FLOW CONTROL MANUAL

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FOREWORD

This manual is intended to give an overview of the role of control valves in the behaviour of a process control loop. This overview consists the information on fluid behaviour and valve installed performance as well as the theory and practices of control valve sizing.

Computer based simulation, design and sizing programs have been developing fast during past years and these are well in used by advanced control valve manufacturers to provide and maximize the benefits of process controlling.

Control valve is a critical element in achieving the desired accuracy in process control. A full understanding of the interaction between the control valves and the process is in a key role in further development of the control performance.

This is the area we believe this manual can provide valuable information and help the plant instrumentation, maintenance and process engineers.

We hope this manual provides valuable aid and we appreciate any comments and suggestions for further improvement.

Valmet

1 INTRODUCTION TO PIPELINE FLOW

1.1 General

This section presents several complicated flow problems in very simple form: firstly, flow behaviour in common fluid systems; and secondly pressure behaviour in different piping components and flow induced dynamic torque.

1.1.1 Turbulent and laminar flow in pipes

The flow in conventional control valve installations is almost always turbulent. Laminar flow can occur with very viscous fluids such as lube oil and with very small flow velocities.

Laminar flow can be explained as a microscopic viscous interaction between several layers of fluid. In laminar flow the fluid particles move in parallel paths or streamlines. A particle has only axial velocity along a streamline. Fluid layers slide relative to each other, and the streamline of an individual particle can be predicted

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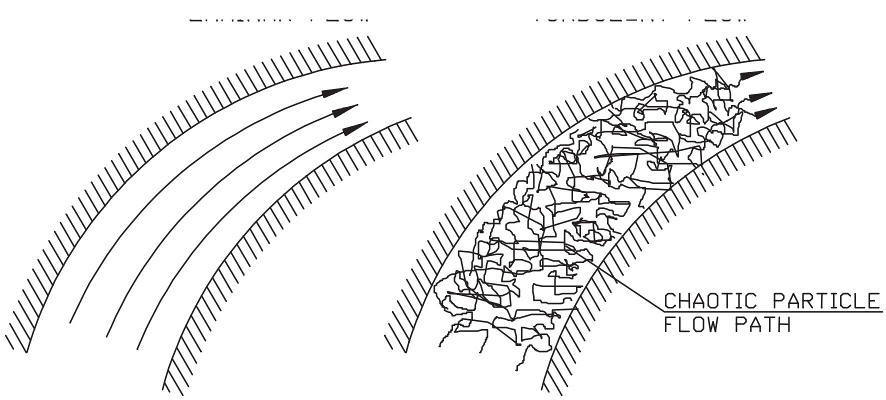


Figure 1. *Laminar and turbulent flow.*

If the velocity is increased above a certain limit the laminar flow field is disturbed and fluid particles start to move in erratic paths. When this happens, the flow becomes turbulent. Transition from laminar to turbulent flow can be predicted by a single parameter, the Reynolds number (Re), as defined in equation (1).

$$Re = V \cdot \frac{d}{v} \tag{1}$$

where d = nominal diameter
V = flow velocity
v = kinematic viscosity

Above a critical value of the Reynolds number the flow goes through a transition to become turbulent. The critical value of Reynolds number is around 3000 for flow in a straight pipe. When the flow is turbulent, part of the flow energy is used to create eddies, which cause increasing pressure losses. Turbulent flow can be described as irregular secondary motion of fluid particles. Secondary motion does not correspond to the principal direction of the flow. The flow path of a single flow particle is irregular.

1.1.2 Velocity distributions

Velocity distribution is affected by several factors, including pipe surface roughness, turbulence level, flow area changes. The shear stress of the fluid creates the velocity distribution in the flow. The flow velocity very close to the pipe wall is zero. This results in a significant change in velocity across the pipe cross section. The laminar and turbulent velocity distributions are significantly different as a result of different shear stresses in the flow. In turbulent flow, the velocity gradient is greater near to the pipe wall than it is in laminar flow. In laminar flow, the center line velocity is higher than in turbulent flow of equal mean velocity (figure 2).

With a smoothly shaped outlet from tank to pipe, the velocity profile is a function of distance. The fully developed profile is constant, if it is not disturbed. In a pipe elbow, the fully developed velocity profile is disturbed by inertia forces in the fluid (figure 3).

A more drastic change in velocity profile can be caused by an orifice or a valve that create a high velocity jet into a flow stream. On the downstream side of an orifice there are strong vortices which cause flow pressure losses. High velocity jets are also accompanied by the noise, especially in the case of compressible flow. The velocity profile after the valve is dependent on the valve type and design, and on valve travel. A fully open ball valve behaves as an extension to the pipe and thus does not disturb the velocity profile in the way that, for instance, butterfly and globe valves do.

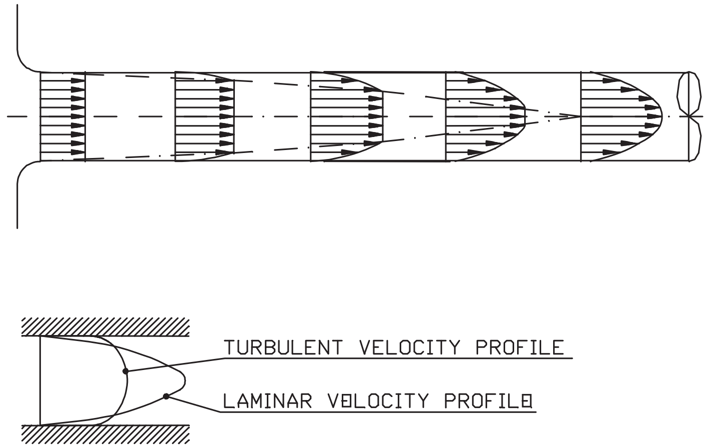


Figure 2. *Development of boundary layer.*

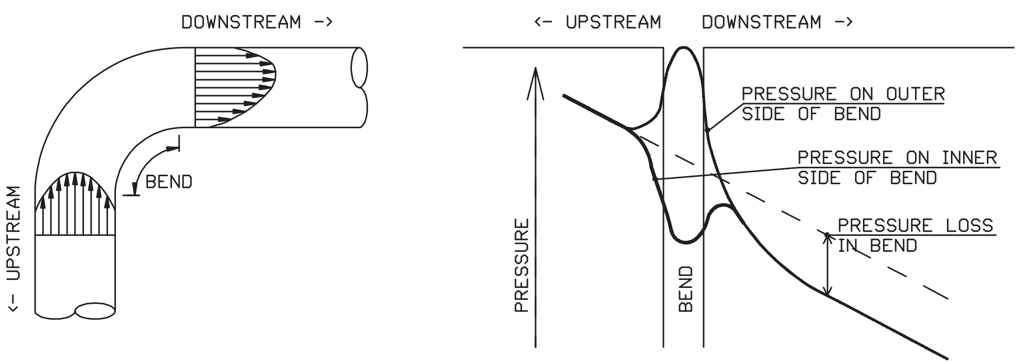


Figure 3. *Pressure in a pipe bend.*

1.1.3 Flow separation

After an abrupt increase in flow area, static pressure at the pipe wall may increase in the direction of flow. This creates a situation in which the forces in the fluid are imbalanced. If this imbalance continues over a sufficient distance, the inner layer of the flow may come to rest and even start to flow backwards. At the point where there is no forward flow the velocity gradient near the wall becomes zero. This point is called the separation point because just beyond it the flow is reversed and the free flow streamlines are diverted from the boundary (figure 4). Large energy losses are associated with separated flow.

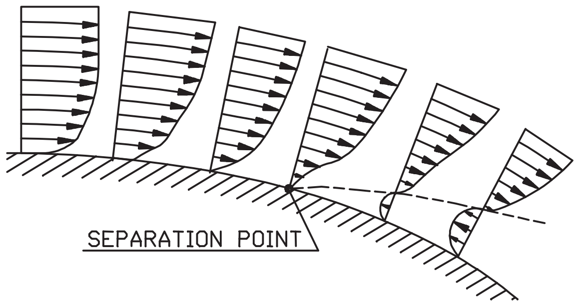


Figure 4. *Flow separation.*

In throttling control, the pressure differences are created through losses caused by separation. In pumps, piping and piping bends flow separation creates unnecessary pressure losses. Improperly designed or installed flange connections, pipe expanders and bends require oversizing the pumps.

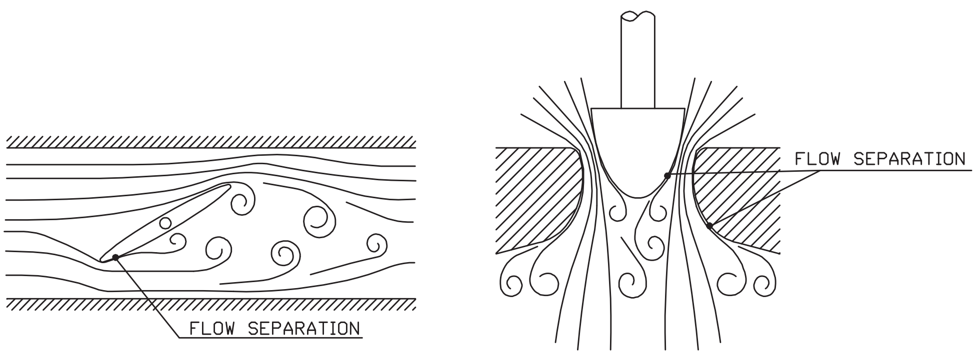


Figure 5. *Flow separation in a butterfly valve and in a flow-to-close single-seat globe valve.*

The idea of a control valve is to control the flow rate by controlling pressure losses across the valve. Flow separation is in practice the physical phenomenon which causes differential pressure across the valve. In future stricter requirements for energy consumption change the design requirements for fluid transport pipelines. This means that permitted pressure losses in systems are going to decrease. Stricter limits on permitted pressure losses will require significant research efforts by engineering companies and valve manufacturers.

1.1.4 Pulsation in flow

Separated flow can produce a large-scale vortex pattern downstream of the valve orifice. The periodic large-scale vortex frequency (f) after a valve plug can be calculated from the equation (2).

$$f = \frac{S \cdot v}{d} \qquad \textbf{(2)}$$

where	S	=	Strouhal number
	v	=	flow velocity
	d	=	obstacle width

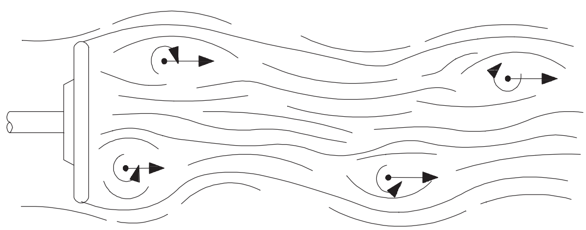


Figure 6. *Vortex pattern behind a poppet.*

The Strouhal number is relatively constant for different Reynolds numbers, and its value depends on the geometry of the obstacle in the flow. Common values for the Strouhal number (S) are from 0.1 to 0.3.

The large-scale vortex shedding forming in certain valve types is illustrated in figure 7. It should be noted that vortex separation can cause disturbances in valve capacity, particularly with certain large-diameter valves in liquid flows.

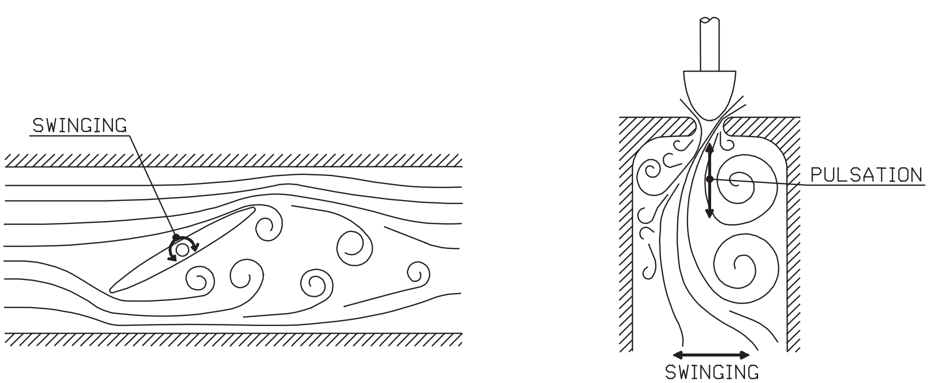


Figure 7. *Periodic vortex pattern after a butterfly and a flow-to-close globe valve.*

The disturbances caused by large-scale vortices due to flow separation can be eliminated or significantly reduced by increasing micro turbulence, or by shaping the flow path. Figure 8 shows examples of valve designs where these principles are used.

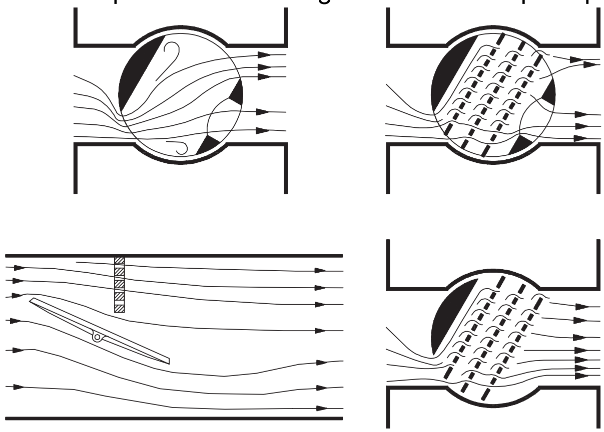


Figure 8. *Examples of flow path modification used in rotary type control valves.*

1.2 Throttling process for incompressible flow

Pipe flows involving valves for throttling control can be studied using simple one-dimensional flow dynamics theory. Incompressible flow through an orifice is taken as a basic example for flow velocity and static pressure behaviour in a valve. Compressible flow can be studied similarly if gas expansion is taken into account.

Bernoulli's equation (4) gives the relation between mean velocity and static pressure. The continuity equation (3), and Bernoulli's equation (4) are all that is required to understand flow through a throttling device.

$$v_1 \cdot A_1 = v_2 \cdot A_2 \tag{3}$$

$$\frac{1}{2} \cdot \rho_1 \cdot v_1^2 + \rho_1 \cdot g \cdot H_1 + p_1 = \frac{1}{2} \cdot \rho_2 \cdot v_2^2 + \rho_2 \cdot g \cdot H_2 + p_2 + \Delta p_h \tag{4}$$

where
- $_1$ refers to upstream conditions
- $_2$ refers to downstream conditions
- v = mean velocity
- p = static pressure
- ρ = density
- Δp_h = pressure loss
- A = flow area
- H = relative height
- g = acceleration of gravity

The continuity equation (3) demonstrates the balance between flow area and mean velocity. As the flow area decreases the mean velocity increases and vice versa. The velocity reaches the highest value at the vena contracta point of an orifice. Vena contracta is the point where the actual flow area reaches its minimum value. Because of contraction caused by fluid dynamics, the area of vena contracta is usually smaller than the physical area of the orifice.

Bernoulli's equation (4) demonstrates the balance between dynamic, hydrostatic and static pressure. The hydrostatic pressure (ρ*g*H) can be assumed constant in the flow through a control valve. As the velocity increases towards the vena contracta point, the static pressure decreases. After the vena contracta, the fluid velocity decreases and static pressure increases. This phenomenon is called pressure recovery. Due to the pressure losses caused by flow separation and vortices, the level of static pressure is decreased across the orifice.

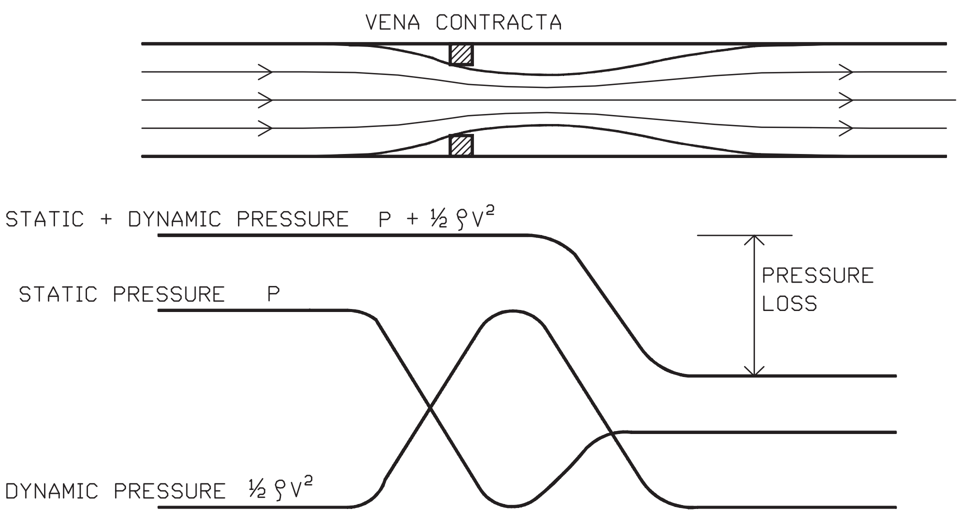


Figure 9. *Static and dynamic pressure around vena contracta point.*

1.3 Defining pressure

In Bernoulli's equation (4) there are three different forms of pressure; static, hydrostatic, and dynamic pressure. Static pressure can be demonstrated with a small cylinder full of fluid and a piston on top of the cylinder, as shown in figure 10. If the piston is pressed downwards with force (F), pressure is exerted onto the cylinder. The value of the pressure equals to force (F) divided by piston area (A). The fluid in the cylinder exerts equal pressure on the walls of the cylinder.

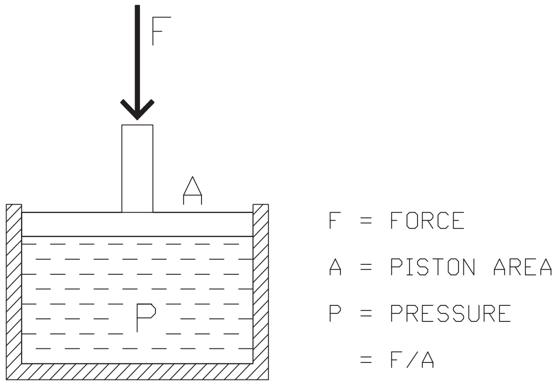


Figure 10. *Static pressure.*

Hydrostatic pressure is caused by gravity. In a tank of liquid, hydrostatic pressure depends on the fluid density and the height of the liquid column. Hydrostatic pressure equals the gravity multiplied by the fluid density multiplied by the height of the liquid column. The hydrostatic pressure is linearly dependent on depth. It is worth mentioning, that hydrostatic pressure does not depend on the area or shape of the tank.

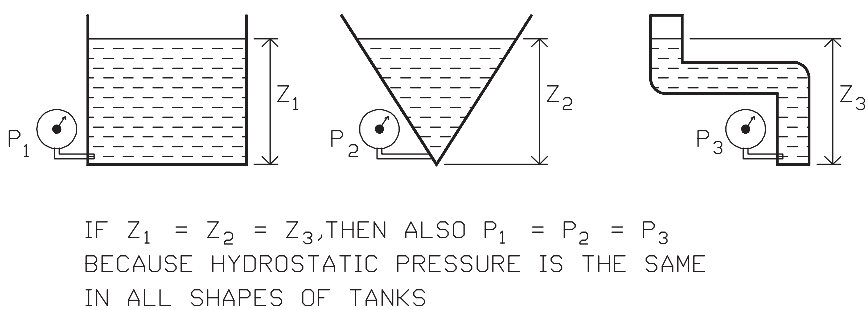


Figure 11. *Hydrostatic pressure.*

Dynamic pressure is related to relative speed of the pressure gauge and the fluid. It makes no difference whether the gauge is moving or the fluid is moving. The dynamic pressure is proportional to the square of the fluid velocity. Static and hydrostatic pressures can be used to measure dynamic pressure and velocity. The measurement can be done within a relatively small range and takes into account only local velocity. To measure the pipe velocity profile or the mean velocity, many measurement points across the fluid cross section are required.

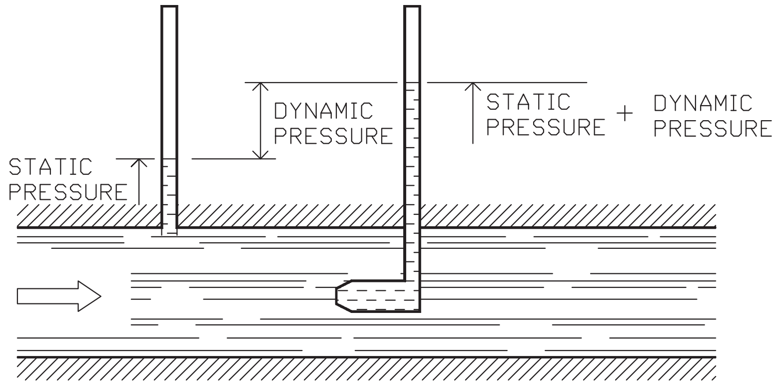


Figure 12. *Dynamic pressure.*

1.4 Pressure loss behaviour of piping components

Pipe components including pumps, control valves, shut-off valves, elbows, joints and the pipe itself have their own typical pressure loss versus flow rate behaviour.

The pump pressure head is typically strongly dependent on the flow rate so that the pressure head decreases when the flow rate increases. Pump manufacturers usually give this dependence in their manuals. In fluid transport systems, the pump is normally sized to operate at maximum efficiency in normal flow conditions. In minimum flow conditions the pump pressure head is higher, and in maximum flow conditions the pressure head is lower than the optimum. This variation is seen in an increase in the pressure at control valve inlet as the flow rate decreases.

In a straight pipe pressure losses are the result of pipe-wall roughness. Pressure loss is proportional to the square of the velocity, as indicated in equation (5), which means that it is also proportional to the flow rate squared if the pipe diameter is constant.

$$\Delta p_h = \frac{1}{2} \cdot \xi \cdot \rho \cdot \frac{L}{d} \cdot v^2 \tag{5}$$

where
Δp_h = pressure loss
ξ = pipe friction factor
ρ = fluid density
L = pipe length
d = pipe inlet diameter
v = fluid velocity

Piping loss increases rapidly as the flow rate increases. Almost all pipe components exhibit a strong dependence of pressure loss on the square of the velocity. For tank inlets and exits, elbows, shut-off valves, orifices etc. pressure loss characteristics can be expressed in the form of equation (6)

$$\Delta p_h = \frac{1}{2} \cdot \xi \cdot \rho \cdot \frac{L}{d} \cdot v^2 \tag{6}$$

where ξ = pressure head loss coefficient

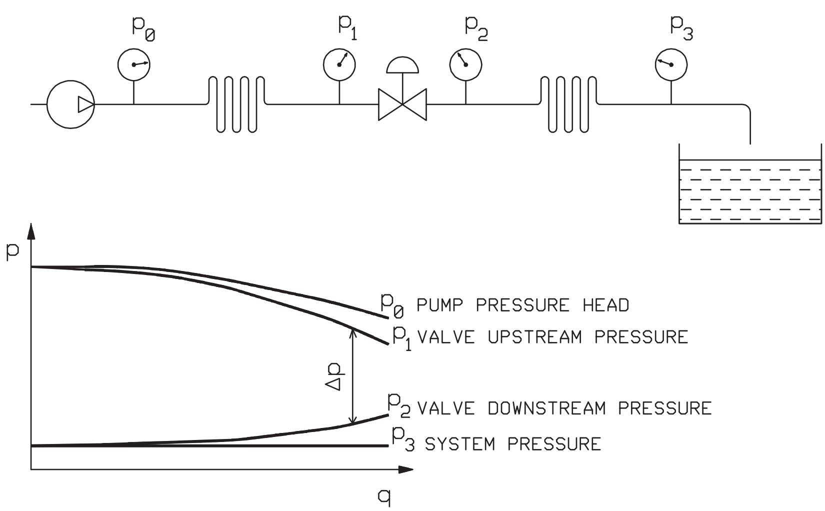


Figure 13. *Pressure losses in a piping system.*

In fluid transport systems, the differential pressure across a flow control valve is strongly dependent on the flow rate. Under normal flow conditions the control valve typically accounts roughly one third of the total pressure loss of the system. In the future, due to energy saving requirements, this will change so that control valves take smaller portion of the total pressure drop.

1.5 Valve dynamic torque

Valve dynamic torque, which tends to close or open the valve depending on the type of valve, is caused by asymmetric pressure distribution on the surface of the throttling element. Theoretical determination of the dynamic torque requires that the pressure distribution on the throttling element surface is known. Figure 14 shows a principle drawing of a pressure distribution that results in dynamic torque in a ball valve. In practice, dynamic torque (M_d) is usually calculated from experimentally defined equation (7).

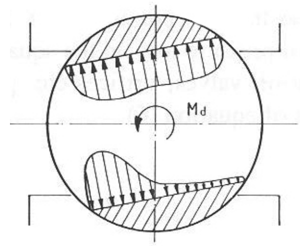


Figure 14. *Ball valve dynamic torque is formed by asymmetric pressure distribution on the inner ball surface.*

$$M_d = C_d(h) \cdot D_N^3 \cdot \Delta p \tag{7}$$

where M_d = dynamic torque
$C_d(h)$ = experimental dimensionless torque coefficient as a function of relative opening travel (h)
D_N = valve diameter
Δp = pressure differential across the valve

The dimensionless torque coefficient (C_d) is determined experimentally using laboratory measurements. Figure 15 shows dimensionless torque coefficient (C_d) for a ball valve measured in a flow control laboratory. As can be seen from figure 15, the torque coefficient depends strongly on the relative travel (h) of the valve. Usually value of dynamic torque reaches a maximum when the valve is close to the fully open position. The dynamic torque is usually insignificant for small valves due to a small valve diameter D_N. In butterfly and ball valves, dynamic torque differs from the torque caused by friction, because dynamic torque tends to close the valve.

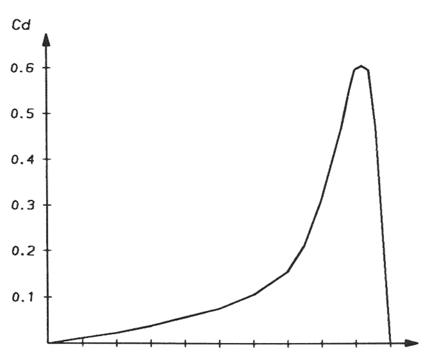


Figure 15. *Dynamic torque coefficient C_d for a ball valve.*

2 CONTROL VALVE INSTALLED PERFORMANCE

2.1 General

This chapter introduces a process pipeline model and calculation procedure that make it possible to predict control valve installed performance accurately and quickly using a valve sizing and selection software. Choked flow conditions are taken into account for all calculated points, a factor that was listed as a limitation in previously presented models. The results and their utilization are discussed in detail, with practical guidelines on how to correct nonlinear installed flow characteristics using a digital controller signal modification block.

It is also important to note that control valve installed accuracy is the method that should be used to judge control valve accuracy, not just the mechanical accuracy. The installed gain multiplied by the mechanical accuracy gives the installed flow accuracy.

