which is called the *vena contracta*. Figure 9.2 shows that the vena contracta does not occur at the

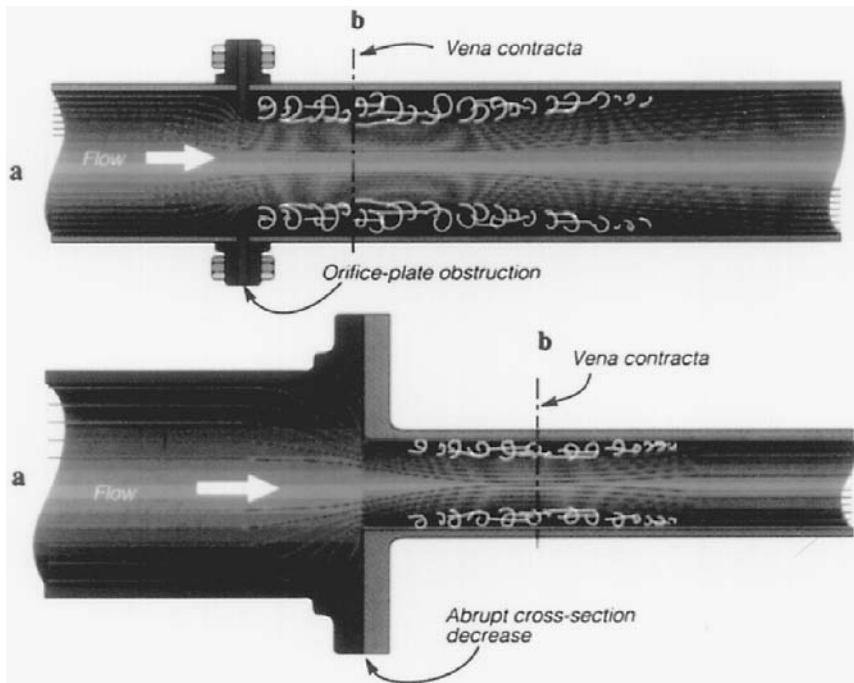


Figure 9.2 Relationship between orifice restriction and turbulence generation. (Courtesy of Fisher Controls International, Inc.)

restriction itself but rather downstream some distance from the restriction. This distance may vary according to the pressures involved. At the vena contracta the flow velocity is at a maximum speed, while the flow area of the fluid stream is at its minimum value.

Following the vena contracta, the fluid slows and pressure builds once again, although not to the original upstream pressure. This difference between the upstream and downstream pressures is caused by frictional losses as the fluid passes through the valve, and is called the *permanent pressure drop*. The difference in pressure from the pressure at the vena contracta and the downstream pressure is called the *pressure recovery*. A simplified profile of the permanent pressure drop and pressure recovery is shown in Fig. 9.3.

The flow rate for a valve can be increased by decreasing the downstream pressure. However, in liquid applications the flow can be limited by the pressure drop falling below the vapor pressure of the fluid, which will create imploding bubbles or pockets of gas (called *cavitation* or *flashing*, respectively). *Choked flow* occurs when the liquid flow

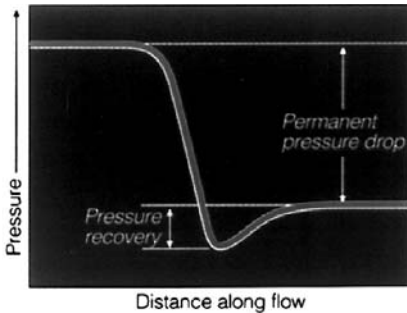


Figure 9.3 Flow curve showing pressure recovery and permanent pressure drop. (Courtesy of Fisher Controls International, Inc.)

is saturated by the fluid itself mixed with the gas bubbles or pockets and can no longer be increased by lowering the downstream pressure. In other words, the formation of gas in a liquid crowds the vena contracta, which limits the amount of flow that can pass through the valve. With gases, as the velocity approaches sonic speeds, choked flow also occurs and the valve will not be able to increase flow despite a reduction in downstream pressure.

9.1.2 Effects of High Pressure Drops

As discussed in Sec. 9.1.1, the flow function of the valve is dependent on the existence of a pressure drop, which allows flow movement from the upstream vessel to the downstream vessel or to atmosphere. Because a pressure drop generated by the valve absorbs energy through frictional losses, the ideal pressure drop allows the full flow to pass through the body without excessive velocity, absorbing less energy. However, some process systems, by virtue of their system requirements, may need to take a larger pressure drop through the valve.

A high pressure drop through a valve creates a number of problems, such as cavitation, flashing, choked flow, high noise levels, and vibration. Such problems present a number of immediate consequences: erosion or cavitation damage to the body and trim, malfunction or poor performance of the valve itself, wandering calibration of attached instrumentation, piping fatigue, or hearing damage to nearby workers. In these instances, valves in high-pressure-drop applications require expensive trims, more frequent maintenance, large spare-part inventories, and piping supports. Such measures drive up maintenance and engineering costs.

Although users typically concentrate on the immediate consequences of high pressure drops, the greatest threat that a high pressure drop presents is lost efficiency to the process system. Because high pressure drops absorb a great deal of energy, that energy is lost from the system. In most process systems, energy is added to the system through heat generated by a boiler or through pressure created by a pump. Both methods generate energy in the system, and as more energy is absorbed by the system—including that energy lost by valves with high pressure drops—larger boilers or pumps must be used. Consequently, if the system is designed with few valves with high pressure drops, the system is more efficient and smaller boilers or pumps can be used.

9.2 Cavitation

9.2.1 Introduction to Cavitation

Cavitation is a phenomenon that occurs only in liquid services. It was first discovered as a problem in the early 1900s, when naval engineers noticed that high-speed boat propellers generated vapor bubbles. These bubbles seemed to lessen the speed of the ship, as well as cause physical deterioration to the propeller.

Whenever the atmospheric pressure is equal to the vapor pressure of a liquid, vapor bubbles are created. This is evident when a liquid is heated, and the vapor pressure rises to where it equals the pressure of the atmosphere. At this point, bubbling occurs. This same phenomenon can also occur by decreasing the atmospheric pressure to equal the vapor pressure of the liquid. In liquid process applications, when the fluid accelerates to pass through the narrow restriction at the vena contracta, the pressure may drop below the vapor pressure of the fluid. This causes vapor bubbles to form. As the flow continues past the vena contracta, the velocity decreases as the flow area expands and pressure builds again. The resulting pressure recovery increases the pressure of the fluid above the vapor pressure. This phenomenon is described in Fig. 9.4.

As a vapor bubble is formed in the vena contracta, it travels downstream until the pressure recovery causes the bubble to implode. This two-step process—the formation of the bubble in the vena contracta and its subsequent implosion downstream—is called *cavitation*. In simple terms, cavitation is a phase that is characterized by a liquid–vapor–liquid process, all contained within a small area of the valve and within microseconds. Minor cavitation damage may be considered normal

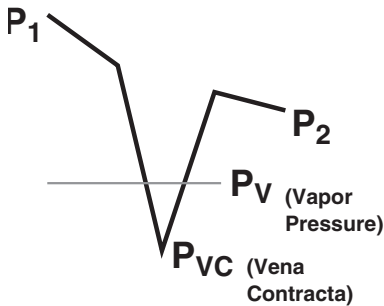


Figure 9.4 Flow curve showing pressure drop falling below the vapor pressure, which results in cavitation. (Courtesy of Valtek International)

for some applications, which can be dealt with during routine maintenance. If unnoticed or unattended, severe cavitation can limit the life expectancy of the valve. It can also create excessive seat leakage, distort flow characteristics, or cause the eventual failure of the pressure vessels (valve body, piping, etc.). In some severe high-pressure-drop applications, valve parts can be destroyed within minutes by cavitation.

In general, five conditions must be present to produce cavitation. First, the fluid must remain a liquid both upstream and downstream from the valve. Second, the liquid must not be at a saturated state when it enters the valve or the pressure drop will create a residual vapor downstream from the valve. Third, the pressure drop at the vena contracta must drop below the vapor pressure of the process fluid. Fourth, the outlet pressure must recover at a level above the vapor pressure of the liquid. Fifth, the liquid must contain some entrained gases or impurities, which act as a “host” for the formation of the vapor bubble. This host is sometimes called the *nuclei*. The nuclei are contained in the process fluid as either microscopic particulates or dissolved gases. Since most process fluids contain either particulates or dissolved gases, the chances of forming vapor bubbles are very likely. In theory, if the liquid was completely nuclei-free, some experts believe that cavitation would not occur; however, this would be nearly impossible, especially considering the effects of thermodynamics.

The creation and implosion of the cavitation bubble involve five stages: First, the liquid’s pressure drops below the vapor pressure as velocity increases through the valve’s restriction. Second, the liquid expands into vapor around a nuclei host, which is either a particulate

or an entrained gas. Third, the bubble grows until the flow moves away from the vena contracta and the increasing pressure recovery inhibits the growth of the bubble. Fourth, as the flow moves away from the vena contracta, the area expands—slowing velocity and increasing pressure. This increased pressure collapses or implodes the bubble vapor back to a liquid. Fifth, if the bubble is near a valve surface, the force of the implosion is directed toward the surface wall, causing material fatigue.

The bubbles created by cavitation are much smaller and more powerful than bubbles caused by normal boiling. This release of energy by the imploding bubbles can easily be heard as noise in the valve or in the downstream piping. The noise generated in the early stages of cavitation is described as a popping or cracking noise, while extensive cavitation produces a steady hiss or sizzling noise. Some describe the noise as gravel rolling down the piping. Noise is normally complemented by excessive vibration, which can cause metal or piping fatigue or miscalibration or malfunctioning of sensitive instrumentation. In some cases, the vibration can be minimized by anchoring the valve or piping securely to floors, walls, etc.

The most permanent damage caused by cavitation is the deterioration of the interior of the valve created by the imploding bubbles. As the bubbles expand in the vena contracta, they move into the downstream portion of the valve and then implode as the pressure recovery occurs. If the bubbles are near a metal surface, such as a body wall, they have a tendency to release the implosion energy toward the wall. This phenomenon occurs when unequal pressures are exerted upon the bubble. Since the fluid pressure is less on the side of the cavitation bubble closest to a nearby object, the energy of the implosion is channeled toward that surface (Fig. 9.5). This principle is identical to the implosion of a depth charge in antisubmarine warfare.

With cavitation, the real damage occurs in the second half of the process, when the bubbles implode. This energy burst toward the metal surface can tear away minute pieces of metal, especially if the pressure intensity reaches or surpasses the tensile strength of the valve material. These shock waves have been reported to be as high as 100,000 psi (6900 bar). This initial destruction is magnified since the drag in torn metal surface attracts and holds other imploding bubbles, causing even more cavitation damage. Valve parts damaged by cavitation have a pitted appearance or feel like a sandblasted surface (Fig. 9.6). The appearance of cavitation damage is far different from flashing or erosion damage, which appears smooth. Another possible long-term effect of cavitation is that it may attack a material's coating, film, or

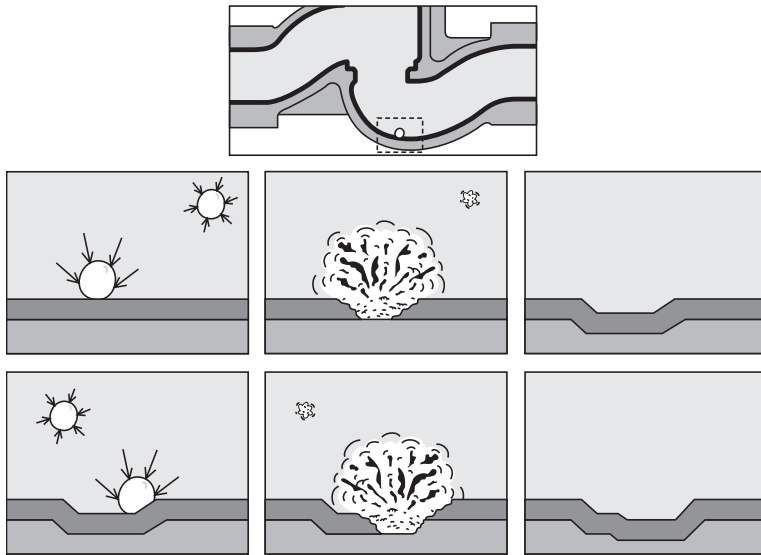


Figure 9.5 Implosion of cavitation bubbles by a valve-body wall.
(Courtesy of Valtek International)

oxide, which will open up the base material to chemical or corrosion attack.

The hardness of the metal plays a large role in how easily the metal can be torn by the cavitation bubbles. Soft materials, such as aluminum, yield easily to the forces generated by cavitation bubbles and

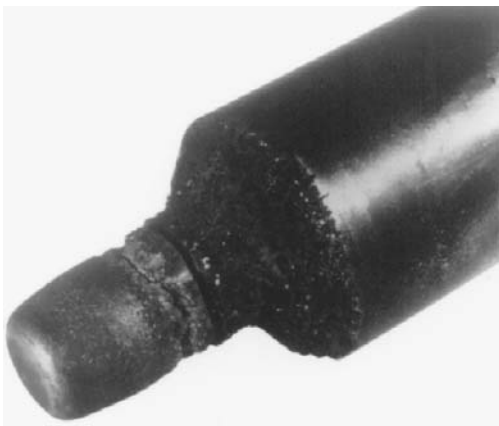


Figure 9.6 Plug damaged by cavitation.
(Courtesy of Fisher Controls International, Inc.)

tear away quickly. Hardened materials are better able to withstand the effects of cavitation, and only after a period of time will they fatigue and begin to wear. No material can resist cavitation indefinitely. Even the hardest materials will eventually wear away.

Another serious side effect of cavitation is decreased performance in the valve and reduced efficiency in the process system. When cavitation occurs, the valve's ability to convert the entire pressure drop to mass flow rate is diminished. In other words, cavitation can cause less flow through the valve, generating a smaller C_v in actual service than what was originally calculated.

Cavitation can be controlled or eliminated by one of three basic methods: first, by modifying the system; second, by making certain internal body parts out of hard or hardened materials; or third, by installing special devices in the valve that are designed to keep cavitation away from valve surfaces or prevent the formation of the cavitation itself.

9.2.2 Incipient and Choked Cavitation

As the downstream pressure is lowered, creating a larger pressure drop, the advent of cavitation is called *incipient cavitation*. When damage occurs to the vessel, that stage is known as *incipient cavitation damage*. As the flow increases, it will eventually become choked, which is called *choked cavitation*. This linear relationship is shown in Fig. 9.7, which is based on the linear relationship between the flow rate Q and

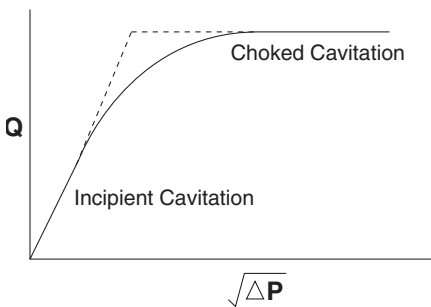


Figure 9.7 Fluid plot of flow vs. $\sqrt{\Delta P}$ and points of incipient and choked cavitation. (Courtesy of Valtek International)

the square root of the pressure drop $\sqrt{\Delta P}$. The constant of proportionality of this relationship is based upon the equation

$$Q = C_v \sqrt{\frac{\Delta P}{G_s}}$$

where Q = flow rate

C_v = flow coefficient

ΔP = pressure drop

G_s = specific gravity

9.2.3 Cavitation Indices

Over the years, cavitation experts have developed a number of cavitation indices to predict the possibility of cavitation in process equipment, including valves. The ability to predict cavitation is critical to the design and application of the valve. For example, if cavitation exists, the valve can be fitted with special trim to minimize the effects or eliminate cavitation altogether. Certain parts, such as the plug or seat, can be made from hard or hardened materials, or the process system can be changed to minimize the pressure drop through the valve so that cavitation does not form.

For many years, the valve industry used the *flow curve cavitation index* K_C , which shows the effect of cavitation on the linear relationship between the flow rate and the square root of pressure drop. The index K_C is still in use today with some manufacturers and is occasionally used in calculations as

$$K_C = \frac{P_1 - P_2}{P_1 - P_v} = \frac{\Delta P}{P_1 - P_v}$$

where K_C = cavitation index

P_1 = valve inlet pressure

P_2 = valve outlet pressure

P_v = vapor pressure of liquid (at valve inlet and vena contracta)

The cavitation index assumes that a valve may function without cavitation at any pressure drop less than the pressure drop calculated with the index K_C . The basic problem associated with the cavitation index K_C is that it does not take into consideration any prechoked cavitation conditions, which may be just as damaging to the valve. Table 9.1 provides several common K_C values for a number of valve styles.

Table 9.1 Typical K_c Values†

Valve Style	K_c
Butterfly	$0.50 K_M^*$
Ball	$0.67 K_M$
Rotary Plug	K_M
Globe with Hardened Trim (Cage Characterized)	K_M
Globe (Plug Characterized)	0.85
Globe with special trim	1.0

†Data courtesy of Fisher Controls International, Inc.

* K_M is equal to F_L^2 (valve recovery coefficient).

A more useful cavitation index for valves is σ , which was approved in 1995 by the Instrument Society of America. In general terms, σ is a ratio of forces that resist cavitation to forces that promote cavitation and is written as

$$\sigma = \frac{P_2 - P_V}{P_1 - P_2}$$

where σ = cavitation index

P_1 = upstream pressure (measured one pipe diameter upstream from the valve)

P_2 = downstream pressure (measured five pipe diameters downstream from the valve)

P_V = liquid vapor pressure (at flowing temperature)

As σ becomes larger, less cavitation damage is occurring inside the valve. Inversely, as σ becomes smaller, cavitation damage is increasing. If σ is at zero or is a negative number, flashing is occurring. σ is expressed in two forms: *Incipient* σ is the value that indicates when cavitation is beginning. *Choked* σ is the value that indicates when

Table 9.2 Typical σ Values^{†,‡}

Valve Style	Flow Direction	Trim Size	Incipient σ	Choked σ
Globe	over the plug	full area	0.73	0.38
	over the plug	reduced	0.93	0.56
	under the plug	full/reduced	0.52	0.52
Butterfly	60° open	full	1.40	0.73
	90° open	full	3.16	2.19
Ball	60° open	full	1.40	0.64
	90° open	full	5.20	2.19
Globe with special trim	under the plug	full/reduced	0.30 to 0.001	*

[†]Data courtesy of Valtek International.

[‡]Note: For estimation only; sigmas may vary according to particular valve design.

*Choking will not occur when properly applied.

choked flow or full cavitation is occurring. If the calculated σ falls between the incipient σ and choked σ values, some measures should be taken (using special trim, hard materials, or process changes) to avoid cavitation damage in the valve. Both incipient σ and choked σ values are determined through laboratory and field testing by the valve manufacturer. Examples of typical σ values for a given valve style are shown in Table 9.2.

9.2.4 σ Example A

To show an application of incipient σ and choked σ , the following example is used:

Fluid	Water
Temperature	80°F
Vapor pressure P_v	0.5 psia
Upstream pressure P_1	200 psia
Downstream pressure P_2	55 psia
Valve type	Single-seated globe valve, 100 percent open, flow-over-the-plug

The value for σ is

$$\sigma = \frac{P_2 - P_v}{P_1 - P_2} = \frac{55 - 0.5}{200 - 55} = 0.38$$

Referring to Table 9.2, incipient σ begins at $\sigma = 0.73$ (for a single-seated globe valve that is at 100 percent open with flow under the plug) and the choked σ occurs at $\sigma = 0.38$. In this example severe cavitation damage is occurring and the valve is choked and cannot increase flow any further.

9.2.5 σ Example B

Using the same valve in example A, new service conditions are applied to illustrate a cavitating, but nonchoking, situation:

Fluid	Water
Temperature	80°F
Vapor pressure P_v	0.5 psia
Upstream pressure P_1	500 psia
Downstream pressure P_2	200 psia
Valve type	Single-seated globe valve, 100 percent open, flow-over-the-plug

Using the σ index equation for these operating conditions, we find that σ is significantly higher:

$$\sigma = \frac{200 - 0.5}{500 - 200} = 0.67$$

This σ value is above the choked σ value for this valve (which is $\sigma = 0.38$) and indicates that the valve is not experiencing choked flow. However, this value is below the incipient σ value, which indicates that the valve is experiencing cavitation and damage may be occurring in the valve.

9.2.6 System Modifications to Prevent Cavitation

To eliminate the formation of cavitation, the answer lies in reducing the pressure from the upstream side to the downstream side, prevent-

ing the pressure at the vena contracta from falling below the vapor pressure. When this reduction is made, vapor bubbles are not formed and cavitation is avoided. This normally requires special trim or modifications of the system to provide a series of smaller pressure drops that result in the required downstream pressure. By taking a series of pressure drops, rather than one large drop, the service can be modified so that the pressure will not fall below the vapor pressure (Fig. 9.8).

In some cases, the process system and related service conditions, or the process equipment used in the system, can be modified to minimize the effects of cavitation. Even the type of valve or number of valves used in one system can modify cavitation effects. One system solution is the injection of air into the system. At first this may appear to worsen a bad situation as the addition of air will provide additional nuclei that can play host to vapor bubbles and increase the damage. However, cavitation studies have shown that at a certain point, additional air content to the process stream disrupts the explosive force of the imploding bubbles and can reduce the overall damage. This solution works best with large valves dumping into tanks or when large particulates in the flow stream prevent the use of cavitation-control trim, anticavitation trims, or downstream devices.

The intensity of cavitation can be modified by varying the downstream pressure, if possible. Increasing the downstream pressure may decrease the pressure drop sufficiently to avoid the pressure falling below the vapor pressure, although this will decrease the process flow capacity. Lowering the downstream pressure may not seem to be an option, since a greater pressure drop would create even more vapor

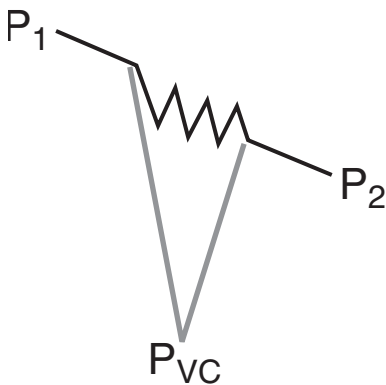


Figure 9.8 Flow curve showing gradual pressure reduction without dropping below vapor pressure. (Courtesy of Valtek International)

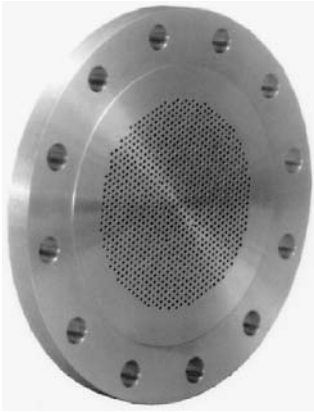


Figure 9.9 Back-pressure device used with globe valves. (Courtesy of Fisher Controls International, Inc.)

bubbles. However, the increased pressure differential provides less cavitation intensity.

