Piping System Fundamentals

The Complete Guide to Gaining a Clear Picture of Your Piping System

SECOND EDITION U.S.

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Chapter Six

Valves and Fittings

Valves and fittings are used to connect pipelines, redirect flow, diverge and converge flow, isolate equipment or parts of a system, or prevent reverse flow in a pipeline. Any valve or fitting installed in a pipeline adds additional resistance to the flow of fluid and therefore the amount of energy dissipated (head loss) and the pressure drop across the pipeline. There is a wide array of types and sizes of valves and fittings.

Many manufacturers perform bench tests on their products to characterize the hydraulic performance of the valves and fittings that they sell. For example, the Crane Valve Company performed a variety of pressure drop tests for a wide range of types and sizes of valves and fittings. The results of this study were published in the Crane Technical Paper 410: Flow of Fluids through Valves, Fittings, and Pipe. For many decades, this publication has served as a key source for engineers needing to calculate head loss for the design, operation, and troubleshooting of piping systems.

Characterizing the Hydraulic Performance of Valves and Fittings

The head loss of a valve or fitting is caused by four factors: the friction between the fluid and the internal surfaces, changes in the direction of the flow path, obstructions in the flow path, and any changes in the cross-sectional size and shape of the flow path.

Because the amount of friction is typically minor compared to the other three factors, the total resistance can be considered independent of the friction factor or Reynolds Number. The Crane Technical Paper No. 410 treats the hydraulic resistance as a constant for any given obstruction under all conditions of flow. Other studies have shown that the resistance increases at lower flow rates, so a correction factor is added based on the Reynolds Number.

Over the years, various methods have been used to characterize the hydraulic performance of valves and fittings. The most common methods are the *Equivalent Length* (*L/D*), *Flow Coefficient* (C_V), and the *Resistance Coefficient* (*K*). More complex methods are available but with marginal improvements.

Equivalent Length (*L/D*)

Valves and fittings are tested under various flow conditions to determine the equivalent length (L_{eqv} in feet) of straight horizontal pipe that will give the same pressure drop that is seen across the component. The equivalent length is typically reported as a ratio of that length of pipe to the inside diameter (*D* in feet). The amount of head loss can then be calculated for a given fluid velocity (*v* in feet/sec) with a modified form of the Darcy Equation, as shown in Equation 6-1.

$$h_L = f \frac{L_{eqv}}{D} \frac{v^2}{2g} \qquad \textit{Equation 6-1}$$

Flow Coefficient (C_V)

Some manufacturers may provide a flow coefficient (C_V) to characterize the hydraulic performance of their equipment. The flow coefficient is determined by measuring the amount of 60 °F water flow (*Q* in gpm) and the pressure drop across the component (P_{in} and P_{out} in psi), then using Equation 6-2. The larger the flow coefficient, the lower the resistance to flow, and therefore the higher the capacity of the component.

$$C_V = \frac{Q}{\sqrt{\frac{P_{in} - P_{out}}{SG}}} \qquad \textit{Equation 6-2}$$

Equation 6-2 is a generalized form of the the flow coefficient equation that is used by control valve manufacturers. A more detailed equation for use with control valves is published in the ISA S75.01 standard, *Flow Equations for Sizing Control Valves*. The concept of the flow coefficient was originally developed in the 1940s by Masonelian to characterize the performance of their control valves.

Other equipment can be characterized by the flow coefficient as well. For example, some heat exchangers, strainers, nozzles, and sprinkler heads use the flow coefficient to define their hydraulic performance.

Other Uses of the Flow Coefficient

The concept of the flow coefficient is a powerful tool in analyzing the performance of equipment in the piping system. It can also be used to characterize the performance of numerous fixed resistance components in a pipeline.

Example 6-1: Using of the Flow Coefficient to Represent the Performance of Fixed Resistance Components **PF**

Consider a portion of a piping system shown in Figure 6-1 with four isolation valves, two elbows, a check valve, a heat exchanger, a venturi flow meter, and various lengths of piping connecting all the components. There are pressure gages measuring the inlet and outlet pressures of the series of fixed resistance components.

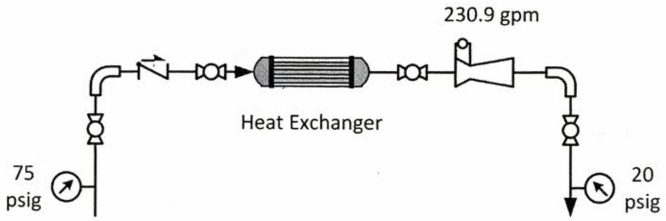
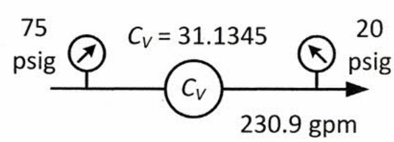


Figure 6-1. Portion of a piping system with fixed resistance components. **PF**

Given the flow rate of 230.9 gpm of water at 60 °F and the inlet and outlet pressures, an equivalent C_V can be calculated to characterize the hydraulic performance of the portion of the system between the two pressure gages, using Equation 6-2:

$$C_V = \frac{Q}{\sqrt{\frac{P_{in} - P_{out}}{SG}}} = \frac{230.9}{\sqrt{\frac{75-20}{1.0}}} = 31.1345$$



This equivalent C_V can be represented graphically as shown in Figure 6-2 as a C_V component in a very short length of pipe (no resistance due to the pipe since this was included in the calculation of the flow coefficient).

Figure 6-2. Graphical representation of the equivalent flow coefficient.

Now consider what happens if the pressure drop across the system is increased to 60 psid by reducing the outlet pressure to 15 psig and increasing the flow while maintaining the inlet pressure at 75 psig, as shown in Figure 6-3.

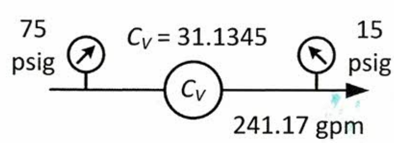


Figure 6-3. Graphical representation of the equivalent flow coefficient at increased flow and dP.

The flow rate can be calculated using the equivalent flow coefficient and the values of pressure at the inlet and outlet, and re-arranging Equation 6-2:

$$Q = C_V\sqrt{\frac{P_{in} - P_{out}}{SG}} = (31.1345)\sqrt{\frac{75-15}{1.0}} = 241.17\ gpm$$

This flow rate would be seen on the flow meter, as shown in Figure 6-4.

One note about using an equivalent flow coefficient to represent fixed resistance components. Because of the square root in Equation 6-2, the value of the flow coefficient is sensitive to the number of significant digits used. To increase the accuracy of calculations done with the equivalent flow coefficient, increase the number of significant digits in its numerical value.