2.2 Operating conditions

Successful control valve sizing and selection depend on knowing the actual process conditions in the system in which the valve is to be installed. It is known that distinct information on operating conditions very seldom exists. The more assumptions one has to make on flow conditions, the less accurate the control valve sizing is going to be. Fortunately, when a few flow conditions are known accurately, an algorithm can be found that can model the behaviour of a given control valve for the whole flow range.

2.3 Selecting a control valve

The selection of a control valve is divided into two parts. First, the mechanical selection in which suitable valve style, materials and pressure rating are picked to guarantee safe and reliable performance according to good engineering practices, local laws and regulations etc. This selection should be made using bulletins and technical information specific for each valve type. Secondly, the sizing of a selected valve, in which the size of the given valve type is determined, while trying to prevent unwanted phenomena like excessive noise or liquid cavitation. After the valve size has been determined, the installed performance of the valve to phase out poor control performance like hunting, excessive slow or fast response and poor accuracy is further predicted.

This control valve selection here concentrates on sizing and installed performance considerations. It should be noted that these calculations do not, by themselves, guarantee the selection of a correct valve. Therefore, careful mechanical and material evaluations should always precede sizing calculations.

2.4 Control valve flow characteristics

In the past, selection of control valves has been based largely on approximate estimation methods and practical experience. Modern sizing and selection software enables faster and more accurate calculation methods to select a control valve with optimum controllability and accuracy for each individual process application. The method is based on modelling or predicting the actual flow characteristic and gain of the installed valve.

2.4.1 Inherent flow characteristic

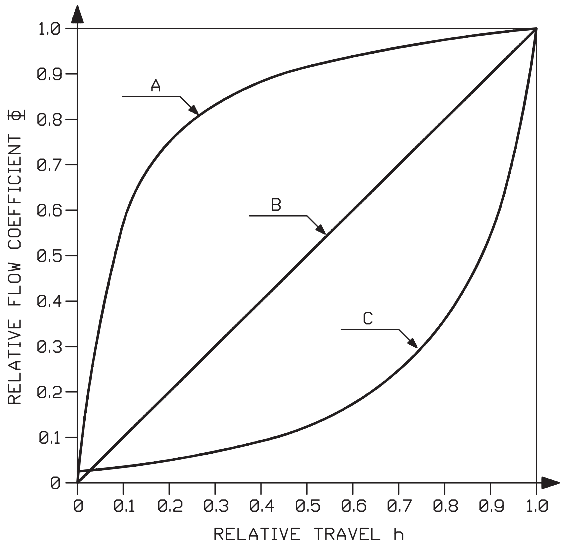
Selection of the optimum size and type of control valve starts with the valve's inherent flow characteristic. For this reason, all control valve types must be laboratory tested to determine their exact inherent flow characteristics.

Valve inherent flow characteristic are defined so that the pressure differential across the valve (Δp) is kept constant. As the differential pressure (Δp) is constant, the flow rate (q) through the valve is proportional to the valve flow coefficient (C_v), as expressed in the simplified equation (8).

$$q = C_v \cdot \sqrt{\Delta p} \qquad \textbf{(8)}$$

where q = flow rate through the valve
C_v = valve flow coefficient
Δp = pressure differential across the valve

Because the valve flow coefficient (C_v) reflects the effective flow crosssection of the valve, the valve inherent flow characteristic shows how the effective flow crosssection changes as a function of relative travel (h).



A QUICK OPENING
B LINEAR (Φ = h)
C EQUAL PERCENTAGE (Φ = Φo * exp(c*h))

Figure 16. *Valve inherent flow characteristics.*

Figure 16 shows the most common valve inherent flow characteristics. The relative flow coefficient (Φ) is determined as shown in equation (9).

$$\Phi(h) = \frac{C_v(h)}{C_v(1,0)} \quad \textbf{(9)}$$

where	$\Phi(h)$	= relative flow coefficient as a function of relative travel
	$C_v(h)$	= valve flow coefficient at relative opening h
	$C_v(1.0)$	= valve flow coefficient when valve is fully open

The inherent flow characteristic is the shape of a flow curve through the valve with a constant pressure drop across the valve. Note that in any case where process piping is attached to the valve, the piping pressure loss, a function of flow rate, will cause the valve pressure drop to vary as a function of flow rate, even if the pressures at the source and receiver are constant.

A. Quickopening inherent flow characteristic

A flow characteristic in which a flow close to a maximum flow is reached with a very small opening. Most commonly used in globe valves for onoff services and in cases where only a limited stroke is available due to the nature of actuator, as in the case of pump governors.

B. Linear inherent flow characteristic

A flow characteristic in which the valve relative opening directly correlates to the percentage flow, e.g. a 50 % open valve gives 50 % of maximum flow, with a constant pressure drop across the valve. If the pressure drop across the valve remains absolutely constant independent of the flow, this will be an optimal inherent flow characteristic, because in that case the inherent flow characteristic equals the installed flow characteristic.

C. Equal percentage inherent flow characteristic

A flow characteristic in which equal increments in the valve opening cause a constant percentage increase in flow with a constant pressure drop across the valve. It is designed to linearize the installed flow characteristic in normal control valve applications, where the amount of pressure drop available to the control valve decreases with the opening of the valve, due to the increase in pressure losses in other parts of the system, such as pipes, pumps, heat exchangers etc.

2.4.2 Installed flow characteristic

Under operating conditions a control valve is part of a process pipeline. The differential pressure across a valve is seldom constant in the valve travel range because dynamic pressure losses in the flow cause the valve inlet pressure to fall and the outlet pressure to rise as the flow rate increases. For an installed valve, the dependence of the flow rate (q) on the travel (h), i.e. the shape of the installed flow characteristic curve, is therefore a function of the process pipeline and of the inherent flow characteristic of the valve. The changes in differential pressure that take place across a control valve installed in a process pipeline are illustrated in figure 17.

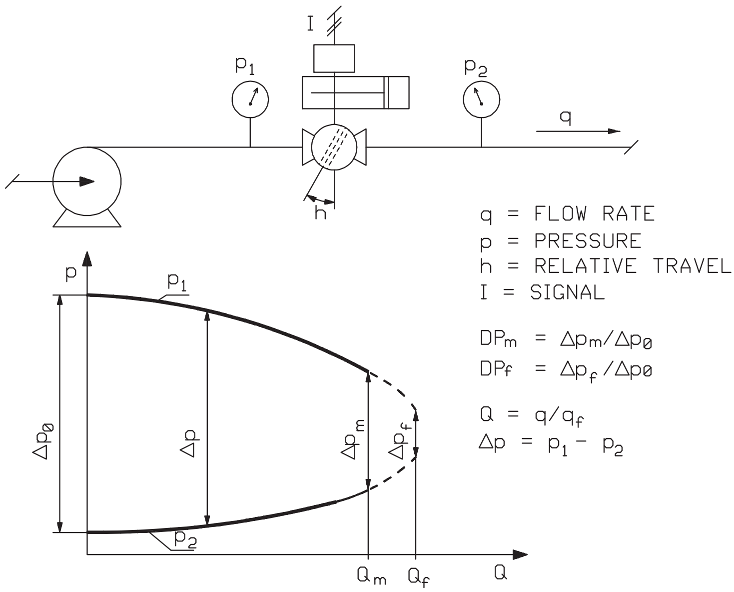


Figure 17. *Change in effective differential pressure across the valve due to a change in the flow rate.*

The character of a process pipeline is described by the process pressure ratio factors DP_f or DP_m. The subscripts refer to situations in which the valve is fully open (subscript $_f$) or opened to allow the maximum flow rate (subscript $_m$) required by the process. The process pressure ratio factor DP_m is the differential pressure across the valve at the process maximum flow rate divided by the differential pressure across the valve when the valve is closed, as expressed in equation (10).

$$DP_m = \frac{\Delta p_m}{\Delta p_0} \qquad \textbf{(10)}$$

where Δp_m = differential pressure across the valve at the maximum designed flow rate

Δp_0 = differential pressure across closed valve

The character of the process pipeline can be determined with a sizing and selection software when at least two different flow conditions in the process are known, or if the process pressure ratio factor DP_m, which describes the character of the pipeline, and the maximum flow conditions are known.

The process pressure losses are relative to the square of the flow velocity, which corresponds to the square of the flow rate. This basic assumption facilitates the creation of a model for pressure variations at the valve inlet and outlet.

Figure 18 shows an installed flow characteristic curve calculated using a sizing and selection software for an eccentric rotary plug control valve in a process application.

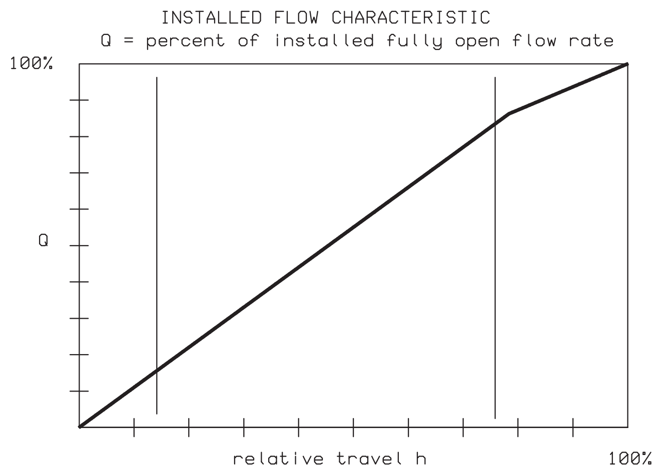


Figure 18. *Installed flow characteristic of an eccentric rotary plug control valve.*

2.4.3 Installed gain

The quality of the installed flow characteristic curve with respect to valve controllability and accuracy can be examined by means of a valve gain curve. The valve gain curve describes the changes that take place in the slope of the installed flow characteristic curve for different amounts of valve travel. The gain of an installed valve is the change in relative flow rate ($dQ_p = d(q/q_m)$) divided by the change in relative travel (dh), as expressed in equation (11).

$$G = \frac{dQ_p}{dh} \tag{11}$$

The installed gain gives the rate of change in flow as a function of the change in signal. A gain of 1 means that a 1 % change in valve opening causes the flow to change by 1 % of the maximum flow. A gain of 2 means that a 1 % change in the signal causes a 2 % change in the relative flow rate (dQ_p), and so forth.

The change in flow rate (dQ_p) can be expressed as in equation (12).

$$dQ_p = G \cdot dh \tag{12}$$

In other words, the change in flow rate (dQ_p) is the gain (G) multiplied by the change in valve travel (dh).

From equation (12) it can be seen that variation of the gain (G) in the required flow range causes different relative flow changes (dQ_p) for the same relative signal change. This is usually undesirable for process controllability.

The installed valve gain is the starting point for selecting an optimum control valve size and an inherent flow characteristic for a certain process application. For the standard PID controllers in most control loops the best possible inherent flow characteristic is the one that gives closest to constant installed gain for the required flow range. In this case the controller's parameter tuning can be kept optimal and unchanged in spite of variations in the load in the process operating range. In applying the above rules it must, however, be noted that only a full dynamic analysis of the control loop is sufficient to include all the nonlinearities of the process and equipment and to absolutely guarantee the choice of an optimum control valve. This kind of analysis is feasible for a control valve simulation program.

The rule of thumb for permissible limits for the installed gain is that a change in gain larger than 2 (equation (13)), or a relative gain smaller than 0.5 (equation (14)) should be avoided in the process operating range.

$$\frac{G_{max}}{G_{min}} < 2 \tag{13}$$

$$G \geq 0{,}5 \tag{14}$$

If the gain is too low or too high, or if it changes considerably in the process operating range, process control will generally become very difficult. Moreover, too high a gain also means problems for the control accuracy of the control valve because the relative flow rate error (ΔQ_ε) for a control valve is the gain (G) multiplied by the relative travel error (Δh_ε) of the control valve.

Figures 19 and 20 illustrate the importance of selecting the proper size control valve for each application. An oversized control valve causes high gain with small flows resulting in poor control accuracy and possible hunting, as shown in figure 19. Selecting a smaller valve will lower the gain, resulting in high accuracy and stable control, as shown in figure 20.

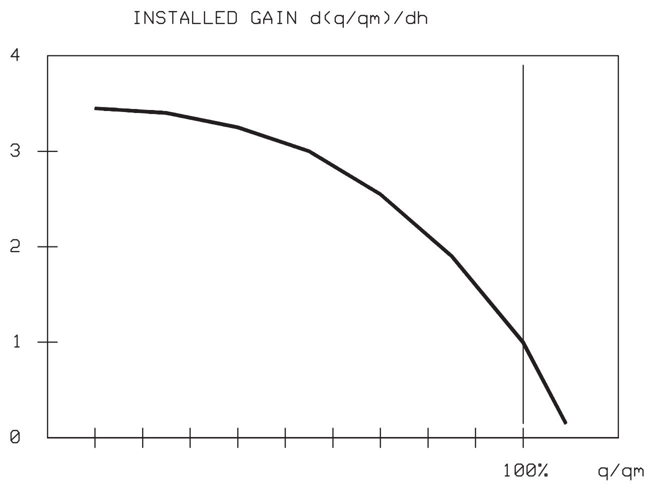


Figure 19. *Installed gain of an oversized control valve.*

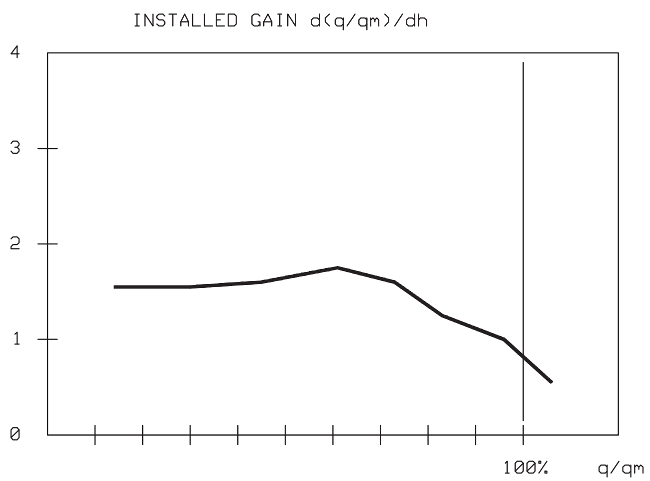


Figure 20. *Installed gain of a proper size of control valve.*

Figure 21 shows a sample installed gain curve for a new type of eccentric rotary plug control valve. The figure demonstrates that an almost constant gain is achieved on the inherent flow characteristics of the eccentric rotary plug control valve. This is often the optimum gain in the process operating range. Furthermore, a smooth gain results in excellent control accuracy.

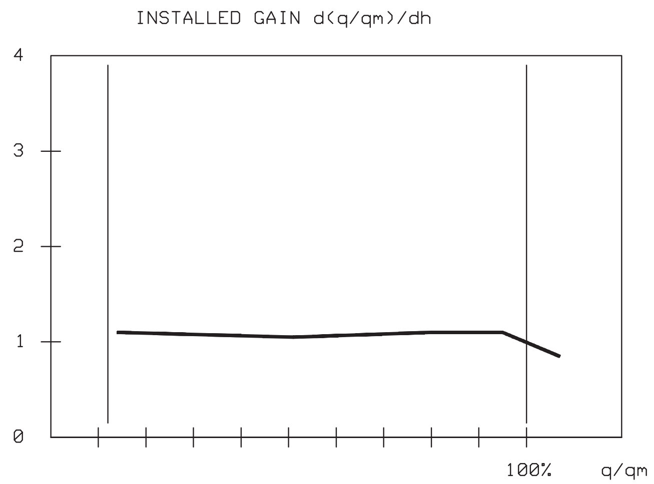


Figure 21. *Installed gain of an eccentric rotary plug control valve.*

2.4.4 Calculation methods

Procedure

With a sizing and selection software it is possible to predict the installed control valve flow characteristic and gain curves, if at least two separate sets of flow conditions or if the process maximum flow and DP_m factor are known.

The program accurately predicts installed control valve performance, if the following conditions are true:

Condition 1: The process is such that a specific flow corresponds to a specific pressure drop across the valve.

Condition 2: Pressure drop changes across the control valve are dependent on pressure losses in the piping system such that the sum of those pressure losses is proportional to the square of flow velocity.

The program uses following procedures to calculate the installed flow characteristic and gain:

- Using given flow data the program generates a process model for the pressures, and checks whether conditions 1 and 2 above are fulfilled. The process model for the valve inlet and outlet pressures is created using 'the sum of least squares' method such that the derivative equals zero at the origin of the pressure curves.
- The program predicts the flow through a fully open control valve (q_f) using the process model to determine pressure drop across the fully open control valve. Possible choked liquid flow, determined by pressure recovery factor (F_L), or critical gas flow, determined by pressure drop ratio factor (x_T), are taken into account when determining flow through a fully open control valve (q_f).
- The valve pressure drop, opening and predicted noise level for the whole flow range are calculated at intervals of 5 % of the maximum flow (q_f). At each of the calculation points possible choked flow is taken into account in the calculations. The installed flow characteristic displayed on the screen is the relative flow rate (q/q_f), which is the actual flow rate divided by the flow in a fully open valve, given as a function of relative travel (h).
- The installed control valve gain is the slope of the installed flow characteristic curve defined using finite difference method. It is important to note that the installed gain curve is calculated and drawn as a function of the maximum process flow (q/q_m) and not as a function of the flow with the valve fully open (q/q_f). This is important, as it allows different valve types and sizes to be compared based on process requirements, and not just from the valve viewpoint. It is worth mentioning, that this method increases the gain as compared with the gain presented as a function of flow in a fully open valve.

Process model and DP_m selection

In order to calculate the installed control valve flow characteristic and gain, a model of the process according to which the pressures involved in the process piping change has to be available, since the program determines the process model based on flow data. In most cases the model of process pressures used is such that the pressure drop across the control valve decreases with increasing flow caused by opening the control valve.

The pressure behaviour of the process can be estimated accurately enough for practical purposes, when the pressure conditions (upstream pressure, pressure drop across valve) and flow rates are known for at least two flow conditions, such as the maximum and minimum flows. If only the maximum flow for the process (q_m) and the corresponding pressure drop (Δp_m) are known, the process model can be found by using the process pressure ratio DP_m, as in figure 22.

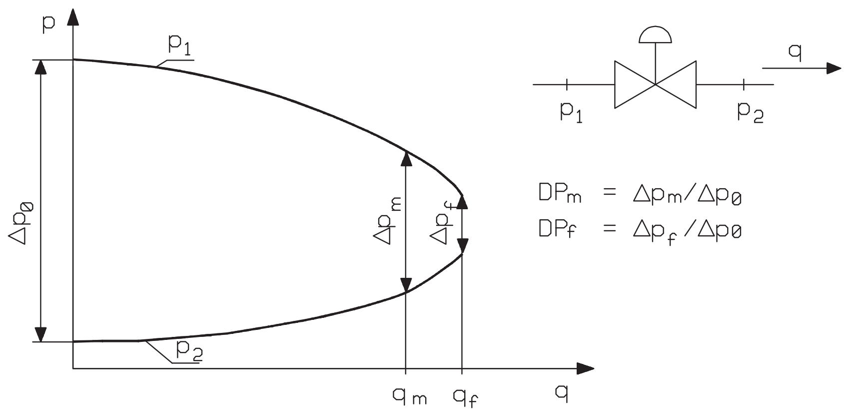


Figure 22. *The process pressure model.*

DP_m is the valve pressure loss portion of the total dynamic pressure loss of the system with maximum process flow. The DP_m information should thus be easily available from system pressure loss calculations.

The program is able to calculate the pressure process model using two or more given flow conditions, if these flow conditions comply with conditions 1 and 2 of the procedure. Figure 23 gives examples of defined process models the program can handle, and that of undefined process models that it cannot.

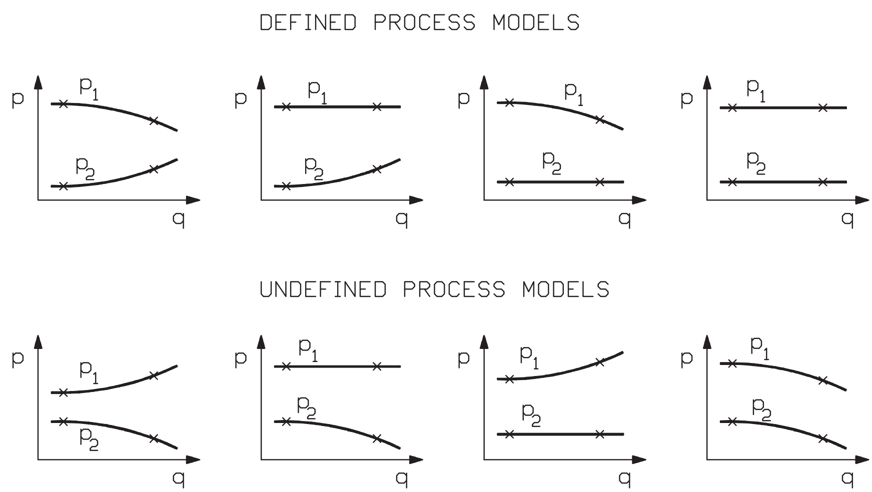


Figure 23. *Examples of defined and undefined process models.*

The process pressure ratio DP_m is affected by the dynamic pressure loss in the piping. The piping pressure loss is generated by the length of the pipe, the diameter, surface roughness, and individual resistances of piping components. Additionally, the pressure drop across a control valve depends on the flow velocity and piping boundary conditions, e.g. if the fluid is a liquid, the pump curve needs to be taken into account when evaluating the change in pressure drop across the valve.

DP_m ratio gives the program user a chance to evaluate and optimize piping structure options and the effect different valves have on overall process total function. The DP_m ratio can be optimized such that a minimum amount of energy is used in the control valve, leading to significant saving in the energy cost of pumping. The following A and B give examples of the selection of DP_m ratios for liquids and gases.

A. Liquid flow

In practice most process piping systems are now designed such that the control valve pressure loss at maximum designed flow is roughly one third of the total dynamic pressure loss in the piping. In that case, DP_m ratio as a rule of thumb for liquid piping is as presented in equation (15).

$$DP_m = 0{,}30 \tag{15}$$

As mentioned before, piping designers are an invaluable asset in calculation the real DP_m factor, particularly for new piping systems. Old systems can be evaluated using common sense to determine a DP_m ratio accurate enough for present purposes. Figure 24 presents an estimation of DP_m ratio in two different kinds of process piping.

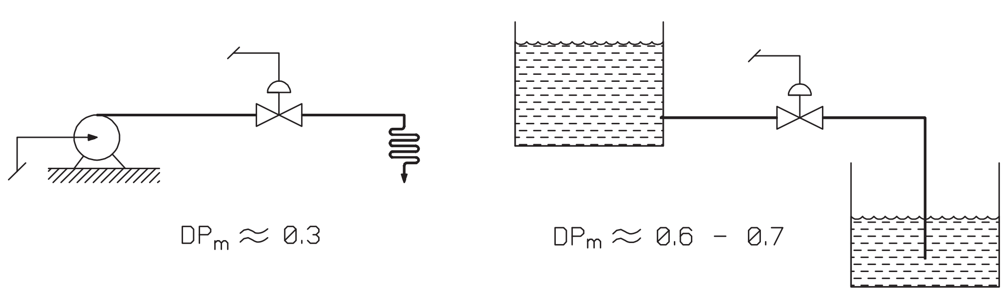


Figure 24. *Evaluating DP_m ratio for liquid flow.*

B. Gas flow

For compressible flows, such as those of gases and steam, the pressure drop changes across a control valve are usually smaller than those for liquid flows. The difficulties involved in presenting 'rules of thumb' for gas and steam flows are firstly, that there is huge variation in different process and piping systems and, secondly, that fluid compressibility, particularly at large flow velocities, has a big effect on the pressure drop across a control valve. Equation (16) gives 'a rule of thumb', which can be used when no accurate information is available.

$$DP_m = 0{,}50 \tag{16}$$

Trying out several possible DP_m ratios to view the effect this has on the installed flow characteristic and gain is suggested in such cases. Some example estimates for DP_m ratios for gas or steam flows in different process piping systems are given in figure 25.

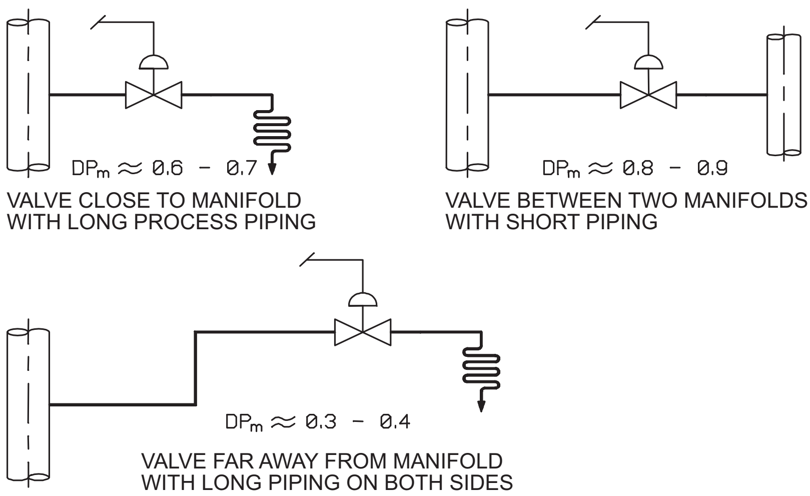


Figure 25. *Evaluation of DP_m ratio for gas and steam flows.*

2.4.5 Control valve accuracy

A control valve in a control loop determines much of the quality of the overall control loop function. Specific attention should thus be paid to control valve selection in order to get the full benefits of the extensive product development that has occurred in other components of the control loop, such as measuring devices and DCS process control systems.

A control valve is the final control element, a mechanical device that effects the changes in flow or pressure deemed necessary by the rest of the system. Therefore, the control valve accuracy is a main element in defining the accuracy of the whole control loop.

The relative flow error of a control valve (ΔQ_ε) is the valve gain (G) multiplied by the relative valve travel error (Δh_ε), as expressed in equation (17).

$$\Delta Q_\varepsilon = G \cdot \Delta h_\varepsilon \tag{17}$$

where
- ΔQ_ε = relative flow rate error
- G = valve installed gain = dQ_p/dh
- Q_p = relative flow rate
- dQ_p = change in relative flow rate
- Δh_ε = relative valve travel error

The relative flow error of a control valve (ΔQ_ε) can also be expressed as in the following equation (18).

$$\Delta Q_\varepsilon = \frac{\Delta q_\varepsilon}{q_m} \tag{18}$$

where
- Δq_ε = flow rate error
- q_m = maximum flow rate required by the process

The installed gain (G) in the equation (17) is the control valve installed gain, which can be calculated by a sizing and selection software.

The accuracy of the flow controlled by a control valve is the mechanical error of the control valve assembly multiplied by the installed flow gain. A figure for the mechanical error alone does not indicate the accuracy of the flow being controlled.