A *downstream back-pressure device* is a device that is installed between the valve and the downstream piping to lower the pressure drop taken by the valve while increasing the resistance of the downstream and the downstream pressure. Although a wide variety of back-pressure devices are available today, a typical device is shown in Fig. 9.9. Because back-pressure devices may limit the flow capacity of the valve, a larger valve or different trim reduction may be required. The device must be examined periodically to ensure that it is not wearing out through erosion or minimal cavitation. A worn back-pressure device will ultimately decrease the downstream pressure, increase the pressure drop, and create cavitation. In addition, the user must be careful to use the back-pressure device within the limits of the flow range; otherwise cavitation can occur after the device in the downstream piping. A back-pressure device is commonly used with a rotary valve (Fig. 9.10), which cannot be designed to include an internal anti-cavitation device because of design limitations. Not only does the device perform the function of raising the downstream pressure, but it also controls existing cavitation by allowing it to occur in the small tubes where cavitation intensity is lower and can be absorbed by the tubes themselves. One caution applies when using a downstream cavitation-control device with a rotary valve: As the rotary valve begins to open (less than 30° open), the most severe cavitation intensity may occur in the outlet half of the body before it reaches the downstream device, causing serious damage between the valve and the device.



Figure 9.10 Back-pressure device used with rotary valves (Courtesy of Fisher Controls International, Inc.)

Some valve designs can be used to minimize cavitation damage. For example, while a globe-style linear valve exposes the bottom of the valve body to the cavitation, an angle-style linear valve may experience less damage since the flow continues straight down from the vena contracta and is directed into the center of the piping—no valve or piping surface is directly bombarded with vapor bubbles.

As a general rule, the face-to-face dimension of rotary valves—such as butterfly, eccentric plug, and ball valves—is far less than comparably sized globe valves. Therefore, the vena contracta generated by a rotary valve is most likely to occur not in the valve itself, but in the downstream piping. In this case, cavitation might be allowed and a segment of the downstream piping routinely replaced as part of periodic maintenance. Another option is to install two or three valves in lieu of one valve, allowing the pressure drop to be taken over more than one restriction and preventing a large pressure drop from falling below the vapor pressure. This option is more expensive in terms of additional valves, but may still be less expensive than obtaining a specially engineered valve. This solution has one disadvantage, however, that may occur when the first valve opens against a high upstream pressure. For a very short time, the first valve will take the entire pressure drop until the flow reaches the second valve. This may result in

cavitation damage to the first valve in some unusual cases. In such an application, installing anticavitation trim in the valve may be a better option.

9.2.7 Materials of Construction

Cavitation easily attacks softer materials, which have a lower tensile strength than harder materials. One of the most common methods of dealing with cavitation is to make the valve out of hard or hardened materials (those materials exceeding a Rockwell hardness of 40). Generally, materials such as chrome–molybdenum and steel alloys (ASTM SA-217 Grade WC9 and C5) are used for the body, while solid alloy hard-facing, a solid alloy overlay with 316 stainless steel or 416 stainless steel, is used on trim parts.

One advantage to using angle-style valves in cavitating service is that one of three options—a hardened seat ring, an extended *Venturi seat ring*, or body liner—can be installed in the downstream portion of the valve. This part can then be replaced periodically after cavitation damage compromises the part. These liners can be made from Alloy 6 or 17-4ph stainless steels.

Because nonmetallic materials, such as PTFE liners or bodies made from plastic, have lower yield values than metal, they are more prone to cavitation damage and are not recommended for cavitating services.

9.2.8 Cavitation-Control Devices

Some valves can be equipped with special trims that will direct the cavitating flow, along with vapor bubbles, away from critical metal surfaces. Since cavitation-control trims are not as highly engineered as trims designed to prevent cavitation, they generally cost less and are simpler in concept.

The design shown in Fig. 9.11 illustrates how this principle works. In flow-over-the-plug applications, a special retainer with specially designed holes is placed inside the valve. As the close-fitting plug lifts out of the seat, the holes in the special retainer are exposed and allow the flow to pass through the seat. In this case, the holes in the retainer are the restrictions and cavitation occurs at that region. Because the holes are directly opposite each other, the cavitating flow from one hole impinges on the opposite hole's flow, thus keeping the cavitation in the center of the retainer. At this point, the only metal surface affected by the cavitation is the bottom of the plug, which can be made from hardened material. Since the middle of the bottom of the plug is flat



Figure 9.11 Laboratory experiment showing diversion of cavitation away from boundary surfaces using cavitation control trim. (Courtesy of Valtek International)

and not necessary for shutoff, it can be sacrificed over a period of time. Only when the deterioration reaches the plug's seating surface will the plug need replacement.

Such cavitation-control designs can be engineered with a wide range of C_v s and in either linear or equal-percentage flow characteristics. Because flow must always be over the plug, pressure-balanced trim is necessary in fail-open applications to prevent instability near the seat.

9.2.9 Cavitation-Elimination Devices

Some valves are designed to prevent the formation of cavitation altogether. Although it is a more expensive option, in some applications anticavitation design features are the only choice. Globe-style valves can be designed with special retainers or cages, which use either (or a combination of) a tortuous path, pressure-drop staging, and/or expanded flow areas to decrease the pressure drop through the valve and to prevent cavitation.

A *tortuous-path device* uses a series of holes and/or channels to increase the flow resistance through the trim (Fig. 9.12). This decreases the overall velocity through the valve, thereby reducing the pressure recovery. In addition, a tortuous path creates pockets of high and low

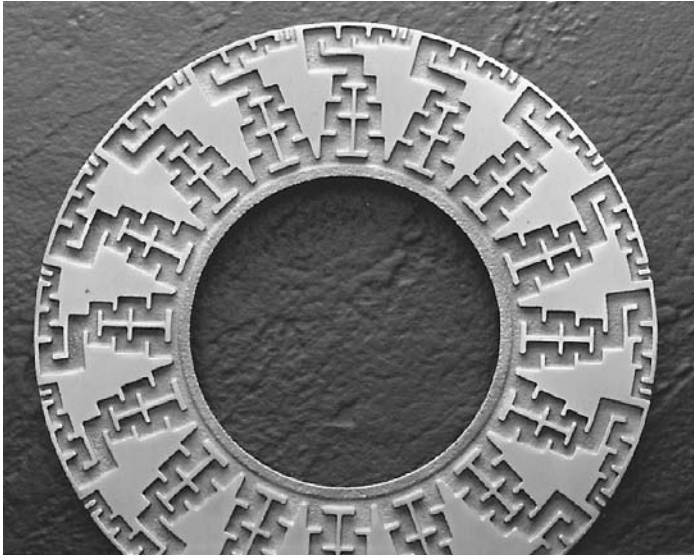


Figure 9.12 Tortuous-path trim for velocity reduction. (Courtesy of Control Components Inc., an IMI company)

pressures as the flow moves through the trim, creating substantial frictional losses. To illustrate the effect of frictional losses in this trim, the losses associated with a single 90° piping elbow are equal to 60 ft of straight pipe. The typical tortuous path uses a series of right-angle turns—similar in principle to a 90° elbow—to create frictional losses and lower velocities. Each turn reduces the velocity by one velocity head ($V_H = \rho V^2/2$). This velocity reduction can be calculated by changing the velocity equation as follows:

$$V = \sqrt{2SGV_H} \quad \text{to} \quad V = \sqrt{\frac{2SG V_H}{N}}$$

where V = required velocity (below sonic or generally below 300 ft/s)

SG = specific gravity

V_H = velocity head

N = number of turns (in series) in each passageway

Determining the number of turns is critical in the design of tortuous-path designs, since they determine the overall velocity-head loss, as well as the diameter of the stack.

Another method of decreasing the pressure drop is by *staged pressure reduction*, in which several smaller restrictions are taken through a trim

rather than one large restriction. In effect, this creates a number of small pressure drops in lieu of one large pressure drop (refer again to Fig. 9.8). As the flow moves through the trim, it reaches the first restriction or stage, absorbing a certain amount of energy and taking a small pressure drop. As the flow continues, it provides a lower inlet pressure to the next stage where another pressure drop is taken, and so forth. The net result is that the entire pressure drop is taken over a series of small pressure drops without falling below the vapor pressure at the vena contracta, yet the overall pressure drop remains unchanged. In some cases, for whatever reason, systems pressures may change. This change may exceed the operating parameters of the valve and create cavitation in the valve, even if a staged-pressure-reduction trim is used. In this case, although cavitation is occurring, the anticavitation trim will continue to modify the pressure differential and the cavitation will not be as severe.

Related to the staged-pressure-drop concept is the *expanded flow-area* concept, in which the flow continues through several restrictions in the trim, the flow area is increased at each stage (Fig. 9.13). With compressible fluids, as dictated by the law of conservation of mass flow, the flow area must increase as the fluid pressure and density are reduced. In this concept, the largest portion of the pressure drop is taken at the first restriction, and then succeeding smaller portions of the pressure drop are taken over the following restrictions. When the flow reaches the last restriction, a minimal pressure drop is taken and

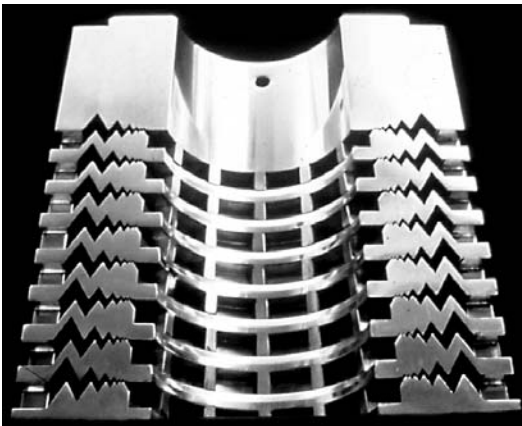


Figure 9.13 Expanding tooth trim for staged pressure reduction. (Courtesy of Valtek International)

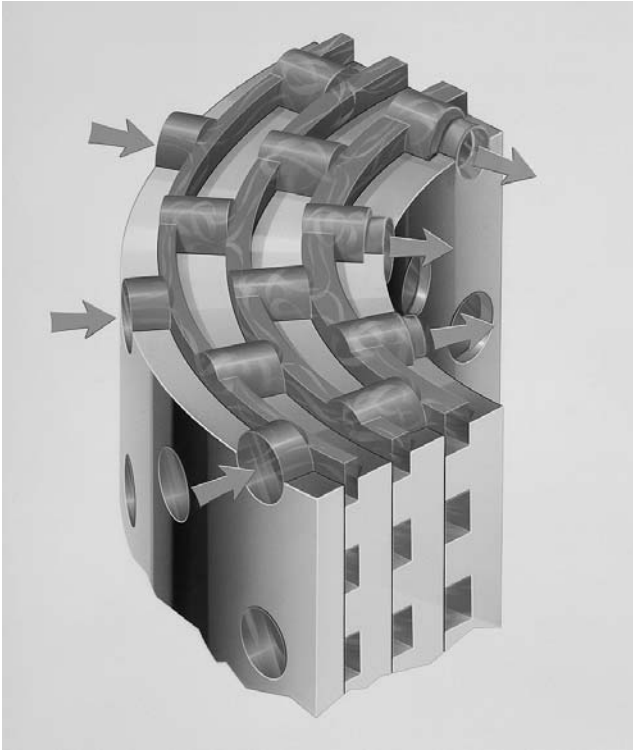


Figure 9.14 Anticavitation trim with multiple pressure-reduction mechanisms. (Courtesy of Valtek International)

the pressure recovery at that point is significantly decreased, preventing cavitation from occurring.

Valve manufacturers have developed a variety of sophisticated trims that use one or a combination of these concepts (tortuous path, staged pressure reduction, and expanded flow areas). For example, Fig. 9.14 shows a flow-over-the-plug trim that directs the flow through a series of close-fitting cylinders with each cylinder acting as a stage. The flow must follow a tortuous path as it travels through a series of 90° angles via the narrow channels and drilled holes, increasing the frictional losses. Pressure reduction is staged through the number of cylinders, allowing the pressure to remain above the vapor pressure. In addition, the channels become progressively deeper and the number of holes increase with each stage, providing expanded flow areas.

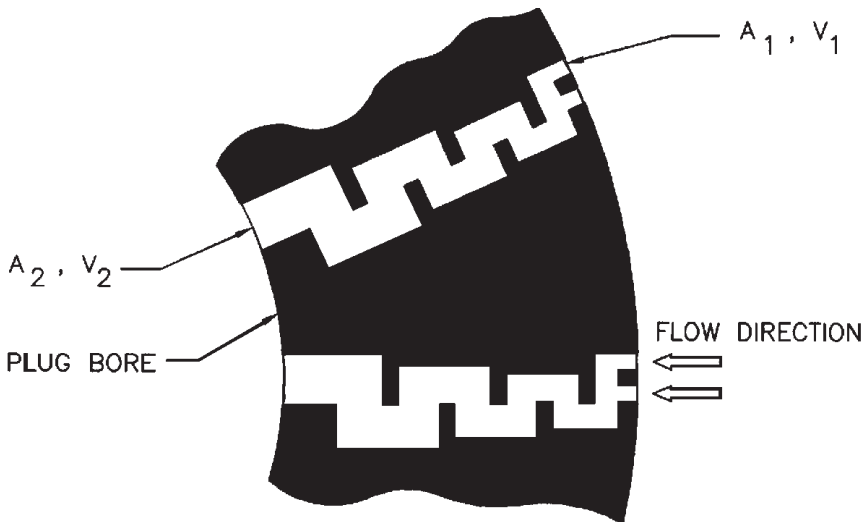


Figure 9.15 Expanding flow area of tortuous-path trim. (Courtesy of Control Components Inc., an IMI company)

Another common design that uses these principles is the *expanding tortuous-path trim*. In addition to the velocity control through the right-hand turns, the tortuous pathways can be enlarged, allowing for expanded flow areas (Fig. 9.15). The tortuous path can follow either a horizontal direction with etched disks (Fig. 9.16) or disks made from a punched plate (Fig. 9.17).

Most anticavitation trims follow a linear characteristic, although some designs allow for an equal-percentage characteristic. When the disks or flow areas of the trim are identical throughout the entire stack, the trim follows a linear characteristic. An equal-percentage characteristic is generally obtained by using different disks or passage-ways that increase the flow as the stroke continues. Another method of modifying an anticavitation linear characteristic is by using a shaped cam in the actuator positioner.

9.2.10 Anticavitation-Trim Sizing

Although methods of sizing a valve with anticavitation trim vary according to different valve manufacturers, the following procedure utilizes σ values and provides a general idea of the steps involved. The first step is to calculate the required C_v for the given application (see

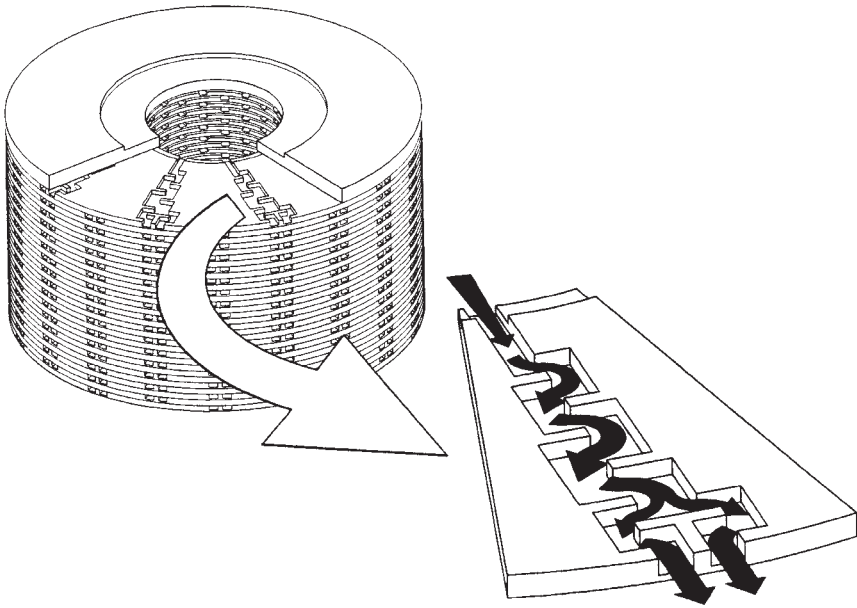


Figure 9.16 Etched tortuous-path trim for horizontal flow. (Courtesy of Control Components Inc., an IMI company)

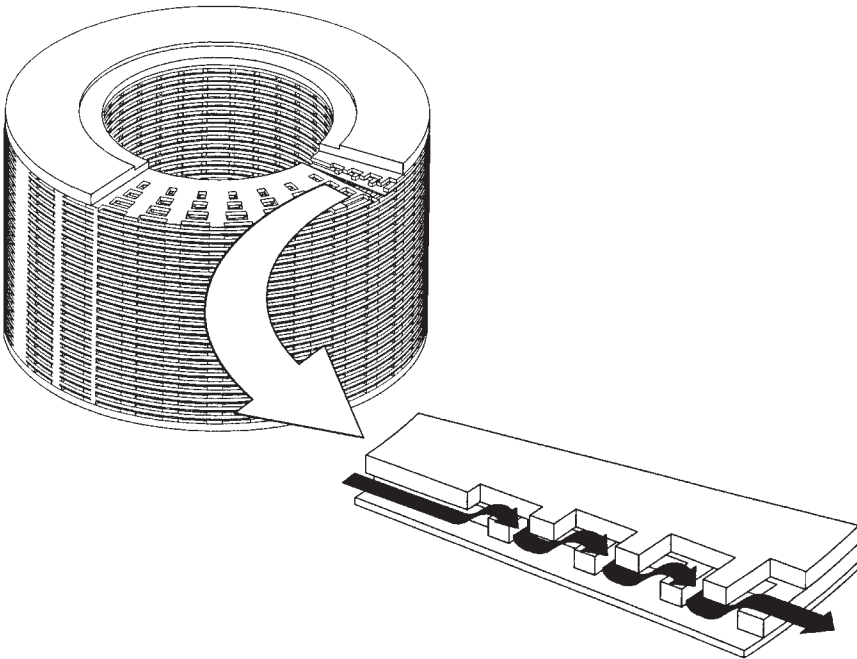


Figure 9.17 Punched tortuous-path trim for vertical and horizontal flows. (Courtesy of Control Components Inc., an IMI company)

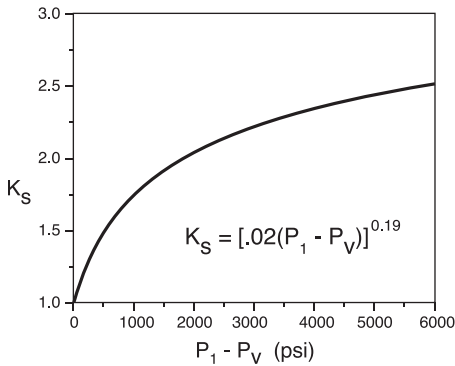


Figure 9.18 Pressure scale for factor K_S (water service), where $K_S = [0.02 (P_1 - P_V)]^{0.19}$. (Courtesy of Valtek International)

Chap. 7). Choked-flow conditions should be considered, and the F_L factor should be adjusted to compensate for the use of the anticavitation trim. In this case, the required C_v for a valve with anticavitation trim will most likely be smaller than a conventional valve. Using the o formula, the operating o can be calculated from the flow conditions for the required C_v . The difference between the upstream pressure and the vapor pressure should be calculated ($P_1 - P_V$). Following this calculation, the K_S factor can be determined by referring to Fig. 9.18 or by using the following calculation:

$$K_S = [0.02(P_1 - P_V)]^{0.19}$$

The service σ can now be calculated for each pressure:

$$\sigma_{\text{service}} = \frac{\sigma}{K_S}$$

The manufacturer of the anticavitation trim provides tables (Table 9.3) that provide C_v and σ values. If the requirement of the service σ is less than the minimum requirement of the calculated σ , then a larger valve with more stages will mostly likely be needed. The velocity of the flow should also be considered to ensure that it does not approach the maximum velocity capacity of the trim.

Table 9.3 Cavitation Trim Sizing Table*

Body Size	Trim No. (Seat Dia.)	Stages	Stroke	C _v	σ _{min.}	Bore Area
1½	1.38	2	1.50	17	.170	1.77
	1.25	3	1.50	11	.070	1.48
	1.12	4	1.50	6	.020	.99
2	1.38	2	1.50	18	.170	1.77
	1.25	3	1.50	12	.070	1.49
	1.12	4	1.50	7	.020	.99
3	2.50	2	2.50	50	.200	5.41
	2.38	3	2.50	34	.080	4.91
	2.00	4	2.50	20	.025	3.55
	1.62	5	2.50	12	.007	2.41
	1.25	6	2.50	7	.002	1.49
4	3.50	2	3.00	85	.200	10.3
	3.12	3	3.00	54	.080	8.3
	2.75	4	3.00	33	.025	6.51
	2.38	5	3.00	21	.007	4.91
	1.88	6	3.00	12	.002	3.14
6	5.25	2	4.00	175	.200	22.7
	4.75	3	4.00	105	.080	18.7
	4.25	4	4.00	65	.025	15.1
	3.50	5	4.00	40	.007	10.3
	3.00	6	4.00	25	.002	7.67
8	6.50	2	6.00	320	.200	34.5
	6.00	3	6.00	200	.080	29.5
	5.50	4	6.00	130	.025	24.8
	5.00	5	6.00	85	.007	20.6
	4.50	6	6.00	55	.002	16.8
10	8.75	2	7.50	530	.230	61.9
	8.38	3	7.50	350	.090	56.7
	7.88	4	7.50	230	.028	50.3
	7.38	5	7.50	155	.008	44.2
	6.88	6	7.50	105	.002	38.5
12	9.75	2	8.00	640	.230	76.6
	9.00	3	8.00	400	.090	65.4
	8.38	4	8.00	260	.028	56.7
	7.88	5	8.00	180	.008	50.3
	7.38	6	8.00	125	.002	44.2
14	11.00	2	8.00	720	.240	97.2
	10.25	3	8.00	460	.095	84.5
	9.50	4	8.00	300	.030	72.8
	8.75	5	8.00	200	.008	61.9
	8.00	6	8.00	135	.002	51.8

*Courtesy of Valtek International.

