Piping System Fundamentals

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

SECOND EDITION U.S.

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Chapter Seven

Control Valves

Control valves play a crucial role in the total piping system. Isolation valves and control valves share many physical characteristics, but where an isolation valve's primary purpose is to allow flow when fully open or to stop flow when fully closed, the control valve operates between the extremes to regulate the flow rate, pressure, temperature, tank level, chemical composition, or some other system process variable.

Role of the Control Valve

A piping system exists to transport mass and energy in order to perform work or make a product, and the working fluid may undergo many processes that change its physical, thermal, or chemical properties. Various instruments are used to measure processes occuring in the piping system, processes that are subject to disturbances that cause the property of the fluid to deviate from a desired value. Control loops are installed to automatically respond to these disturbances and a control valve is a common device used to maintain the measured property in a steady state condition.

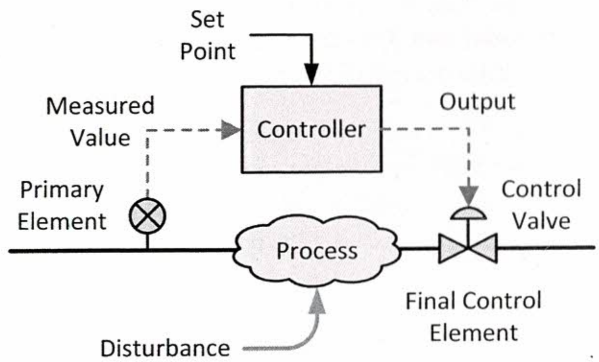

Figure 7-1. Block diagram of a typical process control loop.

Figure 7-1 shows the key elements of a typical process control loop, including the primary element (or sensor), the controller, and the final control element. These elements will be explained in detail in Chapter 8: Process Measurements and Controls, and the actual processes that occur in the piping system will be covered in Chapter 9: Processes and Process Equipment.

The control valve plays a major role as the final control element of the process control loop. In a piping system, a control valve will be called upon to regulate the flow rate, upstream or downstream pressure, fluid temperature or pH, tank level, chemical reaction rate, or some other important process parameter.

How a Control Valve Works

The control valve is designed to change the shape and size of the flow passage within the valve, thereby adjusting its hydraulic resistance and the amount of energy dissipated across the valve. This energy, or head loss, is dissipated in the form of heat, noise, and vibration in the valve.

A control valve's variable resistance is accomplished by inserting an adjustable obstruction in the flow path within the valve. This obstruction can be a moveable plug such as in a globe valve, a fluted disc in butterfly valve, or a vee-notched ball in a ball valve, just to name a few.

To better understand how a control valve dissipates hydraulic energy to control a process occuring in a piping system, consider the generalized energy grade profile, the hydraulic grade profile (pressure profile), and velocity profile shown in Figure 7-2 as the fluid travels through a control valve with an inlet and outlet reducer attached. (Note: The valve shown is a single seated globe valve, but the same principles apply to all types of control valves.)

The energy grade profile shows the total hydraulic energy in the fluid and consists of the velocity head, pressure head, and elevation head as described by the Bernoulli equation. Any reduction in the energy grade line indicates energy loss due to friction and changes in fluid momentum and is energy dissipated as head loss in the form of heat, noise and vibration.

Chapter 2 mentioned that hydraulic grade represents static head, which is the sum of the elevation and pressure head of the fluid. Since the elevation is not changing from the inlet to the outlet, the hydraulic grade line in Figure 7-2 represents just the pressure head component, or the fluid's static pressure.

The velocity profile represents the dynamic head component of the fluid's total energy. The sum of the static head (hydraulic grade) and dynamic head (velocity) is the total energy of the fluid.

In Figure 7-2, Position 1 represents a point in the inlet pipe upstream from the valve assembly. Pipeline head loss from Position 1 to the inlet of the reducer at Position 2 results in a small drop in the total energy grade line. Since the velocity is constant in the inlet pipeline, this energy loss results in an equivalent drop in the hydraulic grade line, which is seen as a pressure drop in the fluid. The energy grade line and hydraulic grade lines are parallel.

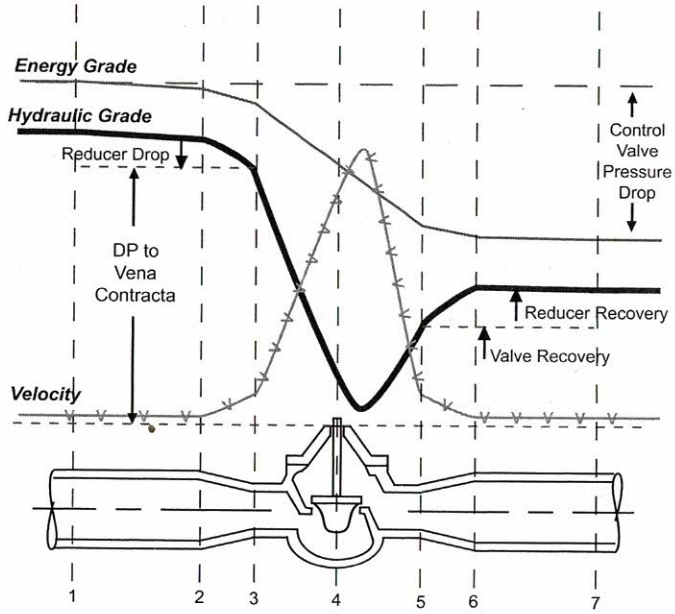

Figure 7-2. Total energy grade, hydraulic grade (pressure), and velocity profiles through a control valve.

As the fluid enters the inlet reducer of the control valve at Position 2, the flow area decreases, resulting in an increase in the fluid velocity according to the Bernoulli Theorem. This increased fluid velocity results in a momentum change and head loss across the reducer, which is seen as a decrease in the energy grade and hydraulic grade lines from Position 2 to Position 3.

As the fluid flows past the inlet at Position 3 it quickly changes direction by 90° to flow up through the valve seat and past the plug at Position 4, resulting in momentum change, head loss, and a reduction in the fluid's total energy. In addition to the change in direction, there is often a reduction of the cross-sectional flow area between the valve inlet and the flow passage through the valve seat, resulting in a large increase in the fluid velocity, additional change in fluid momentum, additional head loss, and associated drop in pressure and total energy. For the globe valve shown in Figure 7-2, the flow path is actually a donut-shaped annulus between the seat and plug. The point of highest velocity and lowest pressure occurs at the vena contracta, which is slightly downstream of the narrowest point in the flow path.

Between Position 4 and 5, there is another 90° change of direction as the fluid moves through the valve seat to the valve outlet. The change of direction results in a momentum change and head loss. In addition, the fluid velocity decreases from the vena contracta to the valve outlet and this momentum change results in head loss. However, the decreased fluid velocity results in an increase in the static pressure of the fluid according to the Bernoulli Theorem, so there is an overall pressure recovery from the vena contracta to the valve outlet, even though there is a continuous drop in the total energy of the fluid.

As the fluid flows past the valve outlet at Position 5 and through the reducer to Position 6, there is an increase in the flow path's cross-sectional area, which decreases the velocity and increases the fluid static pressure. This pressure recovery across the reducer is the conversion of the velocity head to pressure head in accordance with the Bernoulli Theorem. However, this change in velocity is also a change in the momentum of the fluid, resulting in additional head loss and an associated drop in total fluid energy grade line across the outlet reducer.

From Position 6 at the discharge of the outlet reducer to a point farther downstream represented by Position 7, the pipeline head loss results in a constant decrease in the fluid's total energy and static pressure.

The position of the valve plug in relation to the seat will determine the size of the flow passage and the amount of resistance the valve offers to the fluid flow, and therefore the amount of head loss across the control valve. Consider the change to the energy profiles for a control valve that is farther open, as shown in Figure 7-3.

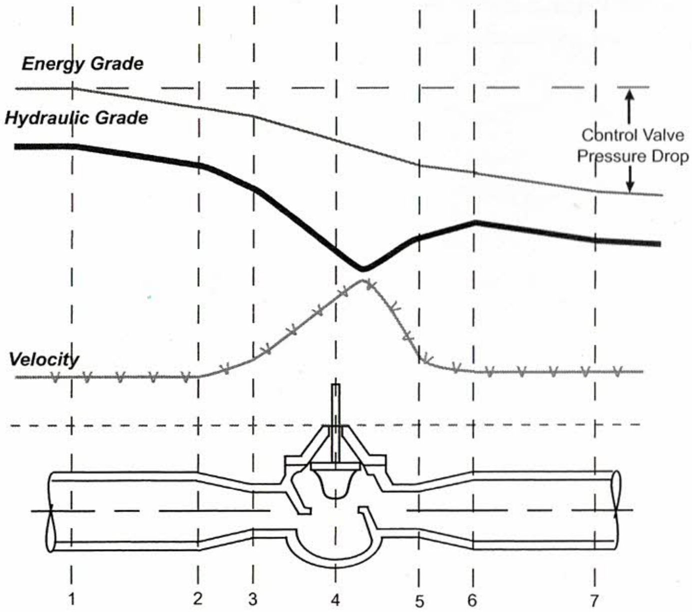

Figure 7-3. Total energy grade, hydraulic grade (pressure), and velocity profiles through a control valve that is farther open.