2.4.6 Control valve characterization

The inherent flow characteristic of a control valve is traditionally chosen so that in the process operating range the controller tuning can be kept constant in spite of load changes. If the process being controlled is nonlinear, the control valve flow characteristic should be inversely nonlinear. Thus the control valve flow characteristic will compensate for a nonlinear process.

The rapid development of process control systems has, however, made it possible to apply highly developed control strategies and algorithms. In this case, there is a tendency to select various subsections of the control loop to be as linear as possible in the process operating range. For control valves this can be done by selecting a valve type and size that give a linear installed flow characteristic and constant gain in the given process flow range.

In certain applications, due to accuracy requirements or process conditions, the desired installed flow characteristics or gain cannot be achieved with the normal range of valves. In this case the flow characteristic of the valve can be modified using characterization, as illustrated in figure 26.

Most control systems assume that the installed flow characteristic of the control valve is linear. Therefore, for optimal performance of the control valve measures should be taken to ensure installed flow characteristic to be as close to linear as possible by selecting the valve style and type accordingly.

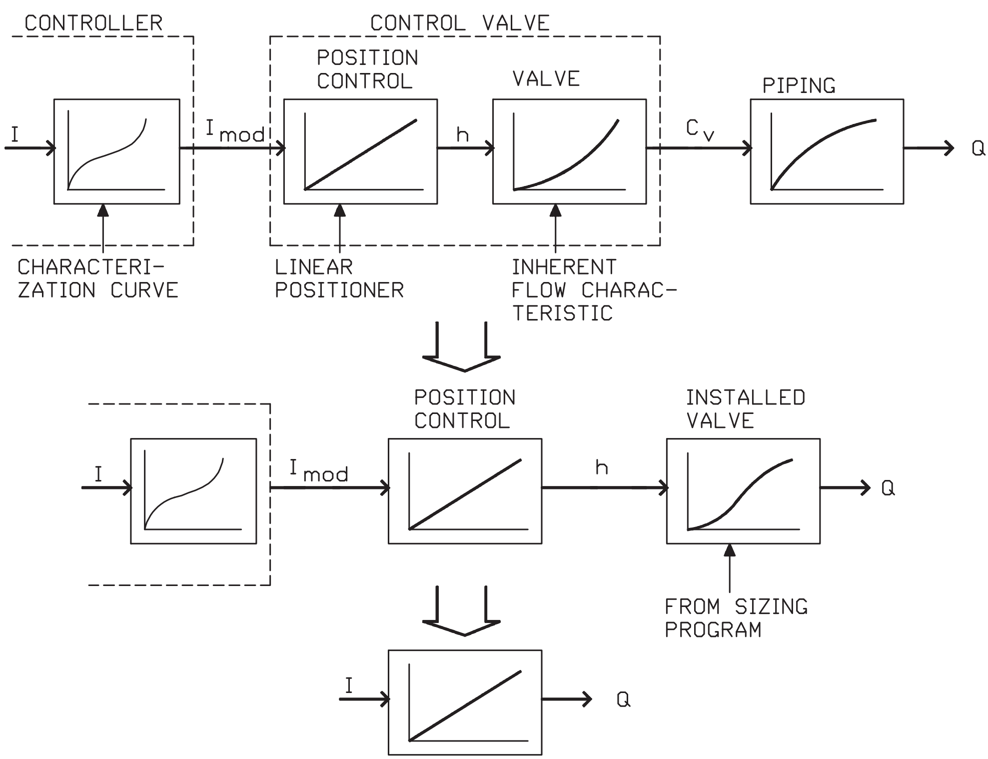


Figure 26. *Control valve characterization by modifying controller output.*

Characterization of control valve performance can basically be achieved in four different ways, namely by modifying:

1. the controller output, as in figure 26
2. the signal in the positioner
3. the valve control orifice or the plug or both
4. the slope of the analog positioner feedback cam or use of the characterization feature of the digital positioner.

Figure 26 shows the control valve characterization performed by controller output modification in the case where a constant gain is required for the control valve. In this case the control valve is fitted with an electro-pneumatic positioner with a linear feedback cam.

The controller characterization curve in figure 27 can be plotted using a sizing software, when the installed flow characteristic graph relative flow rate (Q) is replaced by the input signal (I) for the linearization block, and relative travel (h) is replaced by the output of the linearization block (I_{mod}):

$Q \rightarrow I$

$h \rightarrow I_{mod}$

The linearization curve derived this way can be fed into the controller point by point, as presented in figure 27.

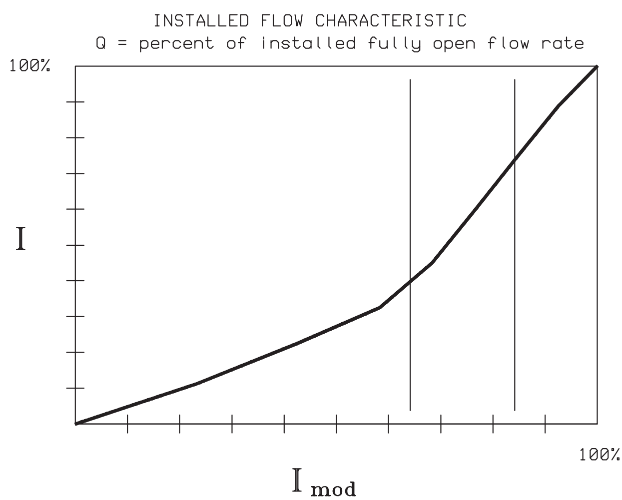


Figure 27. *A characterization curve produced from a sizing software.*

Modern process controllers are capable of linearizing the control valve installed characteristic by signal modification. This method, which is applied by using a graph, like the one shown in figure 27, provides accurate linearization for the particular application. Positioner cam modification simply distorts the valve inherent flow characteristic without taking into account the actual requirements of the specific system being considered.

Modification of control valve characteristics by modifying the positioner feedback cam can be studied using the block diagram in figure 28.

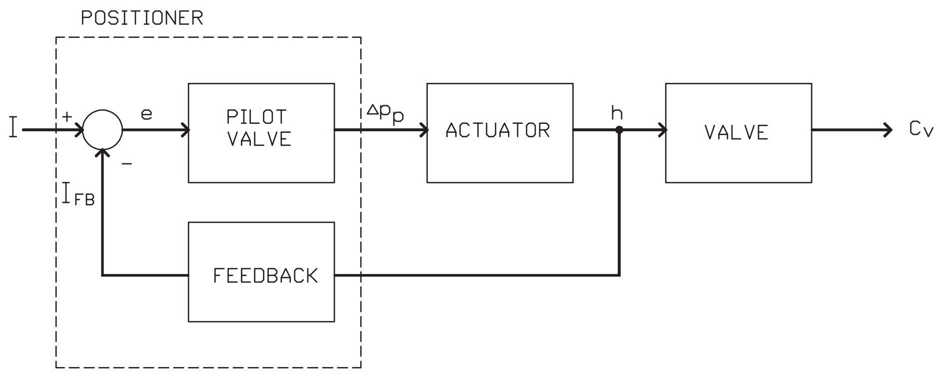


Figure 28. *Block diagram for a control valve.*

Control valve function can be divided into subsections, as in the block diagram in figure 28. The function of each block of the control valve can be described in terms of its individual gain (G), the ratio of the change in output to input. The figure 29 shows the position control block diagram in which G_{PV} is pilot valve gain, G_{AC} is actuator gain and G_{FB} is feedback loop gain. In figure 29, a linear feedback cam is used. A linear feedback cam gives the constant feedback gain G_{FB}=1.

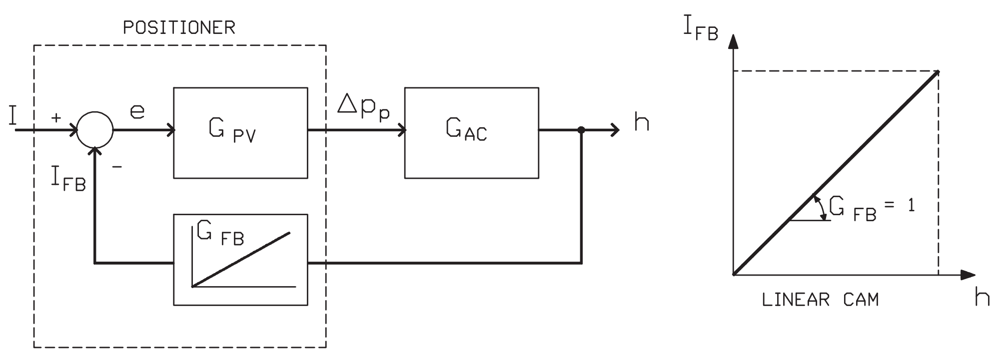


Figure 29. *Linear position control.*

The behaviour of a feedback system can be studied through its transfer function. Figure 29 illustrates the equation (19) for the position closed loop gain.

$$G_{PC} = \frac{h}{I} = \frac{G_{PV} \cdot G_{AC}}{1 + G_{FB} \cdot G_{PV} \cdot G_{AC}} \tag{19}$$

where G_{PC} = total position control gain
G_{PV} = pilot valve gain
G_{AC} = actuator gain
G_{FB} = constant feedback gain

As shown in equation (19), the gain in position control depends on the gains in the individual blocks of the system.

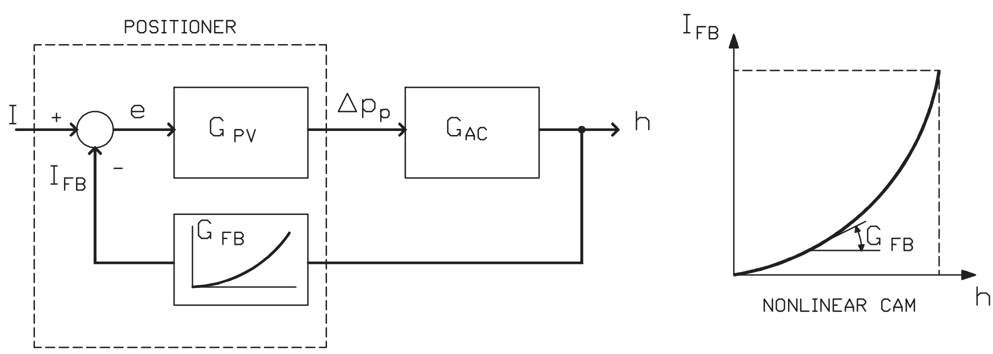


Figure 30. *Nonlinear position control.*

Figure 30 represents nonlinear position control performed by modifying the positioner feedback cam.

The problem with positioner cam modification is that the correction is made in the mechanical feedback loop of the positioner. Thus, the correction is made after the point where it was required. Cam modification is thus only suitable for very slow control loops. Moreover, it has the following disadvantages:

- The control valve operates very rapidly in one flow range and very slowly in another flow range, due to the feedback gain (G_{FB}) change with different valve openings (h); in other words, the control loop time constant changes depending on the valve position.
- Control valve position control can become unstable in a certain opening range, due to the stability criterium given in equation (20).

$$1 + G_{FB} + G_{PV} \cdot G_{AC} \neq 0 \tag{20}$$

If G_{FB} changes, there can be an opening for which equation (20) equals 0 and thus the position control gain approaches infinity: $G_{PC} \rightarrow \infty$.

- Nonlinear, mechanically modified feedback also reduces the static performance of a control valve, because the error in position control (Δh_ε) changes as a function of the input signal as shown in figure 31.

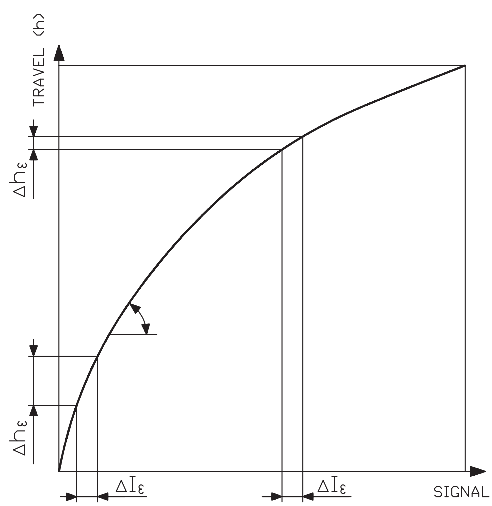


Figure 31. *Nonlinear position control error.*

If characterization of the control valve is done using controller output modification instead of nonlinear mechanical feedback, the situation changes considerably. The situation can now be characterized by the block diagram in figure 32.

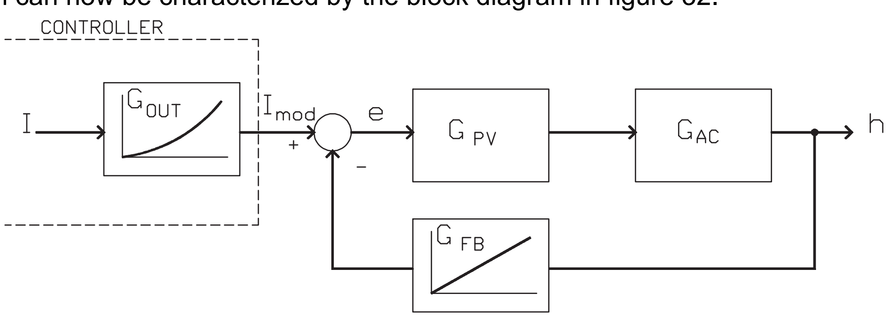


Figure 32. *Control valve characterization by controller output signal.*

The transfer function for control valve characterization using the controller output signal can now be found from figure 32.

Equation (21) shows that with the nonlinear characterization block ($G_{OUT} \neq 1$) of a controller output and linear feedback cam ($G_{FB} \neq 1$) there is a small chance for the position control to become unstable ($G_{PC} \rightarrow \infty$).

$$G_{PC} = \frac{h}{I} = G_{OUT} \cdot \frac{G_{PV} \cdot G_{AC}}{1 + G_{FB} \cdot G_{PV} \cdot G_{AC}} \quad \textbf{(21)}$$

In practice, characterization by modifying the controller output means:

- The installed characteristic modification is done on the signal level very rapidly compared with changes on the pneumatic and mechanical level.
- The error in positioner is constant, and does not change according to the opening.
- The chance for position control instability is small.

3 LIQUID FLOW

The flow of incompressible fluids through control valves has been well defined by universal, standardized sizing equations (ISA/IEC). A description of these equations and the basic fluid mechanics of incompressible flows forms the first half of this chapter. In the second half, there is explained the phenomenon, and the prevention of cavitation in a fresh way, providing new insight into theories that are well-known in the industry.

Determination of levels of damaging cavitation from the predicted noise level is a starting point for making recommendations on when cavitation prevention techniques are required. Methods of abating cavitation using both valve trim variation and external means like back-pressure baffles are explained further on in the text.The chapter ends with hydrodynamic noise prediction, which is presented in the form of the equations used. Guidance on maximum flow velocities for liquid flow valves is given in the last paragraph.

3.1 General

Liquid flow is an incompressible type of flow. A typical process installation for a liquid flow control valve is illustrated in figure 33.

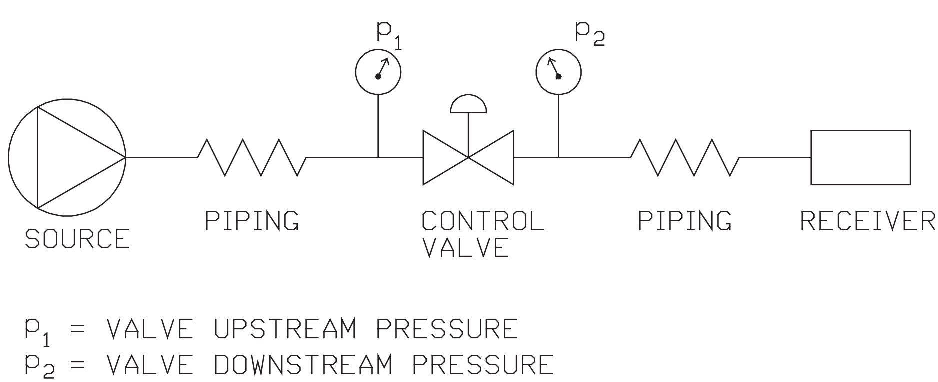


Figure 33. *Typical liquid control valve installation.*

The following part of this manual concentrates on the throttling process between upstream pressure p_1 and downstream pressure p_2, as shown in figure 33, where upstream pressure p_1 is the pressure measured by a tap located two pipe diameters upstream of the upstream valve flange, and downstream pressure p_2 is the pressure measured by a tap located six pipe diameters downstream of the valve downstream flange (per std. IEC 60534/ISA S75).

To displaying the general behaviour of flow through a control valve itself, the valve is simplified to an orifice in a pipeline, as shown in figure 34. Naturally, continuity equation (3) and Bernoulli's equation (4) are also valid in this context.

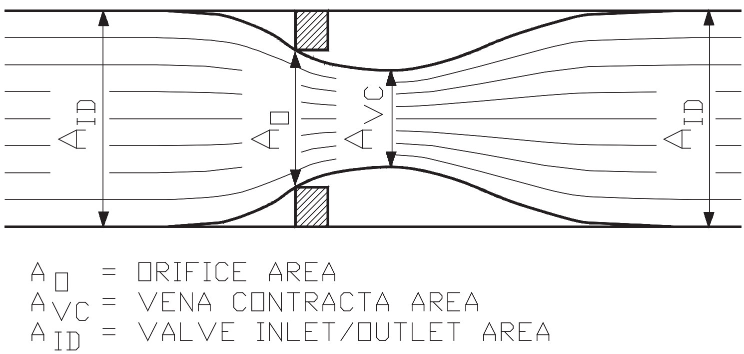


Figure 34. *Flow through an orifice.*

Figure 34 shows the change in the cross-section area of the actual flow when the flow goes through a control valve. In a control valve the flow is forced through the control valve orifice, or a series of orifices, by the pressure difference across the valve. The actual flow area is smallest at the point called the vena contracta (A_{VC}), the location of which is typically a little downstream the valve orifice, but can be extended even into the downstream piping, depending on pressure conditions across the valve, and on the valve type and size.

Due to the reduction in flow area a significant increase in flow velocity has to occur to give equal amounts of flow through the valve inlet area (A_{ID}) and vena contracta area (AV_C). The energy for this velocity change is taken from the valve inlet pressure, which gives a typical pressure profile inside the valve (figure 35).

The pressure inside the valve drops as the effective flow area is reduced, up to the vena contracta point, which is the smallest flow area available to the flow inside the valve. After reaching the vena contracta point the velocity of the flow is reduced due to the fact that more flow area becomes available to the flow. Thus some of the pressure lost up to the vena contracta point is recovered.

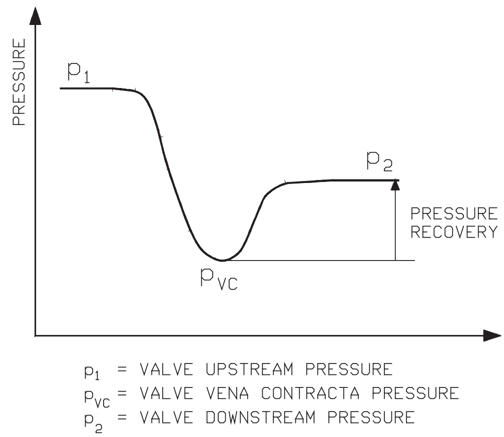


Figure 35. *Pressure profile inside the valve in the valve trim.*

The pressure recovery after the vena contracta point depends on the valve style, and is represented by valve pressure recovery factor (F_L), as given in equation (22). The closer the valve pressure recovery factor (F_L) is to 1.0, the less the pressure recovery is.

$$F_L = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}} \tag{22}$$

When upstream pressure (p_1) is constant, the interrelation between the pressure differential (Δp) across the control valve at a constant opening and the resulting flow rate (q) can be expressed by figure 36.

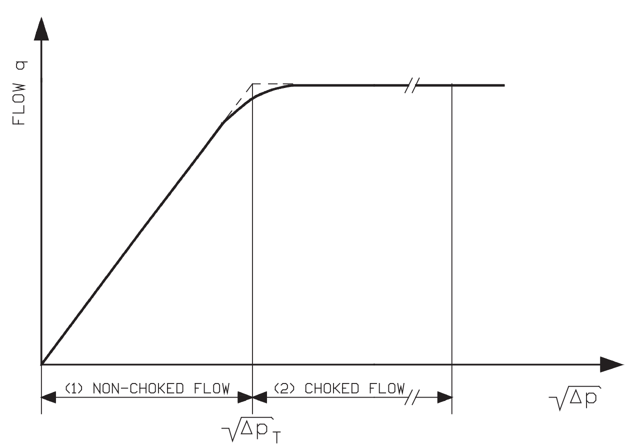


Figure 36. *Interrelation between flow rate and square root of pressure drop across a control valve.*

Up to a certain pressure drop value the flow through a control valve follows the square root of the pressure drop applied across the valve in a direct line equation, where the slope of the line is expressed as the valve flow coefficient (C_v).

With larger relative pressure drops across the valve, the gain in flow decreases, and ultimately becomes zero. The cavitation begins where the flow starts to deviate from the straight line, as illustrated in figure 36. In other words, when upstream pressure (p_1) is constant, an increase in pressure drop does not produce any increase in the flow rate through the valve. The cause of this phenomenon is choking. When the valve vena contracta pressure drops below the vapour pressure of the flowing liquid (figure 37), it causes formation of vapour bubbles in the control valve orifice, thereby restricting the liquid flow through the orifice.

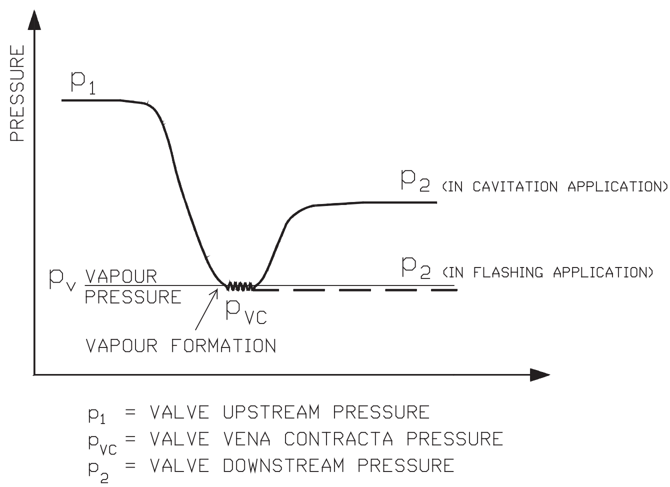


Figure 37. *Pressure behaviour in a control valve in cavitating and flashing application.*

The flow is considered to be choked when the valve actual pressure drop is above the terminal pressure drop (Δp_T), sometimes called the allowed pressure drop, derived as the corner point of the lines expressingnon-choked and choked flows in figure 36.

In choked flow cases where valve downstream pressure recovers above the liquid vapour pressure is produced a phenomenon called cavitation. If the valve downstream pressure remains below the liquid vapour pressure, the phenomenon is called flashing.

3.2 Sizing equations for liquid flow

Liquid flow in non-cavitating or non-flashing conditions is given by equation (23).

$$q = N_1 \cdot F_P \cdot F_R \cdot C_v \cdot \sqrt{\frac{p_1 - p_2}{G_f}} \qquad \textbf{(23)}$$

where
- q = liquid volumetric flow rate (SI units: [m^3/h]; US units: [gpm])
- N_1 = 0.865 (SI units: [m^3/h],[barA])
  = 1.0 (US units: [gpm],[psiA])
- F_P = piping geometry factor
- F_R = Reynold's number factor
- C_v = valve flow coefficient
- p_1 = upstream pressure (SI units: [barA]; US units: [psiA])
- p_2 = downstream pressure (SI units: [barA]; US units: [psiA])
- G_f = specific gravity ($G_f = \rho/\rho_0 = 1$ for water at 15.6°C or at 60°F)
- ρ = fluid density (SI units: [kg/m^3]; US units: [lb/ft^3]
- p_0 = density of water at 15.6°C or at 60°F
  = 999 (SI units: [kg/m^3])
  = 62.36 (US units: [lb/ft^3]

With an increasing pressure drop (p_1-p_2), the vena contracta pressure in the valve drops below the vapour pressure of the liquid, and the vapour cavities begin to form in the liquid. At a certain point, when sufficient vapour has formed, the flow becomes completely choked, and the flow rate does not increase with increasing pressure drop, if upstream pressure is constant. The corresponding pressure drop is called the terminal pressure drop (Δp_T).

$$\Delta p_T = \left(\frac{F_{LP}}{F_P}\right)^2 \cdot (p_1 - F_F \cdot p_v) \qquad \textbf{(24)}$$

where
- Δp_T = terminal pressure drop (SI units: [bar]; US units [psi])
- F_{LP} = product of the liquid pressure recovery factor of a valve with attached fittings and the piping geometry factor
- F_F = liquid critical pressure ratio factor
- p_v = liquid vapour pressure at inlet temperature (SI units: [barA]; US units [psiA])
- F_P = piping geometry factor

Consequently, the maximum flow rate (q_{max}) under choked conditions is reached when (p_1-p_2) ≥ Δp_T

$$q_{max} = N_1 \cdot F_{LP} \cdot F_R \cdot C_v \cdot \sqrt{\frac{p_1 - F_F \cdot p_v}{G_f}} \tag{25}$$

When a valve is operated in such a way that the pressure drop is close to the terminal pressure drop (Δp_T), the rate of flow may be less than that predicted by equations (23) or (25) because the real behaviour of the flow rate is approximated using straight lines, as shown in figure 36. However, the error is insignificant.

Factors F_F, F_R, F_P, and F_{LP}

Factor F_F, which is the critical pressure ratio factor for liquids, has an approximate value as given in equation (26).

$$F_F = 0{,}96 - 0{,}28 \cdot \sqrt{\frac{p_v}{p_c}} \tag{26}$$

where p_v = liquid vapour pressure at inlet temperature (SI units: [barA]; US units: [psiA])
p_c = thermodynamical critical pressure (SI units: [barA]; US units: [psiA])

Factor F_R, which is the Reynolds number factor, usually has a value of 1.0. The Reynolds number factor has an effect in non-turbulent flow, which usually occurs with high viscosity fluids, such as oil, and with low flow velocities and small valve sizes. Reynolds number factor (F_R) can be defined using the valve Reynolds number and figure 38. The valve Reynolds number is derived from the following equation (27).

$$Re_v = \frac{N_4 \cdot F_d \cdot q}{\nu \cdot \sqrt{F_p \cdot F_L \cdot C_v}} \cdot \left[\frac{(F_p \cdot F_L \cdot C_v)^2}{N_2 \cdot d^4} + 1\right]^{\frac{1}{4}} \tag{27}$$

where F_d = valve style modifier
= 1 for ball, segment, eccentric rotary plug, and single seated globe valves
= 0.7 for butterfly and double seated globe valves
= 0.5 for Q-trims and other noise attenuating trims
F_L = valve pressure recovery factor
ν = kinematic viscosity (SI and US unit: [centistoke])

d	=	inlet diameter of a valve (SI units: [mm]; US units: [in])
N_2	=	0.00214 (SI units: [mm])
	=	890 (US units: [in])
N4	=	76 000 (SI units: [m^3/h], [mm], [centistoke])
	=	17 300 (US units: [gpm], [in], [centistoke])

After calculating the Reynolds number for the valve (Re_v), the Reynold's number factor (F_R) can be determined from the diagram in figure 38.