9.2.11 Anticavitation-Trim Sizing Example

The following service conditions apply to this example:

Fluid	Water
Maximum flow	515 gal/min
Inlet temperature	208°F
Inlet pressure	287 psia
Outlet pressure	24 psia
Vapor pressure	13.57
Specific gravity	0.92

Using the flow capacity calculations in Chap. 7, the required C_v is calculated at $C_v = 30$. σ is calculated as follows:

$$\sigma = \frac{P_2 - P_v}{P_1 - P_2} = \frac{287 - 13.57}{287 - 24} = 0.04$$

Using the K_s chart (Fig. 9.18), the K_s is 1.38. Knowing K_s , the allowable σ can be calculated as follows:

$$\sigma_{\text{service}} = \frac{0.04}{1.38} = 0.029$$

Using an anticavitation trim table from the manufacturer (Table 9.3) for an application requiring a C_v of 33 and a σ value of 0.029, the required valve would be a 4-in (DN 100) valve with a four-stage anti-cavitation trim.

9.2.12 Other Cavitation-Control Solutions

A number of other solutions to cavitation control or elimination exist, such as characterized cages or separation of the valve's seat and the throttling mechanism. In applications where the pressure drop decreases as the plug travel and flow rate increase, a characterizable cage can be used. For example, a typical characterizable cage would have two stages of pressure reduction, the middle portion would have one stage of pressure reduction, and the top portion would have straight-through flow. With this design, cavitation control is provided at the early stages of plug travel, when it is needed most. As the travel continues and the

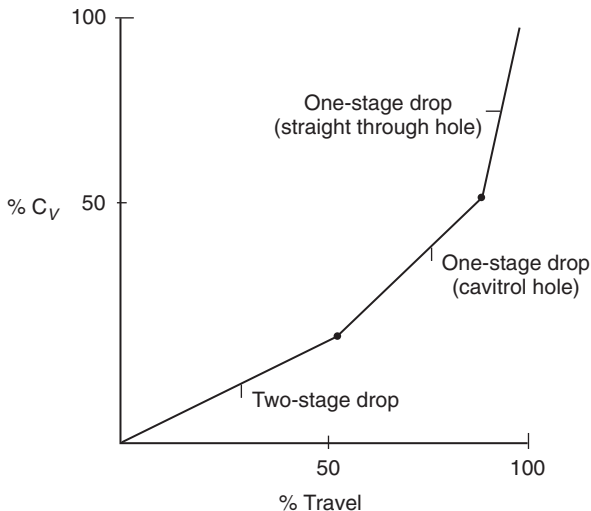


Figure 9.19 Flow curve showing effects of two-stage characterized cage. (Courtesy of Fisher Controls International, Inc.)

pressure drop and chance of cavitation decrease, the mechanism allows greater flow with less restriction. Figure 9.19 plots how a characterizable cage works in relation with flow versus travel.

In most flow-over-the-plug applications, the pressure reduction device is located above the seat in the body gallery. However, in some cavitating applications where tight shutoff is important, the body can be designed with the seat separate from the pressure-reduction mechanism. As shown in Fig. 9.20, the seat is located above the anticavitation trim, which is contained in the downstream portion of the valve. The trim area above the seat is designed to take a large flow, hence a lower pressure drop. This design keeps the velocities at a minimum through the seat, which improves the stability of the valve plug close to the seat and makes for easier shutoff.

Traditionally, anticavitation trim is associated with linear throttling valves, although some designs exist for quarter-turn valves. For example, a plug valve can utilize a special plug (Fig. 9.21) to take an additional stage of pressure reduction for those applications where the pressure drop falls just below the vapor pressure. As the plug closes, the grid turns into the flow, taking a pressure drop and preventing cavitation from forming. The grid prevents severe cavitation from forming and channels remaining cavitation away from metal bound-

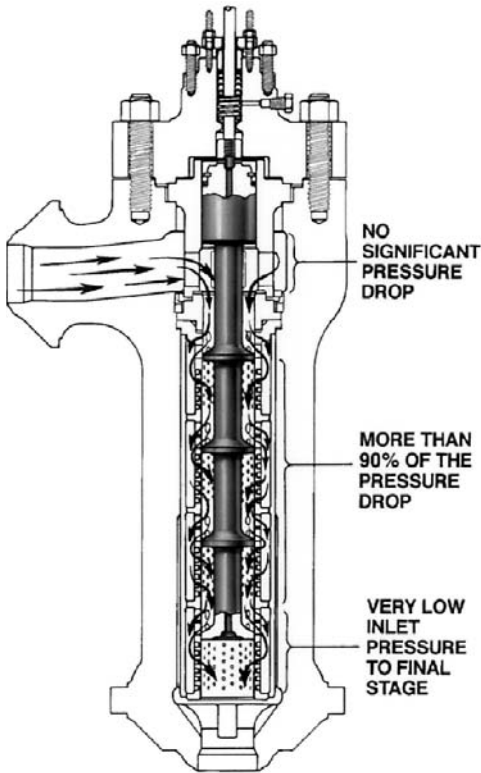


Figure 9.20 Anticavitation trim located downstream from the seating surface. (Courtesy of Fisher Controls International, Inc.)

aries. This design also allows large particulates to bounce off the grid and be flushed downstream.

9.3 Flashing

9.3.1 Introduction to Flashing

In liquid applications, when the downstream pressure is equal to or less than the vapor pressure, the vapor bubbles generated at the vena contracta stay intact and do not collapse. This happens because the pressure recovery is high enough for this to happen. As shown in Fig. 9.22, this phenomenon is known as *flashing*. When flashing occurs, the fluid downstream is a mixture of vapor and liquid moving at very



Figure 9.21 Anticavitation plug for quarter-turn plug valves. (Courtesy of The Duriron Company, Valve Division)

high velocities, resulting in erosion in the valve and in the downstream piping (Fig. 9.23).

9.3.2 Controlling Flashing

Unfortunately, eliminating flashing completely involves modifying the system itself, in particular the downstream pressure or the vapor pressure. However, not all systems are easily modified and this may not be an option. The location of the valve may be considered—especially if the valve empties the downstream flashing flow into a large vessel, tank, or condenser. Placing the valve closer to the larger vessel will allow the flow to impinge into the larger volume of the vessel and away from any critical surfaces. When flashing occurs, no solution can be designed into the valve, such as is the case with anticavitation or cavitation-control trim, except to offer hardened trim materials.

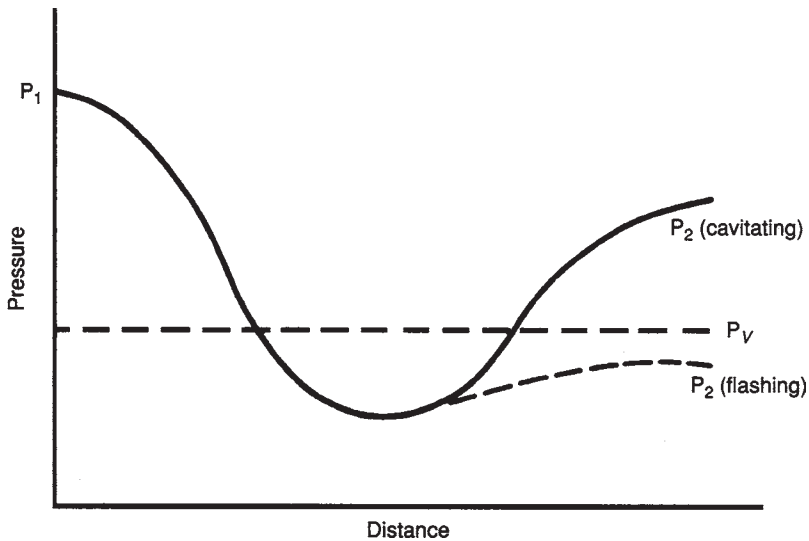


Figure 9.22 Pressure curve showing outlet pressure below the vapor pressure, resulting in flashing. (Courtesy of Fisher Controls International, Inc.)

9.4 Choked Flow

9.4.1 Introduction to Choked Flow

Choked flow occurs in gases and vapors when the velocity of a process fluid achieves sonic speeds in the valve or the downstream piping. As the fluid in the valve reaches the valve restriction, the pressure



Figure 9.23 Plug damaged by flashing. (Courtesy of Fisher Controls International, Inc.)

decreases and the specific volume increases until sonic velocities are achieved. When choked flow occurs, the flow rate is limited to the amount of flow that can pass through the valve at that point and cannot be increased unless the service conditions are changed.

In liquid applications, the presence of vapor bubbles caused by cavitation or flashing significantly increases the specific volume of the fluid. This increase rises at a faster rate than that generated by the pressure differential. In liquid choked flow conditions, if upstream pressure remains constant, decreasing the downstream pressure will not increase the flow rate. In gas applications, the velocity at any portion of the valve or downstream piping is limited to Mach 1 (sonic speed). Hence, the gaseous flow rate is limited to the flow that is achieved at sonic velocity in the valve's trim or the downstream piping.

As noted in Sec. 7.2, choked flow must be considered when sizing a valve, especially when considering $\Delta P_{\text{allowable}}$ and the valve recovery coefficient K_M .

9.5 High Velocities

9.5.1 Introduction to High Velocities

In general, large pressure differentials create high velocities through a valve and in downstream piping. This in turn creates turbulence and vibration in liquid applications and high noise levels in gas applications. The velocity is inversely related to the pressure losses and gains as the flow moves through the vena contracta (Fig. 9.24). The velocity

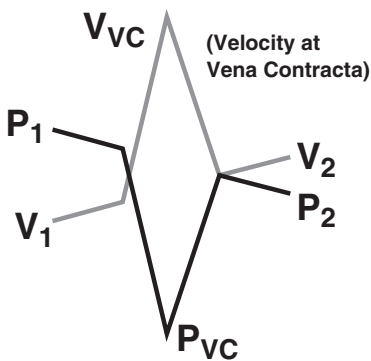


Figure 9.24 Velocity and pressure profiles as flow travels through an orifice restriction.
(Courtesy of Valtek International)

reaches its maximum peak just slightly after the vena contracta, which is when the pressure is at its lowest point.

9.5.2 Velocity Limits

The following general rules apply to velocities: Liquids should generally not exceed 50 ft/s (15.2 m/s) (or 30 ft/s or 9 m/s in cavitating services). Gases should not exceed sonic speeds (Mach 1.0). And, mixtures of gases and liquids (such as flashing applications) should not exceed 500 ft/s (152 m/s). These general rules can vary, however, according to the size of the valve. For example, smaller-sized valves can normally handle higher velocities, while larger valves only handle lower velocities.

Generally, process liquids that have temperatures close to the saturation point must keep the velocity at or under 30 ft/s (9 m/s) to avoid the fluid pressure from falling below the vapor pressure and creating cavitation. The 30-ft/s rule also applies to cavitating services, where higher velocities result in greater cavitation damage to downstream piping. Lower velocities will also reduce problems associated with flashing and erosion.

9.6 Water-Hammer Effects

9.6.1 Definition of the Water-Hammer Effect

In liquid applications, whenever flow suddenly stops, shock waves of a large magnitude are generated both upstream and downstream. This phenomena is known as the *water-hammer effect*. It is typically caused by a sudden pump shutoff or a valve slamming shut when the closure element is suddenly sucked into the seat ("bathtub stopper effect") as the valve nears shutoff. In control valves, the bathtub stopper effect is caused by a low-thrust actuator that does not have the stiffness to hold a position close to the seat. In some cases, valves with a quick-open or an installed linear flow characteristic can also cause water-hammer effects.

Although water hammer generates considerable noise, the real damage occurs through mechanical failure. Because of the drastic changes from kinetic energy to the static pipe pressure, water hammer has been known to burst piping or damage piping supports as well as damage piping connections. In valves, water hammer can create severe shock through the trim, which can cause trim, gasket, or packing failure.

9.6.2 Water-Hammer Control

With valves, the best defense against water hammer is to prevent any sudden pressure changes to the system. This may involve slowing the closure of the valve itself or providing a greater degree of stiffness as the closure element approaches the seat. To avoid pressure surges, the valve should be closed with a uniform rate of change. In some cases, when a quick-open or installed linear characteristic (which approaches the quick-open characteristic) is used, a change to an equal-percentage characteristic may be required. With control valves that must throttle close to the seat, using an exceptionally stiff actuator—such as a spring cylinder pneumatic actuator or a hydraulic actuator—or a special notch in the stroke collar of a manual quarter-turn operator will minimize or prevent the bathtub stopper effect. Adding some type of surge protection to the piping system can also reduce water hammer. This may be accomplished with a pressure-relief valve or a rubber hose containing a gas, which can be run down the length of the piping. In addition, gas may be injected into the system. Gas injection reduces the density of the fluid and provides some compressibility to handle any unexpected surge.

9.7 High Noise Levels

9.7.1 Introduction to Noise

One of the most noticeable and uncomfortable problems associated with valves is noise. To the human, not only can noise be annoying, but it can also cause permanent hearing loss and unsafe working conditions. Extensive studies have shown that human hearing is damaged by long exposures to high noise levels. Hearing damage is cumulative and irreversible and begins with the loss of high frequencies. As hearing loss continues, lower frequencies are eventually lost, which affects the ability to understand normal speech patterns. When subjected to noise at lower frequencies, the performance of human organs, such as the heart or the liver, can also be affected. In addition, noise and the accompanying vibration can affect the valve's performance and cause fatigue in the valve, piping, and nearby process equipment.

In essence, noise is generated when vibration produces wide variations in atmospheric pressure, which are then transferred to the eardrums as noise. Noise spreads at the speed of sound [which is 1100 ft/s (335 m/s) or 750 mi/h (1200 km/h)]. Noise in valves can be created in a number of different ways; however, the most common cause is

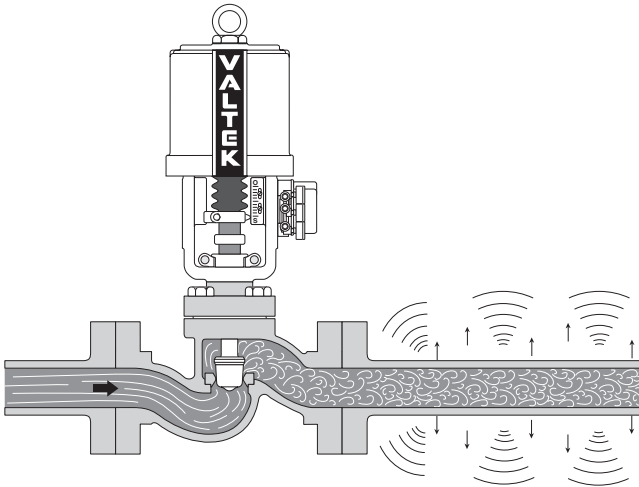


Figure 9.25 Downstream pipeline vibration caused by valve turbulence. (Courtesy of Valtek International)

turbulence generated by the geometry of the valve, which is radiated by the downstream piping (Fig. 9.25). In many cases, noise does not radiate from the valve itself, because the body itself is stiff and unyielding.

Process turbulence can create mechanical vibration of the valve or valve components. Such noise is caused by vibrations created by random pressure fluctuations within the body assembly or the fluid impinging on obstacles in the fluid stream, such as the plug, disk, or other closure element. This often causes a rattling noise, as the closure element impacts continually against its guides. Because the frequency level is less than 1500 hertz (Hz), it is normally not annoying to the hearer. However, this rattling of the stem or shaft with the guides can damage critical guiding or seating surfaces. One side benefit of a rattling noise associated with valve parts is that such secondary noise provides a warning signal that turbulence is taking place inside the valve and that corrections may be necessary before failure occurs. Vibration can also be caused by certain valve parts or accessories that resonate at their natural frequency, which is often found in lower noise levels—less than 100 dBA. This type of noise is characterized by a single tone or hum (with a frequency between 3000 and 7000 Hz). Although this noise is not an annoyance, it does produce high levels of stress in the material, which may fatigue the material of the component and cause it to weaken. Noise can also be generated by hydrody-

namic and aerodynamic fluid sound. With liquid applications, hydrodynamic noise is caused by the turbulence of the flow, cavitation, flashing, or the high velocities that occur as the flow moves through the vena contracta. Generally, however, the noise generated by the liquid flow does not occur at high levels and can be tolerated by workers. In severe cavitating or flashing applications, noise levels can reach higher levels and must be dealt with by changing the process or installing anticavitation components in the valve.

When cavitation occurs in liquid services, the noise generated by the implosion of the bubbles occurs just slightly downstream from the valve and sounds similar to rocks flowing down the pipe. Overall, this noise is simply irritating and does not reach levels that cause harm. On the other hand, aerodynamic noise is often a problem for nearby workers when dealing with gaseous services. It generates frequencies in the range between 1000 and 8000 Hz, the range that is most sensitive to the human ear. In many cases, gaseous noise levels rise above 100 dBA (decibels for human hearing) and in some extreme cases, above 150 dBA.

In general, the noise level is a function of the velocity of the flow stream. As the pressure profile indicates, when pressure drops at the vena contracta, the velocity increases proportionately. Because of the vena contracta, high noise levels can be generated as velocity increases through the restriction, even though the velocity decreases as low as Mach 0.4 as the flow reaches the downstream side of the valve.

The mechanisms used in cavitation control—tortuous paths, staged pressure drops, and expanding flow areas—can also be applied in order to lower sound levels in gas services. In addition, the mechanism in providing a flow path with sudden expansions and contractions is also used to lower aerodynamic noise.

9.7.2 Sound Pressure Level

Vibrations or atmospheric pressure changes are based upon the number of cycles per second (hertz). A young hearer has a hearing range of 20 to 20,000 cycles per second (20 kilocycles or 20 kHz). The intensity of sound that is heard by a hearer is expressed as in units as *decibels*. In order to understand decibels, the relationship of microbars to 1 Newton per square meter must be understood. One μbar is one-millionth of a normal atmospheric pressure and 10 μbar equal 1 N/m². Zero decibel (dB) is defined as 0.00002 N/m², which is considered the absolute threshold of hearing for a young hearer. Decibels are applied to three common weighted sound levels; dBA for human hearing, dBB

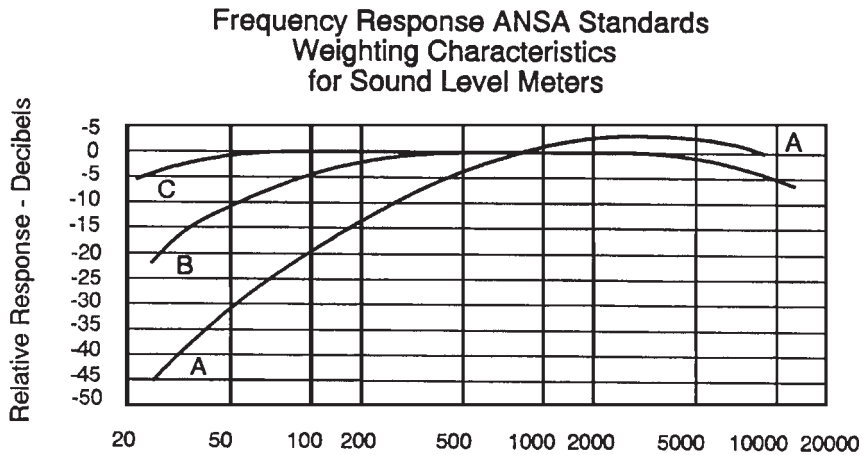


Figure 9.26 Decibel curves for A, B, and C scales. (Courtesy of Valtek International)

for an intermediate range, and dBC for equipment (Fig. 9.26). In nearly all cases, dBA is the most commonly applied sound level because it is weighted to account for the sensitivity of human hearing. With the dBA-weighted scale, the loudness of a particular noise at a certain frequency is compared to the loudness given for a 1000-Hz level. In other words, at 1000 Hz, the dBA value is zero. With the 1000-Hz scale, the sound pressure level is equal to the actual dB level. However, if a different hertz level is applied, the noise may sound less loud. For example, with 200 Hz, a sound measured at approximately 120 dB is lower in loudness (110 dB). Or in other words, the correction for dBA at 200 Hz is -10 , as shown in Fig. 9.26. Table 9.4 indicates a number of common sounds measured in dBA levels.

Valve noise is calculated as a *sound pressure level*, which is defined as

$$\text{SPL} = 20 \log_{10} \frac{P}{0.0002 \mu\text{bar}} \text{ dB}$$

where SPL = sound pressure level

P = root-mean-square sound pressure (N/m^2)

Approximately 90 dB equals one sound pressure level, and this level doubles every 6 dB. Therefore, 96 dB is two times the sound pressure level and 102 dB is four times the sound pressure level. To illustrate the magnitude of this change, the range between 80 and 120 dB is

Table 9.4 Typical dBA Sound Levels*

Sound	dBA
Threshold of hearing	0
Soft whisper at 5 feet (1.3 meters)	10
Average home residence	50
Busy highway	57
Freight train at 100 feet (25.4 meters)	67
Subway train at 20 feet (5.1 meters)	80
Textile weaving plant	83
Electric furnace	90
Pneumatic peen hammer	94
Riveting machine	100
Discomfort threshold	110
Jet take-off at 200 feet (50.8 meters)	123
50 HP siren at 100 feet (25.4 meters)	135
Pain threshold	140

**Courtesy of Valtek International.*

100 times the sound pressure level. Noise radiating from a single point decays at a rate of 6 dB for every doubling of distance. However, if the noise is radiating from a radial line source—such as noise radiating from a pipeline—the noise decays at half that rate or 3 dB for every doubling of distance. Conversely, hard surfaces close to the noise source can increase the noise by reflecting sound. A single hard surface, such as a floor, increases the noise level by 3 dBA. Two hard surfaces, such as a floor and wall, reflect an additional 6 dBA and three hard surfaces (a corner) add 9 dBA. Theoretically, if the noise was enclosed in a completely sealed room with hard surfaces, noise levels would approach infinity—although this is highly unlikely with atmospheric friction. However, the possibility exists that a loud valve installed in a small metal building could easily achieve the pain threshold of 140 dBA.

Sound pressure levels are measured by a sound-level meter, which is normally held 1 m downstream from the valve's outlet and 1 m away from the pipe itself. Because of the effect reflective surfaces can have on the sound pressure levels, the measurement must be taken in a free-field area without any reflective surfaces. In some cases, sound intensity levels may be preferred for measuring or comparing sound intensities. This is calculated as

$$\text{sound intensity level} = 10 \log_{10} \frac{\frac{P_s^2}{\rho C}}{10^{-16}} \text{ dB}$$

where P_s = amplitude of sound pressure
 ρ = density
 C = sonic velocity

In some cases, two noise sources may be occurring at the same time, which will increase the overall sound pressure level. The energy of the two sources is logarithmically combined as one noise source. Table 9.5 represents a simple method of determining the increase in noise when two noise sources are combined. After sound pressure levels are taken at each source, the difference between the two readings is used to find the correct dB factor, which is then added to the loudest noise source. As Table 9.5 shows, as the difference in the sound pressure level between two sources widens, the overall noise increase lessens. Therefore, the obvious solution is to concentrate on correcting the source with the loudest noise.

