Module 6.1

Control Valves

Introduction to Electric/Pneumatic Controls

Block 6 of *The Steam and Condensate Loop* considers the practical aspects of control, putting the basic control theory discussed in Block 5 into practice.

A basic control system would normally consist of the following components:
- Control valves.
- Actuators.
- Controllers.
- Sensors.

All of these terms are generic and each can include many variations and characteristics. With the advance of technology, the dividing line between individual items of equipment and their definitions are becoming less clear. For example, the positioner, which traditionally adjusted the valve to a particular position within its range of travel, can now:

- Take input directly from a sensor and provide a control function.
- Interface with a computer to alter the control functions, and perform diagnostic routines.
- Modify the valve movements to alter the characteristics of the control valve.
- Interface with plant digital communication systems.

However, for the sake of clarity at this point, each item of equipment will be considered separately.

Control Valves

Whilst a wide variety of valve types exist, this document will concentrate on those which are most widely used in the automatic control of steam and other industrial fluids. These include valve types which have linear and rotary spindle movement.

Linear types include globe valves and slide valves.

Rotary types include ball valves, butterfly valves, plug valves and their variants.

The first choice to be made is between two-port and three-port valves.
- Two-port valves 'throttle' (restrict) the fluid passing through them.
- Three-port valves can be used to 'mix' or 'divert' liquid passing through them.

Two-port valves

Globe valves

Globe valves are frequently used for control applications because of their suitability for throttling flow and the ease with which they can be given a specific 'characteristic', relating valve opening to flow.

Two typical globe valve types are shown in Figure 6.1.1. An actuator coupled to the valve spindle would provide valve movement.

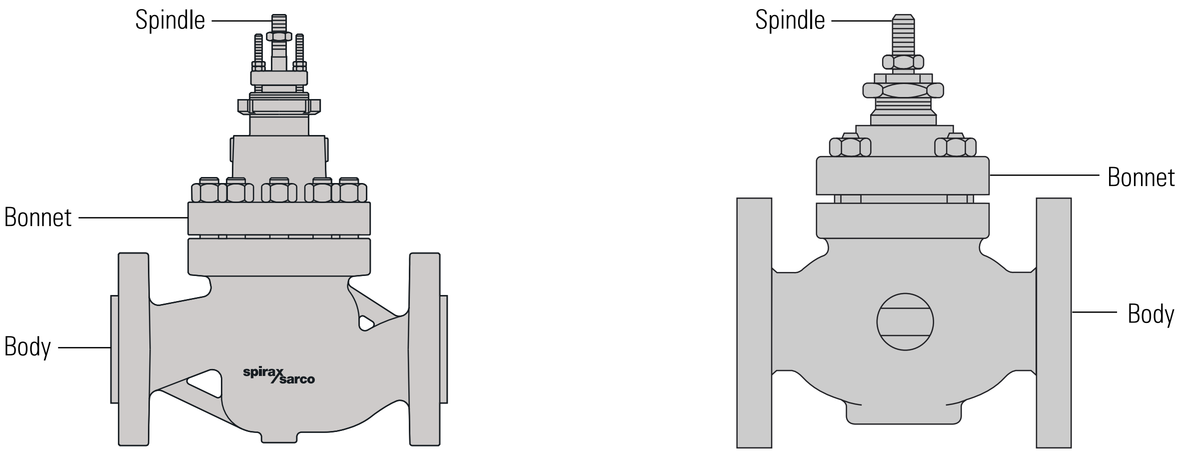


Fig. 6.1.1 Two differently shaped globe valves

The major constituent parts of globe valves are:

- The body.
- The bonnet.
- The valve seat and valve plug, or trim.
- The valve spindle (which connects to the actuator).
- The sealing arrangement between the valve stem and the bonnet.

Figure 6.1.2 is a diagrammatic representation of a single seat two-port globe valve. In this case the fluid flow is pushing against the valve plug and tending to keep the plug off the valve seat.

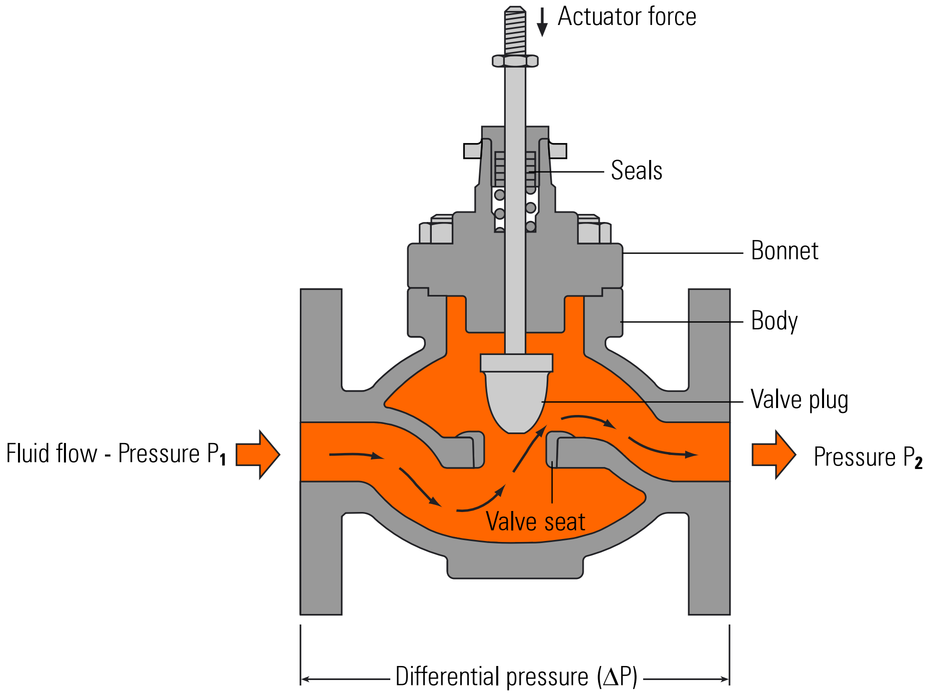


Fig. 6.1.2 Flow through a single seat, two-port globe valve

The difference in pressure upstream (P_1) and downstream (P_2) of the valve, against which the valve must close, is known as the differential pressure (ΔP). The maximum differential pressure against which a valve can close will depend upon the size and type of valve and the actuator operating it.

In broad terms, the force required from the actuator may be determined using Equation 6.1.1.

$$(A \times \Delta P) + \text{Friction allowance} = F \qquad \textbf{Equation 6.1.1}$$

Where:

A = Valve seating area (m^2)

ΔP = Differential pressure (kPa)

F = Closing force required (kN)

In a steam system, the maximum differential pressure is usually assumed to be the same as the upstream absolute pressure. This allows for possible vacuum conditions downstream of the valve when the valve closes. The differential pressure in a closed water system is the maximum pump differential head.

If a larger valve, having a larger orifice, is used to pass greater volumes of the medium, then the force that the actuator must develop in order to close the valve will also increase. Where very large capacities must be passed using large valves, or where very high differential pressures exist, the point will be reached where it becomes impractical to provide sufficient force to close a conventional single seat valve. In such circumstances, the traditional solution to this problem is the double seat two-port valve.

As the name implies, the double seat valve has two valve plugs on a common spindle, with two valve seats. Not only can the valve seats be kept smaller (since there are two of them) but also, as can be seen in Figure 6.1.3, the forces are partially balanced. This means that although the differential pressure is trying to keep the top valve plug off its seat (as with a single seat valve) it is also trying to push down and close the lower valve plug.

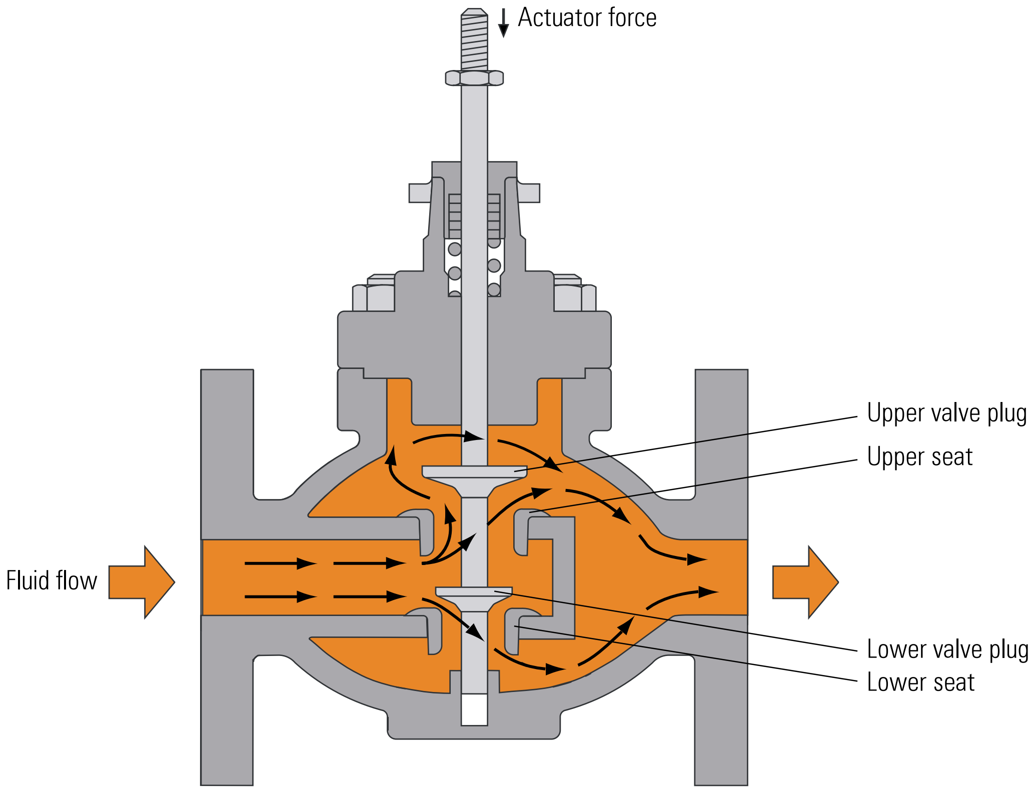


Fig. 6.1.3 Flow through a double seat, two-port valve

However, a potential problem exists with any double seat valve. Because of manufacturing tolerances and differing coefficients of expansion, few double seat valves can be guaranteed to give good shut-off tightness.

Shut-off tightness

Control valve leakage is classified with respect to how much the valve will leak when fully closed. The leakage rate across a standard double seat valve is at best Class III, (a leakage of 0.1% of full flow) which may be too much to make it suitable for certain applications. Consequently, because the flow paths through the two-ports are different, the forces may not remain in balance when the valve opens.

Various international standards exist that formalise leakage rates in control valves. The following leakage rates are taken from the British Standard BS 5793 Part 4 (IEC 60534-4). For an unbalanced standard single seat valve, the leakage rate will normally be Class IV, (0.01% of full flow), although it is possible to obtain Class V, (1.8 x 10^{-5} x differential pressure (bar) x seat diameter (mm). Generally, the lower the leakage rate the more the cost.

Balanced single seat valves

Because of the leakage problem associated with double seat valves, when a tight shut-off is required a single seat valve should be specified. The forces required to shut a single seat globe valve increase considerably with valve size. Some valves are designed with a balancing mechanism to reduce the closing force necessary, especially on valves operating with large differential pressures. In a piston-balanced valve, some of the upstream fluid pressure is transmitted via internal pathways into a space above the valve plug, which acts as a pressure balancing chamber. The pressure contained in this chamber provides a downforce on the valve plug as shown in Figure 6.1.4, balancing the upstream pressure and assisting the normal force exerted by the actuator, to close the valve.

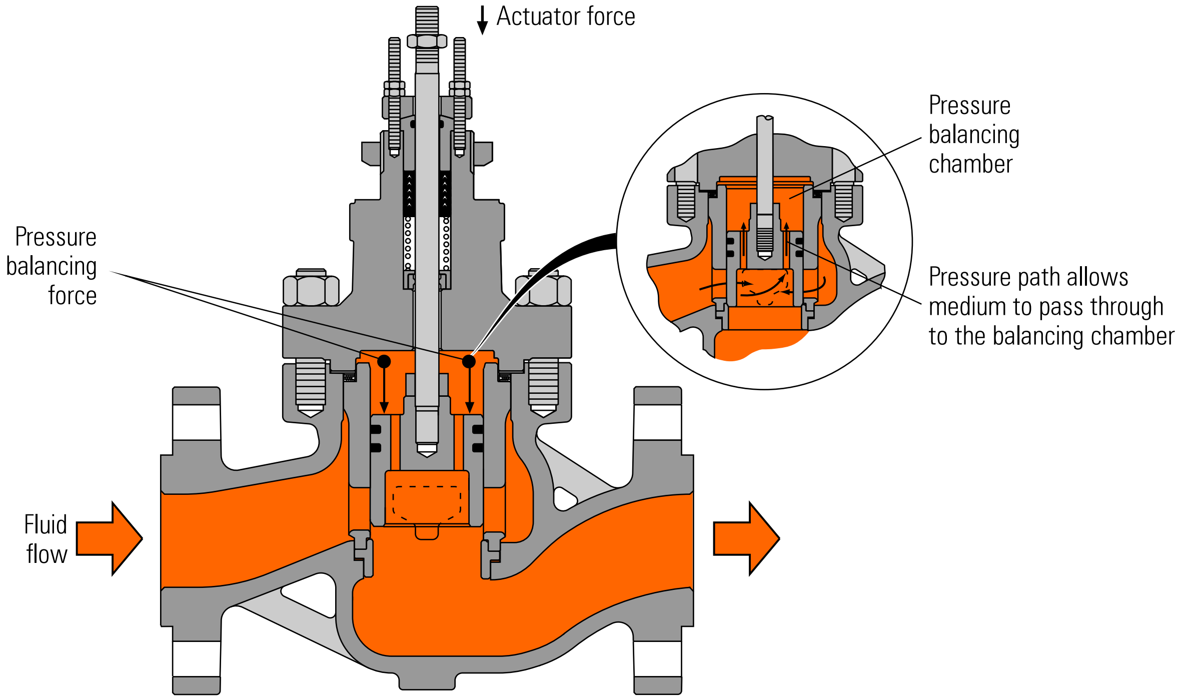


Fig. 6.1.4 A steam control valve with piston balancing

Slide valves, spindle operated

Slide valves tend to come in two different designs; wedge gate type and parallel slide type. Both types are well suited for isolating fluid flow, as they give a tight shut-off and, when open, the pressure drop across them is very small. Both types are used as manually operated valves, but if automatic actuation is required, the parallel slide valve is usually chosen, whether for isolation or control. Typical valves are shown in Figure 6.1.5.

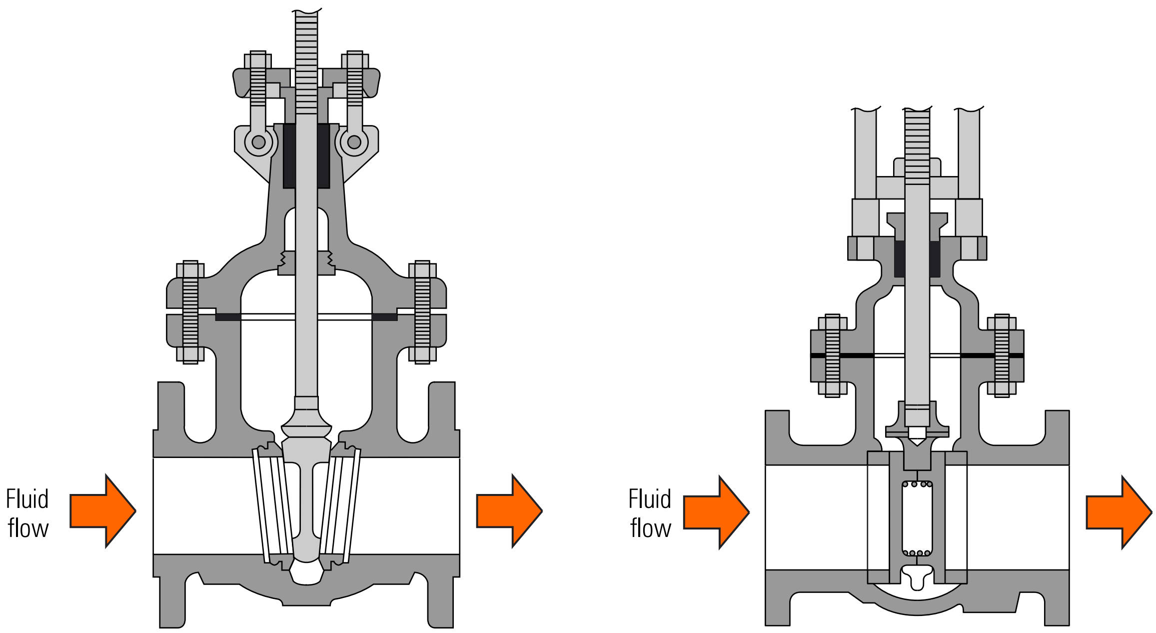


Fig. 6.1.5 Wedge gate valve and parallel slide valve (manual operation)

The parallel slide valve closes by means of two spring loaded sliding disks (springs not shown), which pass across the flow-path of the fluid, the fluid pressure ensuring a tight joint between the downstream disk and its seat. Large size parallel slide valves are used in main steam and feedlines in the power and process industries to isolate sections of the plant. Small-bore parallel slides are also used for the control of ancillary steam and water services although, mainly due to cost, these tasks are often carried out using actuated ball valves and piston type valves.

Rotary type valves

Rotary type valves, often called quarter-turn valves, include plug valves, ball valves and butterfly valves. All require a rotary motion to open and close, and can easily be fitted with actuators.

Eccentric plug valves

Figure 6.1.6 shows a typical eccentric plug valve. These valves are normally installed with the plug spindle horizontal as shown, and the attached actuator situated alongside the valve.

Plug valves may include linkages between the plug and actuator to improve the leverage and closing force, and special positioners that modify the inherent valve characteristic to a more useful equal percentage characteristic (valve characteristics are discussed in Module 6.5).

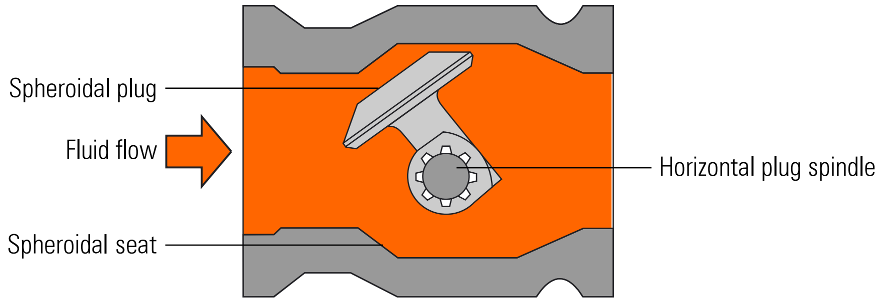


Fig. 6.1.6 Side view of an eccentric plug valve (shown in a partially open position)

Ball valves

Figure 6.1.7 shows a ball valve consisting of a spherical ball located between two sealing rings in a simple body form. The ball has a hole allowing fluid to pass through. When aligned with the pipe ends, this gives either full bore or nearly full bore flow with very little pressure drop. Rotating the ball through 90° opens and closes the flow passage. Ball valves designed specifically for control purposes will have characterized balls or seats, to give a predictable flow pattern.

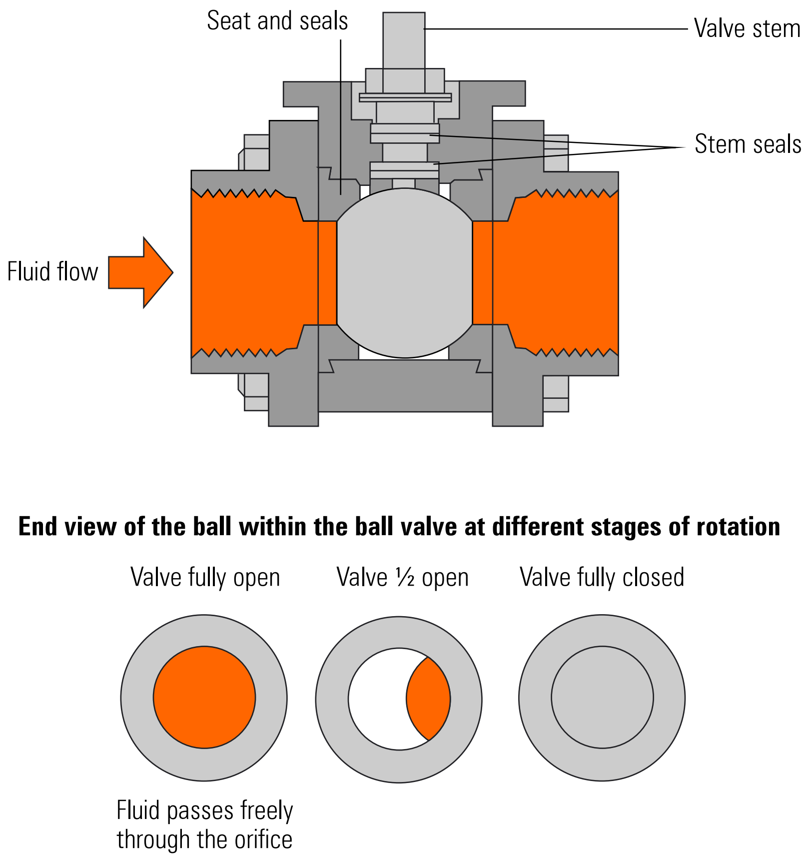
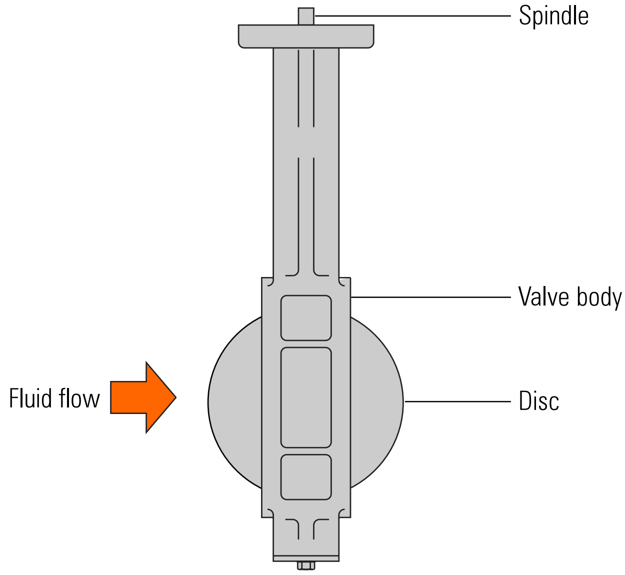


Fig. 6.1.7 Ball valve (shown in a fully open position)

Ball valves are an economic means of providing control with tight shut-off for many fluids including steam at temperatures up to 250°C (38 bar g, saturated steam). Above this temperature, special seat materials or metal-to-metal seatings are necessary, which can be expensive. Ball valves are easily actuated and often used for remote isolation and control. For critical control applications, segmented balls and balls with specially shaped holes are available to provide different flow characteristics.

Butterfly valves

Figure 6.1.8 is a simple schematic diagram of a butterfly valve, which consists of a disc rotating in trunnion bearings. In the open position the disc is parallel to the pipe wall, allowing full flow through the valve. In the closed position it is rotated against a seat, and perpendicular to the pipe wall.



End view of the disc within the butterfly valve at different stages of rotation

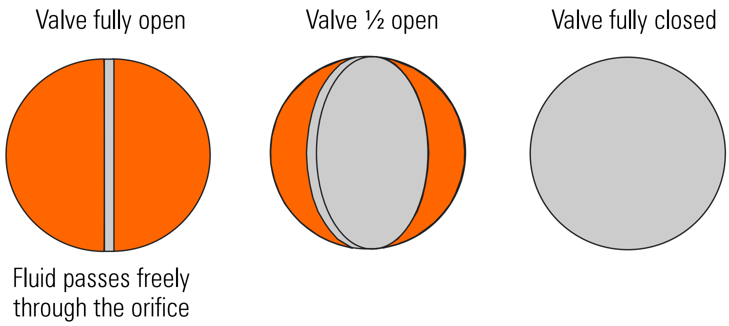


Fig. 6.1.8 Butterfly valve (shown in its open position)

Traditionally, butterfly valves were limited to low pressures and temperatures, due to the inherent limitations of the soft seats used. Currently, valves with higher temperature seats or high quality and specially machined metal-to-metal seats are available to overcome these drawbacks. Standard butterfly valves are now used in simple control applications, particularly in larger sizes and where limited turndown is required.

Special butterfly valves are available for more demanding duties.

A fluid flowing through a butterfly valve creates a low pressure drop, in that the valve presents little resistance to flow when open. In general however, their differential pressure limits are lower than those for globe valves. Ball valves are similar except that, due to their different sealing arrangements, they can operate against higher differential pressures than equivalent butterfly valves.

Options

There are always a number of options to consider when choosing a control valve. For globe valves, these include a choice of spindle gland packing material and gland packing configurations, which are designed to make the valve suitable for use on higher temperatures or for different fluids. Some examples of these can be seen in the simple schematic diagrams in Figure 6.1.9. It is worth noting that certain types of gland packing produce a greater friction with the valve spindle than others. For example, the traditional stuffing box type of packing will create greater friction than the PTFE spring-loaded chevron type or bellows sealed type. Greater friction requires a higher actuator force and will have an increased propensity for haphazard movement.

Spring-loaded packing re-adjusts itself as it wears. This reduces the need for regular manual maintenance. Bellows sealed valves are the most expensive of these three types, but provide minimal friction with the best stem sealing mechanism. As can be seen in Figure 6.1.9, bellows sealed valves usually have another set of traditional packing at the top of the valve spindle housing. This will act as a final defence against any chance of leaking through the spindle to atmosphere.

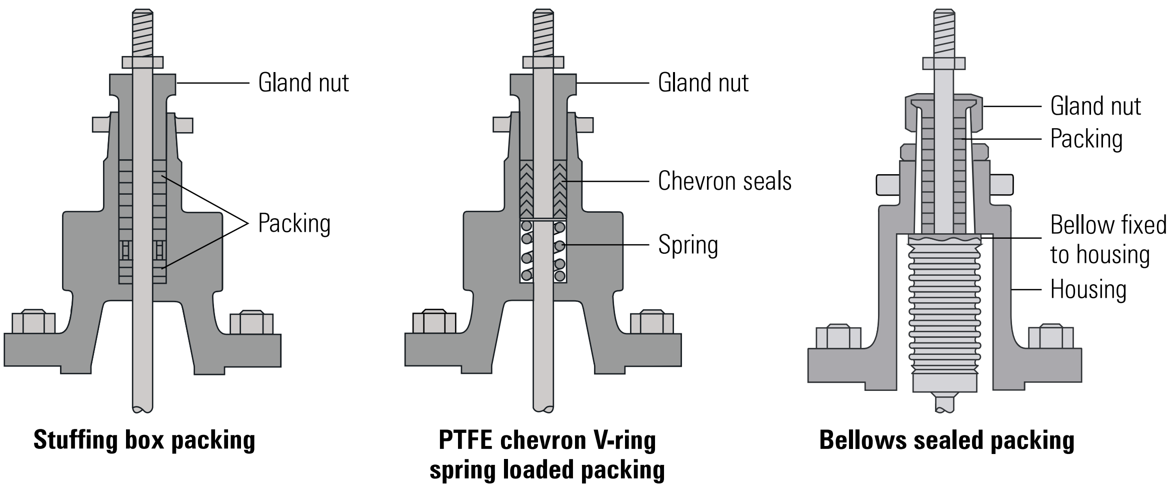


Fig. 6.1.9 Alternative gland packings

Valves also have different ways of guiding the valve plug inside the body. One common guidance method, as depicted in Figure 6.1.10, is the 'double guided' method, where the spindle is guided at both the top and the bottom of its length. Another type is the 'guided plug' method where the plug may be guided by a cage or a frame. Some valves can employ perforated plugs, which combine plug guidance and noise reduction.

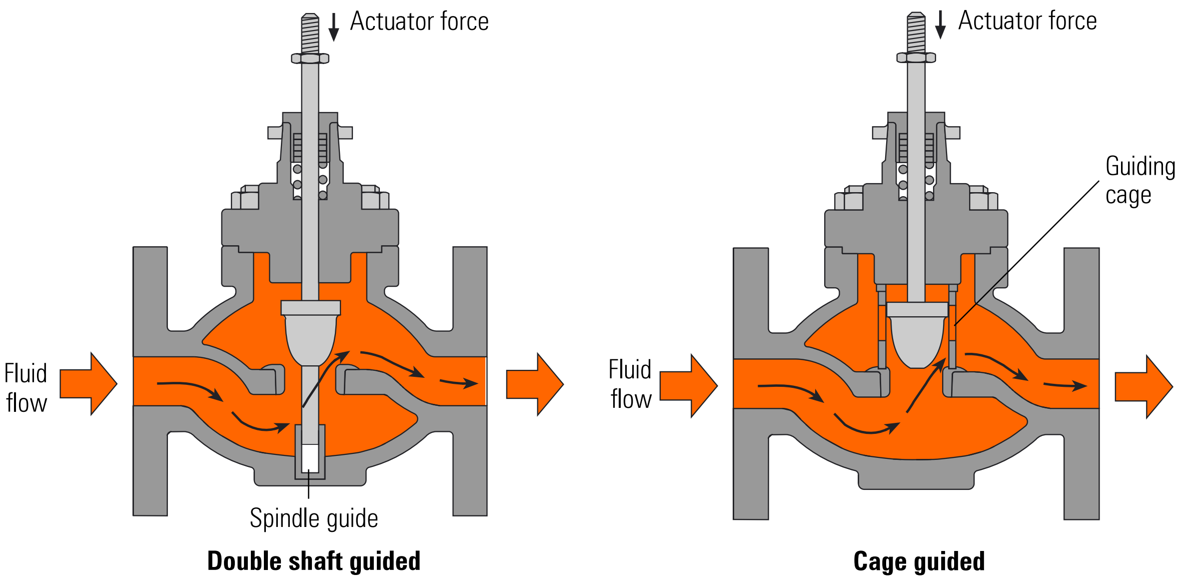


Fig. 6.1.10 Guiding arrangements

Summary of two-port valves used for automatic control

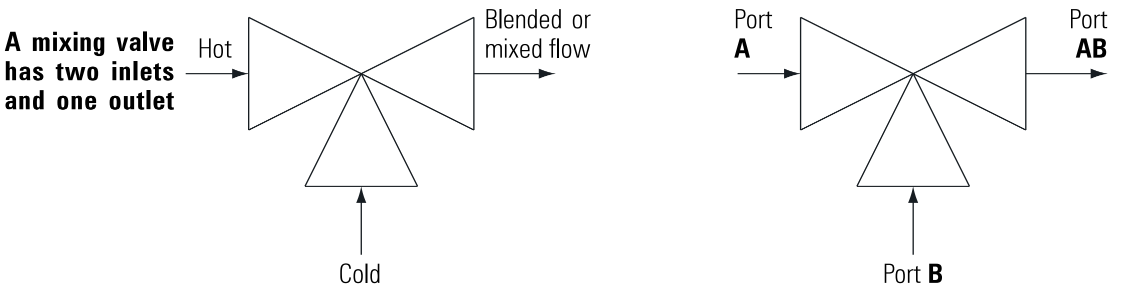
By far the most widely used valve type for the automatic control of steam processes and applications is the globe valve. It is relatively easy to actuate, it is versatile, and has inherent characteristics well suited to the automatic control needs of steam.

It should also be said that two-port automatic control valves are also used within liquid systems, such as low, medium and high temperature hot water systems, and thermal oil systems. Liquid systems carry an inherent need to be balanced with regard to mass flows. In many instances, systems are designed where two-port valves can be used without destroying the balance of distribution networks.

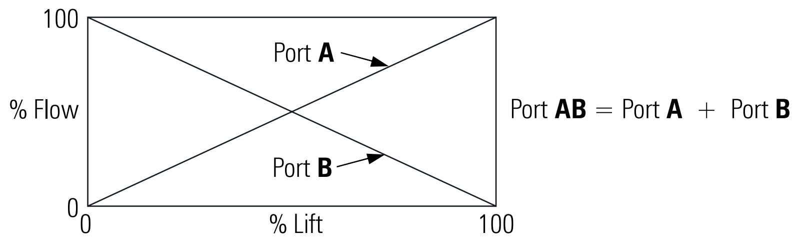
However, when two-port valves cannot be used on a liquid system, three-port valves are installed, which inherently maintain a balance across the distribution system, by acting in a diverting or mixing fashion.

Three-port valves

Three-port valves can be used for either mixing or diverting service depending upon the plug and seat arrangement inside the valve. A simple definition of each function is shown in Figure 6.1.11.



Port **AB** is termed the constant volume port. Its amount of opening is fixed by the sum of ports **A** and **B** and is not changed by the movement of the internal mechanism within the valve when the degree of opening of ports **A** and **B** is varied. A linear characteristic is normally used to provide the constant output volume condition.



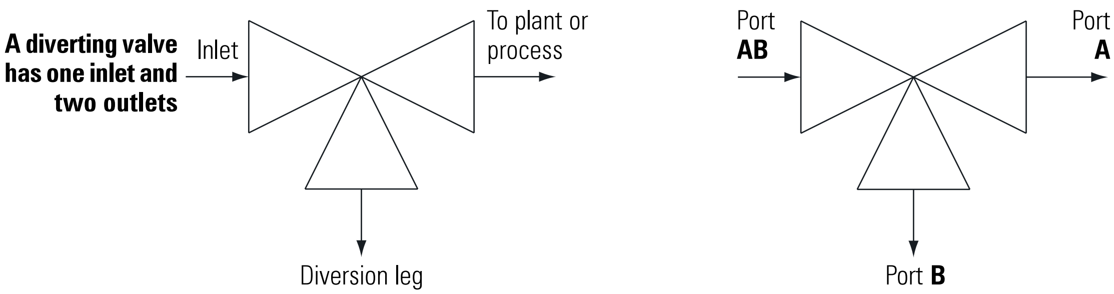


Fig. 6.1.11 Three-port valve definition

There are three basic types of three-port valve:
- Piston valve type.
- Globe plug type.
- Rotating shoe type.

Piston valves

This type of valve has a hollow piston, (Figure 6.1.12), which is moved up and down by the actuator, covering and correspondingly uncovering the two-ports **A** and **B**. Port **A** and port **B** have the same overall fluid transit area and, at any time, the cumulative cross-sectional area of both is always equal. For instance, if port **A** is 30% open, port **B** is 70% open, and vice versa. This type of valve is inherently balanced and is powered by a self-acting control system. **Note:** The porting configuration may differ between manufacturers.

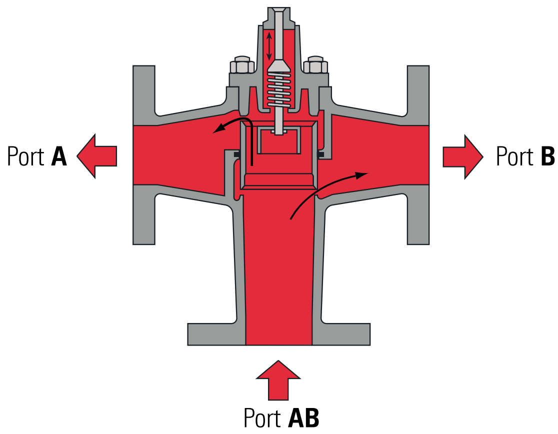


Fig. 6.1.12 Piston valve (shown as a diverting valve)

Globe type three-port valves (also called 'lift and lay')

Here, the actuator pushes a disc or pair of valve plugs between two seats (Figure 6.1.13), increasing or decreasing the flow through ports **A** and **B** in a corresponding manner.

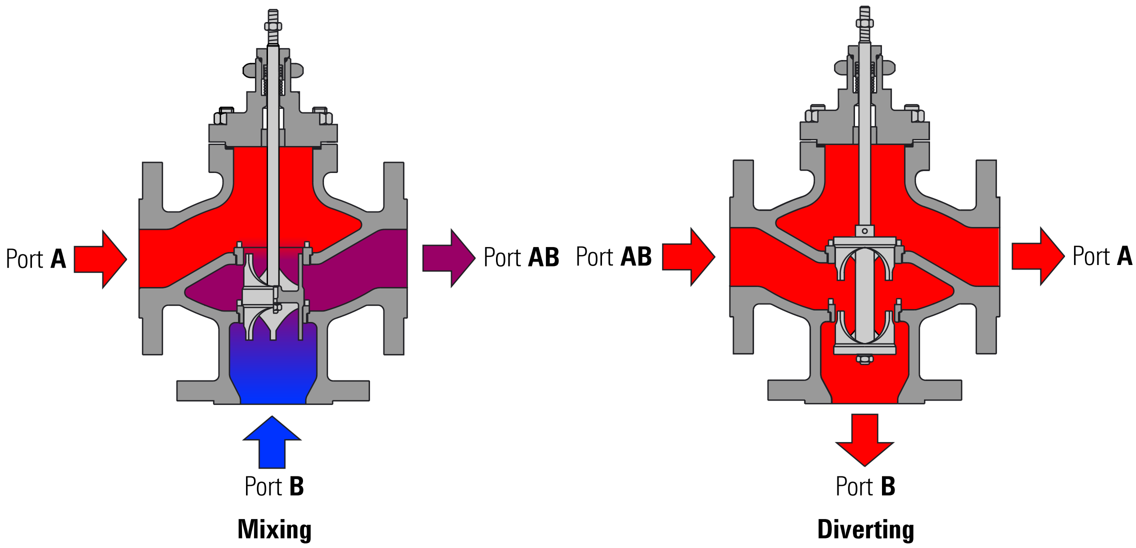


Fig. 6.1.13 Globe type three-port valves

Note: A linear characteristic is achieved by profiling the plug skirt (see Figure 6.1.14).

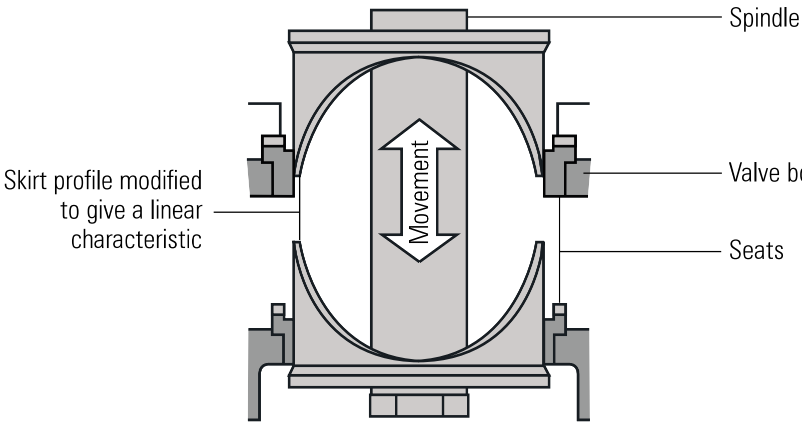


Fig. 6.1.14 Plug skirt modified to give a linear characteristic

Rotating shoe three-port valve

This type of valve employs a rotating shoe, which shuttles across the port faces. The schematic arrangement in Figure 6.1.15 illustrates a mixing application with approximately 80% flowing through port **A** and 20% through port **B,** 100% to exit through port **AB**.

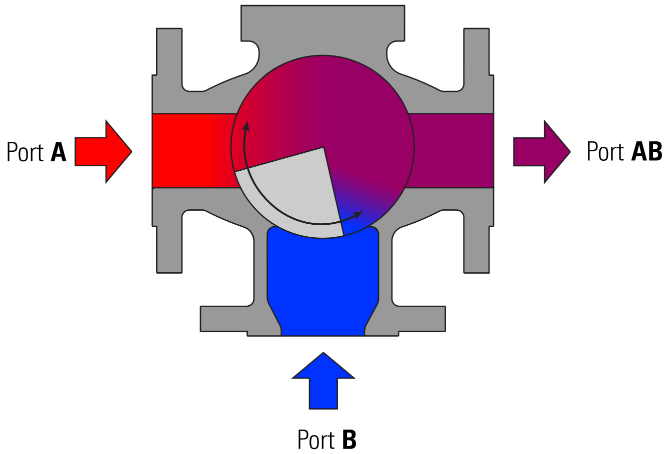


Fig. 6.1.15 Rotating shoe on a mixing application

Using three-port valves

Not all types can be used for both mixing and diverting service. Figure 6.1.16 shows the incorrect application of a globe valve manufactured as a mixing valve but used as a diverting valve.

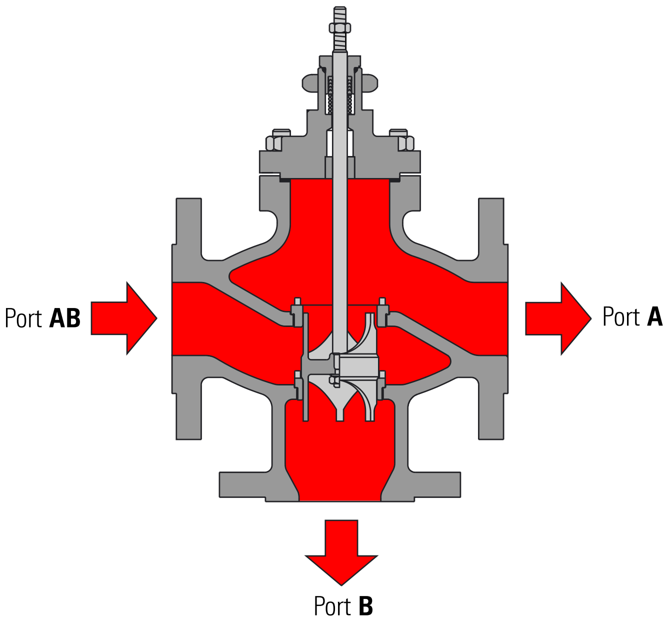


Fig. 6.1.16 Three-port mixing valve used incorrectly as a diverting valve

The flow entering the valve through port **AB** can leave from either of the two outlet ports **A** or **B**, or a proportion may leave from each. With port **A** open and port **B** closed, the differential pressure of the system will be applied to one side of the plug.