Because there is more distance between the seat and plug, there is a larger flow path and less resistance to flow through the valve and a higher flow rate through the piping and control valve (assuming the same inlet pressure at Position 1 as in Figure 7-2). The fluid velocity at Position 1 will be higher than in Figure 7-2 due to the higher flow rate. The hydraulic grade and energy grade lines for the pipelines will have a steeper slope due to the greater amount of head loss in the pipeline from Position 1 to Position 2 and from Position 6 to Position 7.

In additon, because the flow path is larger in the valve, the velocity at the vena contracta does not increase as high as it did in the previous example, so there is less momentum change and therefore less head loss across the valve. This results in a smaller drop in the hydraulic grade line from Position 2 to just past Position 4, and a smaller pressure recovery from Position 4 to Position 5.

The net effect of a wider open control valve is a higher flow rate through the valve, less head loss (energy dissipated) across the valve, and a smaller pressure drop from the inlet to the outlet of the valve.

Major Parts of a Control Valve

The major parts of a control valve can be seen in Figure 7-4 which shows an air operated sliding stem globe valve. The two main components are the valve body assembly and the valve actuator.

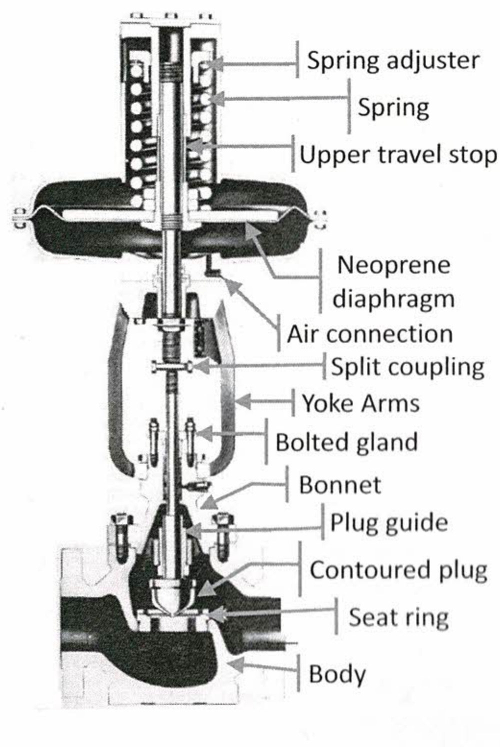

Figure 7-4. Cross-sectional drawing showing major parts of a control valve (courtesy of Fisher Controls International, LLC).

Valve Body Assembly

The valve body assembly consists of the valve body and the valve bonnet, both of which form the fluid pressure boundary. The valve trim components are contained within the body assembly.

Valve Body

The valve body contains the inlet and outlet connections to the piping system and its shape forms the internal fluid flow passage. The body also contains supports for the valve seat and provides room for the valve closure members (plug or disc).

Valve Bonnet

The bonnet is bolted on top of the valve body and contains the packing box, stem seal, and a means to attach the valve actuator. The bonnet also provides support for the plug guide.

The valve gland consists of packing that is inserted into the packing box between the bonnet and the stem. The packing is held into place by a bolted gland.

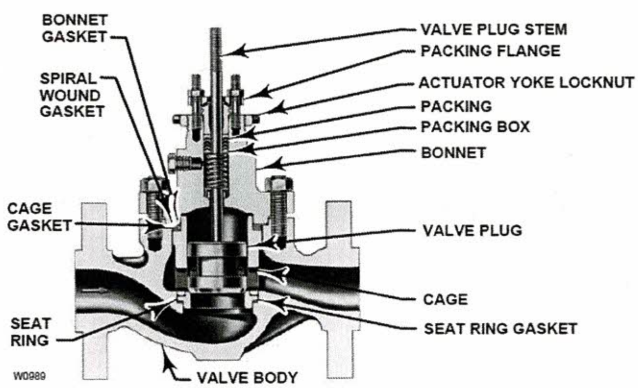

Figure 7-5. Cage guided globe valve (courtesy of Emerson Process Management).

Valve Trim

The valve trim consists of the components that are in contact with the process fluid flowing through the valve and include the valve seat, the plug or disc, plug guide, and valve stem. The valve seat is typically screwed into position in the valve body, allowing the seat to be replaced when necessary. In addition, the valve seat and plug can be replaced as a set and come in reduced sizes, allowing for a wider range of performance for the same valve body.

For cage guided valves, as shown in Figure 7-5, the cage is also a part of the valve trim and can be designed to give the valve various hydraulic characteristics.

Valve Actuator Assembly

The most common actuator installed on a control valve is a pneumatic actuator as shown in Figures 7-4 and 7-6. The actuator can also be an electric motor or solonoid, or use a hydraulic piston to move the valve stem.

The pneumatic actuator consists of a diaphragm motor, a spring chamber, an actuator stem, and yoke arms to connect the actuator to the valve bonnet. The actuator may also have a positioner installed to control the position of the valve stem.

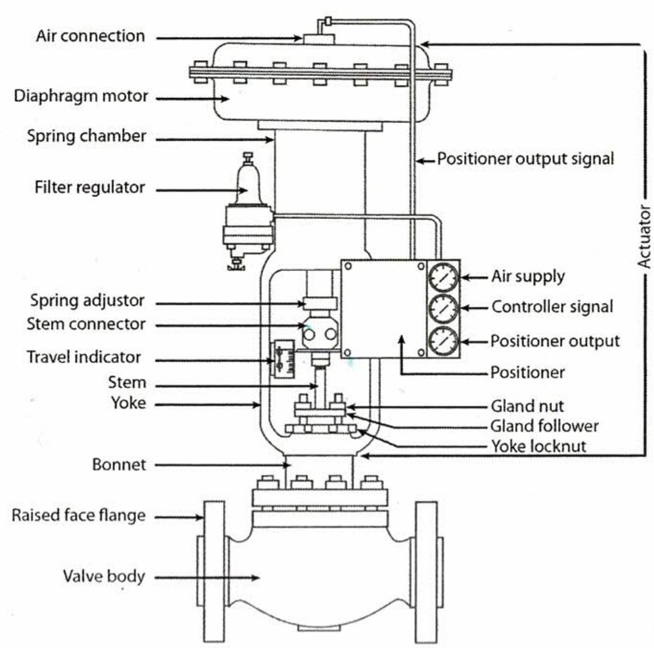

Figure 7-6. External components of a linear control valve (courtesy Fisher Controls International LLC).

Diaphragm Motor

The diaphragm motor contains a neoprene diaphragm that separates the spring chamber from a chamber that is pressurized by an external supply of instrument air. Air pressure can either act to open the valve (Figure 7-4) or close the valve (Figure 7-6).

Spring Chamber

The spring chamber contains a spring, a spring adjuster, and an upper travel stop. The spring chamber can be mounted at the top of the actuator as shown in Figure 7-4, or below the diaphragm as shown in Figure 7-6. The force exerted by the spring is in the opposite direction of the force exerted by the air pressure on the diagphragm.

Actuator Stem

The net force exerted by the air pressure and spring is tranmitted by the actuator stem to the valve stem to position the plug within the valve body. In steady state conditions, the two forces are equalized and valve plug is held in a fixed position. If the control valve position needs to be adjusted by the loop controller, the air pressure will be increased or decreased to create an unbalanced force between the air pressure and spring force, causing the valve to either open farther or close slightly until the process reaches steady state at the desired value.

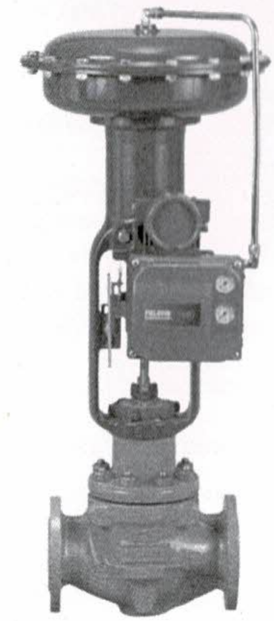

Figure 7-7. Pneumatically operated automatic control valve with digital positioner (courtesy of Emerson Process Management).

The actuator stem may also contain a position indicator that can be compared to a scale mounted on the yoke arm to give a visual indication of the approximate valve position, as can be seen in Figure 7-6.

Positioner

The control valve may also have a positioner installed on the valve actuator that has a magnetic or mechanical linkage to the shaft to measure the actual position of the valve travel. The positioner is a position controller that receives its set point from the loop controller, compares it to the measured valve position, and adjusts the air pressure to the actuator to adjust the valve to the desired position.

The positioner can be pneumatic, electro-pneumatic, or a digital positioner (intelligent or smart position), as shown in Figure 7-7. Positioners will be covered in more detail in Chapter 8: Process Measurement and Controls.

Classifying Control Valves

There are numerous ways to classify control valves with the various attributes that can be used to describe them. They can be classified according to how the valve disc or plug is caused to move, how the valve performs hydraulically, by its body style, or by the ASME or ANSI ratings.

Classifying Control Valves Based on Valve Actuation

Control valves fall into three generaly types according to how the force is applied to the valve stem and disc to cause them to move and change the shape and size of the flow path. Actuated valves use an external source of power to apply the force, self-contained regulators use the energy of the working fluid, and manually-adjusted valves use the force applied by a human "actuator" to move the valve stem and plug.