9.7.3 Turbulence

To achieve an understanding of how to decrease valve noise, the causes of turbulence must be examined. As the flow moves through the valve, the flow stream is interrupted by the valve geometry, such as the presence of a seat, disk, plug, or a sharp contour of the body. Turbulence causes pressure fluctuations in a variety of ways; however, in simple terms the pressures work against the wall of the downstream piping and cause wall fluctuations, which radiates the noise frequencies to the atmosphere. Figure 9.27 shows the pressure profile of a throttling valve as the upstream pressure is released to atmosphere. The profile shows a wide range of fluctuations in the downstream pressure that can vary by more than 15 psi (1.0 bar). As the upstream pressure decreases, the pressure drop decreases, and the variations of downstream pressure and resultant noise are less. Using the same test data, Fig. 9.28 shows a downstream test plot of the sound pressure

Table 9.5 dB Factors for Two Noise Sources†

Difference in dB between two sources	dB Factor*
0	3.01
1	2.54
2	2.12
3	1.76
4	1.46
5	1.20
6	0.97
7	0.79
8	0.64
9	0.52
10	0.42
11	0.33
12	0.27
13	0.22
14	0.17
15	0.14
16	0.11
17	0.09
18	0.07
19	0.06
20	0.05

†Data courtesy of Fisher Controls International, Inc.

*Added to loudest source to provide overall sound pressure level.

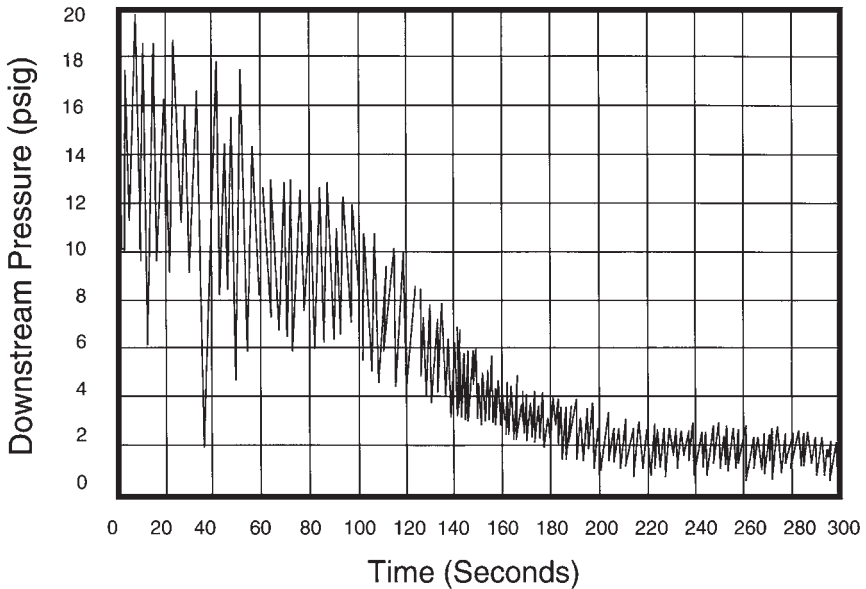


Figure 9.27 Pressure vs. time profile—downstream from valve. (Courtesy of Valtek International)

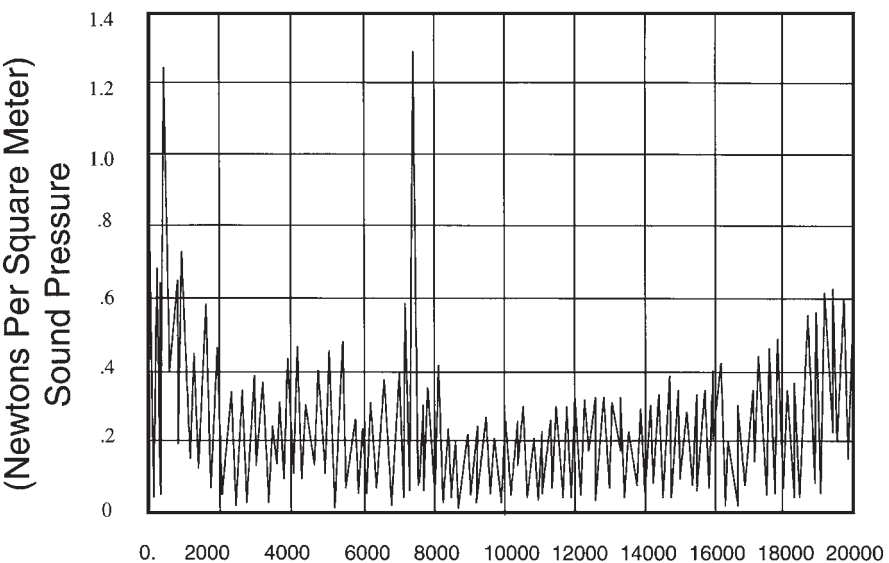


Figure 9.28 Plot of sound pressure level—downstream from valve. (Courtesy of Valtek International)

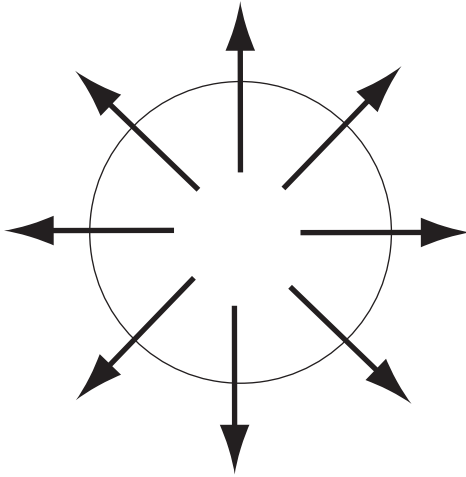


Figure 9.29 Monopole noise source. (Courtesy of Valtek International)

level (in hertz). The plot shows one discrete frequency peak occurring at 7500 Hz. Such peaks in that range are commonplace with valves that experience prolonged high noise levels. Although the test data indicate the presence of a wide range of subharmonics, the discrete peak frequency is principally responsible for the valve noise.

Turbulence is designated as one of three classifications: monopole, dipole, and quadrupole. *Monopole turbulence* is often described as an expanding and contracting source of noise (Fig. 9.29). The energy generated by a monopole-turbulent source is directly proportional to the flowing energy of the process fluid times the Mach number of the fluid, or in equation form:

$$(\text{turbulent energy}) \propto (\text{flowing energy}) \times (\text{Mach number})$$

The formula of monopole turbulence indicates that the greater speed of the flow stream will convert to more turbulent energy. Monopole energy can be easily illustrated by using a Hartmann generator (Fig. 9.30). Air flows through the nozzle (d_0) into the bore (d), causing shock waves to form inside the bore and attach to the flat surface at the bottom of the bore. As these shock waves resonate back and forth, they create discrete peak frequencies, resulting in noise that can increase by as much as 24 dBA. The importance of the Hartmann generator can be

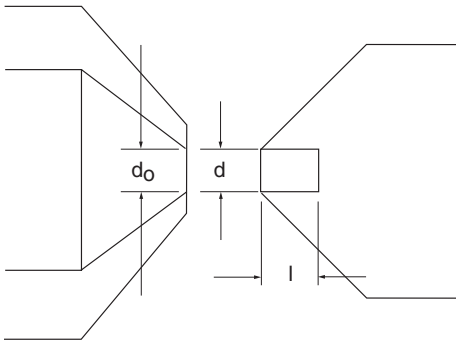


Figure 9.30 Hartmann generator (monopole noise source). (Courtesy of Valtek International)

seen, for example, if one envisions the shapes inside a globe valve that is made from barstock. The fluid follows through a small opening (seat ring) to a flat surface in the cavity leading to the outlet port (the bottom of the valve body). In an open situation, flow moves past the seat ring into the flat bottom portion of the body, where shock waves can attach and resonate. The position of the plug plays a large role in how much flow and velocity occur as well as the resulting noise (Fig. 9.31). However, studies have shown that if the seat-ring design is modified to a very narrow surface on the inside diameter, the shock waves

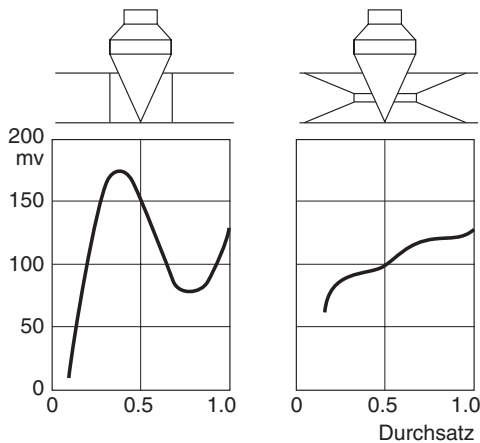


Figure 9.31 Effect of monopole noise with conventional globe valve's closure element. (Courtesy of Valtek International)

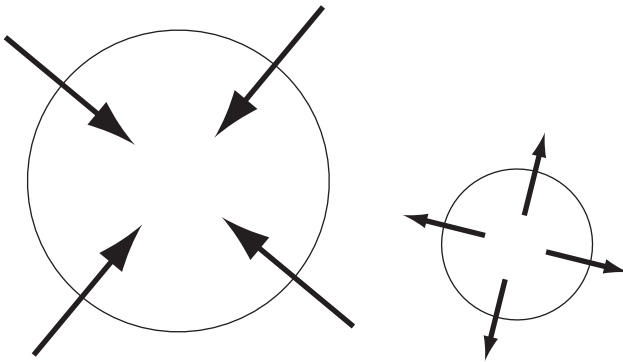


Figure 9.32 Dipole noise source. (Courtesy of Valtek International)

seemed to attach themselves to the sharp point and do not resonate in the body cavity.

Dipole turbulence is defined by two energy sources, one contracting in size as the other expands inversely (Fig. 9.32). With dipole turbulence, the energy of the turbulence is proportional to the Mach number cubed or in equation form:

$$(\text{turbulent energy}) \propto (\text{flowing energy}) \times (\text{Mach number})^3$$

Because of the cubed Mach number, higher velocities are much more critical in dipole turbulence than monopole turbulence. A common example of dipole turbulence is the “singing” telephone line (Fig. 9.33), in which alternate vortices are generated from both the top and bottom of the wire. These alternate vortices produce a discrete frequency, which can vary in pitch as the velocity changes. Dipole turbu-

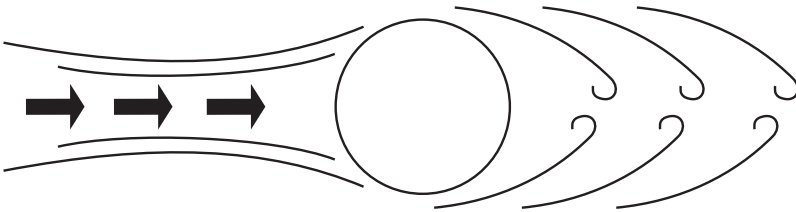


Figure 9.33 Karmen vortex street (dipole noise source). (Courtesy of Valtek International)

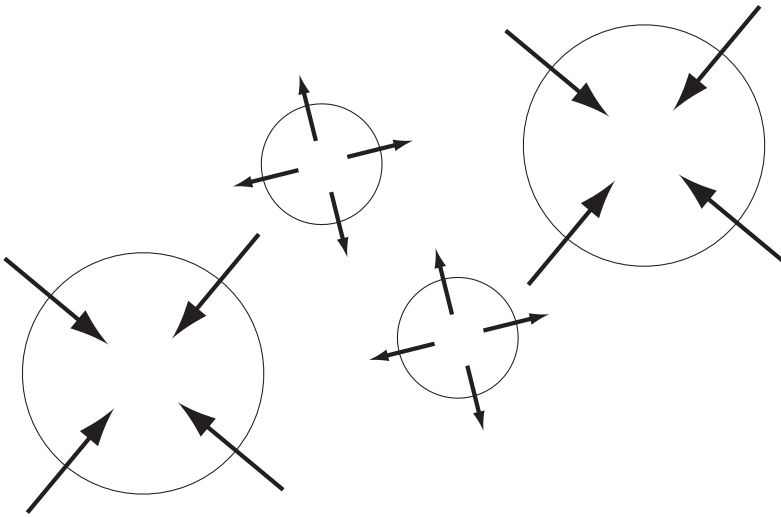


Figure 9.34 Quadrupole noise source. (Courtesy of Valtek International)

lence can be created in valves, such as with the sharp edges of a butterfly valve body and the disk. In addition, some trim-hole designs can generate dipole noise—a good example is the large flow characteristic holes designed into cage-guided trim.

Quadrupole turbulence is related to dipole noise; however, it involves two pairs of dipole turbulent energy. Although each pair is in phase (contracting and expanding inversely) with each turbulent source, the two pairs are out of phase with each other (Fig. 9.34). In this case, the turbulent energy varies according to the Mach number to the fifth power, or in equation form:

$$(\text{turbulent energy}) \propto (\text{flowing energy}) \times (\text{Mach number})^5$$

Even more than in dipole turbulence, velocity is critical to the formation of quadrupole turbulence. One important difference with quadrupole turbulence is that it involves a number of random peak frequencies rather than one discrete frequency. Nearly all noise radiating from a downstream pipe is related to quadrupole turbulence. As shown in Fig. 9.25, as turbulence generated by a valve travels downstream inside the pipe, the turbulence has a tendency to move to the outer wall while smoother portions of the flow stay in the center of the pipe.

9.7.4 Noise Regulations

A growing number of organizations monitor the amount of noise workers can be safely exposed to. For example, in the United States, the Occupational Safety and Health Act (OSHA) and the Environmental Protection Agency (EPA) both regulate noise as it affects workers and the surrounding community. Initially, the Occupational Safety and Health Act (1970) stipulated that workers could be exposed to no more than 90 dBA for an 8-hour work day. Later, the Walsh Healy Public Contracts Act was enacted to further protect workers. It regulates the exact amount of time workers may work around noise. According to this legislation, the higher the dBA level, the less time workers can spend in that area, as outlined in Table 9.6.

Table 9.6 Permissible Noise Levels*
Walsh Healy Public Contracts Act

Duration Per Day (hours)	dBA
0.25 or less	115
0.5	110
1.0	100
2.0	97
4.0	95
6.0	92
8.0	90

*Data courtesy of Valtek International.

9.7.5 Hydrodynamic Noise Prediction

Similar in some aspects to calculating the advent of cavitation and flashing, the prediction of noise levels in liquid services is based upon a number of common factors, including the pressure drop and flow capacity. In addition, the factors associated with pipe attenuation and distance from hearers are also considered. Using these factors, the following empirical equation can be used to predict hydrodynamic noise:

$$\text{dBA} = DP_S + C_S + R_S + K_S + D_S$$

where dBA = sound pressure level

DP_S = pressure-drop factor

C_S = flow capacity factor

R_S = ratio factor

K_S = pipe attenuation factor

D_S = distance factor

To calculate R_S and DP_S the pressure-drop ratio (DP_F) must be determined, which involves the following equation:

$$DP_F = \frac{\Delta P}{P_1 - P_v}$$

where DP_F = pressure-drop ratio

ΔP = pressure drop

P_1 = upstream pressure

P_v = vapor pressure

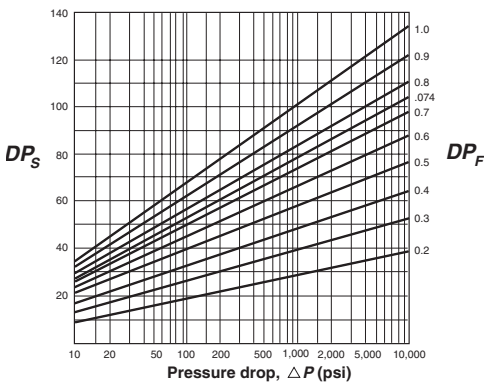


Figure 9.35 Pressure-drop factor. (Courtesy of Valtek International)

If DP_F is calculated to be 1 or greater, a flashing situation is occurring in the valve. Because flashing is indicative of a system problem, no modification to the valve will abate flashing and the resultant noise.

Once the pressure-drop ratio DP_F is determined, the pressure-drop factor DP_S can be determined using Fig. 9.35 and the ratio factor R_S can then be found using Fig. 9.36. Figure 9.37 provides a typical representation of the flow-capacity factor C_S . Table 9.7 provides typical distance factors D_S . Pipe attenuation factors K_S are found in Table 9.8.

9.7.6 Hydrodynamic Noise Example

The following service conditions apply for this example:

Fluid	Water
Upstream pressure	300 psig
Downstream pressure	90 psig
Vapor pressure	29.89 psia
Required C_v	34.8
Pipe size	2 in
Pipe schedule	Schedule 40
Distance of hearer	3 ft

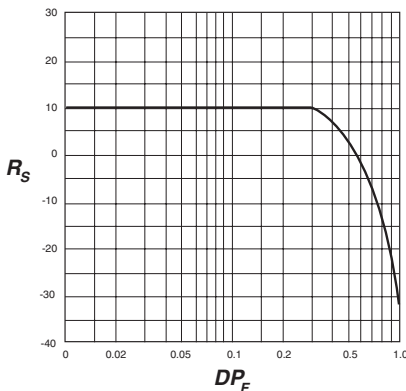


Figure 9.36 Ratio factor. (Courtesy of Valtek International)

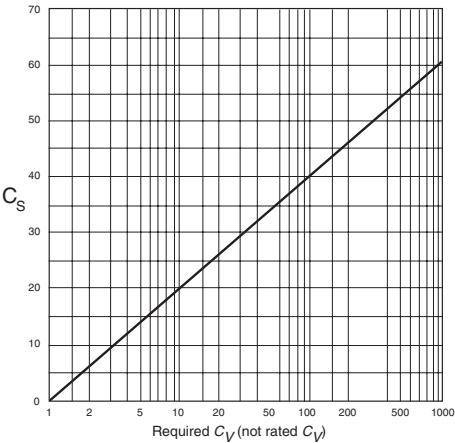


Figure 9.37 Flow-capacity factor. (Courtesy of Valtek International)

Table 9.7 Distance Factors*

Distance of hearer from noise source (feet/meters)	D_s
3 feet 0.9 meters	0 dBA
6 feet 1.8 meters	-5 dBA
12 feet 3.6 meters	-10 dBA
24 feet 7.2 meters	-15 dBA
48 feet 14.4 meters	-20 dBA
96 feet 28.8 meters	-25 dBA

*Data courtesy of Valtek International.

Note: Factors are affected by type of noise source, as well as any reflecting surfaces close to the valve.

Table 9.8 Pipe Attenuation Factors for Liquids*

Pipe Schedule													
	10	20	30	40	60	80	100	120	140	160	STD.	XS	XXS
0.5				0		-5				-11	0	-5	-15
0.75				0		-5				-11	0	-5	-15
1.0				0		-6				-12	0	-6	-15
1.5				0		-6				-12	0	-6	-14
2				0		-6				-12	0	-6	-14
3				0		-7				-13	0	-7	-16
4				0		-7		-9		-13	0	-7	-14
6				0		-8		-10		-14	0	-8	
8		4	3	0	-3	-9	-8	-12	-13	-18	0	-9	
10		5	3	0	-5	-9	-9	-13	-14	-19	0	-7	
12		6	2	-1	-6	-10	-11	-14	-15	-20	0	-6	
14	6	3	0	-2	-6	-11	-12	-15	-16	-22	0	-4	
16	6	3	0	-4	-8	-12	-13	-16	-18	-24	0	-4	
18	5	3	-2	-6	-9	-13	-15	-18	-19	-25	0	-4	
20	5	0	-4	-6	-10	-14	-16	-19	-21	-26	0	-4	
24	5	0	-6	-8	-12	-15	-19	-21	-23	-27	0	-4	
30	3	-4	-7	-8		-15				-27	0	-4	
36	3	-4	-7	-9		-15				-27	0	-4	
42		-4	-7			-15							

*Courtesy of Valtek International.

By using the pressure-drop ratio equation, DP_F is calculated as 0.74:

$$DP_F = \frac{DP}{P_1 - P_v} = \frac{314.7 - 104.7}{314.7 - 29.89} = 0.74$$

From Figs. 9.35 to 9.37 and Tables 9.7 and 9.8, the following factors apply: $DP_S = 60$, $R_S = -10$, $C_S = 31$, $D_S = 0$, and $K_S = 0$. Therefore, the hydrodynamic noise equation can be used to predict the noise from this application:

$$\text{dBA} = DP_S + R_S + C_S + D_S + K_S = 60 + (-10) + 31 + 0 + 0 = 81 \text{ dBA}$$

With a predicted sound pressure level at 81 dB, hearers could safely work in the vicinity of the valve for 8 h per day (as outlined by the Walsh Healy Act).

9.7.7 Aerodynamic Noise Prediction

Because aerodynamic noise is the most irritating type of noise to nearby hearers and communities, predicting the noise level emitted from a valve is critical to the sizing and selection process. The noise prediction for gas services varies from the hydrodynamic noise equation in that factors relating to pressure, temperature, and gas properties must also be considered. The following empirical equation is used:

$$\text{dBA} = V_s + P_s + E_s + T_s + G_s + A_s + D_s$$

where V_s = flow factor

P_s = pressure factor

E_s = pressure ratio factor

T_s = temperature correction factor

G_s = gas property factor

A_s = attenuation factor

The flow factor V_s is determined by using the valve's required C_v , as shown in Fig. 9.38. The pressure factor P_s is found by using the valve's upstream pressure (Fig. 9.39). To determine the pressure ratio factor E_s , the ratio between the upstream and downstream pressures must be calculated (Fig. 9.40). The temperature correction factor T_s is determined by Table 9.9. The gas property factor G_s is found by applying the molecular weight of the gas against Fig. 9.41. The attenuation factor A_s is found for a given pipe size and schedule in Table 9.10. The same distance factor table (Table 9.7) that was used in the hydrodynamic calculations still applies for D_s .