When port **A** is closed, port **B** is open, and differential pressure will be applied across the other side of the plug. At some intermediate plug position, the differential pressure will reverse. This reversal of pressure can cause the plug to move out of position, giving poor control and possible noise as the plug 'chatters' against its seat.

To overcome this problem on a plug type valve designed for diverting, a different seat configuration is used, as shown in Fig. 6.1.17. Here, the differential pressure is equally applied to the same sides of both valve plugs at all times.

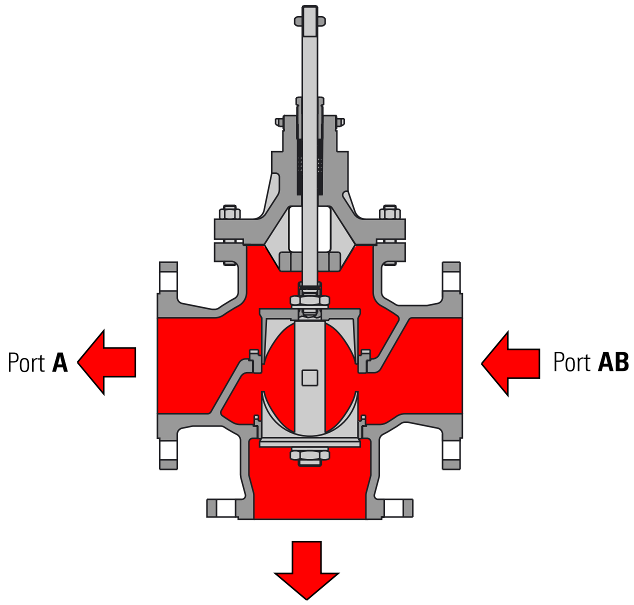


Fig. 6.1.17 Plug type diverting valve

In closed circuits, it is possible to use mixing valves or diverting valves, depending upon the system design, as depicted in Figures 6.1.18 and 6.1.19.

In Figure 6.1.18, the valve is designed as a mixing valve as it has two inlets and one outlet. However, when placed in the return pipework from the load, it actually performs a diverting function, as it diverts hot water away from the heat exchanger.

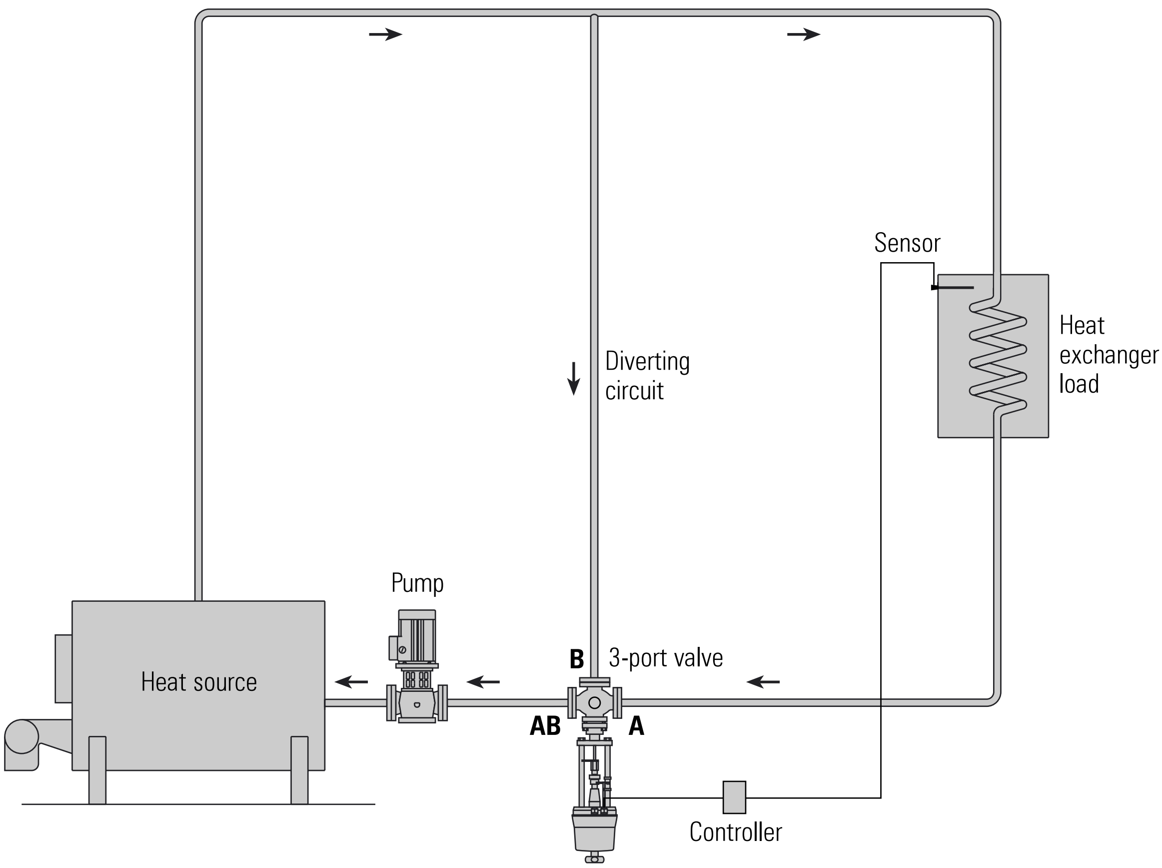


Fig. 6.1.18 Mixing Valve installed on the return pipework

Consider the mixing valve used in Figure 6.1.18, when the heat exchanger is calling for maximum heat, perhaps at start-up, port **A** will be fully open, and port **B** fully closed. The whole of the water passing from the boiler is passed through the heat exchanger and passes through the valve via ports **AB** and **A**. When the heat load is satisfied, port **A** will be fully closed and port **B** fully open, and the whole of the water passing from the boiler bypasses the load and passes through the valve via ports **AB** and **B**. In this sense, the water is being diverted from the heat exchanger in relation to the requirements of the heat load.

The same effect can be achieved by installing a diverting valve in the flow pipework, as depicted by Figure 6.1.19.

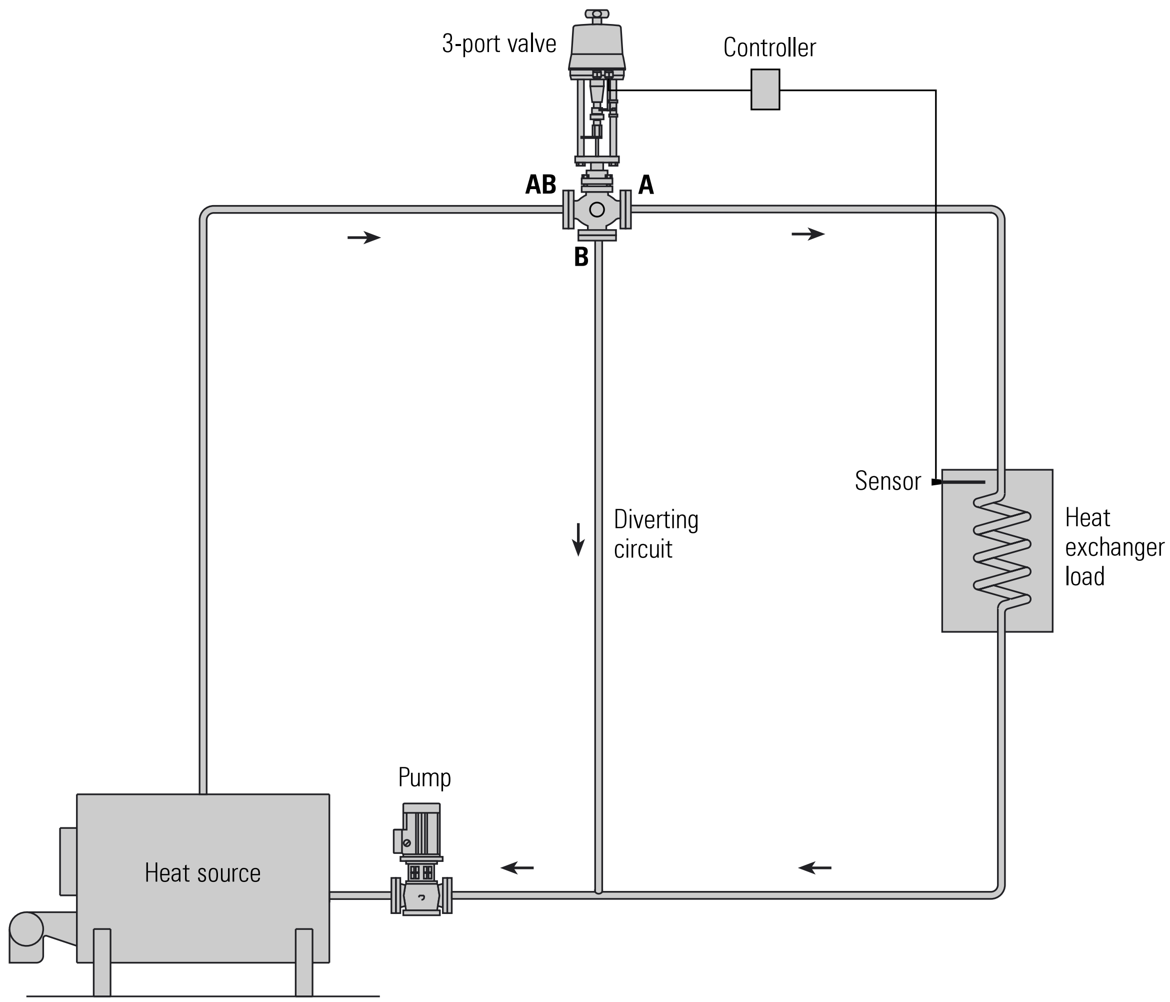


Fig. 6.1.19 Diverting valve installed on the flow pipework

Questions

1. What would an operating control system normally consist of?

a| Valve ☐

b| Valve and actuator ☐

c| Valve, actuator and controller ☐

d| Valve, actuator, controller and sensor ☐

2. What is the basic difference between 2-port and 3-port control valves?

a| 2-port valves restrict the fluid flow, 3-port valves mix or divert ☐

b| 2-port valves are only for gases, 3-port valves are only for liquids ☐

c| 2-port valves use electrical actuators, 3-port valves use pneumatic ☐

d| 2-port valves are steel, 3-port valves are bronze ☐

3. What is the basic difference between a spindle valve and a rotary valve?

a| Spindle valves have higher capacity for the same physical size ☐

b| Plug movement is in/out for spindle, side/side for rotary ☐

c| Spindle valves can only operate in a vertical plane ☐

d| Only spindle valves need valve packing ☐

4. A valve has a plug area of 500 mm², a differential pressure of 1 000 kPa, and a friction allowance of 10%. What is the minimum actuator closing force?

a| 55 kN ☐

b| 550 kN ☐

c| 0.55 kN ☐

d| 5.5 kN ☐

5. What is the main disadvantage of a double seat valve?

a| It costs more than a single seat valve ☐

b| The valve body is larger than a single seat valve of the same capacity ☐

c| It is more difficult to maintain ☐

d| It does not give a tight shut-off when fully closed ☐

6. What benefit does the bellows seal arrangement have over a traditional type of stuffing box valve packing?

a| The spindle movement produces less friction ☐

b| Fluid is less likely to leak through the spindle bonnet ☐

c| The valve operation is smoother ☐

d| All of the above ☐

Answers

1: d, 2: a, 3: b, 4: d, 5: d, 6: d

Module 6.2

Control Valve Capacity

Introduction to Valve Capacity

A control valve must, as its name suggests, have a controlling influence on the process. Whilst details such as connection sizes and materials of construction are vitally important, they do not give any indication of the control exerted by the valve.

Control valves adjust processes by altering:

- **Flowrate -** For example, the amount of steam or water that enters the process equipment.

  With a two-port valve for example, as the valve moves to the closed position, less steam flows, and less heat is added to the process.

  With a three-port valve for example, as the valve plug moves to a new position, it diverts hot water away from the process.

And/or

- **Differential pressure -** This is defined as the difference between the pressure at the valve inlet and the pressure at the valve outlet (see Figure 6.2.1).

  For any given valve orifice size, the greater the differential pressure the greater the flowrate, within certain limitations.

  With saturated steam, the lower its pressure, the lower its temperature, and less heat transfer will occur in the heat exchanger.

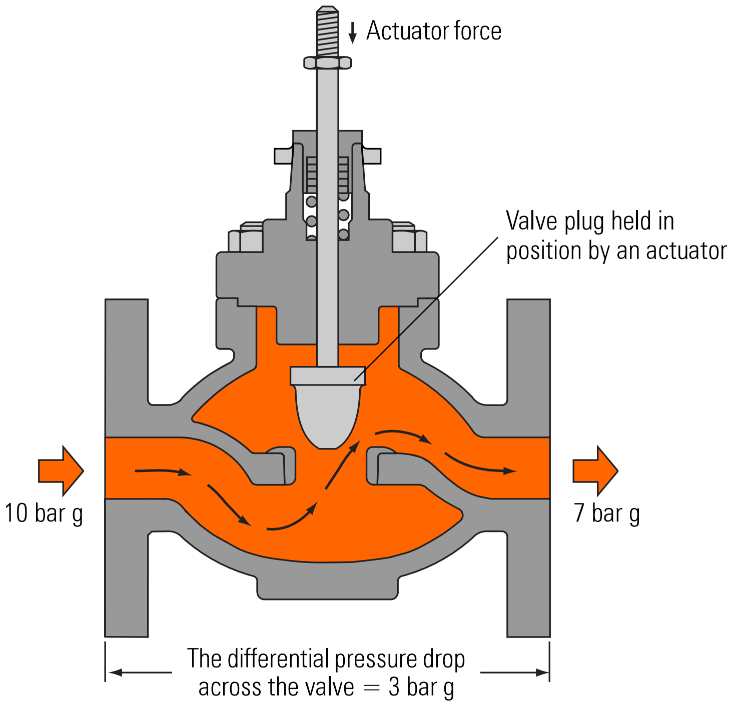


Fig. 6.2.1 Differential pressure across a valve

These two factors (**a**) **Flowrate** and (**b**) **Differential pressure** are brought together as a **flow coefficient** or 'capacity index' as it is sometimes termed.

The flow coefficient allows:

- The performance of valves to be compared.
- The differential pressure across a valve to be determined from any flowrate.
- The flowrate through a control valve to be determined for a given differential pressure.

Because many different units of measurement are used around the world, a number of flow coefficients are available, and it is worthwhile understanding their definitions. Table 6.2.1 identifies and defines the most commonly encountered capacity indices.

Table 6.2.1 Symbols and definitions used to identify and quantify flow through a control valve

K_v	Flowrate in m^3/h of water at a defined temperature, typically between 5°C and 40°C, that will create a pressure drop of one bar across a valve orifice. (Widely used in Europe)
K_{vs}	The actual or stated K_v value of a particular valve when fully open, constituting the valve flow coefficient, or capacity index.
K_{vr}	The K_{vr} is the flow coefficient required by the application.
C_v	The flowrate in gallons per minute of water at a defined temperature, typically between 40°F and 100°F that will create a pressure drop of one pound per square inch. (Widely used in the US, and certain other parts of the world). Care needs to be taken with this term, as both C_v Imperial and C_v US exist. Whilst the basic definition is the same, the actual values are slightly different because of the difference between Imperial and US gallons.
A_v	Flowrate in m^3/s of water that will create a pressure drop of one Pascal.

For conversion:

C_v (Imperial) = K_v x 0.962 658

C_v (US) = K_v x 1.156 099

A_v = 2.88 x 10^{-5} C_v (Imperial)

The flow coefficient, K_{vs} for a control valve is essential information, and is usually stated, along with its other data, on the manufacturer's technical data sheets.

Control valve manufacturers will usually offer a number of trim sizes (combination of valve seat and valve plug) for a particular valve size. This may be to simplify the pipework by eliminating the need for reducers, or to reduce noise.

A typical range of K_{vs} flow coefficients available for a selection of valves is shown in Table 6.2.2

Table 6.2.2 K_{vs} values for a typical range of valves

Sizes	DN15	DN20	DN25	DN32	DN40	DN50	DN65	DN80	DN100
K_{vs}	4.0	6.3	10.0	16.0	25.0	36.0	63.0	100.0	160.0
	2.5	4.0	6.3	10.0	16.0	25.0	36.0	63.0	100.0
	1.6	2.5	4.0	6.3	10.0	16.0	25.0	36.0	63.0
	1.0	1.6	2.5	4.0	6.3	10.0	16.0	25.0	36.0

The relationship between flowrates, differential pressures, and the flow coefficients will vary depending upon the type of fluid flowing through the valve. These relationships are predictable and satisfied by equations, and are discussed in further detail in:

- Module 6.3 - Control Valve Sizing for Water Systems.
- Module 6.4 - Control Valve Sizing for Steam Systems.

Questions

1. What two basic properties enable control valves to 'control'?

a| Temperature and pressure ☐

b| Pressure and valve movement ☐

c| Pressure and flowrate ☐

d| Temperature and flowrate ☐

2. For a given orifice size, which of the following is true?

a| The greater the pressure drop, the less the flow ☐

b| The greater the flow, the less the pressure drop ☐

c| The greater the pressure drop, the greater the flow ☐

d| The less the flow, the greater the pressure drop ☐

3. Which of the following is recognised as a valve flow coefficient for a fully open valve?

a| K_v ☐

b| C_v ☐

c| A_v ☐

d| K_{vs} ☐

Answers

1: c, 2: c, 3: d,

Module 6.3

Control Valve Sizing for Water Systems

Control Valve Sizing For Water Systems

Sizing valves for water service

In order to size a valve for a water application, the following must be known:
- The volumetric flowrate through the valve.
- The differential pressure across the valve.

The control valve can be sized to operate at a certain differential pressure by using a graph relating flowrate, pressure drop, and valve flow coefficients.

Alternatively, the flow coefficient may be calculated using a formula. Once determined, the flow coefficient is used to select the correct sized valve from the manufacturer's technical data.

Historically, the formula for flow coefficient was derived using Imperial units, offering measurement in terms of gallons/minute with a differential pressure of one pound per square inch. There are two versions of the Imperial coefficient, a British version and an American version, and care must be taken when using them because each one is different, even though the adopted symbol for both versions is 'C_v'. The British version uses Imperial gallons, whilst the American version uses American gallons, which is 0.833 the volume of an Imperial gallon. The adopted symbol for both versions is C_v.

The metric version of flow coefficient was originally derived in terms of cubic metres an hour (m^3/h) of flow for a differential pressure measured in kilogram force per square metre (kgf/m^2). This definition had been derived before an agreed European standard existed that defined K_v in terms of SI units (bar). However, an SI standard has existed since 1987 in the form of IEC 534 -1 (Now EN 60534 -1). The standard definition now relates flowrate in terms of m^3/h for a differential pressure of 1 bar. Both metric versions are still used with the adopted symbol K_v, and although the difference between them is quite small, it is important to be certain or to make clear which one is being used. Some manufacturers mistakenly quote K_v conversion values without qualifying the unit of pressure differential.

Table 6.3.1 converts the different types of flow coefficient mentioned above:

Table 6.3.1 Multiplication factors for flow coefficient conversion between K_v and C_v

Multiply	K_v (bar)	K_v (kgf)	C_v (UK)	C_v (US)
K_v (bar)	1.00	1.01	0.96	1.16
K_v (kgf)	0.99	1.00	0.97	1.17
C_v (UK)	1.04	1.05	1.00	1.20
C_v (US)	0.87	0.88	0.83	1.00

For example, multiply K_v (bar) by 1.16 to convert to C_v (US).

The K_v version quoted in these Modules is always measured in terms of K_v (bar), that is units of m^3/h bar, unless otherwise stated.

For liquid flow generally, the formula for K_v is shown in Equation 6.3.1.

$$K_v = \dot{V}\sqrt{\frac{G}{\Delta P}} \qquad \textbf{Equation 6.3.1}$$

Where:

K_v = Flow of liquid that will create a pressure drop of 1 bar (m^3/ h bar)
$\dot{V}$ = Flowrate (m^3/h)
G = Relative density/specific gravity of the liquid (dimensionless). Note: Relative density is a ratio of the mass of a liquid to the mass of an equal volume of water at 4°C
ΔP = Pressure drop across the valve (bar)

Sometimes, the volumetric flowrate needs to be determined, using the valve flow coefficient and differential pressure.

Rearranging Equation 6.3.1 gives: $\dot{V} = K_v \sqrt{\dfrac{\Delta P}{G}}$

For water, G = 1, consequently the equation for water may be simplified to that shown in Equation 6.3.2.

$$\dot{V} = K_v \sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Example 6.3.1

10 m³/h of water is pumped around a circuit; determine the pressure drop across a valve with a K_v of 16 by using Equation 6.3.2:

$$\dot{V} = K_v \sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Where:

$\dot{V}$ = 10 m³/h

K_v = 16

$$10 = 16\sqrt{\Delta P}$$

$$\Delta P = \left(\frac{10}{16}\right)^2$$

$$\mathbf{\Delta P = 0.39\ bar}$$

Alternatively, for this example the chart shown in Figure 6.3.1, may be used. (Note: a more comprehensive water K_v chart is shown in Figure 6.3.2):

1. Enter the chart on the left hand side at 10 m³/h.
2. Project a line horizontally to the right until it intersects the K_v = 16 (estimated).
3. Project a line vertically downwards and read the pressure drop from the 'X' axis (approximately 40 kPa or 0.4 bar).

Note: Before sizing valves for liquid systems, it is necessary to be aware of the characteristics of the system and its constituent apparatus such as pumps.

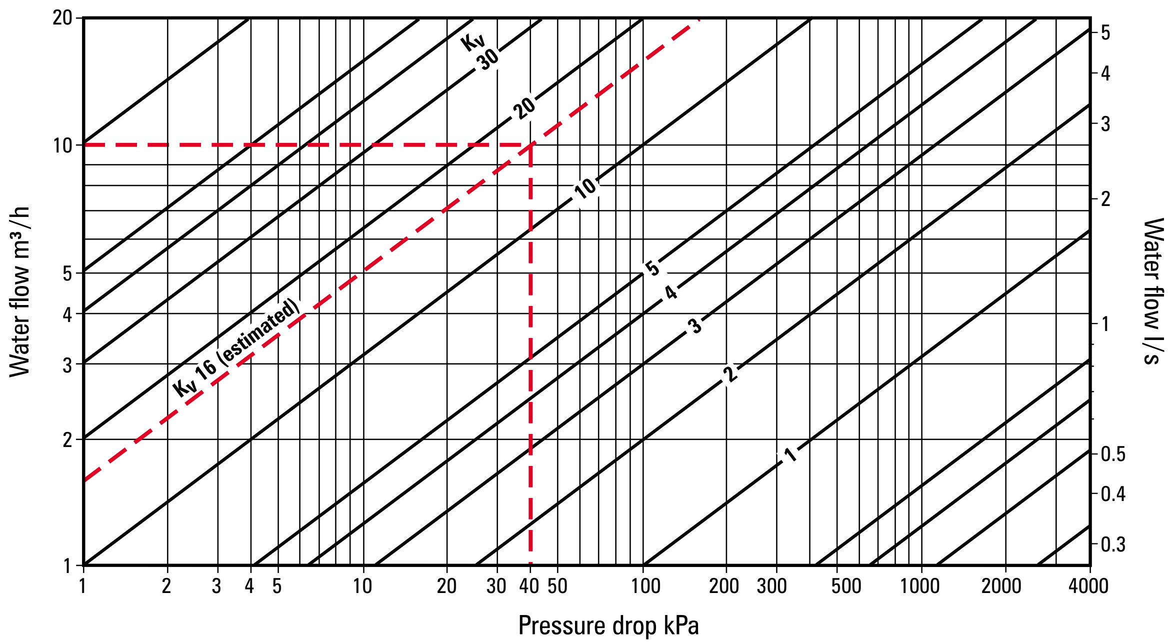


Fig. 6.3.1 Extract from the water K_v chart Figure 6.3.2

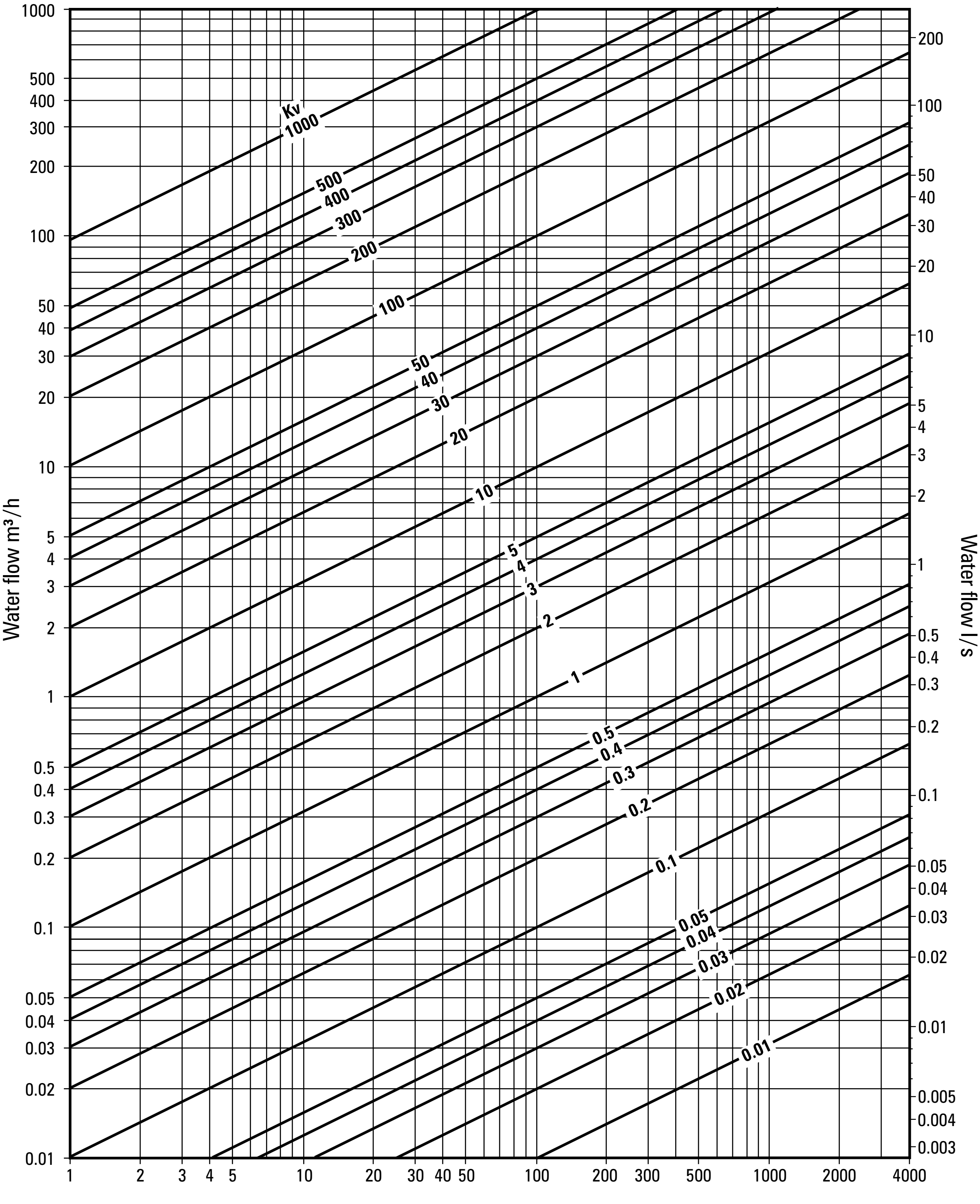


Fig. 6.3.2 Water K_v chart

Pumps

Unlike steam systems, liquid systems require a pump to circulate the liquid. Centrifugal pumps are often used, which have a characteristic curve similar to the one shown in Figure 6.3.3. Note that as the flowrate increases, the pump discharge pressure falls.

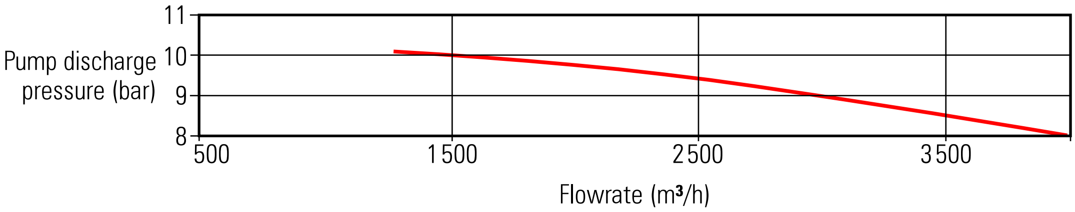


Fig. 6.3.3 Typical pump performance curve

Circulation system characteristics

It is important not only to consider the size of a water control valve, but also the system in which the water circulates; this can have a bearing on which type and size of valve is used, and where it should be positioned within the circuit.

As water is circulated through a system, it will incur frictional losses. These frictional losses may be expressed as pressure loss, and will increase in proportion to the square of the velocity and the flowrate. The flowrate can be calculated at any other pressure loss by using Equation 6.3.3, where V_1 and V_2 must be in the same units, and P_1 and P_2 must be in the same units. V_1, V_2, P_1 and P_2 are defined below.

$$\frac{V_1^2}{V_2^2} = \frac{P_1}{P_2} \qquad \textbf{Equation 6.3.3}$$

Where:

V_1 = Flowrate at pressure loss P_1

V_2 = Flowrate at pressure loss P_2

Example 6.3.2

It is observed that the flowrate (V_1) through a certain sized pipe is 2 500 m³/h when the pressure loss (P_1) is 4 bar. Determine the pressure loss (P_2) if the flowrate (V_2) were 3 500 m³/h, using Equation 6.3.3.

$$\frac{V_1^2}{V_2^2} = \frac{P_2}{P_1}$$

$$P_2 = P_1 \times \frac{V_1^2}{V_2^2}$$

$$P_2 = 4 \times \frac{3\,500^2}{2\,500^2}$$

$$P_2 = 4 \times \frac{1225 \times 10^4}{625 \times 10^4}$$

$$\mathbf{P_2 = 7.84\ bar}$$

It can be seen that as more liquid is pumped through the same size pipe, the flowrate will increase. On this basis, a system characteristic curve, like the one shown in Figure 6.3.4, can be created using Equation 6.3.3, where the flowrate increases in accordance to the square law.

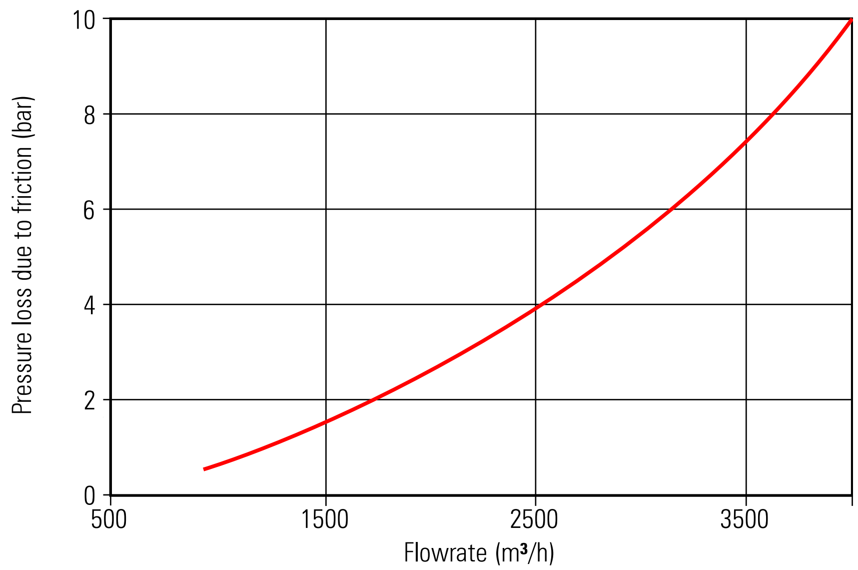


Fig. 6.3.4 Typical system curve

Actual performance

It can be observed from the pump and system characteristics, that as the flowrate and friction increase, the pump provides less pressure. A situation is eventually reached where the pump pressure equals the friction around the circuit, and the flowrate can increase no further.

If the pump curve and the system characteristic curve are plotted on the same chart - Figure 6.3.5, the point at which the pump curve and the system characteristic curve intersect will be the actual performance of the pump/circuit combination.

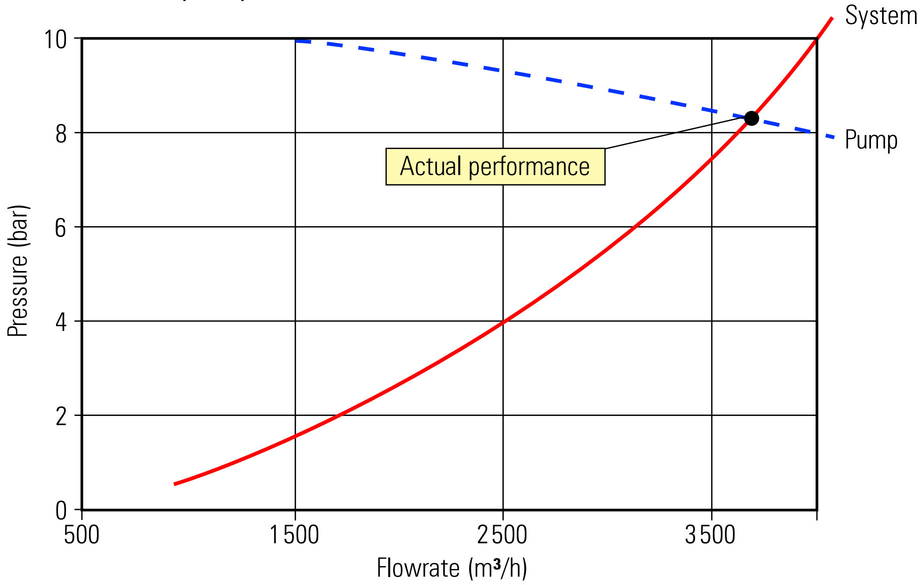


Fig. 6.3.5 Typical system performance curve

Three-port valve

A three-port valve can be considered as a constant flowrate valve, because, whether it is used to mix or divert, the total flow through the valve remains constant. In applications where such valves are employed, the water circuit will naturally split into two separate loops, constant flowrate and variable flowrate.

The simple system shown in Figure 6.3.6 depicts a mixing valve maintaining a constant flowrate of water through the 'load' circuit. In a heating system, the load circuit refers to the circuit containing the heat emitters, such as radiators in a building.

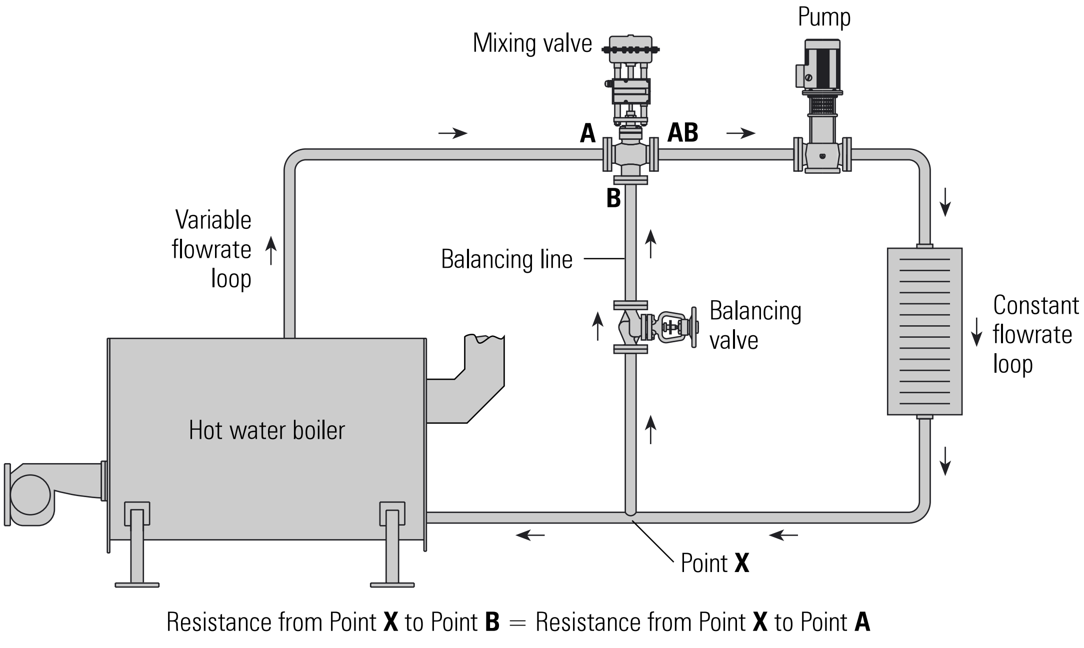


Fig. 6.3.6 Mixing valve (constant flowrate, variable temperature)

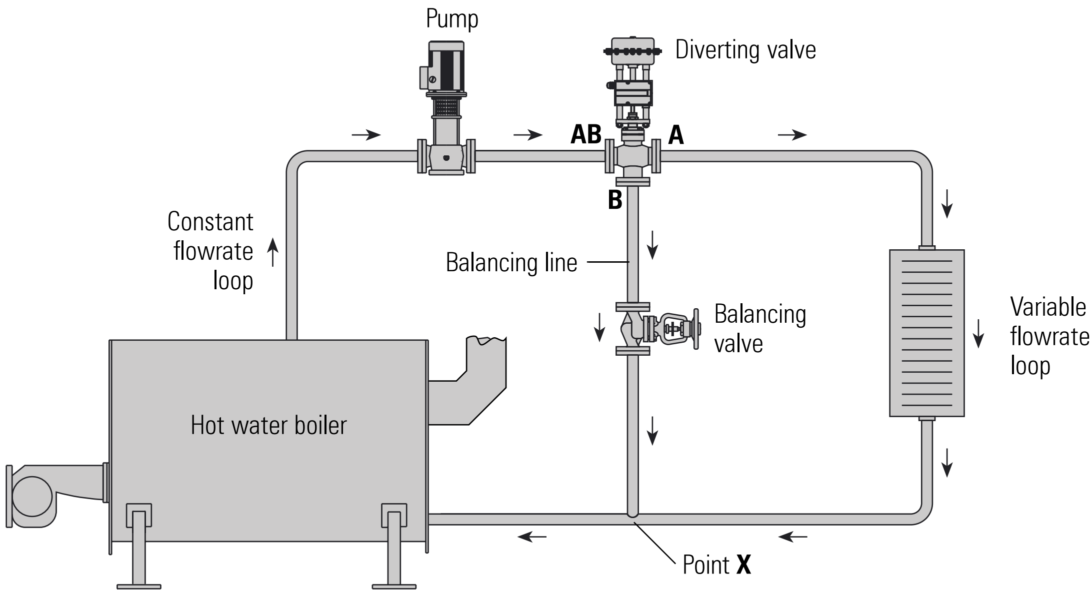
The amount of heat emitted from the radiators depends on the temperature of the water flowing through the load circuit, which in turn, depends upon how much water flows into the mixing valve from the boiler, and how much is returned to the mixing valve via the balancing line.

It is necessary to fit a balance valve in the balance line. The balance valve is set to maintain the same resistance to flow in the variable flowrate part of the piping network, as illustrated in Figures 6.3.6 and 6.3.7. This helps to maintain smooth regulation by the valve as it changes position.

In practice, the mixing valve is sometimes designed not to shut port **A** completely; this ensures that a minimum flowrate will pass through the boiler at all times under the influence of the pump. Alternatively, the boiler may employ a primary circuit, which is also pumped to allow a constant flow of water through the boiler, preventing the boiler from overheating.

The simple system shown in Figure 6.3.7 shows a diverting valve maintaining a constant flowrate of water through the constant flowrate loop. In this system, the load circuit receives a varying flowrate of water depending on the valve position.

The temperature of water in the load circuit will be constant, as it receives water from the boiler circuit whatever the valve position. The amount of heat available to the radiators depends on the amount of water flowing through the load circuit, which in turn, depends on the degree of opening of the diverting valve.



Resistance from Point **B** to Point **X** = Resistance from Point **A** to Point **X**

Fig. 6.3.7 Diverting valve (constant temperature in load circuit with variable flow)

The effect of not fitting and setting a balance valve can be seen in Figure 6.3.8. This shows the pump curve and system curve changing with valve position. The two system curves illustrate the difference in pump pressure required between the load circuit P_1 and the bypass circuit P_2, as a result of the lower resistance offered by the balancing circuit, if no balance valve is fitted. If the circuit is not correctly balanced then short-circuiting and starvation of any other sub-circuits (not shown) can result, and the load circuit may be deprived of water.