Actuated Control Valves

Figure 7-7 shows a typical actuated control valve that uses air as the external source of energy to adjust the position of the valve stem and plug. The valve actuator can be pneumatically powered, use an electric solonoid or motor, or use a hydraulic cylinder to move the valve stem.

It is important to remember that an actuated control valve is part of an external control loop, consisting of a sensor and a transmitter. Because of the variety of supporting equipment required for the control loop, using a control valve to maintain a process variable is expensive. However when the process variable must be maintained within a tight range, using an external control loop and properly sized control valve is necessary because of the degree of control they provide.

The actuated control valve will be the main focus of the majority of this chapter.

Self-contained Regulators

A self-contained regulator, such as the pressure regulator shown in Figure 7-8, is a control valve in which the process fluid provides the energy required to adjust the valve stem and plug. The set screw at the top adjusts the force exerted by the main spring on the diaphragm element. The downstream fluid pressure is applied to the underside of the diaphragm via a pitot tube. A spring below the valve plug provides a counter force that adds to the force of the fluid pressure and balances the force of the main spring to maintain the position of the valve plug.

As the downstream fluid pressure drops due to an increase in flow demand to downstream users, the main spring pushes the diaphragm and valve plug down to further open the valve, allowing more of the higher pressure upstream fluid to flow through the valve, thus maintaining the downstream pressure at a constant value.

Advantages of self-contained regulators include cost, excellent frequency response, good rangeability, space savings, limited valve leakage, and not requiring electricity or air to operate.

The disadvantages of self-contained regulators are a fixed proportional band, no reset action, limited size, and pressure rating, along with a limited choice of material and end connections.

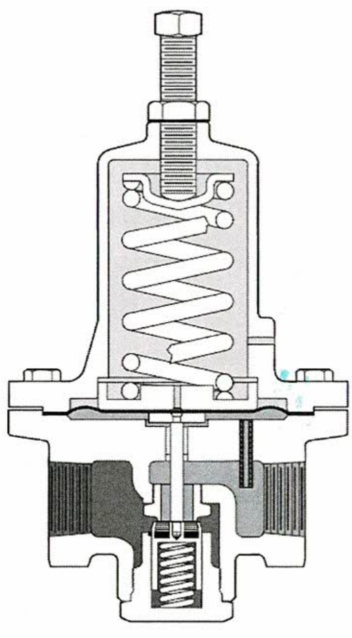

Figure 7-8. Self-contained pressure regulator (courtesy of Emerson Process Management).

Manually Operated Throttle Valves

In many applications, the cost of an automatic control valve or a regulator is not justified for the process. In these cases, a manually operated valve is often used to allow an operator to adjust the position of the valve based on a process variable that the operator observes.

For example, if a desired level is required for a tank with a site glass installed, the operator can adjust the flow rate of the liquid entering the tank based on the actual tank level and the desired tank level. For these manually operated valves, the operator acts as the transmitter, controller, and valve actuator.

Manually operated throttle valves are often used in systems where the process variables do not significantly change. The tolerances and speed of a manual control are limited. As a result, manually operated throttle valves are used on steady-state processes that do not require a tight range of control.

Any isolation valve that is used by an operator to manually adjust the flow rate, pressure, tank level, temperature, or any other process variable can be considered a control valve.

Classifying Control Valves Based on Characteristic Trim

The design of the valve trim affects the hydraulic performance of the valve, called the characteristic trim. Different types of valves will have a specific characteristic trim, but there are some design changes the manufacturer can make to obtain specific characteristics for some valve types.

Inherent Characteristic Curves

The manufacturer tests their control valves in a test system with 60 °F water to determine the *inherent characteristic* of the valve, or the characteristic with a constant differential pressure across the valve. This is done by measuring the flow rate through the valve with a 1 psi pressure drop from the valve inlet to the valve outlet and calculating the valve's flow coefficient (C_V). The test is repeated at various valve positions to obtain a profile of the hydraulic performance of the valve defined by its flow coefficient as a function of the valve position, or the inherent characteristic curve.

The most common characteristic curves are the quick opening, linear, and equal percentage curves shown in Figure 7-9. Trim can be installed to produce a modified linear and modified equal percentage curve as well.

Installed Characteristic Curves

When the control valve is installed in a piping system with a centrifugal pump, the differential pressure across the control valve will not be held constant as it is in the initial testing of the valve. Due to the shape of the pump curve, the *installed characteristic curve* will shift because the inlet pressure of the control valve changes with the flow rate. How the curve shift can be seen in Figure 7-10.

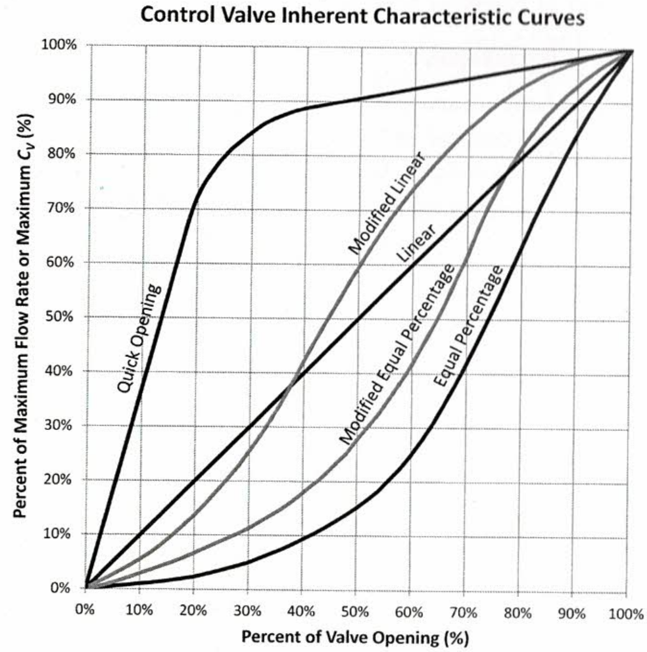

Figure 7-9. Control valve flow characteristic curves.

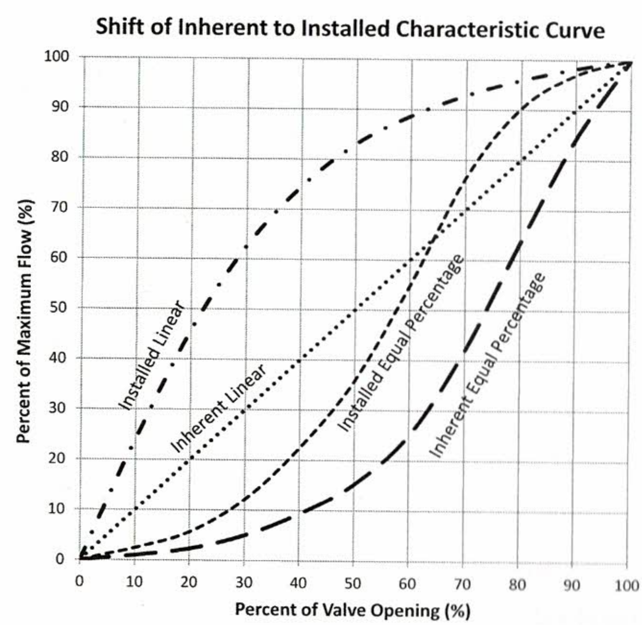

Figure 7-10. Shift of the inherent characteristic curve when installed in a piping system with a pump.

Quick Opening Characteristic

Quick opening valves are used in applications when a large change in the flow rate is needed for very little valve travel, such as on-off valves, pressure relief valves, or safety valves.

Linear Characteristic

Linear valves are used when the pressure at the valve inlet is expected to remain fairly constant, such as with level control and some flow control applications.

Equal Percentage Characteristic

Equal percentage valves are ones in which the flow coefficient changes by an equal percentage for a given change in valve position. This is the most common type of characteristic used in flow, pressure, and temperature control applications due to how inherent characteristic curve shifts when installed in a piping system.

Trim Designs to Obtain Various Characteristics

The geometry of the internal trim will determine how the control valve performs over its range of travel. For the globe valve shown in Figure 7-4, the shape of the contoured plug will determine the valve's inherent characteristics, as shown in Figure 7-11. A relatively flat plug will produces a quick opening characteristic, a parabolic shaped plug gives a linear characteristic, and a linear shaped plug produces an equal percentage characteristic.

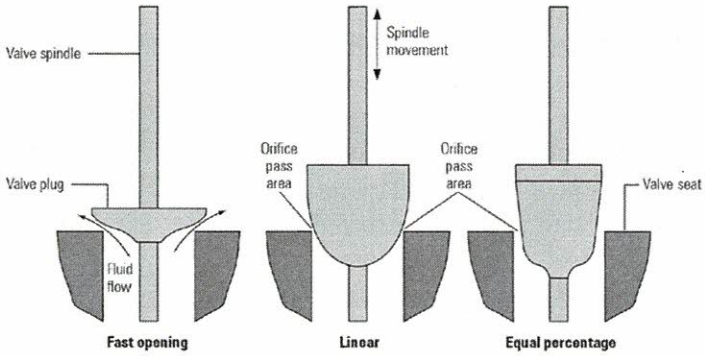

Figure 7-11. Contour of the plug determines the inherent flow characteristic of the control valve (courtesy of Spirax Sarco).