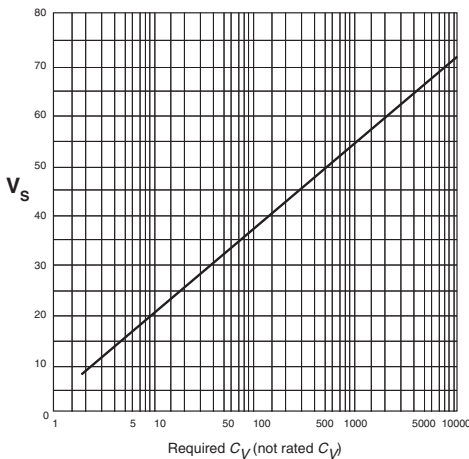


Figure 9.38 Flow factor. (Courtesy of Valtek International)

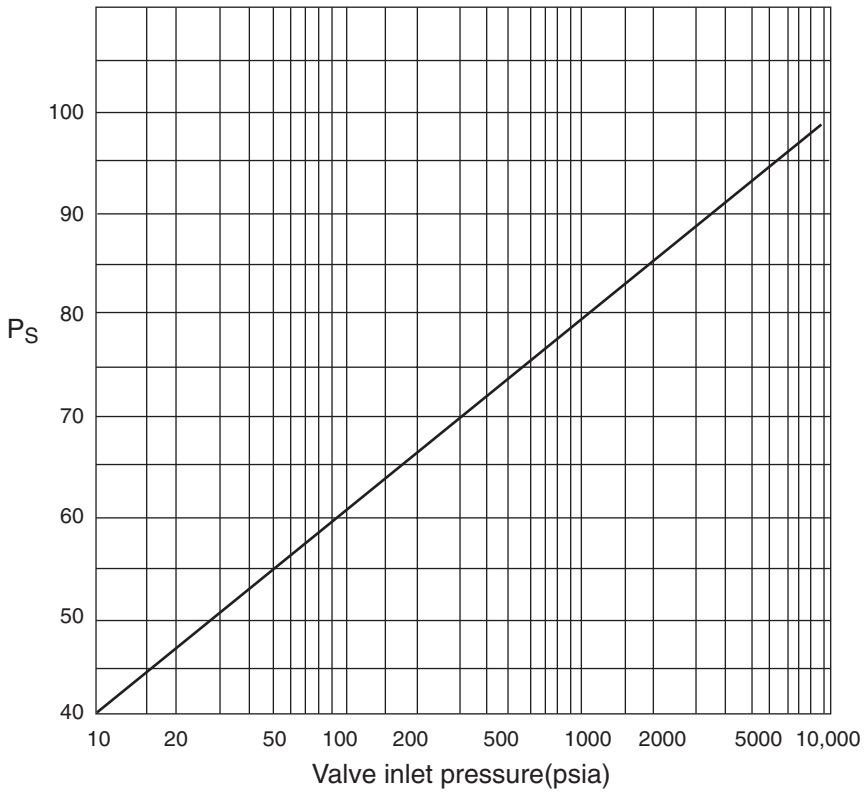


Figure 9.39 Pressure factor. (Courtesy of Valtek International)

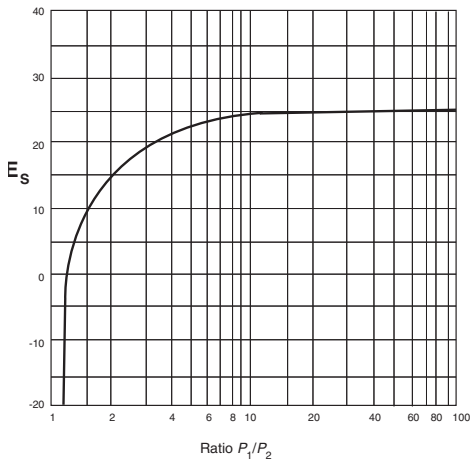


Figure 9.40 Pressure ratio factor. (Courtesy of Valtek International)

Table 9.9 Temperature Correction Factors*

Flowing Temperature of Gas	T_s
70° F / 21° C	0.0
100° F / 38° C	0.0
200° F / 93° C	-1.0
300° F / 150° C	-1.5
500° F / 260° C	-2.0
700° F / 370° C	-3.0
1000° F / 540° C	-3.5
1200° F / 490° C	-4.0

*Courtesy of Valtek International.

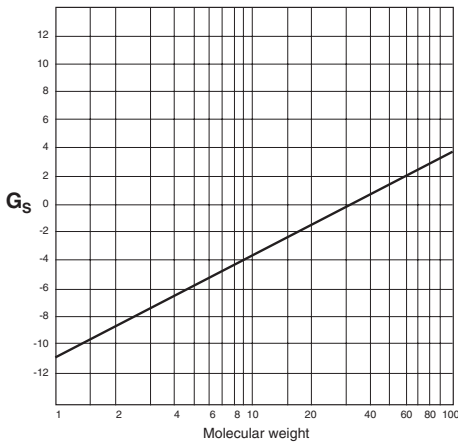
**Figure 9.41** Gas property factor. (Courtesy of Valtek International)

Table 9.10 Pipe Attenuation Factors for Gases*

Pipe Schedule													
Pipe Size	10	20	30	40	60	80	100	120	140	160	Std.	XS	XXS
1/2				-8.0		-11.5				-13.1	-8.0	-11.5	-18.5
3/4				-10.0		-13.7				-17.0	-10.0	-13.7	-21.0
1				-11.4		-15.0				-18.8	-11.4	-15.0	-22.2
1 1/2				-13.6		-17.6				-21.6	-13.6	-17.6	-25.5
2				-14.8		-19.2				-24.6	-14.8	-19.2	-27.4
3				-16.6		-20.8				-25.4	-16.8	-20.8	-29.1
4				-18.0		-22.4		-25.7		-28.0	-18.0	-22.4	-30.9
6				-20.0		-25.5		-28.8		-31.8	-20.0	-25.5	-34.1
8		-18.1	-19.4	-21.4	-24.3	-27.0	-29.1	-31.5	-33.1	-34.4	-21.4	-27.0	-34.0
10		-17.6	-20.3	-22.5	-26.6	-28.7	-31.2	-33.2	-35.3	-36.8	-22.5	-26.6	-35.3
12		-18.2	-21.8	-24.5	-28.5	-31.2	-33.8	-36.0	-37.4	-39.3	-24.5	-28.7	-36.0
14	-18.8	-21.6	-24.0	-26.0	-29.9	-32.9	-35.5	-37.7	-39.3	-40.8	-24.0	-28.5	
16	-19.5	-22.4	-24.8	-28.5	-32.0	-35.2	-37.7	-39.8	-41.9	-43.2	-24.8	-28.5	
18	-20.2	-23.1	-27.4	-30.7	-35.4	-37.2	-39.9	-42.1	-43.7	-45.3	-25.4	-29.0	
20	-20.8	-26.1	-29.8	-32.0	-36.0	-39.1	-41.8	-43.8	-45.7	-47.2	-26.1	-29.8	
24	-21.9	-27.1	-32.3	-34.9	-39.3	-42.3	-45.2	-47.3	-48.9	-50.5	-27.1	-29.5	
30	-26.1	-32.2	-35.1	-38.5	-42.7	-45.5	-48.3	-50.5			-26.0	-32.2	
36	-27.2	-33.3	-36.2	-42.0	-45.5	-48.5	-51.2				-26.4	-33.3	
42	-28.7	-37.0	-40.3	-44.5	-48.0	-50.7	-53.7				-26.7	-30.4	
48	-29.8	-39.0	-42.5	-46.5	-50.3	-53.0					-27.0	-30.5	
54	-30.5	-41.0	-44.3	-48.5	-52.2								
60	-31.2	-42.5	-45.7	-50.3	-53.5								

*Courtesy of Valtek International.

9.7.8 Aerodynamic Noise Example

The following service conditions apply to this example:

Fluid	Steam
Upstream pressure	139.7 psig
Downstream pressure	29.7 psig
Required C_v	46.2
Pipe size	2 in
Pipe schedule	Schedule 40
Distance of hearer	3 ft
Molecular weight	18.02

Using the upstream and downstream pressures, the ratio P_1/P_2 is:

$$\frac{P_1}{P_2} = \frac{139.7}{29.7} = 4.70$$

From Figs. 9.38 to 9.41, and Tables 9.7 and 9.8, the following factors are applied: $V_s = 31$, $P_s = 61$, $E_s = 22.5$, $T_s = -2$, $G_s = -1.0$, $D_s = 0$,

and $A_s = -18.0$. With these factors, the aerodynamic noise equation can be used to predict the noise from this application:

$$\begin{aligned} \text{dBA} &= V_s + P_s + E_s + T_s + G_s + A_s + D_s \\ &= 31 + 61 + 22.5 + (-2) + (-1.0) + (-18.0) + 0 \\ &= 93.5 \text{ dBA} \end{aligned}$$

According to the Walsh Healy Act, at 93.5 dB, hearers could remain in the vicinity of the valve for 4 h per day.

9.8 Noise Attenuation

9.8.1 Introduction to Attenuation

Because hydrodynamic noise is often associated with cavitating services, it can be controlled with anticavitation measures. Generally, however, noise is associated with gas applications. This section emphasizes methods to lower noise levels in gaseous applications, although some methods may be applicable to liquid applications also. The process of lowering noise or sound pressure levels is called *attenuation*. Noise pollution is a primary environmental concern, for both plant and community environments. In many cases, the sound pressure levels must be reduced by noise attenuation of the source itself (the valve) or the path (downstream piping). Correcting the offending source is the ideal situation, but this involves sophisticated attenuation devices that reduce sound pressure levels to comfortable levels. Unfortunately, the costs associated with these special attenuation devices are high. Depending on the size of the valve, the cost could increase anywhere from 40 to 200 percent. If material fatigue or diminished performance is not a concern, path attenuation may be a less expensive, easier option, although it is only treating the symptom rather than the root cause.

Valve manufacturers, especially those that offer sizing and selection software programs, routinely predict noise as part of the valve selection process. However, the user should be aware that these predicted sound pressure levels assume that the valve is installed in a completely nonreflective environment and do not consider the additional noise levels associated with walls, floors, and ceilings. For example, a valve installed in a natural-gas pressure-reduction application is predicted to produce 85 dBA, which is within the safety standards of most regulations. However, because the valve is installed in a metal building,

which is highly sound-reflective, the sound pressure level rises to 115 dBA. Therefore, the location of the valve should always be considered before determining that noise-attenuation devices or preventative measures are not necessary.

As this section outlines, a great deal of options are available to either reduce or eliminate noise. Some are more expensive than others, while some present additional problems, such as increased maintenance or added potential leak paths. Because of the costs and safety factors involved, the user should examine all options before deciding on installing an expensive antinoise valve.

9.8.2 Valve Attenuation Options

Although many users consider expensive valve trims the only solution to valve noise, a number of less expensive options exist that should be explored prior to specifying a specially engineered valve. The most simple, but overlooked option would be to restrict the access of workers to a high noise level or to provide ear protection while in that area. If equipment damage is not an issue, the main benefit of reducing noise levels is to protect the hearing of nearby workers. If workers do not need access to the affected area, then safety warnings and requirements for ear protection can be mandated and the process left alone.

Changes to the process may also be an option. The velocity may be varied by slightly changing the upstream or downstream pressures. In many cases, a discrete signal, which is within the range of hearers, is often prevented by a slight pressure variation to either side of the valve. The valve's position can also be slightly increased or decreased, allowing a minor change in flow that may disrupt the retention of shock waves on a given surface.

An interesting aspect of noise is that some linear valve styles, such as a globe valve, produce a discrete signal at 30 percent lift, despite the valve size or length of stroke. One way to deal with this phenomenon is to use a special diverting seat ring that has a special lip built into the bottom of the seat ring and breaks up the formation of shock waves.

As discussed in Sec. 9.7, velocities are directly related to turbulence and noise and can be controlled through right-angle turns. Frictional losses associated with 60 ft (18 m) of straight pipe are equal to the frictional forces produced by one 90° elbow, which will slow the velocity. Designing the system with several elbows can produce attenuation. In addition, placing two or more globe valves in series will produce a staged pressure drop and also add two or more right-angle turns per valve.

When a gas process is vented to atmosphere, high noise levels above 100 dBA can be generated. Such noise can be channeled in the opposite direction using shields or shrouds, which may lower the sound pressure level within acceptable limits. Moving the vent to a distant location may be an option in some cases, although the additional piping may be cost prohibitive.

The valve style can have some bearing on the type of noise that is generated. As explained in the preceding section, rotary valves are more apt to produce sharp dipole vortices as the flow travels past the sharp edges of the body. In addition, valve bodies produced from bar-stock commonly cause monopole noise as the flow moves through the seat and attaches to the flat surface of the outlet port. On the other hand, the conventional casting design of a globe body would avoid sharp edges associated with rotary valves and the flat surfaces of bar-stock bodies. The user should remember that different valve styles and internal geometries react differently to the same process. As a last resort, trial and error may be required to discover the one valve style that is able to handle the service without producing turbulence and subsequent pressure fluctuations that lead to noise.

When the flow direction is not a critical element of the application, the valve can be installed backwards (inlet port is installed downstream, and the downstream port is installed upstream), so that the flow direction is opposite the normal operation. (For example, a flow-over-the-plug linear valve will become a flow-under-the-plug valve.) When this is done, the fluid will then flow through a different valve geometry, in which monopole, dipole, or quadrupole noise is less likely to be created. For example, changing from flow-over-the-plug to flow-under-the-plug may avoid monopole noise that would be created from the Hartmann generator effect (Sec. 9.7.2). (The process stream flows up through the seat into the upper gallery, where the geometry provides no flat surface perpendicular to the flow where shock waves can form.)

In some cases, modifications can be made to the existing valve trim to attenuate the noise without installing expensive trims or downstream attenuation equipment. As discussed earlier, monopole noise will attach itself to a very narrow landing on the seating surface. Reducing this landing through machining may be possible, as long as the seat's seating surface and overall strength is not affected.

The location of the valve is vitally critical to the amount of noise generated by turbulence. Often noise is generated by turbulence in the valve and is then carried to downstream piping. The noise radiates the pressure fluctuations through the downstream pipe wall to the envi-

ronment as sound waves. This phenomenon occurs with long, straight sections of thin-walled piping that are more apt to flex. Conversely, piping elbows and other nonlinear piping configurations are stiffer and are not apt to allow wall fluctuations. If a valve is included in a long stretch of piping, the preferred arrangement would be a long length of pipe on the upstream side of the valve and on the downstream side an elbow or a shorter length of pipe. The longer the pipe, the more sound radiation is possible. Piping supports can also be used to stiffen long lengths of piping, preventing the flex of the pipe wall.

In some process services in which the valve discharges fluid into a large vessel, the valve can be located next to the vessel without a long expanse of pipe. This will allow the valve to discharge the fluid into the vessel and the noise to be absorbed in a larger area.

If the valve and downstream piping are located in a room or protective shed with a number of close-by hard reflective surfaces, the sound pressure levels may increase significantly, upwards of 30 to 40 dB. However, by moving the location of the valve to the wall, the downstream side of the pipe can be placed outside of the room. Not only will the noise be eliminated from the room, but the noise radiated to the environment outside of the room will also be less.

Another option is to specify a thicker wall schedule in the downstream piping, which provides greater stiffness. For example, using a schedule 80 pipe instead of a schedule 40 pipe will lower the sound pressure level by approximately 4 dB. Table 9.11 provides a correction factor for noise attenuation for piping that has a heavier wall schedule (assuming schedule 40 pipe wall thickness is standard.)

One of the more common methods of dealing with high sound pressure levels is to absorb the noise with thermal or acoustic insulation, which can be wrapped around the valve or downstream piping. This is the best solution only when high sound pressure levels offer no threat of fatigue to materials or substandard performance of instrumentation. Generally, 1 in (2.5 cm) of normal thermal insulation will provide a reduction in sound pressure level of between 3 and 5 dB. Acoustic insulation is manufactured to absorb more sound energy and can provide a reduction of 8 to 10 dBA per inch of insulation. Depending on the *R* value of the insulation, a 3-in insulation will provide the maximum attenuation anywhere from 15 to 24 dB. (Additional insulation does not attenuate the noise any further.) Table 9.12 outlines typical insulation factors.

One caution should be noted, however. As previously explained, noise levels close to a valve and its immediate downstream piping may be reduced with an elbow, thick schedule pipe, or insulation.

Table 9.11 Pipe-Wall Attenuation*

Pipe Size	Schedule 40	Schedule 80	Schedule 120	Schedule 160
2-inch (DN 50)	0 dBA	- 6 dBA	-8 dBA	-12 dBA
3-inch (DN 75)	0 dBA	-7 dBA	-9 dBA	-13 dBA
4-inch (DN 100)	0 dBA	-7 dBA	-10 dBA	-13 dBA
6-inch (DN 150)	0 dBA	-8 dBA	-12 dBA	-15 dBA
8-inch (DN 200)	0 dBA	-9 dBA	-14 dBA	-18 dBA
10-inch (DN 250)	0 dBA	-10 dBA	-14 dBA	-19 dBA
12-inch (DN 300)	0 dBA	-11 dBA	-16 dBA	-20 dBA

*Data courtesy of Fisher Controls International, Inc.

However, these methods only protect the hearer in the immediate vicinity of the valve. Since these methods do not attenuate the source of the noise, sound will continue in the downstream piping and may surface at an unprotected point further downstream (Fig. 9.42). At that point, either the noise must be tolerated or additional corrective action must be taken.

9.8.3 Downstream Antinoise Equipment

In some applications, adding an antinoise element immediately downstream from the valve may be effective in attenuating the noise to reasonable levels. In addition, these elements can absorb energy or straighten turbulent flow so that noise is not carried downstream. The cost associated with these supplemental devices is less than or equal to special valve trim. Access to a downstream element is much easier for

Table 9.12 Insulation Factors*

Depth of Insulation	dBA Reduction
1 in / 2.5 cm	-5 dBA
2 in / 5.1 cm	-10 dBA
3 in / 7.6 cm	-15 dBA

*Data courtesy of Valtek International.

maintenance purposes than gaining access to special trim. Common antinoise elements include attenuator plates, diffusers, silencers, and external stacks. Because these devices all utilize small holes or flow paths, they are susceptible to plugging if the process fluid contains particulate matter, which may require additional maintenance.

Placed downstream in series with the valve, the *attenuator plate* is a downstream antinoise element (Fig. 9.43) that provides anywhere from single to multiple stages of pressure reduction (Fig. 9.44). Attenuator plates typically reduce the overall sound pressure level by up to 15 dB. Using a pattern of holes, each stage of the attenuator plate has its own

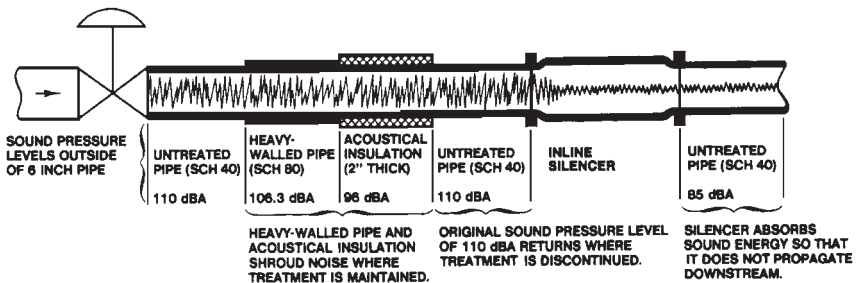


Figure 9.42 Multiple methods of path treatment of noise. (Courtesy of Fisher Controls International, Inc.)

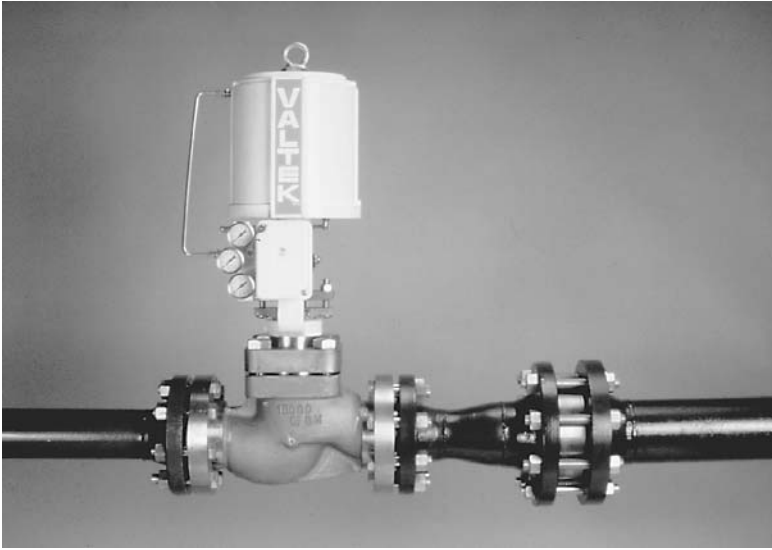


Figure 9.43 Attenuator plate mounted downstream from a globe control valve. (Courtesy of Valtek International)

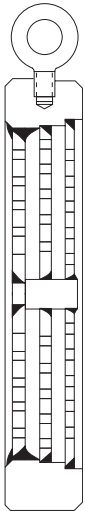


Figure 9.44 Three-stage attenuator plate. (Courtesy of Valtek International)

individual flow capacity. As Fig. 9.44 illustrates, each succeeding stage has a larger flow area, which provides the staged pressure reduction and maintains velocities at lower levels. The multiple holes act as a straightening device for the turbulent fluid, providing a series of controlled, smaller fluid streams instead of a large turbulent eddy. Although these smaller fluid streams still have some turbulence, they are more easily dissipated throughout the overall process stream because of their size. Since the area of the plate is limited to the inside diameter of the pipe, as well as the hole pattern, only so much flow can pass through the first stage. The maximum flow capacity through the attenuator plate is achieved with a pressure ratio of 4.5 to 1 (or less). High rangeability is highly unlikely with attenuator plates; therefore, they should be installed only in moderate to low rangeability applications. Because the flow capacity is limited, attenuator plates should be considered only for those applications that can handle such a reduction in flow. In some applications where additional flow is needed, a larger plate can be specified with more flow area, but pipe expanders or reducers must be used to allow the installation of the larger plate in a smaller pipeline. Not only does this raise costs, but it also adds a number of line penetrations that could leak.

For applications that require greater flow than offered by an attenuator plate, a *diffuser* is often specified, which also offers reductions of up to 15 dB. As shown in Fig. 9.45, a diffuser is a long cylinder tube with a closed end that can vary in length according to the flow needed. As with attenuator plates, the diffuser is installed downstream in series with the valve. The diffuser is designed to fit inside the pipeline, allowing for a specific clearance between the inside diameter of the pipe and outside diameter of the diffuser. The diffuser is held in place between the raised face flanges of the valve and pipeline, or it can be welded in place. A diffuser can also be directly bolted or welded to the valve and be used to vent to atmosphere. When venting to atmosphere, a diffuser can be equipped with shrouds to direct the noise away from hearers. Although the diffuser shares the overall pressure drop with the valve, its flow capacity can be expanded by making the diffuser longer and adding more holes. These holes control the sound pressure level by passing the flow through the holes to absorb sound energy and minimize turbulence. The major disadvantage of a diffuser is the maintenance problems associated with the small holes, which can become plugged if the process contains oversized particulates. Because the holes in the diffuser are perpendicular to the piping, they have a tendency to impinge condensates and particulates directly on the piping wall, which may lead to erosion.