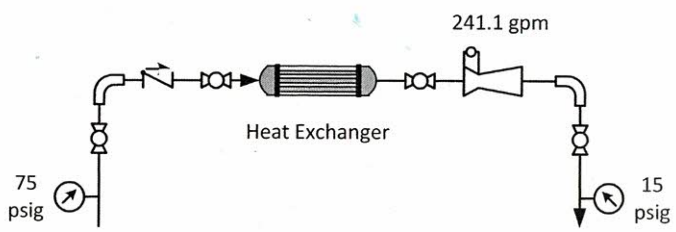


Figure 6-4. Increased pressure differential results in an increased flow rate. **PF**

Adding Flow Coefficients

The flow coefficients of components in series and parallel can be added together to obtain an equivalent flow coefficient that represents the performance of the numerous devices.

Equation 6-3 can be used to calculate the equivalent flow coefficient for *n* components in series.

$$\frac{1}{C_{v\,Total}^2} = \frac{1}{C_{v\,1}^2} + \frac{1}{C_{v\,2}^2} + \cdots \frac{1}{C_{v\,n}^2} \qquad \text{Equation 6-3}$$

Equation 6-4 can be used to calculate the equivalent flow coefficient for *n* components in parallel.

$$C_{v\,Total} = C_{v_1} + C_{v_2} + \cdots C_{v_n} \qquad \text{Equation 6-4}$$

Resistance Coefficient (*K*)

Another way to characterize the hydraulic performance of valves and fittings is with the resistance coefficient (*K*). The resistance coefficient combines the equivalent length with the completely turbulent friction factor of clean commercial steel pipe, as shown in Equation 6-5.

$$K = f_T \frac{L}{D} \qquad \text{Equation 6-5}$$

Given the value of the resistance coefficient, the amount of head loss can be calculated for a given fluid velocity using Equation 6-6.

$$h_L = K \frac{v^2}{2g} \qquad \text{Equation 6-6}$$

Using the more common units of flow rate in gallons per minute, Equation 6-7 can be used to calculate the head loss for a given resistance coefficient using the inside diameter (*d* in inches) of the connecting piping.

$$h_L = 0.00259 \frac{KQ^2}{d^4} \qquad \text{Equation 6-7}$$

The resistance coefficient is the opposite way to view hydraulic performance compared to the flow coefficient. Where the flow coefficient describes how much flow capacity the valve or fitting allows, the resistance coefficient describes how much resistance it offers to the flow. Equation 6-8 shows the mathematical relationship between the two concepts.

$$K = 891 \frac{d^4}{C_V^2} \qquad \text{Equation 6-8}$$

The Crane Technical Paper No. 410 treats the resistance coefficient as constant for all flow conditions since the flow is in the transitional or fully turbulent fluid zone in most industrial systems, but other hydraulic references apply a correction factor to increase the K value at low Reynolds Numbers.

Adding Resistance Coefficients

Just as with the flow coefficients, the resistance coefficients of components in series and parallel can be added together to obtain an equivalent value that represents the performance of the numerous devices together.

Equation 6-9 can be used to calculate and equivalent resistance coefficient for *n* components in series.

$$K_{Total} = K_1 + K_2 + K_3 + \cdots K_n \qquad \textit{Equation 6-9}$$

Equation 6-10 can be used to calculate and equivalent resistance coefficient for *n* components in parallel.

$$\frac{1}{\sqrt{K_{Total}}} = \frac{1}{\sqrt{K_1}} + \frac{1}{\sqrt{K_2}} + \frac{1}{\sqrt{K_3}} + \cdots \frac{1}{\sqrt{K_n}} \qquad \textit{Equation 6-10}$$

Determining the Turbulent Friction Factor

For calculating the resistance coefficient using Equation 6-5, the completely turbulent friction factor for clean commercial steel pipe is used as the scaling factor for the valve and fitting size.

Table 6-1 shows the completely turbulent friction factor for various sizes of valves and fittings. These values are obtained by using the Swamee-Jain equation, Equation 5-9, and taking the Reynolds Number to infinity so that it drops out of the equation. Equation 6-11 shows the resulting equation that can be used to calculate the turbulent friction factor.

$$f_T = \frac{0.25}{\left[log\left(\frac{\varepsilon}{3.7d}\right)\right]^2} \qquad \textit{Equation 6-11}$$

The turbulent friction factor is the value on the Moody diagram where the friction factor becomes constant at sufficiently high Reynolds numbers.

Notice that as the pipe size increases, the turbulent friction factor decreases. In other words, larger valves offer less resistance to flow than smaller valves.

Table 6-1: Turbulent Friction Factor for Various Valve and Fitting Sizes

Nominal Size	f_T
½"	0.026
¾"	0.024
1"	0.022
1 ¼"	0.021
1 ½"	0.020
2"	0.019
2 ½"	0.018
3"	0.017
4"	0.016
5", 6"	0.015
8"	0.014
10-14"	0.013
16-22"	0.012
24-36"	0.011

Resistance of Valves and Fittings

Valves and fittings are selected to perform a certain function in the piping system, whether it is to isolate equipment, redirect flow, or prevent reverse flow. When installed, they add additional resistance to the flow of fluid through the pipeline. The resistance of some fittings depends on size, for others the resistance is only a function of geometry.

The following graphics and resistance values are taken from the Crane Technical Paper No. 410, with permission.

Reducers and Enlargers

Reducers (Figure 6-5) and enlargers (Figure 6-6) are used to connect pipelines of different sizes. When the fluid passes through the fitting, a change in fluid momentum occurs, resulting in an energy loss (head loss), and an associated pressure drop.

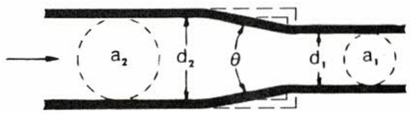


Figure 6-5. Reducer (courtesy of Crane Valve Co.)

The amount of resistance offered by the change in pipe size depends on how much the pipe diameter changes (beta ratio, β = the smaller pipe diameter divided by the larger pipe diameter), how quickly the change occurs (indicated by the angle of approach, θ, and approach length, L) and in which direction the fluid flows.

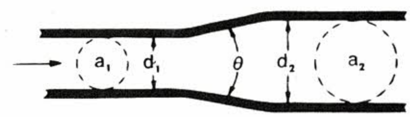


Figure 6-6. Enlarger (courtesy of Crane Valve Co.)

For a reducer or enlarger specified by dimensions d_2 x d_1 x L, the angle of approach can be calculated using trigonometry and is given by Equation 6-12.

$$\theta = 2\,tan^{-1}\left(\frac{d_2 - d_1}{2L}\right) \qquad \textit{Equation 6-12}$$

Equation 6-13 gives the resistance coefficient (K_2) for reducers with an angle of approach $\theta \leq 45°$, expressed in terms of the fluid velocity in the larger pipe size.

$$K_2 = \frac{0.8\left(sin\frac{\theta}{2}\right)(1-\beta^2)}{\beta^4} \qquad \textit{Equation 6-13}$$

Equation 6-14 gives the resistance coefficient (K_2) for enlargers with an angle of approach $\theta \leq 45°$, expressed in terms of the fluid velocity in the larger pipe size.

$$K_2 = \frac{2.6\left(sin\frac{\theta}{2}\right)(1-\beta^2)^2}{\beta^4} \qquad \textit{Equation 6-14}$$

Consult the Crane Technical Paper No. 410 for the appropriate equations for reducers and enlargers with the angle of approach $\theta > 45°$.