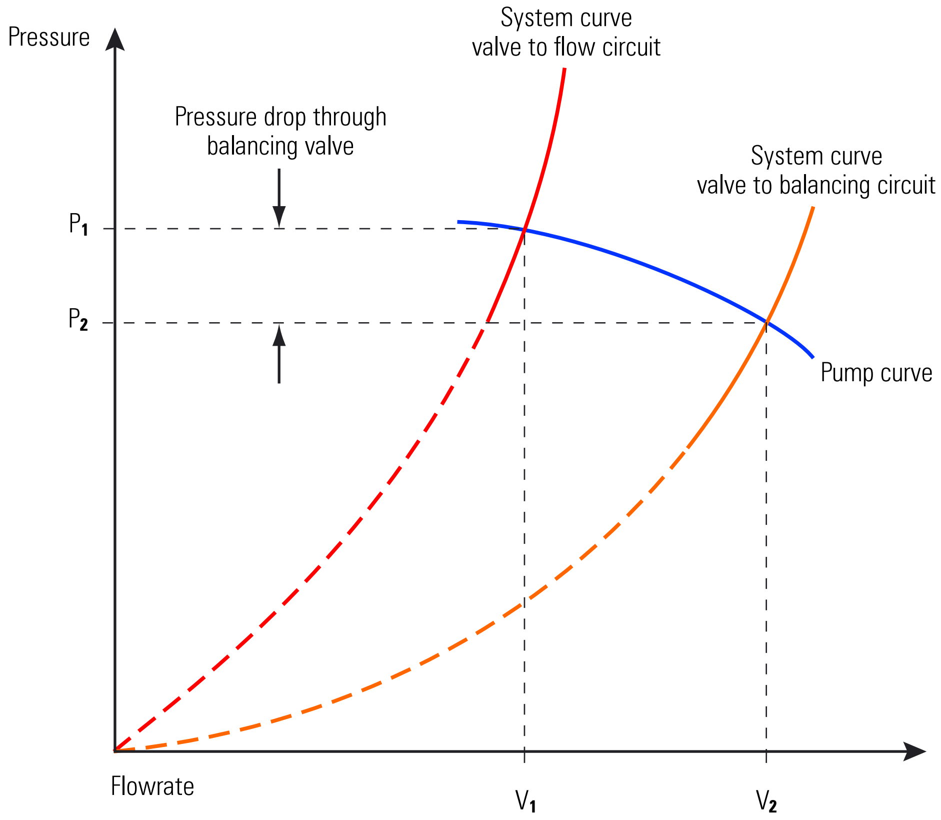


Fig. 6.3.8 Effect of not fitting a balance valve

Two-port Valves

When a two-port valve is used on a water system, as the valve closes, flow will decrease and the pressure upstream of the valve will increase. Changes in pump head will occur as the control valve throttles towards a closed position. The effects are illustrated in Figure 6.3.9.

A fall in flowrate not only increases the pump pressure but may also increase the power consumed by the pump. The change in pump pressure may be used as a signal to operate two or more pumps of varying duties, or to provide a signal to variable speed pump drive(s). This enables pumping rates to be matched to demand, saving pumping power costs.

Two port control valves are used to control water flow to a process, for example, for steam boiler level control, or to maintain the water level in a feedtank.

They may also be used on heat exchange processes, however, when the two-port valve is closed, the flow of water in the section of pipe preceding the control valve is stopped, creating a 'dead-leg'. The water in the dead-leg may lose temperature to the environment. When the control valve is opened again, the cooler water will enter the heat exchange coils, and disturb the process temperature. To avoid this situation, the control system may include an arrangement to maintain a minimum flow via a small bore pipe and adjustable globe valve, which bypass the control valve and load circuit.

Two-port valves are used successfully on large heating circuits, where a multitude of valves are incorporated into the overall system. On large systems it is highly unlikely that all the two-port valves are closed at the same time, resulting in an inherent 'self-balancing' characteristic. These types of systems also tend to use variable speed pumps that alter their flow characteristics relative to the system load requirements; this assists the self-balancing operation.

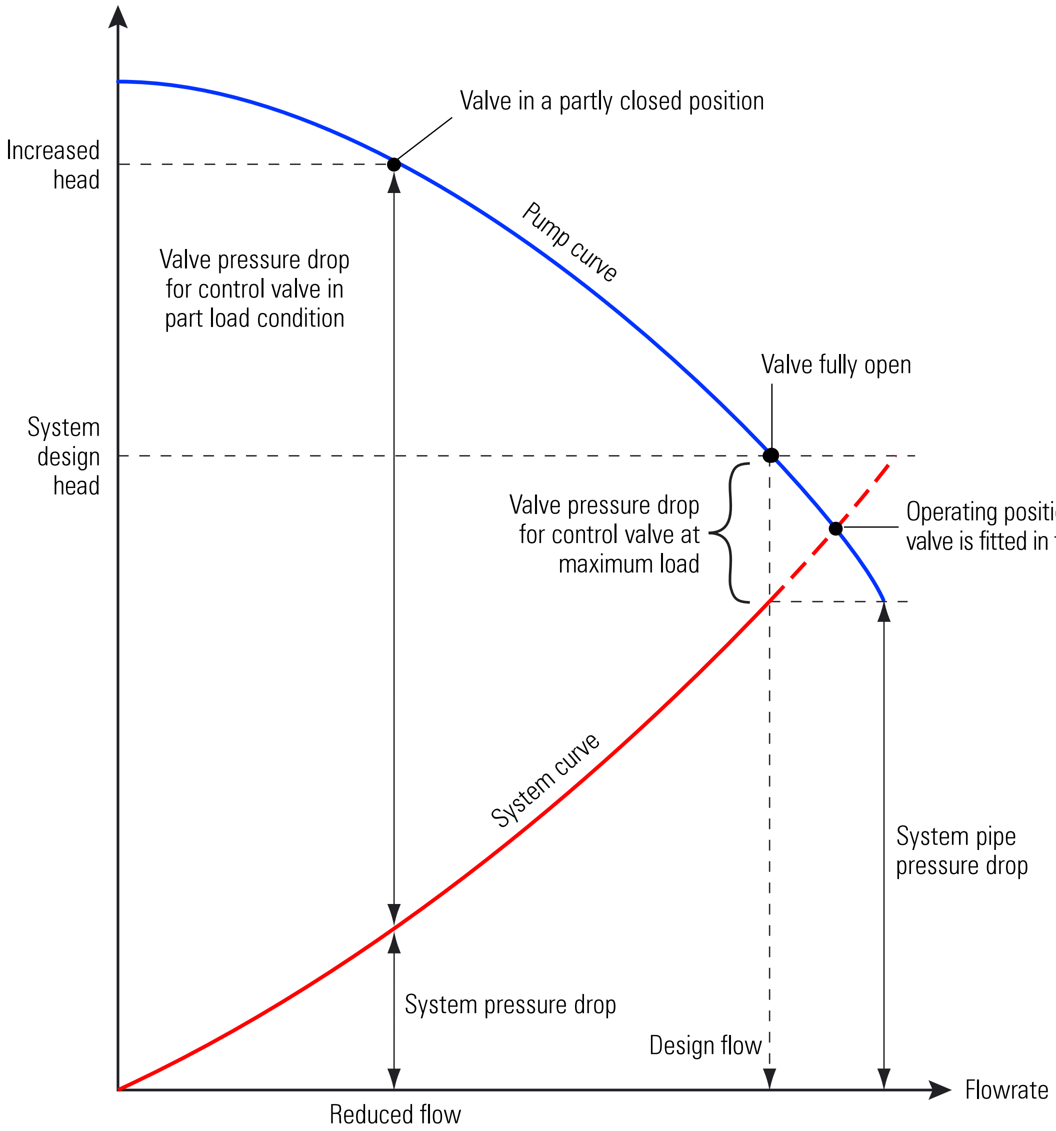


Fig. 6.3.9 Effect of two-port valve on pump head and pressure

When selecting a two-port control valve for an application:

- If a hugely undersized two-port control valve were installed in a system, the pump would use a large amount of energy simply to pass sufficient water through the valve.

  Assuming sufficient water could be forced through the valve, control would be accurate because even small increments of valve movement would result in changes in flowrate. This means that the entire travel of the valve might be utilised to achieve control.

- If a hugely oversized two-port control valve were installed in the same system, the energy required from the pump would be reduced, with little pressure drop across the valve in the fully open position.

However, the initial valve travel from fully open towards the closed position would have little effect on the flowrate to the process. When the point was reached where control was achieved, the large valve orifice would mean that very small increments of valve travel would have a large effect on flowrate. This could result in erratic control with poor stability and accuracy.

A compromise is required, which balances the good control achieved with a small valve against the reduced energy loss from a large valve. The choice of valve will influence the size of pump, and the capital and running costs. It is good practice to consider these parameters, as they will have a bearing on the overall lifetime cost of the system.

These balances can be realised by calculating the 'valve authority' relative to the system in which it is installed.

Valve authority

Valve authority may be determined using Equation 6.3.4.

$$N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2} \quad \textbf{Equation 6.3.4}$$

Where:

N = Valve authority
ΔP_1 = Pressure drop across a fully open control valve
ΔP_2 = Pressure drop across the remainder of the circuit
$\Delta P_1 + \Delta P_2$ = Pressure drop across the whole circuit

The value of N should be near to 0.5 (but not greater than), and certainly not lower than 0.2.

This will ensure that each increment of valve movement will have an effect on the flowrate without excessively increasing the cost of pumping power.

Example 6.3.3
A circuit has a total pressure drop ($\Delta P_1 + \Delta P_2$) of 125 kPa, which includes the control valve.
a) If the control valve must have a valve authority (N) of 0.4, what pressure drop is used to size the valve?
b) If the circuit/system flowrate ($\dot{V}$) is 3.61 l/s, what is the required valve K_v?

Part a) Determine the ΔP

$$N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2} \quad \textbf{Equation 6.3.4}$$

$$N = 0.4$$

$$\Delta P_1 + \Delta P_2 = 125 \text{ kPa}$$

$$N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2}$$

$$\Delta P_1 = N\,(\Delta P_1 + \Delta P_2)$$

$$\Delta P_1 = 0.4 \times 125 \text{ kPa}$$

$$\mathbf{\Delta P_1 = 50\,kPa}$$

Consequently, a valve ΔP of 50 kPa is used to size the valve, leaving 75 kPa (125 kPa - 50 kPa) for the remainder of the circuit.

Part b) Determine the required K_v

$$\dot{V} = K_v \sqrt{\Delta P} \quad \textbf{Equation 6.3.2}$$

Where:
$\dot{V}$ = 3.61 l/s (13m³/h)
ΔP = 50 kPa (0.5 bar)

$$13 = K_v \sqrt{0.5}$$

$$K_v = \frac{13}{\sqrt{0.5}}$$

$$\mathbf{K_v = 18.38}$$

Alternatively, the water K_v chart (Figure 6.3.2) may be used.

Three-port control valves and valve authority

Three-port control valves are used in either mixing or diverting applications, as explained previously in this Module. When selecting a valve for a diverting application:

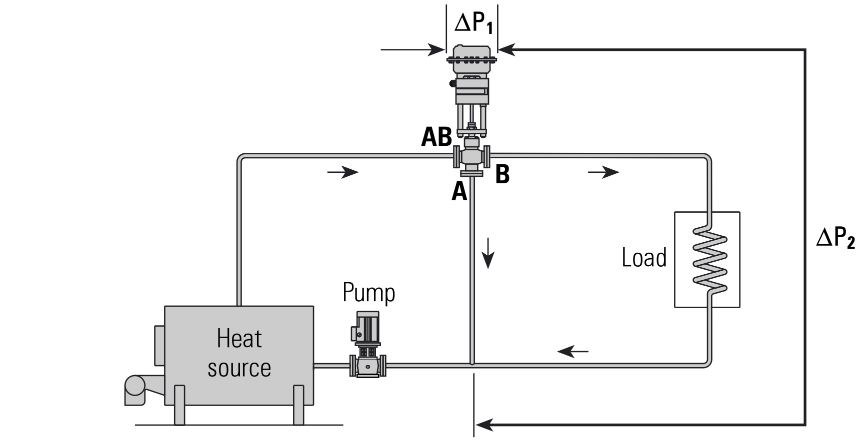
- A hugely undersized three-port control valve will incur high pumping costs, and small increments of movement will have an effect on the quantity of liquid directed through each of the discharge ports.
- A hugely oversized valve will reduce the pumping costs, but valve movement at the beginning, and end, of the valve travel will have minimal effect on the distribution of the liquid. This could result in inaccurate control with large sudden changes in load. An unnecessarily oversized valve will also be more expensive than one adequately sized.

The same logic can be applied to mixing applications.

Again, the valve authority will provide a compromise between these two extremes.

With three-port valves, valve authority is always calculated using P_2 in relation to the circuit with the variable flowrate. Figure 6.3.10 shows this schematically.

Three-port diverting valve



Three-port mixing valve

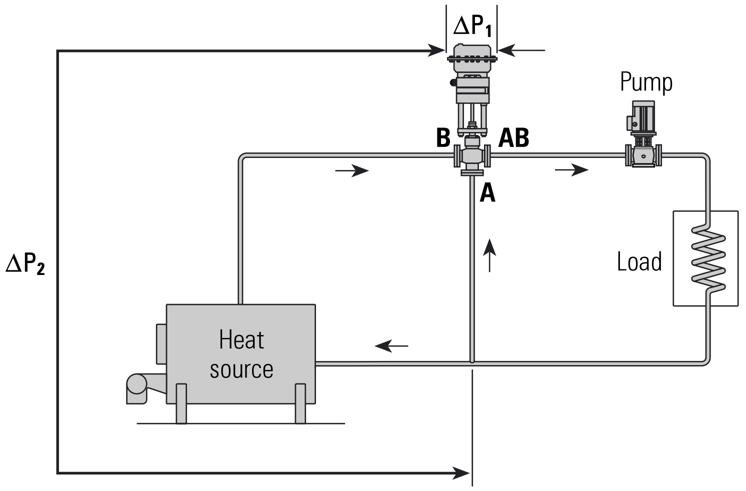


Fig. 6.3.10 Valve authority diagrams showing three-port valves

Note: Because mixing and diverting applications use three-port valves in a 'balanced' circuit, the pressure drop expected over a three-port valve is usually significantly less than with a two-port valve.

As a rough guide:

- A three-port valve will be 'line sized' when based on water travelling at recommended velocities (Typically ranging from 1 m/s at DN25 to 2 m/s at DN150).
- 10 kPa may be regarded as typical pressure drop across a three-port control valve.
- Aim for valve authority (N) to be between 0.2 and 0.5, the closer to 0.5 the better.

Cavitation and flashing

Other symptoms sometimes associated with water flowing through two-port valves are due to 'cavitation' and 'flashing'.

Cavitation in liquids

Cavitation can occur in valves controlling the flow of liquid if the pressure drop and hence the velocity of the flow is sufficient to cause the local pressure after the valve seat to drop below the vapour pressure of the liquid. This causes vapour bubbles to form. Pressure may then recover further downstream causing vapour bubbles to rapidly collapse. As the bubbles collapse very high local pressures are generated which, if adjacent to metal surfaces can cause damage to the valve trim, the valve body or downstream pipework. This damage typically has a very rough, porous or sponge-like appearance which is easily recognised. Other effects which may be noticed include noise, vibration and accelerated corrosion due to the repeated removal of protective oxide layers.

Cavitation will tend to occur in control valves:

- On high pressure drop applications, due to the high velocity in the valve seat area causing a local reduction in pressure.
- Where the downstream pressure is not much higher than the vapour pressure of the liquid. This means that cavitation is more likely with hot liquids and/or low downstream pressure.

Cavitation damage is likely to be more severe with larger valves sizes due to the increased power in the flow.

Flashing in liquids

Flashing is a similar symptom to cavitation, but occurs when the valve outlet pressure is lower than the vapour pressure condition. Under these conditions, the pressure does not recover in the valve body, and the vapour will continue to flow into the connecting pipe. The vapour pressure will eventually recover in the pipe and the collapsing vapour will cause noise similar to that experienced with cavitation. Flashing will reduce the capacity of the valve due to the throttling effect of the vapour having a larger volume than the water. Figure 6.3.11 illustrates typical pressure profiles through valves due to the phenomenon of cavitation and flashing.

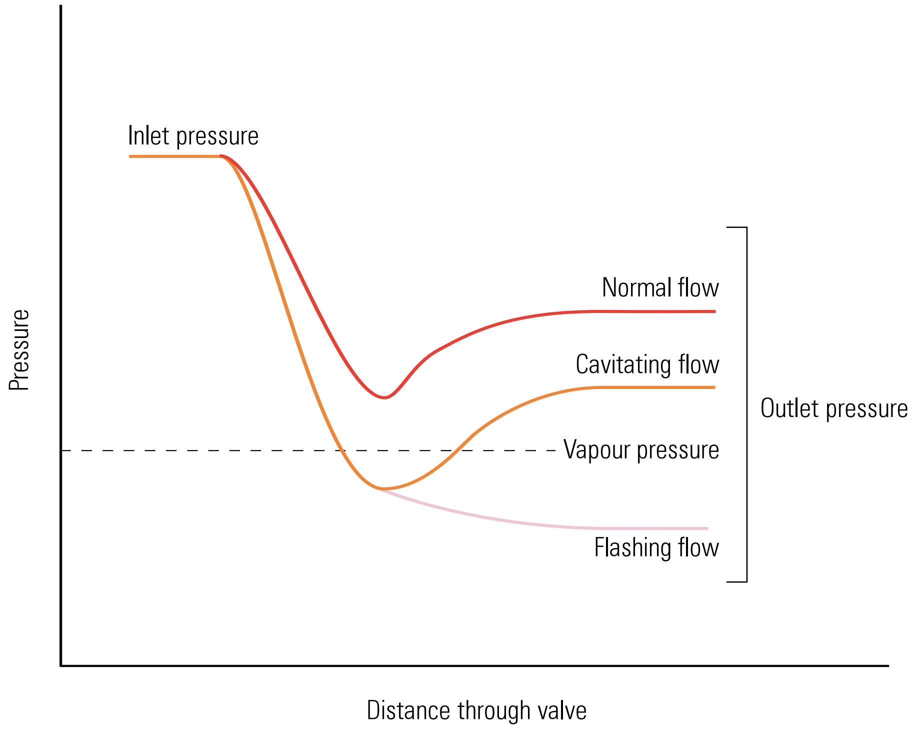


Fig. 6.3.11 Cavitation and flashing through a water control valve

Avoiding cavitation

It is not always possible to ensure that the pressure drop across a valve and the temperature of the water is such that cavitation will not occur. Under these circumstances, one possible solution is to install a valve with a valve plug and seat especially designed to overcome the problem. Such a set of internals would be classified as an 'anti-cavitation' trim.

The anti-cavitation trim consists of the standard equal percentage valve plug operating inside a valve seat fitted with a perforated cage. Normal flow direction is used. The pressure drop is split between the characterised plug and the cage which limits the pressure drop in each stage and hence the lowest pressures occur. The multiple flow paths in the perforated cage also increase turbulence and reduce the pressure recovery in the valve. These effects both act to prevent cavitation occuring in case of minor cavitation, or to reduce the intensity of cavitation in slightly more severe conditions. A typical characterised plug and cage are shown in Figure 6.3.12.

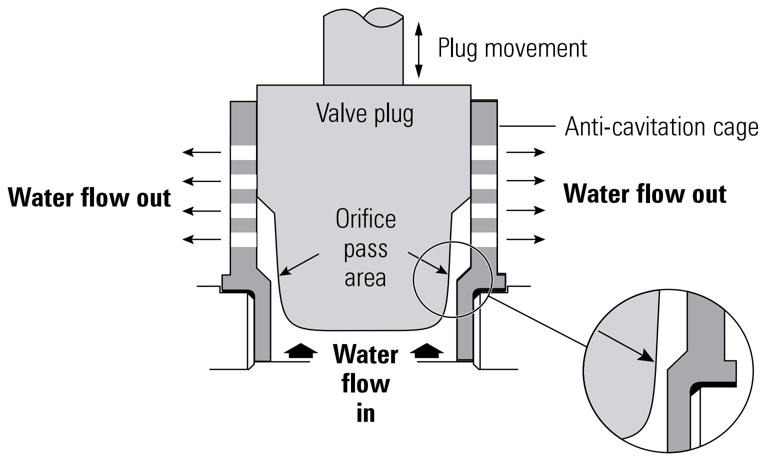


Fig. 6.3.12 A typical two-port valve anti-cavitation trim

The pressure drop is split between the orifice pass area and the cage. In many applications the pressure does not drop below the vapour pressure of the liquid and cavitation is avoided. Figure 6.3.12 shows how the situation is improved.

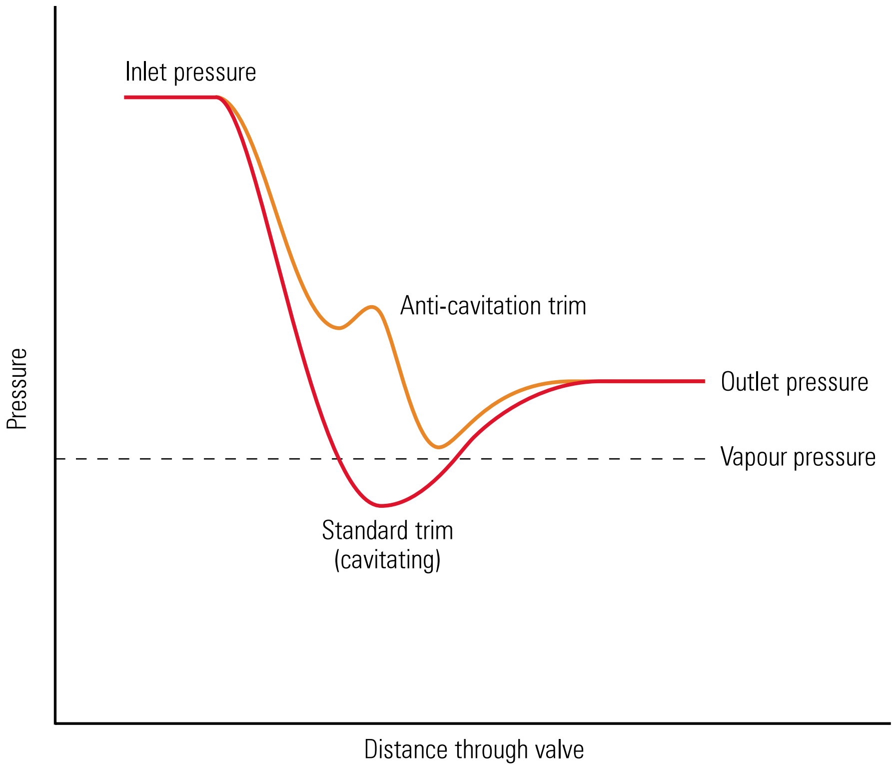
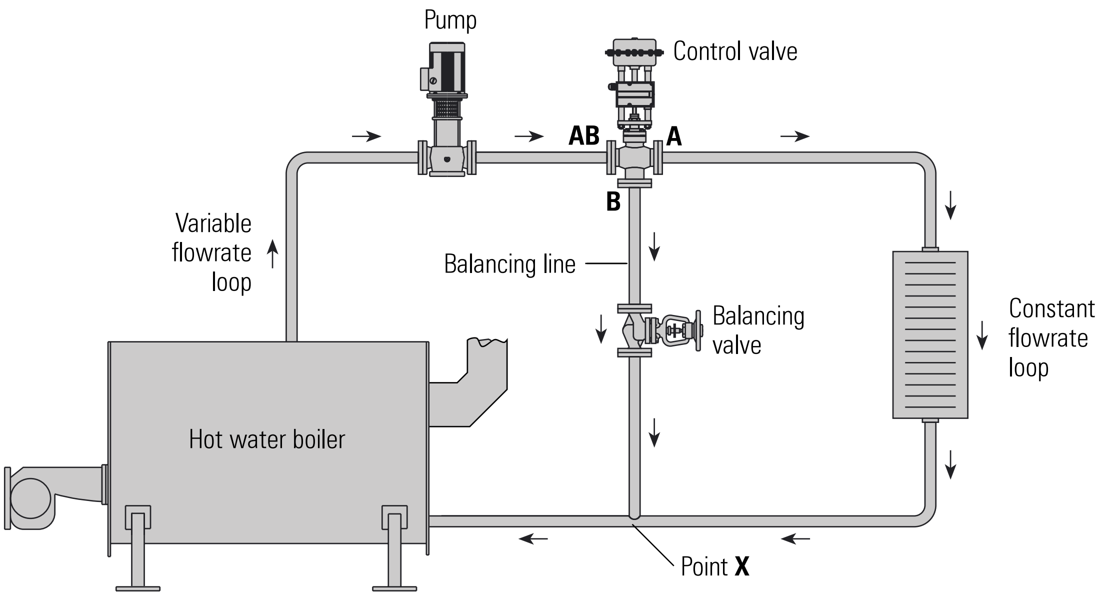


Fig. 6.3.13 Cavitation is alleviated by anti-cavitation valve trim

Questions

1. In the arrangement shown below, what will be the effect of omitting the balance valve?



a| The pump curve will change as the control valve diverts more of the flow through the balancing pipe ☐

b| Short circuiting and starvation of water to the process ☐

c| The pump must be repositioned to the process outlet ☐

d| None ☐

2. What is the optimum range of valve authority?

a| 0 – 0.2 ☐

b| 0.2 – 1.0 ☐

c| 0.5 – 1.0 ☐

d| 0.2 – 0.5 ☐

3. Calculate the valve authority if ΔP_1 = 15 kPa and ΔP_2 = 45 kPa

a| 0.75 ☐

b| 0.25 ☐

c| 0.33 ☐

d| 3.0 ☐

4. Water flowing through a fully open valve at a rate of 5 m^3/h creates a differential pressure of 0.25 bar across the valve. What is the valve K_{vs}?

a| 20 ☐

b| 1.25 ☐

c| 10 ☐

d| 80 ☐

5. It is noticed that the pressure loss along a certain sized pipe is 1.0 bar when the flowrate of water is 1 L/s. Using Equation 6.3.3, determine the flowrate of water along the same pipe if the pressure loss falls to 0.75 bar.

a| 1.155 L/s ☐

b| 0.500 L/s ☐

c| 1.333 L/s ☐

d| 0.866 L/s ☐

6. What are the two basic configurations for which a three-port valve is used?

a| Hot and cold ☐

b| Flow and return ☐

c| Series and parallel ☐

d| Mixing and diverting ☐

Answers

1: a, 2: d, 3: b, 4: c, 5: d, 6: d

Module 6.4

Control Valve Sizing for Steam Systems

Control Valve Sizing for Steam Systems

Before discussing the sizing of control valves for steam systems, it is useful to review the characteristics of steam in a heat transfer application.

- Steam is supplied at a specific pressure to the upstream side of the control valve through which it passes to a heat exchanger, also operating at a specific pressure.
- Steam passes through the control valve and into the steam space of the equipment where it comes into contact with the heat transfer surfaces.
- Steam condenses on the heat transfer surfaces, creating condensate.
- The volume of condensate is very much less than steam. This means that when steam condenses, the pressure in the steam space is reduced.
- The reduced pressure in the steam space means that a pressure difference exists across the control valve, and steam will flow from the high-pressure zone (upstream of the control valve) to the lower pressure zone (the steam space in the equipment) in some proportion to the pressure difference and, ideally, balancing the rate at which steam is condensing.
- The rate of steam flow into the equipment is governed by this pressure difference and the valve orifice size. Should, at any time, the flowrate of steam through the valve be less than the condensing rate (perhaps the valve is too small), the steam pressure and the heat transfer rate in the heat exchanger will fall below that which is required; the heat exchanger will not be able to satisfy the heat load.
- If a modulating control system is used, as the temperature of the process approaches the controller set point, the controller will close the valve by a related amount, thereby reducing the steam flowrate to maintain the lower pressure required to sustain a lower heat load. (The action of opening and closing the valve is often referred to as increasing or decreasing the 'valve lift'; this is explained in more detail in Module 6.5, 'Control Valve Characteristics').
- Closing the valve reduces the mass flow. The steam pressure falls in the steam space and so too the steam temperature. This means that a smaller difference in temperature exists between the steam and the process, so the rate of heat transfer is reduced, in accordance with Equation 2.5.3.

$$\dot{Q} = U\,A\,\Delta T_M \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W (J/s))
U = Overall heat transfer coefficient (W/m²°C)
A = Heat transfer area (m²)
ΔT_M = Mean temperature difference between the steam and secondary fluid (°C)

The overall heat transfer coefficient (U) does not change very much during the process, and the area (A) is fixed, so if the mean temperature difference (ΔT_M) is reduced, then the heat transfer from the steam to the secondary fluid is also reduced.

Saturated steam flow through a control valve

A heat exchanger manufacturer will design equipment to give a certain heat output. To achieve this heat output, a certain saturated steam temperature will be required at the heat transfer surface (such as the inside of a heating coil in a shell and tube heat exchanger). With saturated steam, temperature and pressure are strictly related; therefore controlling the steam pressure easily regulates the temperature.

Consider an application where steam at 10 bar g is supplied to a control valve, and a given mass flow of steam passes through the valve to a heat exchanger. The valve is held fully open (see Figure 6.4.1).

- If a DN50 valve is fitted and the valve is fully open, the pressure drop is relatively small across the valve, and the steam supplied to the heat exchanger is at a fairly high pressure (and temperature). Because of this, the heating coil required to achieve the design load is relatively small.
- Consider now, a fully open DN40 valve in the steam supply line passing the same flowrate as the DN50 valve. As the valve orifice is smaller the pressure drop across the valve must be greater, leading to a lower pressure (and temperature) in the heat exchanger. Because of this, the heat transfer area required to achieve the same heat load must be increased. In other words, a larger heating coil or heat exchanger will be required.
- Further reduction of the valve size will require more pressure drop across the control valve for the same mass flow, and the need for an increased heat transfer surface area to maintain the same heat output.

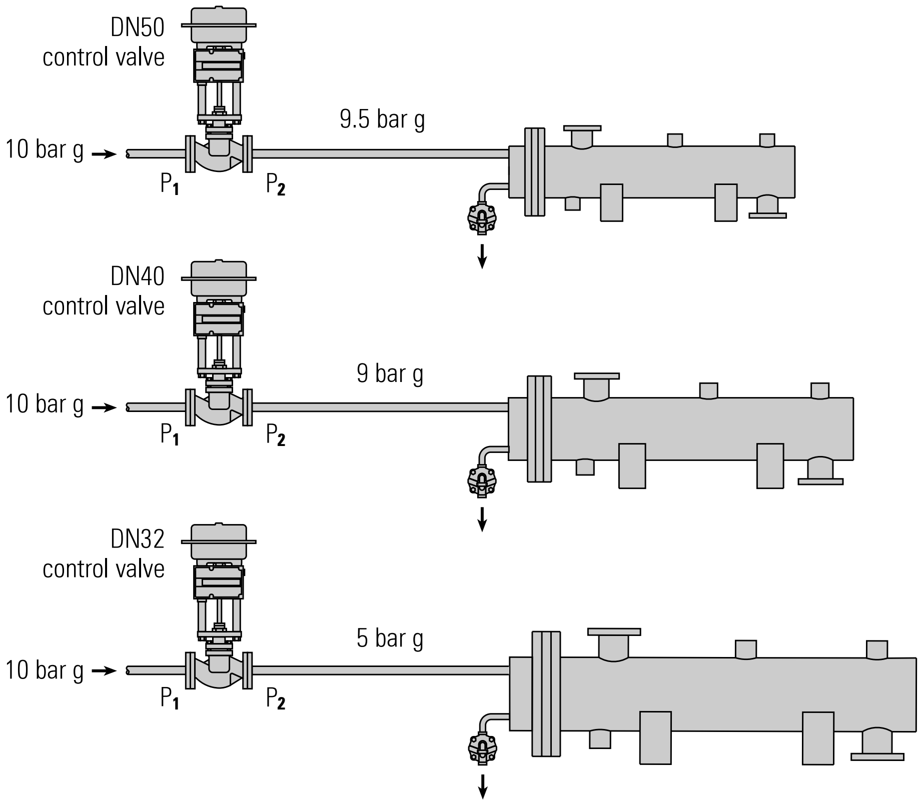


Fig. 6.4.1 Flow through a fully open control valve

Whatever the size of the control valve, if the process demand is reduced, the valve must modulate from the fully open position towards closed. However, the first part of the travel has only a small regulating effect, with any percentage change in valve lift producing a lesser percentage change in flowrate. Typically, a 10% change in lift might produce only a 5% change in flowrate. With further travel, as the valve plug approaches the seat, this effect reverses such that perhaps a 5% change in lift might produce a 10% change in flowrate, and better regulation is achieved.

The initial part of the control valve travel, during which this lowered control effect is seen, is greater with the selection of the larger control valves and the accompanying small pressure drop at full load. When the control valve chosen is small enough to require a 'critical pressure drop' at full load the effect disappears. Critical pressure is explained in the Section below.

Further, if a larger control valve is selected, the greater size of the valve orifice means that a given change in flowrate is achieved with a smaller percentage change in lift than is needed with a smaller control valve.

This can often make the control unstable, increasing the possibility of 'hunting', especially on reduced loads.

Critical pressure

The mass flow of steam passing through the valve will increase in line with differential pressure until a condition known as 'critical pressure' is reached. The principle can be explained by looking at how nozzles work and how they compare to control valves.

Consider an almost perfect orifice, such as a convergent-divergent nozzle shown in Figure 6.4.2. Its shape, if designed correctly to match the upstream and downstream pressure conditions and the condition of the supplied steam, will allow it to operate at high efficiency.

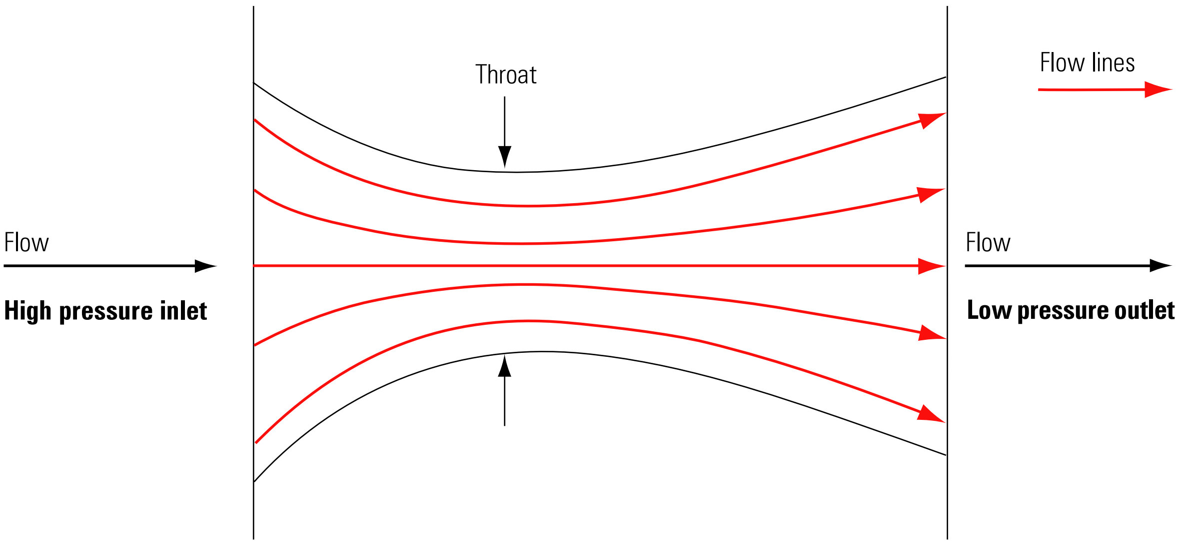


Fig. 6.4.2 A convergent-divergent nozzle

Such a nozzle can be thought of as a type of heat engine, changing heat energy into mechanical (kinetic) energy. It is designed to discharge the required weight of steam with a given pressure drop, and with minimum turbulence and friction losses.

In the convergent section, the steam velocity increases as the pressure falls, though the specific volume of the steam also increases with the lowered pressures. At first, the velocity increases more quickly than the specific volume, and the required flow area through this part of the nozzle becomes less. At a certain point, the specific volume begins to increase more rapidly than does the velocity and the flow area must become greater. At this point, the steam velocity will be sonic and the flow area is at a minimum. The steam pressure at this minimum flow area or 'throat' is described as the 'critical pressure', and the ratio of this pressure to the initial (absolute) pressure is found to be close to 0.58 when saturated steam is passing.

Critical pressure varies slightly according to the fluid properties, specifically in relation to the ratio of the specific heats c_p/c_v of the steam (or other gaseous fluid), which is termed the adiabatic index or isentropic exponent of the fluid, often depicted by the symbols 'n', 'k' or 'γ'. With superheated steam the ratio is about 0.55, and for air about 0.53.

Point of interest:

Critical pressure ratio can also be determined by Equation 6.4.1.

$$\text{Critical pressure ratio} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma + 1}} \qquad \textbf{Equation 6.4.1}$$

γ is given as follows:

Wet steam: $\gamma = 1.035 + 0.1(x)$
where 'x' is dryness fraction, $0.8 > x > 1$.

Dry saturated steam: $\gamma = 1.135$

Superheated steam: $\gamma = 1.3$

For dry saturated steam, using Equation 6.4.1:

$$\begin{aligned}
\text{Critical pressure ratio} &= \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135 - 1}} \\
&= \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}} \\
&= (0.937)^{8.41} \\
&= 0.58
\end{aligned}$$

Clearly, the mass flow through the throat of a given size is at a maximum at this 'critical pressure drop'. To achieve a greater flow, either:

a. The velocity would have to be greater, which could only be reached with a greater pressure drop – but this would also increase the specific volume by an even greater amount, or:

b. The specific volume would have to be less, which could only be the case with a lesser pressure drop – but this would reduce the velocity by an even greater amount.

Thus, once the critical pressure drop is reached at the throat of the nozzle, or at the 'vena contracta' when an orifice is used, further lowering of the downstream pressure cannot increase the mass flow through the device.

If the pressure drop across the whole nozzle is greater than the critical pressure drop, critical pressure will always occur at the throat. The steam will expand after passing the throat such that, if the outlet area has been correctly sized, the required downstream pressure is achieved at the nozzle outlet, and little turbulence is produced as the steam exits the nozzle at high velocity.

Should the nozzle outlet be too big or too small, turbulence will occur at the nozzle outlet, reducing capacity and increasing noise:

- If the nozzle outlet is too small, the steam has not expanded enough, and has to continue expanding outside the nozzle until it reaches the required downstream pressure in the low pressure region.
- If the nozzle outlet is too large, the steam will expand too far in the nozzle and the steam pressure in the nozzle outlet will be lower than the required pressure, causing the steam to recompress outside the outlet in the low pressure region.

The shape of the nozzle (Figure 6.4.3) is gently contoured such that the vena contracta occurs at the nozzle throat. (This is in contrast to a sharp-edged orifice, where a vena contracta occurs downstream of the orifice. The vena contracta effect is discussed in more detail in Module 4.2 'Principles of Flowmetering').

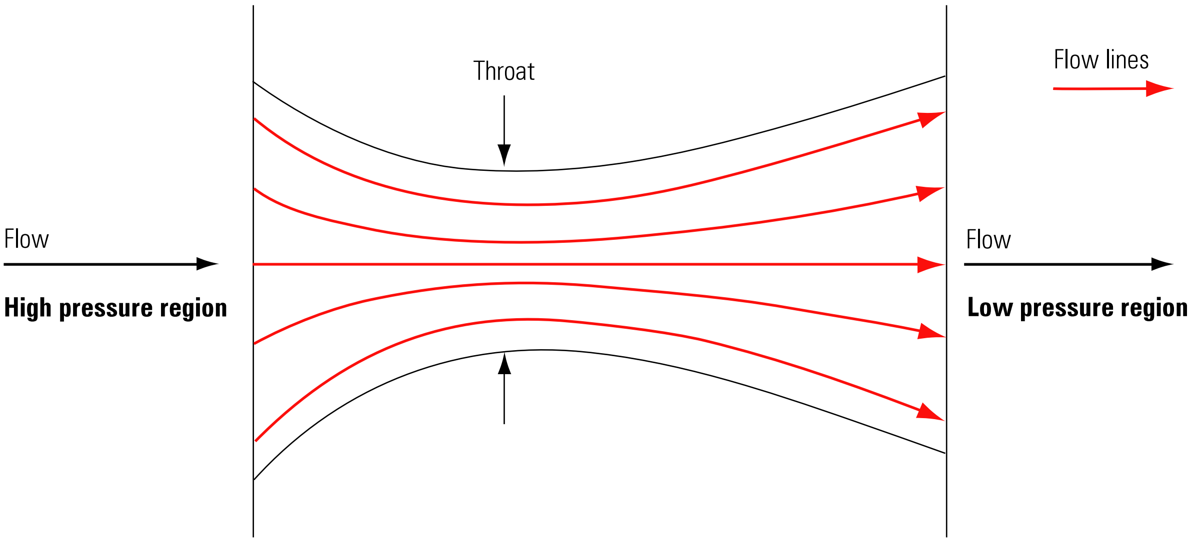


Fig. 6.4.3 The convergent-divergent nozzle

Control valves can be compared to convergent-divergent nozzles, in that each has a high-pressure region (the valve inlet), a convergent area (the inlet between the valve plug and its seat), a throat (the narrowest gap between the valve plug and its seat), a divergent area (the outlet from the valve plug and its seat, and a low-pressure region (the downstream valve body). See Figure 6.4.4.

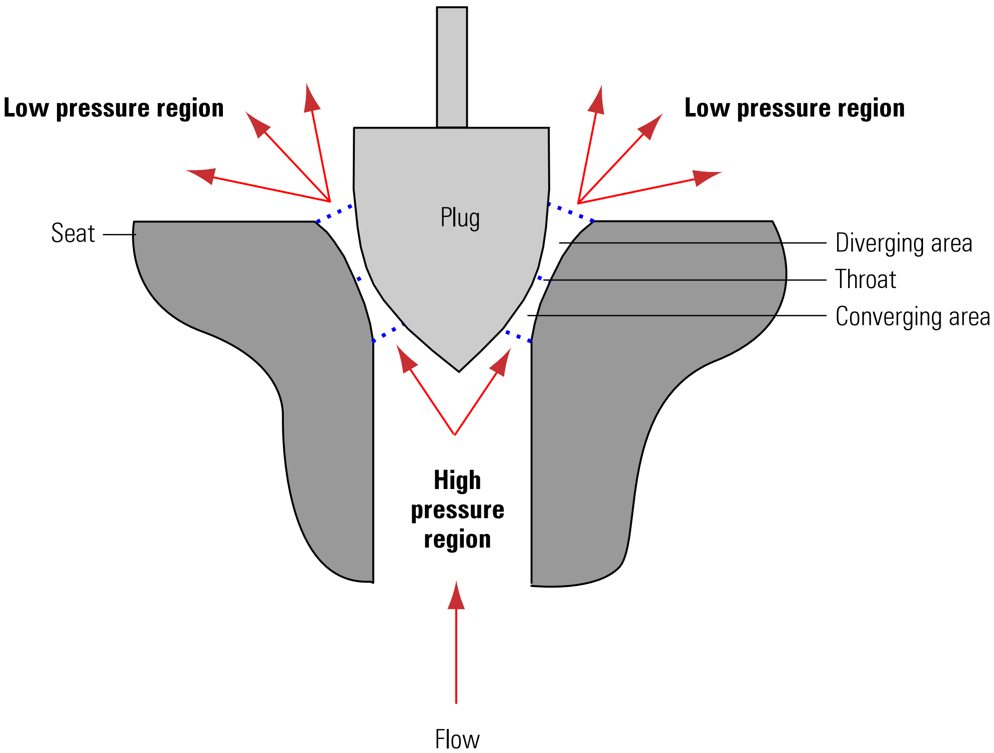


Fig. 6.4.4 The convergent-divergent principle in a control valve

Nozzles and control valves have different purposes. The nozzle is primarily designed to increase steam velocity in order to produce work (perhaps to turn a turbine blade), so the velocity of steam leaving the nozzle is required to remain high.