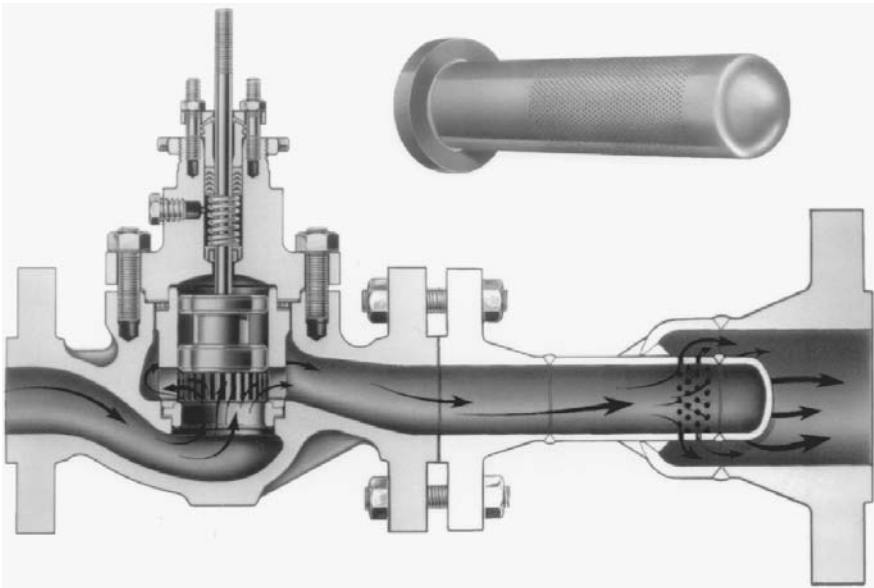


Figure 9.45 Downstream diffuser. (Courtesy of Fisher Controls International, Inc.)

A *silencer* is used when reductions of sound pressure levels of more than 15 dB are required, which are beyond the capability of attenuator plates or diffusers. Depending on the design and process service conditions, attenuations as high as 35 dB can be achieved with a silencer. Similar to a diffuser in that it shares the pressure drop with the valve, a silencer also lowers the sound pressure level by absorbing noise. As shown in Fig. 9.46, a common silencer incorporates a series of compartments that use tubes with holes, much like minidiffusers. Acoustic material is used throughout the silencer to absorb sound and process energy. The primary disadvantage of silencers is that they are designed to attenuate a particular frequency. Overall, silencers are good for applications with a constant flow. However, if the application is such that the flow varies routinely, the frequency will also vary and may render the silencer ineffective. While a silencer is less expensive than other antinoise options, it requires some piping modifications, including piping supports. Depending on the application, the size of the silencer can be quite large. This may become a factor where space is limited. Silencers are normally flanged and bolted to the pipeline, although they can also be used to vent to atmosphere.

Some valves use an *external stack* (also known as an *atmospheric resistor*) as a downstream element to reduce noise in venting or blowdown

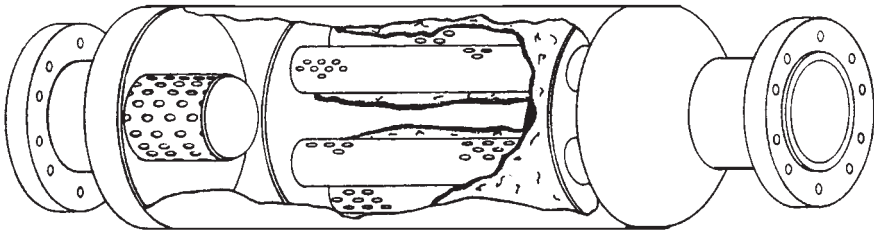


Figure 9.46 In-line silencer. (Courtesy of Fisher Controls International, Inc.)

applications (Fig. 9.47). Instead of installing antinoise trim inside the valve, the stack is placed immediately downstream from the valve's outlet port. This design provides several benefits. First, the physical characteristics of the stack can be larger, allowing a greater outside diameter and stack height. This allows greater flow and increased



Figure 9.47 External stack mounted on outlet of angle body control valve. (Courtesy of Control Components Inc., an IMI company)

attenuation than a stack inside the valve—which is limited by the body's gallery height. Second, the antinoise mechanisms built into the stack—such as expanding flow areas, tortuous paths, etc.—can lower the exit velocity and share the pressure drop with the valve. This design provides greater attenuation while not affecting the overall flow rate.

9.8.4 Downstream Antinoise Equipment Sizing

Sizing for the flow capacity of downstream equipment is based on the number of stages of pressure drop taken. These stages can be taken through one element (such as an attenuator plate) or a number of single-stage elements in series (such as two diffusers). A common equation for attenuation plates follows:

$$C_v = \frac{1}{\sqrt{\left(\frac{1}{C_{v_1}}\right)^2 + \left(\frac{1}{C_{v_2}}\right)^2 + \left(\frac{1}{C_{v_3}}\right)^2 + \left(\frac{1}{C_{v_N}}\right)^2}}$$

where C_v = total flow capacity

C_{v_1} = flow coefficient of the first control element (or first stage)

C_{v_2} = flow coefficient of the second control element (or second stage)

C_{v_3} = flow coefficient of the third control element (or third stage)

C_{v_N} = flow coefficients of any additional control elements

9.8.5 Downstream Antinoise Equipment Sound-Pressure-Level Prediction

Predicting the overall sound pressure level is determined by the following two equations:

$$\begin{aligned} \text{SPL} = & 22 + 12 \log_{10} \left(\frac{P_1}{P_O} - 1.05 \right) + 10 \log_{10} (C_v F_L) + 10 \log_{10} (P_1 P_2) \\ & + 30 \log_{10} \left(\frac{t_{40}}{t} \right) - G_s + T_L \end{aligned}$$

$$\frac{P_1}{P_O} = \frac{\frac{P_1}{P_2} + Z}{Z}$$

where SPL = sound pressure level

Z = number of elements (or stages)

P_1 = inlet pressure

P_2 = outlet pressure

P_O = outlet pressure for each stage

t_{40} = schedule 40 pipe wall thickness

t = wall thickness for given wall pipe

G_s = gas property correction factor (Table 9.12)

T_L = SPL velocity-correction factor for gas discharges above Mach 0.15 ($T_L = 20 \log_{10} [1/(1.1-M)]$)

M = Mach number of outlet pipe

When two elements are combined in series, up to 3 dB should be added to the total sound pressure level to compensate for having two separate noise sources. Figure 9.48 provides this data. If two noise sources have identical sound pressure levels, the overall intensity will not be equal to that level, but will be greater than either noise source. A 6-dB insertion loss factor should be included in the overall sound pressure level to compensate for a close connection between the element and the valve. *Close connection* is defined as one pipe reducer length. The sound pressure level of venting applications can also be determined. Although the general rule is that spherical radiation of

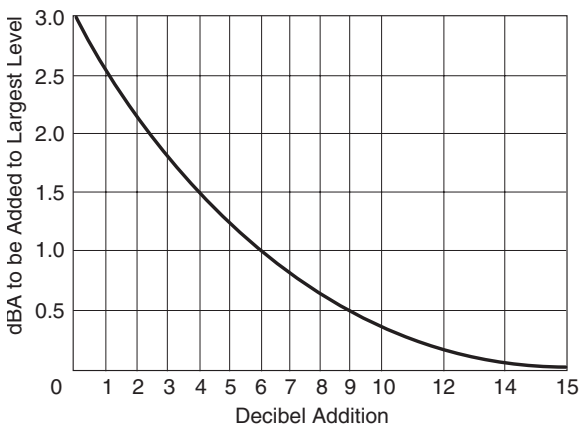


Figure 9.48 dBA addition for two elements installed in series. (Courtesy of Valtek International)

noise reduces the sound pressure level by 6 dB for every doubling of distance, noise emitted over long distances can absorb even more sound because of atmospheric absorption, and the attenuating effects of nearby objects and the ground.

The following equation can be used to calculate the sound pressure level for all gaseous venting applications, except steam:

$$\text{SPL}_{\text{intermediate}} = \text{SPL}_{\text{elements}} - 10 \log_{10} \left(\frac{(3.2)(10^{-11}) P_2 D^2 T^{0.5} R^2}{t^3} \right) - \frac{G_s}{2}$$

where $\text{SPL}_{\text{intermediate}}$ = uncorrected sound pressure level from vent

$\text{SPL}_{\text{elements}}$ = sound pressure level emitted from control elements

D = downstream nominal pipe diameter

P_2 = valve downstream pressure

R = distance from vent

T = absolute temperature

Although this equation is used to find the total sound pressure level emitting from the vent, the sound pressure level can also be lowered by the direction of the noise.

The following equation is used to calculate the sound pressure level for steam-venting applications:

$\text{SPL}_{\text{intermediate}}$

$$= \text{SPL}_{\text{elements}} - 10 \log_{10} \left(\frac{(3.2)(10^{-10}) P_2 D^2 R^2 (1 + 0.00126 T_{\text{SN}})^3}{t^3} \right)$$

where T_{SN} = superheated steam temperature

Sound pressure levels are also reduced if the noise radiates in a directional nature rather than spherical. In other words, the farther the noise is pointed away from the hearer, the less noise is heard. This phenomena is called *directivity*. This concept is illustrated in Fig. 9.49 and Table 9.13. With particular venting applications, directivity can occur if the vent is pointed away from workers or nearby communities or if a resistor shroud is used to direct the sound upward (or away from the hearer). In venting applications where directivity occurs, the reduction of the sound pressure level can be determined by the following equation:

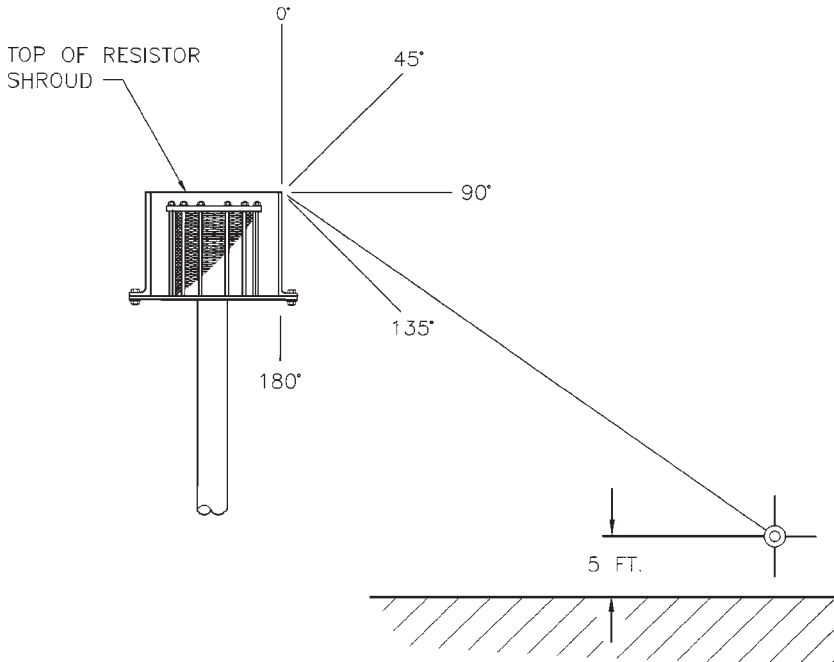


Figure 9.49 Angle location of hearer from noise source as associated with the directivity index. (Courtesy of Control Components Inc., an IMI company)

$$\text{SPL}_{\text{total}} = \text{SPL}_{\text{intermediate}} + \text{DI}$$

where $\text{SPL}_{\text{total}}$ = total noise emitting from the vent
 DI = directivity index (Table 9.13)

Atmospheric noise can be divided into *near field* and *far field*. The near-field noise is the noise that is generated within 3 to 10 ft (1 to 3 m) of the source, while the far-field noise is that generated beyond 10 ft from the source. In far-field situations where sound spreads in a radial, homogeneous pattern, the noise intensity is attenuated by the distance—intensity decreases inversely to the (distance)² from the vent. This relationship is found in the following equation:

$$I = \frac{W}{4\pi r^2}$$

Table 9.13 Typical
Directivity Index*

Angle Away from Axis of Resister Shroud	dBA Addition or Subtraction
0°	0
20°	+1
40°	+8
60°	+2
80°	-4
100°	-8
120°	-11
140°	-13
160°	-15
180°	-17

**Data courtesy of Control Components, Inc.*

where I = sound intensity (W/m^2)
 W = sound power (W)
 r = distance from sound source

This calculation applies only to far-field situations, in which large distances are involved and can be affected by a number of different factors, including humidity, wind, presence of trees, etc.

9.8.6 Antinoise Valve Trims

In difficult gaseous applications, noise must be treated at the source rather than treating the symptom with insulation, heavier or nonlinear piping, or ear protection. This means that modifications must be made to the valve to minimize or eliminate the high sound pressure levels

and resultant vibration that can fatigue metal or affect the performance of nearby instrumentation. The antinoise trim must reduce the pressure drop, so that the resultant high velocities do not approach sonic levels. The most common approach to this problem is to install special trims in globe valves. In principle, these trims channel the fluid through a series of turns, which affects the velocities and pressures involved. Each turn is typically called a *stage*. Antinoise trims can include anywhere from 1 to 40 or even 50 stages, based on the design. While anticavitation trims are designed to flow over the plug in linear valves, antinoise trims are designed to flow under the plug. This direction allows an expanding flow area in the later stages of the antinoise device, which slows the velocity to subsonic levels.

A number of different antinoise trim devices are in existence, but for the most part they can be categorized into four different styles: slotted, multihole, tortuous path, and expanding teeth. *Slotted trim* use is a single-stage cage or retainer that contains long, narrow slots around the entire diameter (Fig. 9.50). As the fluid passes through the slots, turbu-

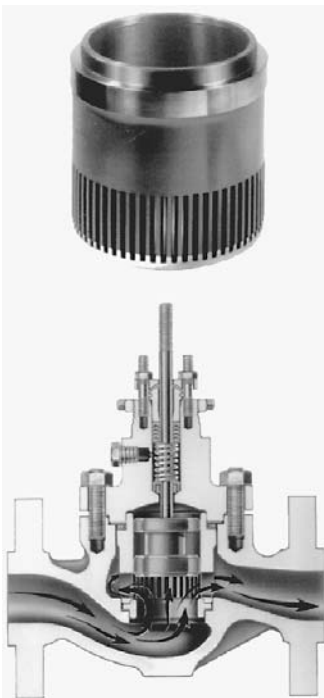


Figure 9.50 Single-stage multiple-slotted cage.
(Courtesy of Fisher Controls International, Inc.)

lence is broken up into smaller eddies, and the velocity is distributed evenly throughout the gallery of the globe body. This design works best when the pressure drop to inlet pressure ratio ($\Delta P/P_1$) is equal to or less than 0.65 and when the maximum downstream pressure (P_2) is less than half of the fluid's sonic velocity. If the $\Delta P/P_1$ ratio is higher than 0.65, the pressure drop may be handled by adding a second device (such as a downstream element) to share the pressure drop. Slotted cages or retainers offer noise attenuation up to 15 dB and are relatively inexpensive when compared to other antinoise trims. Outlet velocity is limited to below Mach 0.5. Additional dB reduction can be handled by adding an attenuation plate or diffuser downstream to the valve, which can also be cost effective when compared to other antinoise trims.

Multihole trim utilizes a number of cylinders, also known as stages, with drilled or punched holes that control turbulence in the flow stream (Fig. 9.51). This device also has a secondary use as a seat-ring retainer, which allows a clearance between the plug and the inside diameter of the retainer (Fig. 9.52). This device can also be designed as a cage, where the plug guides on the inside diameter (Fig. 9.53). One of



Figure 9.51 Single- and multiple-stage attenuators. (Courtesy of Valtek International)

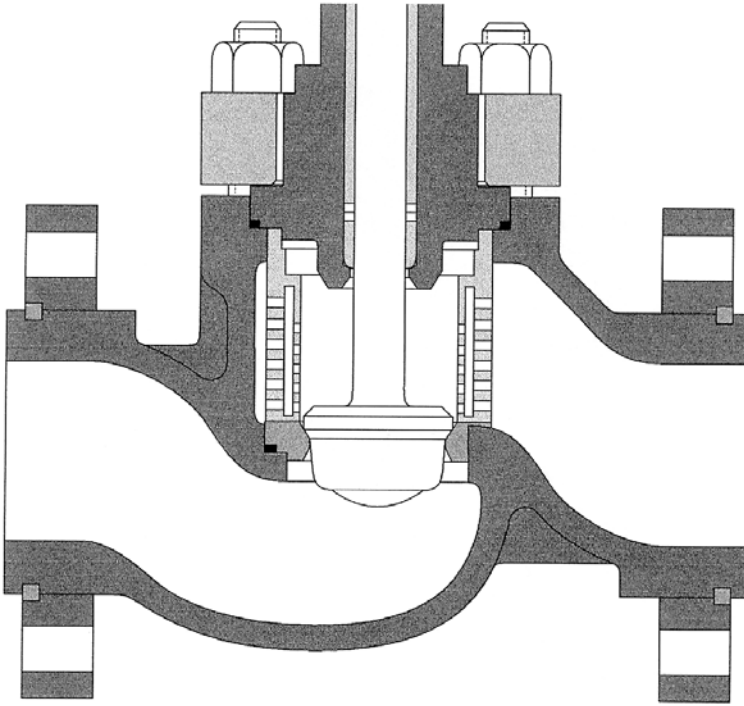


Figure 9.52 Globe valve equipped with two-stage attenuator. (Courtesy of Valtek International.)

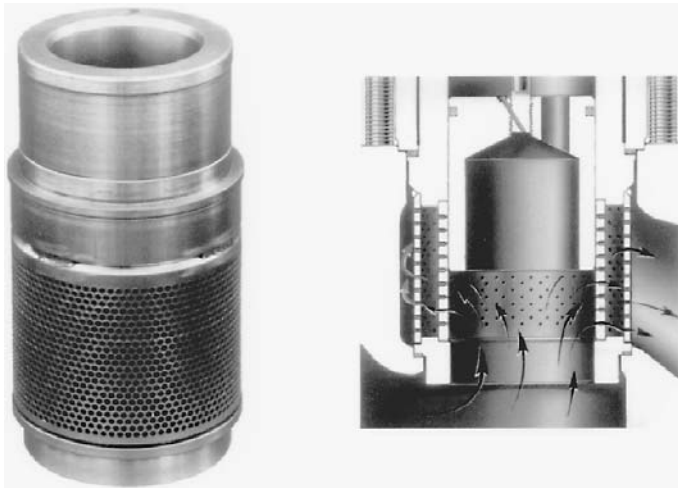


Figure 9.53 Globe valve equipped with two-stage attenuation cage. (Courtesy of Fisher Controls International, Inc.)

the key engineering elements of multihole trim is its utilization of sudden expansions and contractions. With flow under the plug, the pressure drop occurs as the flow moves through the seat to the inside diameter (an expansion), through the first stage cylinder (a contraction), through the area between cylinders (an expansion), through the second-stage cylinder (a contraction), and so forth. With this method, a portion of the pressure drop is taken at each stage. As the pressure drop is taken in stages, velocity is maintained at acceptable and reasonable levels of around Mach 0.33. The number of stages, flow areas, and flow-area ratios are determined by the velocity control required to avoid high sound pressure levels. In other words, the greater the control, the more stages and flow area that are required. The only limitations to the number of stages are the inside dimensions of the globe-body gallery and the amount of flow required to pass through the valves.

As the flow moves through the valve's vena contracta, the increased velocity, along with the geometry of the seat, creates turbulence. If untreated, this may create pressure fluctuations and eventually noise as the flow carries down the pipe. With multihole devices, the large turbulent eddy is broken up into smaller eddies. As the flow moves through the entire trim, the resulting small eddies are easily dissipated into the overall flow stream, which is illustrated in Fig. 9.54. The use of

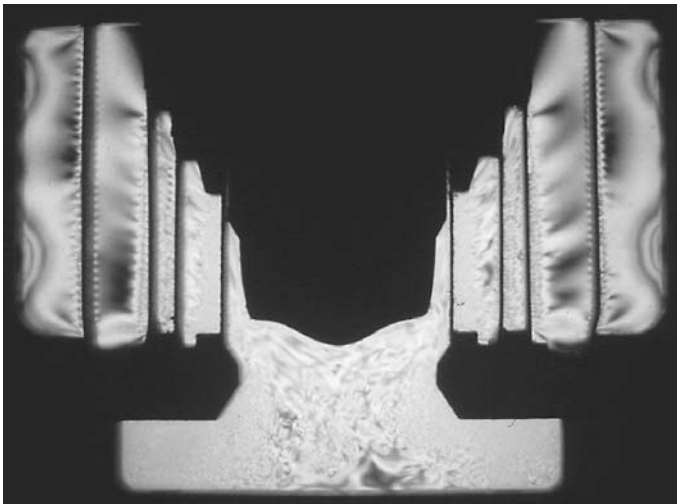


Figure 9.54 Schlieren display showing dissipation of turbulent eddies with attenuation trim. (Courtesy of Valtek International)

smaller holes also decreases the noise energy significantly. If one hole in a cage generates 90 dB, studies have shown that two smaller holes (which add up to the total area of the original hole) will generate less noise—in this case 84 dB. This is due, in part, to the principle that the energy generated by noise is proportional to the square of the hole area. Therefore, using one hole instead of two will provide the desired flow, but will also double the sound pressure level. Each succeeding cylinder is designed with more or larger holes. Not only does this provide an increased flow area, but it also handles the increased gas volume that results from the pressure drop. In addition, the materials and overall design of multistage devices are selected to provide maximum acoustic impedance, avoiding any geometry that may create monopole, dipole, or quadrupole noise. This is especially important when the plug is throttled close to the seating surface where noise is most likely to occur.

When used as part of the valve's trim, multihole devices can achieve attenuation of sound pressure levels up to 15 dB for one- and two-stage devices, while multistage devices can achieve up to 30 dB. When high-pressure ratios ($\Delta P/P_1$) are greater than 0.8, the addition of a downstream element (in conjunction with the valve) can divide the pressure drop between the two. However, both should be engineered to produce the same noise level so as to not increase the overall sound pressure level. As discussed in Sec. 9.2, quarter-turn plug valves can be equipped with severe service grids, similar to multihole devices, which take an additional pressure drop and control turbulence.

As detailed extensively in Sec. 9.2.7, a tortuous-path device uses a series of 90° turns etched or machined into a stack of metal disks to slow velocity to acceptable levels. For gas service, this same device can be used, although the flow direction is opposite that of liquid applications. The flow direction moves from the inside diameter of the stack to the outside diameter. The tortuous path becomes wider and/or deeper as it progresses, widening the flow area. Each turn in the tortuous-path device is considered to be one stage. With some mazelike paths, upwards of 40 right-hand turns are possible, achieving the same number of stages and providing extremely high attenuation. Tortuous-path devices typically provide attenuation up to 30 dB.