The numerator in Equations 6-13 and 6-14 expresses the resistance coefficient in terms of the fluid velocity in the smaller pipe, designated as K_1. In other words, if the head loss is calculated using Equation 6-6, the K_2 value would be used if the calculation uses the velocity in the larger pipe. The K_1 value would be used if the head loss is calculated with the velocity of the smaller pipe.

Pipe Entrances and Exits

The change in fluid momentum that occurs at a tank entrance or exit also adds resistance to flow.

The losses for the pipe entrances from a tank into the pipeline shown in Figure 6-7 vary based on the smoothness of the flow transition. A pipe projecting into a tank has the highest loss, and losses decrease if the entrance is sharp-edged and as the entrance becomes more rounded.

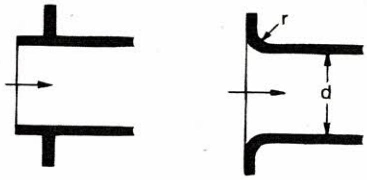


Figure 6-7. Pipe entrances: protruding (left) and bell-mouthed with rounding using r/d (right).(courtesy of Crane Valve Co.)

For the case in which the pipe protrudes into the tank, as shown in left-hand drawing in Figure 6-7, the resistance coefficient, $K = 0.78$, regardless of the pipe size.

For pipe entrances that are bell-mouthed, as shown in the right-hand drawing of Figure 6-7, Table 6-2 shows the resistance coefficient as a function of the r/d value. For sharp-edge pipe entrances, the r/d value is zero and the resistance coefficient, $K = 0.5$, regardless of the pipe size.

For all fluid exits from a pipeline into the tank, as shown in Figure 6-8, the resistance coefficient, $K = 1.0$ for all designs because this takes into account the fact that the fluid will slow down from the velocity it has in the pipeline to zero feet/sec in the tank.

Table 6-2: Sharp-edged and Bell-mouthed Pipe Entrance Resistance Coefficients

r/d	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 and up	0.04

*sharp-edged entrance

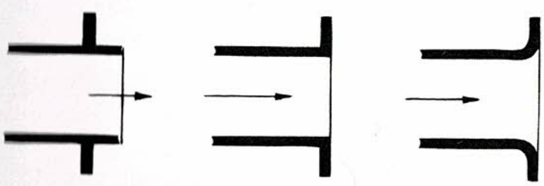


Figure 6-8. K = 1.0 for all pipe exits into a tank: protruding (left), sharp edged (center), and bell-mouthed (right). (courtesy of Crane Valve Co.)

Elbows and Bends

The losses for elbows and bends are caused by a change of fluid momentum when the fluid changes direction as it flows through the bend. The larger the change in direction (i.e. the larger the angle), the greater the change in fluid momentum, the greater the resistance offered by the elbow, and therefore the greater the head loss. The resistance due to the distance traveled through the elbow is also included in the head loss.

Standard Elbows

For the standard elbows shown in Figure 6-9, the resistance coefficient depends on the angle and the pipe size of the elbow.

For 90° standard elbows, the resistance coefficient is calculated using Equation 6-15. The

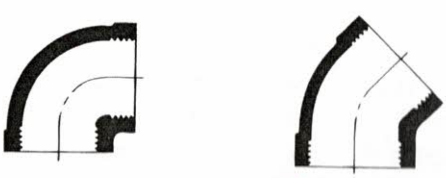


Figure 6-9. Standard threaded elbows: 90° (left), 45° (right). (courtesy of Crane Valve Co.)

constant in Equation 6-15 is the *L/D* value. The turbulent friction factor (f_T) is determined by the elbow size using Table 6-1.

$$K = 30\ f_T \qquad \textit{Equation 6-15}$$

For 45° standard elbows, the resistance coefficient is calculated using Equation 6-16. Notice that the *L/D* value is smaller than for the 90° elbow, which means that for the same size, the 45° elbow has less resistance to flow and less head loss than the 90° elbow because there is a smaller change in fluid momentum.

$$K = 16\ f_T \qquad \textit{Equation 6-16}$$

90° Pipe Bends and Flanged or Butt-Welded 90° Elbows

For 90° pipe bends that are flanged or butt-welded, as shown in Figure 6-10, the resistance coefficient depends not only on the pipe size, but also on the r/d of the bend. Table 6-3 shows the equations to calculate the resistance coefficient based on the r/d value. Notice that the *L/D* value initially decreases with increasing r/d for r/d < 3 as the change is more gradual, but then increases with increasing r/d above 4. This occurs because the length of pipe required to make a 90° turn increases with higher r/d, which adds more resistance to the flow.

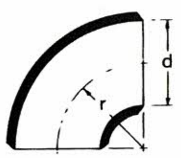


Figure 6-10. Flanged or butt-welded 90° elbow (courtesy of Crane Valve Co.)

Table 6-3: Resistance Coefficients for Flanged or Butt-welded 90° Elbows as a Function of r/d

r/d	K	r/d	K
1	$20\ f_T$	8	$24\ f_T$
1.5	$14\ f_T$	10	$30\ f_T$
2	$12\ f_T$	12	$34\ f_T$
3	$12\ f_T$	14	$38\ f_T$
4	$14\ f_T$	16	$42\ f_T$
6	$17\ f_T$	20	$50\ f_T$

Close Pattern Return Bends

For the 180° close pattern return bend shown in Figure 6-11, the resistance coefficient is dependent on the pipe size and can be calculated with Equation 6-17.

$$K = 50\ f_T \qquad \textit{Equation 6-17}$$

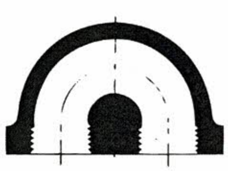


Figure 6-11. Close pattern return bend (courtesy of Crane Valve Co.)

Mitre Bends

The resistance coefficient of a mitre bend shown in Figure 6-12 depends on the mitre angle and the pipe size. Table 6-4 shows the equations to calculate the resistance coefficient of the bend as a function of the mitre angle. Notice that the greater the mitre angle, the larger the *L/D* value.

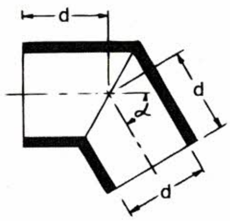


Figure 6-12. Mitre bend (courtesy of Crane Valve Co.)