In contrast, the control valve is a flow restricting or 'throttling' device designed to produce a significant pressure drop in the steam. The velocity of steam passing out of a control valve throat will behave in a similar fashion to that of the steam passing out of the throat of a convergent-divergent nozzle; in that it will increase as the steam expands in the diverging area between the plug and seat immediately after the throat. If the pressure drop across the valve is greater than critical pressure drop, the steam velocity will increase to supersonic in this area, as the pressure here is less than that at the throat.

Past this point, the steam passes into the relatively large chamber encased by the valve body (the low pressure region), which is at a higher pressure due to the backpressure imposed by the connecting pipework, causing the velocity and kinetic energy to fall rapidly. In accordance with the steady flow energy equation (SFEE), this increases the steam enthalpy to almost that at the valve entrance port. A slight difference is due to energy lost to friction in passing through the valve.

From this point, the valve body converges to port the steam flow to the valve outlet, and the pressure (and density) approach the pressure (and density) in the downstream pipe. As this pressure stabilises, so does the velocity, relative to the cross sectional area of the valve outlet port.

The relative change in volume through the valve is represented by the dotted lines in the schematic diagram shown in Figure 6.4.5.

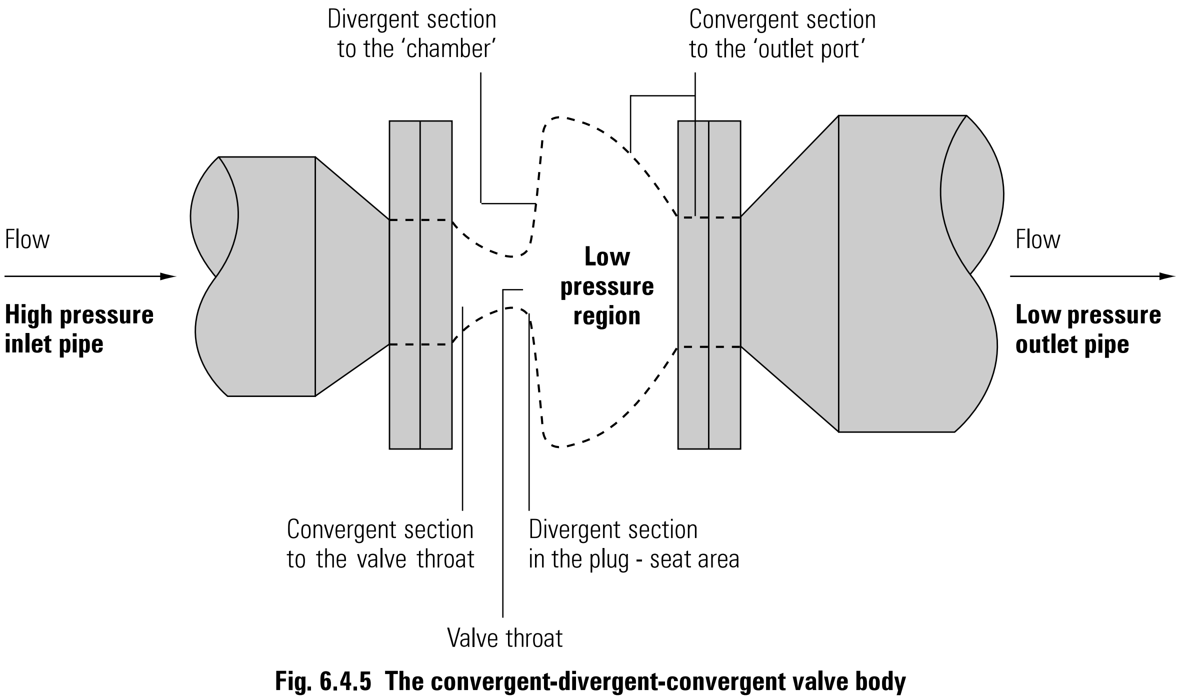


Fig. 6.4.5 The convergent-divergent-convergent valve body

When the pressure drop across a valve is greater than critical, noise can be generated by the large instantaneous exchange from kinetic energy to heat energy in the low pressure region, sometimes exacerbated by the presence of supersonic steam.

Valve outlet velocity, noise, erosion, drying and superheating effect

Noise can be an important consideration when sizing control valves. Special noise-reducing valve trims are available but, sometimes, a less expensive solution is to fit a larger valve body than required. Complicated equations are required to calculate noise emitted from control valves and these are difficult to use manually. It is usually considered that the control valve will produce unacceptable noise if the velocity of dry saturated steam in the control valve outlet is greater than 0.3 Mach. The speed of sound in steam will depend upon the steam temperature and the quality of the steam, but can be calculated from Equation 6.4.2 if the conditions are known (Mach 1 = speed of sound).

$$C = 31.6\sqrt{\gamma\, R\, T}$$ **Equation 6.4.2**

Where:
C = Speed of sound in steam (m/s)
31.6 = Constant of proportionality
γ = Steam isentropic exponent (1.135 : saturated, 1.3 : superheated)
R = 0.4615 the gas constant for steam (kJ/kg)
T = Absolute steam temperature (K)

A less accurate but useful method to estimate whether noise will be a problem is by calculating the velocity in the valve outlet port. In simplistic terms and for dry saturated steam, if this is greater than 150 m/s, there is a chance that the valve body is too small (even though the valve trim size suits the required capacity). Higher velocities also cause erosion in the downstream valve body, especially if the steam is wet at this point. It is recommended that the maximum exit velocity for wet steam is 40 m/s in the outlet port.

Another result of dropping steam pressure across a control valve is to dry or superheat the steam, depending upon its condition as it enters the valve. Large degrees of superheat are usually unwanted in heating processes, and so it is useful to be able to determine if this will occur. Superheated steam (and dry gas) velocities, however, may be allowed to reach 0.5 Mach in the outlet port; whereas, at the other end of the scale, liquids might be restricted to a maximum outlet velocity of 10 m/s.

Example 6.4.1 The valve outlet velocity and drying/superheating effect

A control valve is supplied with dry saturated steam from a separator at 12 bar g and used to drop steam pressure to 4 bar g at full load. The full load flowrate is 1 300 kg/h requiring a K_{vr} of 8.3. A DN25 (1") valve is initially considered for selection, which has a K_{vs} of 10 and a valve outlet area of 0.000 9 m². What is the steam velocity in the valve outlet?

Determine the state of the steam in the valve outlet at 4 bar g.

The degree of drying and superheating can be calculated from the following procedure:

From steam tables, total heat (h_g) in the upsteam dry saturated steam at 12 bar g = 2 787 kJ/kg

As the supply steam is in a dry saturated state, the steam will certainly be superheated after it passes through the valve; therefore the superheated steam table should be used to quantify its properties.

Using the Spirax Sarco website steam tables, it is possible to calculate the condition of the downstream steam at 4 bar g by selecting 'Superheated steam' and entering a pressure of '4 bar g' and a total heat (h) of 2 787 kJ/kg.

By entering these values, the steam table returns the result of superheated steam at 4 bar g with 16.9 degrees of superheat (442 K). (Further details on how to determine the downstream state are given in Module 2.3 'Superheated steam'.

Specific volume of superheated steam, 4 bar g, 442 K is 0.391 8 m^3/kg.

$$\text{The volumetric flow} = 1\,300 \text{ kg/h} \times 0.391\,8 \text{ m}^3\text{/kg}$$

$$= 509.3 \text{ m}^3\text{/h}$$

$$= 0.1415 \text{ m}^3\text{/s}$$

$$\text{Valve outlet velocity} = \frac{\text{Volumetric flowrate}}{\text{Outlet area}}$$

$$= \frac{0.141\,5 \text{ m}^3\text{/s}}{0.000\,9 \text{ m}^2}$$

$$= 157 \text{ m/s}$$

It is necessary to see if this velocity is less than 0.5 Mach, the limit placed on valve outlet velocities for superheated steam.

The speed of sound (Mach 1) can be calculated from Equation 6.4.2.

$$C = 31.6\sqrt{\gamma\, R\, T} \qquad \textbf{Equation 6.4.2}$$

A value of 1.3 is chosen for the isentropic exponent 'γ' due to the steam in the valve outlet being superheated.

R is the gas constant for steam 0.461 5 kJ/kg

T is the absolute temperature of 442 K

Therefore the speed of sound in the valve outlet:

$$C = 31.6\sqrt{\gamma\, R\, T}$$

$$C = 31.6\sqrt{1.3 \times 0.461\,5 \times 442}$$

$$C = 31.6 \times 16.28$$

$$C = 515 \text{ m/s}$$

As the steam is superheated in the valve outlet, the criterion of 0.5 Mach is used to determine whether the valve will be noisy.

$$0.5 \times 515 = 257.5 \text{ m/s}$$

As the expected velocity is 157 m/s and below the limit of 257.5 m/s, and the outlet steam is superheated, the DN25 valve would be suitable for this application.

The same procedure can be used to determine the conditions of the downstream steam for other upstream conditions. For instance, if the upstream steam is known to be wet, the downstream condition might be wet, dry saturated or superheated, depending on the pressure drop. The allowable outlet velocity will depend on the downstream steam condition as previously outlined in this section, and observed in Example 6.4.2.

Erosion

Another problem is the possibility of erosion in the valve body caused by excessive velocity in the valve outlet. In Example 6.4.1, due to the drying and superheating effect of the pressure drop from 12 bar g to 4 bar g, the steam is in a dry gaseous state containing absolutely no moisture, and erosion should not be an issue.

Simplistically, if it can be guaranteed that the steam leaving a control valve is superheated, then 250 m/s is an appropriate limit to place on the outlet velocity.

Sometimes, when saturated steam is supplied to a control valve, it will be carrying a certain amount of water and the steam may be, for example, 97% or 98% dry. If it has just passed through a properly designed separator it will be close to 100% dry, as in Example 6.4.1.

With anything more than a small pressure drop and wet steam, the steam will probably be dried to saturation point or even slightly superheated.

If the supply steam is dry and/or the valve encounters quite a large pressure drop, (as in Example 6.4.1), the steam will be more superheated.

Equations for sizing control valves

Control valves are not as efficient as nozzles in changing heat into kinetic energy. The path taken by steam through the valve inlet, the throat and into the valve outlet is relatively tortuous.

In a control valve a great deal more energy is lost to friction than in a nozzle, and, because...

- The outlet area of the valve body is unlikely to match the downstream pressure condition.
- The relationship between the plug position and the seat is continually changing.

. . . turbulence is always likely to be present in the valve outlet.

It seems that control valves of differing types may appear to reach critical flow conditions at pressure drops other than those quoted above for nozzles. Restricted flow passages through the seat of a valve and on the downstream side of the throat may mean that maximum flowrates may only be reached with somewhat greater pressure drops. A ball valve or butterfly valve may be so shaped that some pressure recovery is achieved downstream of the throat, so that maximum flow conditions are reached with an overall pressure drop rather less than expected.

Complicated valve sizing equations can be used to take these and other criteria into consideration, and more than one standard exists incorporating such equations.

One such standard is IEC 60534. Unfortunately, the calculations are so complicated, they can only be used by computer software; manual calculation would be tedious and slow.

Nevertheless, when sizing a control valve for a critical process application, such software is indispensable. For example, IEC 60534 is designed to calculate other symptoms such as the noise levels generated by control valves, which are subjected to high pressure drops. Control valve manufacturers will usually have computer sizing and selection software complementing their own range of valves.

However, a simple steam valve sizing equation, such as that shown in Equation 3.21.2 for saturated steam, is perfectly adequate for the vast majority of steam applications with globe valves.

Also, if consideration is given to critical pressure occurring at 58% of the upstream absolute pressure, a globe valve is unlikely to be undersized.

For simplicity, the rest of this Module assumes critical pressure for saturated steam occurs at 58% of the upstream absolute pressure.

For example, if the pressure upstream of a control valve is 10 bar a, the maximum flowrate through the valve occurs when the downstream pressure is:

$$10 \text{ bar a} \times 58\% = 5.8 \text{ bar a}$$

Equally, critical pressure drop is 42% of the upstream pressure, that is, a pressure drop ratio of 0.42. As shown in the previous text, once this downstream pressure is reached, any further increase in pressure drop does not cause an increase in mass flowrate.

This effect can be observed in Figure 6.4.6 showing how, in the case of a globe valve, the flowrate increases with falling downstream pressure until critical pressure drop is achieved.

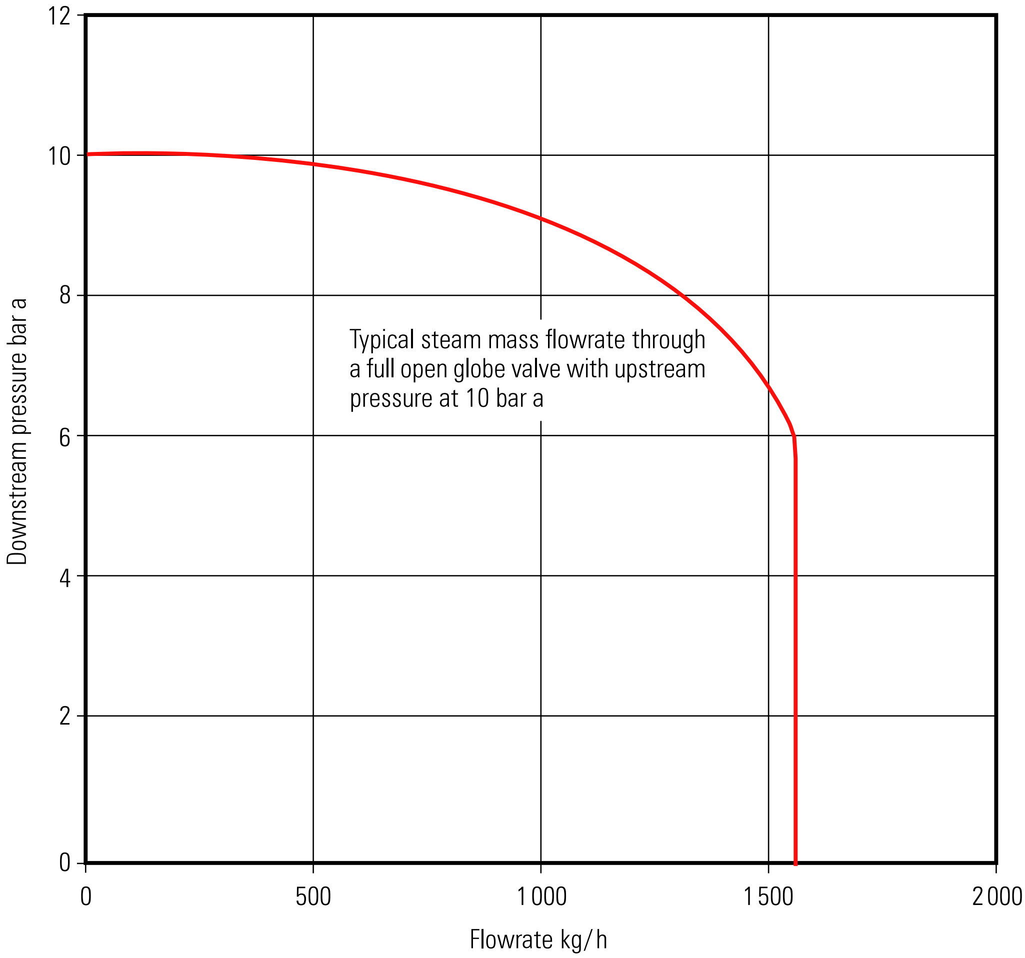


Fig. 6.4.6 The mass flowrate through a steam valve increases until critical pressure is reached

Sizing a control valve for a steam heat exchanger is a compromise between:

1. A smaller pressure drop that will minimise the size (and perhaps the cost) of the heat exchanger.

2. A larger pressure drop that allows the valve to apply effective and accurate control over the pressure and flowrate for most of its travel.

If the pressure drop is less than 10% at full load, three problems can occur:

- Depending upon the controller settings and secondary temperature, and system time lags, 'hunting' of the temperature around the set value may occur because the valve is effectively oversized; small changes in lift will cause large changes in flowrate, especially in the case of a valve with a linear characteristic.
- Running loads are often much less than the full load, and the valve may operate for very long periods with the valve plug close to its seat. This creates a risk of wiredrawing, (erosion caused by high velocity water droplets squeezing through the narrow orifice). Wiredrawing will result in a reduced valve service life.
- The system will not control well at low heat loads, effectively reducing the 'turndown' capability of the valve.

Simple sizing routine for globe valves in steam service

The flow and expansion of steam through a control valve is a complex process. There are a variety of very complex sizing formulae available, but a pragmatic approach, based on the 'best fit' of a mathematical curve to empirical results, is shown in Equation 3.21.2 for globe valves throttling saturated steam. The advantage of this relatively simple formula is that it can be used with the aid of a simple calculator. It assumes that critical pressure drop occurs at 58% of the upstream pressure.

$$\dot{m}_s = 12\,K_v\,P_1\sqrt{1 - 5.67\,(0.42 - \varkappa)^2} \qquad \textbf{Equation 3.21.2}$$

Where:

$\dot{m}$ = Mass flowrate (kg/h)

K_v = Valve flow coefficient (m³/h bar)

P_1 = Upstream pressure (bar a)

$\varkappa$ = Pressure drop ratio = $\dfrac{P_1 - P_2}{P_1}$

P_2 = Downstream pressure (bar a)

Note: If Equation 3.21.2 is used when P_2 is less than the critical pressure, then the term within the bracket (0.42 - $\varkappa$) becomes negative. This is then taken as zero and the function within the square root sign becomes unity, and the equation is simplified as shown in Equation 6.4.3.

$$\dot{m} = 12\,K_v\,P_1 \qquad \textbf{Equation 6.4.3}$$

Alternatively, valve-sizing or K_v charts can be used.

Terminology

Normally the full lift value of the valve will be stated using the term K_{vs}, thus:

K_{vr} = Actual value required for an application

K_{vs} = Full lift capacity stated for a particular valve

Manufacturers give the maximum lift K_{vs} values for their range of valves. Hence the K_v value is not only used for sizing valves but also as a means of comparing the capacity of alternative valve types and makes. Comparing two DN15 valves from different sources shows that valve 'A' has a K_{vs} of 10 and valve 'B' a K_{vs} of 8. Valve 'A' will give a higher flowrate for the same pressure drop.

Bringing together the information for steam valve sizing

Certain minimum information is required to determine the correct valve size:

- The pressure of the steam supply must be known.
- The steam pressure in the heat exchanger to meet the maximum heat load must be known.

The difference between the above criteria defines the differential pressure across the valve at its full load condition.

- The heat output of the equipment must be known, along with the enthalpy of evaporation (h_{fg}) at the working pressure in the heat exchanger. These factors are required to determine the steam mass flowrate.

Example 6.4.2

A control valve is required for the application shown in Figure 6.4.7.

The shell and tube heat exchanger manufacturer specifies that a steam pressure of 5 bar absolute is required in the tube bundle to satisfy a process demand of 500 kW.

Wet steam, at dryness 0.96 and 10 bar a, is available upstream of the control valve. Enthalpy of evaporation (h_{fg}) at 5 bar a is 2 108.23 kJ/kg.

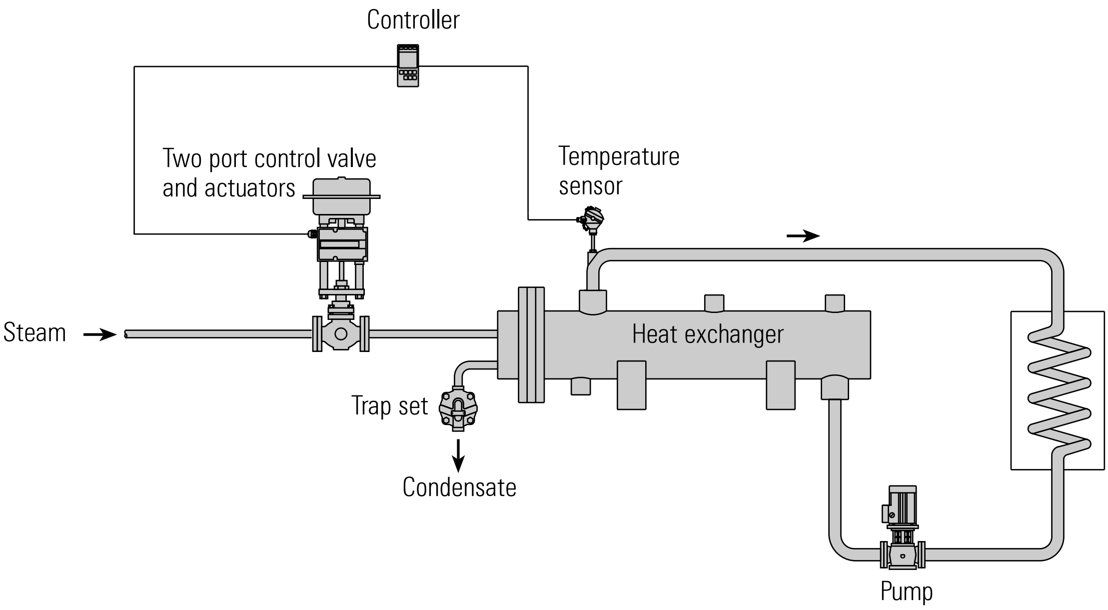


Fig. 6.4.7 Control valve on steam supply to a shell and tube heat exchanger

Determine the steam flowrate

First, it is necessary to determine the steam state for the downstream condition of 5 bar a. By entering wet steam at 10 bar a, and 0.96 dryness into the Spirax Sarco website wet steam table, it can be seen that the total heat hg held in the 10 bar wet steam is 2 697.15 kJ/kg.

The heat exchanger design pressure is 5 bar a, and the total heat in dry saturated steam at this pressure is 2 748.65 kJ/kg (from the steam table).

The total heat in the 10 bar g steam (due to its 'wetness'), is less than the total heat in saturated steam at 5 bar g, and so the lower pressure steam will not contain enough heat to be totally dry. The dryness fraction of the lower pressure steam is the quotient of the two total heat figures.

$$\text{Dryness fraction of the 5 bar a steam} = 2\,697.15 / 2\,748.65$$

$$= 0.98$$

The energy available for heat transfer at 5 bar a is 0.98 x h_{fg} at 5 bar a

$$= 0.98 \times 2\,108.23 \text{ kJ/kg}$$

$$= 2\,066 \text{ kJ/kg}$$

The steam flowrate can now be determined from Equation 2.8.1, where h_{fg} is the enthalpy of evaporation available after accounting for wet steam.

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW} \times 3\,600}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\text{Steam flowrate} = \frac{500 \text{ kW}}{2\,066 \text{ kJ/kg}} \left(\frac{\text{kJ/s}}{\text{kW}}\right) \times 3\,600 \text{ s/h}$$

Steam flowrate = 871 kg/hof wet steam

Determine the pressure drop ratio ($\varkappa$) at full load

$$\text{Pressure drop ratio } (\varkappa) = \frac{10 \text{ bar a} - 5 \text{ bar a}}{10 \text{ bar a}} = 0.5$$

Determine the required K_{vr}
The pressure drop ratio at full load is larger than 0.42, so critical conditions apply and Equation 6.4.3 may be used to find the required K_{vr}.

$$\dot{m} = 12\, K_v\, P_1 \qquad \textbf{Equation 6.4.3}$$

$$\dot{m} = 12\, K_v\, P_1$$
$$871 \text{ kg/h} = 12\, k_{vr}\, 10 \text{ bar a}$$
$$K_{vr} = \frac{871}{12 \times 10}$$
$$K_{vr} = 7.26$$

A DN25 control valve with a K_{vs} of 10 is initially selected. A calculation can now be carried out to determine if noise is an issue with this sized valve passing wet steam in the valve outlet.

The speed of sound in the valve outlet:

$$C = 31.6 \sqrt{\gamma\, R\, T}$$
As the steam is wet, $\gamma = 1.035 + 0.1(x)$ where 'x' is the dryness fraction
$$\gamma = 1.035 + 0.1(0.98)$$
$$\gamma = 1.133$$

R = 0.461 5 kJ/kg (the gas constant for steam)

The temperature of wet steam at 5 bar a is the same as dry saturated steam at the same pressure;

T = 425 K.

The speed of sound in the wet steam in the value outlet:

Speed of sound $C = 31.6 \sqrt{\gamma\, R\, T}$
$$C = 31.6 \sqrt{1.133 \times 0.4615 \times 425}$$
$$C = 31.6 \times 14.91$$
$$C = 471 \text{ m/s}$$

A DN25 valve has an outlet area of 0.000 9 m²

The specific volume of wet steam at 5 bar a, and 0.98 dry = 0.367 4 m³/kg

The volumetric flow = 871 kg/h x 0.367 4 m³/kg
= 320 m³/h

The volumetric flow = 0.088 8 m³/s

$$\text{Valve outlet velocity} = \frac{\text{Volumetric flowrate}}{\text{Outlet area}}$$
$$= \frac{0.0888 \text{ m}^3/\text{s}}{0.0009 \text{ m}^2}$$

Valve outlet velocity = 99 m/s

The noise criterion for wet steam in the valve outlet = 40 m/s

As this outlet velocity is higher than 40 m/s, it suggests that the DN25 control valve might:

1. Create an unacceptable noise.

2. Cause unreasonable erosion in the valve outlet.

The DN25 control valve will therefore be unsuitable for this application where wet steam passes through the valve outlet.

One solution to this problem is to fit a larger bodied valve with the same K_{vs} of 10 to reduce the wet steam outlet velocity.

$$\text{As, valve outlet velocity} = \frac{\text{Volumetric flowrate}}{\text{Outlet area}}$$

$$\text{Minimum outlet area} = \frac{\text{Volumetric flowrate}}{\text{Valve outlet velocity}}$$

$$\text{Minimum outlet area} = \frac{0.088\,8 \text{ m}^3/\text{s}}{40 \text{ m/s}}$$

$$\text{Minimum outlet area} = 0.002\,22 \text{ m}^2$$

Consider Table 6.4.1 to determine the minimum sized control valve with an outlet area greater than 0.002 22 m².

Table 6.4.1 Typical valve outlet areas DN15 - DN200 control valves

Control valve size	Outlet areas (m²)
DN15	0.000 4
DN20	0.000 6
DN25	0.000 9
DN32	0.001 4
DN40	0.001 8
DN50	0.002 9
DN65	0.004 2
DN80	0.006 2
DN100	0.010 3
DN125	0.015 7
DN150	0.022 2
DN200	0.037 7

It can be seen from Table 6.4.1 that the smallest valve required to satisfy the maximum outlet velocity of 40 m/s for wet steam is a DN50 valve, having an outlet area of 0.002 9 m².

Therefore, due to wet steam passing through the valve outlet, the size of the control valve would double from, in this instance a DN25 (1") to DN50 (2").

A better solution might be to fit a separator before the control valve. This will allow the smaller DN25 control valve to be used, and is preferred because:

- It will give better regulation as it is more appropriately sized to handle changes in the steam load.
- It will ensure dry steam passes through the control valve, thereby reducing the propensity for erosion at the valve seat and valve outlet.
- It will ensure optimal performance of the heat exchanger, as the heating surface is not thermally insulated by moisture from wet steam.
- The cost of the smaller valve and its actuator plus separator will probably be the same as the larger valve with a larger actuator.

Sizing on an arbitrary pressure drop

If the apparatus working pressure is not known, it is sometimes possible to compromise.

It should be stressed that this method should only be used as a last resort, and that every effort should be made to determine the working pressures and flowrate.

Under these circumstances, it is suggested that the control valve be selected using a pressure drop of 10% to 20% of the upstream pressure. In this way, the selected control valve will more than likely be oversized.

To help this situation, an equal percentage valve will give better operational performance than a linear valve (this is discussed in more detail in Module 6.5 'Control valve characteristics'.

Sizing on an arbitrary pressure drop is not recommended for critical applications.

The higher the pressure drop the better?

It is usually better to size a steam valve with critical pressure drop occurring across the control valve at maximum load. This helps to reduce the size and cost of the control valve.

However, the application conditions may not allow this.

For example, if the heat exchanger working pressure is 4.5 bar a, and the maximum available steam pressure is only 5 bar a, the valve can only be sized on a 10% pressure drop ([5 – 4.5] / 5) = 0.1. In this situation, sizing on critical pressure drop would have unduly reduced the size of the control valve, and the heat exchanger would be starved of steam.

If it is impossible to increase the steam supply pressure, one solution is to install a larger heat exchanger operating at a lower pressure. In this way, the pressure drop will increase across the control valve.

This could result in a smaller valve but, unfortunately, a larger heat exchanger, because the heat exchanger operating pressure (and temperature) is now lower.

However, a larger heat exchanger working at a lower pressure brings some advantages:

- There is less tendency for the heating surfaces to scale and foul as the required steam temperature is lower.
- Less flash steam is produced in the condensate system leading to less backpressure in the condensate return pipework.

It is important to balance the cost of the valve and heat exchanger, the ability of the valve to control properly, and the effects on the rest of the system, as explained previously.

On steam systems, equal percentage valves will usually be a better choice than linear valves, as low pressure drops will have less effect on their operating performance.

Types of steam heated heat exchangers

This subject is outside the scope of this Module, but it is useful to have a brief look at the two main types of heat exchanger used for steam heating and process applications.

The shell and tube heat exchanger

Traditionally, the shell-and-tube heat exchanger has been used for many steam heating and process applications across a broad spectrum of industries. It is robust and often 'over-engineered' for the job. It tends to have an inherently high mass and large thermal hysteresis, which can make it unwieldy for certain critical applications.

Shell-and-tube heat exchangers are often greatly oversized on initial installation, mainly because of large fouling factors applied to the calculation. They tend to have low steam velocity in the steam tube, which reduces:

- Turbulence.
- The sheer stress between the flowing steam and the tube wall.
- Heat transfer.

Low sheer stress also tends not to clean the tube surfaces; hence high fouling factors are usually applied at the design stage leading to oversizing. Due to oversizing, the actual steam pressure after installation is often much less than predicted. If this is not anticipated, the steam trap might not be correctly sized and the steam tubes might flood with condensate, causing erratic control and poor performance.

The plate (and frame) heat exchanger

Plate heat exchangers are a useful alternative; being relatively small and light, they have a small mass and are extremely quick to respond to changes in heat load.

When properly designed, they tend not to foul, but if they do, they are easily disassembled, cleaned and recommissioned. Compared to shell-and-tube exchangers, they can operate at lower pressures for the same duty, but because of their high heat transfer characteristics, and a lower requirement for oversizing, they are still smaller and less expensive than a comparable shell-and-tube exchanger.

Plate heat exchangers (when properly engineered to use steam) are therefore more economically suited to high pressure drops across control valves than their shell-and-tube counterparts. This can give the advantage of smaller and less expensive control valves, whilst minimising the cost of the heat exchanger itself. Generally, it is better to design the system so that the plate exchanger operates with critical pressure drop (or the highest possible pressure drop) across the control valve at full load.

It must be stressed that not all plate heat exchangers are suitable for steam use. It is very easy to buy a heat exchanger designed for liquid use and wrongly assume that it will perform perfectly when heated with steam. Correct selection for steam is not just a matter of pressure/temperature compatibility. Proper expertise is available from bona fide manufacturers, and this should always be sought when steam is the prime energy source.

Steam sizing examples using charts

The required 'flow coefficient' (K_{vr}) may be determined in a number of ways, including calculation using Equation 3.21.2 or Equation 6.4.3 or via computer software. An alternative method of simple valve sizing is to use a K_v chart, Figure 6.4.8. A few examples of how these may be used are shown below:

Saturated steam

Example 6.4.3 Critical pressure drop application

Steam demand of heat exchanger = 800 kg/h

Steam pressure upstream of valve = 9 bar a

Steam pressure required in heat exchanger = 4 bar a

Reference steam K_v chart (Figure 6.4.8)

1. Draw a line from 800 kg/h on the steam flow ordinate.
2. Draw a horizontal line from 9 bar on the inlet pressure ordinate.
3. At the point where this crosses the critical pressure drop line (top right diagonal) draw a vertical line downwards until it intersects the horizontal 800 kg/h line.
4. Read the K_v at this crossing point, i.e. $K_{vr} \approx 7.5$

Example 6.4.4 A non critical-pressure-drop application

Steam demand of heat exchanger = 200 kg/h

Steam pressure upstream of valve = 6 bar a

Steam pressure required in heat exchanger = 5 bar a

Reference steam K_v chart (Appendix 1)

As in example 6.4.3, draw a line across from the 200 kg/h steam flow ordinate, and then draw another line from the 6 bar inlet pressure ordinate to the 1 bar pressure drop line.

Drop a vertical line from the resulting intersection point, to meet the 200 kg/h horizontal and read the K_v at this crossing point i.e. $K_{vr} \approx 3.8$

Example 6.4.5 Find the pressure drop (ΔP) across the valve having a known K_{vs} value

Steam demand of heat exchanger = 3 000 kg/h

Steam pressure upstream of valve = 10 bar a

K_{vs} of valve to be used = 36

Reference steam K_v chart (Appendix 1)

Draw a horizontal line from 3 000 kg/h to meet at the K_v 36 line. Draw a vertical line upward from this intersection to meet the 10 bar horizontal line.

Read the pressure drop at this crossing point, $\Delta P \approx 1.6$ bar.

Note: In the examples, to convert gauge pressure (bar g) to absolute pressure (bar a) simply add ‘1’ to the gauge pressure, for example, 10 bar g = 11 bar a.

Saturated steam sizing chart

This sizing chart is empirical and should not be used for critical applications

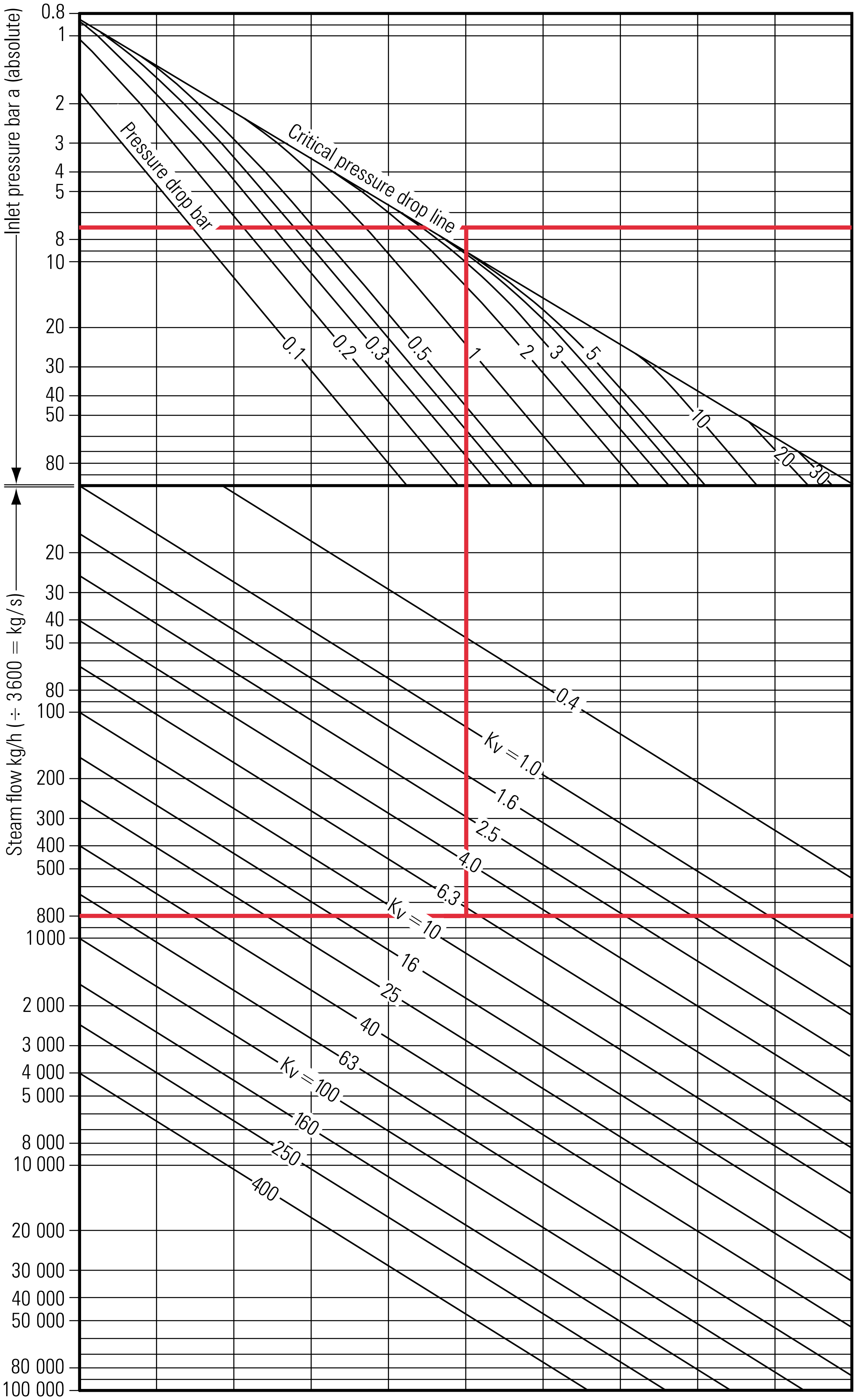


Fig. 6.4.8 Steam K_v chart

Superheated steam

To size a valve for use with superheated steam refer to Example 6.4.6 and the superheated steam chart, Figure 6.4.9.

Example 6.4.6

The following example shows how to use the chart for 100°C of superheat: follow the respective steam flow line on the left to the vertical line which represents 100°C of superheat, then draw a horizontal line across as normal from the resulting intersection. By doing this, the graph introduces a correction factor for the superheat and corrects the K_v value.

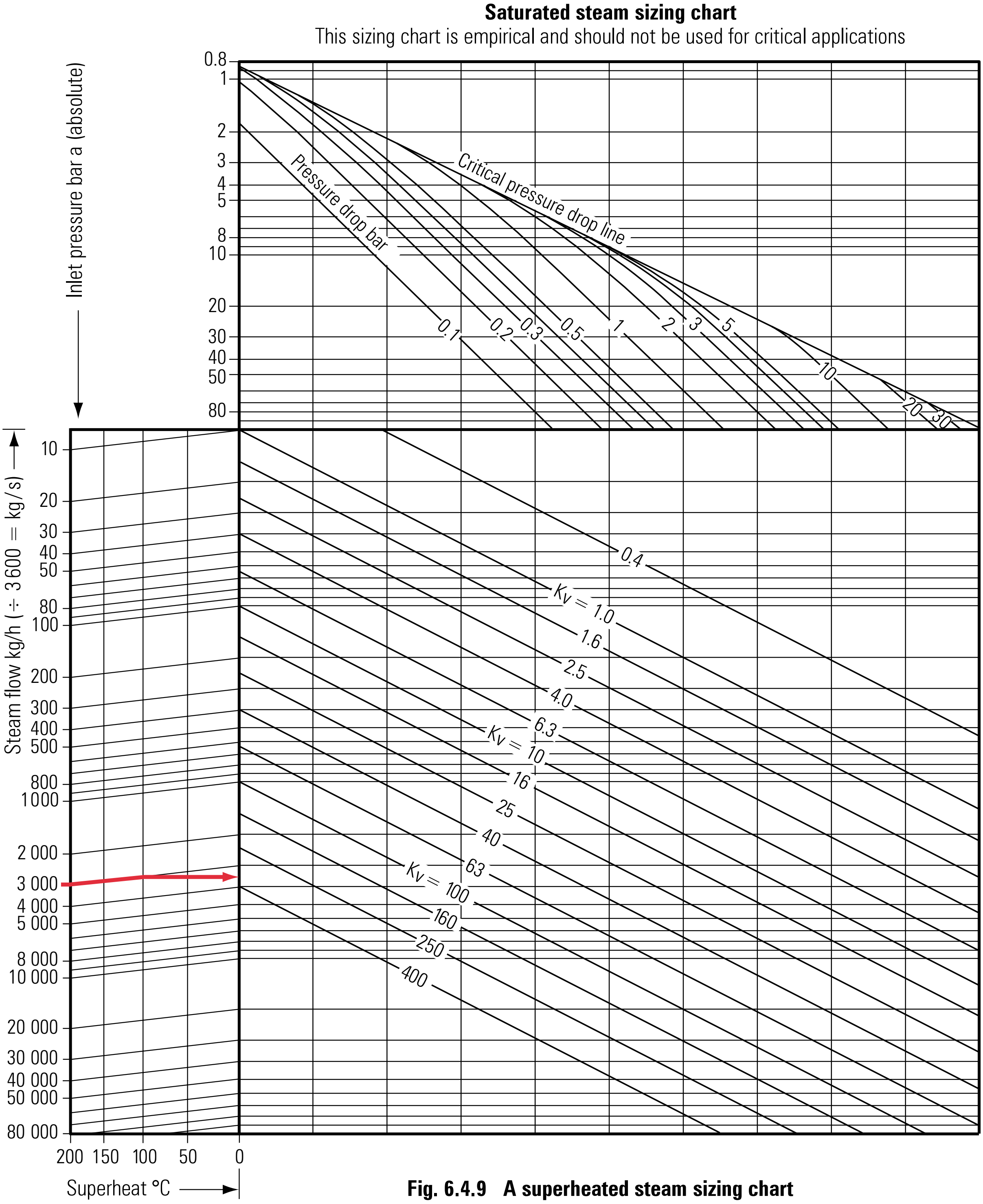


Fig. 6.4.9 A superheated steam sizing chart

Selecting a control valve for steam service

The previous Section covered the procedure for sizing a control valve based on the flowrate it needs to pass, and the pressure drop across the valve. From this data, the K_{vs} value of the control valve can be obtained. Reference to the appropriate product literature will provide the information needed to select the required valve size.