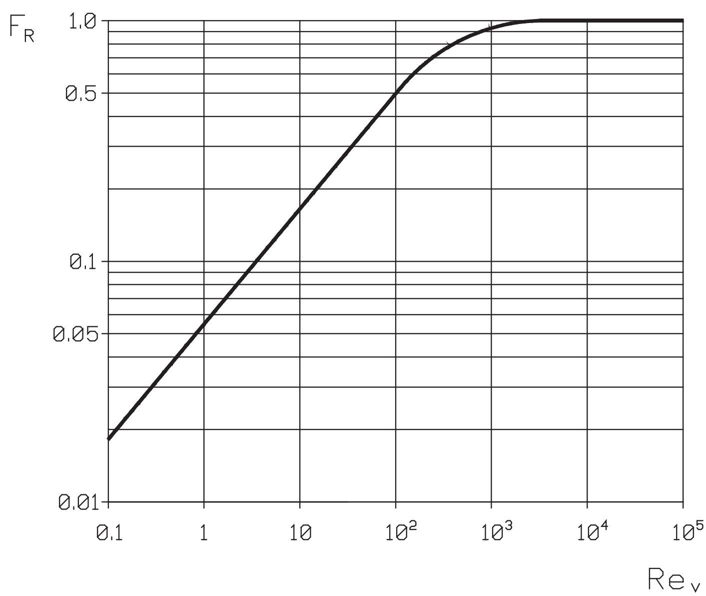


Figure 38. *Reynolds number factor (F_R) for liquid flow.*

The piping geometry factor (F_p) can be calculated as shown in equation (28).

$$F_p = \frac{1}{\sqrt{1 + \frac{\Sigma K}{N_2} \cdot \left(\frac{C_v}{d^2}\right)^2}} \qquad \textbf{(28)}$$

where	ΣK	=	$K_1+K_2+K_{B1}-K_{B2}$
	K_1	=	upstream resistance coefficient
	K_2	=	downstream resistance coefficient
	K_{B1}	=	inlet Bernoulli coefficient
	K_{B2}	=	outlet B ernoulli coefficient
	N_2	=	0.00214 (SI units: [mm])
		=	890 (US units: [in])

When the inlet and outlet fittings are identical, $K_{B1} = K_{B2}$, and they cancel out at the equation. If the pipe diameters approaching and leaving the control valve are different, the K_B coefficients are calculated as in equation (29).

$$K_B = 1 - \left(\frac{d}{D}\right)^4 \qquad \textbf{(29)}$$

where	d	=	valve inlet diameter for K_{B1} and outlet diameter for K_{B2} (SI units: [mm]; US units: [in])
	D	=	pipe inlet diameter for K_{B1} and outlet diameter for K_{B2} (SI units: [mm]; US units: [in])

For standard concentric reducers, the upstream and downstream resistance coefficients (K_1) and (K_2) can be approximated as shown in equations (30) and (31).

K_1 for inlet reducers:

$$K_1 = 0{,}5 \cdot \left[1 - \left(\frac{d}{D}\right)^2\right]^2 \qquad \textbf{(30)}$$

K_2 for outlet reducers:

$$K_2 = 1 \cdot \left[1 - \left(\frac{d}{D}\right)^2\right]^2 \qquad \textbf{(31)}$$

Factor F_{LP} is the product of the liquid pressure recovery factor of a valve with attached fittings and the piping geometry factor (F_P). It is calculated using the given valve pressure recovery factor (F_L) and piping geometry factor (F_P) as shown in equation (32).

$$F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2}{N_2} \cdot \left(\frac{C_v}{d^2}\right)^2 \cdot \Sigma K_i}} \qquad \textbf{(32)}$$

where $\sum K_i$ = head loss coefficient for an inlet fitting = $K_1 + K_{B1}$
N_2 = 0.00214 (SI units: [mm])
= 890 (US units: [in])

3.3 Cavitation and flashing

3.3.1 Cavitation phenomenon

Cavitation is a two-phase phenomenon appearing in liquids under certain flow conditions. The cavitation within a control valve can most easily be described using figure 39, where changes in pressure and velocity conditions across a control valve are displayed using a simplified model of an orifice plate in the pipe.

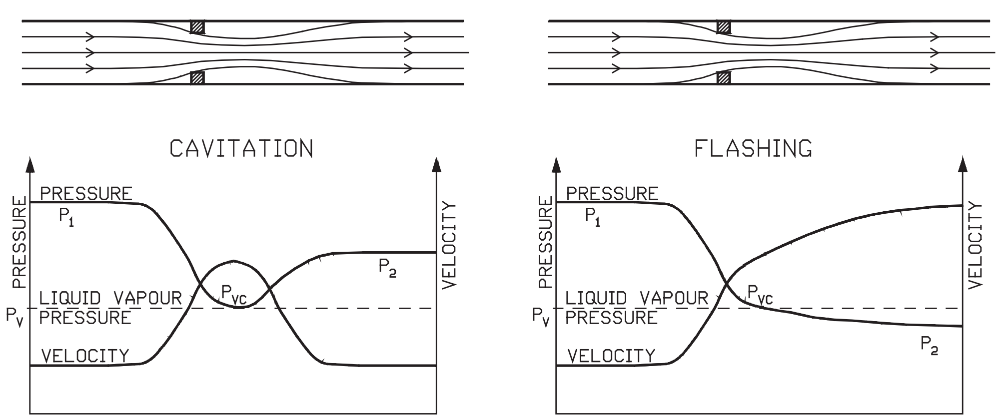


Figure 39. *A simplified model of pressure and velocity changes in control valves in cavitating and flashing conditions.*

A reduction in the area available for the flow leads to a large increase in fluid velocity, which in turn causes a drop in the fluid pressure due to the conservation of energy. The pressure is at its lowest level at the vena contracta point, which is located a little downstream of the physical valve orifice. When the pressure at the valve vena contracta point is lower than the liquid vapour pressure (p_v) and the downstream pressure of the valve (p_2) is higher than the liquid vapour pressure (p_v), cavitation takes place.

Cavitation occurs in two stages. Firstly, as the pressure of the liquid is reduced below the liquid vapour pressure, vapour bubbles form in the liquid. The formation of vapour bubbles starts from boundary layers of the nuclei, minuscule solid particles or gas bubbles that are carried by the fluid. Secondly, recovery of pressure after the vena contracta raises the pressure above the liquid vapour pressure, causing collapse of the vapour bubbles. The collapsing vapour bubbles cause large pressure shocks, due to either a liquid microjet or a spherical shock wave as shown in figure 40.

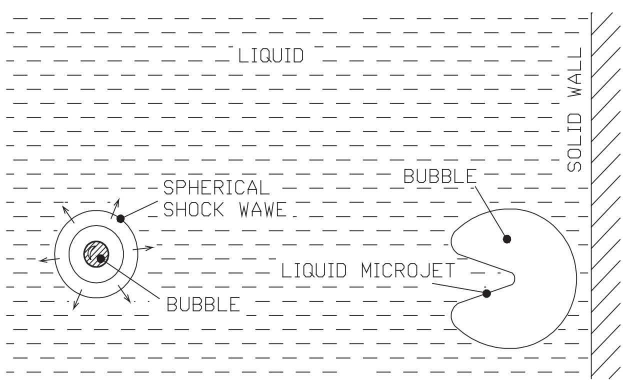


Figure 40. *Pressure shock generation mechanisms.*

If the vapour bubble is very close to or in contact with a solid wall, the microjet generated can hit the wall with a pressure shock of the order of 10^4 MPa ($1.5 * 10^6$ psi). Spherical pressure shock waves are generated when the bubbles collapse further away from solid walls, in the body of the liquid. The pressure shock in this case is of the order 10^3 MPa ($1.5 * 10^5$ psi) and the effect of the microjet does not reach a solid wall. If bubbles collapse close to a solid wall, the microjet or shock wave can cause yield or surface cracks on the material. Cavitation-damaged surfaces are rough and spongy.

Figure 39 right side shows a situation in which the valve outlet pressure is lower than the liquid vapour pressure. In this case vapour bubbles cannot collapse and the fluid exits the valve as a two-phase mixture. The fluid is not cavitating, as the second phase (collapse of the bubbles) does not occur. This phenomenon is called flashing, and it is not nearly as damaging as cavitation. Selecting all stainless steel valve body and trim materials and limiting the flow velocity at the valve inlet and outlet typically gives satisfactory control valve life.

Mechanical damages to cavitating valves has been shown to correlate very closely with the noise level produced. The maximum noise level is usually reached before the valve chokes. When choking and any flashing takes place, the bubble collapse location and, thus, the source of noise, moves further downstream. Cavitation damage also correlates with the materials and coatings used in the valve and downstream piping.

Although cavitation damage can be a purely mechanical phenomenon, it is often related to corrosion or erosion. In cavitation corrosion the microjets or shock waves break the passivating film covering a metal surface. The existence of this passivating film is essential to the corrosion resistance of the material. The base metal is consumed in renewing this passivating film, and, as an ongoing process, this causes considerable loss of material from the cavitating location, as illustrated in figure 41.

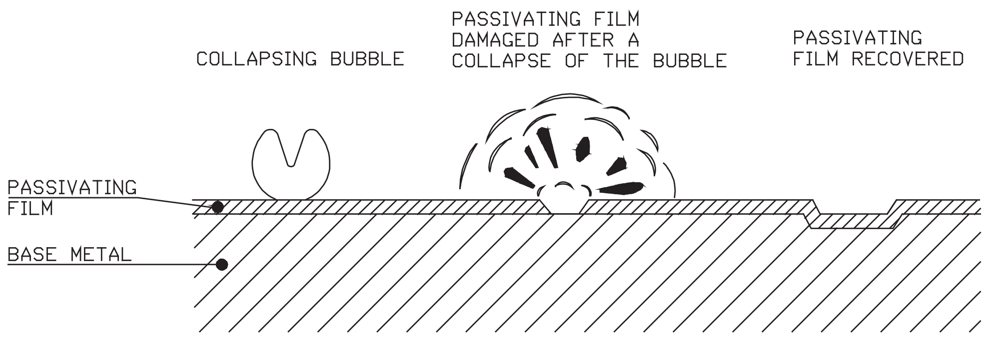


Figure 41. *Wear on passivating film in cavitation corrosion.*

Erosion related to cavitation or cavitation corrosion causes a substantial increase in material wear and damage. The metal material is soften by the joint effect of cavitation and possible corrosion, and thus is subject to wear by erosion. The erosion is strongly dependent on the flow velocity. Equation (33) expresses the relation between wear and flow velocity.

$$\in = k \cdot V^n \qquad \textbf{(33)}$$

where
$\in$ = loss of material
k = constant
v = flow velocity
n = wear exponent

The value of exponent n in equation (33) can be as high as 7 in cases of erosion related to cavitation corrosion. In normal erosion the value of exponent (n) is roughly 2.5. The wear is also strongly correlated with the materials used, with stainless steels suffering much lower wear than carbon steels.

In applications where cavitation exists, close attention should thus be paid to selecting the correct trim style and materials to avoid erosion related to cavitation corrosion. Equation (33) suggests the following:

- Reduce the flow velocity inside the valve by using multistage pressure drop trim to minimize valve trim velocities.
- Reduce the wear by using stainless and hardened materials.

3.3.2 Investigating the existence of cavitation

One way to estimate the level of cavitation is to use the pressure recovery factor (F_L). The F_L factor is determined according to the C_v measurement. The definition for the F_L factor is shown in equation (22). On the basis of that equation, the terminal pressure drop for the valve can be calculated. When the terminal pressure drop is reached, the flow is fully choked.

Some manufacturers publish a “cavitation index” (K_C) for their valves. This number is the ratio (p_1-p_2) / (p_1-p_V). The value for the cavitation index is based on C_v measurement with water. The point where the valve is considered to cavitate is the point where the curve in Figure 36 diverges 2 % from a straight line. The value of the cavitation index varies with valve opening. The deviation is caused by cavitation, i.e. the valve is cavitating to some extent even before the limit set by the cavitation index is reached.

The intensity of the cavitation depends on the valve type and pressure level, so it is not easy to give an absolute limit for non-cavitating flow. For this reason, Valmet has decided to use both noise level and terminal pressure drop to predict cavitation level.

Cavitation and noise

Determining whether cavitation causes mechanical damage for a given set of flow conditions is complicated. Several different factors affect the degree of the cavitation damage: the valve material, the nature of the medium, pipeline layout and possible solids in the flow. When the actual valve pressure drop is at or above the calculated terminal pressure drop and the downstream pressure is higher than the vapour pressure, it is likely that significant cavitation damage will take place and any available means of preventing this should be used.

On the other hand, Figure 42 shows that cavitation starts before the terminal pressure drop is reached. The curve shows how the sound pressure level (SPL) of a valve develops as a function of the ratio $(p_1-p_2) / (p_1-p_V)$. It can be seen from next picture that the region of laminar flow is practically negligible, and noise levels are low. Following the laminar region, the curve is characterized by turbulence of the liquid flow and a moderate increase in the sound pressure level. From a certain value of the valve pressure differential, $\Delta p = z(p_1-p_V)$ onwards, the noise level curve starts to rise rapidly. The valve-specific characteristic pressure drop ratio (z), for incipient cavitation, is determined by the manufacturer's laboratory measurement of hydrodynamic noise. These values are typically shown in the valve manufacturer's sizing coefficient documents. The reason for this increase in sound pressure level is cavitation. It can be seen from laboratory tests that the sound pressure levels start to rise even before actual steam bubbles can be observed visually.

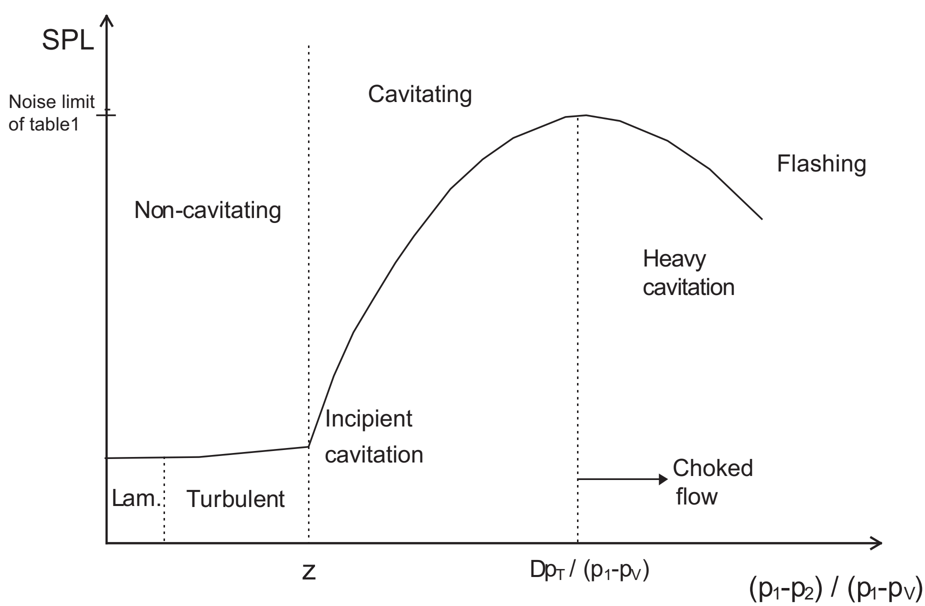


Figure 42. *Liquid flow valve noise emission.*

When the Δp across the valve is close to the terminal pressure drop (Δp_T), the sound pressure level reaches its maximum. Further increase in pressure ratio $(p_1-p_2)/(p_1-p_V)$ will, as a rule, cause the noise pressure level curve to level off again, as can seen from figure. This is predicted by the standard VDMA 24422 (1979) which, like most other standards, predicts the noise level one metre downstream from the valve. When the valve is cavitating heavily the implosion of the cavitation bubbles occurs further away from the valve in the downstream piping. That is the reason for the reduction in sound pressure levels predicted by VDMA 24422 (1979). The noise does not disappear, but simply moves to another place (downstream piping). Also, the length of the area where the implosions occur increases, so the intensity of the noise declines, even though the number of bubbles increases. On the other hand, the nature of the flow approaches more closely that of a gas/liquid mixture when the pressure differential increases.

Typically, a valve which is noisy in liquid service is cavitating. Cavitation bubble implosions are the main source of disturbing noise, and noise level is directly related to cavitation intensity. In the early stages of cavitation (incipient cavitation) the noise can sound like sand going through the valve . When the pressure difference across the valve is increased, the intensity of the cavitation rises and severe cavitation and rough noise occurs. Cavitation noise can also be predicted accurately. Therefore, the predicted noise level is a good indicator of mechanical cavitation damage.

The pressure drop needed to mechanically damage valves or piping depends on the valve type, size and material. Various studies suggest that the lower the recovery of the valve, the closer the terminal pressure drop (Δp_T) can be approached without cavitation damage. Terminal pressure drop (Δp_T) occurs at a much higher level with a low recovery valve. It is not always true that when $\Delta p > \Delta p_T$ there will be damage. If the pressure differential is very small and the time that the valve is exposed to the cavitation is short, there may not be a problem. Corrosion is also a factor, which depends on the material and the medium used.

There are two factors to consider when cavitation intensity is estimated:

- the pressure drop across the valve may not exceed the terminal pressure drop (Δp_T) (If Δp is small, Δp_T may be exceeded, but then the material selection is critical)
- the noise level of the valve may not exceed the values given in Table 1

Noise limits for preventing cavitation damage in valves for non-insulated schedule 40 piping are given in Table 1. It is important to note that even if the sound pressure level is below the limits given in Table 1, the valve might be cavitating heavily if the pressure difference over the valve is greater than the terminal pressure drop.

Table 1. Noise limits for liquid flow.

Valve size		Noise limit
DN	Inch	
up to 80	up to 3	80 dB(A)
100 - 150	4 - 6	85 dB(A)
200 - 350	8 - 14	90 dB(A)
400 and larger	16 and larger	95 dB(A)

3.3.3 Flashing of liquids

When the downstream pressure of a control valve is less than the liquid vapour pressure, part of the liquid is vapourized and remains as vapour downstream of the valve. Flow downstream of valve is part liquid and part vapour. The flow is choked and therefore a decrease in downstream pressure does not increase the flow rate. Flashing flow sizing is done according to liquid choked flow equations. In some cases where upstream pressure is very close to vapour pressure, flow may occur in two phases upstream of the valve, and sizing is not as accurate. Two-phase flow sizing can be used in such cases.

Flashing flow may cause mechanical difficulties, like erosion and vibration, but unlike cavitation, the reason is the high velocity of the two-phase flow stream. High velocity is due to the larger vapour volume compared with liquid. High velocity can be very erosive and it is recommended that more resistant materials be used in such cases. Valve downstream velocity can be approximated according to the following calculations.

The amount of vapour in the flow stream is expressed as a fraction of flash, so that one hundred percent of flash is totally vapourized. The fraction of flash (x_v) can be calculated from the enthalpy changes in the vapourizing process, as given in equation (34).

$$x_v = \frac{h_{f1} - h_{f2}}{h_{fg2}} \quad \textbf{(34)}$$

where x_v = fraction of liquid flashed to vapour
h_{f1} = enthalpy of saturated liquid at inlet temperature conditions
h_{f2} = enthalpy of saturated liquid at outlet pressure conditions
h_{fg2} = evapouration enthalpy at outlet pressure conditions

The density of the vapour in downstream conditions (ρ_{g2}), equation (35), is needed to calculate downstream velocity. Vapour density given for upstream conditions may be significantly different from the density in downstream conditions because of vapour compressibility.

$$\rho_{g2} = N_{12} \cdot \frac{p_2 \cdot M}{Z \cdot T_2} \tag{35}$$

where
- ρ_{g2} = vapour downstream density (SI units: [kg/m^3]; US units: [lb/ft^3])
- N_{12} = 12.03 for SI units
  = 0.09325 for US units
- p_2 = valve downstream pressure (SI units: [barA]; US units: [psiA])
- M = molecular weight of vapour
- Z = vapour compressibility; typically Z = 1.0
- T_2 = downstream temperature; usually assumed to be same as upstream temperature (SI units: [K]; US units: [R])

The outlet velocity of the liquid and vapour mixture (v_{fg2}) can be calculated from equation (36).

$$v_{fg2} = \frac{N_{13}}{d^2} \cdot w \cdot \left[\frac{1 - x_v}{\rho_{f2}} + \frac{x_v}{\rho_{g2}}\right] \tag{36}$$

where
- v_{fg2} = outlet velocity of the liquid and vapour mixture (SI units [m/s]; US units [ft/s])
- N_{13} = 353.9 for SI units
  = 0.05103 for US units
- w = mass flow (SI units: [kg/h]; US units: [lb/h])
- d = valve outlet diameter (SI units: [mm]; US units: [in])
- x_v = fraction of liquid flashed to vapour
- ρ_{g2} = gas downstream density (SI units: [kg/m^3]; US units: [lb/ft^3])
- ρ_{f2} = liquid downstream density (SI units: [kg/m^3]; US units: [lb/ft^3])

The recommended maximum flow velocity (v_{max}) at the valve exit port is given in terms of sonic velocity (v_s) as follows.

$v_{max} \leq 0.2 * v_s$ for continuous throttling duty

$v_{max} \leq 0.3 * v_s$ for infrequent use

$$v_s = N_{14} \cdot \sqrt{\frac{k \cdot T}{M}} \tag{37}$$

where vs = sonic velocity (SI units [m/s]; US units [ft/s])
k = ratio of specific heats
M = molecular weight
N14 = 91 for SI units
= 223 for US units

In practice, it is recommended that the valve downstream piping length is minimized in flashing flow applications by locating the valve as close to the receiving vessel as possible. Vertical upward flow should be avoided to prevent any slug flow, which may cause strong vibrations.

There is currently no noise prediction method for flashing control valves. However, the noise level of control valves in flashing conditions generally stays below 85 db(A).

3.4 Cavitation and hydrodynamic noise abatement

The most significant source of control valve noise in liquid applications is cavitation. The abatement of noise and cavitation are therefore dealt together here. The basic division between methods of noise and cavitation abatement in liquid valves is between 'source' treatment, i.e. modification of the valve and its trim so that the cavitation intensity is reduced or the existence of cavitation is prevented, and 'path' treatment, i.e. dampening of the noise generated. The source treatment of hydrodynamic noise and cavitation abatement can be performed using at least four different methods, as indicated in figure 43.

Source treatment, whenever possible and feasible, is the preferred way of dampening liquid cavitation and the associated hydrodynamic noise, because only then can excessive generation of cavitation bubbles be avoided across a wide flow range. Path treatment reduces the noise radiated by the piping system, but it does not eliminate cavitation inside the valve and adjacent piping.

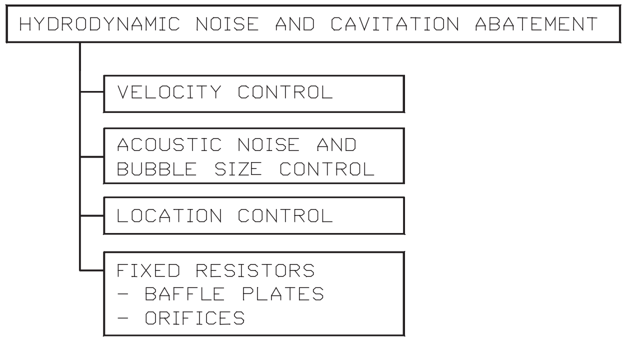


Figure 43. *Hydrodynamic noise and cavitation abatement source treatment.*

Velocity control

Controlling the fluid velocity, which is equivalent to controlling the pressure in the valve trim, is an effective mean avoiding cavitation. The aim of this is to get the lowest pressure in the valve trim above the vapour pressure for the liquid in question. This is done by dividing the valve pressure drop into several stages. There are two basic methods for pressure drop staging:

- Identical pressure drop across each stage
- Identical minimum pressure in each stage

The first principle requires a constant flow area trim. Due to the equal area, the flow velocity in each stage is constant and the pressure drop is equal in each step. The second principle involves an expanding area trim, in which the pressure drop in the first stages is very high, but a higher minimum pressure is maintained in the last stages, thereby better avoiding cavitation within the valve trim. Figure 44 shows the two principles.

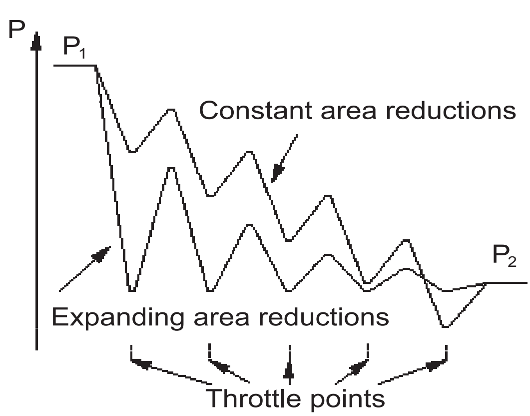


Figure 44. *Alternative pressure staging principles.*

Expanding area trim is recommended when cavitation is to be eliminated to a very great extend. However, the velocity during the first stages of this trim will be quite high and may create erosion problems. In practice, valve designers usually combine the two methods to arrive at an optimum solution.

Acoustic noise and bubble size control

The division of flow into multiple small streams actually has two effects; first it acts to create a pressure stage and secondly it reduces the size of vapour bubbles in the fluid. Smaller bubbles cause higher frequency noise fields in the fluid and reduce the external vibration intensity of the pipe, as well as reducing the mechanical and erosion effects of cavitation. The concept of using small jets to reduce control valve noise is based on the principle that the jet produces a characteristic frequency above the pipe ring frequency, thereby reducing the sound reradiated by the pipe.

Location control

Location control is a principle of flow path design to over come that bubble implosions can be directed far away from solid boundaries.

Flat baffle plates

When the pressure drop ratio $x_F = \Delta p/(p_1\text{-}p_v)$ becomes very high it is possible that even a valve with an anti-cavitation trim cannot alone do a good job of reducing liquid cavitation and noise.

If the valve outlet pressure is increased with a baffle plate, the pressure drop ratio across the control valve can be brought back to the most effective valve operating range. The baffle plate is thereby used to create the required back-pressure for the valve.