Table 6-4: Resistance Coefficients for Mitre Bends as a Function of Angle

α	K
0°	2 f_T
15°	4 f_T
30°	8 f_T
45°	15 f_T
60°	25 f_T
75°	40 f_T
90°	60 f_T

Isolation Valves

Isolation valves are used to start and stop flow and to isolate a component or part of the piping system. There are a variety of types of isolation valves to choose from and the amount of resistance they offer when in the fully open position varies with their design.

The resistance increases when there are restrictions in the cross sectional area through the valve, the valve disk remains in the flow stream, or there are changes in direction of flow within the valve.

Ball Valves

A ball valve consists of a spherical ball with a hole drilled through it to allow passage of the fluid. A ball valve is a quarter turn rotary valve that goes from fully open to fully closed with a quarter turn of the handle. The advantage of a ball valve is that it offers the lowest resistance to flow of any standard valve design when fully open.

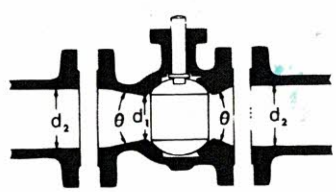


Figure 6-13. Reduced ported ball valve (courtesy of Crane Valve Co.)

The valve shown in Figure 6-13 is a reduced ported ball valve, also known as a venturi ball valve, in which the diameter of the port is smaller than the diameter of the connecting piping ($\beta < 1.0$). A full ported ball valve is one in which the port diameter is equal to the connecting pipe diameter.

The resistance coefficient for a fully open full port ball valve is a function of the valve size and can be calculated with Equation 6-18. The Crane Technical Paper No. 410 should be consulted for calculating the resistance coefficient for reduced ported ball valves.

$$K = 3\, f_T \qquad \textit{Equation 6-18}$$

Ball valves are used for on/off isolation valves and can be used for moderate throttling, but do not have the best throttling characteristics. They can be used in applications with non-abrasive liquids or gases, slurries, and chemicals in piping systems that are pressurized or under a vacuum.

Gate Valves

A gate valve is a linear motion valve that has a flat disk that is inserted and withdrawn from the valve seat perpendicular to the direction of the flow. When the valve is open, the valve disk is fully removed from the flow stream through the valve. A gate valve can also be full ported or reduced ported, as shown in Figure 6-14

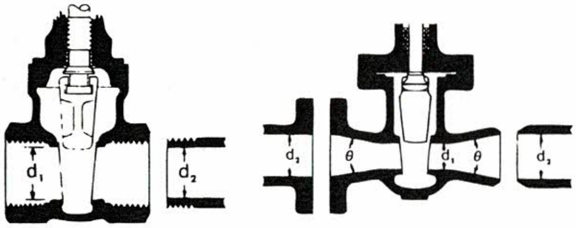


Figure 6-14. Full port and reduced ported gate valves (courtesy of Crane Valve Co.)

Gate valves also have a very low resistance to flow because the disc is completely removed from the flow stream when the valve is fully open. The resistance coefficient for full ported gate valves can be calculated with Equation 6-19. The Crane Technical Paper No. 410 should be consulted for calculating the resistance coefficient for reduced ported gate valves.

$$K = 8\, f_T \qquad \textit{Equation 6-19}$$

Gate valves are primarily used for on/off isolation valves because they have extremely poor throttling characteristics. They are used in liquid and gas applications, for powders, or with slurries that have entrained solids. They can be used in pressurized systems or systems under vacuum.

Gate valves are difficult to open when there is a high differential pressure across the valve. Large gate valves typically have a bypass valve installed for system startup to equalize the pressure across the valve prior to opening. Equalizing the differential pressure by "cracking open" the gate valve could lead to excessive erosion of the seat and disc, especially in high temperature and pressure steam applications.

In addition, the disc guide is susceptible to plugging and fouling in applications with fluids that have particulates.

Plug Valves

Plug valves are also a quarter turn rotary valve in which the plug rotates 90° from fully closed to fully open. There is a passage through the valve plug allowing fluid to flow, and can be a straight through design or a 3-way design, as shown in Figure 6-15. Plug valves can also be full ported or reduced ported.

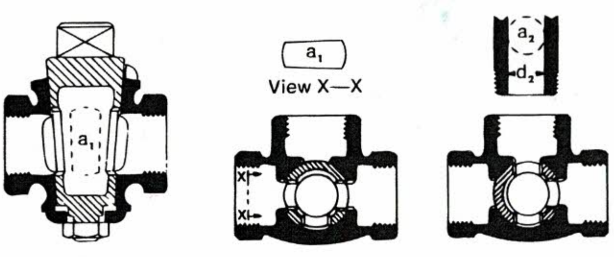


Figure 6-15. Straigh through plug valve (left), 3-way plug valve positioned for straight through flow (center), and positioned for 90° flow (right). (courtesy of Crane Valve Co.)

The large opening through the plug yields a low resistance to flow. The resistance coefficient for the straight through design in which the flow area of the plug equal to the pipeline flow area ($\beta = 1.0$) is given by Equation 6-20.

$$K = 18\, f_T \qquad \textit{Equation 6-20}$$

For the 3-way plug valve with straight through flow, shown in the center drawing of Figure 6-15, the resistance coefficient is calculated using Equation 6-21. For the case in which the flow takes a 90° turn through the 3-way plug valve, the resistance coefficient is calculated using Equation 6-22.

$$K = 30\, f_T \qquad \textit{Equation 6-21}$$

$$K = 90\, f_T \qquad \textit{Equation 6-22}$$

The Crane Technical Paper No. 410 should be consulted for calculating the resistance coefficient for reduced ported plug valves.

Plug valves can be used as on/off isolation valves and can also be used in moderate throttling applications. Multi-port plug valves are used for diverting applications.

Plug valves are typically used in systems under vacuum and at low pressures and temperatures, although some designs allow for high pressures and temperatures. They can also be used in non-abrasive liquids or gases, or with slurries and chemicals.

Butterfly Valves

Butterfly valves are rotary valves with a waffle shaped valve disc that closes and opens. The centerline of the shaft of the centric butterfly valve shown on the left is located at the centerline of disc, which is the same as the centerline of the pipeline, as shown in Figure 6-16.

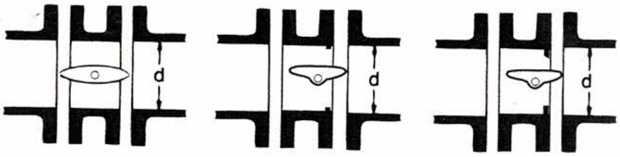


Figure 6-16. Various butterfly valve designs: centric (left), double offset (center), and tripple offset (right). (courtesy of Crane Valve Co.)

With double offset butterfly valves (center image of Figure 6-16), the disc and seat are offset from the shaft centerline, and the shaft itself is offset from the centerline of the pipeline. The triple offset design (image on the right of Figure 6-16) also incorporates an offset between the valve seat and the disc sealing surface to minimize rubbing during opening and closing and to improve the sealing integrity over the life of the valve.