Control valve selection requires several other factors to be taken into account. The body material must be selected to suit the application. Valves are available in cast iron, SG iron, bronze, steel, stainless steel, and exotic materials for very special applications, for example titanium steel.

The design and material of the control valve must be suitable for the pressure of the system in which it will be fitted. In Europe, most valves have a nominal pressure body rating, stipulated by the letters 'PN' which actually means 'Pression Nominale'. This relates to the maximum pressure (bar gauge) the valve can withstand at a temperature of 120°C. The higher the temperature, the lower the allowable pressure, resulting in a typical pressure / temperature graph as shown in Figure 6.4.10.

It should be noted that the type of material used in manufacturing the control valve plays an important part in the pressure / temperature chart. Typical limiting conditions are:

PN16 - Cast iron	PN25 - SG iron	PN40 - Cast steel

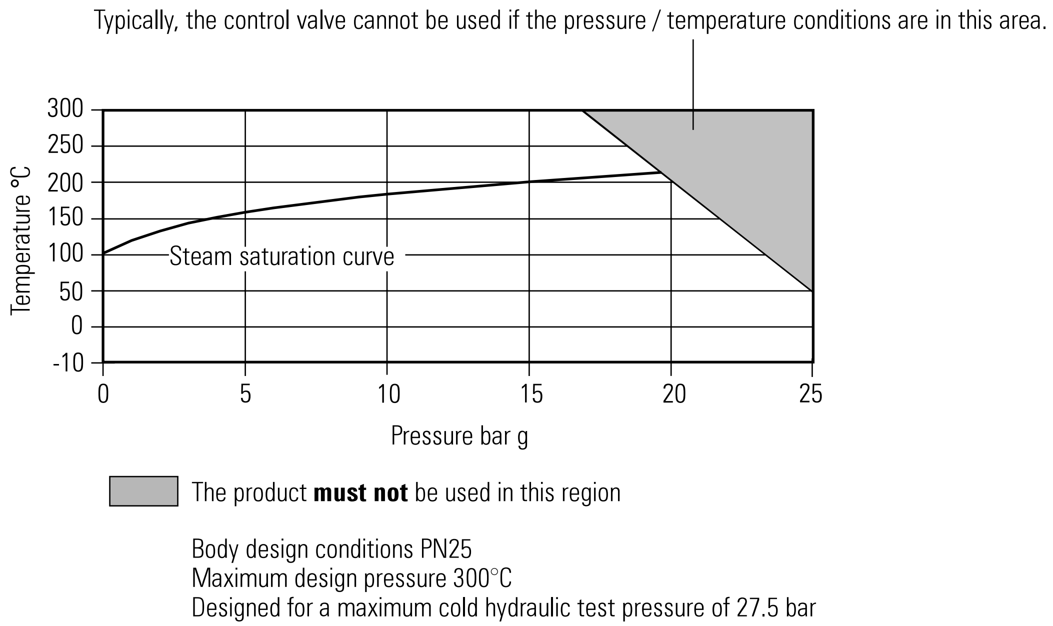


Fig. 6.4.10 An example of PN25 temperature / pressure limiting conditions

The design thickness and body jointing methods also have an effect. For example, an SG iron valve could have a PN16 rating and may also be available with a slightly different design, with a PN25 rating. Local or national regulations may affect the limits, as may the type of connection which is used.

A checklist of the major factors to be taken into account when selecting a control valve for steam service include:

1. Mass flow or volumetric flow to be considered (typically maximum, normal or minimum).

2. Flow medium (this may affect the type of material used for the valve body and internals).

3. Upstream pressure available at maximum, normal and minimum loads.

4. Downstream pressure for maximum, normal and minimum loads.

5. K_v value required.

6. Pressure drop across the valve at maximum, normal and minimum loads.

7. Body size of valve.

8. Body material and nominal pressure rating.

9. Maximum differential pressure for shut-off.

10. Connection required. Which pipe connections are required on the inlet and outlet of the valve? Screwed or flanged connections, and which type of flange, for example, ANSI, EN 1092 or DIN?

11. Maximum temperature of the medium flowing through the valve.

12. Any special requirements, for example, special gland packing variations; hardened valve seat and plug, soft seats for absolutely tight shut-off; and others.

Note: Manufacturers restrict the leakage rates of control valves to agreed limits and / or they are sometimes the subject of national standards. Also see point 17.

13. Details of the application control requirements. This is explained in more detail in Module 6.5. Briefly, an application needing on / off control (either fully-open or fully-closed) may require a valve characteristic suited to that purpose, whereas an application calling for continuous control (any degree of opening or closing), might perform better with a different type of valve characteristic.

14. Method of actuation and type of control to be used; for example, self-acting, electric, pneumatic, electropneumatic.

15. Noise levels. It is often a requirement to keep noise below 85 dBA at 1 m from the pipe if people are to work unprotected in the area. Keeping the same size internals but increasing the size of the connections may achieve this. (Many control valves have the option of reduced trim variants, alternatively special noise-reducing trims are available, and / or acoustic lagging can be applied to the valve and pipework. Valves for critical process applications should be sized using computer software utilising the IEC 60534 standard or national equivalent.

16. Pressure drops, sizes of valve body and noise level are related and should be considered. It is good practice to keep the downstream steam velocity in the valve body typically below 150 m/s for saturated steam and 250 m/s for superheated steam. This can be achieved by increasing the valve body size, which will also reduce the velocity in the valve outlet and the likelihood of excess noise. It is possible to consider a saturated steam exit velocity of 150 m/s to 200 m/s if the steam is always guaranteed to be dry saturated at the valve inlet. This is because, under these circumstances, the steam leaving the control valve will be superheated due to the superheating affect of reducing the pressure of dry saturated steam. Please note that these are general figures, different standards will quote different guidelines.

17. Leakage and isolation. Control valves are meant to control flowrate rather than isolate the supply, and are likely to leak slightly when fully shut. Control valves will be manufactured to a standard relating to shut-off tightness. Generally, the better the shut-off, the higher the cost of the valve. For steam control valves, a leakage rate of 0.01% is perfectly adequate for most applications.

18. Turndown. Usually expressed as a ratio of the application maximum expected flow to the minimum controllable flow through a control valve.

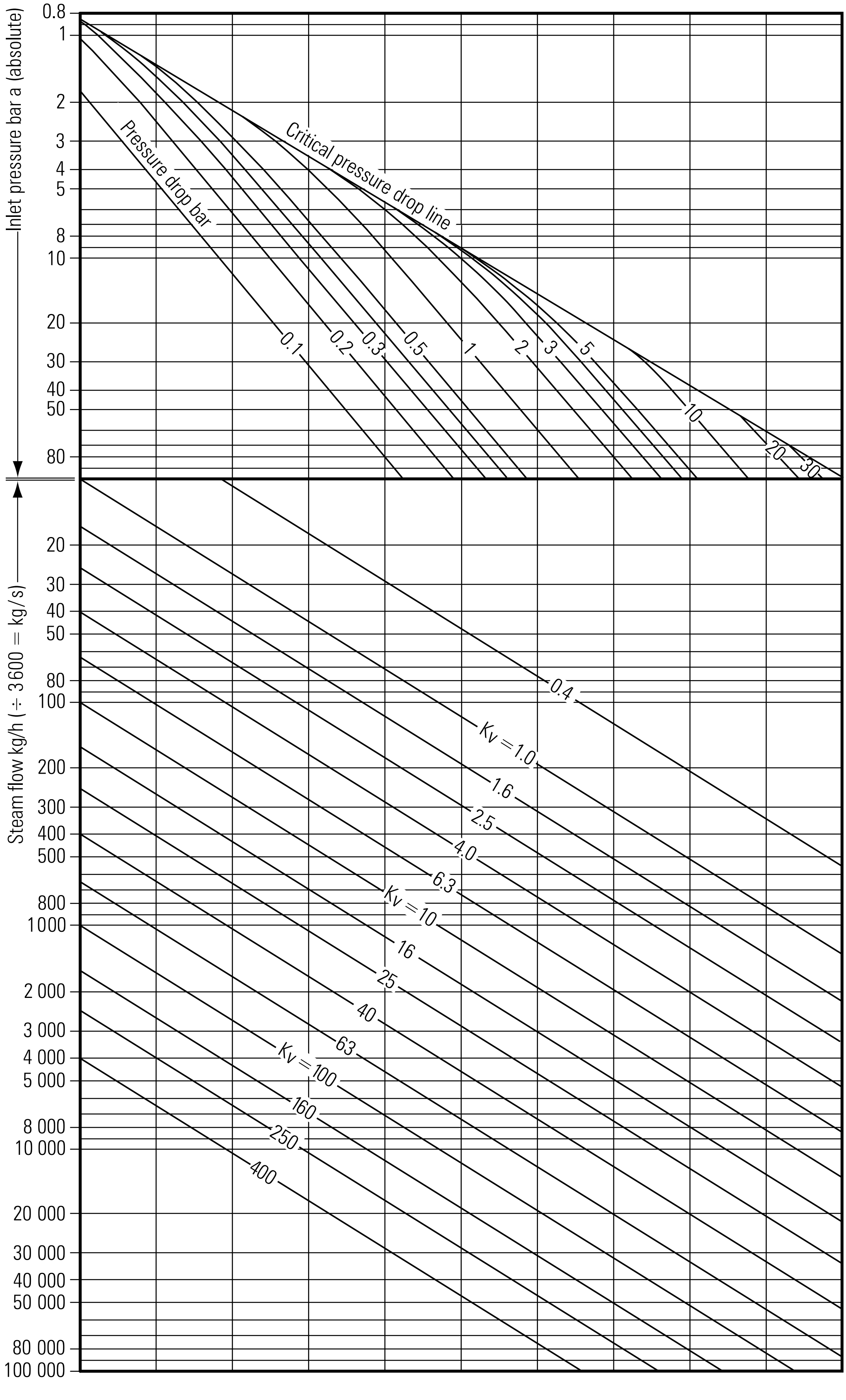
19. Rangeability. Usually expressed as a ratio of the valve maximum controllable flow to the minimum controllable flow, between which the characteristics of the control valve are maintained. Typically, a rangeability of 50:1 is acceptable for steam applications.

20. It would be wrong to end this Module on control valves without mentioning cost. The type of valve, its materials of construction, variations in design and special requirements will inevitably result in cost variations. For optimum economy the selected valve should be correct for that application and not over-specified.

Appendix 1 Saturated steam valve sizing chart

Saturated steam sizing chart

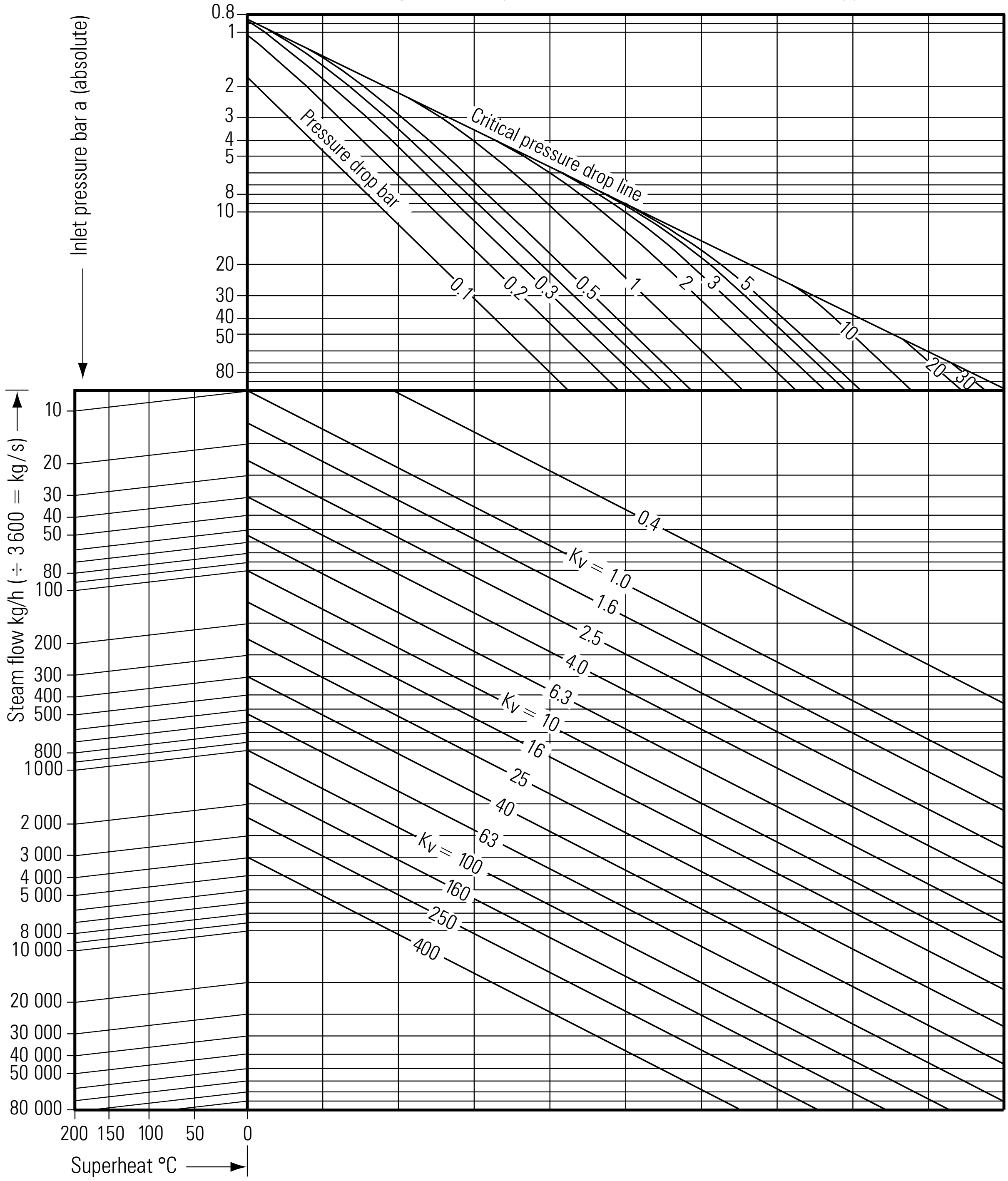
This sizing chart is empirical and should not be used for critical applications



Appendix 2 Superheated steam valve sizing chart

Saturated steam sizing chart

This sizing chart is empirical and should not be used for critical applications



Questions

1. What factor determines the rate of heat transfer between fluids across a barrier?

a| The overall heat transfer coefficient 'U' ☐

b| The area of the heat transfer surface ☐

c| The mean temperature difference between the fluids ☐

d| All of the above ☐

2. **The upstream saturated steam pressure before a control valve is 7 bar g, the downstream pressure is 4 bar g, and the valve K_{vs} is 4. What is the pressure drop ratio?**

a| 0.429 ☐

b| 0.75 ☐

c| 0.375 ☐

d| 0.6 ☐

3. **Using Appendix 1, what is the flow of saturated steam through a valve of K_{vs} 10, when the upstream pressure is 9 bar g, and the downstream pressures are (i) 2 bar g (ii) 4.5 bar g (iii) 8 bar g.**

a| (i) 1 080 kg/h (ii) 1 000 kg/h (iii) 1 000 kg/h ☐

b| (i) 40 kg/h (ii) 120 kg/h (iii) 120 kg/h ☐

c| (i) 1 200 kg/h (ii) 695 kg/h (iii) 695 kg/h ☐

d| (i) 1 200 kg/h (ii) 1 200 kg/h (iii) 695 kg/h ☐

4. **A heat exchanger control valve is supplied with wet steam at 4 bar g. If the steam is dry in the heat exchanger, its flowrate is 97 kg/h and the heat exchanger is delivering 60 kW, what is the steam pressure in the heat exchanger? (Steam tables are required). Use Equation 2.8.1.**

a| 2.1 bar g ☐

b| 0.48 bar g ☐

c| 0.48 bar a ☐

d| 2.1 bar a ☐

5. In the above example, what is the K_{vr}?

a| 17 ☐

b| 1.6 ☐

c| 5.4 ☐

d| 0.7 ☐

6. For Figure 6.4.7; with an upstream pressure of 3 bar g, determine the pressure drop across a control valve with a K_{vs} of 16 passing 700 kg/h of dry saturated steam. Use Spirax Sarco on-line valve sizing calculator in the Engineering Support Centre.

a| 0.981 bar ☐

b| Critical pressure drop ☐

c| 0.5 bar ☐

d| 0.1 bar ☐

Answers

1: d, 2: c, 3: d, 4: b 5: b, 6: a

Module 6.5

Control Valve Characteristics

Control Valve Characteristics

Flow characteristics

All control valves have an inherent flow characteristic that defines the relationship between 'valve opening' and flowrate under constant pressure conditions. Please note that 'valve opening' in this context refers to the relative position of the valve plug to its closed position against the valve seat. It does not refer to the orifice pass area. The orifice pass area is sometimes called the 'valve throat' and is the narrowest point between the valve plug and seat through which the fluid passes at any time. For any valve, however it is characterised, the relationship between flowrate and orifice pass area is always directly proportional.

Valves of any size or inherent flow characteristic which are subjected to the same volumetric flowrate and differential pressure will have exactly the same orifice pass area. However, different valve characteristics will give different 'valve openings' for the same pass area. Comparing linear and equal percentage valves, a linear valve might have a 25% valve opening for a certain pressure drop and flowrate, whilst an equal percentage valve might have a 65% valve opening for exactly the same conditions. The orifice pass areas will be the same.

The physical shape of the plug and seat arrangement, sometimes referred to as the valve 'trim', causes the difference in valve opening between these valves. Typical trim shapes for spindle operated globe valves are compared in Figure 6.5.1.

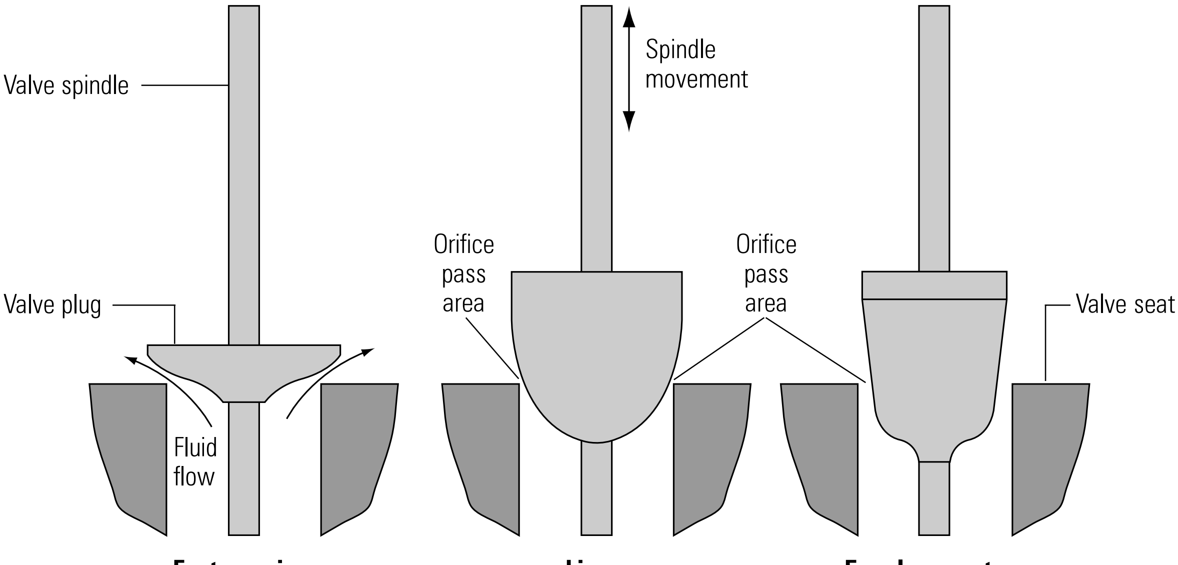


Fig. 6.5.1 The shape of the trim determines the valve characteristic

In this Module, the term 'valve lift' is used to define valve opening, whether the valve is a globe valve (up and down movement of the plug relative to the seat) or a rotary valve (lateral movement of the plug relative to the seat).

Rotary valves (for example, ball and butterfly) each have a basic characteristic curve, but altering the details of the ball or butterfly plug may modify this. The inherent flow characteristics of typical globe valves and rotary valves are compared in Figure 6.5.2.

Globe valves may be fitted with plugs of differing shapes, each of which has its own inherent flow/opening characteristic. The three main types available are usually designated:

- Fast opening.
- Linear.
- Equal percentage.

Examples of these and their inherent characteristics are shown in Figures 6.5.1 and 6.5.2.

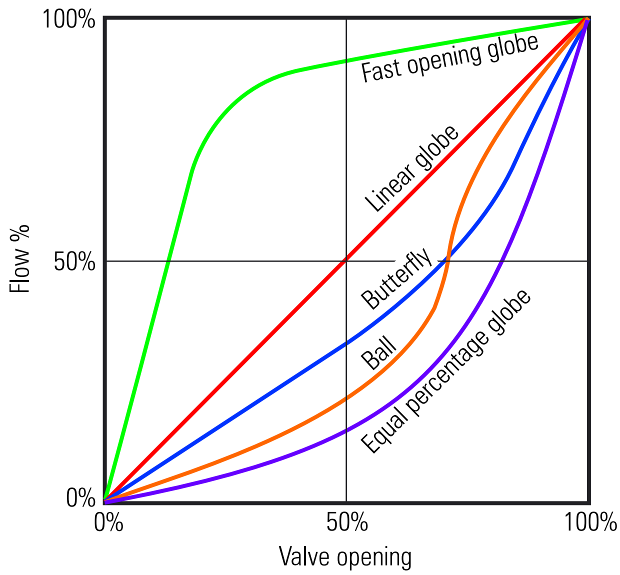


Fig. 6.5.2 Inherent flow characteristics of typical globe valves and rotary valves

Fast opening characteristic

The fast opening characteristic valve plug will give a large change in flowrate for a small valve lift from the closed position. For example, a valve lift of 50% may result in an orifice pass area and flowrate up to 90% of its maximum potential.

A valve using this type of plug is sometimes referred to as having an 'on / off' characteristic.

Unlike linear and equal percentage characteristics, the exact shape of the fast opening curve is not defined in standards. Therefore, two valves, one giving a 80% flow for 50% lift, the other 90% flow for 60% lift, may both be regarded as having a fast opening characteristic.

Fast opening valves tend to be electrically or pneumatically actuated and used for 'on / off' control.

The self-acting type of control valve tends to have a plug shape similar to the fast opening plug in Figure 6.5.1. The plug position responds to changes in liquid or vapour pressure in the control system. The movement of this type of valve plug can be extremely small relative to small changes in the controlled condition, and consequently the valve has an inherently high rangeability. The valve plug is therefore able to reproduce small changes in flowrate, and should not be regarded as a fast opening control valve.

Linear characteristic

The linear characteristic valve plug is shaped so that the flowrate is directly proportional to the valve lift (H), at a constant differential pressure. A linear valve achieves this by having a linear relationship between the valve lift and the orifice pass area (see Figure 6.5.3).

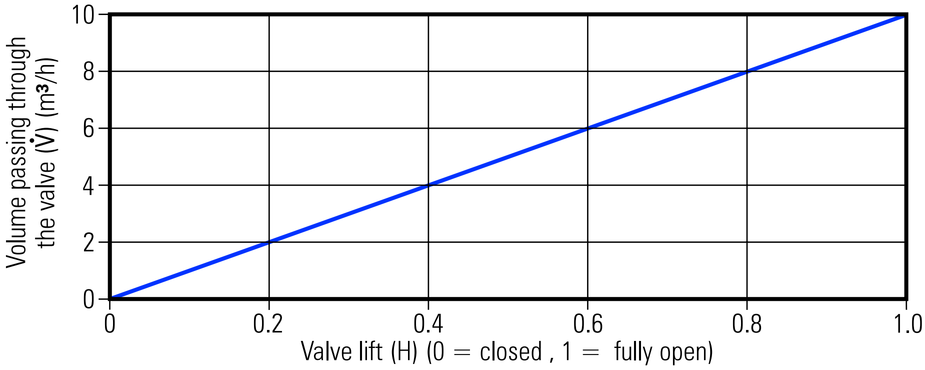


Fig. 6.5.3 Flow / lift curve for a linear valve

For example, at 40% valve lift, a 40% orifice size allows 40% of the full flow to pass.

Equal percentage characteristic (or logarithmic characteristic)

These valves have a valve plug shaped so that each increment in valve lift increases the flowrate by a certain percentage of the previous flow. The relationship between valve lift and orifice size (and therefore flowrate) is not linear but logarithmic, and is expressed mathematically in Equation 6.5.1:

$$\dot{V} = \frac{e^x}{\tau}\,\dot{V}_{max} \qquad \textbf{Equation 6.5.1}$$

Where:

$\dot{V}$ = Volumetric flow through the valve at lift H.

x = (ln τ) H

Note: 'ln' is a mathematical function known as 'natural logarithm'.

τ = Valve rangeability (ratio of the maximum to minimum controllable flowrate, typically 50 for a globe type control valve)

H = Valve lift (0 = closed, 1 = fully open)

$\dot{V}_{max}$ = Maximum volumetric flow through the valve

Example 6.5.1

The maximum flowrate through a control valve with an equal percentage characteristic is 10 m³/h. If the valve has a turndown of 50:1, and is subjected to a constant differential pressure, by using Equation 6.5.1 what quantity will pass through the valve with lifts of 40%, 50%, and 60% respectively?

$\dot{V}_{max}$ = Maximum volumetric flow through the valve = 10 m³/h

H = Valve lift (0 closed to 1 fully open) = 0.4; 0.5; 0.6

τ = Valve rangeability = 50

$$\dot{V} = \frac{e^x}{\tau}\,\dot{V}_{max} \qquad \textbf{Equation 6.5.1}$$

40% open, H = 0.4

$x = (\ln \tau) \times H$

$x = (\ln 50) \times 0.4$

$x = 3.912 \times 0.4$

$\mathbf{x = 1.564\,8}$

$\dot{V} = \dfrac{e^{1.564\,8}}{\tau} \times 10$

$\dot{V} = \dfrac{4.781\,7}{50} \times 10$

$\dot{V} = 0.095\,6 \times 10$

$\mathbf{\dot{V} = 0.956\ m^3/h}$

50% open, H = 0.5

$x = (\ln \tau) \times H$

$x = (\ln 50) \times 0.5$

$x = 3.912 \times 0.5$

$\mathbf{x = 1.956}$

$\dot{V} = \dfrac{e^{1.956}}{\tau} \times 10$

$\dot{V} = \dfrac{7.071}{50} \times 10$

$\dot{V} = 0.141\,4 \times 10$

$\mathbf{\dot{V} = 1.414\ m^3/h}$

60% open, H = 0.6

$x = (\ln \tau) \times H$

$x = (\ln 50) \times 0.6$

$x = 3.912 \times 0.6$

$\mathbf{x = 2.347}$

$\dot{V} = \dfrac{e^{2.347}}{\tau} \times 10$

$\dot{V} = \dfrac{10.45}{50} \times 10$

$\dot{V} = 0.209\,1 \times 10$

$\mathbf{\dot{V} = 2.091\ m^3/h}$

The increase in volumetric flowrate through this type of control valve increases by an equal percentage per equal increment of valve movement:

- When the valve is 50% open, it will pass 1.414 m³/h, an increase of 48% over the flow of 0.956 m³/h when the valve is 40% open.
- When the valve is 60% open, it will pass 2.091 m³/h, an increase of 48% over the flow of 1.414 m³/h when the valve is 50% open.

It can be seen that (with a constant differential pressure) for any 10% increase in valve lift, there is a 48% increase in flowrate through the control valve. This will always be the case for an equal percentage valve with rangeability of 50. For interest, if a valve has a rangeability of 100, the incremental increase in flowrate for a 10% change in valve lift is 58%.

Table 6.5.1 shows how the change in flowrate alters across the range of valve lift for the equal percentage valve in Example 6.5.1 with a rangeability of 50 and with a constant differential pressure.

Table 6.5.1
Change in flowrate and valve lift for an equal percentage characteristic with constant differential pressure

Valve Lift (H)	Flowrate ($\dot{V}$ m³/h)	Increase in flow from previous increment (%)
0.0	0.20 *	-
0.1	0.30	48%
0.2	0.44	48%
0.3	0.65	48%
0.4	0.96	48%
0.5	1.41	48%
0.6	2.09	48%
0.7	3.09	48%
0.8	4.57	48%
0.9	6.76	48%
1.0	10.00	48%

* Flowrate according to theoretical characteristic due to rangeability. In practice the valve will be fully shut at zero lift.

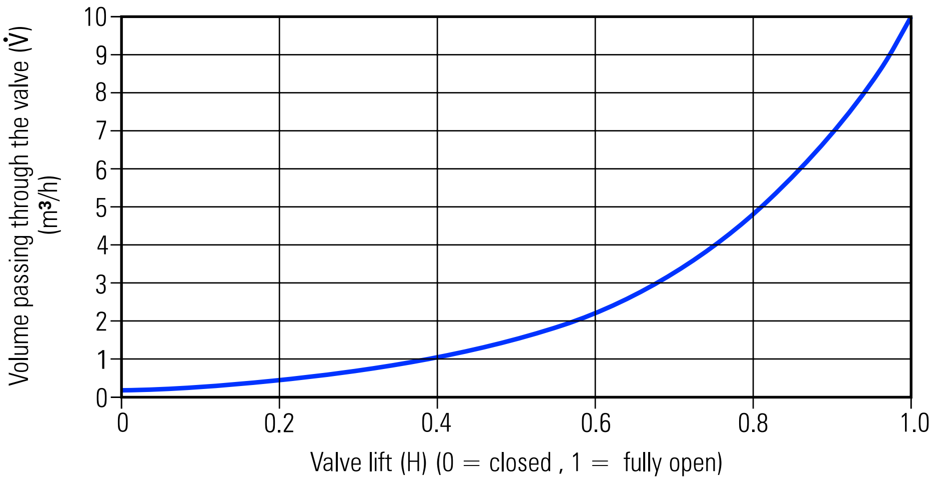


Fig. 6.5.4 Flowrate and valve lift for an equal percentage characteristic with constant differential pressure for Example 6.5.1

A few other inherent valve characteristics are sometimes used, such as parabolic, modified linear or hyperbolic, but the most common types in manufacture are fast opening, linear, and equal percentage.

Matching the valve characteristic to the installation characteristic

Each application will have a unique installation characteristic that relates fluid flow to heat demand. The pressure differential across the valve controlling the flow of the heating fluid may also vary:

- In water systems, the pump characteristic curve means that as flow is reduced, the upstream valve pressure is increased (refer to Example 6.5.2, and Module 6.3).
- In steam temperature control systems, the pressure drop over the control valve is deliberately varied to satisfy the required heat load.

The characteristic of the control valve chosen for an application should result in a direct relationship between valve opening and flow, over as much of the travel of the valve as possible.

This section will consider the various options of valve characteristics for controlling water and steam systems. In general, linear valves are used for water systems whilst steam systems tend to operate better with equal percentage valves.

1. A water circulating heating system with three-port valve

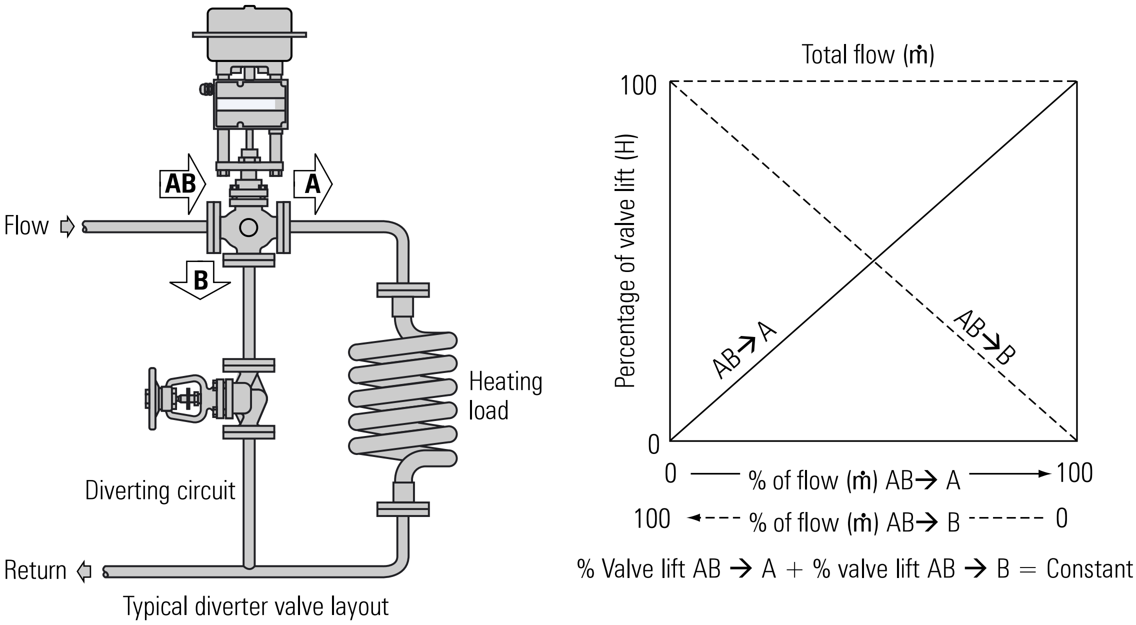


Fig. 6.5.5 A three-port diverting valve on a water heating system

In water systems where a constant flowrate of water is mixed or diverted by a three-port valve into a balanced circuit, the pressure loss over the valve is kept as stable as possible to maintain balance in the system.

Conclusion - The best choice in these applications is usually a valve with a linear characteristic. Because of this, the installed and inherent characteristics are always similar and linear, and there will be limited gain in the control loop.

2. A boiler water level control system – a water system with a two-port valve

In systems of this type (an example is shown in Figure 6.5.6), where a two-port feedwater control valve varies the flowrate of water, the pressure drop across the control valve will vary with flow. This variation is caused by:

- The pump characteristic. As flowrate is decreased, the differential pressure between the pump and boiler is increased (this phenomenon is discussed in further detail in Module 6.3).
- The frictional resistance of the pipework changes with flowrate. The head lost to friction is proportional to the square of the velocity. (This phenomenon is discussed in further detail in Module 6.3).
- The pressure within the boiler will vary as a function of the steam load, the type of burner control system and its mode of control.

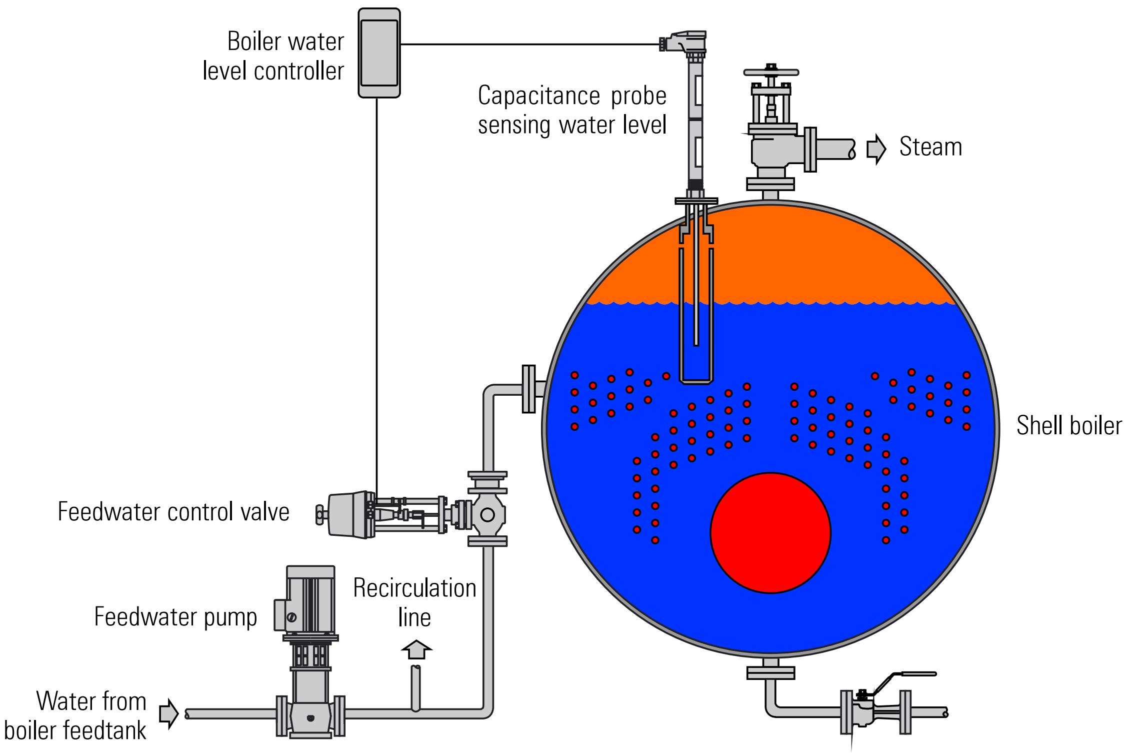


Fig. 6.5.6 A modulating boiler water level control system (not to scale)

Example 6.5.2 Select and size the feedwater valve in Figure 6.5.6

In a simplified example (which assumes a constant boiler pressure and constant friction loss in the pipework), a boiler is rated to produce 10 tonnes of steam per hour. The boiler feedpump performance characteristic is tabulated in Table 6.5.2, along with the resulting differential pressure (ΔP) across the feedwater valve at various flowrates at, and below, the maximum flow requirement of 10 m^3/h of feedwater.

Note: The valve ΔP is the difference between the pump discharge pressure and a constant boiler pressure of 10 bar g. Note that the pump discharge pressure will fall as the feedwater flow increases. This means that the water pressure before the feedwater valve also falls with increased flowrate, which will affect the relationship between the pressure drop and the flowrate through the valve.

It can be determined from Table 6.5.2 that the fall in the pump discharge pressure is about 26% from no-load to full-load, but the fall in differential pressure across the feedwater valve is a lot greater at 72%. If the falling differential pressure across the valve is not taken into consideration when sizing the valve, the valve could be undersized.

Table 6.5.2 Feedwater flowrate, pump discharge pressure, and valve differential pressure (ΔP)

Flow (m^3/h)	0	1	2	3	4	5	6	7	8	9	10
Pump discharge pressure (bar)	15.58	15.54	15.42	15.23	14.95	14.58	14.41	13.61	13.00	12.31	11.54
Valve ΔP (bar)	5.58	5.54	5.42	5.23	4.95	4.58	4.41	3.61	3.00	2.31	1.54

As discussed in Modules 6.2 and 6.3, valve capacities are generally measured in terms of K_v. More specifically, K_{vs} relates to the pass area of the valve when fully open, whilst K_{vr} relates to the pass area of the valve as required by the application.

Consider if the pass area of a fully open valve with a K_{vs} of 10 is 100%. If the valve closes so the pass area is 60% of the full-open pass area, the K_{vr} is also 60% of 10 = 6. This applies regardless of the inherent valve characteristic. The flowrate through the valve at each opening will depend upon the differential pressure at the time.

Using the data in Table 6.5.2, the required valve capacity, K_{vr}, can be calculated for each incremental flowrate and valve differential pressure, by using Equation 6.5.2, which is derived from Equation 6.3.2.

The K_{vr} can be thought of as being the actual valve capacity required by the installation and, if plotted against the required flowrate, the resulting graph can be referred to as the 'installation curve'.

$$\dot{V} = K_v \sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Where:
$\dot{V}$ = Flowrate through the valve (m^3/h)
K_v = Valve K_{vr} (m^3/h bar)
ΔP = The differential pressure across the valve (bar)

Equation 6.3.2 is transposed into Equation 6.5.2 to solve for K_{vr}:

$$K_{vr} = \frac{\dot{V}}{\sqrt{\Delta P}} \qquad \textbf{Equation 6.5.2}$$

Where
K_{vr} = The actual valve capacity required by the installation
$\dot{V}$ = Flowrate through the valve (m^3/h)
ΔP = The differential pressure across the valve (bar)

At the full-load condition, from Table 6.5.2:

Required flow through the valve = 10 m^3/h

ΔP across the valve = 1.54 bar

From Equation 6.5.2: $K_{vr} = \dfrac{10}{\sqrt{1.54}}$

$$\mathbf{K_{vr} = 8.06\ m^3/h\ bar}$$

Taking the valve flowrate and valve ΔP from Table 6.5.2, a K_{vr} for each increment can be determined from Equation 6.5.2; and these are tabulated in Table 6.5.3.

Table 6.5.3 The relationship between flowrate, differential pressure (ΔP), and K_{vr}

Flow m^3/h	0*	1	2	3	4	5	6	7	8	9	10
Valve ΔP bar	5.58*	5.54	5.42	5.23	4.95	4.58	4.14	3.61	3.00	2.31	1.54
K_{vr} m^3/h bar	0*	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06

* Assumes the valve is fully shut and the pump produces maximum discharge pressure at no flow.

Constructing the installation curve

The K_{vr} of 8.06 satisfies the maximum flow condition of 10 m^3/h for this example.

The installation curve could be constructed by comparing flowrate to K_{vr}, but it is usually more convenient to view the installation curve in percentage terms. This simply means the percentage of K_{vr} to K_{vs}, or in other words, the percentage of actual pass area relative to the full open pass area.

For this example: The installation curve is constructed, by taking the ratio of K_{vr} at any load relative to the K_{vs} of 8.06. A valve with a K_{vs} of 8.06 would be 'perfectly sized', and would describe the installation curve, as tabulated in Table 6.5.4, and drawn in Figure 6.5.7. This installation curve can be thought of as the valve capacity of a perfectly sized valve for this example.