Like the tortuous-path trim, *expanding-teeth trim* uses a stack of disks. Instead of a tortuous path, however, expanding-teeth trim uses a series of concentric grooves (referred to as *teeth*) that are machined onto both sides (face and backside) of the disk (refer again to Fig. 9.13). Flow arrives from under the plug to the inside diameter where it passes through the wavelike teeth in a radial manner. As shown in that figure,

the spacing between the teeth grows significantly larger as the flow moves to the outside diameter, permitting flow expansion, increasing pressure, and decreasing velocity. In addition, as the flow moves over the grooves, the phenomenon of sudden expansions and contractions takes place, which provides staged pressure-drop reduction and increased frictional losses. Figure 9.55 shows where the fluid expansions and contractions take place as the flow moves over the grooves. One advantage of the expanding tooth design over the tortuous-path or multiple-hole trims is that its passages are wider than the beginning of a tortuous path or a hole, which allows for particulates to flow through the stack without clogging the inlet passages. Each groove (or tooth) in the stack is considered to be a stage, and in most cases this trim can have up to seven grooves, providing seven stages of pressure drop. Depending on the number of teeth in the design, this trim can provide up to 30 dB attenuation.

An antinoise trim can often be used in series with a downstream element to attenuate noise to acceptable levels. For example, when noise is close to the threshold of pain (140 dBA) and the valve cannot be removed from a reverberate chamber, such as a metal building, installing antinoise trim may make a significant reduction of up to 30 dB. However, to reduce the sound pressure level down to 85 dBA (allowing employees to work an entire 8-h shift), a downstream element must be installed to reduce the noise by another 15 dBA, which

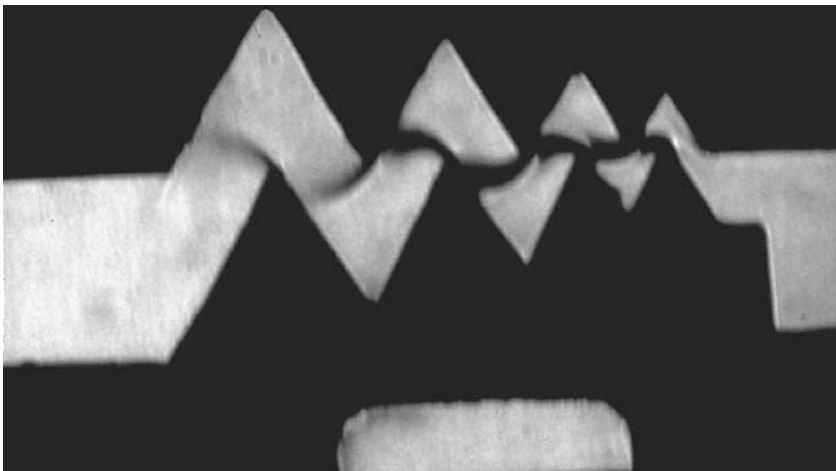


Figure 9.55 Schlieren display showing pressure reduction through sudden expansions and contractions with expanding-teeth trim. (Courtesy of Valtek International)

brings the noise level to 95 dBA. Insulation will most likely be needed to bring the level close to the desired 85 dBA. Although this is an extreme application, it does indicate the multiple options necessary to bring extremely high noise down to acceptable levels.

Antinoise trims with multiple stages that provide attenuation up to 30 dB are often the most expensive method of noise control. One- and two-stage devices are often less expensive but only provide attenuation up to 15 dB. The addition of a diffuser, for example, with a two-stage trim may provide the same attenuation as the more expensive multistage trim. When the required noise attenuation is 15 dB or less, in many cases a downstream element will accomplish what a two-stage trim can but at a lower cost. And in some cases, simple modifications to the process system or the orientation of the valve or changing of the pipe configuration may be even better cost-effective options—as long as the noise is only a hearing concern and is not destructive to the equipment.

9.9 Fugitive Emissions

9.9.1 Introduction to Fugitive Emissions

In many industrial regions of the world, increasing levels of environmental pollution have led to enactment of strict antipollution laws, which target emissions from automobiles, home heating systems, and industry. In particular, process industries have been under legislative mandate to reduce or eliminated fugitive emissions from their process systems. These antipollution laws target all devices that penetrate a process line, such as valves, sensors, regulators, flow meters, etc. Although many users see such legislation as costly and labor-consuming, a side benefit to tighter fugitive-emissions control is a more efficient system, with less lost product and greater efficiency. Even if a user is not under legislative mandate to reduce emissions, maintaining a strict antifugitive emissions program can provide greater production savings than the actual cost of the program. A case in point is the power-generation industry that, in the past, has accepted leakage of steam applications as standard operating procedure. Although steam (being water-based) is not a fugitive emission, power plants have discovered that using high-temperature seals prevents significant steam losses, which in turn lowers operating costs. In addition, power plants are operating more in the range of high-pressure superheated steam to improve energy efficiencies, which requires new sealing systems for safety reasons.

9.9.2 Clean Air Legislation

In the United States, the Clean Air Act was amended in 1990 to include some of the strictest laws regarding industrial pollution. In general terms, it mandates lower fugitive emissions from process equipment, including valves. Because most valves in today's chemical plants were installed prior to the new standards, maintenance personnel face a choice of retrofitting existing valves to the new standard or replacing them with new valves equipped with packing-box designs that comply with the Environmental Protection Agency (EPA). The Clean Air Act mandates a 500-parts-per-million (ppm) standard on all valves. As compared to past leakage standards, this new standard is 20 times more stringent. The Clean Air Act lists 189 hazardous materials that must be monitored by the law; 149 of these hazardous materials are volatile organic compounds (VOC), which can be easily monitored using an organic sniffer (Fig. 9.56). The Clean Air Act provides an



Figure 9.56 Organic sniffer used to detect fugitive emissions. (Courtesy of Valtek International)

incentive of fewer inspections if the valves are tested below the mandated 500 ppm. On the other hand, process systems with fugitive emissions higher than 500 ppm must increase the number of inspections and/or implement programs designed to improve the quality of the system.

The final phase of the Clean Air Act began in April 1997. A 500-ppm standard applies, but quarterly testing is permitted if less than 2 percent of all valves fail to meet the standard. If the failure rate is higher, monthly testing is mandatory unless a quality-improvement program is instituted. A plant can earn semiannual testing status if less than 1 percent of the valves fail to meet the standard. And finally, if less than 0.5 percent of the valves do not meet the standard, the plant can earn an annual test status. With the number of valves in a typical plant numbering in the hundreds and even thousands, achieving the higher semiannual or annual test status is important in order for the plant to avoid additional paperwork, testing, and maintenance. A graph indicating the program as outlined by the Clean Air Act is shown in Fig. 9.57.

9.9.3 Detection Standards

Clean air legislation calls for field monitoring of all line penetrations. Static seals at the flanges or body gaskets retain their seals for some

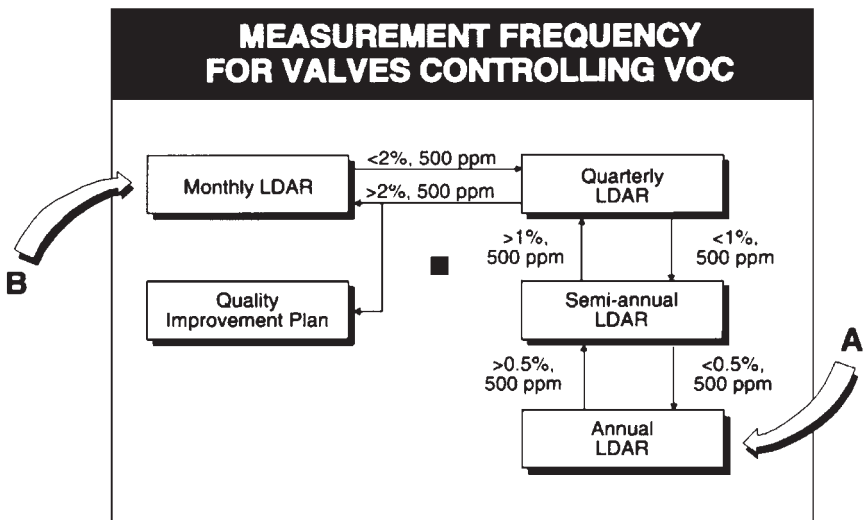


Figure 9.57 Monitoring frequency required by the Clean Air Act (United States). (Courtesy of Fisher Controls International, Inc.)

time. However, the sliding seal at the stem or shaft is more apt to leak because packing damage occurs over time due to friction. Because of the potential leak paths, valve packing boxes attract the most attention when fugitive emissions testing is performed.

The leak detection and repair (LDAR) procedure outlines the procedures for inspections and leak repairs. In addition to the LDAR procedure, a related regulation is "Method 21: Determination of Volatile Organic Compound Leaks." In general terms, Method 21 provides leakage definitions, as well as the proper procedures for using an organic sniffer to detect a leak or measure the leakage from the valve's static seals and the dynamic seal at the packing box. With linear-motion valves, the leakage reading is taken where the rising-stem slides out of the bonnet. With rotary valves, the reading is taken where the shaft penetrates the body. Measurements are also taken at all static seals around the body, bonnet, and flange gaskets.

When metal bellows-sealed valves are used (Sec. 9.9.6), they can be equipped with a leak-detection port, which can be used to monitor any fluid leakage between the bellows and the packing box. Although a negative (no emissions) measurement can be taken at the seal, the user can also read a leak-detection gauge for visual verification that the bellows has remained pressurized.

9.9.4 Packing-Box Upgrades

The user may replace an existing valve with one that has EPA-compliant designs. However, before the valve is replaced, its design should be reviewed to determine if the valve packing box can be upgraded to an improved packing or a live-loaded configuration. Overall, upgrades are more cost effective than purchasing a newer design. However, the upgrade may affect valve performance with more stem friction that can create sticking or erratic stroking. Upgrading the valve also means that continual monitoring is required during a period of break-in. Maintenance costs will also increase.

Because the packing box is the valve's primary dynamic seal it usually receives the most attention rather than the static seals (body, bonnet, and flange gaskets). One criteria for the new packing-box design should be its ability to compensate for packing consolidation, which occurs when the packing volume is reduced by wear, cold flow, plastic deformation, or extrusion. When packing is first installed, a certain amount of space can be found between the rings. As the packing is compressed to form a seal, these gaps slowly collapse. As packing loses its seal through friction, more force is applied to once again pro-

vide a seal. After several tightenings, all available space between the rings is exhausted. The packing is now one solid block and is incapable of further compression. When continued force is applied to consolidated packing, if the packing is soft and fluid, it may extrude up or down the stem or shaft. A photograph of packing that has extruded is found in Chap. 2 (Fig. 2.41).

Since 1990 when the Clean Air Act was amended, valve manufacturers introduced a number of packing-box designs that comply with the EPA requirements, many of which can be upgraded or retrofitted into existing valves. In nearly all cases, the costs associated with upgrading an existing valve are far less than installing a new valve. The following criteria should be evaluated before determining if a valve can be upgraded to an EPA-compliant packing. First, the user should ensure that the upgrade can be accomplished easily, safely, and economically. In some cases, the valve can remain in the line while the retrofit takes place—although the line should be drained and decontaminated, if necessary, for safety reasons. In some cases, the retrofit procedure may be so complicated that the valve must be sent to the factory or an authorized repair center for the conversion. This may present a problem if the valve is a critical final element of the system or if a replacement valve is not available. Second, the user should ensure that the upgraded packing box will meet the 500-ppm standard without continual packing readjustments. In addition, the packing box should continue to perform under the 500-ppm standard for long periods of time. Third, some consideration should be given to whether the upgraded packing-box design requires new maintenance procedures or installation equipment (which may require additional training for maintenance personnel). The best solution, and the least costly, is an upgrade that permits using the original bonnet, body, stem, or shaft. If live-loading is necessary, space for the fasteners and live-loading mechanism must be available above the bonnet or body. In some cases, an upgrade requires a new bonnet for linear valves or a new body for rotary valves. Unfortunately, the introduction of these expensive new parts often increases costs so much that an overall new valve is the best option.

A careful review of an existing valve's packing-box features should be conducted to reveal upgrade possibilities and the probability of success. Some packing-box designs have features that are better suited for upgrading, while others have features that may result in leakage or premature failure. A number of design features improve the likelihood of success in upgrading packing boxes. Bonnets manufactured from forgings or barstock inherently seal better than bonnets made from

castings. Although less expensive, bonnets made from castings may have minuscule cracks or porosity, which sometimes cannot be detected during manufacture and inspection without use of a dye penetrate test. The problem with these minute cracks or porosity is that leakage from these avenues cannot be halted by tightening the packing. Double-top stem guiding is commonly used in linear-motion valves to contain the packing with both the top and bottom guides. This arrangement provides a concentric and constant alignment between the plug stem and the bonnet bore. The lower guide also acts as a barrier against particulates or other impurities, which may affect the integrity of the packing. Double-top stem guiding also avoids the problems inherent to caged-guided trim, which may lead to increased fugitive emissions. The longer distance between the two guiding elements (the upper guide and the cage) allows column loading and stem flex. Plug stems with small diameters can create side loading in the packing box and possible leakage. Because the packing box itself lacks a bottom guide, particulates in the fluid can damage the “wiper” set of packing.

Deep packing boxes are designed to allow for a wider separation of upper and lower guides in the double-top stem guiding design, which provides accurate guiding of the plug head into the seat. Regarding fugitive emissions, a side benefit of a deep packing box is that it allows the upper set of packing to be completely separated from the lower set, which is designed to protect and “wipe” the fluid medium from the plug stem. This wide spacing of the packing sets avoids contact with any part of the plug stem exposed to the flowing medium. Shallow packing-box designs permit the exposed plug stem to contaminate the upper seal. A buildup of process material could also damage the dynamic seal between the stem and packing.

Packing works best with a highly polished plug stem or shaft. A typical plug stem or shaft will be approximately $8\text{ }\mu\text{in}$ root mean squared (rms). On the other hand, a static seal (such as a bonnet bore in a linear valve or a body bore in a rotary valve) would be designed with a surface finish of $32\text{ }\mu\text{in}$ rms.

If the application requiring low fugitive emissions can utilize either a linear or rotary valve, a rotary valve may be the best choice. Because of the circular action of the ball or disk, the seal between the packing and the shaft travels around the shaft circumference instead of linearly up the shaft. This shorter action produces less friction and wear and in the long term promotes packing life. Additionally, consolidation of the packing is far less because the individual rings are stressed in a tangential direction rather than an axial direction.

9.9.5 Live-Loading

Live-loading is often installed to apply a constant packing load without requiring continual retightening of the packing bolting. Live-loading is designed to compensate for packing load losses due to consolidation as well as thermal contraction and expansion. If space exists between the gland flange and the top-works of the valve, live-loading can be retrofitted on most linear and rotary valves. As illustrated in Fig. 9.58, a typical live-loading design uses disk springs above the packing flange to provide a constant load to the packing when properly torqued. The typical disk spring is a metal washer, with the inside diameter formed so that it rises higher than the outside diameter. Two disk springs are placed from inside diameter to inside diameter and stacked with other sets, allowing for a springlike configuration. Disk springs are normally made from corrosion-resistant stainless steel, although Inconel is sometimes used for highly corrosive environments.

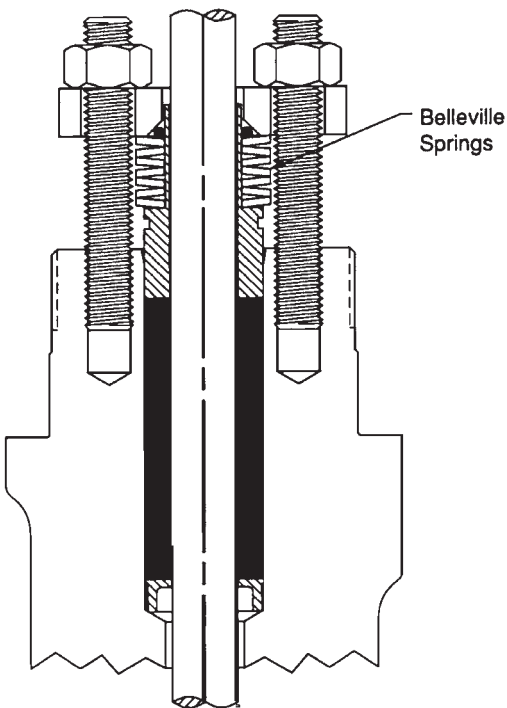


Figure 9.58 Conventional live-loading design with single stack of disk springs (Courtesy of Fisher Controls International, Inc.)

In live-loading, the disk springs are compressed by the gland-flange bolting, allowing a certain percentage of possible travel (typically 80 to 85 percent). As the packing volume decreases due to extrusion or friction, the disk spring's action continues to provide a load to the packing without retorquing. This is especially important since most packings can lose at least 0.02 in (0.5 mm) during the early stage of compression. Without live-loading this height loss would result in the relaxation of the packing and eventual leakage, unless the user retightens the packing. The use of live-loading compensates for this first initial loss in height. As packing settles over time, causing the springs to return to their natural position, the spring force will decrease slightly. However, the overall loss is so low that the seal is not normally affected. The amount of force applied by the live-loading can be controlled by the type of disk spring as well as the compression of the disk spring.

In addition to the reduced need for retorquing, live-loading is ideal for applications in which thermal cycling is a problem. With normal packing configurations, if the packing is tightened when the temperature is high, the packing will leak when the temperature lowers. If the packing is tightened when the temperature is low, the stem or shaft may grab or stick due to thermal expansion when the temperature increases.

Live-loading has other disadvantages than the initial cost as well as the acquisition and installation of new parts. With some valves, little or no room exists between the packing box and the top-works of the valve for upgrading to live-loading, although some manufacturers provide special live-loading configurations for limited space applications, as shown in Fig. 9.59. This design uses an upper plate as the gland flange and a lower plate as the packing compressor with stacks of disk springs located on the outside fringes of the two plates.

The torque values provided by the manufacturer to maintain the proper spring compression of the washers may be affected by the condition of the bolting. If the bolting is new and lubricated, the resulting torque value will be much different than if the threads are corroded and nonlubricated. Some packings may not respond to live-loading as well. For example, because of its high density, graphite packing requires a greater load than the manufacturer specifies for normal packings. If the live-loading is placed in a corrosive atmosphere, the disk springs can also lose strength through corrosion or even bond together, restricting free movement of the disk springs.

Some users argue that the use of live-loading actually contributes to early failure of packing through extrusion by applying more force to the packing than is required to achieve an adequate seal. If extrusion

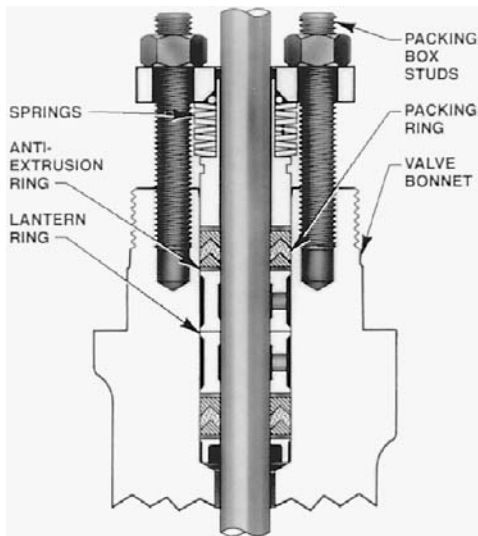


Figure 9.59 Live-loading design used for limited space applications. (Courtesy of Fisher Controls International, Inc.)

occurs, after some time the packing box will lose so much packing material that a seal will not be possible. Consequently, if live-loading is desired, antiextrusion rings should be included inside the packing box, especially if a soft packing is used. Too much compression may also be the problem. In that case, a thinner disk spring (which will apply less force) can be specified.

Another argument against live-loading is that, unless the live-loading provides equal amounts of force on the packing, it can cause stem-alignment problems with linear valves. This can occur if tolerance buildup occurs on some disk-spring stacks and not others, causing an unbalanced packing load and slightly affecting the stem alignment (especially with extremely thin stems or shafts which can flex). Such misalignment can affect both the shutoff and packing seal. This may be remedied, however, by using stem guides that have close-fitting guide liners or by using linear valves with oversized stems.

9.9.6 Metal-Bellows Seals

As a safety measure to workers and the general community, hazardous and corrosive applications must not be allowed to leak any fugitive emissions. In some toxic or lethal processes, however, the

Environmental Protection Agency (EPA) can designate a portion of the plant as a nonattainment area where a small amount of fugitive emissions are allowed. If a process is expanded to include more line penetrations, the parameters of the nonattainment area are often not easily expanded by regulations; therefore the user must not introduce new fugitive emissions. In this case, valves that are incapable of leaking are often required.

Linear valves equipped with a standard packing box always present a risk of leakage. When zero leakage is required, a *metal-bellows seal* is usually specified. A typical metal-bellows seal design contains the fluid with a specially formed metal-bellows welded to the stem of the plug. As shown in Fig. 9.60, the bellows is designed to expand or contract with the linear stroke of the valve, while providing a solid, permanent barrier between the fluid medium in the body and any potential leak paths to atmosphere. A metal bellows presents the best solution against fugitive emissions, as long as the body gaskets hold

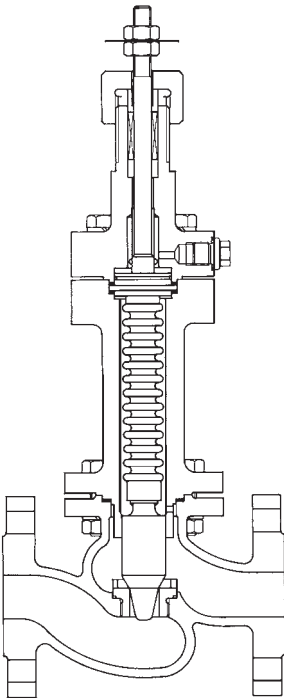


Figure 9.60 Hydroformed bellows with extended bonnet design. (Courtesy of Kammer Valves)

their static seals and the metal-bellows seal remain intact. Valves equipped with bellows seals do have some limitations, such as shorter strokes, decreased stroking life, and greater height. Valves with bellows seals can also cost 20 to 40 percent more, although that cost is offset by less monitoring and packing maintenance.

In throttling applications, the bellows is welded to the stem in the middle of the stroke. In the middle of the stroke the bellows is in a “relaxed” state and is equally stretched at the full-open and full-closed positions. This maximizes the life of the bellows. In applications in which a majority of the throttling is done between the 25 and 75 percent range, a bellows life of up to 200,000 strokes is possible. If a full stroke is required (0 to 100 percent), the life drops dramatically—up to 60,000 strokes. On the other hand, in applications in which the valve remains shut (or wide open) for a good portion of the time, the bellows can be welded at different locations in the plug. The bellows stays in the relaxed position for a majority of the service, prolonging its life. A metal-bellows cycle life is expressed as the number of times that the bellows can be stretched to its full limit and then compressed without failure. Because a full cycle involves a complete expansion and contraction, a bellows rated at 10,000 cycles actually translates into 20,000 full valve strokes. Because throttling service may not require a full-open or full-closed position, the bellows may be stretched or compressed less than a full stroke, which will further prolong the bellows life. The bellows life can also be prolonged by changing the tuning setting on the process controller. Process controllers can be so highly tuned that they continually search for the correct signal, sending minute signals to the valve that varies in position with each signal. Although minimal, this continual movement of the valve will shorten the overall life of the bellows. The rated bellows life number is determined by the minimum number of cycles that a bellows can withstand at the maximum operating temperature and pressure. Although a bellows is designed for the operating services, the actual operating conditions are usually less than the maximum temperature and pressure, which further prolongs bellows life. This means that the bellows life can be many more times than expected. Some applications require minimal stroke travel in a service with lower-than-rated service conditions. For example, a bellows rated at 10,000 cycles can provide beyond 100,000 strokes, given the right conditions. Table 9.14 shows how reducing the stroke by half significantly prolongs the life of the bellows.