For the cage guided globe valve shown in Figure 7-5, the shape of the flow windows determines the flow characteristic of the valve. Figure 7-12 shows the various shapes that produce a quick opening, linear, and equal percentage characteristic.

For ball valves, the ball can be contoured with a V-notch or designed with various shapes of the flow passage to obtain different flow characteristics. Similarly, for butterfly valves, the disc can be designed to provide the desired flow characteristic.

Figure 7-12. Shape of the flow window determines the inherent flow characteristic of the control valve (courtesy of Emerson Process Management).

Classifying Control Valves Based on Valve Body Style

Control valves can also be classified according the style of the valve body. In general, valve body styles fall into two categories based on the motion used to change the shape of the flow path: linear motion and rotary motion.

In a linear motion control valve, the flow rate is controlled by the linear motion of the plug attached to a valve stem. The plug is moved closer to or farther away from the valve seat, resulting in a change of the flow path area through the seat. When the valve plug is moved closer to the seat, the flow area decreases and the velocity of the fluid flowing past the valve seat increases. This increase in velocity causes greater turbulence in the valve, resulting in more energy dissipated across the valve (head loss) and a larger pressure drop. Linear motion control valves include single and dual ported globe valves, cage trim globe valves, diaphragm valves, and pinch valves.

In a rotary motion control valve, the flow rate is controlled by turning the control element (ball, disc, or plug) in the flow stream. When the valve is fully open, the controlling element presents the smallest resistance to flow. As the controlling element is rotated, the cross-sectional area of the flow passage decreases. Rotary valves have a larger flow capacity and some are less expensive than linear motion control valves. In addition, since there are fewer obstructions in the flow path, the rotary valve design works well with slurries, paper stock, and solids in suspension. Rotary valves are more susceptible to flashing and cavitation in high-pressure drop applications due to their high-pressure recovery. Under these conditions, the fluid velocity through the valve can cause noise and excess wear. Rotary motion control valves include butterfly valves, ball valves, and rotating plug valves.

Single Port Globe Valves

Globe valve bodies tend to have a compact spherical shape. Figure 7-13 shows a typical single ported globe valve. The valve consists of a valve body, bonnet, stem, and plug. As previously mentioned, the seat can be removed for repair. In addition, a reduced ported plug can be inserted into the valve in order to change its control characteristics.

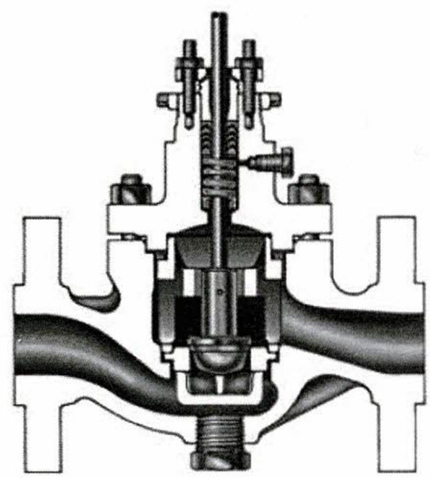

Figure 7-13. Single port globe valve (courtesy of Fisher Controls International, LLC).

As the fluid flows through the valve, the turbulence between the valve seat and plug can cause lateral forces on the plug, resulting in a bend of the stem, as well as vibration. A guide is used in the bonnet in order to steady the plug under varying flow rates and minimize its lateral movement.

Finally, there is considerable unbalanced force on the plug because of the difference in area between the bottom and top of the plug caused by the differential pressure across the valve. This exerts additional force upon the valve stem and may require a thicker stem and a larger actuator.

Single ported globe valves are also manufactured in a 90° angle and inline style. The 90° angle valve is useful when the pipeline with the control valve needs to make a change of direction, eliminating the need for an additional elbow. The inline valve pattern has the valve stem, plug, and seat on an oblique angle, minimizing the number of directional changes of the fluid.

Dual Port Globe Valves

Figure 7-14 shows a cross section of a dual port globe valve with two seats and a plug that has two seating areas. The flow enters the valve through the inlet port, diverges through each of the valve seats, travels around the plug, and then converges before leaving the valve.

The advantage of the dual port valve is its ability to balance the hydrostatic and hydrodynamic forces. The pressure differential acts both upward and downward upon the upper and lower plug areas, canceling out the hydrostatic and hydrodynamic forces upon the plug. The only unbalanced force on the dual ported plug is that caused by the difference in area of the valve stem passing through the valve gland to the atmosphere. Any difference in pressure must be overcome by the valve actuator. Since this unbalanced force is smaller on the dual port valve than the single port valve, the actuator can be smaller.

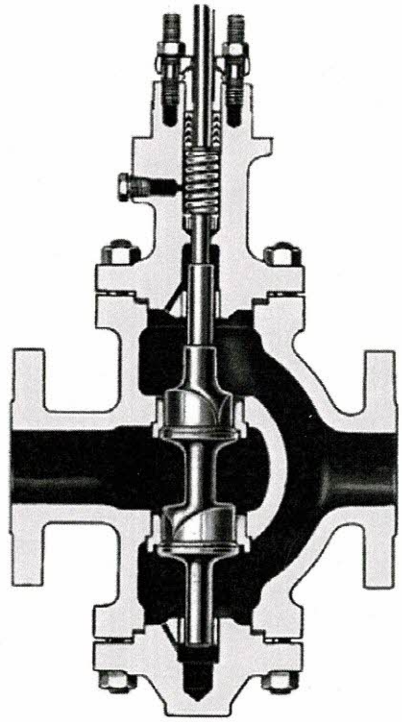

Figure 7-14. Dual port globe valve (courtesy of Fisher Controls International, LLC).

Cage-Trim Globe Valves

The cage trim design eliminates many of the problems associated with the single port and dual port globe valves. Cage trim valves have the following advantages over single and dual port valves:

- Greater capacity for a given body size
- Less prone to vibration from the fluid passing through the valve
- Reduced susceptibility to cavitation and the damage caused by cavitation
- Less noise
- Greater serviceability

As shown in Figure 7-15, the plug is guided in the cage as it opens and closes. This increases the lateral stability and reducing vibrations in the valve. By drilling balancing holes in the plug, the process pressure acts equally upon both sides of the plug, thus minimizing problems with unbalanced pressures.

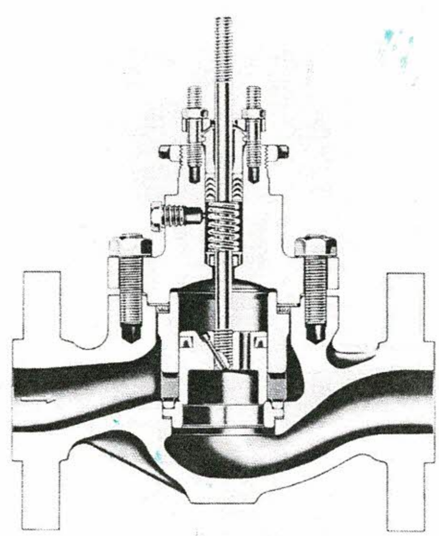

Figure 7-15. Cage trim globe valve (courtesy of Fisher Controls International, LLC).

As was seen in Figure 7-12, the shape of the ports in the cage can be varied to provide the desired characteristic trim. In addition, special cages are designed to reduce noise or cavitation in the valve. Also, the manufacturer can change the number and size of the holes in the cage to change the capacity of the valve without having to change out the valve body.

Multi Port Globe Valve

Multi port globe valves, such as the one shown in Figure 7-16, allow diverting or mixing of fluids. These valves always have two seats. The plug can be a double-ended single plug or two separate plugs. The valve body has three ports. For diverting applications, one port serves as the inlet and the other two as the outlets. These valves are often used in HVAC chilled water applications. When the multi port valve is used for mixing two fluids, two of the ports serve as inlets, and the mixture exits the remaining port.

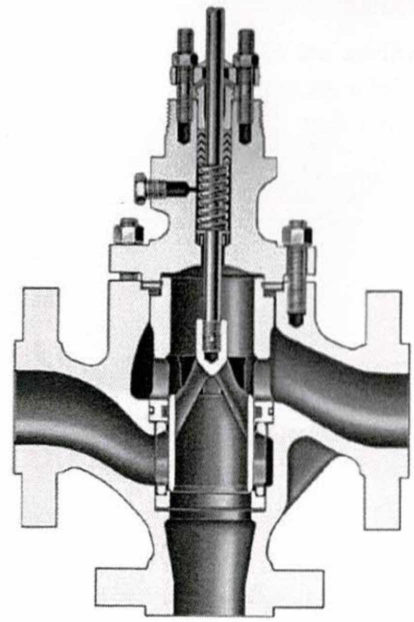

Figure 7-16. Multi-port globe valve (courtesy of Fisher Controls International, LLC).

Pinch Valves

The pinch valve shown in Figure 7-17 consists of a valve body that holds a flexible sleeve, which is the only component exposed to the process fluid. The valve operates when the sleeve is compressed, reducing the cross-sectional area of the sleeve. Compression of the sleeve can occur by a mechanical bar that pinches the flexible sleeve and restricts the flow, or by an external pressure source around the sleeve that compresses the sleeve and reduces the flow area to throttle the flow.