Table 6.5.4 Installation curve plotted by the valve K_{vs} equalling the full-load K_{vr}

Flow m^3/h	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06
Valve K_{vs}	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06
% K_{vr} / K_{vs} (Installation curve)	0	5.2	10.7	16.3	22.3	29.0	36.6	45.7	57.3	73.4	100

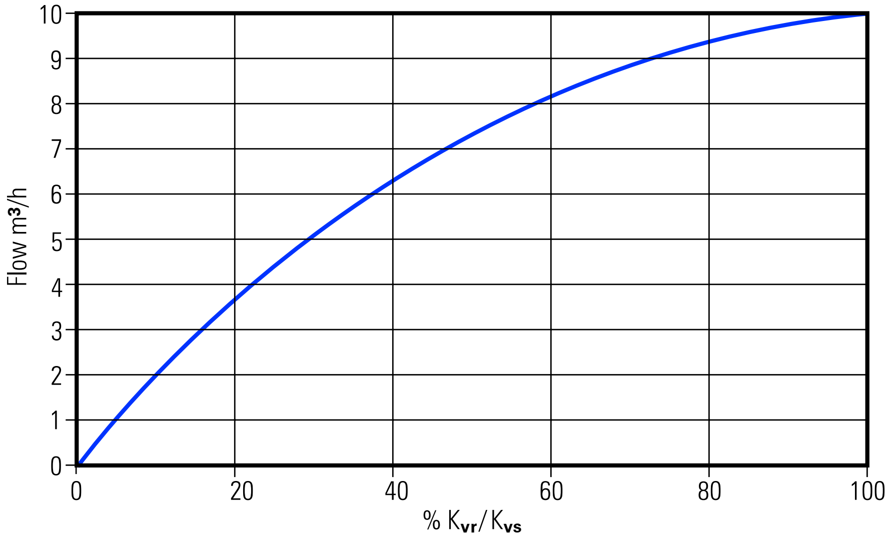


Fig. 6.5.7 The installation curve for Example 6.5.2

It can be seen that, as the valve is 'perfectly sized' for this installation, the maximum flowrate is satisfied when the valve is fully open.

However, it is unlikely and undesirable to select a perfectly sized valve. In practice, the selected valve would usually be at least one size larger, and therefore have a K_{vs} larger than the installation K_{vr}.

As a valve with a K_{vs} of 8.06 is not commercially available, the next larger standard valve would have a K_{vs} of 10 with nominal DN25 connections.

It is interesting to compare linear and equal percentage valves having a K_{vs} of 10 against the installation curve for this example.

Consider a valve with a linear inherent characteristic

A valve with a linear characteristic means that the relationship between valve lift and orifice pass area is linear. Therefore, both the pass area and valve lift at any flow condition is simply the K_{vr} expressed as a proportion of the valve K_{vs}. For example:

$$\text{Percentage valve lift} = \frac{K_{vr}}{K_{vs}} \times \frac{100}{1}$$

It can be seen from Table 6.5.4, that at the maximum flowrate of 10 m³/h, the K_{vr} is 8.06. If the linear valve has a K_{vs} of 10, for the valve to satisfy the required maximum flowrate, the valve will lift:

$$\frac{8.06}{10.0} \times \frac{100}{1} = 80.6\%$$

Using the same routine, the orifice size and valve lift required at various flowrates may be determined for the linear valve, as shown in Table 6.5.5.

Table 6.5.5 Pass area and valve lift for a linear valve with K_{vs} 10

Flow m³/h	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06
Valve K_{vs}	10	10	10	10	10	10	10	10	10	10	10
% Pass area	0	4.20	8.60	13.10	18.00	23.40	29.50	36.80	46.20	59.20	80.60
% Valve lift	0	4.20	8.60	13.10	18.00	23.40	29.50	36.80	46.20	59.20	80.60

An equal percentage valve will require exactly the same pass area to satisfy the same maximum flowrate, but its lift will be different to that of the linear valve.

Consider a valve with an equal percentage inherent characteristic

Given a valve rangeability of 50:1, $\tau = 50$, the lift (H) may be determined using Equation 6.5.1:

$$\dot{V} = \frac{e^x}{\tau}\,\dot{V}_{max}$$ **Equation 6.5.1**

Where:

$\dot{V}$ = Flow through the valve at lift H.

x = $(\ln \tau)\,H$

Note: 'In' is a mathematical function known as 'natural logarithm'.

τ = Valve rangeability (ratio of the maximum to minimum controllable flowrate, typically 50 for a globe type control valve)

H = Valve lift (0 = closed, 1 = fully open)

$\dot{V}_{max}$ = Maximum flow through the valve

Transposing from Equation 6.5.1: $e^x = \dfrac{\dot{V}\tau}{\dot{V}_{max}}$

By taking logarithms on both sides: $x = \ln\left[\dfrac{\dot{V}\tau}{\dot{V}_{max}}\right]$

As: $x = (\ln \tau)\,H$

$$(\ln \tau)\;H = \ln\left[\frac{\dot{V}\tau}{\dot{V}_{max}}\right]$$

$$H = \frac{\ln\left[\dfrac{\dot{V}\tau}{\dot{V}_{max}}\right]}{\ln \tau}$$

Percentage valve lift is denoted by Equation 6.5.3.

$$H\,\% = \frac{\ln\left[\dfrac{\dot{V}\tau}{\dot{V}_{max}}\right]}{\ln \tau} \times 100$$ **Equation 6.5.3**

As the volumetric flowrate through any valve is proportional to the orifice pass area, Equation 6.5.3 can be modified to give the equal percentage valve lift in terms of pass area and therefore K_v.

This is shown by Equation 6.5.4.

$$H\,\% = \frac{\ln\left[\dfrac{K_{vr}\,\tau}{K_{vs}}\right]}{\ln \tau} \times 100$$ **Equation 6.5.4**

As already calculated, the K_{vr} at the maximum flowrate of 10 m³/h is 8.06, and the K_{vs} of the DN25 valve is 10. By using Equation 6.5.4 the required valve lift at full-load is therefore:

$$H\,\% = \frac{\ln\left[\dfrac{8.06 \times 50}{10}\right]}{\ln 50} \times 100$$

$$H\,\% = \frac{\ln 40.3}{\ln 50} \times 100$$

$$H\,\% = \frac{3.696}{3.912} \times 100$$

$$\mathbf{H\,\% = 94.5\%}$$

Using the same routine, the valve lift required at various flowrates can be determined from Equation 6.5.4 and is shown in Table 6.5.6.

Table 6.5.6 Pass area and valve lift for the equal % valve with K_{vs} 10.

Flow m³/h	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06
Valve K_{vs}	10	10	10	10	10	10	10	10	10	10	10
% Pass area	0	4.2	8.6	13.1	18.00	23.4	29.5	36.8	46.2	59.2	80.6
% Valve lift	0	19.0	37.0	48.0	56.20	62.9	68.8	74.4	80.3	86.6	94.5

Comparing the linear and equal percentage valves for this application

The resulting application curve and valve curves for the application in Example 6.5.2 for both the linear and equal percentage inherent valve characteristics are shown in Figure 6.5.8.

Note that the equal percentage valve has a significantly higher lift than the linear valve to achieve the same flowrate. It is also interesting to see that, although each of these valves has a K_{vs} larger than a 'perfectly sized valve' (which would produce the installation curve), the equal percentage valve gives a significantly higher lift than the installation curve. In comparison, the linear valve always has a lower lift than the installation curve.

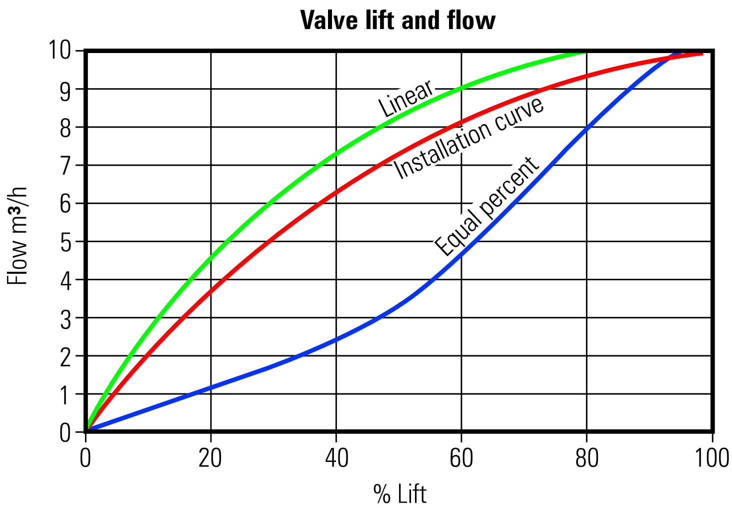


Fig. 6.5.8 Comparing linear and equal percent valve lift and the installation curve for Example 6.5.2

The rounded nature of the curve for the linear valve is due to the differential pressure falling across the valve as the flow increases. If the pump pressure had remained constant across the whole range of flowrates, the installation curve and the curve for the linear valve would both have been straight lines.

By observing the curve for the equal percentage valve, it can be seen that, although a linear relationship is not achieved throughout its whole travel, it is above 50% of the flowrate.

The equal percentage valve offers an advantage over the linear valve at low flowrates. Consider, at a 10% flowrate of 1 m³/h, the linear valve only lifts roughly 4%, whereas the equal percentage valve lifts roughly 20%. Although the orifice pass area of both valves will be exactly the same, the shape of the equal percentage valve plug means that it operates further away from its seat, reducing the risk of impact damage between the valve plug and seat due to quick reductions in load at low flowrates.

An oversized equal percentage valve will still give good control over its full range, whereas an oversized linear valve might perform less effectively by causing fast changes in flowrate for small changes in lift.

Conclusion - In most applications, an equal percentage valve will provide good results, and is very tolerant of over-sizing. It will offer a more constant gain as the load changes, helping to provide a more stable control loop at all times. However, it can be observed from Figure 6.5.8, that if the linear valve is properly sized, it will perform perfectly well in this type of water application.

3. Temperature control of a steam application with a two-port valve

In heat exchangers, which use steam as the primary heating agent, temperature control is achieved by varying the flow of steam through a two-port control valve to match the rate at which steam condenses on the heating surfaces. This varying steam flow varies the pressure (and hence temperature) of the steam in the heat exchanger and thus the rate of heat transfer.

Example 6.5.3

In a particular steam-to-water heat exchange process, it is proposed that:

- Water is heated from 10°C to a constant 60°C.
- The water flowrate varies between 0 and 10 L/s (kg/s).
- At full-load, steam is required at 4 bar a in the heat exchanger coils.
- The overall heat transfer coefficient (U) is 1 500 W/m²°C at full-load, and reduces by 4% for every 10% drop in secondary water flowrate.

Using this data, and by applying the correct equations, the following properties can be determined:

- The heat transfer area to satisfy the maximum load. Not until this is established can the following be found:
- The steam temperature at various heat loads.
- The steam pressure at various heat loads.
- The steam flowrate at various heat loads.

The heat transfer area must be capable of satisfying the maximum load.

At maximum load:

- Find the heat load.

Heat load is determined from Equation 2.6.5:

$$\dot{Q} = \dot{m}\, c_p\, \Delta T \qquad \textbf{Equation 2.6.5}$$

Where:
$\dot{Q}$ = Mean heat transfer rate (kW)
$\dot{m}$ = Mean seconday fluid flowrate (kg/s)
c_p = Specific heat capacity of water (4.19 kJ/kg°C)
ΔT = Temperature rise of the secondary fluid (°C)

$$\dot{\mathbf{Q}} = 10 \text{ kg/s} \times 4.19 \text{ kJ/kg°C} \times (60 - 10)\text{°C} = \mathbf{2\,095\ kW}$$

- Find the corresponding steam flowrate.

The steam flowrate may be calculated from Equation 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Heat load in kW x 3 600 s/h}}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

h_{fg} for steam at 4 bar a = 2 133.6 kJ/kg, consequently:

$$\textbf{Steam flowrate} = \frac{2\,095 \text{ kW x } 3\,600 \text{ s/h}}{2\,133.6 \text{ kJ/kg}} = \mathbf{3\,535\ kg/h}$$

- Find the heat transfer area required to satisfy the maximum load.

The heat transfer area (A) can be determined from Equation 2.5.3:

$$\dot{Q} = U\,A\,\Delta T_{LM} \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W (J/s))
U = Overall heat transfer coefficient (W/m² K or W/m²°C)
A = Heat transfer area (m²)
ΔT_{LM} = Log mean temperature difference (K or °C)

At this stage, ΔT_{LM} is unknown, but can be calculated from the primary steam and secondary water temperatures, using Equation 2.5.5.

- Find the log mean temperature difference.

ΔT_{LM} may be determined from Equation 2.5.5:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\dfrac{T_s - T_1}{T_s - T_2}\right)} \qquad \textbf{Equation 2.5.5}$$

Where:
T_1 = 10°C
T_2 = 60°C
T_s = Saturation temperature at 4 bar a = 143.6°C
ln = A mathematical function known as 'natural logarithm'

$$\Delta T_{LM} = \frac{(T_2 - T_1)}{\ln\left(\dfrac{T_s - T_1}{T_s - T_2}\right)}$$

$$\Delta T_{LM} = \frac{(60 - 10)}{\ln\left(\dfrac{143.6 - 10}{143.6 - 60}\right)}$$

$$\Delta T_{LM} = \frac{50}{\ln 1.598\,1}$$

$$\Delta T_{LM} = \frac{50}{0.469}$$

$$\mathbf{\Delta T_{LM} = 106.6°C}$$

- The heat transfer area must satisfy the maximum design load, consequently from Equation 2.5.3:

$$\dot{Q} = U\,A\,\Delta T_{LM} \qquad \textbf{Equation 2.5.3}$$

$$2\,095\text{ kW}\left[1\,000\ \frac{\text{W}}{\text{kW}}\right] = 1\,500\text{ W/m}^2\text{°C x Area (m}^2\text{) x }106.6\text{°C}$$

The heat transfer (A) = 13.1 m²

Find the conditions at other heat loads at a 10% reduced water flowrate:

- Find the heat load.

If the water flowrate falls by 10% to 9 kg/s, the heat load reduces to:

$\dot{Q}$ = 9 kg/s x (60 – 10°C) x 4.19 kJ/kg °C = **1 885.5 kW**

The initial 'U' value of 1 500 W/m²°C is reduced by 4%, so the temperature required in the steam space may be calculated from Equation 2.5.3:

$$\dot{Q} = U\,A\,\Delta T_{LM} \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = 1 885.5 kW
U = 1500 kW/m² °C x 0.96 (representing the 4% decrease in U value)
A = 13.1 m²

$$1\,885.5\text{ kW}\left[1000\ \frac{\text{W}}{\text{kW}}\right] = 1\,500\text{ W/m}^2\,{}^\circ\text{C} \times 0.96 \times 13.1\text{ m}^2 \times \Delta T_{LM}$$

$$\boldsymbol{\Delta T_{LM} = 100^\circ C}$$

- Find the steam temperature at this reduced load.

If ΔT_{LM} = 100°C, and T_1, T_2 are already known, then T_s may be determined from Equation 2.5.5:

$$\Delta T_{LM} = \frac{(T_2 - T_1)}{\ln\left(\dfrac{T_s - T_1}{T_s - T_2}\right)} \qquad \textbf{Equation 2.5.5}$$

$$100 = \frac{(60 - 10)}{\ln\left(\dfrac{T_s - 10}{T_s - 60}\right)}$$

$$\ln\left(\frac{T_s - 10}{T_s - 60}\right) = 0.5$$

By taking antilogs on either side:

$$\left(\frac{T_s - 10}{T_s - 60}\right) = e^{0.5}$$

$$\left(\frac{T_s - 10}{T_s - 60}\right) = 1.65$$

$$T_s - 10 = 1.65 \times (Ts - 60)$$

$$\boldsymbol{T_s = 137^\circ C}$$

- Find the steam flowrate.

The saturated steam pressure for 137°C is 3.32 bar a (from the Spirax Sarco steam tables).

At 3.32 bar a, h_{fg} = 2 153.5 kJ/kg, consequently from Equation 2.8.1:

$$\textbf{Steam flowrate} = \frac{1\,885.5\text{ kW} \times 3\,600\text{ s/h}}{2\,153.5\text{ kJ/kg}} = \textbf{3 152 kg/h}$$

Using this routine, a set of values may be determined over the operating range of the heat exchanger, as shown in Table 6.5.7.

Table 6.5.7 The heat transfer, steam pressure in the coil, and steam flowrate

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
Energy (kW)	0	210	419	629	838	1 048	1 257	1 467	1 676	1 886	2 095
Steam Pressure (bar a)	0	0.22	0.27	0.37	0.54	0.81	1.19	1.71	2.42	3.35	4.0
Steam flowrate (kg/h)	0	321	644	974	1312	1 659	2 016	2 383	2 762	3 152	3 535

If the steam pressure supplying the control valve is given as 5.0 bar a, and using the steam pressure and steam flowrate information from Table 6.5.7; the K_{vr} can be calculated from Equation 6.5.6, which is derived from the steam flow formula, Equation 3.21.2.

$$\dot{m} = 12 \times K_v \times P_1 \times \sqrt{1 - 5.67\,(0.42 - x)^2} \qquad \textbf{Equation 3.21.2}$$

Where:

$\dot{m}$ = Mass flowrate (kg/h)

K_v = Valve flow coefficient (m³/h. bar)

P_1 = Upstream pressure (bar a)

X = Pressure drop ratio = $\dfrac{P_1 - P_2}{P_1}$

P_2 = Downstream pressure (bar a)

Equation 3.21.2 is transposed to give Equation 6.5.5.

$$K_{vr} = \frac{\dot{m}}{12 \times P_1 \times \sqrt{1 - 5.67\,(0.42 - x)^2}} \qquad \textbf{Equation 6.5.5}$$

Known information at full-load includes:

$\dot{m}$ = 3 535 kg/h

P_1 = 5 bar a

P_2 = 4 bar a

$$x = \frac{P_1 - P_2}{P_2}$$

$$x = \frac{5 - 4}{5}$$

$$\mathbf{x = 0.2}$$

$$\text{Full-load } K_{vr} = \frac{3\,535}{12 \times 5 \times \sqrt{1 - 5.67\,(0.42 - 0.2)^2}}$$

$$K_{vr} = \frac{3\,535}{60 \times \sqrt{0.726}}$$

Full - load K_{vr} = 69.2 (to one decimal point)

Using this routine, the K_{vr} for each increment of flow can be determined, as shown in Table 6.5.8.

The installation curve can also be defined by considering the K_{vr} at all loads against the 'perfectly sized' K_{vs} of 69.2.

Table 6.5.8

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0.0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.2
Valve K_{vs}	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2
% Installation curve	0.0	7.7	15.5	23.4	31.6	39.9	48.6	57.4	66.5	77.7	100

The K_{vr} of 69.2 satisfies the maximum secondary flow of 10 kg/s.

In the same way as in Example 6.5.2, the installation curve is described by taking the ratio of K_{vr} at any load relative to a K_{vs} of 69.2.

Such a valve would be 'perfectly sized' for the example, and would describe the installation curve, as tabulated in Table 6.5.8, and drawn in Figure 6.5.9.

The installation curve can be thought of as the valve capacity of a valve perfectly sized to match the application requirement.

It can be seen that, as the valve with a K_{vs} of 69.2 is 'perfectly sized' for this application, the maximum flowrate is satisfied when the valve is fully open.

However, as in the water valve sizing Example 6.5.2, it is undesirable to select a perfectly sized valve. In practice, it would always be the case that the selected valve would be at least one size larger than that required, and therefore have a K_{vs} larger than the application K_{vr}.

A valve with a K_{vs} of 69.2 is not commercially available, and the next larger standard valve has a K_{vs} of 100 with nominal DN80 connections.

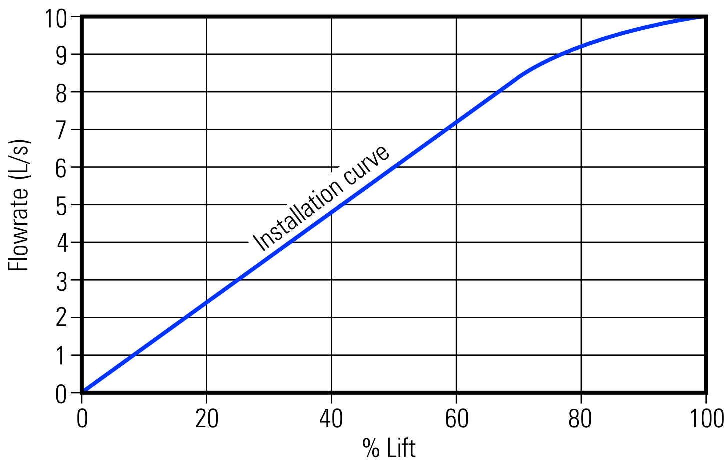


Fig. 6.5.9 The installation curve for Example 6.5.3

It is interesting to compare linear and equal percentage valves having a K_{vs} of 100 against the installation curve for this example.

Consider a valve with a linear inherent characteristic

A valve with a linear characteristic means that the relationship between valve lift and orifice pass area is linear. Therefore, both the pass area and valve lift at any flow condition is simply the K_{vr} expressed as a proportion of the valve K_{vs}. For example.

$$\text{Percentage valve lift} = \frac{K_{vr}}{K_{vs}} \times \frac{100}{1}$$

At the maximum water flowrate of 10 kg/s, the steam valve K_{vr} is 69.2. The K_{vs} of the selected valve is 100, consequently the lift is:

$$\frac{69.2}{100} \times 100 = 69.2\%$$

Using the same procedure, the linear valve lifts can be determined for a range of flows, and are tabulated in Table 6.5.9.

Table 6.5.9 Comparing valve lifts (K_{vs} 100) the K_{vr}, and the installation curve

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.2
Valve K_{vs}	100	100	100	100	100	100	100	100	100	100	100
% Lift - Linear valve	0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.2
% Lift - Equal percentage valve	0	25.1	43.0	53.5	61.1	67.1	72.1	76.4	80.2	84.2	90.6
% installation curve*	0	7.7	15.5	23.5	31.6	40.0	48.6	57.4	66.5	77.8	100.0

* The installation curve is the percentage of K_{vr} at any load to the K_{vr} at maximum load

Consider a valve with an equal percentage inherent characteristic

An equal percentage valve will require exactly the same pass area to satisfy the same maximum flowrate, but its lift will be different to that of the linear valve.

Given that the valve turndown ratio, $\tau = 50$, the lift (H) may be determined using Equation 6.5.4.

$$H\,\% = \frac{\ln\left[\frac{K_{vr}\,\tau}{K_{vs}}\right]}{\ln \tau} \times 100 \qquad \textbf{Equation 6.5.4}$$

For example, at the maximum water flowrate of 10 kg/s, the K_{vr} is 69.2. The K_{vs} of the selected valve is 100, consequently the lift is:

$$H\,\% = \frac{\ln\left[\frac{69.2 \times 50}{100}\right]}{\ln 50} \times 100$$

$$H\,\% = \frac{\ln 34.6}{\ln 50} \times 100$$

$$H\,\% = \frac{3.544}{3.912} \times 100$$

$$H\,\% = 90.6\%$$

Using the same procedure, the percentage valve lift can be determined from Equation 6.5.4 for a range of flows for this installation.

The corresponding lifts for linear and equal percentage valves are shown in Table 6.5.9 along with the installation curve.

As in Example 6.5.2, the equal percentage valve requires a much higher lift than the linear valve to achieve the same flowrate. The results are graphed in Figure 6.5.10.

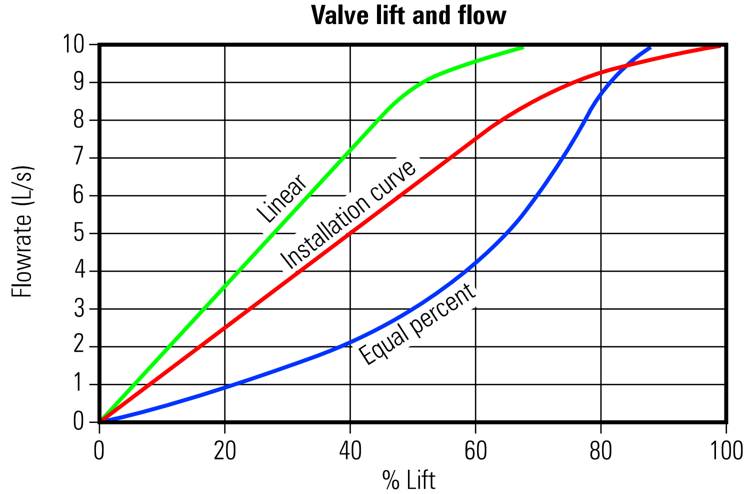


Fig. 6.5.10 Comparing linear and equal % valve lift and the installation curve for Example 6.5.3

There is a sudden change in the shape of the graphs at roughly 90% of the load; this is due to the effect of critical pressure drop across the control valve which occurs at this point.

Above 86% load in this example, it can be shown that the steam pressure in the heat exchanger is above 2.9 bar a which, with 5 bar a feeding the control valve, is the critical pressure value. (For more information on critical pressure, refer to Module 6.4, Control valve sizing for steam).

It is generally agreed that control valves find it difficult to control below 10% of their range, and in practice, it is usual for them to operate between 20% and 80% of their range.

The graphs in Figure 6.5.10 refer to linear and equal percentage valves having a K_{vs} of 100, which are the next larger standard valves with suitable capacity above the application curve (the required K_{vr} of 69.2), and would normally be chosen for this particular example.

The effect of a control valve which is larger than necessary

It is worth while considering what effect the next larger of the linear or equal percentage valves would have had if selected. To accommodate the same steam loads, each of these valves would have had lower lifts than those observed in Figure 6.5.10.

The next larger standard valves have a K_{vs} of 160. It is worth noting how these valves would perform should they have been selected, and as shown in Table 6.5.10 and Figure 6.5.11.

Table 6.5.10 Comparing valve lifts (K_{vs} 160) the K_{vr} and the installation curve

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.0
Valve K_{vs}	160	160	160	160	160	160	160	160	160	160	160
% Lift - Linear valve	0	3.3	6.7	10.1	13.7	17.3	21.0	24.8	28.8	33.6	43.0
% Lift - Equal percentage valve	0	13.1	30.9	41.5	49.1	55.1	60.1	64.4	68.2	72.1	78.0
% Installation curve*	0	7.7	15.5	23.5	31.6	40.0	48.6	57.4	66.5	77.8	100

* The installation curve is the percentage of K_{vr} at any load to the K_{vr} at maximum load

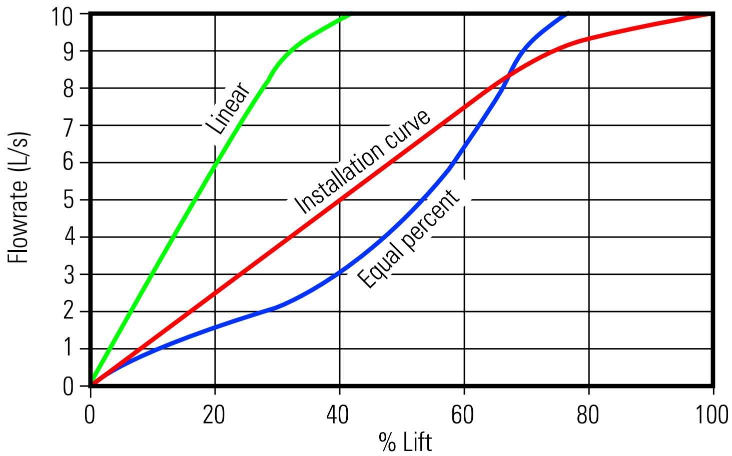


Fig. 6.5.11 Percentage valve lift required for equal percentage and linear valves in Example 6.5.3 with K_{vs} 160

It can be seen from Figure 6.5.11 that both valve curves have moved to the left when compared to the smaller (properly sized) valves in Figure 6.5.10, whilst the installation curve remains static.

The change for the linear valve is quite dramatic; it can be seen that, at 30% load, the valve is only 10% open. Even at 85% load, the valve is only 30% open. It may also be observed that the change in flowrate is large for a relatively small change in the lift. This effectively means that the valve is operating as a fast acting valve for up to 90% of its range. This is not the best type of inherent characteristic for this type of steam installation, as it is usually better for changes in steam flow to occur fairly slowly.

Although the equal percentage valve curve has moved position, it is still to the right of the installation curve and able to provide good control. The lower part of its curve is relatively shallow, offering slower opening during its initial travel, and is better for controlling steam flow than the linear valve in this case.

Circumstances that can lead to over-sizing include:

- The application data is approximate, consequently an additional 'safety factor' is included.
- Sizing routines that include operational 'factors' such as an over-zealous allowance for fouling.
- The calculated K_{vr} is only slightly higher than the K_{vs} of a standard valve, and the next larger size has to be selected.

There are also situations where:

- The available pressure drop over the control valve at full-load is low.

For example, if the steam supply pressure is 4.5 bar a and the steam pressure required in the heat exchanger at full-load is 4 bar a, this only gives an 11% pressure drop at full-load.

- The minimum load is a lot less than the maximum load.

A linear valve characteristic would mean that the valve plug operates close to the seat, with the possibility of damage.

In these common circumstances, the equal percentage valve characteristic will provide a much more flexible and practical solution.

This is why most control valve manufacturers will recommend an equal percentage characteristic for two-port control valves, especially when used on compressible fluids such as steam.

Please note: Given the opportunity, it is better to size steam valves with as high a pressure drop as possible at maximum load; even with critical pressure drop occurring across the control valve if the conditions allow. This helps to reduce the size and cost of the control valve, gives a more linear installation curve, and offers an opportunity to select a linear valve.

However, conditions may not allow this. The valve can only be sized on the application conditions. For example, should the heat exchanger working pressure be 4.5 bar a, and the maximum available steam pressure is only 5 bar a, the valve can only be sized on a 10% pressure drop ([5 – 4.5] / 5). In this situation, sizing the valve on critical pressure drop would have reduced the size of the control valve and starved the heat exchanger of steam.

If it were impossible to increase the steam supply pressure, a solution would be to install a heat exchanger that operates at a lower operating pressure. In this way, the pressure drop would increase across the control valve. This could result in a smaller valve but also a larger heat exchanger, because the heat exchanger operating temperature is now lower.

Another set of advantages accrues from larger heat exchangers operating at lower steam pressures:

- There is less propensity for scaling and fouling on the heating surfaces.
- There is less flash steam produced in the condensate system.
- There is less backpressure in the condensate system.

A balance has to be made between the cost of the control valve and heat exchanger, the ability of the valve to control properly, and the effects on the rest of the system as seen above. On steam systems, equal percentage valves will usually be a better choice than linear valves, because if low pressure drops occur, they will have less of an affect on their performance over the complete range of valve movement.

Questions

1. An equal percentage valve has a certain orifice pass area. For the same flowrate and differential pressure what would be the pass area of a linear valve?

a| More than the equal percentage valve ☐

b| Less than the equal percentage valve ☐

c| Almost the same as the equal percentage valve ☐

d| Exactly the same as the equal percentage valve ☐

2. **An equal percentage valve has a certain orifice pass area. For the same flowrate and differential pressure what would be the lift of a linear valve?**

a| More than the equal percentage valve ☐

b| Less than the equal percentage valve ☐

c| Almost the same as the equal percentage valve ☐

d| Exactly the same as the equal percentage valve ☐

3. **A linear valve with K_{vs} 4 and rangeability 50 passes 10 m^3/h of water when fully open. What will be the percentage orifice pass area, the K_{vr}, and the valve lift with a flow of 5 m^3/h with the same differential pressure across the same valve?**

a| Pass area 50%; K_{vr} 2; lift 50% ☐

b| Pass area 40%; K_{vr} 2; lift 40% ☐

c| Pass area 60%; K_{vr} 2; lift 60% ☐

d| Pass area 50%; K_{vr} 1; lift 50% ☐

4. **An equal percentage valve with K_{vs} 4 and rangeability 50 passes 10 m^3/h of water when fully open. What will be the percentage orifice pass area, the K_{vr}, and the valve lift with a flow of 5 m^3/h with the same differential pressure across the same valve?**

a| Pass area 50%; K_{vr} 2; lift 82.3% ☐

b| Pass area 40%; K_{vr} 3.29; lift 41.1% ☐

c| Pass area 60%; K_{vr} 2; lift 60% ☐

d| Pass area 82.3%; K_{vr} 2; lift 82.3% ☐

5. What is the effect on the control performance of a linear valve when it is oversized?

a| None ☐

b| The valve tends to control better ☐

c| The valve will tend to act as a fast opening valve ☐

d| The valve will tend to act as a slow opening valve ☐

6. What is the effect on the control performance of an equal percentage valve when it is oversized?

a| None ☐

b| The valve tends to control better ☐

c| The valve will tend to act as a slow opening valve ☐

d| The valve is still likely to perform with a reasonable degree of control ☐

Answers

1: d, 2: b, 3: a, 4: a, 5: c, 6: d

Module 6.6

Control Valve Actuators and Positioners

Actuators

In Block 5, 'Basic Control Theory', an analogy was used to describe simple process control:

- The arm muscle and hand (the actuator) turned the valve - (the controlled device).

One form of controlling device, the control valve, has now been covered. The actuator is the next logical area of interest.

The operation of a control valve involves positioning its movable part (the plug, ball or vane) relative to the stationary seat of the valve. The purpose of the valve actuator is to accurately locate the valve plug in a position dictated by the control signal.

The actuator accepts a signal from the control system and, in response, moves the valve to a fully-open or fully-closed position, or a more open or a more closed position (depending on whether 'on/off' or 'continuous' control action is used).

There are several ways of providing this actuation. This Module will concentrate on the two major ones:

- Pneumatic
- Electric.

Other significant actuators include the hydraulic and the direct acting types. These are discussed in Block 7, 'Control Hardware: Self-Acting Actuation'.

Pneumatic actuators – operation and options

Pneumatic actuators are commonly used to actuate control valves and are available in two main forms; piston actuators (Figure 6.6.1) and diaphragm actuators (Figure 6.6.2)

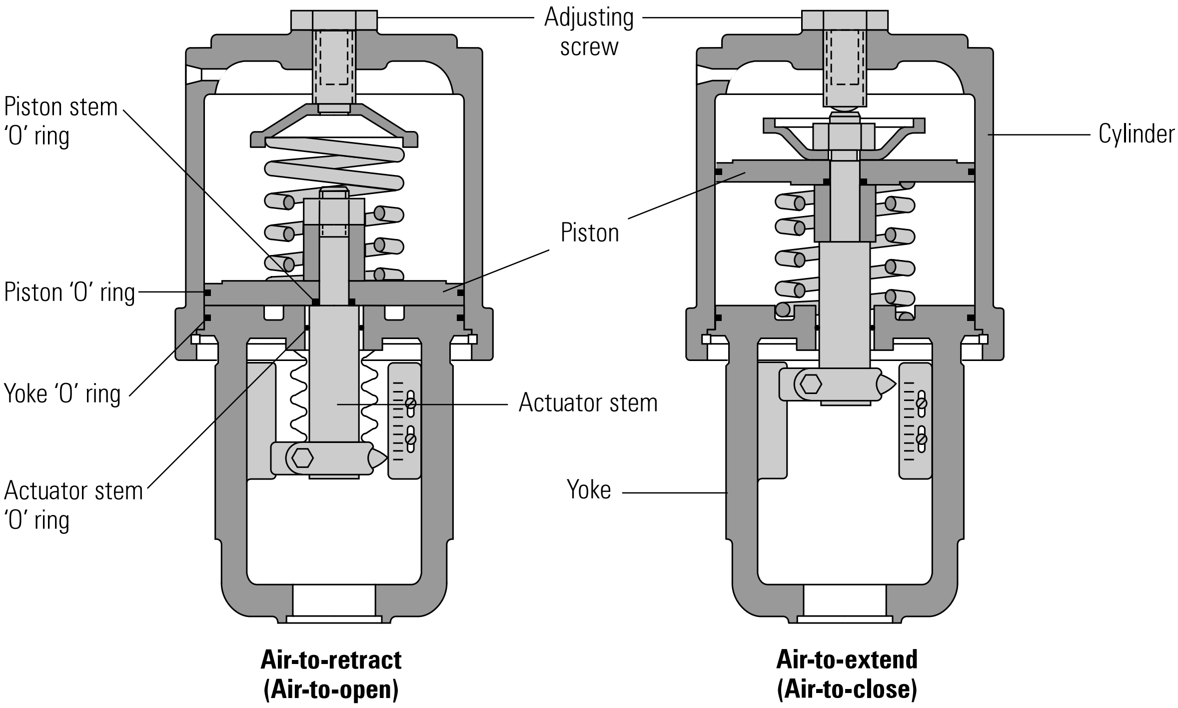


Fig. 6.6.1 Typical piston actuators

Piston actuators

Piston actuators are generally used where the stroke of a diaphragm actuator would be too short or the thrust is too small. The compressed air is applied to a solid piston contained within a solid cylinder. Piston actuators can be single acting or double acting, can withstand higher input pressures and can offer smaller cylinder volumes, which can act at high speed.

Diaphragm actuators

Diaphragm actuators have compressed air applied to a flexible membrane called the diaphragm. Figure 6.6.2 shows a rolling diaphragm where the effective diaphragm area is virtually constant throughout the actuator stroke. These types of actuators are single acting, in that air is only supplied to one side of the diaphragm, and they can be either direct acting (spring-to-retract) or reverse acting (spring-to-extend).

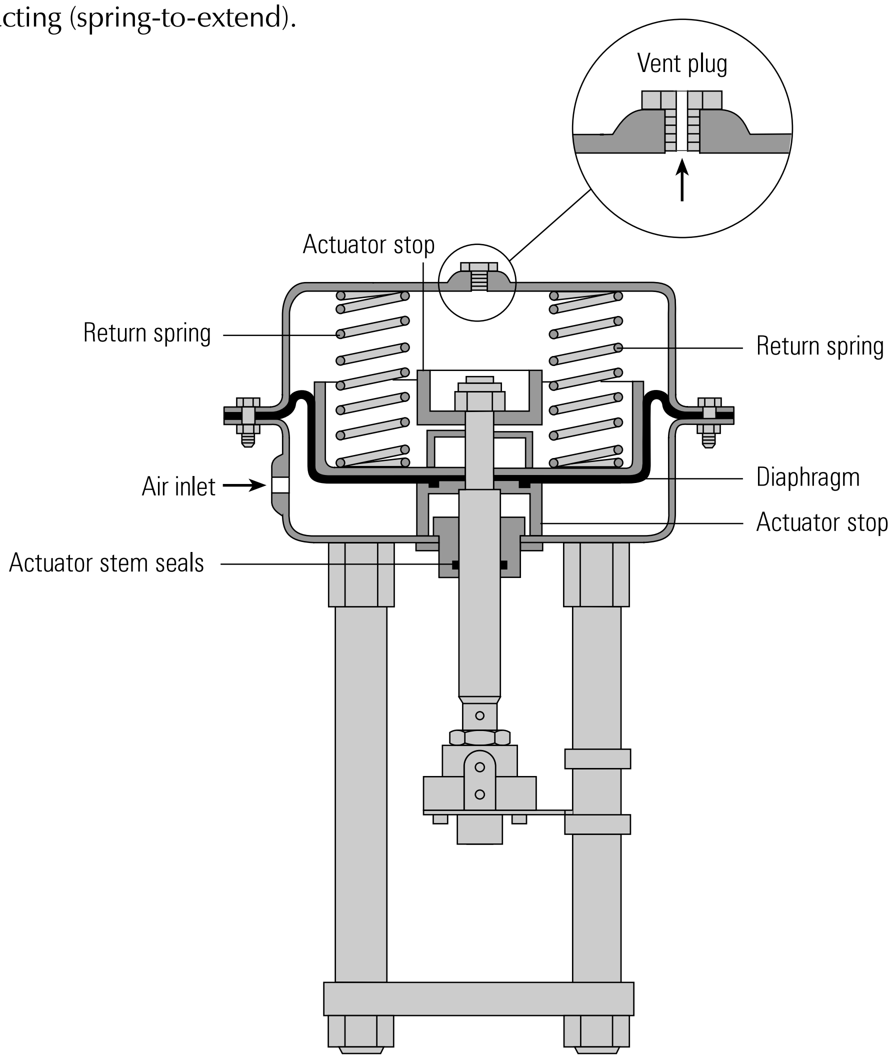


Fig. 6.6.2 A pneumatic diaphragm actuator

Reverse acting (spring-to-extend)

The operating force is derived from compressed air pressure, which is applied to a flexible diaphragm. The actuator is designed so that the force resulting from the air pressure, multiplied by the area of the diaphragm, overcomes the force exerted (in the opposite direction) by the spring(s).

The diaphragm (Figure 6.6.2) is pushed upwards, pulling the spindle up, and if the spindle is connected to a direct acting valve, the plug is opened. The actuator is designed so that with a specific change of air pressure, the spindle will move sufficiently to move the valve through its complete stroke from fully-closed to fully-open.

As the air pressure decreases, the spring(s) moves the spindle in the opposite direction. The range of air pressure is equal to the stated actuator spring rating, for example 0.2 - 1 bar.

With a larger valve and/or a higher differential pressure to work against, more force is needed to obtain full valve movement.

To create more force, a larger diaphragm area or higher spring range is needed. This is why controls manufacturers offer a range of pneumatic actuators to match a range of valves – comprising increasing diaphragm areas, and a choice of spring ranges to create different forces.

The diagrams in Figure 6.6.3 show the components of a basic pneumatic actuator and the direction of spindle movement with increasing air pressure.