Because the rotating disc of the butterfly valve is always in the flow path, it offers a greater resistance to flow and a larger pressure drop compared to ball and gate valves. The resistance coefficient depends not only on the valve design, but also on the valve size. The equations to calculate the resistance coefficients for fully open butterfly valves are shown in Table 6-5. Notice that for a given valve design, the resistance coefficient decreases with increasing size, not only due to the decreasing turbulent friction factor, but also due to the valve size iteslf. This is because the valve disc has a lesser influence on the overall resistance in an increasingly larger flow passage.

Table 6-5: Resistance Coefficients for Butterfly Valves as a Function of Valve Size and Design

Size Range	Centric	Double Offset	Triple Offset
2" – 8"	45 f_T	74 f_T	218 f_T
10" – 14"	35 f_T	52 f_T	96 f_T
16" – 24"	25 f_T	43 f_T	55 f_T

Butterfly valves are used for on/off isolation valves and in throttling applications when the differential pressure across the valve is relatively small since they are susceptible to cavitation at high differential pressures. They are used for both liquids and gases, as well as for powders and slurries. They can be used in fluids under vacuum or in pressurized systems. Other advantages of butterfly valves include their low cost and small installation width in the pipeline.

Globe and Angle Valves

A globe valve is a linear motion valve in which the fluid flowing through the valve body takes a more tortuous path through the valve, depending on the valve design. In addition, the valve disc remains in the flow path even when the valve is fully open. These two factors cause the globe valve to have a higher resistance to flow.

In the standard straight through design shown in Figure 6-17, the flow actually makes roughly two 90° changes of direction in quick succession. The resistance coefficient for full ported globe valves can be calculated using Equation 6-23.

$$K = 340\, f_T \qquad \textit{Equation 6-23}$$

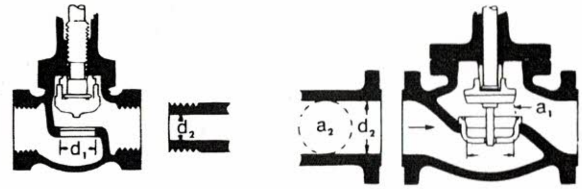


Figure 6-17. Standard straight through globe valves (courtesy of Crane Valve Co.).

Another type of globe valve is the Y-pattern angled globe valve shown in Figure 6-18. The flow essentially makes two 45° changes in fluid direction, so its resistance coefficient would be less than the design shown in Figure 6-17. The resistance coefficient for a full ported Y-pattern angled globe valve is given by Equation 6-24.

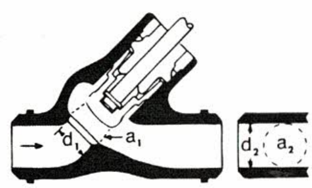


Figure 6-18. 45° Y-pattern angled globe valve (courtesy of Crane Valve Co.).

$$K = 55\, f_T \qquad \textit{Equation 6-24}$$

If a globe valve needs to be installed at the connection between a vertical pipeline and a horizontal pipeline, a 90° angled globe valve such as the ones shown in Figure 6-19 may be used. The flow makes a single 90° change in direction as it flows through the angled globe valve. If the valve plug of the angled globe valve remains in the flow stream as shown on the left, its resistance coefficient can be calculated using Equation 6-25.

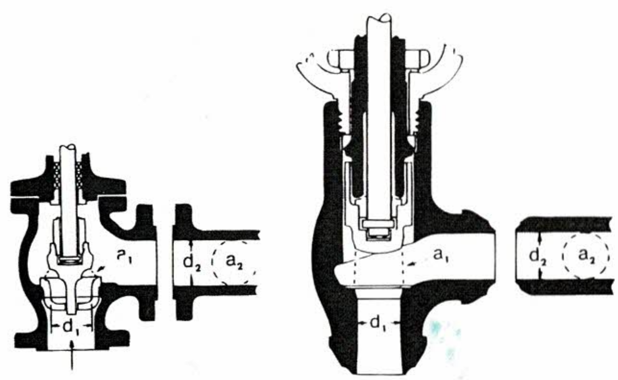


Figure 6-19. 90° angled globe valves. (courtesy of Crane Valve Co.)

$$K = 150\, f_T \qquad \textit{Equation 6-25}$$

The valve plug of the angled globe valve on the right in Figure 6-19 is almost fully removed from the flow stream, so it will offer less resistance to flow, as can be seen in the *L/D* value used for calculating its resistance coefficient in Equation 6-26.

$$K = 55\, f_T \qquad \textit{Equation 6-26}$$

The above equations for calculating the resistance coefficient for globe valves assumes fully open full port valves. The Crane Technical Paper No. 410 should be consulted for calculating the resistance coefficient for reduced ported globe valves.

Globe valves are good choices for on/off isolation valves and for throttling the flow rate through the valve. They can be used in non-abrasive liquids and gases at a wide range of temperature and pressure or under vacuum. They are not as susceptible to cavitation at high differential pressures as butterfly valves.

Diaphragm and Pinch Valves

Diaphragm and pinch valves are very similar in design and operation. Both contain a flexible diaphragm that deforms to change the shape of the flow path and restrict the fluid flow as the valve is closed. Figure 6-20 shows two types of diaphragm valves. The weir type (left) contains a solid weir at the bottom that reduces the amount of travel required by the flexible diaphragm. It's fully open resistance coefficient can be calculated with Equation 6-27.

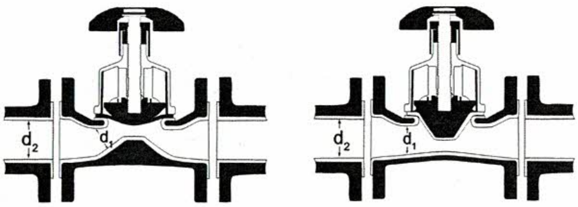


Figure 6-20. Weir type (left) and straight through (right) diaphragm valves (courtesy of Crane Valve Co.).

$$K = 149\ f_T \qquad \textit{Equation 6-27}$$

The straight through diaphragm valve has a more flexible membrane that allows the diaphragm to stretch farther, allowing a larger flow area when fully open. The resistance coefficient for a fully open straight through diaphragm valve is calculated with Equation 6-28.

$$K = 39\ f_T \qquad \textit{Equation 6-28}$$

Pinch valves are similar in that they contain a flexible tube through which the fluid flows. An external bar or pnuematic or hydraulic pressure compresses the tube to pinch off the fluid flow. The valve manufacturer should be consulted to obtain the resistance coefficient for the pinch valves they produce.

Diaphragm and pinch valves are excellent choices to use with fluids with particulates, slurries, or chemicals. Pinch valves are not good choices for systems under vacuum or at high pressures or with a high differential pressure across the valve. Both types are typically used in low pressure applications, but diaphragm valves can be used at higher pressures than pinch valves.