Diaphragm Valves

Diaphragm valves can be a straight through design such as the one shown in Figure 7-18, or have a weir built into the body of the valve. A flexible diaphragm is the closing member that deforms as the actuator is closed to change the shape and size of the flow passage and vary the amount of energy dissipated across the valve and throttle the flow rate. Diaphragm valves can operate at positive pressure or under high vacuum. The flexible diaphragm provides a tight seal at shutoff, even for liquids with solid particles.

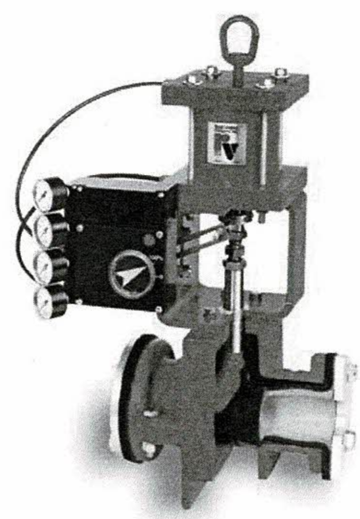

Figure 7-17. A typical pinch valve, (courtesy of the Red Valve Company).

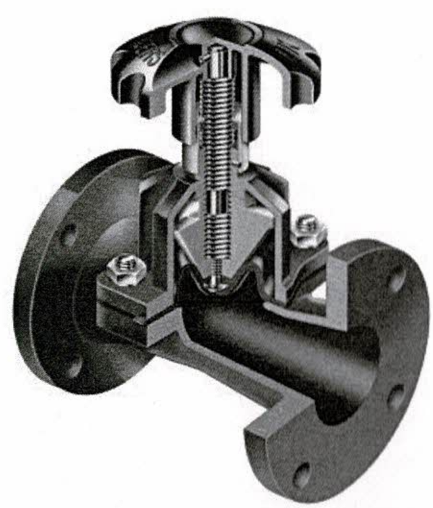

Figure 7-18. A manually operated diaphragm valve, (courtesy of Crane Saunders).

Ball Valves

Ball valves are rotary valves consisting of a spherical closing member with a flow passage, as shown in Figure 7-19. In addition to the V-notch segmented ball shown in Figure 7-19, the rotating member can be a full ball, a characterized V-notch, or a V-notch ball with an attenuator for cavitation and noise reduction, as shown in Figure 7-20.

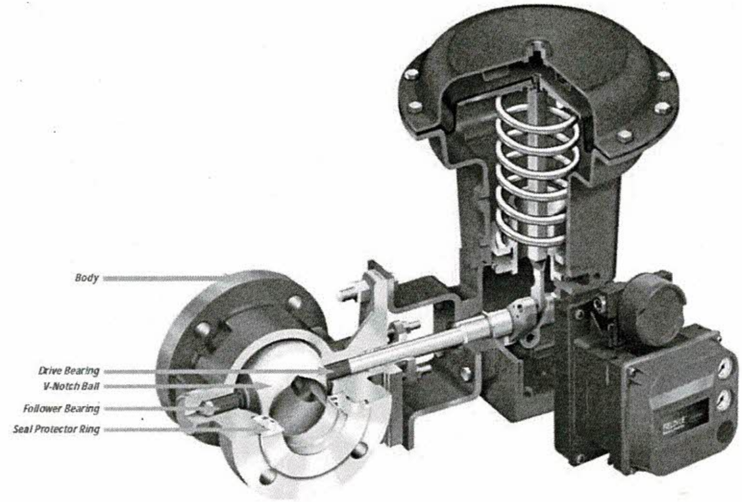

Figure 7-19. Segmented V-notch ball valve (courtesy of Emerson Process Management).

Ball valves are used to control flow, pressure in gas distribution systems, and pressure reduction in gas storage systems.

Ball valves are designed as full port and reduced port valves. The full port ball valves are primarily used in oil and gas distribution systems to allow a cleaning "pig" to be sent through the pipeline and pass through fully open ball valves without requiring the valve to be disassembled. One disadvantage of the full port ball valve is its poor flow control characteristics.

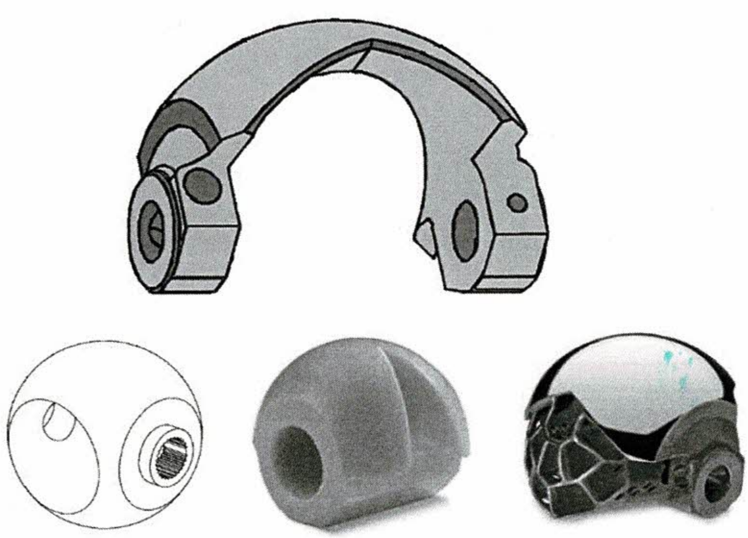

Figure 7-20. Segmented V-notch ball (top), full port ball (bottom left), characterized V-notch (bottom center), and V-notch with cavitation and noise attenuator (bottom right) (courtesy of Emerson Process Management).

The flow control characteristics of ball valves can be greatly improved by characterizing the rotating ball by designing them with a V-notch. The V-notch gives the ball valve an equal percentage inherent flow characteristic.

In addition, the V-notch prevents clogging of partially open ball valves in applications with particulates in the liquid. The V-notch design makes the valve suitable for paper stock, fibrous materials, and slurries.

V-notch ball valves have a high capacity and rangeability. They have a very tight shutoff and can operate in high temperature applications. They can be used for the control of gas, steam, clean or dirty liquids, and abrasive chemicals.

For applications which result in high noise levels through the ball valve, or if there is cavitation occurring in the ball valve, the ball can be designed with a cavitation and noise attenuator as shown in Figure 7-20.

Butterfly Valves

Butterfly valves consist of a thin wafer or contoured disc that rotates 90 degrees from the closed position to the fully open position, as shown in Figure 7-21. They are suitable for gas and liquid applications for a wide range of temperatures and pressures, and work well as both control and isolation valves.

There are a variety of disc designs that can be used for the butterfly valve. The conventional butterfly valve has a disc with geometric symmetry, as shown on the left in Figure 7-22. The disadvantage of this type of disc is the high torque that is produced on the shaft due to the flow of the fluid over the disc from about 60 to 90 degrees of valve travel. This torque can be reduced by changing the shape of the disc, such as with the fish tail design shown in the center drawing of Figure 7-22.

High performance butterfly valves have eccentric contoured discs (shown on the right in Figure 7-22) to produce an equal percentage inherent flow characteristic. These can incorporate a single, double, or triple offset with the valve body, shown in Figure 7-23. A single offset means that the centerline of the shaft is offset from the plane of the sealing surface of the disc. This allows the disc to rotate away from the seal on the body in the initial 10 degrees of rotation, with no contact in the remainder of the rotation.

A double offset incorporates an offset between the centerline of the valve shaft with the centerline of the flow passage, or disc face. This allows the disc to pull away from the seal in the valve body at initial opening, minimizing the torque required to adjust the position of the disc.

The triple offset design has a third offset between the centerline of the shaft and the cut angle of the leading edge of the sealing surface of the disc. This ensures that the disc contacts the body seal only at the final shut-off position.

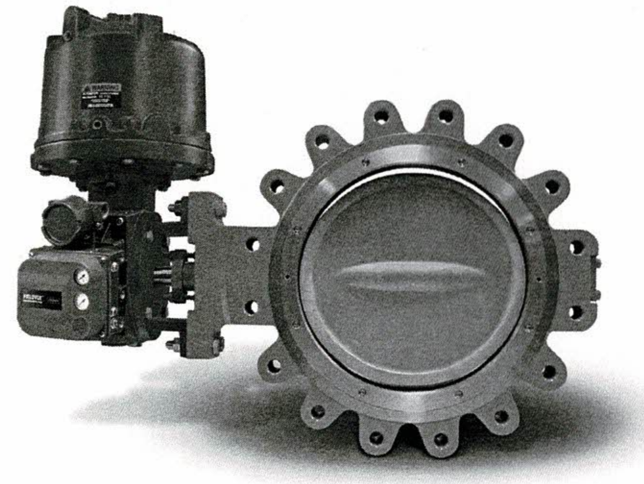

Figure 7-21. Fisher® Control-Disk butterfly valve (courtesy of Emerson Process Management).

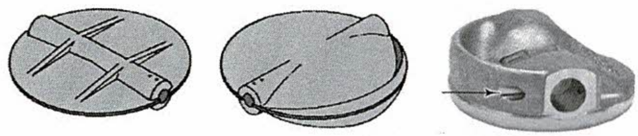

Figure 7-22. Butterfly valve disc designs: conventional (left), fish tail (center), and contoured (right) (courtesy of Emerson Process Management).