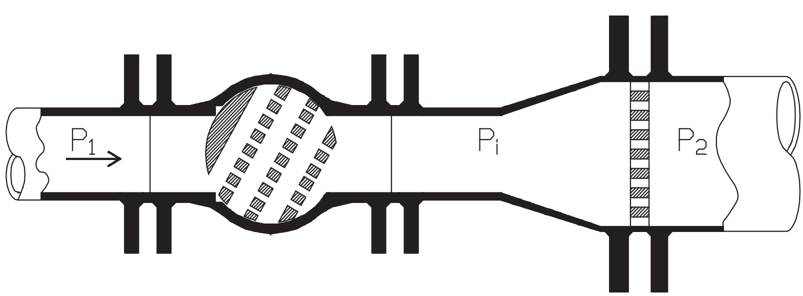


Figure 45. *Typical installation of a noise attenuating valve with a flat baffle plate.*

A flat baffle is a flow restrictor with a custom-made constant flow area and multiple holes for a particular flow condition. The pressure drop across a baffle plate is proportional to the flow squared. If it is assumed that the inlet pressure (p_1) falls and the outlet pressure (p_2) increases in proportion to the flow squared (cf. installed flow characteristics), the division of the total pressure drop (p_1 - p_2) between the valve and baffle plate described is in figure 46.

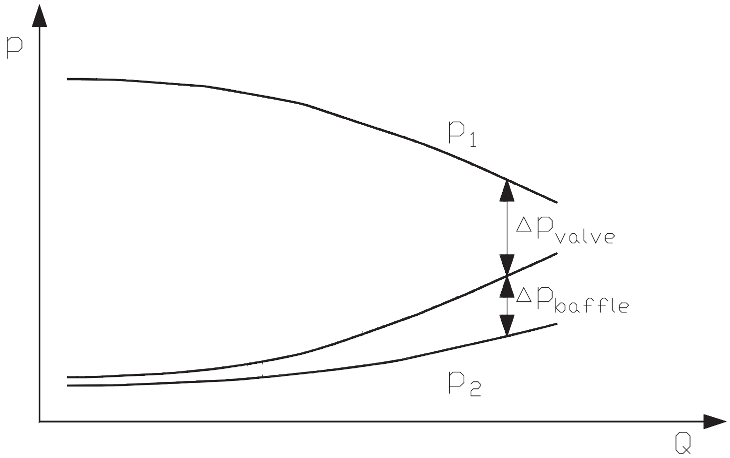


Figure 46. *Typical pressure drop division between valve and baffle plate.*

It has to be remembered that a baffle plate is a fixed restrictor, which produces the desired pressure drop only at one given flow, that used for sizing the baffle. The baffle plates are typically sized for the valve maximum flow. As the flow is reduced the pressure drop across the baffle decreases and the pressure drop across the valve increases. Thus, the baffle plate loses its effect at low flow rates.

Orifice plates

An orifice plate is a constant flow resistance. Unlike a baffle plate, there is only one large hole in an orifice plate, rather than multiple small holes.

An orifice plate can be used for producing linear valve installed flow characteristic, and in limited cases, also for cavitation abatement. An orifice plate can be installed both upstream and downstream of a control valve. Sizing an orifice plate is basically similar to sizing a baffle plate.

Sizing baffle and orifice plates

The following instructions should be followed when sizing a baffle or an orifice plate downstream of the valve. Most easily the sizing is done with a sizing and selection software.

Choose a value for valve back pressure (p_i) under maximum flow conditions. The capacity of the plate (C_v) is then calculated using a pressure drop (p_i - p_2) as in equation (38).

$$q = N_1 \cdot C_v \cdot \sqrt{\frac{p_i - p_2}{G_f}} \tag{38}$$

where
- q = liquid volumetric flow rate (SI units: [m^3/h]; US units: [gpm])
- C_v = flow coefficient of the baffle plate
- p_i - p_2 = pressure drop across the baffle plate (SI units: [bar]; US units: [psi])
- G_f = specific gravity (G_f=ρ/ρ_0 = 1 for water at 15.6°C or at 60°F)
- ρ = fluid density (SI units: [kg/m^3]; US units: [lb/ft^3]
- ρ_0 = density of water at 15.6°C or at 60°F
  = 999 (SI units: [kg/m^3])
  = 62.36 (US units: [lb/ft^3]
- N_1 = 0.865 (SI units: [m^3/h], [barA])
  = 1.0 (US units: [gpm],[psiA])

The back pressure chosen must ensure that the actual pressure drop across the plate does not exceed the terminal pressure drop of the plate, given by the equation (39). The calculation procedure may require iteration.

$$p_i - p_2 \le F_L^2 \cdot (p_i - F_F \cdot p_v) \tag{39}$$

where
- L = liquid pressure recovery factor of a baffle plate (see figure 47)
- F_F = critical pressure drop ratio factor for liquids (see equation (26))
- p_v = liquid vapour pressure (SI units: [barA]; US units: [psiA])

The pressure recovery factor F_L of plates is shown in figure 47.

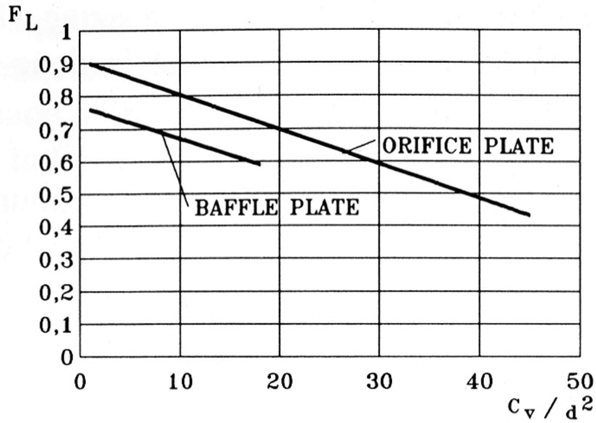


Figure 47. *The pressure recovery factor F_L for orifice and for baffle plates as a function of C_v/d^2 (note that the nominal size (d) of the plate is in inches).*

When the capacity of the plate is determined, the pressure drop across it at flows other than maximum flow can be calculated using liquid flow sizing equations. In non-choked flow conditions liquid flow can be simplified as in equation (40).

$$p_1 - p_2 = (p_1 - p_2)_{max} \cdot \left(\frac{q}{q_{max}}\right)^2 \tag{40}$$

For nonchoked flow the required flow cross-section area for the plate can be calculated using equation (41).

$$A_B = \frac{q}{\alpha \cdot \sqrt{\dfrac{N_{11}^2 \cdot \Delta p}{F_L^2 \cdot G_f} + \dfrac{q^2}{A_p^2}}} \tag{41}$$

where N_{11} = 0.0509 (SI units: [m^3/h], [bar], [mm^2])
= 38 (US units: [gpm], [psi], [inch2])
Δp = pressure drop p_i-p_2 (bar or psi)
A_B = plate cross-section flow area (mm^2 or in^2)
A_P = pipe area at the plate (mm^2 or in^2)
α = 0.70 for baffle plates
= 0.00762 *(C_v/d^2) +0.55 for orifice plates, when (C_v/d^2) < 21
= 0.71 for orifice plates, when (C_v/d^2) Š 21

The maximum ratio of A_B/A_P for mechanical integrity of the baffle is ap proximately 0.4. Single orifices can have a larger area ratio.

In equation (42), it is shown how a single hole orifice plate, when used upstream of the valve, is to be sized to ensure that there is no cavitation.

$$p_1 - p_i \leq z \cdot (p_i - p_v) \qquad \textbf{(42)}$$

Incipient cavitation coefficient (z) can be determined from figure 48. Higher cavitation coefficient (z) value for orifice plates compared to value for baffle plate is due to greater hole diameter ratio to hole length.

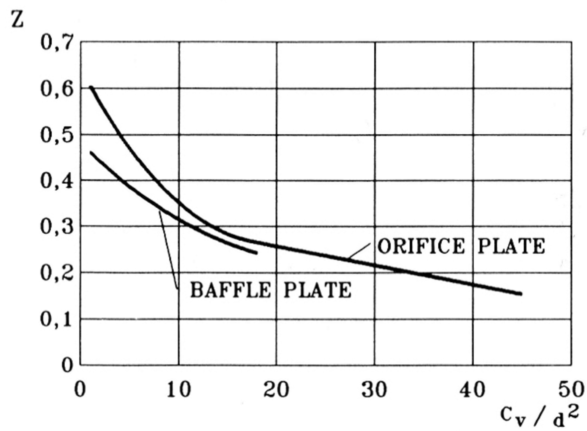


Figure 48. *Incipient cavitation coefficient z for orifice and for baffle plates as a function of C_v/d^2 (note that the nominal size (d) of the plate is in inches).*

Example of noise and cavitation abatement

A Q-trim™ valve (figure 44) utilizes a combination of the effects described in the above paragraphs:

- The pressure drop is taken in multiple stages, leading to lower valve trim velocity and higher trim minimum pressure.
- Division of the flow into multiple streams diffuses the flow and provides acoustic control.
- The trim outlet is designed in such way that the flow is spread out evenly across the downstream flow port.

The rotary motion in rotary type of valves, such as the valve in figure 44, gives additional benefits of being able to handle a very large flow range and pass impurities through the valve.

3.5 Hydrodynamic noise prediction for valves

Noise generated by liquid flow is called hydrodynamic noise. At low pressure ratios, $x_F = \Delta p/(p_1 - p_v)$, with laminar or turbulent flow, the noise level is normally low. At a certain pressure ratio (x_F=z) the noise level increases sharply. The reason for this sharp increase is cavitation. The value of cavitation coefficient (z) depends on the valve type and opening angle.

There are two well known noise prediction methods used in the valve market. One is based on the German VDMA 24422 recommended practise which origins to 1979 (see appendix K). The second method is an international standard for hydrodynamic noise prediction generated by a control valve, IEC 60534-8-4.

According to the VDMA 24422 (1979) the noise level in dB(A) can be calculated using the following equations (equation (43) for SI units and equation (44) for US units) when the flow is turbulent but cavitation has no effect on the noise level; (x_T < z). In equations (43) and (44) $\Delta L_F = 0$

$$L_A = 10 \cdot \log(C_v) + 18 \cdot \log(p_1 - p_v) - 5 \cdot \log(\rho) + 18 \cdot \log\left(\frac{x_F}{z}\right) + 39 + \Delta L_F + \Delta L_P \quad \textbf{(43)}$$

$$L_A = 10 \cdot \log(C_v) + 18 \cdot \log(p_1 - p_v) - 5 \cdot \log(\rho) + 18 \cdot \log\left(\frac{x_F}{z}\right) + 12{,}1 + \Delta L_F + \Delta L_P \quad \textbf{(44)}$$

When cavitation (x_F>z) has an effect, the noise level LA in dB(A) can be calculated using the following equations (equation (45) for SI units and equation (46) for US units)

$$\begin{aligned} L_A = {} & 10 \cdot \log(C_v) + 18 \cdot \log(p_1 - p_v) - 5 \cdot \log(\rho) \\ & + 292 \cdot (x_F - z)^{0,75} \\ & -(268 + 38 \cdot z) \cdot (x_F - z)^{0,935} \\ & + 39 + \Delta L_F + \Delta L_P \end{aligned} \quad \textbf{(45)}$$

$$\begin{aligned} L_A = {} & 10 \cdot \log(C_v) + 18 \cdot \log(p_1 - p_v) - 5 \cdot \log(\rho) \\ & + 292 \cdot (x_F - z)^{0,75} \\ & -(268 + 38 \cdot z) \cdot (x_F - z)^{0,935} \\ & + 12{,}1 + \Delta L_F + \Delta L_P \end{aligned} \quad \textbf{(46)}$$

For each valve, the z and ΔL_F values for equations (45) and (46) are usually presented in the manufacturers C_v-tables and noise prediction graphs.

Correction coefficient ΔLp is the correction for pipe attenuation, and is given by the following graph (figure 49) and table 2.

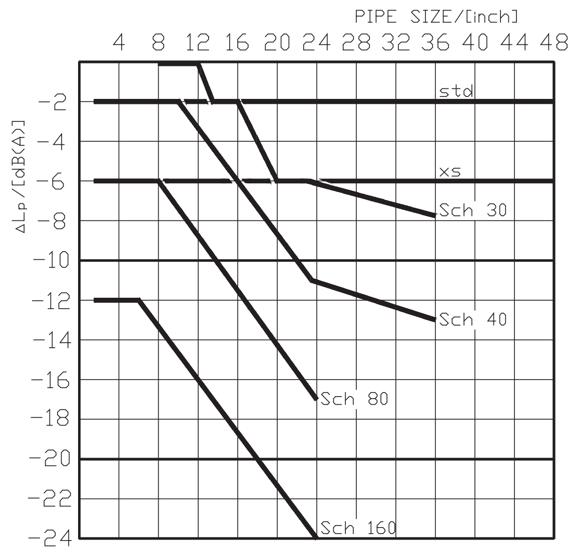


Figure 49. *Correction coefficient ΔLp for pipe schedule number.*

Table 2. Correction coefficient ΔLp for pipe wall thickness

SIZE	Correction coefficient ΔLp for pipe wall thickness (mm)																
DN	Inch	2.9	3.2	3.6	4	4.5	5.6	6.3	7.1	8	10	11	12.5	14.2	16	20	25	30
50	2	0	-0.5	-1	-2	-3.5	-6	-7.5	-9	-11	-13							
80	3		0.5	0	-0.5	-1	-2	-3.5	-5	-6.5	-10	-12	-14					
100	4			0.5	0	-0.5	-1.5	-2.5	-3.5	-5	-8	-9	-11.5	-13	-15			
150	6					0.9	0	-1	-2	-3	-5	-6	-7	-9	-10	-11.5	-17	
200	8						0.5	0	-0.5	-1.5	-3.5	-4.5	-6	-7	-8.5	-11	-15	-17
250	10						1	0.5	0	-1	-3	-4	-5.5	-6.5	-8	-10	-13	-15.5
300	12							1	0.5	0	-2.5	-4	-5.5	-6.5	-8	-10	-13	-15.5
400	16								0.5	0	-2.5	-4	-6	-7	-8	-10.5	-13.5	-16
500	20									0	-2.5	-4	-6	-7	-8.5	-11	-13.5	-16.5
600	24									0	-2.5	-4	-6	-7	-8.5	-11.5	-14	-17
800	32									0	-2.5	-4	-6	-7	-9	-12	-15	-17

The hydrodynamic noise prediction according to the IEC 60534-8-4 is more laborious than noise prediction according to VDMA 24422 (1979). The noise prediction is based on a method where mechanical stream power as well as acoustical efficiency factors are calculated in different flow regimes i.e. turbulent and cavitation conditions. The sound pressure level inside the pipe is estimated and the pipe external noise is attained from inner sound pressure level by reducing the transmission loss of the piping.

In short the IEC hydrodynamic noise prediction can be presented in the five following steps:

1. The noise source is the mechanical energy dissipated in the valve orifice. The mechanical stream power is calculated at vena contracta of the valve.
2. Small part of mechanical power is transferred to acoustical energy. The portion of the mechanical power converted to (acoustical) sound power is defined by using acoustical efficiency factors. Different flow regimes (turbulent and cavitation) have different acoustical efficiency factors.
3. The sound power is converted to internal sound pressure level at the downstream of the valve. The peak frequency of the generated noise is also estimated.
4. The transmission losses of the piping are defined. The transmission loss is dependent of the piping, the peak frequency and the fluid properties. Step 4 also covers the part where the pipe external sound power level is A-weighted.
5. Define the A-weighted sound pressure level at the observation point at a distance of 1 m from the pipe wall.

3.6 Hydrodynamic noise prediction for baffle and orifice plates

The valve and baffle or orifice plate noises are first calculated separately using the VDMA 24422 noise prediction standard. The valve sizing and noise prediction is done using (p_1 - p_i) as the pressure drop for the valve. Pressure pi is the intermediate pressure between the valve and the plate. The required capacity (C_v) for the baffle plate is calculated with the IEC 60534/ISA S75 sizing equations for liquids, using (p_i - p_2) as the pressure drop for the baffle. The pressure recovery coefficient (F_L) and acoustical coefficient (z) are given as a function of C_v/d_2. The hydrodynamic noise coefficients (ΔL_f) for the orifice and baffle plates are given in figures 50 and 51.

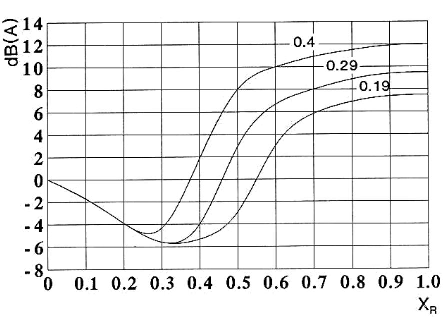


Figure 50. *Hydrodynamic noise coefficient ΔL_f for baffle plates as a function of X_R = (x-z) /(1-z), the three curves representing different ratios of baffle-free cross-section area to pipecross-section area.*

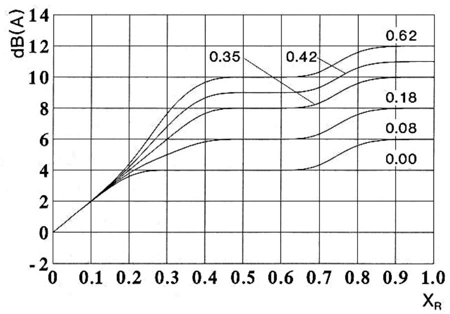


Figure 51. *Hydrodynamic noise coefficient ΔL_f for orifice plates as a function of X_R =(x-z)/(1-z), the six curves representing different ratios of orifice-free cross-section area to pipe cross-section area.*

The total noise L_{pTOT} is calculated by adding the noise levels from different sources logarithmically as in the following equation (47).

$$L_{pTOT} = 10 \cdot \log(10^{LpV/10} + 10^{LpB/10}) \tag{47}$$

where L_{pV} = valve noise level [dB(A)]
L_{pB} = noise level for baffle [dB(A)]

3.7 Recommended flow velocities for liquids

Limitation of liquid flow velocity in pipelines and control valves is mainly used to prevent excessive erosion. Other reasons include piping pressure losses (proportional to velocity squared), and corrosion (wearing away of protective film from metal surface). In the case of butterfly valves, high inlet velocities can also lead to instability of the valve trim.

The following are recommended inlet velocities for liquid service control valves:

1 Segment, ball, globe, eccentric rotary plug, and Q-trim™ valves:
- Inlet velocity ≤ 10m/s (32 ft/s) for continuous duty
- Inlet velocity ≤ 12 m/s (39 ft/s) for infrequent duty

2 Butterfly valves:
- Inlet velocity ≤ 7 m/s (23 ft/s) for continuous duty
- Inlet velocity ≤ 8.5 m/s (27 ft/s) for infrequent duty

Typical piping design velocities range from 2 to 3 m/s (6 to 10 ft/s) for liquids.

The above values are for relatively pure liquids. As well as cavitation and flashing, impurities in the flow may reduce the allowed maximum flow velocities.

4 GAS AND STEAM FLOW

The flow of gas and steam through control valves has been well defined by universal, standardized sizing equations (ISA/IEC). In the first half of this chapter, a description of these equations and the basic fluid mechanics involved in gas or steam flow through control valves is given much emphasis.

In the second half of this chapter, a great deal of attention is placed on techniques of aerodynamic noise generation, prediction and abatement. This is done for two reasons: firstly, noise is again, after some years of relatively low emphasis, becoming a major environmental issue; and secondly, there is no good, easily understandable literature on the actual generation and abatement mechanics of noise in control valves. The details of aerodynamic noise generation and abatement given here should help to explain the reasons for aerodynamic noise generation in control valves and the different means that are available for dampening aerodynamic noise.

4.1 General

The main difference of gas or steam flow compared with liquid flow, is that the interrelation of flow and pressure is typically much weaker. A typical pressure source is a manifold with a relatively constant pressure unless the particular process pipe takes a major share of the gas or steam led into it. A typical process installation model for a compressible-flow such as a gas or steam flow control valve is shown in figure 52.

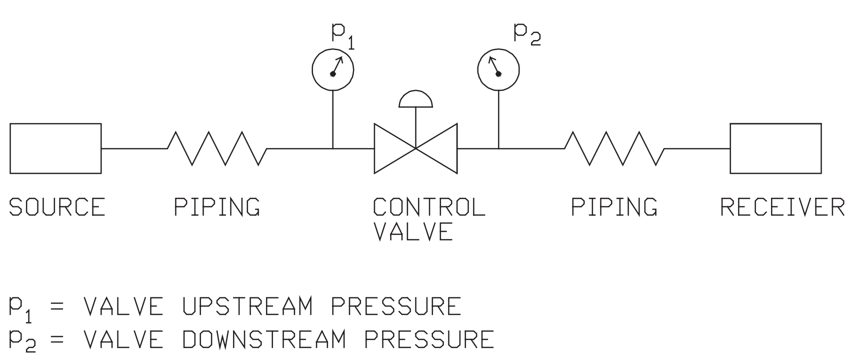


Figure 52. *Typical gas or steam control valve installation.*

In figure 52, the upstream pressure (p_1) is located two pipe diameters upstream of valve inlet flange and downstream pressure (p_2) is located six pipe diameters downstream of the valve downstream flange (per std. IEC 60534/ISA S75). This chapter concentrates on the throttling process between upstream pressure (p_1) and downstream pressure (p_2) of figure 52.

The most important difference between liquid and gas flow is compressibility: when the pressure is increased the volume of the gas decreases and vice versa. For the flow there is used constant flow units like mass flow or standard volumetric flow units. The pressure drop ratio x = Δp/p1 is used instead of pressure drop alone because gas expansion is related to this ratio. The speed of sound is a significant measure of the effects of compressibility compared with the velocity of flow. This introduces a dimensionless parameter which is called the Mach number. The calculation for the Mach number (Ma) is given in equation (48).

$$Ma = \frac{v}{v_s} \tag{48}$$

where	Ma	=	Mach number
	v	=	speed of flow (SI units: [m/s]; US units [ft/s])
	vs	=	speed of sound (SI units: [m/s]; US units [ft/s]), given by equation (49)

$$v_s = N_{14} \cdot \sqrt{\frac{k \cdot T}{M}} \tag{49}$$

where	k	=	ratio of specific heats (dimensionless)
	M	=	molecular weight (g/mol)
	T	=	temperature (SI units: [K]; US units [°R])
	N_{14}	=	91 for SI units
		=	223 for US units

Figure 53 shows the interrelation of flow through a control valve at a constant opening and constant upstream pressure (p_1). It shows the flow through a gas or steam valve starting to deviate from a straight line at fairly low pressure drop due to gas compressibility. Accordingly, the valve inlet pressure is higher and the volume smaller than at the outlet, where lower pressure causes higher volume. This expansion becomes a significant factor at higher pressure drop ratios (Δp/p1).

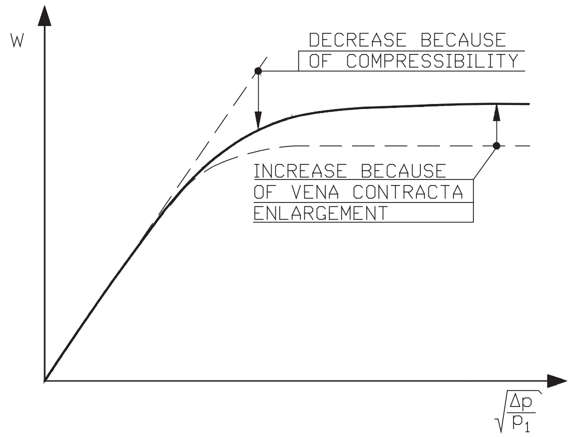


Figure 53. *Interrelation of flow and pressure drop ratio in gas or steam valves.*

The phenomenon of choking, i.e. a situation in which an increase in pressure drop (Δp) across the valve does not produce an increase in flow, also occurs in gas and steam flows, but here the reason is different. Whenever the fluid reaches sonic flow velocity at the vena contracta , the smallest actual flow area location, a further increase in pressure drop (Δp) at constant inlet pressure will not increase the flow velocity at the vena contracta, although supersonic velocities may exist downstream of the vena contracta. The flow can still be increased as a result of vena contracta enlargement. When the pressure drop is further increased beyond the sonic velocity at the vena contracta, the vena contracta moves upstream towards the valve orifice and the area of the vena contracta gets larger, therefore increasing the flow. When the vena contracta reaches the physical valve orifice the flow becomes fully choked, and an increase in the pressure drop will not increase the flow. The terminal pressure drop at which this happens can be calculated using the pressure drop ratio factor of valve (x_T) as in equation (50).

$$\Delta p_T = p_1 \cdot F_k \cdot x_T \tag{50}$$

4.2 Sizing equations for gas and steam flow

The gas or vapour flow through the valve can be calculated using any of these forms of basic equations (51) to (54).

$$w = N_6 \cdot F_p \cdot C_v \cdot Y \cdot \sqrt{x \cdot p_1 \cdot \rho_1} \tag{51}$$

$$q = N_7 \cdot F_p \cdot C_v \cdot p_1 \cdot Y \cdot \sqrt{\frac{x}{G_g \cdot T_1 \cdot Z}} \tag{52}$$

$$w = N_8 \cdot F_p \cdot C_v \cdot p_1 \cdot Y \cdot \sqrt{\frac{x \cdot M}{T_1 \cdot Z}} \tag{53}$$

$$q = N_9 \cdot F_p \cdot C_v \cdot p_1 \cdot Y \cdot \sqrt{\frac{x}{M \cdot T_1 \cdot Z}} \tag{54}$$

$$Y = 1 - \frac{x}{3 \cdot F_k \cdot x_T} \tag{55}$$

$$G_g = \frac{M}{M_{air}} \tag{56}$$

$$x = \frac{p_1 - p_2}{p_1} \tag{57}$$

$$F_k = \frac{k}{1{,}40} \tag{58}$$

where
- w = mass flow rate (SI units: [kg/h]; US units: [lb/h])
- q = volumetric flow rate (SI units:[Nm^3/h], in which a cubic meter is taken at 1.013 barA and 15°C; US units: [scfh], in which a cubic foot is taken at 14.73 psiA and 60°F)
- F_P = piping geometry factor
- C_v = valve flow coefficient
- p_1 = upstream pressure (SI units: [barA]; US units: [psiA])
- N_6 = 27.3 for SI units ([kg/h], [bar], [kg/m^3])
  = 63.3 for US units ([lb/h], [psi], [lb/ft^3])
- N_7 = 417 for SI units
  = 1360 for US units
- N_8 = 94.8 for SI units
  = 19.3 for US units
- N_9 = 2250 for SI units
  = 7320 for US units
- Y = expansion factor
- ρ_1 = upstream density (SI units: [kg/m^3]; US units: [lb/ft^3]
- G_g = gas specific gravity
- T_1 = upstream temperature (SI units: [K]; US units: [°R])
- Z = compressibility factor
- M = gas molecular weight
- M_{air} = gas molecular weight for air =28.96
- Fk = specific heat ratio factor
- k = ratio of specific heats

If $x \geq F_k * x_T$, then the flow becomes critical and Y = 0.667 and $F_k * x_T$ must be use d instead of x in equations (51) to (54).