Diaphragm and pinch valves are good isolation valves and can be used for throttling in the last 50% of their stroke, but throttling close to the closed seat should be avoided because of high velocity erosion of the rubber sealing surfaces.

Check Valves

Check valves prevent the reversal of flow in a pipeline which could result in damage to equipment. When fully open, the check valve still offers resistance to flow depending on the type and design. The check valve resistance tends to be greater if the disc remains within the flow stream or if there are multiple changes of direction within the check valve.

When selecting a check valve it is important to ensure there is sufficient flow through the check valve to fully open the disc.

Swing Check Valves

A swing check valve consists of a hinged disc that allows flow in one direction, as shown in Figure 6-21. The hinged disc is opened by the fluid passing through the valve and is pushed out of the main flow passage by the moving liquid.

The resistance coefficient for a swing check valve depends on the design. For the swing check valve in which the disc seats at an angle, as shown in the left hand drawing of Figure 6-21, the fluid makes two 45° changes in direction, and its resistance coefficient is calculated by Equation 6-29.

$$K = 100\ f_T \qquad \text{Equation 6-29}$$

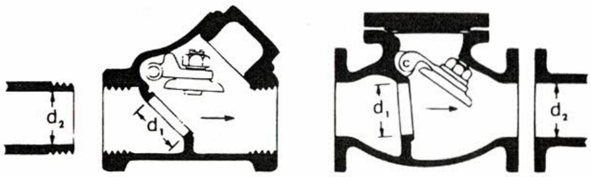


Figure 6-21. Various designs of swing check valves: disc seats at an angle (left) or vertically (right). (courtesy of Crane Valve Co.)

The resistance coefficient for the swing check valve in which the disc seats vertically, as shown in the right hand drawing of Figure 6-21, is calculated by Equation 6-30. Since the fluid essentially passes straight through the valve, its resistance coefficient would be smaller, as indicated by the smaller *L/D* value.

$$K = 50\ f_T \qquad \text{Equation 6-30}$$

Tilting Disc Check Valves

The tilting disc check valve shown in Figure 6-22 is typically the least expensive design and consists of a hinged disc that remains tilted at an angle as the fluid flows through the valve. The resistance to flow is increased because the disc remains in the flow passage. The resistance coefficient for a tilting disc check valve is a function of the angle at which the disc sits when fully open, as well as the check valve size, as shown in the equations for the resistance coefficients in Table 6-6.

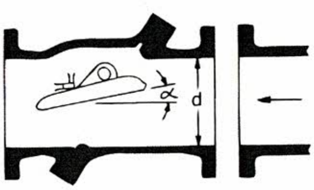


Figure 6-22. Tilting disc check valves. (courtesy of Crane Valve Co.)

Table 6-6: Resistance Coefficients for Tilting Disc Check Valves as a Function of Valve Size and Disc Angle

Size Range	$\alpha = 5°$	$\alpha = 15°$
2" – 8"	40 f_T	120 f_T
10" – 14"	30 f_T	90 f_T
16" – 48"	20 f_T	60 f_T

Lift Check Valves

With lift check valves, shown in Figure 6-23, the force of the fluid flowing through the valve causes the disc to open and the disc "floats" on the fluid as it passes through the valve.

As in every valve or fitting, the greater the change in direction of the flow, the greater the head loss. The lift check valve on the left in Figure 6-23 is similiar to a globe valve in that there is essentially two 90° changes in fluid direction. Its resistance coefficient is calculated using Equation 6-31 for a full ported valve.

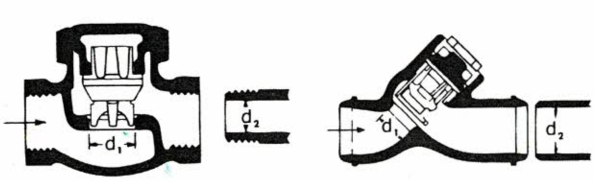


Figure 6-23. Lift check valves: globe type (left) and Y-pattern (right). (courtesy of Crane Valve Co.)

$$K = 600\ f_T \qquad \text{Equation 6-31}$$

Fluid flowing through the Y-pattern lift check valve shown on the right in Figure 6-23 makes two 45° changes in direction. Its resistance coefficient is less than the globe type lift check valve as seen with the smaller *L/D* value used to calculated the resistance coefficient for a full port valve in Equation 6-32.

$$K = 55\ f_T \qquad \textit{Equation 6-32}$$

The Crane Technical Paper No. 410 should be consulted for calculating the resistance coefficient for reduced ported lift check valves.

Foot Check Valves

Foot check valves are typically installed in the suction line of a pump to prevent it from losing its prime when the pump is turned off and the fluid flows back into the supply tank.

The poppet disc type of foot check valve shown in Figure 6-24 has a higher resistance coefficient because the disc remains in the flow stream and is an obstruction to the flow compared to the hinged disc type in which the disc is partially moved out of the flow stream. This can be seen with the higher *L/D* value for calculating the resistance coefficient for the poppet disc foot check valve in Equation 6-33.

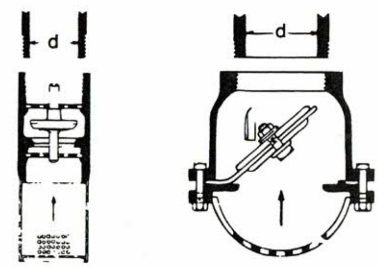


Figure 6-24. Foot check valves: poppet disc type (left) and hinged disc type (right). (courtesy of Crane Valve Co.)

$$K = 420\ f_T \qquad \textit{Equation 6-33}$$

The resistance coefficient for the hinged disc foot check valve is given in Equation 6-34.

$$K = 75\ f_T \qquad \textit{Equation 6-34}$$

Stop Check Valves

A stop check valve fulfills two functions in a single valve in that it is an isolation valve and a check valve combined in one valve body. When fully open, the disc can drop to the closed position automatically in the event of a flow reversal, or it can be manually closed by a valve actuator.

Stop check valves can have either a globe type body or a Y-pattern, as shown in Figures 6-25, and they can be either full ported or reduced ported valves. The design of the valve will have a big impact on the resistance coefficient of the valve.

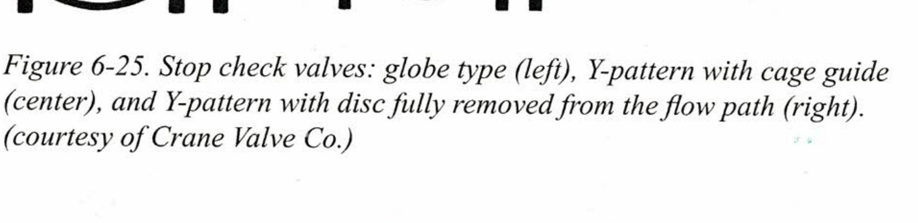


Figure 6-25. Stop check valves: globe type (left), Y-pattern with cage guide (center), and Y-pattern with disc fully removed from the flow path (right). (courtesy of Crane Valve Co.)