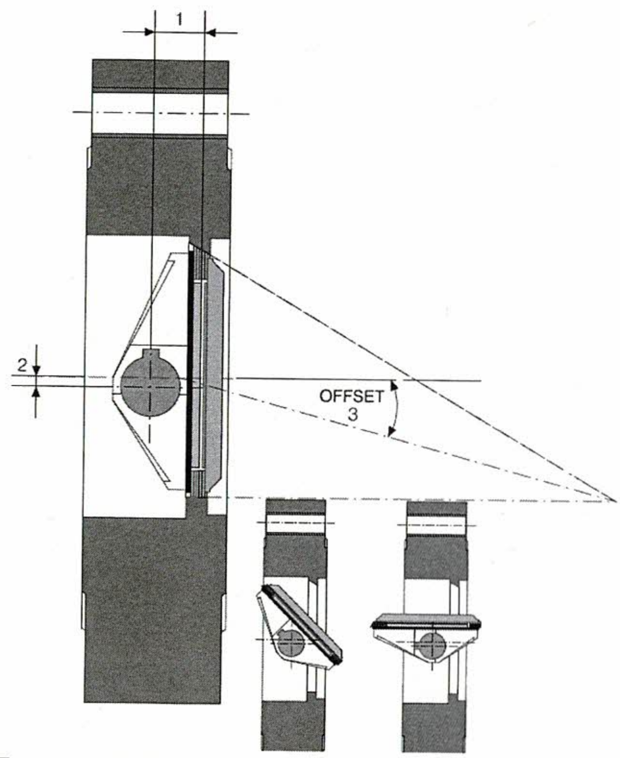

Figure 7-23. Triple offset of butterfly valve disc (courtesy of Flowseal, a Crane Co.).

Plug Valves

Plug valves are also rotary motion valves that are ideally suited for erosive fluid applications. The closing member can be a tapered or cylindrical plug with a rectangular flow passage through it, or it can be an eccentric segmented plug as shown in Figure 7-24. The plug can also incorporate various V-notch features to characterize the hydraulic performance of the valve.

Table 7-1 summarizes the features and differences between the types of control valve styles.

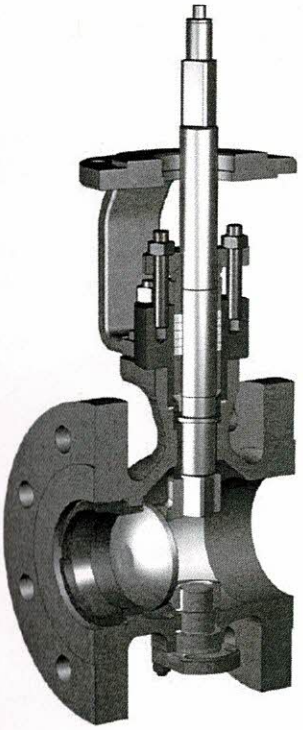

Figure 7-24. Eccentric rotating plug valve (courtesy of Flowserve).

Table 7-1: Capabilities and Features of Various Types of Control Valves (courtesy of European Valves)

	Linear motion	***Rotary Ball Valve***	***Rotary Butterfly Valve***
Size			
Nominal Size Range	¼ - 60"	½" – 48"	2" – 140"
C_V for a 4" valve	240	530	490
Maximum C_V for type	60,000	14,000	400,000
Pressure Rating			
ANSI	6,600 psi	600 psi	2,500 psi
API	20,000 psi		
Max differential pressure			
	10,000 psi	1,500 psi	2,000 psi
Operating temperature			
Maximum °F	100 to 1,500	175 to 800	300 to 1,500
Minimum (normal trim) °F	60 to -100	-20 to -115	20 to -110
End connections			
Screwed	√		
Compression fittings	√		
Flanged	√	√	√
Proprietary clamped fittings	√		
Socket weld	√		√
Butt weld	√	√	√
Wafer	√	√	√
Lug		√	√
Valve characteristics			
Equal Percentage	√	√	√
Linear	√		
Quick opening	√		√
Rangeability			
Maximum / minimum flow	50:1 to 200:1	150:1 to 300:1	100:1
Action on control failure			
Fail open	√	√	√
Fail closed	√	√	√
As-is	√	√	√
Installation options			
Space required	large	smaller	smallest
Trim servicing	in place	removal	removal
Actuator mounting	fixed	variable	variable
Operating conditions			
Corrosive	√	√	√
Abrasive	limited	X	X
Flashing	√	√	√
Cavitating	√	X	X
Hazardous emissions	bellows seal	double packing	double packing
Pulp	X	√	√
Sanitary	√	X	√

Classifying Control Valves Based on Pressure and Temperature Rating

Control valves are also classified according to the pressure and temperature of the working fluid that the valve can handle without affecting the integrity of the valve. The higher the temperature and pressure of the working fluid, the greater the wall thickness, depending on the material of construction. The valve manufacturer can increase the wall thickness of the valve or choose a different material. When the wall thickness is increased, the internal passages of the control valve may decrease which reduces the flow coefficient for a given valve size.

The American Society of Mechanical Engineers has established valve class ratings based on the allowable fluid pressure at the working temperature, as shown in Figure 7-25.

The valve class number increases with higher allowable operating pressures. Class 900, 1500, and 2500 valves are generally referred to as high pressure valves.

Also note in Figure 7-25, as the fluid temperature increases, the allowable operating pressure decreases, especially above about 750 °F. This is because the metal valve bodies become more pliable at higher temperatures, making them more susceptible to failure at higher pressures.

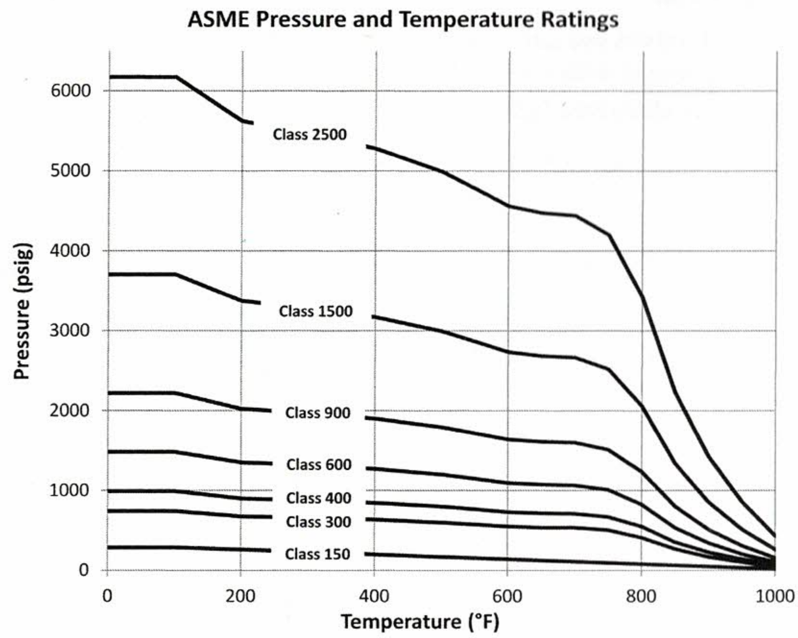

Figure 7-25. Generalized graph of the allowable pressure and temperature based on ASME valve classification.

Classifying Control Valves Based on Leak Tightness at Shutoff

Control valves can also be classified according to the amount of allowable leakage at shutoff. Many factors affect the valve's leak tighness, including whether the valve is single or double seated, how the valve plug is guided, the seating surface material, how much force the actuator can produce, the pressure differential across the closed valve, and the working fluid composition and temperature.

The ANSI valve classifications shown in Table 7-2 defines the valve class based on the tested shutoff leak rate of the valve as specified in the ANSI/FCI 70-2 standard. In addition to the maximum allowable leak rate, the standard defines the test procedure, the test fluid, and the test pressure required to meet the valve classifications.

Table 7-2: ANSI Valve Classifications Based on Leak Rate at Shutoff

Valve Class	Allowable Leak Rate (Percent of Rated Capacity)
Class I	x
Class II	0.5%
Class III	0.1%
Class IV	0.01%
Class V	0.0005 mL / minute per inch port diameter per psid
Class VI	Bubbles/min or mL/min based on valve size

Class I valves are referred to as "dust tight", but there is no actual bench test that needs to be done for these valves.

Class II, III, and IV valves are similar in type and design but have progressively lower allowable leak rates at shutoff. They are typically balanced single port or double ported valves with a metal-to-metal contact between the seat and plug.

Class V valves are tested at higher test pressures and have an extremely low allowable leak rate based on the port diameter and test pressure drop.

Class VI valves are soft seated valves where the seat and/or disc is made of a resilient material like teflon or coated with a resilient material like Stellite, a cobalt-chromium alloy designed for wear resistance. The allowable leak rate for Class VI valves is extemely low and based on the valve size.

Class IV and Class VI valves are the most commonly used control valves.

Considerations for Sizing and Selecting Control Valves

A control valve must meet the requirements of the control loop and the piping system. The design conditions are the primary sizing criteria used to select a control valve and are typically the most demanding conditions the valve will be expected to handle. The design conditions used to size the control valve represent a worst-case operating scenario.

Although sized for specific design conditions, a control valve will typically not operate continually at those conditions. For example, when a system is started, the process fluid temperature and pressure may be considerably different from the design conditions. These conditions result in changes to the density, viscosity, and vapor pressure of the fluid compared to initial design conditions. The set point may also vary greatly during startup conditions. Minimum and maximum flow conditions, as well as extended operation at conditions other than the design conditions, must be taken into account when selecting a control valve.