Bellows life is also dependent on the process pressures that act on the bellows. Bellows can be designed to allow the process fluid to be

Table 9.14 Bellows Cycle Life*

Valve Size	Maximum Bellows Travel	Half Stroke (cycles)	Full Stroke (cycles)
0.5-inch DN 12	0.56 inches 1.42 cm	1,400,000	150,000
2-inch DN 50	0.84 inches 2.13 cm	1,400,000	150,000
3-inch DN 80	1.12 inches 2.84 cm	700,000	300,000
4-inch DN 100	1.50 inches 3.81 cm	450,000	100,000

*Data courtesy of Fisher Controls International, Inc.

Note: Data based on single-wall formed bellows, Inconel 625 material, 100°F (39°C), and 150 psig (10.3 bar).

contained in the inside or on the outside of the bellows. However, because a bellows is harder to compress externally than to expand internally, external pressure can double the life of the bellows. A bellows typically handles process pressures from 250 to 550 psi (17.2 to 37.9 bar). It can also be designed with up to four walls, ranging in wall thickness from 0.004 to 0.006 in (0.1 to 0.15 mm)—depending on the pressure and temperature ratings. Multiwall designs provide longer cycle life, because the multiple walls all share the stress of the process pressure instead of a single wall bearing the entire stress of the pressure. Multiple walls also allow for higher pressures over single-wall designs, as shown in Table 9.15.

Although many standard bellows are designed for pressures between 250 and 550 psi (between 17.2 and 37.9 bar), severe service bellows can be designed for pressures up to 3800 psi (262 bar) and temperature ranges from −320 to 1000°F (−195 to 535°C). Both high temperatures and pressures can affect the cycle life of the bellows, as is shown in Fig. 9.61. As a safety measure, bellows-seal valve manufac-

Table 9.15 Pressure Ratings for Single- and Double-Wall Bellows*

Valve Size	Bellows Walls	100° F 38° C	300° F 149° C	500° F 260° C	800° F 427° C
0.5 - 2-inch DN 12 - 50	Single	550 psig 38 bar	497 psig 34 bar	429 psig 30 bar	396 psig 27 bar
0.5 - 2-inch DN 12 - 50	Double	1,000 psig 69 bar	870 psig 60 bar	780 psig 54 bar	720 psig 50 bar
3 - 4-inch DN 80 - 100	Single	346 psig 24 bar	296 psig 20 bar	265 psig 18 bar	245 psig 17 bar
3 - 4-inch DN 80 - 100	Double	625 psig 43 bar	544 psig 37 bar	488 psig 34 bar	450 psig 31 bar

*Data courtesy of Fisher Controls International, Inc.

turers usually pressure-test each bellows seal at or over the rated service pressure.

Because of corrosion or erosion problems, the bellows is not normally placed in direct contact with the fluid; instead, it is placed just outside the flow stream, usually above the plug. A hole or a number of holes are used to allow the process fluid and pressure to bleed either to the outside or inside of the bellows. One problem that can occur with bellows pressurization is that process fluid leaving the flow stream may enter the area next to the bellows, where it cools and thickens. This can cause maintenance problems or undue bellows fatigue. In this case, external pressurization is preferred (Fig. 9.62), since cleaning the outside surfaces of a bellows during maintenance is much easier. Larger bleed holes can allow more liquid to circulate around the bellows and prevent the fluid from cooling.

Two types of metal bellows are in general use today and each is classified by its method of manufacture. *Welded bellows* (Fig. 9.63), also referred to as *diaphragm bellows*, are fabricated using a series of flat

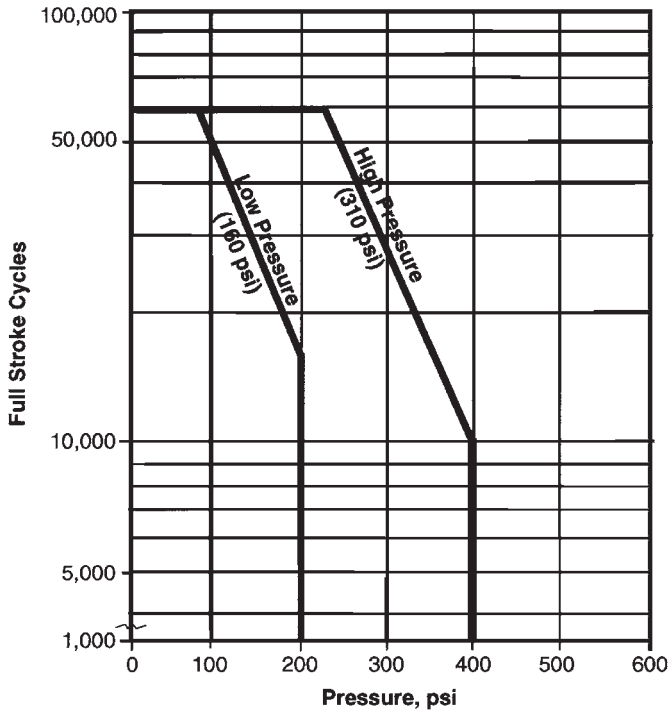


Figure 9.61 Full-stroke cycle life according to pressure.
(Courtesy of Valtek International)

rings that are joined at the outside diameter and inside diameter by a fillerless tungsten inert gas (TIG) weld, creating a series of uniform convolutions. These convolutions have the general appearance of an accordion. Because welded bellows are made from flat rings, the overall height is quite compact and therefore can be contained in a relatively small area, adding only minimal height to the valve. For those applications requiring a small stroke, bellows can be contained inside the body (Fig. 9.64). This is particularly important where space consideration is critical or where seismic requirements limit the height of the valve's top-works. A primary disadvantage of the welded bellows is the welded edges of each convolution, which are easily stressed during expansion or contraction and are usually the first area to fail. Another problem can occur when particulates or solid matter becomes caught in the tight crevices of the convolutions. When this happens, these solids can create stress points in the convolutions and can cause premature failure. Welded bellows are also susceptible to corrosion

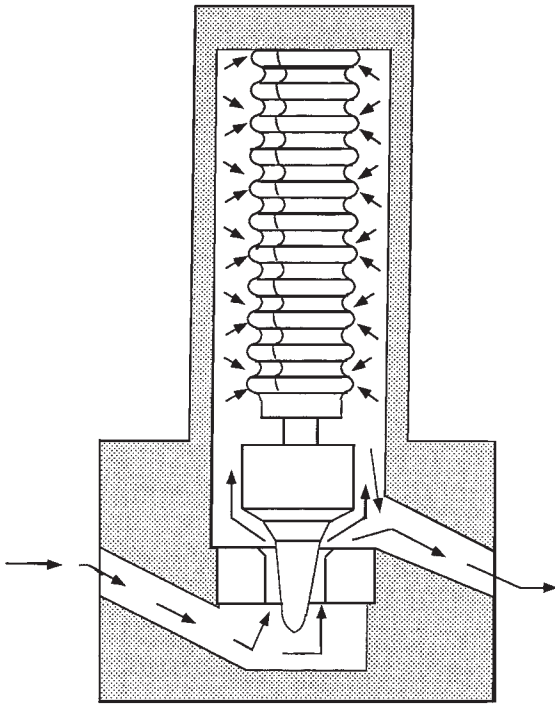


Figure 9.62 External pressurization of bellows.
(Courtesy of Kammer USA)

because of the thin plate used in manufacture, especially when process fluid is continually trapped in the crevices. In addition, due to the difficulties associated with welding some alloys, material selection is limited. Because of the welded edges of the convolutions, the outside diameter of the bellows may restrict their use with some valve styles.

Hydroformed bellows (again refer to Fig. 9.63) is made from a flat metal sheet, which is rolled and fusion welded for solid construction. This tube is then mechanically or hydraulically pressed to create a series of uniform corrugations. More space is required for a complete corrugation—up to three times longer than a single convolution of a welded bellows. For this reason, hydroformed bellows are much longer than welded bellows for the same stroke length. They are encased inside an extended bonnet and have a greater height than normal valves (refer again to Fig. 9.60). One important advantage of the rolled construction is that process matter does not become entrapped in the folds, as is the case with welded bellows. Generally, formed bellows last longer than welded bellows because of the minimal welding,

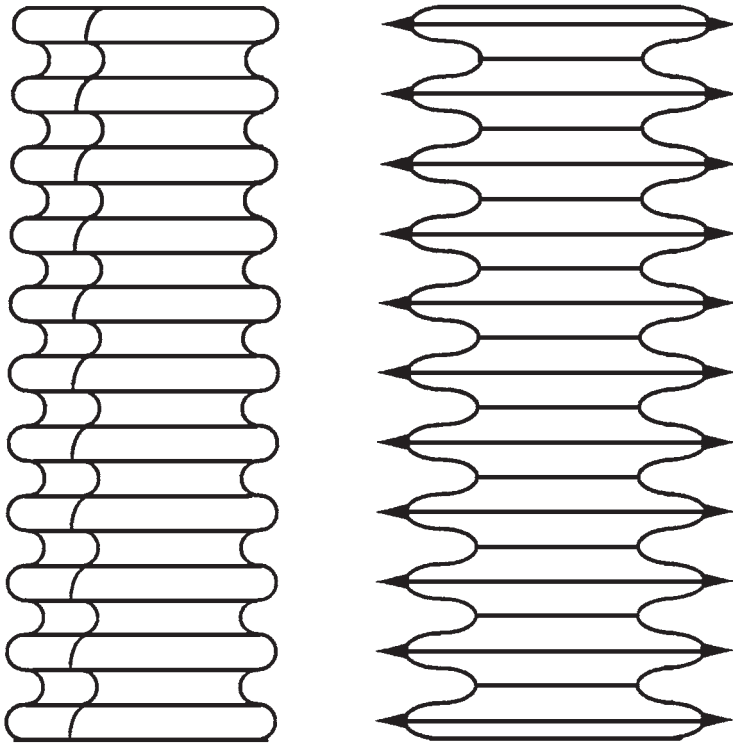


Figure 9.63 Hydroformed (left) and welded bellows. (Courtesy of Kammer USA)

the overall strength of the corrugations, and the limited travel of each fold (as compare to welded flat rings). They also handle higher pressures because of their greater strength. The main disadvantage is that formed bellows must be three times longer than welded bellows to handle the same stroke. The longer length may present problems with upgrading if space restrictions or seismic limitations exist.

In most designs, a packing box is placed above the bellows as a backup, in case the bellows ruptures from mechanical failure. To provide a warning of a bellows failure, a “telltale tap” can be installed in the bonnet, which is connected to an alarm system. Although not fail-proof, a metal-bellows seal provides the most reliable seal against leakage to atmosphere. Bellows can be made from a number of different materials, depending on the application, but 300 series stainless steels, Inconel, or Hastelloy C are standard materials because of their ability to resist stress fatigue and corrosion. Bellows can also be made from titanium, nickel, or Monel.

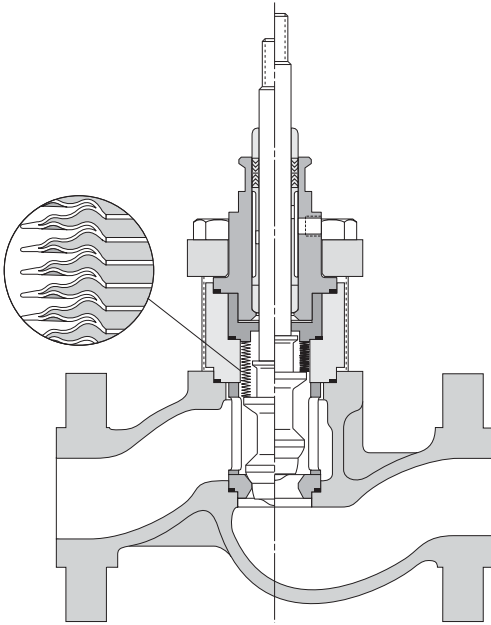


Figure 9.64 Body-contained welded bellows.
(Courtesy of Valtek International)

A metal-bellows sealed valve may be seal-welded between the bonnet and the body as a precautionary measure with lethal or highly toxic services.

Because bellows seals are highly complex, retrofitting a linear valve is equally complex as well as very costly. In most cases, a new bonnet, plug or bellows assembly, and housing must be acquired, which can cost more than a new metal-bellows sealed valve.

9.9.7 Packing-Box Issues

When valves are initially installed in service, their packing boxes normally meet fugitive-emissions requirements. However, over time with continual operation, the packing will consolidate somewhat and begin to leak, requiring retightening of the gland-flange bolting. Most packing boxes will require retightening over time, until the packing reaches full compression. Further retightening only results in crushing the packing, rendering it useless. Manufacturers often provide suggested torque rates for given packing-box designs. This torque is applied to

the gland-flange bolting, which in turn compresses the gland flange against the packing guide, finally resulting in full compression of the packing. Because of the problems associated with exact torque measurements, some designs have been simplified with the packing bolting tightened to just a flat or two past finger-tight. A manufacturer's recommended torque value can be affected by environmental corrosion or a lack of adequate lubrication, which can cause increased thread resistance and a false torque reading. Ideally, correct packing compression can be determined by measuring the packing's height when uncompressed and then applying torque until the manufacturer's ideal packing height is reached. Normally, the manufacturer's recommended packing height requires a 15 to 30 percent compression.

Maintenance technicians will sometimes overtorque the gland-flange bolting, believing that overcompression is better than undercompression. Unfortunately, too much torque can crush the packing, creating even greater leak paths. Because the packing will be compressed against the stem or shaft, high torques will boost the breakout force, causing an uneven (jerky) stroking motion. Due to the severe nature of the process or wide temperature swings, some applications require retightening often. For example, superheated steam applications may require retightening every few days. If this is not done, the packing box may develop a serious leak and be destroyed quickly by the high temperatures and pressures of the superheated steam.

The issue of torque is related directly to balancing leakage rates versus stem friction. As compression is applied to the packing by an axial load, packing deforms radially, pushing against two surfaces: the wall of the packing box and the stem or shaft. With greater compression, the greater the stress will be applied against these surfaces. As the packing deforms against the wall, any voids are closed off, permitting an effective seal. However, more compression also increases stem friction as the inside diameter of the packing grips the plug or shaft stem. This leads to erratic stem movement. Conversely, if the force to the packing is decreased to allow for smoother stroking, the packing may not fully grip the stem or shaft and leakage can occur.

Another factor that plays an important part in packing-box friction is the amount of contact between the packing and the stem or shaft. As the surface area of the packing touching the sliding stem or rotating shaft increases, more friction is produced that must be overcome to produce movement. High levels of friction will require greater force by the actuator, or a longer lever or larger diameter handwheel with manual valves. Some packings are V-shaped, which provides a very narrow point of contact and generates minimal friction. On the other

hand, square packing (such as graphite) provides full contact and increased packing friction.

Most valve manufacturers today provide highly polished valve plug stems or shafts to accommodate the dynamic seal between the inside diameter of the packing and the stem or shaft. However, valve stroking or the service conditions of the process itself can deform, pit, or corrode the stem or shaft. Such wear can significantly increase the stem friction, while decreasing valve performance. This problem is often compounded when the seal is leaking and additional torque is applied to the packing to stop the leak. During routine maintenance, the stem or shaft should be carefully examined to ensure a smooth surface finish. If the finish is not smooth, that part should be replaced or repaired if the scratches or pits are not too deep.

9.9.8 Packings Specified for Fugitive-Emission Control

Today's packing materials are well suited to control fugitive emissions and can be adapted for retrofitting. Although no packing material or design is universal, many different packings exist that have broader applications than in years past. Choosing the correct packing is critical to the successful performance of the packing box. The packing should be compatible with the process fluid and service temperature and pressures as well as provide the desired seal between maintenance checks, without excessively high torque of the gland-flange bolting. The proper packing should also withstand consolidation and should minimize the friction on the stem or shaft, avoiding poor stroking performance.

A number of packing materials are commonly applied to anti-fugitive-emission packing boxes. Recently introduced in the past several years, perfluoroelastomer (PFE) packing is generally regarded by valve users as the best packing for complying with fugitive-emission standards. PFE provides an excellent seal with even the most difficult application. It resists degradation and chemical attack and is very resilient and elastic. PFE is rated to handle service temperatures from 20 to 550°F. A special low-temperature PFE has been developed that handles temperatures down to -40°F. As shown in Fig. 9.65, PFE requires a rigid backup V-ring system to support the packing. A property unique to PFE is that it wears well and compensates for any consolidation that takes place—although PFE can consolidate and eventually extrude if not supported by backup rings. Live-loading is not normally required, since PFE has an ability to return to its precompressed position. However, in applications with large thermal gradi-

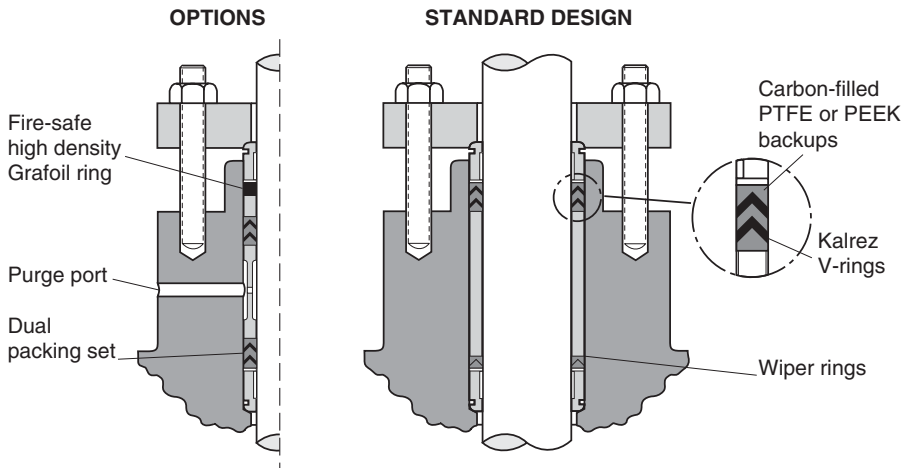


Figure 9.65 PFE backup ring packing configuration. (Courtesy of Valtek International)

ents, live-loading should be considered. The chief disadvantage of PFE is its high cost, although the initial cost of the packing is easily offset by reduced maintenance and increased up-time.

One of the most common and least expensive packings, “virgin” polytetrafluoroethylene (PTFE) packing is typically applied in a V-ring design (see Fig. 2.32 in Chap. 2). Virgin PTFE is often chosen because it has numerous advantages. Due to its pressure-energized design, coupled with “feather” edges, little compression is required to create a strong seal. Overall, virgin PTFE has good elasticity, which minimizes packing consolidation while responding well to a live-loading option. Because it is inert to many chemicals, virgin PTFE is found in a wide range of process services. The surfaces of virgin PTFE are extremely smooth; therefore, little breakout force is required to begin stroking the valve. Despite its wide application, virgin PTFE has some disadvantages. Its performance is limited to temperatures between -20 and 350°F . If the packing bolting is overtightened—providing an excess load on the packing—the voids between the male and female rings can compress and result in consolidation. In addition, the spaces between packing spacers and the plug stem can result in extrusion, although antiextrusion rings or close-fitting spacers can be installed to prevent extrusion. Because of its tendency to cold flow and consolidate over time, virgin PTFE does require retorquing on occasion.

The composition of “filled” PTFE contains 15 to 20 percent glass or carbon, which creates a more rigid V-ring design that is less likely to

produce consolidation (which is common to virgin PTFE). Because its elasticity is less, filled PTFE does not seal as well as virgin PTFE. It also produces greater friction and is slightly abrasive to the stem or shaft. And, it is more expensive than virgin PTFE. Sometimes, as a compromise between virgin and filled PTFE, rings of both materials are alternated in the packing configuration to provide a good seal with reduced consolidation. Live-loading can also be used with filled PTFE to minimize retorquing.

Graphite and other carbon-based packings are commonly manufactured in die-formed or straight braided carbon-ring sets. As a measure against graphite migration, braided rings are often included in die-formed packing sets. This feature also protects the graphite rings from foreign particles. Braided rings are known to cause additional friction and leakage in high-compression applications. The main advantage of graphite packing is its ability to handle high temperatures (up to 800°F with a standard-length bonnet in an oxidizing environment). Graphite packings are usually offered in low-density or high-density graphite. Low-density graphite seals well and has lower friction, but must be retorqued often. High-density graphite has higher friction and provides a marginal seal but allows for a longer retorque cycle. To convert low-density to high-density packing, the packing can be torqued several times over a period of time. Compared to other packings, graphite packings are more expensive and do not respond well to live-loading systems. Also, the higher friction can affect the performance of the valve, requiring high breakout forces that may result in unstable stem movement. Typically, torque requirements for graphite ring packings can be eight to 10 times higher than those of PTFE or PFE packings. This usually requires the use of a torque wrench to ensure that over-compression does not occur. Overcompression will crush the graphite, causing it to extrude from the packing box.

9.9.9 Other Packing Considerations

Some users believe that if using the standard number of packing rings provides a good seal, using more rings should provide an even better seal against fugitive emissions. If a packing box is exceptionally deep, a user may be tempted to double the number of rings during routine maintenance. However, the use of extra-ring compounds several problems. First, multiple rings maximize the adverse affects of thermal expansion of the packing. Second, they increase stem friction substantially. Third, the manufacturer's recommend torque values will now be incorrect, providing far less compression than required. This may

necessitate a trial-and-error approach to determining the correct torque value, which could shorten the life of the packing. Fourth, with more soft packing material in the packing box, unnecessary consolidation and extrusion can take place.

With rotary valves, the closure or regulating element or the actuation unit can apply stresses to the shaft, causing an incorrect center alignment. If a small-diameter stem (linear valves) is used, the force applied by the actuation to the closure element in the seated position can actually flex the stem. Whenever the stem or shaft are off-center with the packing box, a leak path for fugitive emissions can occur on one side. This problem can usually be avoided by using valves that feature oversized stems or shafts. Oversized stems or shafts present a large contact area between the stem or shaft and the packing, which will result in higher friction—although this is not an issue with high-thrust actuation units, such as piston cylinder actuators.