For dry saturated steam, a simplified equation (59) can be used where x may not exceed x_T.

$$w = N_{10} \cdot F_P \cdot C_1 \cdot p_1 \cdot \left(3 - \frac{x}{x_T}\right) \cdot \sqrt{x} \tag{59}$$

where N_{10} = 6.6 for SI units [kg/h]
= 1.0 for US units [lb/h]

If $x \geq x_T$, then pressure drop ratio factor (x_T) must be used instead of pressure drop ratio (x). For a valve with attached fittings pressure drop ratio factor (x_T) has to be replaced by pressure drop ratio factor (x_{TP}).

x_{TP} can be calculated as in equation (60) using a given factor x_T.

$$X_{TP} = \frac{x_T}{F_P^2} \cdot \left[1 + \frac{x_T \cdot K_i}{N_5} \cdot \left(\frac{C_v}{d^2}\right)^2\right]^{-1} \tag{60}$$

where K_I = sum of inlet velocity head coefficients: $K_1 + K_{B1}$ (see equations (29) and (30))
N_5 = 0.00241 for SI units ([mm])
= 1000 for US units ([in])

Factor Z is the compressibility factor, and for perfect gas has a value of 1.0. In many flows, the fluid behaves as a perfect gas and the compressibility factor (Z) will be 1. Compressibility factor (Z) can determined from the graph in appendix G as a function of reduced pressure (p_r) and reduced temperature (T_r).

Reduced pressure (p_r) is defined as the ratio of the actual inlet absolute pressure to the absolute thermodynamic critical pressure, as in equation (61). The reduced temperature Tr is defined similarly, as in equation (62).

$$p_r = \frac{p_1}{p_c} \tag{61}$$

$$T_r = \frac{T_1}{T_c} \tag{62}$$

where p_c = thermodynamic critical pressure (SI units: [barA]; US units: [psiA])
T_c = thermodynamic critical temperature (SI units: [K]; US units: [°R])

4.3 Aerodynamic noise

4.3.1 Aerodynamic noise generation

The noise created by throttling gas or steam is called aerodynamic noise. Aerodynamic noise at subsonic velocities is generated by three forms of aerodynamic sources: monopole, dipole and quadropole. Aerodynamic monopole noise can be described as a pulsating sphere causing the acoustic pressure waves. This type of noise is typical of pulse jets, sirens and propellers. Monopole sources can also be caused by flow instability. The intensity of noise in this case is proportional to the fourth power of the flow velocity.

Control valve aerodynamic noise is mainly caused by dipole sources when the flow velocity is subsonic. An aerodynamic dipole consists of two monopoles pulsing 180°out of phase near a solid boundary. In this case the acoustic intensity is given by the following proportionality equation (63).

$$I \sim \rho^2 \cdot \frac{v^6 \cdot D_j^2}{r^2} \qquad \textbf{(63)}$$

where I = acoustic intensity
ρ = density of jet
v = velocity of jet
D_j = diameter of jet
r = distance from source

A very important implication in equation (63) is that the noise intensity is proportional to the velocity to the sixth power.

High-speed jet noise is generated by aerodynamic quadropoles, which consists of two dipoles pulsing in opposing pairs. According to Lighthill's classic work, the proportionality of noise intensity in this case is as in equation (64).

$$I \sim \rho^2 \cdot \frac{v^8 \cdot D_j^2}{r^2} \quad \textbf{(64)}$$

The noise intensity here is proportional to the eight power of the velocity. This applies to a free jet.

In choked flows, where the flow velocity reaches sonic velocity, the flow becomes critical, and the main sources of noise are the shock waves and turbulence interactions. Shock waves and turbulence interactions have been found to create a 'screech' from acoustic feedback. Figure 54 describes the change in noise intensity as a function of fluid velocity.

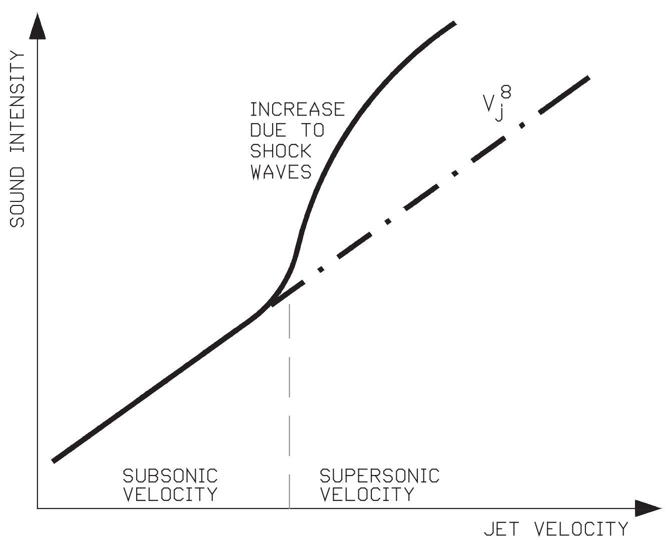


Figure 54. *Velocity and jet noise interrelation.*

Shock waves which are perpendicular to the free jet stream are called normal shocks or direct shocks. A normal shock is illustrated in figure 55.

The shock wave can also be oblique, if the flow passes a wedge or sharp object or if supersonic flow is forced to change direction by a solid boundary. Figure 56 shows an attached and a detached shock wave.

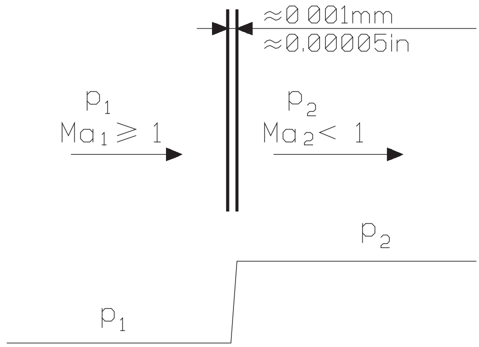


Figure 55. *Pressure change in a normal shock wave.*

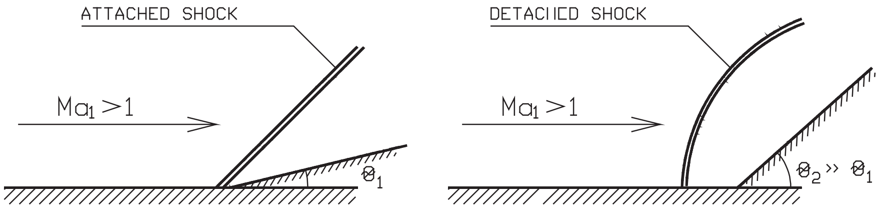


Figure 56. *Attached and detached shock waves.*

Shock waves can be reflected from solid boundaries and can penetrate each other. A control valve outlet flow stream can create an oscillating chain of compression shock waves with declining cycles and intensity, as shown in figure 57. This is a very powerful noise source.

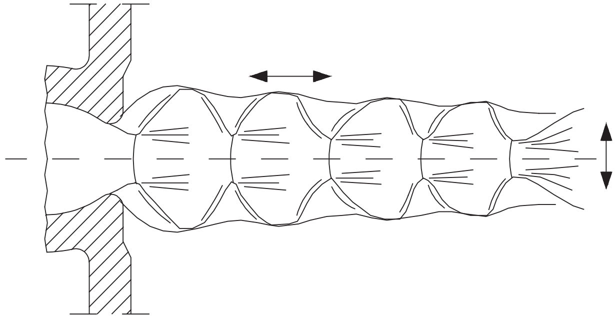


Figure 57. *A chain of compression shocks.*

With computational fluid dynamics it is possible to simulate shock waves in different geometries. Super computers are normally used because the calculation requires a great deal of complicated computations.

Figure 58 is a print-out of a calculation with a rotational symmetrical geometry. The lines in figure 58 represent constant pressure regions. The shock waves are where the lines are dense. The results calculated by the state-of-the-art computer programs are used in the design of low noise trims and attenuators.

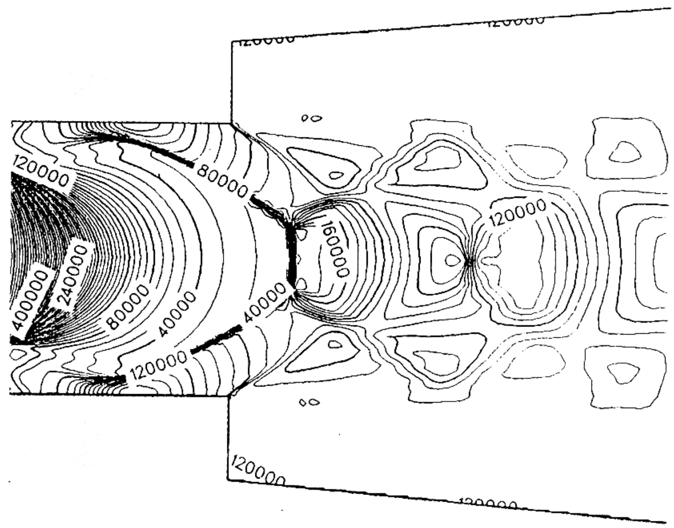


Figure 58. *Computational shock wave pattern downstream of rotational symmetrical geometry.*

4.3.2 Aerodynamic noise prediction

At low pressure ratios (p_1/p_2), the cause of aerodynamic noise is turbulence. At high pressure ratios, the flow becomes critical and the turbulent interactions of the shock waves become the major source of noise.

Calculation method to predict aerodynamic noise level

The noise level LA in dB(A) is calculated using the following equation (65) for SI units and equation (66) for US units. Equations are based on VDMA 24422 (1979 version) standard (see appendix K).

$$L_A = 14 \cdot \log(C_v) + 18 \cdot \log(p_1) + 5 \cdot \log(T_1) - 5 \cdot \log\rho_n + 20 \cdot \log\left[\log\left(\frac{p_1}{p_2}\right)\right] + 51 + \Delta L_G + \Delta L_p + \Delta L_{p2} \tag{65}$$

$$L_A = 14 \cdot \log(C_v) + 18 \cdot \log(p_1) + 5 \cdot \log(T_1) - 5 \cdot \log\rho_n + 20 \cdot \log\left[\log\left(\frac{p_1}{p_2}\right)\right] + 22{,}8 + \Delta L_G + \Delta L_p + \Delta L_{p2} \tag{66}$$

where

- ΔL_G = valve specific correction coefficient; usually presented in the manufacturers noise prediction graphs along the C_v-tables
- ΔL_P = correction coefficient for pipe wall thickness (see figure 48 and table 2)
- ΔL_{P2} = high downstream pressure correction
  - = 0 if p_2 < 30 barA (435 psiA)
  - = - (p_2 - 30 bar)/2.5 if p_2 is between 30 and 55 barA, or - (p_2 - 435 psi)/36.3 if p_2 is between 435 and 798 psiA
  - = - 10 if p_2 > 55 barA (798 psiA)
- C_v = valve flow coefficient
- p_1 = upstream pressure (SI units: [barA]; US units: [psiA])
- p_2 = downstream pressure (SI units: [barA]; US units: [psiA])
- T_1 = upstream temperature (SI units: [K]; US units: [°R])
- ρ_n = gas density (SI units: [kg/m^3]; US units: [lb/ft^3]) at normal conditions; pressure of 1.013 barA at 15.6°C or pressure of 14.7 psiA at 60°F

The aerodynamic noise prediction according to the IEC 60534-8-3 is very similar to IEC hydrodynamic noise prediction. The noise prediction is based on similar method where mechanical stream power as well as acoustical efficiency factors are calculated in five different flow regimes. Standard defines regimes as function of the pressure drop ratio (x). Then the sound pressure level inside the pipe is estimated and the pipe external noise is attained from inner sound pressure level by reducing the transmission loss of the piping.

Acoustical efficiency factors, the peak frequency and the mechanical stream power calculations are depending on the flow regime but otherwise the IEC aerodynamic noise prediction has similar steps than in hydrodynamic noise (see steps in chapter 3.5).

Due to the extent of the standard the reader is recommended to refer to the IEC 60534-8-3 for details.

4.4 Atmospheric venting

Venting gas or steam to the atmosphere usually generates high noise levels around the vent exit. In such cases, the jet noise inside the pipe released into the air and sound pressure levels from 140 up to 170 dBA one meter from the vent exit may be generated. Noise levels of this magnitude are hazardous to personnel and in many countries strict noise abatement laws require action to reduce noise to acceptable levels near the manufacturing facilities.

As a result of high noise level in atmospheric venting, a control valve is normally equiped with a low noise trim. Often a special vent diffuser or vent silencer is required to achieve an acceptable noise level.

The observation point for vent noise is usually far from the vent exit, which can be considered a point noise source, and the sound pressure level decreases as a function of distance, as in equation (67).

$$\Delta L_d = 20 \cdot \log\left(\frac{r}{r_0}\right) \tag{67}$$

where ΔL_d = change in sound pressure level [dBA]
r = distance from the reference point to the observation point
r_0 = distance from the centerline of the vent stackto the reference point; usually r_0 =1 m

When the vent exits are high above ground level, for example, in gas-to-flare applications, the observation direction has a substantial effect on the noise. The directivity effect attenuates most in opposite direction of the vent flow and least in the direction of the vent flow. Attenuation can be as high as 15 dBA depending on the observation direction.

Further away, in addition to the above reductions, more attenuation can be expected owing to environmental and other conditions such as absorption in to the air, attenuation by barriers, vegetation, and topography, or by wind and temperature gradients.

VDMA 24422 (version 1979) calculates noise only in a closed piping system and there are no standard methods for predicting noise at an open pipe end. That is why noise prediction for an atmospheric outlet uses an empirical method.

4.5 Aerodynamic noise abatement

Abatement of the noise produced in the throttling process can be done in several ways. The basic division of these is between 'source' treatment, i.e. valve and trim modification in a way that generation of excessive noise is prevented, and 'path' treatment, i.e. dampening the generated noise.

Source treatment, whenever possible and feasible, is the preferred form of noise abatement. This is because when source treatment is used the high mechanical vibration levels always associated with noise can be prevented, thus ensuring reliable operation of the process.

A combination of source and path treatment often provides the most economical sound dampening system. Figure 59 shows the basic options for aerodynamic noise abatement.

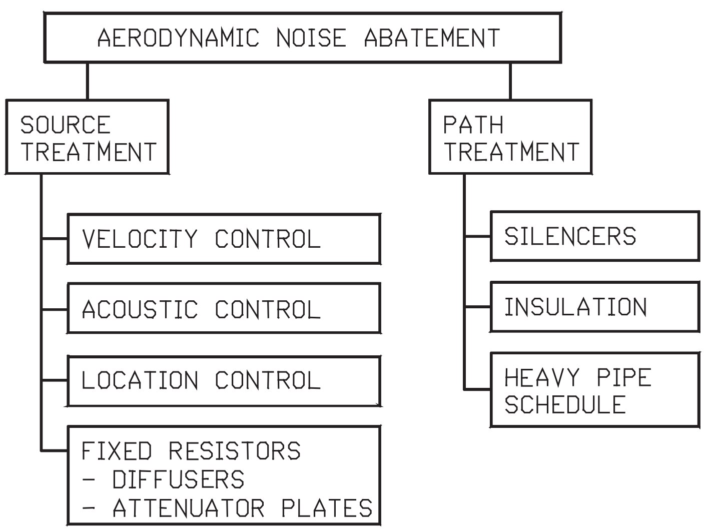


Figure 59. *Options in aerodynamic noise abatement.*

4.5.1 Source treatment

Source treatment of noise, i.e. preventing the generation of excessive noise levels, can be performed by at least four different methods: velocity control, acoustic control, location control and by using diffusers.

Velocity control

Controlling the maximum fluid velocity inside a control valve trim is a very effective way of controlling noise at sub-sonic flow velocities in the trim, as the acoustic intensity of a jet has been shown to be proportional to the sixth power of the flow velocity in a system with solid boundaries like a valve trim or a pipe. The relation between sound pressure level (SPL) and acoustic intensity is given in equation (69).

$$I \sim v_j^6 \qquad \textbf{(68)}$$

$$SPL(dB) = 10 \cdot \log \frac{I}{I_0} \qquad \textbf{(69)}$$

where I = acoustic intensity
I_0 = 10-12W/m^2
v_j = jet velocity

The control valve trim velocity can be most effectively controlled using a multistage pressure drop and by increasing the valve trim outlet area such that the flow velocity and pressure at the valve outlet are the minimum and the gas volume is the maximum. Figure 60 illustrates the basic principle of a staged pressure drop.

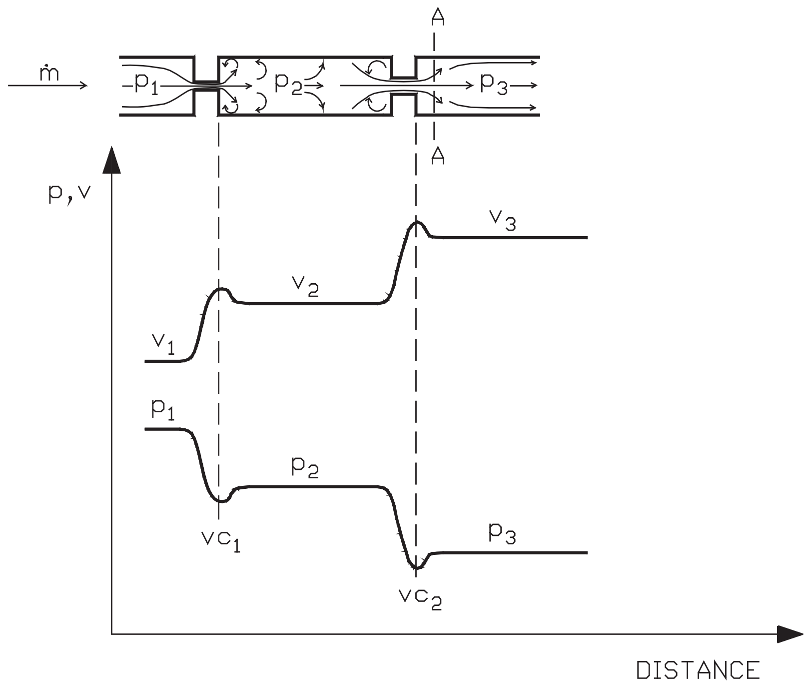


Figure 60. *Effect on flow velocity of a multi-stage pressure drop.*

When the pressure drop across a control valve is taken in multiple stages, as in figure 59, the total pressure drop (p_1 - p_3) is divided into several smaller successive pressure drops (p1 - p*2*, p_2 - p_3). The successive stages are spaced in such a way that the gas pressure is allowed to recover to an intermediate pressure (p_2) and velocity (v_2) before the next throttling stage. This intermediate recovery of pressure and the resulting velocity reduction prevents the fluid from reaching the velocity it would have in a single-stage pressure drop system. In other words, the smaller the pressure drop, the smaller the recovery and fluid velocity increase downstream of the orifice. An infinite number of stages would, therefore, produce a valve with no pressure recovery, in which the velocity at the vena contracta would equal the downstream velocity. However, economics and mechanics limit the number of stages actually used in valves. Note that an expanding flow area is used for each successive stage in gas processes to counter the expansion of the gas as the pressure drops.

Acoustic control

Acoustic control affects the noise level by means of acoustics. Two methods used in control valves are described here: flow division into multiple streams and the modification of acoustic field.

In theory, as given by equations (63) and (64), flow division into multiple streams is effective because intensity of noise generated by a single orifice decreases rapidly when hole diameter is decreased. Thus a number of small holes attenuates noise more effectively than one big hole. A rule of thumb is that each doubling of the number of holes reduces noise by 3 dB, as illustrated in figure 61.

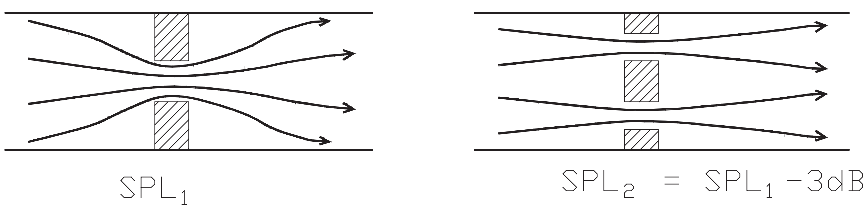


Figure 61. *Effect of increasing number of holes on noise level.*

In practice increasing the number of flow passages by making them smaller also affects the noise frequency distribution. Equation (70) is an experimental equation showing the noise reducing effect of smaller orifices.

$$\Delta SPL = 17 \cdot \log\left(\frac{D_{LHD}}{D_{SHD}}\right) \tag{70}$$

where ΔSPL = difference in sound pressure level [db(A)]
D_{LHD} = large hole diameter
D_{SHD} = small hole diameter

The smaller the passage, the higher the noise frequency. High-frequency noise is easily attenuated by the pipe wall, and does not contribute to the 'A'-weighted sound pressure level scale (figure 62). High frequencies also extend beyond the capabilities of the human ear.

Modification of the acoustic field directs the flow path in such a way that the peak noise field is broken up in a more diffused space. Figure 63 gives the values for a dynamically balanced butterfly valve as an example of acoustic field modification.

Figure 63 shows a typical acoustic field for a conventional butterfly valve (top of the picture), and for a noise and cavitation attenuating equipped butterfly valve which diffuses the checkered acoustic field.

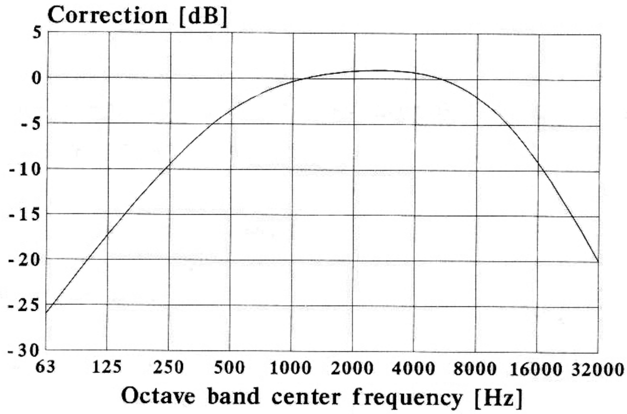


Figure 62. *A-weighted sound pressure curve for human ear.*

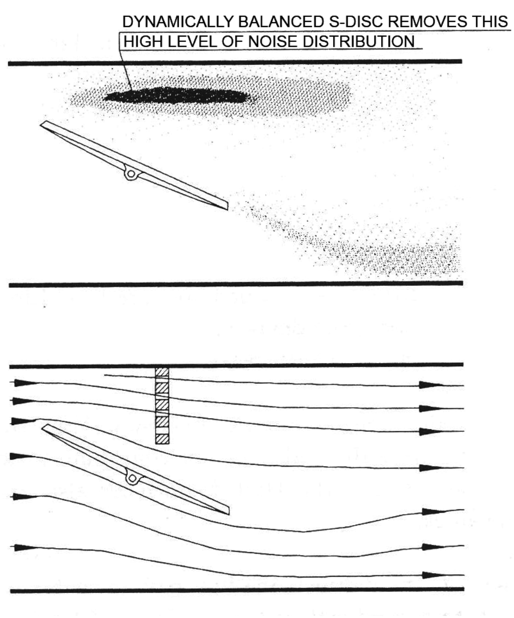


Figure 63. *Modification of acoustic field in a dynamically balanced butterfly valve.*

Location control

Location control involves designing a valve trim in such a way that the location and the shape of the jet streams in the valve trim, and especially leaving the valve trim, are such that the minimum noise is produced.

The formation of turbulence in the mixing region between where the jet exits from an orifice and the gas flow at the outlet region as well as attachment and interaction of shock waves (generated during throttling if the flow reaches sonic velocity in the valve) are major sources of noise that can be controlled to an extent by intelligent valve trim design. One way to do this is to smooth the velocity profile of the jet by introducing a lower velocity gas stream alongside the jet, as shown in figure 64.

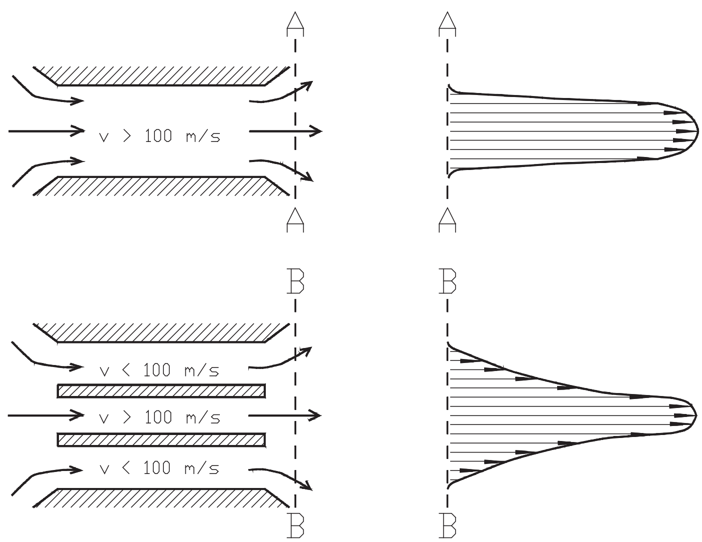


Figure 64. *Jet velocity profile modification.*

Other methods include using orifice shapes and spacing orifices ensuring that the shock wave interactions generate the minimum amount of total noise.

Diffusers

Dividing the pressure drop between a control valve and a downstream diffuser provides an effective way of further increasing the noise attenuation in cases where there is a constant, high pressure drop across the control valve and the flow rate is relatively constant. Diffusers are fixed area flow resistances and are custom-made for a particular flow condition.

A high pressure drop over the valve generally causes noise and vibration problems in the gas and steam flow. An inline diffuser is a constant area flow resistance used at the valve outlet to increase valve downstream pressure.

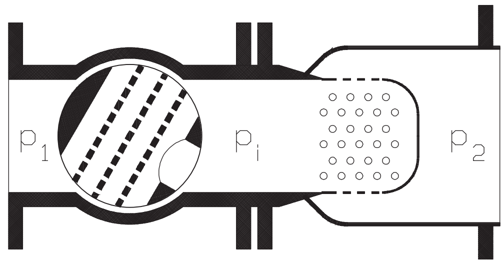


Figure 65. *Figure 65.Valve with single stage diffuser.*

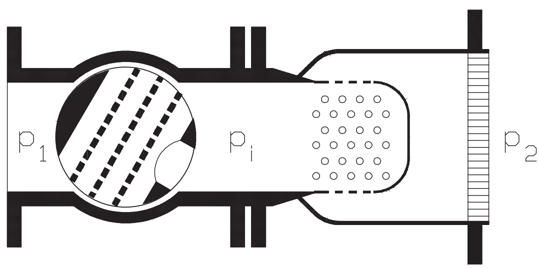


Figure 66. *Valve with double stage diffuser.*

Software tools are available for sizing diffusers.

It is recommended that a single-stage diffuser be used when pressure drop ratio, $(p_1 - p_2)/p_1$, under maximum flow conditions is higher than 0.65 and a double-stage diffuser when the ratio exceeds 0.75. These recommendations are for valves equipped with noise attenuating trims. In the case of a standard valve with a diffuser and a pressure drop ratio, $(p_1 - p_2)/p_1$, smaller than 0.65, a different arrangement may be needed.