There are many things to consider when sizing and selecting a particular control valve for a given application, including:

- Valve size
- Flow characteristics
- Controllability
- Rangeability
- Reliability
- Fluid / material compatability
- Pressure and temperature rating
- Leak tightness

Control Valve Sizing

The valve must be properly sized to ensure it has the capacity to pass the desired flow rate. An iterative method for sizing control valves is outlined in the Instrumentation, System, and Automation Society (ISA) standard ANSI/ISA-75.01.01 *Flow Equations for Sizing Control Valves*. The standard presents the necessary information and equations needed to calculate the required flow coefficient (C_V) to size a control valve for both liquid and gas applications, in both U.S. and metric units. It also allows for adjustment of the flow coefficient for laminar flow, and for valve installations in which another valve or fitting is installed within two pipe diameters upstream of the valve inlet or six pipe diameters downstream of the valve outlet.

Sizing for Incompressible Fluid Flow

For incompressible fluid flow, Equation 7-1 is used to calculate the required flow coefficient of the control valve.

$$C_V = \frac{Q}{N_1 F_P \sqrt{\frac{P_1 - P_2}{S}}}$$ *Equation 7-1*

C_V = nominal flow coefficient

Q = volumetric flow rate

N_1 = numerical constant based on units used for flow rate and pressure

F_P = piping geometry factor

P_1 = absolute pressure measured at the valve inlet

P_2 = absolute pressure measured at the valve outlet

S = specific gravity of the fluid

The value of $N_1 = 1.0$ if units of gpm are used for flow rate and psi are used for pressure. If units other than gpm and psi are used, the standard should be consulted.

The piping geometry factor (F_P) accounts for fittings attached within two pipe diameters of the valve inlet or six pipe diameters of the outlet. The piping geometry factor is the ratio of the flow coefficient with the fittings attached, to the flow coefficient of the valve installed in a straight pipe of the same diameter, and is calculated using Equation 7-2.

$$F_P = \frac{1}{\sqrt{1 + \frac{\Sigma K}{N_2}\left(\frac{C_V}{d^2}\right)^2}}$$ *Equation 7-2*

F_P = piping geometry factor

ΣK = sum of the resistance coefficients for all attached valves and fittings

C_V = valve flow coefficient

N_2 = numerical constant based on units used for nominal valve size

d = nominal valve size

Sizing a control valve is an iterative procedure. The flow coefficient is first calculated assuming no fittings attached to the valve(the piping geometry factor $F_P = 1.0$). Control valve manufacturer data can be checked to find a valve with a fully open C_V greater than the calculated C_V. If a smaller control valve than the pipe size can be selected (for economic reasons as well as others), then reducers will need to be installed at the valve inlet and outlet.

The piping geometry factor will then need to be calculated using Equation 7-2, and the flow coefficient re-calculated using the piping geometry factor. As long as the second calculation of the flow coefficient is within the range of the selected valve, the valve will have sufficient capacity to meet the needs of the application.

If the valve is expected to be used under laminar flow conditions, an additional correction factor called the Reynolds number factor will need to be determined using the standard.

The possibility of choking and cavitation will need to be determined for the control valve being considered. This will be discussed in more detail later in this chapter.

Sizing for Compressible Fluid Flow

The sizing equations for applications involving compressible gas flow are similar to the equations for incompressible flow, with the exception that the compressible nature of the fluid is taken into account with an expansion factor, *Y*. Equation 7-3 is used to calculate the flow coefficient for a control valve in a compressible application.

$$C_V = \frac{w}{N_6 F_P Y \sqrt{x P_1 \rho}} \qquad \textit{Equation 7-3}$$

w = mass flow rate

N_6 = numerical constant based on units used for mass flow rate, pressure, and density

F_P = piping geometry factor

Y = fluid expansion factor

x = ratio of the pressure drop across the valve to the absolute inlet pressure ($\Delta P/P_1$)

P_1 = absolute pressure measured at the valve inlet

ρ = density of the fluid at the inlet pressure and temperature

The expansion factor, *Y*, is calculated using Equation 7-4.

$$Y = 1 - \frac{x}{3 F_k x_T} \qquad \textit{Equation 7-4}$$

Y = expansion factor

x = ratio of the pressure drop across the valve to the absolute inlet pressure ($\Delta P/P_1$)

F_K = specific heat ratio factor (= $k/1.4$, where k = specific heat ratio of the fluid)

x_T = rated pressure drop ratio factor, obtain by manufacturer testing in accordance with ANSI/ISA-75.02

Flow Characteristics

The selected control valve needs to have the right flow characteristics to ensure adequate response over the range of operating flow rates. All components of a control loop have inherent characteristics that define the relationship between the inlet and outlet of each component. The process itself (level, temperature, flow, or pressure control for example) will have inherent characteristics. A key to selecting a good valve for a given process is to choose one with the right inherent characteristics to make the entire loop and process as close to linear as possible over the range of control for the valve.

Controllability

Controllability involves the static aspects of the control valve such as the size and type, the flow characteristics, and the flow direction. It also involves dynamic aspects that determine how the valve responds to step changes in the control signal in response to disturbances in the process. These dynamic aspects include backlash and stiction that result in a dead band between when the control signal changes to when the valve plug or disc starts to move. Backlash is the loss of movement of the closure member in response to a change in control signal due to looseness in the mechanical linkages from the actuator to the valve plug or disc. Stiction is a result of the packing friction that prevents movement of the stem until enough force is applied by the actuator to overcome the friction.

Rangeability

The rangeability of the valve is the range of flow rates that the valve is expected to control from the minimum to the maximum flow rate. A control valve is typically sized to meet the maximum flow requirements but may be expected to operate over a wide range of conditions.

Reliability

Reliability is also a key performance indicator for a control valve. Cavitation, critical damage, noise, high pressure drop, and high outlet velocity are the most common reliability challenges a control valve has to deal with to minimize maintenance on the valve.

Cavitation and Choking

A control valve with incompressible fluid flow is susceptible to cavitation just like a centrifugal pump. Cavitation occurs when the localized static pressure within the control valve drops below the fluid's vapor pressure, causing some of the liquid to flash to a vapor state. When the fluid reaches a region of higher pressure above the vapor pressure, the vapor bubbles collapse, potentially causing severe erosion and damage to the valve trim. Cavitation also results in increased noise levels and vibration of the attached piping and supports.

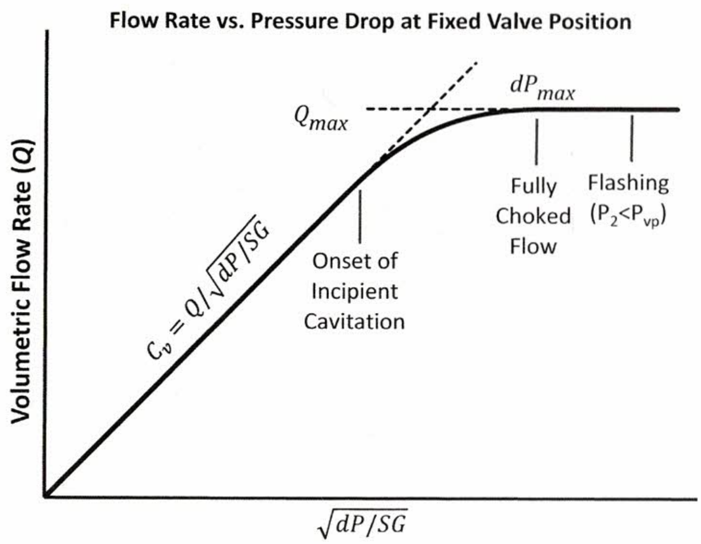

Figure 7-26. Flow vs. pressure drop at a fixed valve position showing the onset of cavitation to fully choked flow.

In addition, since the vapor bubbles occupy more volume than the same mass of liquid, the flow rate through the valve begins to become restricted and the flow rate deviates from that predicted by the flow coefficient equation. This can be seen graphically on Figure 7-26, which shows the relationship between the volumetric flow rate and the square root of the pressure drop for a valve at a fixed position. In the region where there is a linear relationship between the flow rate and the square root of the pressure drop, the fluid remains incompressible all the way through the valve. The slope of this line is the value of the flow coefficient.

As the differential pressure across the valve is increased (by lowering the downstream pressure, for example) the flow rate through the valve increases and the pressure at the vena contracta drops. At the point where the pressure reaches the vapor pressure, the onset of incipient cavitation occurs and the flow is becoming choked. As the downstream pressure continues to be lowered, a greater region around the vena contracta is filled with vapor bubbles, causing a further deviation from the linear C_V relationship. At the point where the entire region of the vena contracta has vapor flowing through it, the flow is fully choked, and no further drop in downstream pressure will result in an increase in the flow rate through the valve.

When the valve is experiencing cavitation, the vapor bubbles collapse within the valve body. If the pressure is not returned above the vapor pressure at the outlet of the valve, a condition of flashing is said to occur.

For compressible fluid applications, choked flow occurs when the fluid velocity at the vena contracta approaches the speed of sound of the compressible gas.