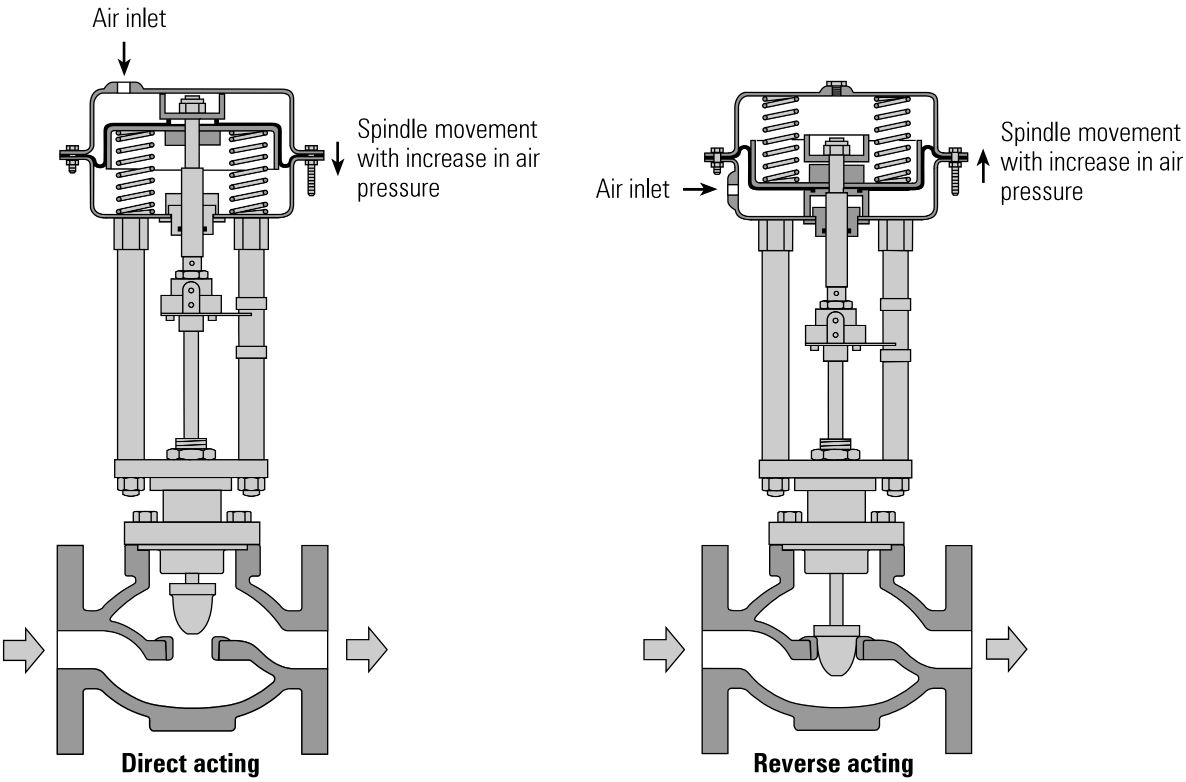


Fig. 6.6.3 Valve and actuator configurations

Direct acting actuator (spring-to-retract)

The direct acting actuator is designed with the spring below the diaphragm, having air supplied to the space above the diaphragm. The result, with increasing air pressure, is spindle movement in the opposite direction to the reverse acting actuator.

The effect of this movement on the valve opening depends on the design and type of valve used, and is illustrated in Figure 6.6.3. There is however, an alternative, which is shown in Figure 6.6.4. A direct acting pneumatic actuator is coupled to a control valve with a reverse acting plug (sometimes called a 'hanging plug').

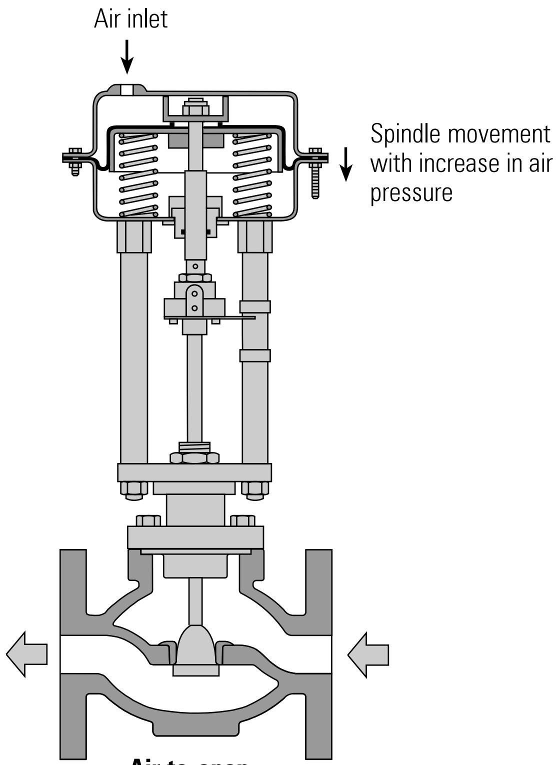


Fig. 6.6.4 Direct acting actuator and reverse acting control valve

The choice between direct acting and reverse acting pneumatic controls depends on what position the valve should revert to in the event of failure of the compressed air supply. Should the valve close or be wide-open? This choice depends upon the nature of the application and safety requirements. It makes sense for steam valves to close on air failure, and cooling valves to open on air failure. The combination of actuator and valve type must be considered. Figure 6.6.5 and Figure 6.6.6 show the net effect of the various combinations.

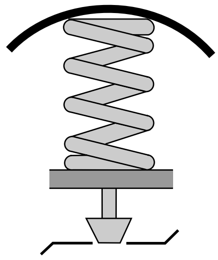
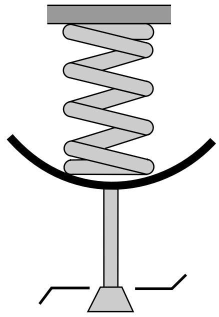
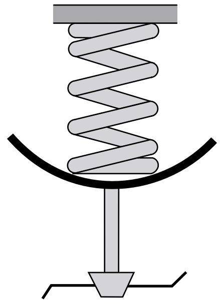
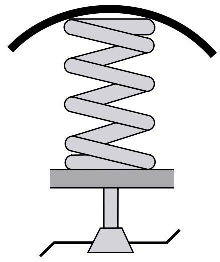
Two port valves				
Actuator action	Direct	Reverse	Reverse	Direct
Valve action	Direct	Reverse	Direct	Reverse
On air failure	Valve opens		Valve closes	

Fig. 6.6.5 Net effect of various combinations for two port valves

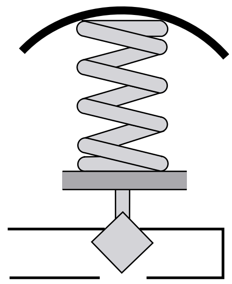
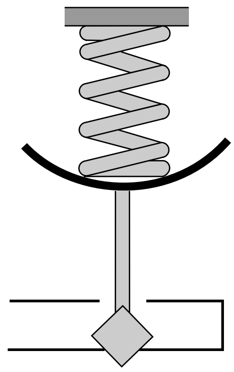
Three port valves (typical mixing valve depicted)		
Actuator action	Direct	Reverse
On air failure	Top seat closes bottom seat opens	Bottom seat closes top seat opens

Fig. 6.6.6 Net effect of the two combinations for three port valves

Effect of differential pressure on the valve lift

The air fed into the diaphragm chamber is the control signal from the pneumatic controller. The most widely used signal air pressure is 0.2 bar to 1 bar. Consider a reverse acting actuator (spring to extend) with standard 0.2 to 1.0 bar spring(s), fitted to a direct acting valve (Figure 6.6.7).

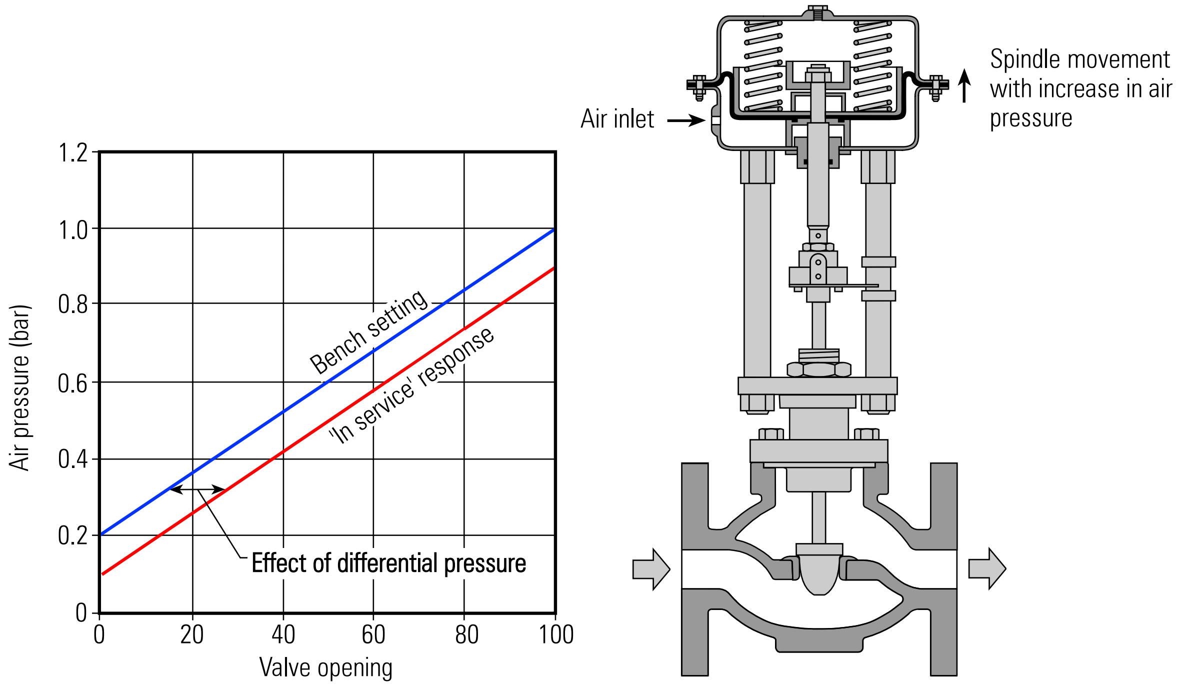


Fig. 6.6.7 Reverse acting actuator, air-to-open, direct acting valve - normally closed

When the valve and actuator assembly is calibrated (or 'bench set'), it is adjusted so that an air pressure of 0.2 bar will just begin to overcome the resistance of the springs and move the valve plug away from its seat.

As the air pressure is increased, the valve plug moves progressively further away from its seat, until finally at 1 bar air pressure, the valve is 100% open. This is shown graphically in Figure 6.6.7.

Now consider this assembly installed in a pipeline in a pressure reducing application, with 10 bar g on the upstream side and controlling the downstream pressure to 4 bar g.

The differential pressure across the valve is 10 - 4 = 6 bar. This pressure is acting on the underside of the valve plug, providing a force tending to open the valve. This force is in addition to the force provided by the air pressure in the actuator.

Therefore, if the actuator is supplied with air at 0.6 bar (halfway between 0.2 and 1 bar), for example, instead of the valve taking up the expected 50% open position, the actual opening will be greater, because of the extra force provided by the differential pressure.

Also, this additional force means that the valve is **not closed** at 0.2 bar. In order to close the valve in this example, the control signal must be reduced to approximately 0.1 bar.

The situation is slightly different with a steam valve controlling temperature in a heat exchanger, as the differential pressure across the valve will vary between:

- A minimum, when the process is calling for maximum heat, and the control valve is 100% open.
- A maximum, when the process is up to temperature and the control valve is closed.

The steam pressure in the heat exchanger increases as the heat load increases. This can be seen in Module 6.5, Example 6.5.3 and Table 6.5.7.

If the pressure upstream of the control valve remains constant, then, as the steam pressure rises in the heat exchanger, the differential pressure across the valve must decrease.

Figure 6.6.8 shows the situation with the air applied to a direct acting actuator. In this case, the force on the valve plug created by the differential pressure works against the air pressure. The effect is that if the actuator is supplied with air at 0.6 bar, for example, instead of the valve taking up the expected 50% open position, the percentage opening will be greater because of the extra force provided by the differential pressure. In this case, the control signal has to be increased to approximately 1.1. bar to fully close the valve.

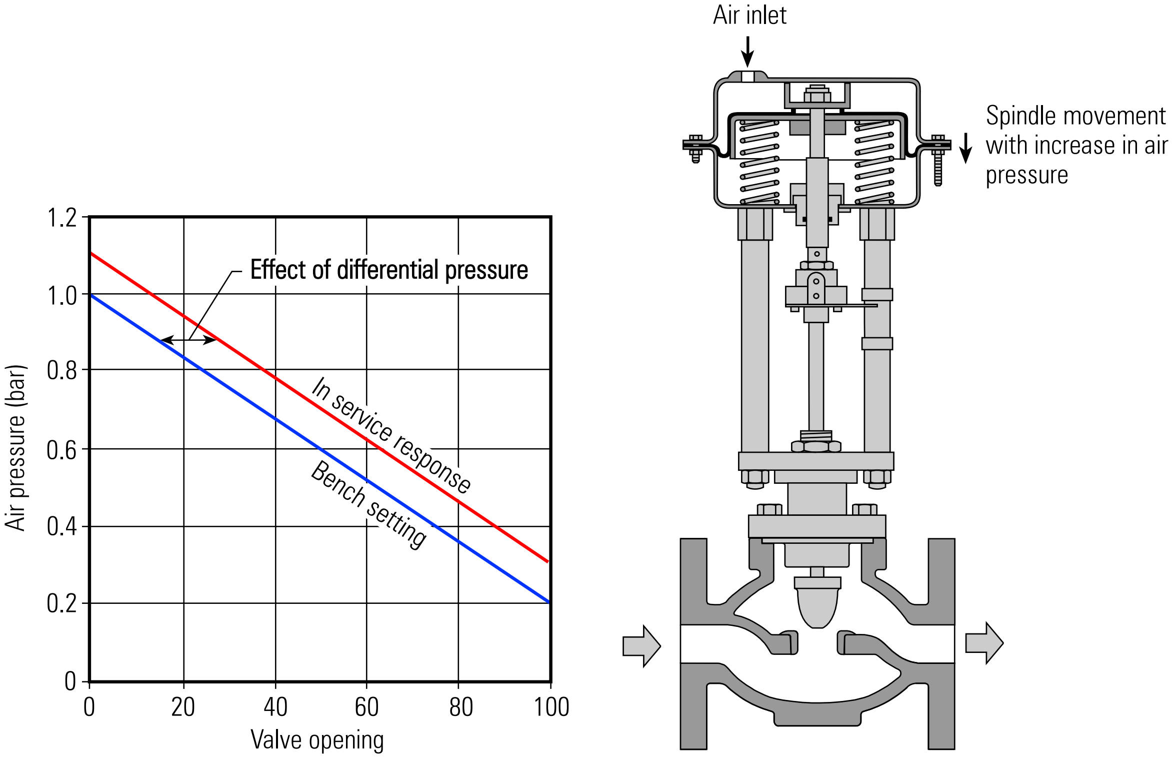


Fig. 6.6.8 Direct acting actuator, air-to-close, direct acting valve - normally open

It may be possible to recalibrate the valve and actuator to take the forces created by differential pressure into account, or perhaps using different springs, air pressure and actuator combinations. This approach can provide an economic solution on small valves, with low differential pressures and where precise control is not required. However, the practicalities are that:

- Larger valves have greater areas for the differential pressure to act over, thus increasing the forces generated, and having an increasing effect on valve position.
- Higher differential pressures mean that higher forces are generated.
- Valves and actuators create friction, causing hysteresis. Smaller valves are likely to have greater friction relative to the total forces involved.

The solution is to fit a positioner to the valve/actuator assembly. (More information is given on positioners later in this Module).

Note: For simplicity, the above examples assume a positioner is not used, and hysteresis is zero. The formulae used to determine the thrust available to hold a valve on its seat for various valve and actuator combinations are shown in Figure 6.6.9.

Where:

A = Effective area of diaphragm

P_{max} = Maximum pressure to actuator (normally 1.2 bar)

S_{max} = Maximum bench setting of spring

P_{min} = Minimum pressure to actuator (normally 0 bar)

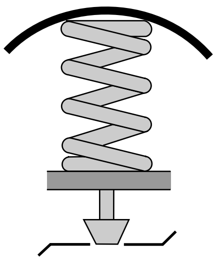
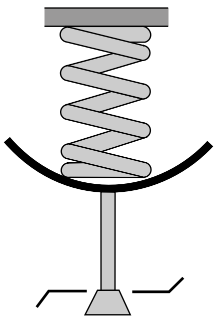
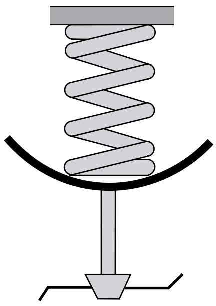
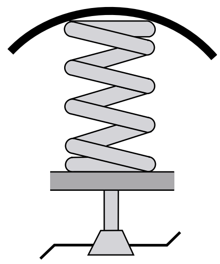
S_{min} = Minimum bench setting of spring

The thrust available to close the valve has to provide three functions:

1. To overcome the fluid differential pressure at the closed position.
2. To overcome friction in the valve and actuator, primarily at the valve and actuator stem seals.
3. To provide a sealing load between the valve plug and valve seat to ensure the required degree of tightness.

Control valve manufacturers will normally provide full details of the maximum differential pressures against which their various valve and actuator/spring combinations will operate; the Table in Figure 6.6.10 is an example of this data.

Note: When using a positioner, it is necessary to refer to the manufacturer's literature for the minimum and maximum air pressures.

Two port valves				
Actuator action	Direct	Reverse	Reverse	Direct
Valve action	Direct	Reverse	Direct	Reverse
Thrust available to close valve	A (P_{max} - S_{max})		A (P_{min} - S_{min})	

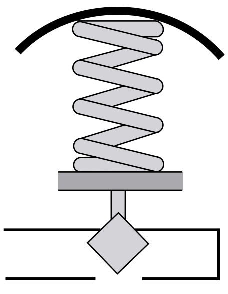
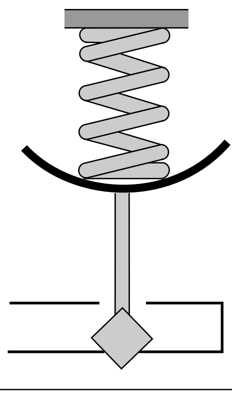
Three port valves (typical mixing valve depicted)		
Actuator action	Direct	Reverse
Thrust available against top seat	A (P_{min} - S_{min})	A (P_{min} - S_{min})
Thrust available against bottom seat	A (P_{max} - S_{max})	A (P_{min} - S_{min})

Fig. 6.6.9 Two and three port formulae

KE and LE valves

Valve size		DN15	DN20	DN25	DN32	DN40	DN50	DN65	DN80	DN100
Actuator	Spring range	Maximum differential pressure (bar)								
PN5123	2.0 to 4.0	40.0	40.0	30.5	14.9	10.3	5.5	-	-	-
PN5126	1.0 to 2.0	34.2	16.1	8.2	3.2	1.1	-	-	-	-
PN5120	0.2 to 1.0	7.7	4.9	-	-	-	-	-	-	-
	0.4 to 1.2	17.6	10.1	4.4	-	-	-	-	-	-
PN5220	0.2 to 1.0	21.3	12.1	5.6	2.2	1.8	0.7	-	-	-
	0.4 to 1.2	40.0	24.6	13.4	6.1	4.5	2.2	-	-	-
PN5226	1.0 to 2.0	40.0	40.0	31.1	14.7	8.0	4.4	-	-	-
PN5223	2.0 to 4.0	40.0	40.0	40.0	38.0	25.6	14.1	-	-	-
PN5320	0.2 to 1.0	34.4	19.1	10.0	4.4	3.3	1.6	-	-	
	0.4 to 1.2	40.0	32.6	22.1	10.6	7.5	3.9	-	-	-
PN5326	1.0 to 2.0	40.0	40.0	40.0	24.0	13.6	7.9	-	-	-
PN5323	2.0 to 4.0	40.0	40.0	40.0	40.0	30.0	22.3	-	-	-
PN5330	0.4 to 1.2	-	-	-	-	-	-	0.7	-	-
PN5336	1.0 to 2.0	-	-	-	-	-	-	4.0	2.3	1.2
PN5333	2.0 to 4.0	-	-	-	-	-	-	11.7	7.4	4.6
PN5420	0.2 to 1.0	40.0	31.3	17.5	8.3	5.9	3.0	-	-	-
	0.4 to 1.2	40.0	40.0	37.2	18.4	12.6	6.8	-	-	-
PN5426	1.0 to 2.0	40.0	40.0	40.0	38.5	22.4	13.3	-	-	-
PN5423	2.0 to 4.0	40.0	40.0	40.0	40.0	30.0	30.0	-	-	-
PN5430	0.4 to 1.2	-	-	-	-	-	-	2.5	1.3	0.6
PN5436	1.0 to 2.0	-	-	-	-	-	-	7.3	4.5	2.6
PN5433	2.0 to 4.0	-	-	-	-	-	-	20.2	13.1	8.3
PN5520	0.2 to 1.0	40.0	40.0	34.0	16.0	11.5	5.6	-	-	-
	0.4 to 1.2	40.0	40.0	40.0	36.0	24.2	13.0	-	-	-
PN5524	0.8 to 1.5	40.0	40.0	40.0	40.0	30.0	27.0	-	-	-
PN5530	0.2 to 1.0	-	-	-	-	-	-	3.8	2.6	1.6
	0.4 to 1.2	-	-	-	-	-	-	7.9	5.2	3.3
PN5534	0.8 to 1.5	-	-	-	-	-	-	15.8	10.4	6.6
PN5620	0.2 to 1.0	40.0	40.0	40.0	22.3	16.0	7.8	-	-	-
	0.4 to 1.2	40.0	40.0	40.0	40.0	30.0	18.1	-	-	-
PN5624	0.8 to 1.5	40.0	40.0	40.0	40.0	30.0	30.0	-	-	-
PN5630	0.2 to 1.0	-	-	-	-	-	-	5.4	3.6	2.3
	0.4 to 1.2*	-	-	-	-	-	-	11.0	7.3	4.6
PN5634	0.8 to 1.5	-	-	-	-	-	-	22.0	14.5	9.2

Fig. 6.6.10 Typical manufacturer's valve/actuator selection chart

Positioners

For many applications, the 0.2 to 1 bar pressure in the diaphragm chamber may not be enough to cope with friction and high differential pressures. A higher control pressure and stronger springs could be used, but the practical solution is to use a positioner.

This is an additional item (see Figure 6.6.11), which is usually fitted to the yoke or pillars of the actuator, and it is linked to the spindle of the actuator by a feedback arm in order to monitor the valve position. It requires its own higher-pressure air supply, which it uses to position the valve.

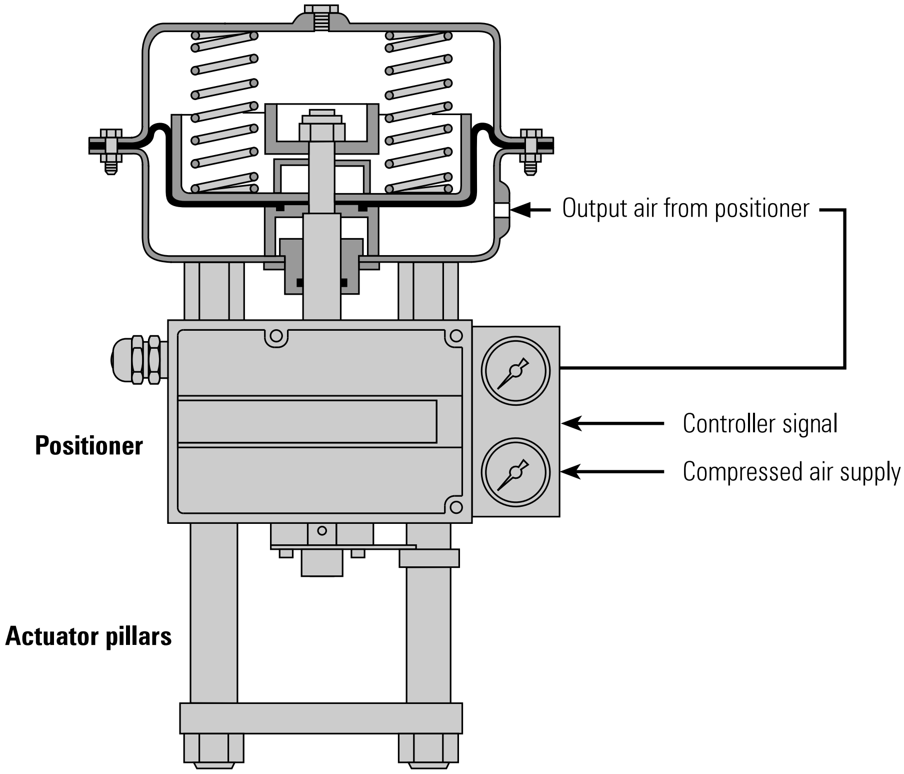


Fig. 6.6.11 Basic pneumatic positioner fitted to actuator pillars (valve not shown)

A valve positioner relates the input signal and the valve position, and will provide any output pressure to the actuator to satisfy this relationship, according to the requirements of the valve, and within the limitations of the maximum supply pressure.

When a positioner is fitted to an 'air-to-open' valve and actuator arrangement, the spring range may be increased to increase the closing force, and hence increase the maximum differential pressure a particular valve can tolerate. The air pressure will also be adjusted as required to overcome friction, therby reducing hysteresis effects.

Example: Taking a PN5400 series actuator fitted to a DN50 valve (see Table in Figure 6.6.10)

1. With a standard 0.2 to 1.0 bar spring range (PN5420), the maximum allowable differential pressure is 3.0 bar.
2. With a 1.0 to 2.0 bar spring set (PN5426), the maximum allowable differential pressure is increased to 13.3 bar.

With the second option, the 0.2 to 1.0 bar signal air pressure applied to the actuator diaphragm cannot provide sufficient force to move an actuator against the force provided by the 1.0 to 2.0 bar springs, and even less able to control it over its full operating range. In these circumstances the positioner acts as an amplifier to the control signal, and modulates the supply air pressure, to move the actuator to a position appropriate to the control signal pressure.

For example, if the control signal was 0.6 bar (50% valve lift), the positioner would need to allow approximately 1.5 bar into the actuator diaphragm chamber. Figure 6.6.12 illustrates this relationship.

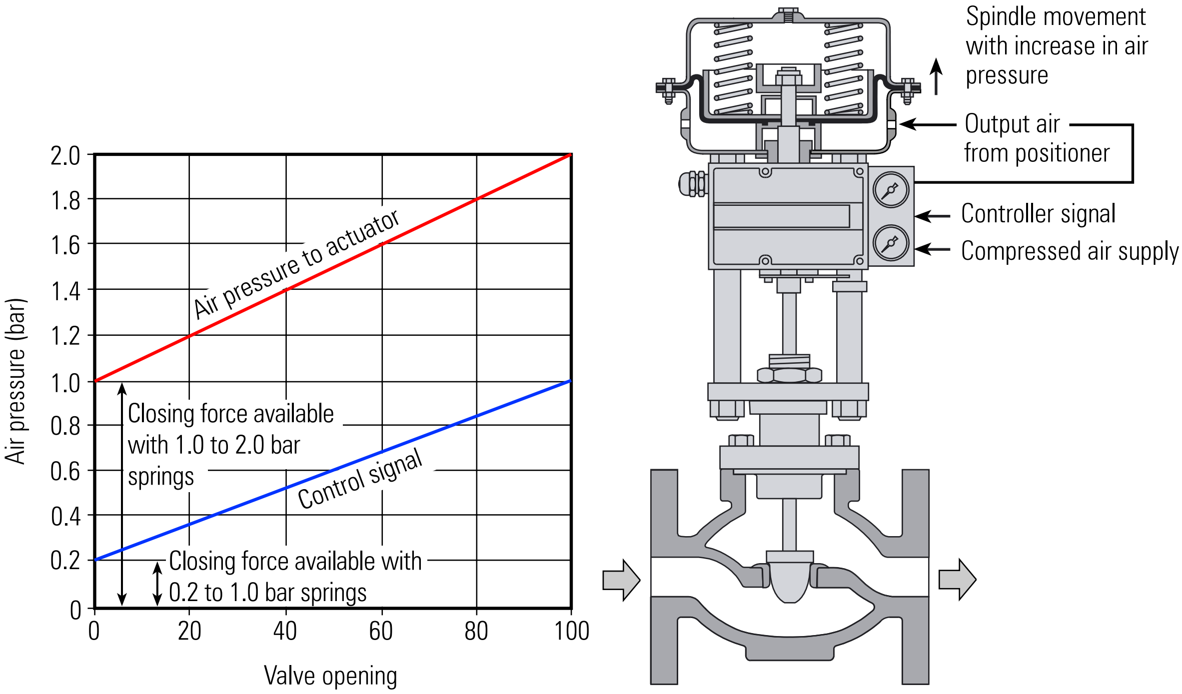


Fig. 6.6.12 The positioner as a signal amplifier

It should be noted that a positioner is a proportional device, and in the same way that a proportional controller will always give an offset, so does a positioner.

On a typical positioner, the proportional band may be between 3 and 6%. The positioner sensitivity can usually be adjusted. It is essential that the installation and maintenance instructions be read prior to the commissioning stage.

Summary - Positioners

1. A positioner ensures that there is a linear relationship between the signal input pressure from the control system and the position of the control valve. This means that for a given input signal, the valve will always attempt to maintain the same position regardless of changes in valve differential pressure, stem friction, diaphragm hysteresis and so on.

2. A positioner may be used as a signal amplifier or booster. It accepts a low pressure air control signal and, by using its own higher pressure input, multiplies this to provide a higher pressure output air signal to the actuator diaphragm, if required, to ensure that the valve reaches the desired position.

3. Some positioners incorporate an electropneumatic converter so that an electrical input (typically 4 - 20 mA) can be used to control a pneumatic valve.

4. Some positioners can also act as basic controllers, accepting input from sensors.

A frequently asked question is, 'When should a positioner be fitted?'

A positioner should be considered in the following circumstances:

1. When accurate valve positioning is required.

2. To speed up the valve response. The positioner uses higher pressure and greater air flow to adjust the valve position.

3. To increase the pressure that a particular actuator and valve can close against. (To act as an amplifier).

4. Where friction in the valve (especially the packing) would cause unacceptable hysteresis.

5. To linearise a non-linear actuator.

6. Where varying differential pressures within the fluid would cause the plug position to vary.

To ensure that the full valve differential pressure can be accepted, it is important to adjust the positioner zero setting so that no air pressure opposes the spring force when the valve is seating.

Figure 6.6.13 shows a typical positioner. Commonly, this would be known as a P to P positioner since it takes a pneumatic signal (P) from the control system and provides a resultant pneumatic output signal (P) to move the actuator.

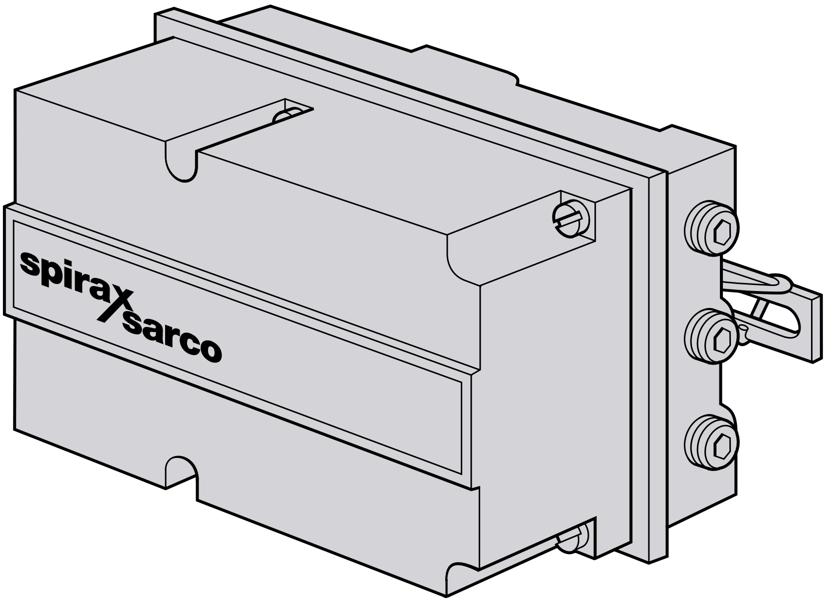


Fig. 6.6.13 Typical P to P positioner (gauges omitted for clarity)

One advantage of a pneumatic control is that it is intrinsically safe, i.e. there is no risk of explosion in a dangerous atmosphere, and it can provide a large amount of force to close a valve against high differential pressure. However, pneumatic control systems themselves have a number of limitations compared with their electronic counterparts.

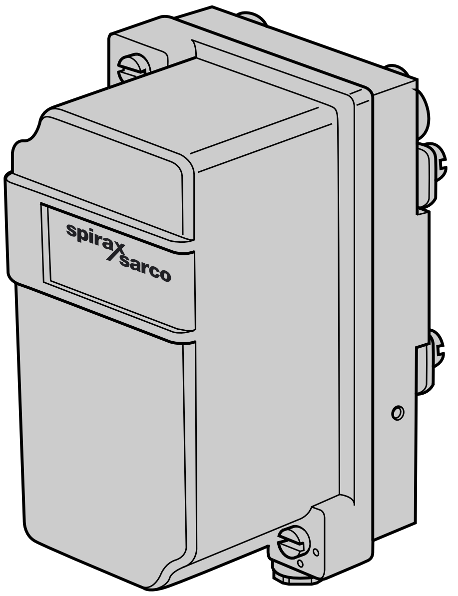


Fig. 6.6.14 Typical I to P converter

To alleviate this, additional components are available to enable the advantages of a pneumatic valve and actuator to be used with an electronic control system.

The basic unit is the I to P converter. This unit takes in an electrical control signal, typically 4 - 20 mA, and converts it to a pneumatic control signal, typically 0.2 - 1 bar, which is then fed into the actuator, or to the P to P positioner, as shown in Figure 6.6.15.

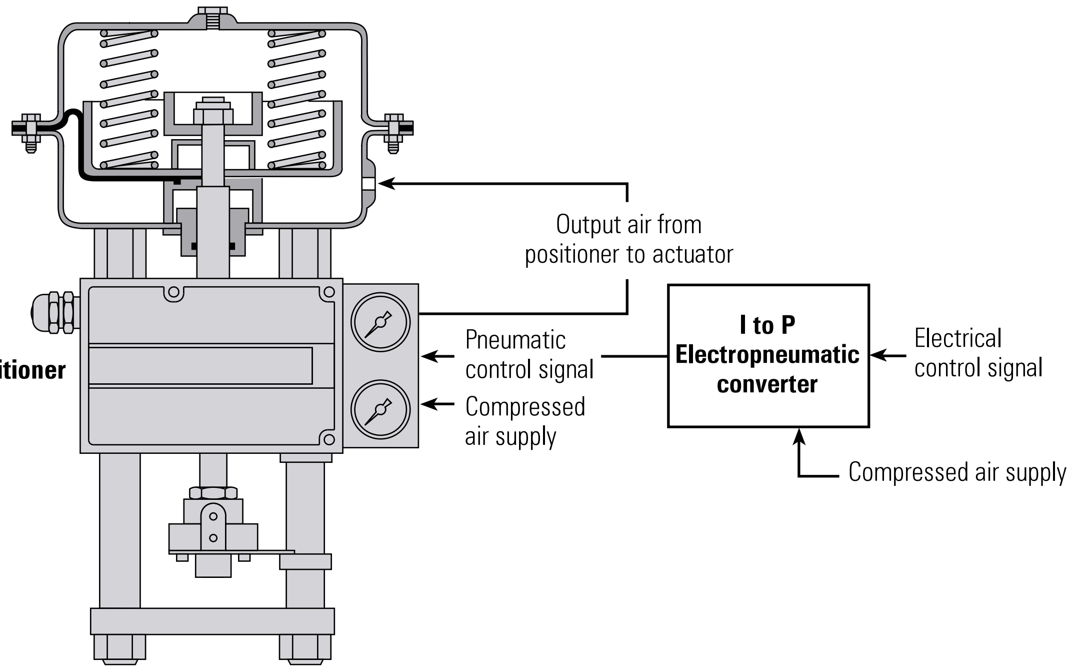


Fig. 6.6.15 Pneumatic valve/actuator operated by a control signal using I to P converter and P to P positioner

With this arrangement, an I to P (electrical to pneumatic) conversion can be carried out outside any hazardous area, or away from any excessive ambient temperatures, which may occur near the valve and pipeline.

However, where the conditions do not present such problems, a much neater solution is to use a single component electropneumatic converter / positioner, which combines the functions of an I to P converter and a P to P positioner, that is a combined valve positioner and electropneumatic converter. Figure 6.6.16 shows a typical I to P converter/positioner.

Fig. 6.6.16 A typical I to P converter/positioner fitted to a pneumatic valve (gauges omitted for clarity)

Most sensors still have analogue outputs (for example 4 - 20 mA or 0 - 10 V), which can be converted to digital form. Usually the controller will perform this analogue-to-digital (A / D) conversion, although technology is now enabling sensors to perform this A/D function themselves. A digital sensor can be directly connected into a communications system, such as Fieldbus, and the digitised data transmitted to the controller over a long distance. Compared to an analogue signal, digital systems are much less susceptible to electrical interference.

Analogue control systems are limited to local transmission over relatively short distances due to the resistive properties of the cabling.

Most electrical actuators still require an analogue control signal input (for example 4 - 20 mA or 0 - 10 V), which further inhibits the completion of a digital communications network between sensors, actuators, and controllers.

Digital positioners

Sometimes referred to as a SMART positioner, the digital positioner monitors valve position, and converts this information into a digital form. With this information, an integrated microprocessor offers advanced user features such as:

- High valve position accuracy.
- Adaptability to changes in control valve condition.
- Many digital positioners use much less air than analogue types.
- An auto stroking routine for easy setting-up and calibration.
- On-line digital diagnostics*
- Centralised monitoring*

*Using digital communications protocols such as HART® ; Fieldbus, or Profibus.

The current industrial trend is to provide equipment with the capability to communicate digitally with networked systems in a Fieldbus environment. It is widely thought that digital communications of this type offer great advantages over traditional analogue systems.

Fig. 6.6.17 Digital positioner

Selecting a pneumatic valve and actuator

In summary, the following is a list of the major factors that must be considered when selecting a pneumatic valve and actuator:

1. Select a valve using the application data.
2. Determine the valve action required in the event of power failure, fail-open or fail-closed.
3. Select the valve actuator and spring combination required to ensure that the valve will open or close against the differential pressure.
4. Determine if a positioner is required.
5. Determine if a pneumatic or electric control signal is to be provided. This will determine whether an I to P converter or, alternatively a combined I to P converter/positioner, is required.

Rotary pneumatic actuators and positioners

Actuators are available to drive rotary action valves, such as ball and butterfly valves. The commonest is the piston type, which comprises a central shaft, two pistons and a central chamber all contained within a casing. The pistons and shaft have a rack and pinion drive system.

In the simplest types, air is fed into the central chamber (Figure 6.6.18a), which forces the pistons outwards.

The rack and pinion arrangement turns the shaft and, because the latter is coupled to the valve stem, the valve opens or closes.

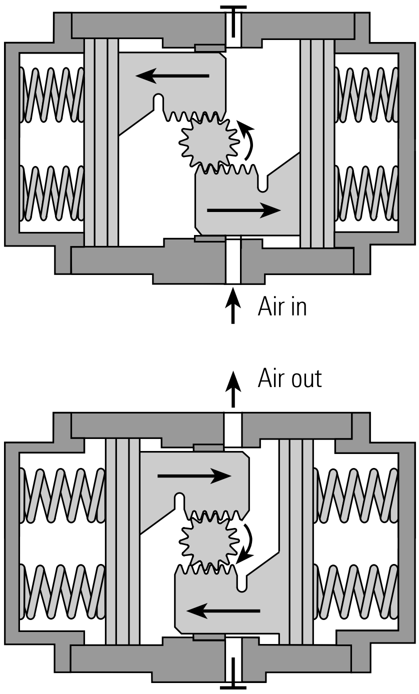
When the air pressure is relieved, movement of the shaft in the opposite direction occurs due to the force of the return springs (Figure 6.6.18b).

It is also possible to obtain double acting versions, which have no return springs. Air can be fed into either side of the pistons to cause movement in either direction. As with diaphragm type actuators, they can also be fitted with positioners.

a

Anticlockwise

Air is supplied forcing the pistons away from each other (towards the ends), rotating the drive pinion anticlockwise.



b

Clockwise

Air failure (loss of pressure) allows compressed springs to force pistons towards each other (toward centre), rotating the drive pinion clockwise and exhausting the air.

Fig. 6.6.18 Spring return rotary pneumatic actuator

Air supply

An adequate compressed air supply system is essential to provide clean and dry air at the right quantity and pressure. It is advantageous to install an individual coalescing filter/regulator unit ahead of the final supply connection to each piece of equipment. Air quality is particularly important for pneumatic instrumentation such as controllers, I to P convertors and positioners.

The decision to opt for a pneumatically operated system may be influenced by the availability and/or the costs to install such a system. An existing air supply would obviously encourage the use of pneumatically powered controls.

Electrical actuators

Where a pneumatic supply is not available or desirable it is possible to use an electric actuator to control the valve. Electric actuators use an electric motor with voltage requirements in the following range: 230 Vac, 110 Vac, 24 Vac and 24 Vdc.

There are two types of electrical actuator; VMD (Valve Motor Drive) and Modulating.