Equation 6-35 is used to calculate the resistance coefficient for the globe type stop check valve shown on the left in Figure 6-25.

$$K = 400\ f_T \qquad \text{Equation 6-35}$$

Equation 6-36 is used to calculate the resistance coefficient for the Y-pattern stop check valve with a cage guide shown in the center in Figure 6-25.

$$K = 300\ f_T \qquad \text{Equation 6-36}$$

Equation 6-37 is used to calculate the resistance coefficient for the Y-pattern stop check valve in which the disc is fully removed from the flow path shown on the right in Figure 6-25.

$$K = 55\ f_T \qquad \text{Equation 6-37}$$

Stop check valves can also be mounted at the corner intersection of a vertical and horizontal pipe and have an angled body design as shown in Figure 6-26.

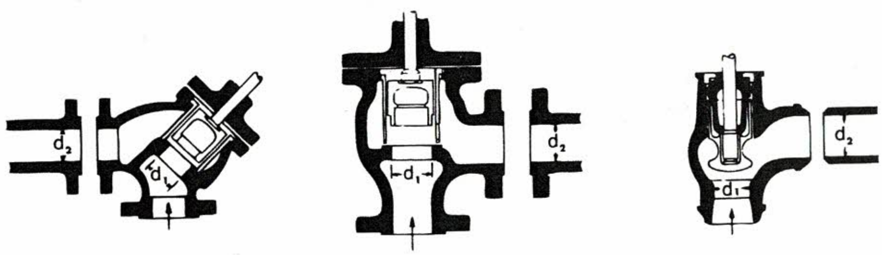


Figure 6-26. Angled body stop check valves: cage guided with actuator mounted at 45° (left), cage guided with actuator mounted at 90° (center), and angled with disc fully removed from the flow path (right). (courtesy of Crane Valve Co.)

Equation 6-38 is used to calculate the resistance coefficient for the cage guided angled stop check valve with the actuator mounted at 45° as shown on the left in Figure 6-26.

$$K = 350\ f_T \qquad \text{Equation 6-38}$$

Equation 6-39 is used to calculate the resistance coefficient for the cage guided angled stop check valve with the actuator mounted at 90° as shown in the center in Figure 6-26.

$$K = 200\ f_T \qquad \text{Equation 6-39}$$

Equation 6-40 is used to calculate the resistance coefficient for the angled stop check valve with the actuator mounted at 90° and the disc fully removed from the flow path as shown on the right in Figure 6-26.

$$K = 55\ f_T \qquad \text{Equation 6-40}$$

Notice that for all the cases for calculating the resistance coefficient, the more tortuous the flow path is through the valve, the more resistance it offers to the fluid and the higher the *L/D* value.

As before, the equations for stop check valves presented above are for full ported valves. The Crane Technical Paper No. 410 should be consulted for calculating the resistance coefficient for reduced ported stop check valves.

Example 6-2: Calculating the Hydraulic Performance of Valves and Fittings PF

To demonstrate how to use these concepts and equations, calculate the resistance coefficient, head loss, associated pressure drop, and equivalent flow coefficient of a standard full port 4-inch globe valve that is fully open with a 400 gpm flow of water at 60°F, as shown in Figure 6-27. The inside diameter of the attached 4-inch nominal pipe is 4.026". The density of the water is 62.4 lb/ft³.

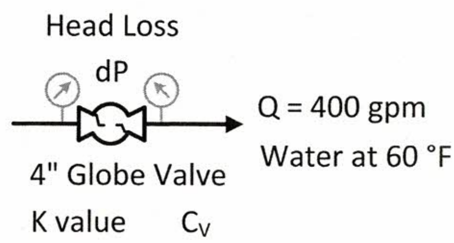


Figure 6-27. 4-inch globe valve with 400 gpm of water flow. PF

Resistance Coefficient

The resistance coefficient is calculated with the proper form of Equation 6-5 for a globe valve. The *L/D* value is 340 as shown in Equation 6-23. The turbulent friction factor (f_T) is determined using Table 6-1 for a 4-inch valve size.

$$K = f_T\frac{L}{D} = 340f_T = (340)(0.016) = 5.44$$

Head Loss

The head loss can be calculated with Equation 6-7 since the flow rate (in gpm) and the inside pipe diameter (in inches) are known:

$$h_L = 0.00259\frac{KQ^2}{d^4} = \frac{(0.00259)(5.44)(400\,gpm)^2}{(4.026")^4} = 8.58\,ft$$

Pressure Drop

The pressure drop associated with the head loss across the valve can be calculated using Equation 2-20.

$$dP = \frac{\rho h_L}{144} = \frac{(62.4\,lb/ft^3)(8.58\,ft)}{144\ in^2/ft^2} = 3.72\,psi$$

Equivalent Flow Coefficient, C_V

The equivalent flow coefficient can be calculated using the flow rate and pressure drop with Equation 6-2, or with the relationship between the resistance coefficient and flow coefficient by re-arranging Equation 6-8.

$$C_V = \frac{Q}{\sqrt{dP/SG}} = \frac{400\,gpm}{\sqrt{3.72\,psid/1.0}} = 207.39$$

$$C_V = \sqrt{891\frac{d^4}{K}} = \sqrt{891\frac{(4.026")^4}{5.44}} = 207.44$$

As mentioned before, the accuracy of the calculation of the flow coefficient is sensitive to the number of significant digits used for the variables because of the square root function.

Example 6-3: Comparing the Hydraulic Performance of Valves and Fittings

Table 6-7 shows the comparison between the *L/D* value, resistance coefficient, flow coefficient, head loss, and pressure drop across various types of 4-inch valves with 400 gpm of 60 °F water flow. Notice that the *L/D* value and resistance coefficient increase linearly with each other, whereas the flow coefficient decreases exponentially with an increasing resistance coefficient. The greater the resistance coefficient, the greater the amount of head loss and associated pressure drop.

Table 6-7: Comparing the Hydraulic Characterization and Performance of Various Types of Valves

Valve Type	L/D	K	C_V	Head Loss (feet)	Pressure Drop (psi)
Ball	3	0.05	2208	0.08	0.03
Gate	8	0.13	1352	0.20	0.09
Plug	18	0.29	902	0.45	0.20
Butterfly	45	0.72	570	1.14	0.49
Globe	340	5.44	207	8.58	3.72

Valve and Fitting Head Loss Graph

The amount of head loss created by the flow of fluid through a valve or fitting can be graphed as a function of the flow rate, as shown in Figure 6-28. The graph is a second order curve as indicated by the head loss equation, Equation 6-7. The steepness of the curve depends on the resistance coefficient of the valve or fitting.