Depending on the type of diffuser, single or double-stage, it is recommended that the pressure differential $(p_1 - p_i)$ across the valve be set according to the graphs in figure 67.

The pressure differential $(p_1 - p_i)$ is set for flows in which the highest capacity (C_v) is needed for the valve. The capacity (C_v) of the diffuser is calculated under maximum flow conditions using equations (51) to (54). Intermediate pressure (p_i) and corresponding density (ρ_i) are used as upstream pressure and density. The pressure drop ratio factor (x_T) is 0.5 for a single-stage and 0.7 for double-stage diffuser. The capacity (C_v) of the diffuser remains constant, and this is used to calculate $(p_1 - p_i)$ and the intermediate pressure (p_i) for other flow conditions.

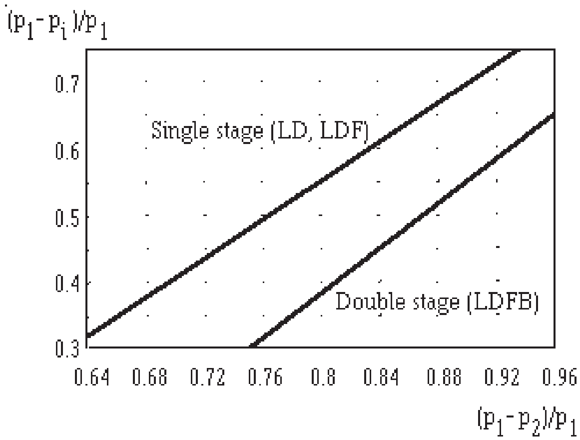


Figure 67. *Pressure differential selection for line diffuser.*

In noise calculations, the noise from the valve and diffuser are first calculated separately using standard VDMA 24422 noise prediction method. The intermediate pressure (p_i) is the valve downstream and diffuser upstream pressure. The noise level of the total package is achieved by summing both noise levels logarithmically. A 6 dBA insertion loss is subtracted from the valve noise level and the total noise level is then given by equation (71).

$$L_{PTOT} = 10 \cdot \log(10^{LpV-6/10} + 10^{LpD/10}) \tag{71}$$

where L_{pV} = predicted sound pressure level of a valve [dBA]
L_{pD} = predicted sound pressure level of a diffuser [dBA]

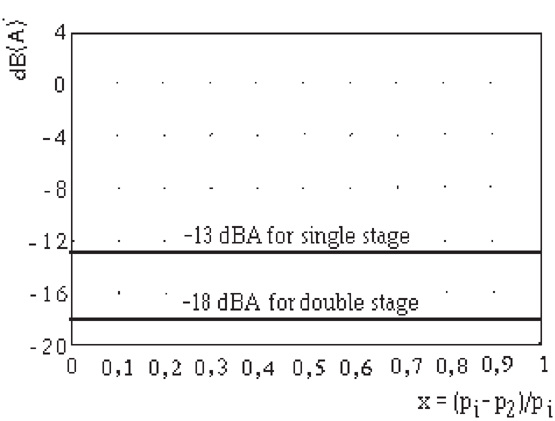


Figure 68. *Aerodynamic noise coefficient ΔL_G for diffusers as a function of pressure drop ratio (x), as $x = (p_i - p_2)/p_i$.*

The outer shell diameter of the diffuser is normally sized to match the downstream piping. However, the flow velocity in the annular area between the inlet tube and the outer shell of the diffuser should not exceed 0.5 * v_s if regenerated noise caused by high flow velocity is to be prevented.

The minimum outer shell diameter to meet this requirement is given by equation (72).

$$D_{out} = \sqrt{D_{in}^2 + \frac{w}{N_{15} \cdot v_s \cdot \rho_2}} \qquad \textbf{(72)}$$

where
- w = mass flow rate (SI units: [kg/h]; US units: [lb/h])
- v_s = speed of sound in downstream conditions, presented in equation (49) (SI units: [m/s]; US units: [ft/s])
- ρ_2 = downstream density (SI units: [kg/m^3]; US units: [lb/ft^3])
- D_{out} = internal diameter of outer shell (SI units: [mm]; US units [in])
- D_{in} = external diameter of in ner tube (SI units: [mm]; US units [in])
- N_{15} = 1.414*10-3 for SI units ([kg/h], [m/s], [kg/m^3], [mm])
  = 9.821 for US units ([lb/h], [ft/s], [lb/ft^3], [in])

A smaller outlet shell than calculated from equation (72) can be used if, for example, the outlet shell should be the same size as the downstream pipe. In these cases the fact that flow velocity in the annular area will be greater than 0.5*vs resulting in limited diffuser noise reduction capability for the diffuser must be taken into account.

When the outlet shell specified by equation (72) is bigger than the downstream piping, the reduction of the downstream pipe can begin straight after the end of the diffuser inlet tube.

Attenuator plates

Dividing the pressure drop between the valve and a downstream device provides an effective methods noise attenuation in cases where the pressure drop is close to constant across the valve. An attenuator plate is a fixed restriction installed downstream of the valve, as shown in figure 70.

High pressure drop across a valve generally causes noise and vibration problems in the gas and steam flow. An attenuator plate is a constant-area flow resistance, which is used at the valve outlet to increase valve downstream pressure. Noise attenuation is achieved by specific hole geometry. This hole geometry is a result of computational and experimental studies. Figure 69 is a printout of a calculation of the flow through one hole in an attenuator plate. The lines in figure 68 represent constant density regions.

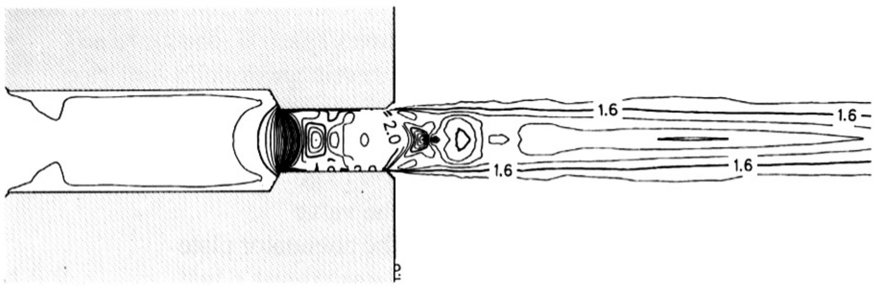


Figure 69. *Constant density regions of flow through a hole in an attenuator plate.*

The attenuator plate capacity is fixed for each nominal size. If a higher capacity is needed, the nominal size of the attenuator plate has to be increased. Normally the nominal size must be selected between the valve and downstream pipe diameter. Pressure and flow conditions with the selected plate are calculated using standard ISA/IEC sizing equations.

The attenuator plate is mounted between flanges downstream of valve. Mounting can be between the valve and pipe flanges or between piping flanges. For some valves an attenuator plate are also available as an integrated unit inside the valve body.

Mounting the attenuator plate between the valve and pipe flanges is not allowed if the valve is a butterfly valve, or flangless segment valve or if the attenuator plate is already integrated into the valve body. In these cases at least one pipe diameter of pipe is required between the valve and the attenuator plate.

The attenuator plate is usually effective for noise reduction when the pressure drop ratio $(p_1-p_2)/p_1$ is less than from 0.8 to 0.9, depending on the valve type and opening. A diffuser is recommended for higher pressure drop ratios.

The pressure differential across the attenuator plate is calculated for each flow according to basic gas sizing equations. The intermediate pressure (p_i) is iterated from equations (51) to (54) replacing upstream pressure (p_1) with intermediate pressure (p_i). Sizing is most easily done with a sizing and selection software. The critical pressure drop ratio for an attenuator plate is 0.39. C_v is obtained by selecting the nominal size of the attenuator.

Noise calculations for valve and attenuator plate are first performed separately using intermediate pressure (p_i) as both the valve downstream and the attenuator plate upstream pressure. The noise level of the total package is found by summing both noise levels logarithmically. A 4 dBA insertion loss is subtracted from the valve noise level. The total noise level is then as given in equation (73).

$$L_{PTOT} = 10 \cdot \log(10^{L_{pV} - 4/10} + 10^{L_{pA}/10}) \tag{73}$$

where L_{PTOT} = total noise level
L_{pV} = calculated noise level of the valve
L_{pA} = calculated noise level of the attenuator plate

Source treatment examples

Source treatment is demonstrated using rotary type noise control trim valves, such as in figure 70.

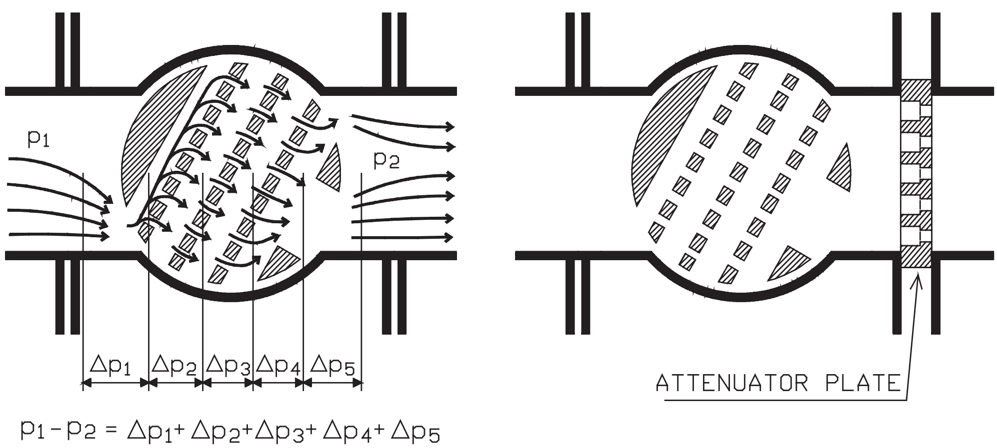


Figure 70. *A noise control trim provides a combination of effects for reducing noise level in gas controlling applications; in right, an attenuator plate is installed between valve and pipe flanges.*

These noise control trims use a combination of the effects listed in the following paragraphs:

- The trim velocity is minimized using a multistage pressure drop. Trim outlet velocity is minimized by opening up the downstream cheek of the trunnion mounted ball valve or by using segment valves with no restriction at the trim outlet.
- Division of the flow into multiple streams in attenuator holes attenuates the noise following the principles of acoustic control.
- The trim outlet is designed to provide a uniform acoustic field in the valve outlet port, thus minimizing disturbances creating high-noise areas in the acoustic field.

The rotary motion of these valves has the additional benefits of allowing very large flow ranges and the passing of impurities through the valve.

4.5.2 Path treatment

The path treatment of noise, i.e. suppressing the transmission of excessive noise levels, can be performed by using least three different methods: silencers, insulation and by heavier pipe schedule.

Silencers

Silencers are mufflers used in-line or at an outlet to the atmosphere to dampen the noise produced. Reactive silencers create frequency interactions to dampen noise, whereas dissipative silencers use sound-absorbing materials like glass-fibre to dampen the noise.

The main use for silencers is on atmospheric outlets for gas or steam. In such cases, the jet noise inside the pipe is released to the open air and very high noise levels can easily develop. Typically, a combination of reactive and dissipative methods are used in atmospheric vent silencers.

Insulation

Pipe insulation may be used to dampen noise in gas and steam lines, particularly in steam lines, where there is already thermal lagging. Thermal insulation can dampen noise 1-2 dB(A) per 10 mm of insulation thickness (3 to 5 dB(A) per inch). The practical maximum is about 12 dB(A) due to leaks and acoustic bridges. Special acoustic insulation reduces noise up to 4 dB(A) per 10 mm of insulation thickness (10 dB(A) per inch). The maximum attenuation for acoustic insulation is 20-25 dB(A).

Two things should be noted in using insulation for dampening noise: First, a noise level greater than 100 dB(A) should primarily be handled by source treatment methods to avoid hiding potential vibration damage by insulation. Second, the true cost of installing acoustic insulation should be calculated before using it for long pipelines. This is because noise is attenuated very slowly in piping filled with gas or vapour, and because long pipe runs may have to be insulated in order to achieve acceptable noise reduction.

Heavy downstream pipe schedule

A heavier downstream pipe schedule can be used to dampen noise, as a part of the noise is generated by vibrating pipe wall. The heavier the pipe wall (bigger schedule), the less it will vibrate. Note that the noise level inside the pipe does not decrease when the change is made into a heavier wall pipe. The heavier wall pipe should therefore be used for the whole downstream piping to avoid recurrence of the high noise level. This in turn can be very costly in the case of long pipe runs.

4.6 Recommended flow velocities and limits for noise levels

The flow velocity for gases or steam downstream of the valve can be calculated as in equation (74).

$$v = N_{16} \cdot \frac{q \cdot T}{p_2 \cdot d^2} \tag{74}$$

where
- v = velocity (SI units: [m/s]; US units [ft/s])
- N_{16} = 1.23 (SI units)
  = 0.00144 (US units)
- q = volume rate of flow (SI units: [Nm^3/h]; US units [scfh])
- T = absolute temperature (SI units: [K]; US units [R])
- p_2 = valve outlet pressure (SI units: [barA]; US units [psiA])
- d = internal diameter of valve exit port (SI units: [mm]; US units: [in])

The recommended flow velocity at the valve exit port is given in terms of sonic velocity, as in inequality equations (75) and (76).

$V_{max} \leq 0{,}5 \cdot v_s$ for continous throttling duty **(75)**

$V_{max} \leq 0{,}7 \cdot v_s$ for infrequent occasions such as gas- to- flare and blow-off valves **(76)**

where v_s = speed of sound, presented in equation (49)

Naturally, the above velocity limits are for relatively pure gases and steam. In the case of impurities in the flow or wet steam, the flow velocities need to be lowered to prevent erosion damage.

To prevent mechanical damages of a valve and to ensure operation of control valve instrumentation, it is recommended that sound pressure levels (SPL's) exceeding 110 dBA (calculated with uninsulated Schedule 40 piping) is never used.

5 MULTI-PHASE FLOW

5.1 General

Control valve sizing theory for multiphase flow is not nearly so well developed as it is for single-phase fluids. Sizing control valves for a pure liquid or gas flow can be done using existing standard sizing equations based on flow dynamics and valve- specific sizing coefficients. When a control valve is sized for a two-phase flow, which is usually a mixture of liquid and gaseous fluid, no generally accepted standard method exists. This is because in two-phase flow liquid and gas cannot be mathematically described in a simple and exact way at the same time. Also, experimental research requires a lot of tests with different kinds of mixtures and mass fractions using various valve types. For this reason it is not possible to size valves for multi-phase flow with the same accuracy as for single component streams.

A large part of this section concentrates on sizing control valves for a particular multiphase fluid, i.e. pulpstock. Valmet has done a lot of the basic research on determining the behaviour of pulpstock flow through control valves. As a result of this research there is now introduced a method for sizing control valves for pulpstock flows that applies to all rotary control valve types.

5.2 Two-phase flow of liquid and gas

The method presented here is based on a homogeneous flow theory which assumes that liquid and gas move with the same velocity and are homogeneously intermixed. The method can be applied in the following two cases of two-phase flow presented below.

Case 1 Single component, two-phase flow (e.g. steam and liquid water)

Case 2 Two component, two-phase flow (e.g. crude oil and natural gas)

Sizing equations

Homogeneous flow theory is based on average properties such as the density and velocity of a two-phase mixture. After necessary average properties have been determined, a valve can be sized using equations similar to the standard equations for single-phase flow.

The density of two-phase flow is calculated using separate densities for both phases on the upstream side of a valve. In addition, gas expansion must be taken into account when gas flows through the valve. The density of the mixture, the so called effective density, can then be formulated as in equation (77).

$$\rho_E = \left(\frac{f_g}{\rho_g \cdot Y^2} + \frac{f_f}{\rho_f}\right)^{-1} \tag{77}$$

where
- f_g = w_g/w ; the fraction of gas mass flow rate of the total mass flow rate
- f_f = w_f/w ; the fraction of liquid mass flow rate of the total mass flow rate
- ρ_f = density of the liquid-phase on the valve inlet side (SI units: [kg/m^3]; US units: [lb/ft^3])
- ρ_g = density of the gas-phase on the valve inlet side (SI units: [kg/m^3]; US units: [lb/ft^3])
- Y = gas expansion factor

$$w = N_6 \cdot F_p \cdot C_v \cdot \sqrt{\Delta p \cdot \rho_E} \tag{78}$$

Capacity for a non-choked two-phase flow is then given by equation (78).

where
- w = $w_g + w_f$; total mass flow rate of the mixture (SI units: [kg/h]; US units: [lb/h])
- F_p = piping geometry factor
- C_v = valve flow coefficient
- ρ_E = effective density (SI units: [kg/m^3]; US units: [lb/ft^3])
- Δp = pressure drop across the valve (SI units: [bar]; US units: [psi])
- N6 = 27.3 for SI units
  = 63.3 for US units

Choked flow

Experimental studies of choked two-phase flow are too few to allow exact terminal pressure drops to be determined. In control valve sizing, the terminal pressure drop approximation is performed using a combination of the terminal pressure drop values of pure gas and pure liquid. Pure gas flow chokes when the pressure drop reaches the value given by equation (79).

$$\Delta p = p_1 \cdot F_k \cdot x_{TP} \qquad \textbf{(79)}$$

The corresponding pressure drop for a pure liquid flow is given by equation (80).

$$\Delta p = \left(\frac{F_{Lp}}{F_p}\right)^2 \cdot (P_1 - F_F \cdot P_v) \qquad \textbf{(80)}$$

In a case where all the fluid is in the liquid-phase flow, choking will start when the pressure drop in equation (80) is reached. When a small fraction of gas is added to the flow, the choked flow pressure differential changes, but is still quite close to the value in equation (80). An increased mass fraction of gas will further change the terminal pressure drop that produces choking, but it is not clear how this happens in control valves. Finally, when all the fluid is in the gas-phase, choking will start with a pressure drop as in equation (79). Theoretical calculations and flow tests have been performed to determine the terminal pressure drop for ideal nozzles. These results show that the linear relationship between the terminal pressure drop in the liquid and gas-phase as a function of gas mass fraction describes the terminal pressure drop in choked two-phase flow accurately enough, and can be formulated as in equation (81).

$$\Delta P_{T-2PH} = (P_1 \cdot F_k \cdot x_{TP}) \cdot f_g + \left(\frac{F_{Lp}}{F_p}\right)^2 \cdot (P_1 - F_F P_v) \cdot f_f \qquad \textbf{(81)}$$

When the actual pressure drop exceeds the value of equation (81) the two-phase flow is considered to be a choked flow, and the pressure drop in equation (81) is used in the valve sizing equation (78), and to calculate the expansion factor Y. Note that the minimum value for expansion factor Y is Y_{min} (= 0.667).

Accuracy

Due to the nature of the two-phase flow of liquid and gas it is impossible to describe the various possible forms of flow adequately with a single mathematical formula. The method presented here is based on so called homogeneous flow theory, which assumes that the velocities of liquid and gas are the same, and that they are completely intermixed. This is a fairly common flow type, so the method described can be applied in many two-phase flow applications.

Accuracy of sizing decreases if the form of the flow deviates from the type described. The following forms of flow are possible in a pipe:

bubble flow:	gas-phase is divided into bubbles and liquid; the bubbles have the velocity of the liquid
plug and slug flow:	when the amount of gas increases, the bubbles unite, forming-bullet shaped plugs and slugs
churn flow:	when the amount of gas increases further, the plug and slug shapes break, leading to a very unstable form of flow
annular flow:	the liquid flows as a thin film along the pipe wall and the gas flows at high speed in the middle
stratified flow:	in a horizontal pipe the phases are completely separate from each other due to gravity. When the speed of the gas increases, waves form on the surface of the liquid.
mist flow:	almost all the liquid is in drops which move together with the gas

Phase changes in the liquid and gas, vapourization of the liquid or condensation of the gas into liquid make the calculation of mass fractions and effective density difficult. These factors in sizing accuracy are especially evident with single-component two-phase flow. When the pressure decreases and the temperature is almost constant, the liquid tends to evapourate whereby the mass fraction of vapour and the required valve capacity increase. On the other hand, the so called metastability phenomenon tends to 'slow down' the phase change, which means that the liquid does not vapourize at once in the vena contracta although the thermodynamic equilibrium of the substance would indicate the case to be such, but vapourization occurs only after the vena contracta.

The effect of mass fractions on sizing accuracy is especially visible with small mass fractions of vapour. For example, a change in the mass fraction from 1 % to 2 % of a saturated 7 barA water and steam mixture causes a 73 % change in the specific volume of the mixture. This means that the required capacity increases by about 30 %. On the other hand, if the mass fraction of the flow changes from 98 % to 99 %, the specific volume of the mixture changes by 1 %. If the mass fractions with single-component two-phase flow are not known exactly, the sizing can be checked by assuming the whole mass flow as vapour flow, which guarantees that the valve capacity is adequate in all situations.

Noise

There are no methods for predicting noise in two-phase flow. In practice, estimation of noise in two-phase flow is very difficult. It is known from experience that cavitation noise in pure liquid flow is lower if, for instance, air is mixed with the liquid. Air bubbles attenuate the pressure waves created by the collapse of cavitation bubbles.

5.3 Pulpstock

Paper and pulp mills use a large number of control valves to throttling the flow of pulpstock. The costs of both building and running the process are largely determined by how the throttling control valves are sized and selected, as the selection process determines the pumping energy 'burnt' in the control valve and the piping. Correct selection of the pulpstock control valves can determine the selection of the optimal size pumps and piping. In the past, over-dimensioning of pumps and valves have been used for all the uncertainties in the behaviour of the pulpstock flow. Rough correction coefficients has been used to account for the greater capacity (C_v) requirement of pulpstock compared with water. Now, new studies go a long way towards predicting exactly how pulpstock behaves in control valves, and towards improving the accuracy of sizing calculations.

Previous studies, mainly performed to evaluate consistency effects on pipe flow, it indicate that pulpstock flow behaviour is highly dependent on flow velocity, as shown in figure 70.

When the flow velocity of the pulpstock is very small, it flows in the pipe like a solid plug and forms 'rolls', which spin between the plug and pipe wall. As the flow velocity is increased, the flow changes into a plug flow, in which the fibers no longer touch the pipe wall as there is a narrow boundary layer between the plug and the pipe wall. In this area the pressure loss remains relatively constant in spite of flow velocity changes, and can even have an inverse relation to the flow velocity increase (unstable area in figure 71). As the flow velocity is increased further, the thickness of the boundary layer increases and turbulence starts appearing in the boundary layer and at the surfaces of the pulp plug. The centre of the pulp plug remains pluglike. This is called mixed flow. When the flow velocity is further increased, the pulp plug disappears and the whole flow crosssection becomes turbulent.

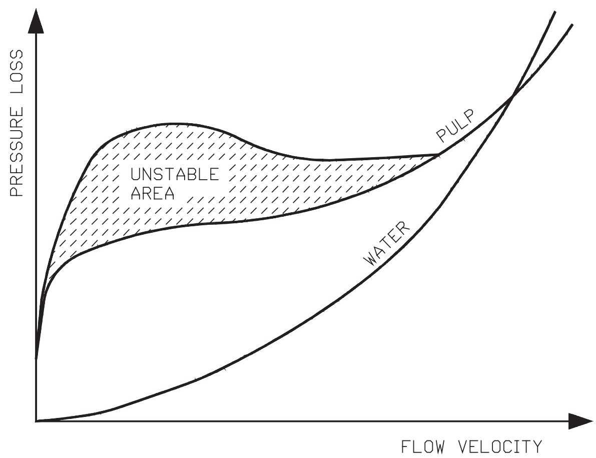


Figure 71. *The piping pressure loss in relation to flow velocity for pulpstock and water describes the changes in piping pressure loss at different flow velocities in pulpstock and water flows.*

The three basic forms of pulpstock flow are illustrated in figure 72.

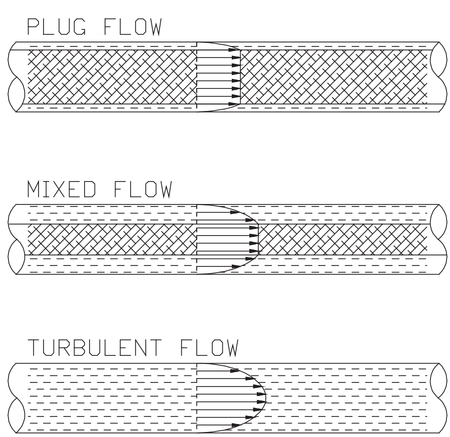


Figure 72. *Pulpstock flow types.*

Note also that at high flow velocities, as in figure 71, the pressure loss of the pulpstock can be lower than that of water.

5.3.1 Studies about pulpstock

There has been performed over 2000 measurements of pulpstock flow through control valves. The tests were performed on three different types of quarterturn control valve in three different sizes (50 mm/2", 100 mm/4" and 200 mm/8"). The pulp stocks used were bleached kraft (sulphate pulp), thermo-mechanical pulp and recycled pulp. The tests resulted an empirical correction coefficient (k), which is presented in appendix J. The coefficient (k) is determined by equation (82).

$$k = \frac{q_{stock}}{q_{water}} \quad \textbf{(82)}$$

where q_{stock} = measured flow rate of pulpstock flow
q_{water} = equivalent flow rate of water

Note that a correction of the flow rate is used instead of a correction in the required $C_v$value. This is done so that valves would be sized using correct sizing parameters pressure recovery factor (F_L) and cavitation index (z), which relate to the real opening of the control valve.

The differences in the results for different size control valves and different quarterturn valve types were found to be insignificant, and thus to justify using a single set of correction curves for each type of pulpstock. The correction coefficients (k) are given in appendix J for both SI units and US units.

5.3.2 Pulpstock behaviour in control valves

There has been establish the following relationships in the pulpstock flow through control valves:

- The coefficient (k) increases with the increase in differential pressure.
- The coefficient (k) decreases with the increase of pulp consistency.

When the consistency is lower than 2 %, there is very little difference between water and pulp flow rates, and sizing can be done without a correction coefficient (k) using only water sizing equations.

The effect of differential pressure on pulp flow

The differential pressure across a pulpstock control valve has the largest effect on the correction coefficient k. In particular, very small differential pressures require a large correction (= small k value).

The effect of consistency on pulp flow

The tests show clearly the effect of pulp consistency on the correction coefficient. When consistency is between 1.5 % and 2 %, the pulpstock behaves much like water. When consistency increases to, say, 12 % the value of the correction coefficient k can be in the range of 0.3 0.6 at small differential pressures, depending on the type of pulp. This finding is very important for selecting valves for MC pump discharge control. Expanded outlet valves provide a higher capacity (C_v) than standard segment valves in order to allow for small k coefficients.

The effect of valve diameter and style on pulp flow

The valve diameter has a very small effect on the correction coefficient, i.e. the larger the valve the lower the k coefficient. This difference is so small that, for all practical purposes, correction coefficients used for sizing purposes can be kept constant for all valve sizes. No measurable differences have been found between the ball, segment and butterfly valve k coefficients. Thus the coefficients are the same for all rotary valve styles.