VMD (Valve Motor Drive)
This basic version of the electric actuator has three states:

1. Driving the valve open.
2. Driving the valve closed.
3. No movement.

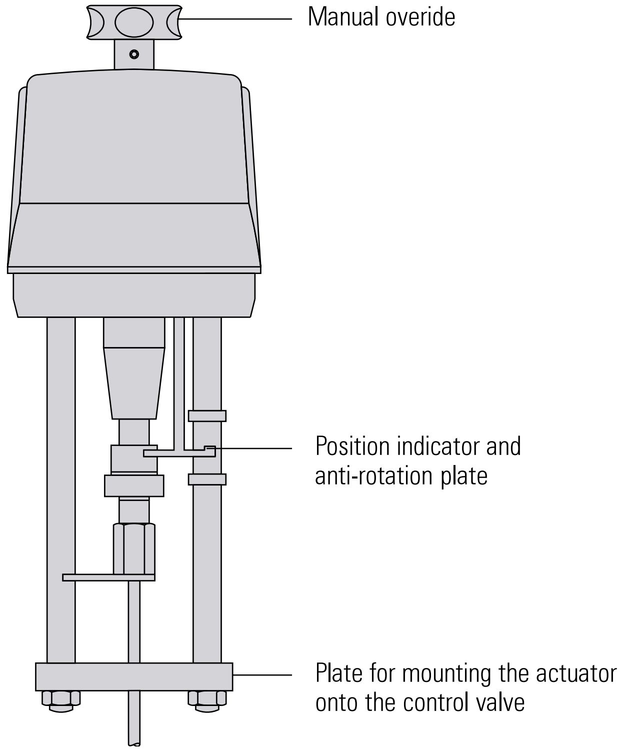


Fig. 6.6.19 Typical electric valve actuator

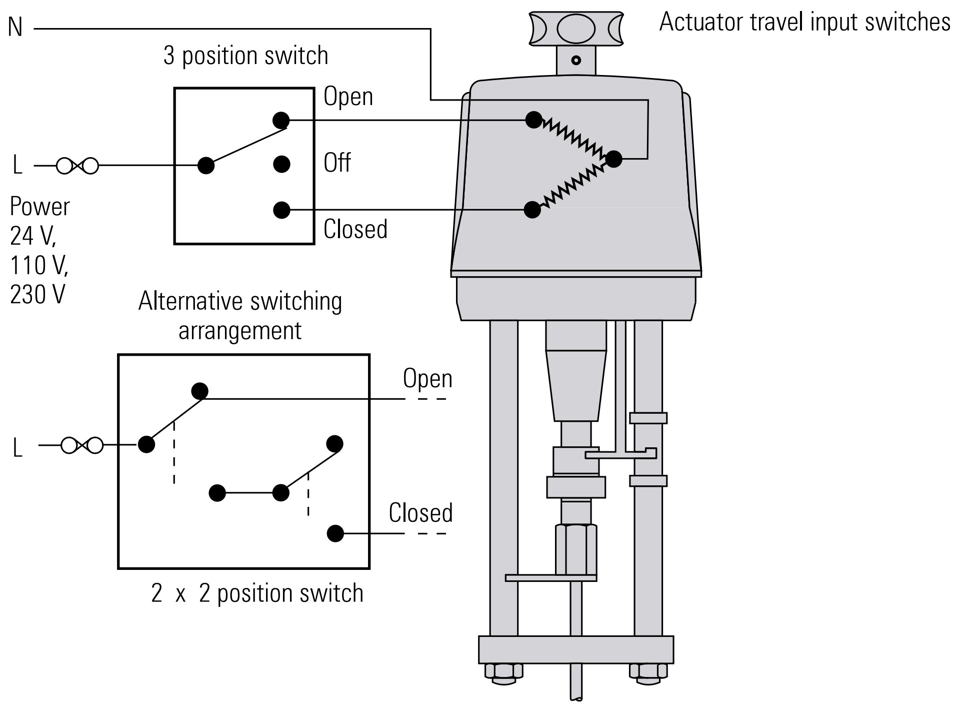


Fig. 6.6.20 Valve motor drive actuator system

Figure 6.6.20 shows the VMD system where the forward and reverse travel of the actuator is controlled directly from any external 3-position or two 2-position switch units. The switches are rated at the actuator voltage and may be replaced by suitable relays.

Limiting devices are fitted within the VMD actuators to protect the motors from over-travel damage. These devices are based on either the maximum motor torque or physical position limit switches. Both devices stop the motor driving by interrupting the motor power supply.

- Position limit switches have the advantage that they can be adjusted to limit valve strokes in oversized valves.
- Torque switches have the advantage of giving a defined closing force on the valve seat, protecting the actuator in the case of valve stem seizure.
- If only position limit switches are used, they may be combined with a spring-loaded coupling to ensure tight valve shut-off.

A VMD actuator may be used for on/off actuation or for modulating control. The controller positions the valve by driving the valve open or closed for a certain time, to ensure that it reaches the desired position. Valve position feedback may be used with some controllers.

Modulating

In order to position the control valve in response to the system requirements a modulating actuator can be used. These units may have higher rated motors (typically 1 200 starts/hour) and may have built-in electronics.

A positioning circuit may be included in the modulating actuator, which accepts an analogue control signal (typically 0-10 V or 4-20 mA). The actuator then interprets this control signal, as the valve position between the limit switches.

To achieve this, the actuator has a position sensor (usually a potentiometer), which feeds the actual valve position back to the positioning circuit. In this way the actuator can be positioned along its stroke in proportion to the control signal. A schematic of the modulating actuator is shown in Figure 6.6.21.

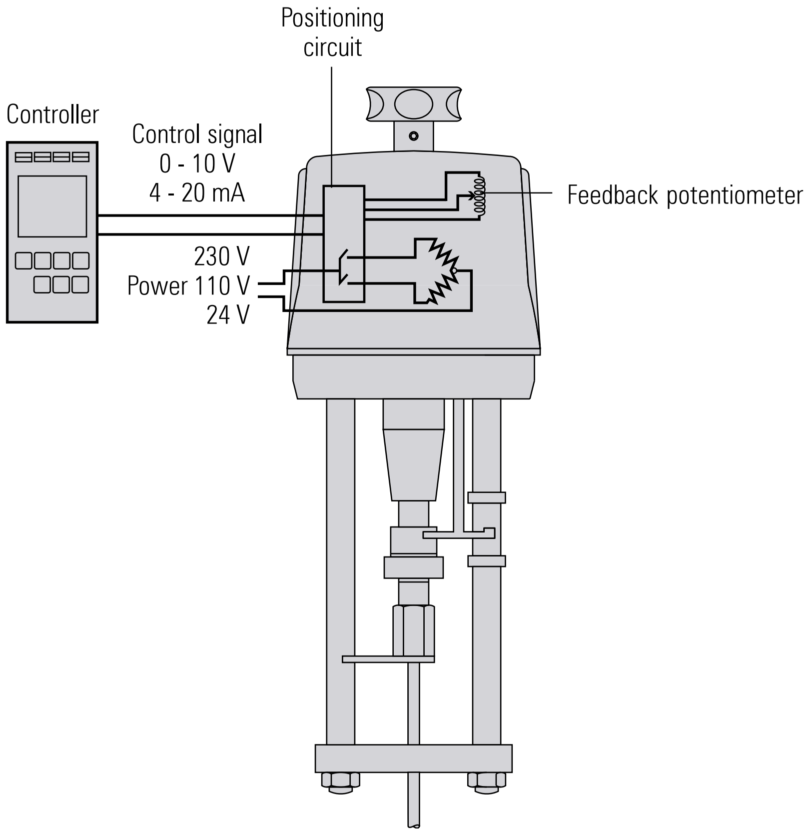


Fig. 6.6.21 Integral positioning circuit for modulating electric actuators

Pneumatic actuators have an inherent fail-safe feature; should the air supply or control signal fail the valve will close. To provide this function in electric actuators, 'spring reserve' versions are available which will open or close the valve on power or control signal failure. Alternatively, fail-safe can be provided with battery power.

Electric actuators offer specified forces, which may be limited on spring reserve versions. The manufacturer's charts should always be consulted during selection.

When sizing an actuator, it is wise to refer to the manufacturer's technical data sheets for maximum differential pressure across the valve (see Figure 6.6.22).

Another limitation of an electric actuator is the speed of valve movement, which can be as low as 4 seconds/mm, which in rapidly varying systems may be too slow.

EL series actuators

Valve size		**DN15**	**DN20**	**DN25**	**DN32**	**DN40**	**DN50**	**DN65**	**DN80**	**DN100**
Actuator	**Voltage**	**Maximum differential pressure (bar)**								
EL5601	230	40.0	30.3	18.3	9.3	5.4	2.9	1.2	0.6	0.3
EL5602	110	40.0	30.3	18.3	9.3	5.4	2.9	1.2	0.6	0.3
EL5603	24	40.0	30.3	18.3	9.3	5.4	2.9	1.2	0.6	0.3
EL5611	230	40.0	40.0	38.3	19.8	12.0	6.7	3.5	2.2	1.3
EL5612	110	40.0	40.0	38.3	19.8	12.0	6.7	3.5	2.2	1.3
EL5613	24	40.0	40.0	38.3	19.8	12.0	6.7	3.5	2.2	1.3
EL5621	230			40.0	40.0	28.5	16.3	9.3	6.1	3.8
EL5622	110			40.0	40.0	28.5	16.3	9.3	6.1	3.8
EL5623	24			40.0	40.0	28.5	16.3	9.3	6.1	3.8
EL5631	230					40.0	29.7	17.5	11.5	7.4
EL5632	110				40.0	29.7	17.5	11.5	7.4	
EL5633	24					40.0	29.7	17.5	11.5	7.4
EL5641	230						40.0	26.7	17.8	11.4
EL5642	110						40.0	26.7	17.8	11.4
EL5643	24						40.0	26.7	17.8	11.4
EL5651	230							40.0	38.0	24.6
EL5652	110							40.0	38.0	24.6
EL5653	24							40.0	38.0	24.6

Fig. 6.6.22 Typical manufacturer's electric actuator selection chart

Questions

1. In a reverse acting actuator what happens upon air failure?

a| The valve spindle does not move ☐
b| The valve spindle retracts ☐
c| The valve spindle extends ☐
d| The valve will always close ☐

2. In a direct acting actuator what happens upon air failure?

a| The valve spindle does not move ☐
b| The valve spindle retracts ☐
c| The valve spindle extends ☐
d| The valve will always open ☐

3. With a direct acting actuator on a reverse acting valve, what happens upon air failure?

a| The valve spindle does not move ☐
b| The valve closes ☐
c| The valve opens ☐
d| It is not possible to fit this combination of actuator and valve ☐

4. With a reverse acting actuator on a direct acting 2-port valve, what is required due to the effect of differential pressure?

a| The closing force must decrease ☐
b| The air pressure must decrease ☐
c| The air pressure must increase ☐
d| It is not possible to fit this combination of actuator and valve ☐

5. What is the difference between an I to P positioner and I to P converter?

a| The positioner is fitted off the valve, the converter on the valve ☐
b| The positioner and converter are both fitted on the valve ☐
c| The positioner and converter are both fitted off the valve ☐
d| The positioner is fitted on the valve, the converter off the valve ☐

6. A VMD electric actuator can only be used for on/off control – true or false?

a| True ☐
b| False ☐

Answers

1: c, 2: b, 3: b, 4: b, 5: b, 6: b

Module 6.7

Controllers and Sensors

Controllers

It is important to state at the outset that not all control applications need a sophisticated controller.

An on/off valve and actuator, for example, can be operated directly from a thermostat. Another example is the operation of high limit safety controls, which have a 'snap' action to close valves or to switch off fuel supplies.

However, when the control requirements become more sophisticated, a controller is needed to match these requirements.

The controller receives a signal, decides what action is needed and then sends a signal to the actuator to make it move.

In the age of the microchip, integrated circuits and computers, the functions performed by the controller can be very complex indeed.

However, since an analogy between the human brain and controllers/computers has been made in previous Modules, the renowned IBM motto can be paraphrased:

Computer - Fast, accurate and stupid

Human being - Slow, slovenly and brilliant

To summarise, the controller will not solve all problems. It must be properly selected and commissioned, subjects which will be dealt with later.

Although most controllers are now electronic digital/microprocessor based, a range of pneumatic controllers is commercially available. These might be used in hazardous areas where the risk of explosion precludes the use of electrics/electronics. It is possible to make electrical equipment 'intrinsically safe' or explosion-proof or flameproof, however, there is usually a substantial cost implication.

As previously mentioned, the functions carried out by the controller can be very complex and it is beyond the scope of this publication to list them in detail, or to explain how they operate.

The major variations that require consideration are as follows:

Single loop controller
Operates one valve/actuator from a single sensor.

Multi-loop controller
May operate more than one valve/actuator from more than one sensor.

Single input/output
Can accept only one signal from the sensor and send only one to the actuator.

Multi-input/output (multi-channel)
Can accept several signals and send out several signals.

Real time
May include a time clock to switch at pre-determined, pre-set times.

Elapsed time
May switch at some predetermined, pre-set length of time before or after other items of plant have been switched on or off.

Ramp and dwell
Using temperature as an example, the capability to raise the temperature of a controlled medium over a specified time period and then to hold it at a pre-set value. Such controllers frequently incorporate a series of ramps and dwells.

Figure 6.7.1, shows a typical electronic, single loop controller. This has P + I + D action (discussed in Modules 5.2 and 5.4), suitable for 110 or 230 volt supply.

Figure 6.7.2 shows a pneumatic single loop controller with P action.

Different models can be selected to control either temperature or pressure.

Fig. 6.7.1 Electronic single loop controller

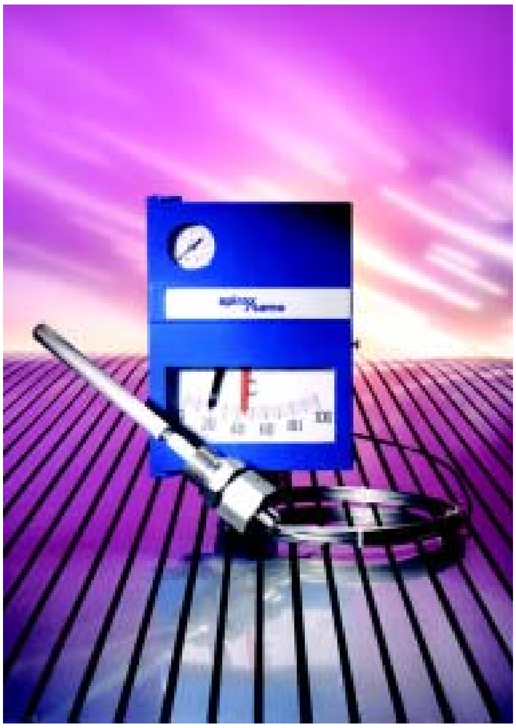


Fig. 6.7.2 Pneumatic single loop temperature controller

A single loop controller, which has the ability to perform ramp and dwell functions, may have a typical sequence pattern like the one shown in Figure 6.7.3. This shows a series of ramps (temperature change) and dwell (maintaining temperature) functions, carried out over a period of time.

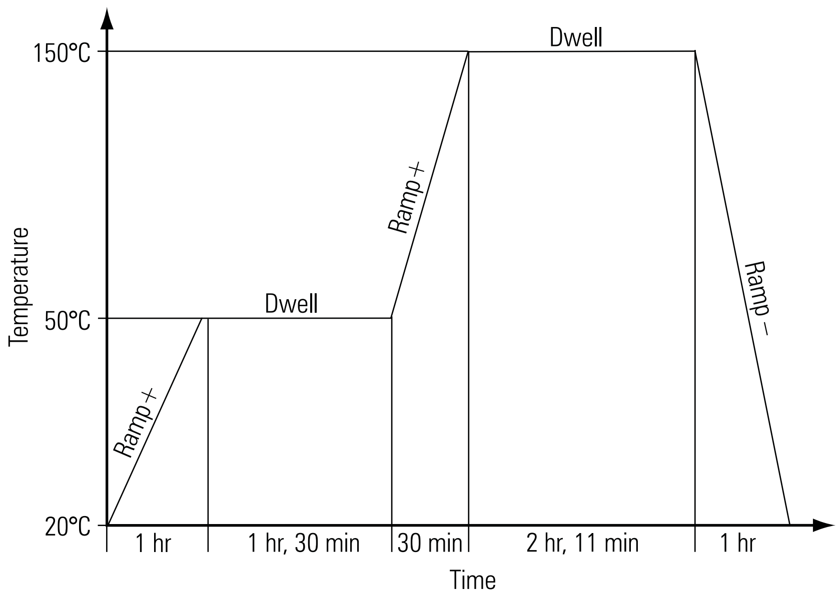


Fig. 6.7.3 Typical multi-sequence ramp and dwell pattern

One term frequently found in control literature is 'Programmable Logic Controller (PLC)'. In a batch process, the controller must trigger a sequence of actions, for example, turning valves or pumps on or off. In some cases the whole sequence is on a timed basis, but often the various steps may be triggered by a specific condition being reached and maintained for a certain time period; for example a certain temperature being reached or a vessel filled. These sequences can be controlled by a PLC, a microcomputer-based device that utilises standard interfaces for sensors and actuators to control the process.

Another type of complex controller is the plant room controller, which might be used to control the boiler, pump, heating control valve, HWS valve, as well as providing a number of other features.

Sensors

In this Section the subject of temperature measurement will be covered more broadly. There are a wide variety of sensors and transducers available for measuring pressure, level, humidity, and other physical properties. The sensor is the part of the control system, which experiences the change in the controlled variable.

The sensor may be of a type where a change in temperature results in a change of voltage or perhaps a change in resistance.

The signal from the sensor may be very small, creating the need for local signal conditioning and amplification to read it effectively. A small change in resistance signalled by a sensor in response to a change in temperature, may, for example, be converted to an electrical voltage or current for onward transmission to the controller.

The transmission system itself is a potential source of error.

Wiring incurs electrical resistance (measured in ohms), as well as being subject to electrical interference (noise). In a comparable pneumatic system, there may also be minute leaks in the piping system.

The term 'thermostat' is generally used to describe a temperature sensor with on/off switching. 'Transducer' is another common term, and refers to a device that converts one physical characteristic into another; for example, temperature into voltage (millivolts).

An example of a transducer is a device that converts a change in temperature to a change in electrical resistance.

With pneumatic devices, the word 'transmitter' is frequently encountered. It is simply another description of transducer or sensor, but usually with some additional signal conditioning.

However, the actual measuring device is usually termed as the sensor, and the more common types will be outlined in the following Section.

Filled system sensors

With pneumatic controllers, filled system sensors are employed. Figure 6.7.4 illustrates the principles of such a system.

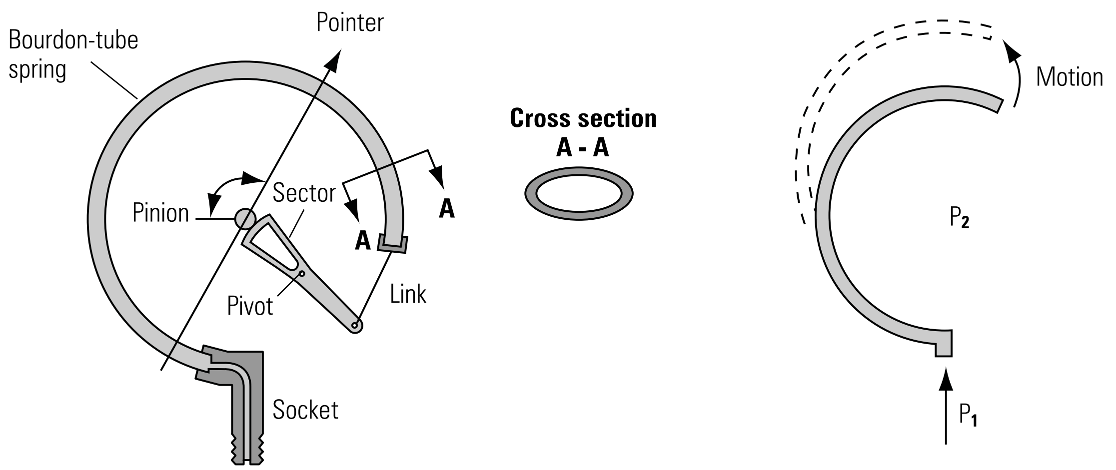


Fig. 6.7.4 Liquid filled system sensor and gas filled or vapour pressure system

When the temperature changes, the fluid expands or contracts, causing the Bourdon tube to tend to straighten out. Sometimes a bellows is used instead of a Bourdon tube.

In the past, the filling has often been mercury. When heated, it expands, causing the Bourdon tube to uncoil; cooling causes contraction and forces the Bourdon tube to coil more tightly. This coil movement is used to operate levers within the pneumatic controller enabling it to perform its task. A pressure sensing version will simply utilise a pressure pipe connected to the Bourdon tube. Note: for health and safety reasons, mercury is now used less often. Instead, an inert gas such as nitrogen is often employed.

Resistance temperature detectors (RTDs)

RTDs (Figure 6.7.5) employ the fact that the electrical resistance of certain metals change as the temperature alters. They act as electrical transducers, converting temperature changes to changes in electrical resistance. Platinum, copper, and nickel are three metals that meet RTD requirements and Figure 6.7.6 shows the relationship between resistance and temperature.

A resistance temperature detector is specified in terms of its resistance at 0°C and the change in resistance from 0°C to 100°C. The most widely used RTD for the typical applications covered in these Modules are platinum RTDs. These are constructed with a resistance of 100 ohms at 0°C and are often referred to as Pt100 sensors. They can be used over a temperature range of -200°C to +800°C with high accuracy (± 0.5%) between 0°C and 100°C.

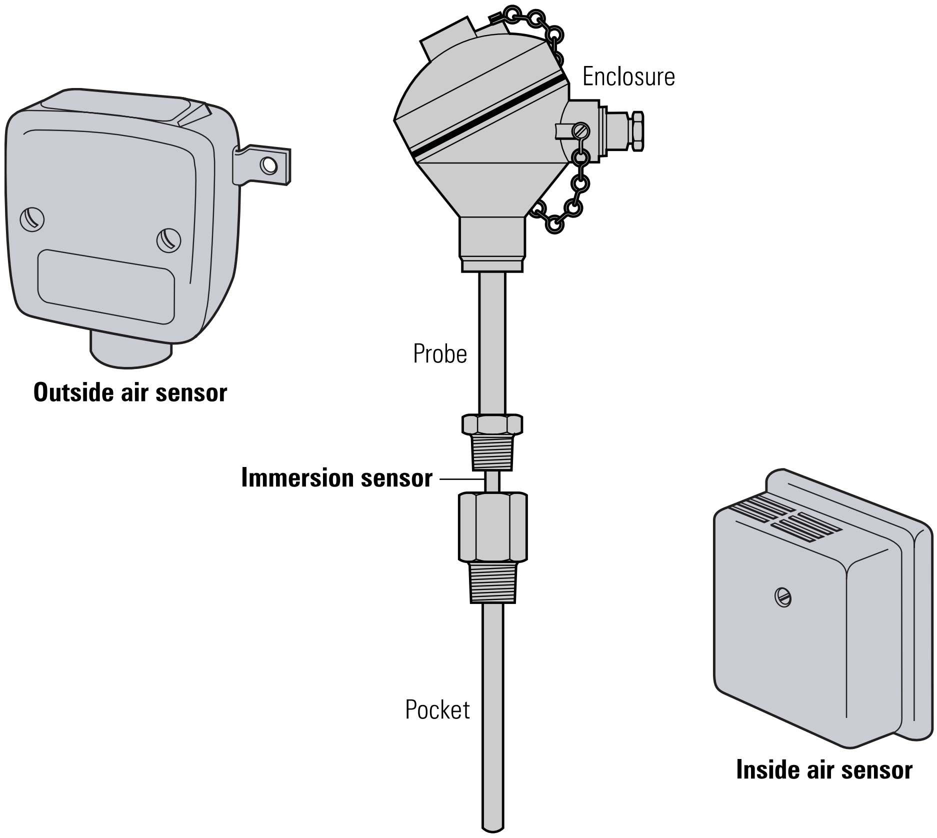


Fig. 6.7.5 Typical resistance temperature sensors

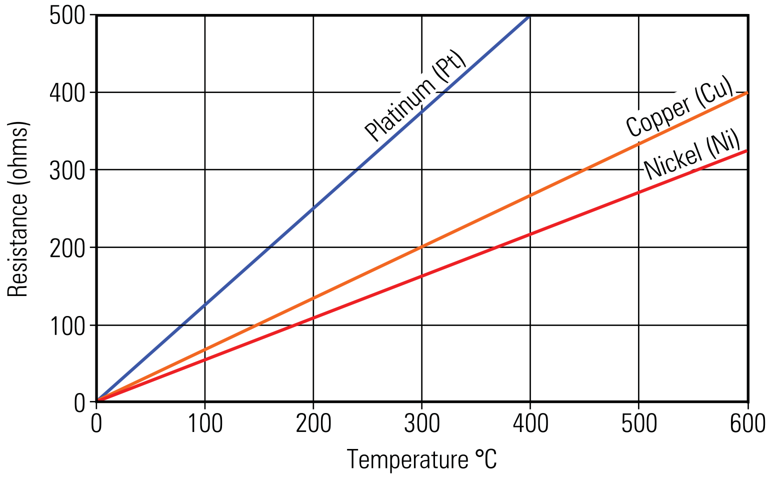


Fig. 6.7.6 RTD element typical resistance/temperature graphs

As can be seen from Figure 6.7.6, the increase of resistance with temperature is virtually linear. RTDs have a relatively small change in resistance, which requires careful measurement. Resistance in the connecting cables needs to be properly compensated for.

Thermistors

Thermistors use semi-conductor materials, which have a large change in resistance with increasing temperature, but are non-linear. The resistance decreases in response to rising temperatures (negative coefficient thermistor), as shown in Figure 6.7.7.

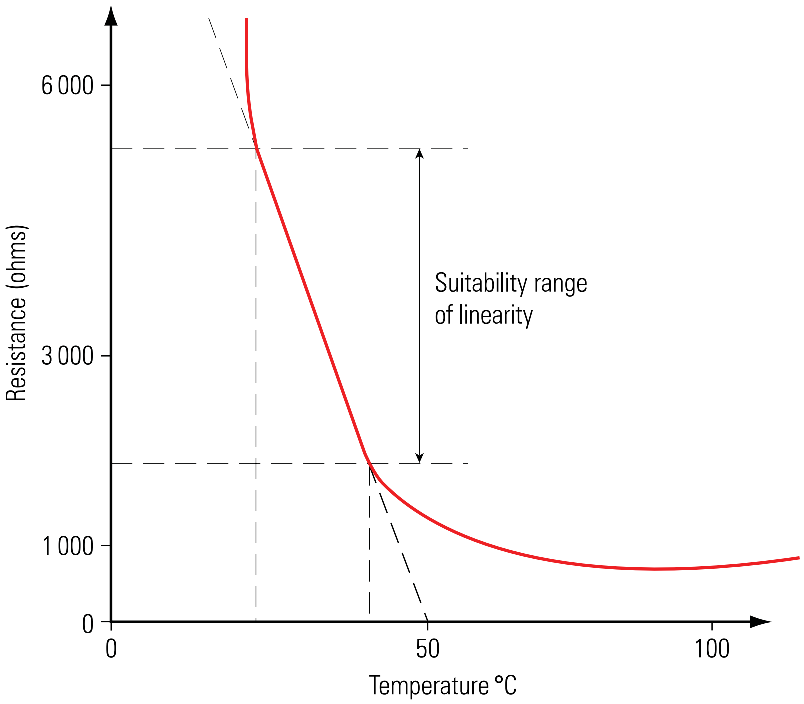


Fig. 6.7.7 Negative coefficient thermistor

Positive coefficient thermistors can be manufactured where the resistance increases with rising temperature (Figure 6.7.8) but their response curve makes them generally unsuitable for temperature sensing.

Thermistors are less complex and less expensive than RTDs but do not have the same high accuracy and repeatability. Their high resistance means that the resistance of the connecting cable is less important.

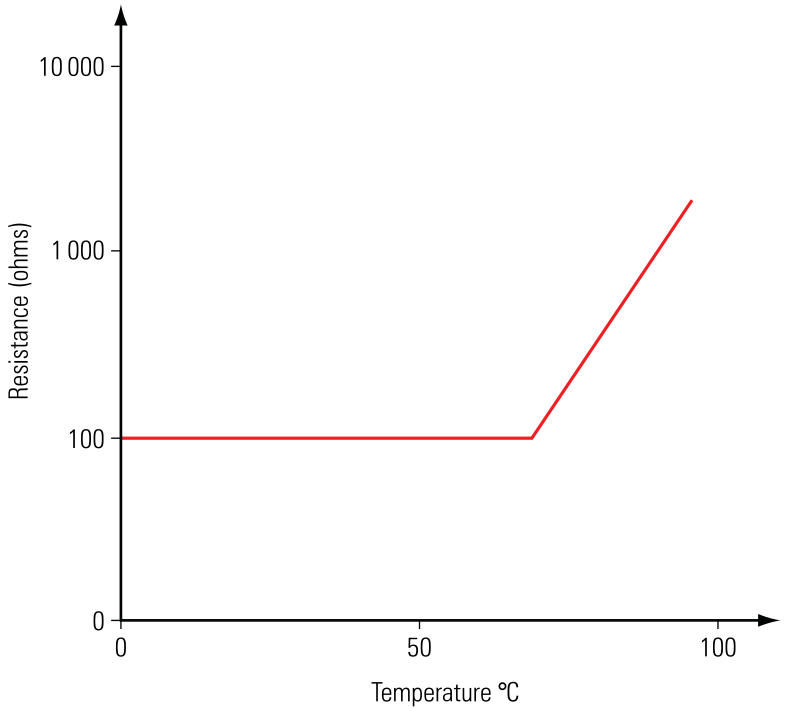


Fig. 6.7.8 Positive coefficient thermistor

Thermocouples

If two dissimilar metals are joined at two points and heat is applied to one junction (as shown in Figure 6.7.9), an electric current will flow around the circuit. Thermocouples produce a voltage corresponding to the temperature difference between the measuring junction (hot) and the reference junction (cold).

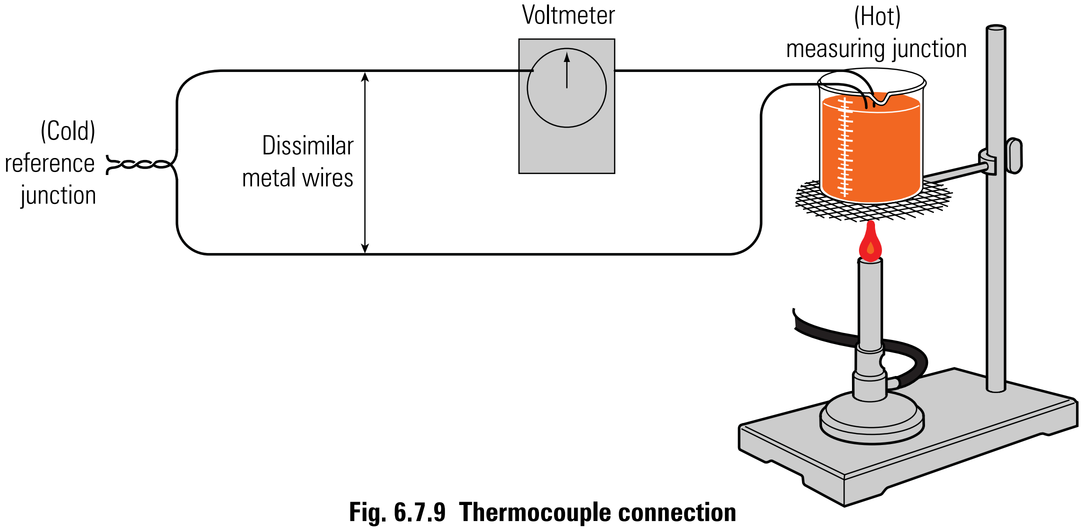


Fig. 6.7.9 Thermocouple connection

The cold reference junction temperature must be accurately known if the thermocouple itself is to provide accurate sensing.

Traditionally, the cold junction was immersed in melting ice (0°C), but the temperature of the cold junction is now measured by a thermistor or an RTD and, from this, the indicated temperature, generally at the measuring junction, is corrected. This is known as cold junction compensation.

Any pair of dissimilar metals could be used to make a thermocouple. But over the years, a number of standard types have evolved which have a documented voltage and temperature relationship. The standard types are referred to by the use of letters, that is, Type J, K, T and others.

Table 6.7.1 Standard range of thermocouples and their range (°C)

Thermocouple ISA Type designation	J	K	T	R	S	N	B	L
Temperature Range (°C)	-200 to +1 000	0 to 1 260	-200 to +400	0 to 1 760	0 to 1 760	0 to 1 760	0 to 1 760	0 to 500

The most widely used general-purpose thermocouple is Type K.

The dissimilar metals used in this type are Chrome (90% nickel, 10% chromium) and Alumel (94% nickel, 3% manganese, 2% aluminium and 1% silicon) and can be used between the range 0°C to 1 260°C. Figure 6.7.10 illustrates the sensitivity of Type K thermocouples, and it can be seen that the output voltage is linear across the complete range.

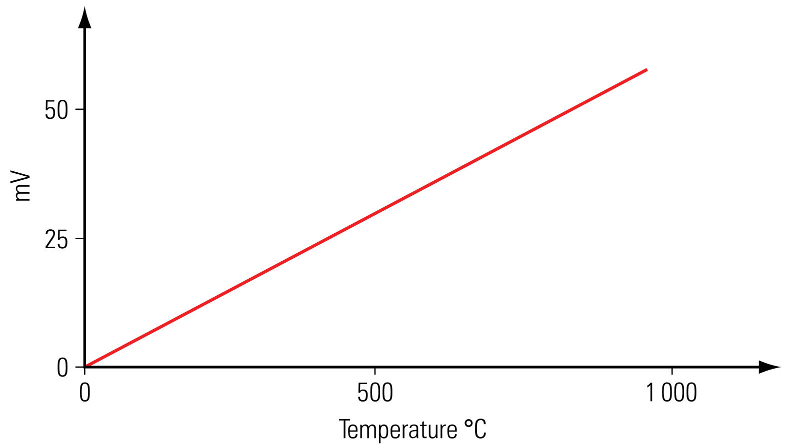


Fig. 6.7.10 Sensitivity of Type K thermocouple

Extension tail wires are used to connect the measuring junction to the reference junction in the instrument case. These extension tails may be of the same material as the wires in the thermocouple itself, or may be a compensating cable made of copper and copper-nickel alloy. Both extension tails must be of the same material.

Thermocouples are available in a wide variety of sizes and shapes. They are inexpensive and rugged and reasonably accurate, with wide temperature ranges. However, the reference junction temperature must be held at a constant value otherwise deviations must be compensated for. The low junction voltages mean that special screened cable and careful installation must be used to prevent electrical interference or 'noise' from distorting signals.

Electrical communication signals

The output signals from most control systems are low power analogue signals but there is a growing use of digital systems such as 'Fieldbus®' or 'Profibus®'.

An analogue system provides a continuous but modulating signal whereas a digital system provides a stream of binary numeric values represented by a change between two specific voltage levels or frequencies.

A comparison between digital and analogue systems can be made using Example 6.7.1 and Example 6.7.2:

Example 6.7.1

Imagine two people, person **A** and person **B**, each on opposite hilltops and each with a flag and a flag-pole. The aim is for person **A** to communicate to person **B** by raising his flag to a certain height. Person **A** raises his flag half way up his pole. Person **B** sees this and also raises his flag halfway. As person **A** moves his flag up or down so does person **B** to match. This would be similar to an analogue system.

Example 6.7.2

Now assume that person **A** does not have a pole but instead has two boards, one with the figure '0' and the other with the figure '1', and again wants person **B** to raise his flag half way, that is to a height of 50% of his flag-pole. The binary number for 50 is 110010, so he displays his boards, two at a time, in the corresponding order. Person **B** reads these boards, translates them to mean 50 and raises his flag exactly half way. This would be similar to a digital system.

It can be seen that the digital system is more precise as the information is either a '1' or a '0' and the position can be accurately defined. The analogue example is not so precise because person **B** cannot determine if person **A**'s flag is at exactly 50%. It could be at 49% or 51%. It is for this reason, together with higher integration of microprocessor circuitry that digital signals are becoming more widely used.

Digital addressing

Digital addressing allows a controller to send information over a set of wires onto which several receivers are connected and yet be able to communicate with only one of those receivers if required. This is done by allocating an address to each receiver, which the controller must broadcast first.

To explain this, consider the digital example above but now assume that there is another person, person **C** on a third hill. Person **B** and person **C** can both see person **A**, so person **A** must first indicate to whom he is communicating.

This is done with the first board. If the first board is a '0' then all subsequent data is intended for person **B** who adjusts his flag accordingly. Conversely, if the first board is a '1' then all subsequent data is intended for person **C**. Hence person **B** has a digital address of '0' and person **C** has a digital address of '1'; each person knows that the first number to be seen by them refers to the address not the message.

HART®, Profibus® and Foundation™ Fieldbus.

What is HART®?

HART® stands for 'Highway Addressable Remote Transducer' and is a standard originally developed as a communications protocol for control field devices operating on a 4-20 mA control signal.

The HART® protocol uses 1200 baud Frequency Shift Keying (FSK) based on the Bell 202 standard to superimpose digital information on the conventional 4-20 mA analogue signal. Maintained by an independent organisation, the HART® Communication Foundation, the HART® protocol is an industry standard developed to define the communications protocol between intelligent field devices and a control system.

HART® is probably the most widely used digital communication protocol in the process industries, and:

- Is supported by all of the major suppliers of process field instruments.
- Preserves existing control strategies by allowing 4-20 mA signals to co-exist with digital communication on existing 2-wire loops.
- Is compatible with analogue devices.
- Provides important information for installation and maintenance, such as Tag-IDs, measured values, range and span data, product information and diagnostics.
- Can support cabling savings through use of multidrop networks.
- Reduces operating costs via improved management and utilisation of smart instrument networks.

What is Profibus®?

Profibus® is an open fieldbus standard for a wide range of applications in manufacturing and process automation independent of manufacturers. Manufacture independence and transparency are ensured by the international standards EN 50170, EN 50254 and IEC 61158.

It allows communication between devices of different manufacturers without any special interface adjustment. Profibus® can be used for both high-speed time critical applications and complex communication tasks. Profibus® offers functionally graduated communication protocols DP and FMS. Depending on the application, the transmission technologies RS-485, IEC 1158-2 or fibre optics can be used.

It defines the technical characteristics of a serial Fieldbus® system with which distributed digital programmable controllers can be networked, from field level to cell level. Profibus® is a multi-master system and thus allows the joint operation of several automation, engineering or visualization systems with their distributed peripherals on one bus.

At sensor/actuator level, signals of the binary sensors and actuators are transmitted via a sensor/actuator bus. Data are transmitted purely cyclically.

At field level, the distributed peripherals, such as I/O modules, measuring transducers, drive units, valves and operator terminals communicate with the automation systems via an efficient, real-time communication system. As with data, alarms, parameters and diagnostic data can also be transmitted cyclically if necessary.

At cell level, programmable controllers such as PLC and IPC can communicate with each other. The information flow requires large data packets and a large number of powerful communication functions, such as smooth integration into company-wide communication systems, such as Intranet and Internet via TCP/IP and Ethernet.

What is Foundation™ Fieldbus?

Foundation™ Fieldbus is an all-digital, serial, two-way communications system that serves as a Local Area Network (LAN) for factory/plant instrumentation and control devices. The Fieldbus® environment is the base level group of the digital networks in the hierarchy of plant networks. Foundation™ Fieldbus is used in both process and manufacturing automation applications and has a built-in capability to distribute the control application across the network.

Unlike proprietary network protocols, Foundation™ Fieldbus is neither owned by any individual company, nor regulated by a single nation or standards body. The Foundation™ Fieldbus, a not-for-profit organization consisting of more than 100 of the world's leading controls and instrumentation suppliers and end users, controls the technology.

While Foundation™ Fieldbus retains many of the desirable features of the 4-20 mA analogue system, such as a standardized physical interface to the wire, bus-powered devices on a single wire, and intrinsic safety options, it also offers many other benefits.

Device interoperability
Foundation™ Fieldbus offers interoperability; one Fieldbus® device can be replaced by a similar device with added functionality from a different supplier on the same Fieldbus® network while maintaining specified operations. This permits users to 'mix and match' field devices and host systems from various suppliers. Individual Fieldbus® devices can also transmit and receive multivariable information, and communicate directly with each other over a common Fieldbus®, allowing new devices to be added to the Fieldbus® without disrupting services.

Enhanced process data
With Foundation™ Fieldbus, multiple variables from each device can be brought into the plant control system to analyse trends, optimise processes, and generate reports. Access to accurate, high-resolution data enables processes to be fine-tuned for better productivity, less downtime, and higher plant performance.

Overall view of the process
Modern Fieldbus® devices, with powerful microprocessor-based communications capabilities, permit process errors to be recognized faster and with greater certainty. As a result, plant operators are notified of abnormal conditions or the need for preventive maintenance, allowing personnel to consider pro-active decisions. Lower operating efficiencies are corrected more quickly, enabling production to rise while raw material costs and regulatory problems fall.

Improved in plant safety
Fieldbus technology helps manufacturing plants keep up with stringent safety requirements. It can provide operators with earlier warning of potential hazardous conditions, thereby allowing corrective action to be taken to reduce unplanned shutdowns. Enhanced plant diagnostic capabilities also offer less frequent access to hazardous areas, thus minimizing the risks to personnel.

Easier predictive maintenance
Enhanced device diagnostics capabilities make it possible to monitor and track insidious conditions such as valve wear and transmitter fouling. Plant personnel are able to perform predictive maintenance without waiting for a scheduled shutdown, thus reducing or even avoiding downtime.

Reduced wiring and maintenance costs
The use of existing wiring and multi-drop connections provides significant savings in network installation costs. This includes reductions in intrinsic safety barriers and cabling costs, particularly in areas where wiring is already in situ.

Additional cost savings can be achieved through the decreased time required for construction and start-up, as well as simplified programming of control and logic functions using software control blocks built into Fieldbus® devices.

Questions

1. If the temperature of a RTD sensor increases by 150°C, what happens to its electrical resistance?

a| The resistance falls ☐

b| The resistance remains the same ☐

c| The resistance rises ☐

2. What main advantage does a thermistor have over a RTD sensor?

a| It is more accurate ☐

b| It has a higher repeatability ☐

c| It is cheaper to buy ☐

d| It is linear over its complete range ☐

3. What main advantage does a thermocouple have over a RTD sensor?

a| It is more accurate ☐

b| It has a higher repeatability ☐

c| It is cheaper to buy ☐

d| It is linear over its complete range ☐

Answers

1: c, 2: c, 3: c