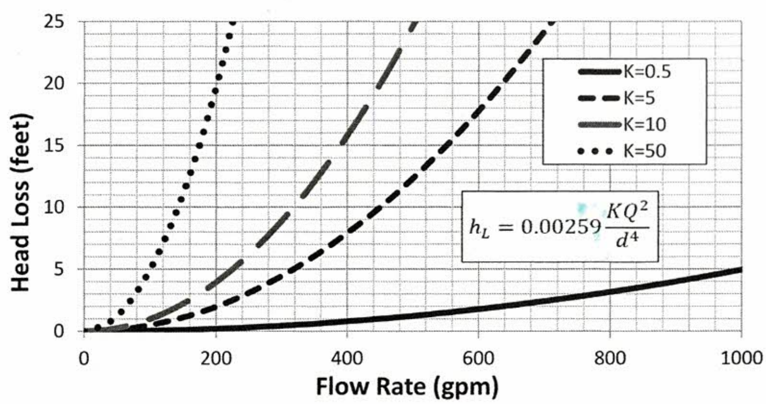


Figure 6-28. Graph of valve and fitting head loss as a function of flow rate and increasing resistance coefficient.

As with the pipeline head loss graph, the graph of the valve and fitting head loss is a basis for the system resistance curve that was discussed in Chapter 4 on centrifugal pumps.

Cost of Head Loss Across Valves and Fittings

The head loss across a valve or fitting is dissipated energy that was initially added to the fluid by the pump. A portion of the operating cost of the energy added by the pump can be allocated to the head loss across each valve and fitting in the system using Equation 6-41.

$$Cost\ of\ Head\ Loss = \frac{(0.746)\ Q\ h_L\ \rho}{(247{,}000)\ \eta_P\ \eta_m\ \eta_{VFD}}\begin{pmatrix}Operating\\ Hours\end{pmatrix}(\$/kWh) \qquad \textit{Equation 6-41}$$

Comparing the Operating Cost of Valves and Fittings

Table 6-8 puts into perspective the cost of the energy being dissipated as head loss across various valves and fittings. This table shows the amount of head loss and the associated energy cost for a 4" fitting with 400 gpm of 60 °F water flow with a pump efficiency of 70%, motor efficiency of 90%, and a utility rate of $0.10 / kWh, operating for 1,000 hours, using Equations 6-7 and 6-41.

The head loss across a long radius elbow is 0.35 feet at an energy cost of $4.23, compared to 0.50 feet of head loss at a cost of $6.04 for a short radius elbow.

For entrances, the head loss of a rounded entrance costs $0.75, a sharp edged entrance costs $9.43, and an inward projecting entrance costs $14.72.

The head loss across a ball valve costs $0.91, $2.41 for a gate valve, $5.43 for a plug valve, $13.58 for a butterfly valve, and $102.63 for a globe valve.

Table 6-8: Comparing the Operating Cost of Various Types of Valves and Fittings

Valve Type	Head Loss (feet) of a 4-inch Fitting With 400 gpm	Cost ($) per 1,000 hours of Operation
Elbow LR	0.35	$4.23
Elbow SR	0.50	$6.04
Entrance (rounded)	0.06	$0.75
Entrance (sharp edged)	0.79	$9.43
Entrance (protruding)	1.23	$14.72
Ball	0.08	$0.91
Gate	0.20	$2.41
Plug	0.45	$5.43
Butterfly	1.14	$13.58
Globe	8.58	$102.63

This is not an argument to replace all short radius elbows in an existing piping system with long radius elbows, or all globe valves with ball valves. Operating cost isn't always the deciding factor when designing a system and selecting valves and fittings. A particular valve or fitting is selected for a specific function in a system and that function has to be accomplished in order for the system to work as intended. However, if there are multiple valve types that are suitable for a particular application, the operating cost should be considered in the selection.

These operating costs, and potential energy savings, will be paid for the entire life of the system.

Graphing the Energy Costs of Valves and Fittings

Another way to view the operating costs of valve and fitting head loss is using a graph of energy cost versus flow rate, as shown in Figure 6-29. This curve is based on a 4-inch valve with 60 °F water flow with a pump efficiency of 70%, motor efficiency of 90%, and a utility rate of $0.10 / kWh, operating for 1,000 hours.

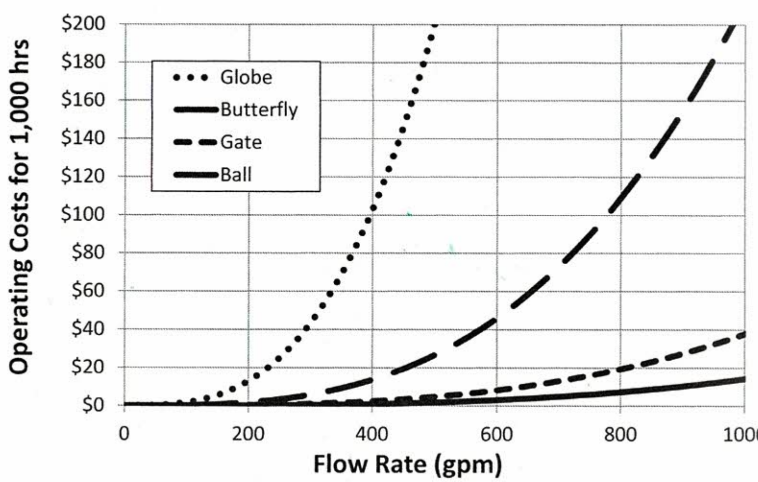


Figure 6-29. Graph of the energy cost of the head loss for various valves as a function of flow rate.

For a given flow rate, the operating costs vary dramatically with the valve type. In addition, the operating cost for a particular valve increases by the second order of the flow rate. This last observation emphasizes the need to select low resistance valves and fittings wherever possible for a particular application.

Selecting Valves & Fittings for a Given Application

Valves and fittings are required to redirect and isolate the flow in a piping system. There are many things to consider when choosing a valve or fitting for a given application. The selection of a valve type has an effect on the amount of head loss across the valve for a given flow rate, but the head loss should not be the sole consideration for the valve selected.

When selecting a valve type for an application in a piping system, the service requirements must be considered, including the valve's sealing capability under various pressures, the number of times a valve is opened and closed, how much time it takes to close the valve, hand-wheel torque, and the consequences of valve stem leakage, among other considerations.

The costs of the valve or fitting are often a primary concern. These costs include not only the life time energy cost, but the initial capital cost and anticipated maintenance cost as well.

Valves and fittings come in a wide array of types and sizes and are produced by many manufacturers around the world. The *L/D* values and the equations for resistance coefficients presented in this chapter are based on the pressure drop tests performed by the Crane Valve Company. When selecting and purchasing a particular valve or fitting, the manufacturer should be consulted to determine if they have equivalent lengths, resistance coefficients, or flow coefficients for their products. In the absence of specific data, the resistance coefficients presented in this chapter may be used, or approximations may be made based on good engineering principles and experience.