When selecting a control valve for a given application, it is important to check if cavitation or choking will occur throughout the range of operation of the valve. For incompressible flow, choking can be determined if the expected flow rate is above the maximum flow rate shown in Figure 7-26, or if the pressure drop across the valve is greater than the maximum pressure drop corresponding to the maximum flow rate.

Equation 7-5 can be used to calculate the maximum flow rate and Equation 7-6 can be used to calculate the maximum pressure drop for a valve without attached fittings.

$$Q_{max} = N_1 F_L C_V \sqrt{\frac{P_1 - F_F P_{vp}}{S}}$$ *Equation 7-5*

$$\Delta P_{max} = F_L^2 (P_1 - F_F P_{vp})$$ *Equation 7-6*

Q_{max} = volumetric flow rate
N_1 = numerical constant based on units used for flow rate and pressure
F_L = liquid pressure recovery factor
C_V = flow coefficient at a given valve position
P_1 = absolute pressure measured at the valve inlet
F_F = liquid critical pressure ratio factor
P_{vp} = fluid's vapor pressure
S = fluid's specific gravity

The liquid pressure recovery factor (F_L) is a function of the valve type, size, and valve position, and is determined by the valve manufacturer by testing and should be provided in the manufacturer's valve data table. For globe valves, typical values of F_L range from about 0.82 to 0.95, for ball valves from 0.55 to 0.75, and from 0.50 to 0.70 for butterfly valves.

The liquid critical pressure ratio factor (F_F) is a function of the fluid's vapor pressure and critical pressure and can be calculated using Equation 7-7.

$$F_F = 0.96 - 0.28 \sqrt{\frac{P_{vp}}{P_c}}$$ *Equation 7-7*

F_F = liquid critical pressure ratio factor
P_{vp} = fluid's vapor pressure
P_c = fluid's critical pressure

For compressible gas flow, choked flow occurs when the velocity at the vena contracta approaches the speed of sound. This occurs when $x = F_k X_T$ and therefore Y approaches 2/3 (refer to Equation 7-4).

For applications where cavitation is occuring in a control valve, or is predicted to occur, special anti-cavitation trim is available for most types of control valves. Anti-cavitation trim causes the total pressure drop across the valve to occur in stages rather than all at once, allowing some pressure recovery to occur between the stages so that the localized static pressure doesn't fall below the fluid's vapor pressure.

Fluid / Material Compatibility

The process fluid and the material of construction for the control valve must be compatible to ensure the internal trim is not adversely affected by severe duty conditions. The valve internals can be stellated to handle wear, erosion, corrosion and high temperature process conditions. Stellite alloy is a cobalt-chromium alloy designed for wear resistance.

Pressure and Temperature

When selecting a control valve, the maximum pressure and temperature must be known in order to choose the correct ASME rating for the valve.

Leak Tightness

The allowable leakage at shutoff is also a consideration for selecting a control valve. The ANSI valve classifications may influence the decision on which valve is appropriate for a given application.

Cost of Head Loss Across a Control Valve

The pressure drop across a control valve is hydraulic energy that is dissipated across the valve as head loss in the form of heat, noise, and vibration. This energy was originally added to the fluid at the pump and a portion of the operating cost of the energy can be allocated to the control valve using Equation 7-8.

$$\begin{matrix} Cost\ of\ Control \\ Valve\ Head\ Loss \end{matrix} = \frac{(0.746)\ Q\ h_L\ \rho}{(247{,}000)\ \eta_P\ \eta_m\ \eta_{VFD}} \begin{pmatrix} Operating \\ Hours \end{pmatrix} (\$/kWh) \qquad \text{Equation 7-8}$$

Examples Using the Control Valve Equations

In addition to using the control valve equations to size and select a control valve for a particular application, they can also be used to determine various operating parameters if enough information is known about the valve.

Example 7-1: Sizing a Control Valve for a Given Flow Rate and Pressure Drop

For the piping system shown in Figure 7-27, water is heated to 160 °F and pumped from the Supply Tank to the Product Tank. The flow control valve (FCV-271) must allow a maximum flow rate of 950 gpm with an inlet pressure of 75 psig and an outlet pressure of 65 psig. Fluid density at 160 °F is 61.0 lb/ft^3. Select the appropriate valve size from the manufacturer's list of equal percentage globe valves shown in Table 7-3. Assume N_I= 1.0, and F_P= 1.0.

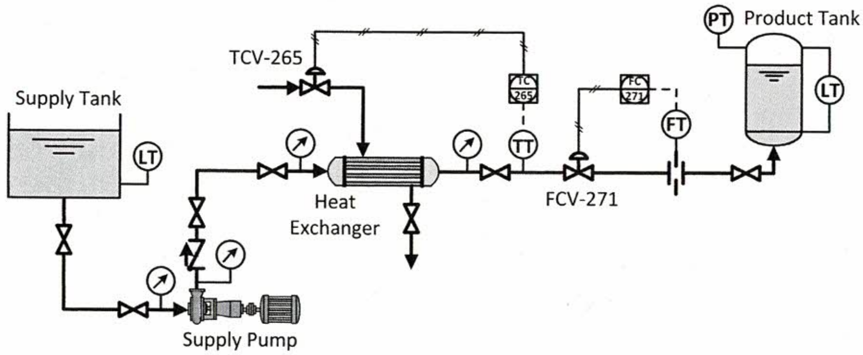

Figure 7-27. Typical piping system with a flow control valve.

Table 7-3: Valve Sizes

Valve Size	Fully Open C_V
4″	224
6″	394
8″	818

Determine Specific Gravity

$$SG = \frac{\rho_{water\ at\ 160\ °F}}{\rho_{water\ at\ 60\ °F}} = \frac{61.0\ lb/ft^3}{62.4\ lb/ft^3} = 0.978$$

Calculate the Valve's Required Flow Coefficient

$$C_V = \frac{Q}{N_1 F_P \sqrt{\frac{P_1 - P_2}{S}}} = \frac{950}{(1.0)(1.0)\sqrt{\frac{75-65}{0.978}}} = 297.1$$

From Table 7-3, a 4-inch valve would be too small, but the 6-inch and 8-inch valves have a flow coefficient greater than what is required. From an economic perspective, there is no reason to buy an 8-inch valve when a 6-inch valve would satisfy the application; also, an 8-inch valve cannot be installed in a 6-inch pipeline. The 6-inch control valve should be selected. No adjustment to the flow coefficient would be needed for piping geometry (F_P= 1.0).

Example 7-2: Calculating Pressure Drop and Head Loss

The flow control valve (FCV-271) is 65% open with a measured flow rate of 700 gpm. It is a 6-inch globe valve installed in a 6-inch pipeline with no fittings attached. A portion of its C_V profile is given in Table 7-4.

Calculate the pressure drop and associated head loss dissipated across the control valve.

Table 7-4: Control Valve Data

Position	C_V
50%	65
60%	106
70%	178
80%	270

Determine the Valve's Flow Coefficient at 65% Open

The flow coefficient at 65% open can be estimated using the values at the 60% and 70% open positions, or can be determined by linear interpolation between those values.

$$C_{V_{act}} = C_{V_1} + \frac{(Position_{act} - Position_1)}{(Position_2 - Position_1)}\left(C_{V_2} - C_{V_1}\right) = 106 + \frac{(65-60)}{(70-60)}(178-106) = 142$$

Calculate the Pressure Drop

The differential pressure across the control valve can be calculated by re-arranging Equation 7-1 and solving for the pressure drop (P_1 - P_2):

$$dP = P_1 - P_2 = \frac{Q^2\ SG}{(C_V N_1 F_P)^2} = \frac{(700)^2\ (0.978)}{(142 \times 1 \times 1)^2} = 23.8\ psi$$

Convert Pressure Drop to Head Loss

Since the pressure drop across the valve represents hydraulic energy that is dissipated across the valve, Equation 2-19 can be used:

$$h_L = \frac{144\ dP}{\rho} = \frac{144\ (23.8)}{61.0} = 56.2\ ft$$

Example 7-3: Calculating the Cost of Control Valve Head Loss

For the control valve in Example 7-2, calculate the cost of the head loss across the control valve using 8,000 hours of operation, a utility rate of \$0.10/kWh, a fixed speed pump efficiency of 70%, and motor efficiency of 95%.

$$\begin{matrix} Cost\ of\ Control \\ Valve\ Head\ Loss \end{matrix} = \frac{(0.746)\ (700)(56.2)\ (61.0)}{(247{,}000)(0.70)\ (0.95)\ (1.0)} (8{,}000\ )(\$0.10/kWh) = \$8{,}719$$

Example 7-4: Determine the Flow Rate Through a Control Valve

The flow rate through a control valve can be determined if the valve's flow coefficient and the inlet and outlet pressures of the valve are known. The flow coefficient can be determined by the valve position and the manufacturer's valve data table. The valve's inlet and outlet pressures may be measured directly with pressure gages or can be estimated using nearby pressure gages.

Determine the flow rate through FCV-271 in Figure 7-27 with the valve at 80% open and a measured inlet pressure of 80 psig and an outlet pressure of 67 psig.

From Table 7-4, at 80% open the C_V = 270. Re-arranging Equation 7-1 and solving for flow rate:

$$Q = C_V N_1 F_P \sqrt{\frac{P_1 - P_2}{S}} = (270)(1.0)(1.0) \sqrt{\frac{80 - 67}{0.978}} = 984.4\ gpm$$