The effect of other factors on pulp flow

The tests performed to develop the coefficient k curves were done using valve inlet pressures common to typical pulp and paper mill typical pump heads, and can be considered constant for typical pulp and paper mill environments, independent of inlet pressure. Various freeness factor samples from each pulp stock brand were used for testing. No significant effect was observed due to freeness variation.

In light of these experiments, k coefficients can be expressed as a function of pressure drop, pulp consistency and pulpstock type with sufficient accuracy for control valve sizing calculations.

5.3.3 Control valve sizing for pulpstock applications

Control valve sizing for pulpstock applications is performed using an 'equivalent water flow rate' and water sizing equations. The 'equivalent water flow rate' is derived from equation (83).

$$q_{water} = \frac{q_{stock}}{k} \tag{83}$$

where q_{stock} = pulp stock flow rate
k = correction coefficient (see appendix J)

After calculating an equivalent water flow rate, the sizing is done utilizing standard IEC/ ISA sizing equations for water, as presented in chapter 3.

Figure 73 shows an example of an installed flow characteristic for a 10 % mechanical pulp control valve in a typical process piping system.

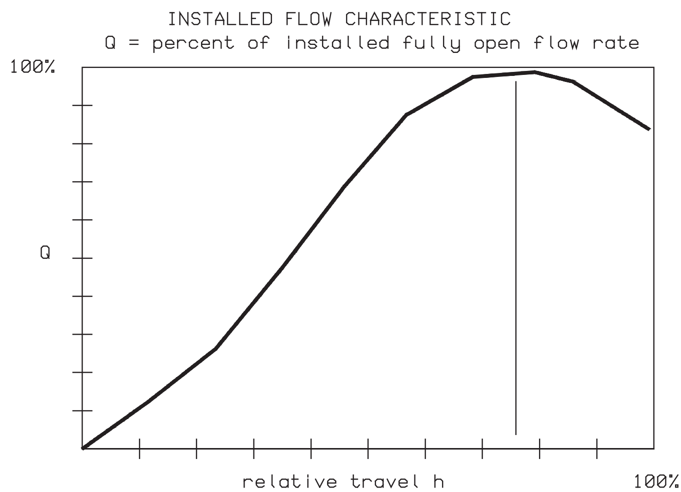


Figure 73. *Installed flow characteristic for a 10 % pulpstock control valve.*

Figure 72 shows that when the valve opening is already relatively large, the actual flow decreases if the valve is opened further. This kind of situation has to be avoided in order to maintain the fluid flow control of the process.

In the described case, the valve maximum opening should be set at about 55 % to guarantee process controllability, and a valve with an expanded outlet should be used. In the above case, when the valve maximum opening is set at 55 %, increasing the valve opening, the capacity (C_v) increase produces a constantly larger flow.

5.4 Slurries

A typical twophase flow of a liquid and solids is wet slurry. Wet slurry is a mixture of a liquid and solid particles. Suspensions are slurries if the particle sizes are so small that the mixture remains unseparated for a long time. Suspensions can be treated as non-Newtonian fluids, where basic liquid equations can be applied to the calculated 'apparent viscosity' for the suspension.

Settling slurries are suspensions in which particle size is bigger than 0.25 mm (0.01 in). These can be calculated as liquids, as long as the two phases can be assumed to flow through the control valve as a homogenous mixture; with the densities of different phases close to each other, and viscosity small enough to guarantee effective mixing. The density (ρ) in that case is taken as the effective density (ρ_E) weighted mass ratios of liquid and solids.

Solving the twophase flow of suspensions is complicated because it is a complex flow type. It involves calculating the 'apparent viscosity' and determining the necessary parameters, as well as a liquid calculation taking into account the Reynolds number. The details of the calculation are not presented here, but are available from valve manufacturers.

A typical twophase flow of dry slurry, is a mixture of gas and solids. This kind of mixture is rarely controlled by conventional valves due to mechanical problems like erosion and plugging. If a control valve is to be sized for dry slurry, equation (78) gives the best results, although it is slightly conservative. In this case, the density of the mixture is taken as the effective density in equation (77), where the liquid is replaced by a solid.

6 MATHEMATICAL SIMULATION OF CONTROL VALVE BEHAVIOUR

6.1 General

The use of computational science has resulted in a control valve simulation program that can be used to describe the dynamic behaviour of control valves.

6.2 Use of simulation

Use of simulation can be divided into three areas:

1. In development of new products.
2. In support for customer's process applications.
3. In training of personnel about fluid flow control and control valves.

In the development of new products simulation can be used to test new ideas, develop and optimize constructions before a prototype is manufactured. Simulation thus reduces the number of prototypes required. It can also be used to study the effects of different disturbances on control valves.

Before the delivery to a customer, the operation of a control valve packages can be simulated in conditions that equal the real process conditions. Simulation makes it possible to study the operation of the whole control loop including the customer's process. The simulation program can also be used to study various questions relating to the safety of process plants.

In training of personnel about control valves, simulation can be used to illustrate the operation of different control valve units in process pipelines. In addition, simulation makes it possible to examine different control valve characteristics in the whole control loop.

Simulation of customer's control applications is state-of-the-art service which is designed to study the control valve dynamic behaviour before delivery. Detailed dynamic analysis performed in simulated process conditions enhances the optimal control valve selection for customer's demanding control valve applications.

6.3 Mathematical model of a control valve

The starting point for the control valve simulation program is a comprehensive mathematical model describing the operation of the control valve, right from the control signal from the controller to the fluid flow through the valve.

The dynamics equation of a quarter-turn control valve can be found using the basic equations (84) and (85).

$$\sum \bar{F}_i = m_{red} \cdot \frac{d^2 \bar{x}}{dt^2} \tag{84}$$

$$\sum \bar{M}_i = J \cdot \frac{d^2 \bar{\psi}}{dt^2} \tag{85}$$

where
- x = distance covered by actuator piston
- ψ = valve turning angle
- F = force
- M = torque
- m_{red} = reduced mass for actuator piston and lever mechanism
- J = combined inertia moment of actuator and valve

Equations (84) and (85) can be used to produce the equations for the dynamics of an installed quarter-turn control valve as in equation (86).

$$\begin{aligned}
&\left[m_{red} \cdot \frac{dx}{d\psi} + J \cdot b(\psi, \mu)\right] \cdot \frac{d^2\psi}{dt^2} + m_{red} \cdot \frac{d^2 x}{d\Psi^2} \cdot \left(\frac{d\psi}{dt}\right)^2 \\
&+ \left[f_m \cdot \frac{dx}{d\psi} + f_v \cdot b(\psi, \mu)\right] \cdot \frac{d\psi}{dt} \\
&+ \operatorname{sign}\left(\frac{d\psi}{dt}\right) \cdot b(\psi, \mu) \cdot M_{v\mu}(\psi, \mu, \Delta p) \\
&+ b(\psi, \mu) \cdot M_d(\psi, \Delta p) - F_m(P_A P_B) = 0
\end{aligned} \tag{86}$$

where
- μ = friction coefficient
- b = actuator coefficient
- $M_{v\mu}$ = valve friction torque
- M_d = dynamic torque
- f_m = actuator damping coefficient
- f_v = valve damping coefficient
- F_m = piston force
- p_A, p_B = pressures on piston
- Δp = pressure difference across the valve

The actuator coefficient (b) used in equation (86) is a function of the actuator turning angle (ψ) and friction coefficient (μ). Figure 74 shows the actuator coefficient for one manufacturer's actuator as a function of actuator relative travel (h) when the friction coefficient (μ) is assumed to be constant.

The non-linearity of the actuator coefficient (b) depends on the lever mechanism of the actuator. In practice, actuator friction coefficients vary greatly as a function of velocity. This means that the actuator coefficient is in reality less linear than indicated in figure 74.

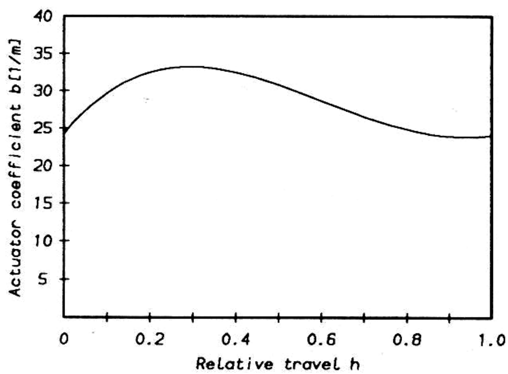


Figure 74. *Actuator coefficient of the actuator examined.*

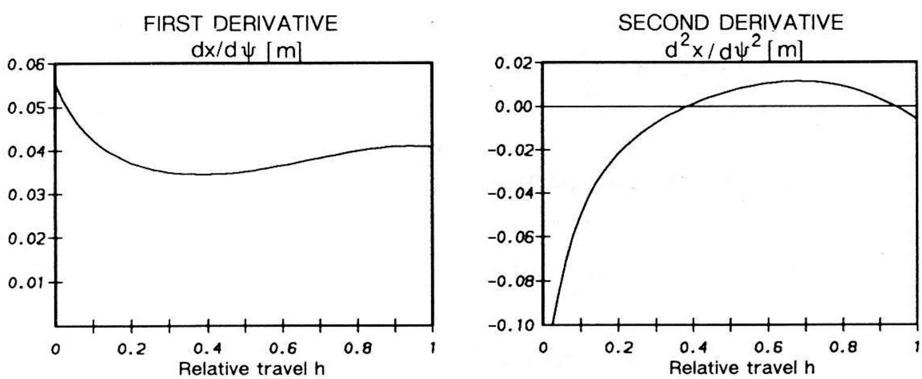


Figure 75. *Derivatives in equation (86) as a function of relative travel*

Figure 75 shows the non-linearity of the derivatives in equation (86) for the actuator examined, as a function of actuator relative travel (h). The first derivative in figure 74 describes the non-linearity between the actuator piston position (x) and the valve turning angle (ψ). The non-linearity of the derivatives depends on the lever mechanism of the actuator.

As the control valve dynamic equation (86) includes pressures p_A and p_B, which affect both sides of the actuator piston, they must be solved from the mathematical models describing positioner dynamics and actuator thermodynamics.

In equation (86) the valve load has been divided into the load caused by friction ($M_{v\mu}$) and the load caused by dynamic torque (M_d) since the dynamic torque usually tends to close the valve. The valve load consists of the seat friction, gland packing friction, bearing frictions and dynamic torque of the valve. A mathematical model is needed to solve these torques as a function of valve travel (h) and the pressure difference (Δp) across the valve throttling element. Because the pressure difference (Δp) over the throttling element changes considerably in a control situation especially with large travel changes, the mathematical model of the control valve must include a dynamic model of pipe flow.

Unsteady pipe flow control is calculated using the continuity equation (87) and the equation of motion, equation (88).

$$\frac{\partial h}{\partial t} + v \cdot \frac{\partial h}{\partial x} + \frac{a^2}{g} \cdot \frac{\partial v}{\partial x} = 0 \tag{87}$$

$$\frac{\partial v}{\partial t} + v \cdot \frac{\partial v}{\partial x} + g \cdot \frac{\partial h}{\partial x} + \xi \cdot \frac{v \cdot |v|}{2 \cdot d} = 0 \tag{88}$$

where
h = pressure head
g = acceleration of gravity
ρ = density
v = fluid velocity
a = speed of pressure pulse
ξ = pipe friction factor
d = pipe diameter

6.4 Control valve simulation program

The solution of mathematical models describing control valve action in space-time is not, however, mathematically possible as such because the equations are complicated and nonlinear. For this reason, a numerical solution produced by a computer program, has to be adopted. One advantage of the computer program is that the non-linearities in the control valve do not have to be linearized. Furthermore, the computer program is flexible because different changes and disturbance factors, depending on time and state, are easy to insert.

6.5 Friction model

A large decrease in the friction coefficient right after the valve starts to move is an important factor in the dynamics of a control valve equipped with a pneumatic cylinder actuator. This means that the correct modelling of friction behaviour is decisive for proper functioning of the simulation program.

Lubrication mechanisms can be divided into contact lubrication and fluid lubrication. In contact lubrication, the sliding bearing surfaces touch each other, although the lubricant limits the contact area by penetrating between the contact points. In fluid lubrication the pressure generated in the bearings prevents any contact between surfaces, and the friction force is created only by the shear stress of the lubricant.

The starting point for the contact friction model used is the exponential friction model, in which the friction force depends exponentially on relative slide velocity (v_r). Applying the model gives the equations (89) and (90) for the contact friction coefficients.

$$\mu_d 1 = \mu_{dmin} + (\mu_s - \mu_{dmin}) \cdot e^{-\lambda v_r^{\delta}} \tag{89}$$

$$\mu_d 2 = \mu_{dmin} + (\mu_{0s} - \mu_{dmin}) \cdot e^{-\lambda v_r^{\delta}} \tag{90}$$

where
μ_{d1} = contact friction coefficient for accelerating motion
μ_{d2} = contact friction coefficient for decelerating motion
μ_s = static friction coefficient for accelerating motion
μ_0 = static friction coefficient for decelerating motion
μ_{dmin} = minimum value for dynamic friction coefficient
δ = sliding materials constant
λ = stick-slip parameter
v_r = relative slide velocity

By approximating the contact friction using the exponential friction coefficient and by assuming that the fluid friction is directly proportional to velocity, the control valve friction model can be determined.

6.6 Testing and implementing the simulation program

Testing position control

The simulation program created was first tested with a single-stage pneumatic positioner, a double-acting cylinder actuator and a valve with a load factor (L_p). In this case, the actuator load factor (L_p) is the valve load divided by the actuator nominal output torque. Figures 76 and 77 show one form of step response calculated with the simulation program, and the corresponding valve relative travel (h) measured when the relative input signal (I) changes from value 0.4 to value 0.45. As becomes apparent from the step responses presented in figure 76, the form of the response calculated theoretically using the simulation program is fairly consistent with the corresponding response measured in a laboratory by means of an oscilloscope.

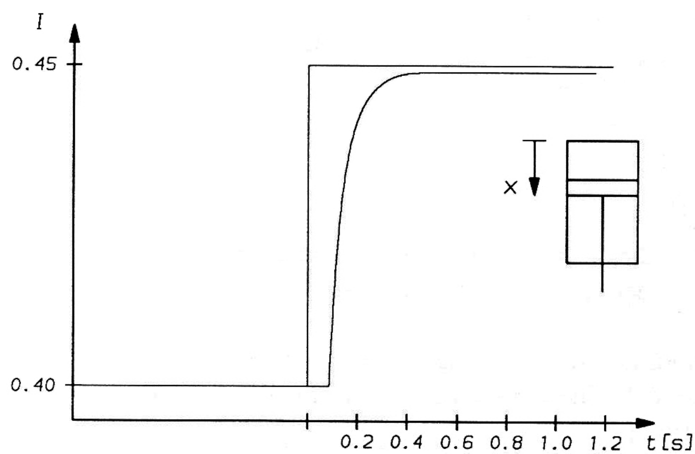


Figure 76. *Step response calculated theoretically with the simulation program when the value of the relative signal changes from 40 % to 45 %.*

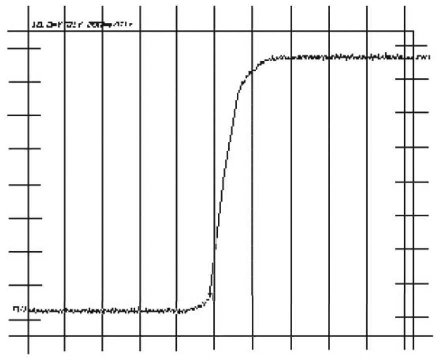


Figure 77. *The corresponding step response measured in the laboratory using an oscilloscope.*

Studies of an installed control valve

Figure 78 shows a preliminary study of a water-hammer with three different inherent flow characteristic curves. The calculations are based on a single-stage pneumatic positioner, a double-acting cylinder actuator and linear, equal percentage and constant gain inherent flow characteristics.

The characteristic pressure difference ratio (DP_f) describing the pipeline is determined using equation (91).

$$DP_f = \frac{\Delta P_f}{\Delta P_0} \tag{91}$$

where Δp_0 = pressure difference across a closed valve
Δp_f = pressure difference across a fully open valve

The behaviour is studied in a pipeline in which the characteristic pressure difference ratio $DP_f = 0.1$. The constant gain inherent flow characteristic can be determined using equation (92).

$$\phi(h) = \frac{C_v(h)}{C_{vf}} = \frac{h \cdot \sqrt{DP_f}}{\sqrt{1 + h^2 \cdot (DP_f - 1)}} \tag{92}$$

where C_{vf} = maximum capacity coefficient of valve

The starting point for the case is a constant gain inherent flow characteristic valve which performs the signal change from 100 % to 10 %. For the equal percentage opening valve with a corresponding steady-state pressure difference and flow rate change, the signal change is from 100 % to 12.0 %. For a linear valve the corresponding signal change is from 100 % to 3.2 %.

Figure 78 shows that the water-hammer caused by a valve with a linear inherent flow characteristic is very strong and sharp. With equal percentage and constant gain inherent flow characteristics, water-hammers are considerably smaller. Note that in the constant gain inherent flow characteristics the change in flow rate is almost linear as a function of time.

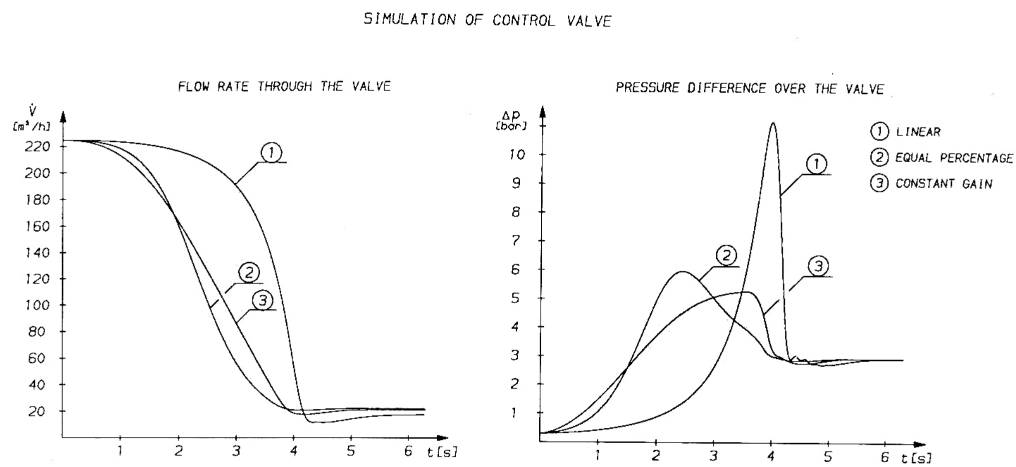


Figure 78. *Water-hammer caused by a valve with different inherent flow characteristics.*

SIZING EXAMPLES

The sizing examples presented here are based on sizing coefficients of Neles control valves. The sizing coefficients can be found from Valmet sizing coefficient tables.

Liquid flow sizing example

D series ball valve is controlling high pressure viscous oil flow. Valve size is 100 mm. What is required capacity C_v and opening of the valve with given flow data?

Flow data:

Water volumetric flow rate	$q = 100\ m^3/h$
Upstream temperature of fluid	$T_1 = 85\ °C$
Valve upstream pressure	$p_1 = 42$ barA
Valve downstream pressure	$p_2 = 41.2$ barA

Fluid properties:

Liquid specific gravity	$G_f = 0.9$
Liquid kinematic viscosity	$v = 10000$ cs

Piping configuration:

Upstream pipe diameter	$D_1 = 100$ mm
Downstream pipe diameter	$D_2 = 100$ mm
Pipe schedule	sch = 40

Valve sizing coefficients, when $C_v/d^2 = 20$:

Pressure recovery factor	$F_L = 0.76$

Calculation

Pressure recovery factor, including pipe fittings:

$F_{LP} = F_L = 0.76$(valve and pipe diameter are the same)

Valve style modifier:

$F_d = 1$ (for ball valves)

Estimated valve capacity:

$C_v = 320$

Valve Reynolds number:

$$Re_v = \frac{N_4 \cdot F_d \cdot q}{v \cdot \sqrt{F_p \cdot F_L \cdot C_v}} \cdot \left[\frac{(F_P \cdot F_L \cdot C_v)^2}{N_2 \cdot d^4} + 1\right]^{\frac{1}{4}}$$

$$= \frac{76000 \cdot 1 \cdot 100}{10000 \cdot \sqrt{1 \cdot 0{,}76 \cdot 320}} \cdot \left[\frac{(1 \cdot 0{,}76 \cdot 320)^2}{0{,}00214 \cdot 100^4} + 1\right]^{\frac{1}{4}}$$

$$= 51{,}8$$

Reynolds number factor:

$F_R = 0.35$ (from the graph in figure 38)

Valve required capacity:

$$C_v = \frac{q}{N_1 \cdot F_P \cdot F_R \cdot \sqrt{\frac{\Delta p}{G_f}}}$$

$$= \frac{100}{0{,}865 \cdot 1 \cdot 0{,}35 \cdot \sqrt{\frac{0{,}8}{0{,}9}}}$$

$$= 323$$

Valve relative opening h:

h = 75 %

Water flow sizing example

Top entry rotary control valve with Q-Trim is controlling cooling water flow. Valve size is 50 mm. What is required capacity C_v and opening of the valve with given flow data?

Flow data:

Water volumetric flow rate	$q = 40\ m^3/h$
Upstream temperature of fluid	$T_1 = 20\ °C$
Valve upstream pressure	$p_1 = 10$ barA
Valve downstream pressure	$p_2 = 1.5$ barA

Fluid properties:

Water specific gravity	$G_f = 1$
Water critical pressure	$p_c = 221.2$ barA
Water vapor pressure in T_1	$p_v = 0.03$ barA

Piping configuration:

Upstream pipe diameter	$D_1 = 50$ mm
Downstream pipe diameter	$D_2 = 50$ mm
Pipe schedule	sch = 40

Valve sizing coefficients, when $C_v/d^2 = 4.0$:

Pressure recovery factor	$F_L = 0.9$

Calculation

Pressure recovery factor, including pipe fittings:

$F_{LP} = F_L = 0.9$ (valve and pipe diameter are the same)

Water critical pressure ratio factor:

$$F_F = 0{,}96 - 0{,}28 \cdot \sqrt{\frac{P_v}{P_c}} = 0{,}96 - 0{,}28 \cdot \sqrt{\frac{0{,}023}{221{,}2}} = 0{,}957$$

Terminal pressure drop:

$$\Delta P_T = \left(\frac{F_{LP}}{F_P}\right)^2 \cdot (P_1 - F_F \cdot P_v)$$

$$= \left(\frac{0{,}9}{1}\right)^2 \cdot (10 - 0{,}957 \cdot 0{,}023)$$

$$= 8{,}08$$

Check if flow is choked:

$\Delta p > \Delta p_T \rightarrow$ the flow is choked $\rightarrow$ Δp_T must be used instead of Δp in sizing

Valve required capacity:

$$C_v = \frac{q}{N_1 \cdot F_P \cdot F_R \cdot \sqrt{\frac{\Delta p_T}{G_f}}}$$

$$= \frac{40}{0{,}865 \cdot 1 \cdot 1 \cdot \sqrt{\frac{8{,}08}{1}}}$$

$$= 16{,}3$$

Valve relative opening h:

h = 68 %

Gas flow sizing example

R-series segment valve with Q-Trim is controlling air flow to atmosphere. Valve size is 80 mm. What is required capacity C_v and opening of the valve with given flow data?

Flow data:

Gas volumetric flow rate (in NTP)	$q = 3700\ Nm^3/h$
Upstream temperature of gas	$T_1 = 20\ °C$
Valve upstream pressure	$p_1 = 4.7$ barA
Valve downstream pressure	$p_2 = 1.2$ barA

Fluid properties:

Molecular weight of gas	$M = 28.9$
Gas compressibility	$Z = 1$
Gas ratio of specific heats	$k = 1.4$

Piping configuration:

Upstream pipe diameter	$D_1 = 100$ mm
Downstream pipe diameter	$D_2 = 150$ mm
Pipe schedule	sch = 80

Valve sizing coefficients, when $C_v/d^2 = 6.5$:

Pressure drop ratio factor	$x_T = 0.69$

Calculation

Inlet Bernoulli coefficient K_{B1}:

$$K_B1 = 1 - \left(\frac{d}{D_1}\right)^4 = 1 - \left(\frac{80}{100}\right)^4 = 0,59$$

Outlet Bernoulli coefficient K_{B2}:

$$K_B2 = 1 - \left(\frac{d}{D_2}\right)^4 = 1 - \left(\frac{80}{150}\right)^4 = 0{,}92$$

Inlet reducer K_1:

$$K_1 = 0{,}5 \cdot \left[1 - \left(\frac{d}{D_1}\right)^2\right]^2 = 0{,}5 \cdot \left[1 - \left(\frac{80}{100}\right)^2\right]^2 = 0{,}06$$

Outlet reducer K_2:

$$K_2 = 1{,}0 \cdot \left[1 - \left(\frac{d}{D_2}\right)^2\right]^2 = 1{,}0 \cdot \left[1 - \left(\frac{80}{150}\right)^2\right]^2 = 0{,}51$$

Sum of K coefficients:

$$\begin{aligned} \Sigma K &= K_1 + K_2 + K_{B1} - K_{B2} \\ &= 0{,}06 + 0{,}51 + 0{,}59 - 0{,}92 \\ &= 0{,}24 \end{aligned}$$

Piping geometry factor F_p:

$$F_P = \frac{1}{\sqrt{1 + \frac{\Sigma K}{N_2} \cdot \left(\frac{C_v}{d^2}\right)^2}} = \frac{1}{\sqrt{1 + \frac{0{,}24}{890} \cdot 6{,}5^2}} = 0{,}99$$

Pressure drop ratio factor which includes pipe fittings x_{TP}:

$$x_{TP} = \frac{x_T}{F_P^2} \cdot \left[1 + \frac{x_T \cdot (K_1 + K_B1)}{N_5} \cdot \left(\frac{C_v}{d^2}\right)^2\right]^{-1}$$

$$= \frac{0{,}69}{0{,}99^2} \cdot \left[1 + \frac{0{,}69 \cdot (0{,}06 + 0{,}59)}{1000} \cdot 6{,}5^2\right]^{-1} = 0{,}69$$

Specific heat ratio factor F_k:

$$F_k = \frac{k}{1,4} = \frac{1,4}{1,4} = 1,0$$

Checking if flow is choked:

$$x = \frac{\Delta p}{p_1} = \frac{3,5}{4,7} = 0,745$$

$$F_k \cdot x_{TP} = 1,0 \cdot 0,69 = 0,69$$

Flow is choked because x > F_k * x_{TP}. Therefore F_k * x_{TP} (= 0.69) is used instead of x (= 0.745) in sizing.

Gas expansion factor Y: (for choked flow)

Y = 0.667

Valve required capacity C_v: (for choked flow)

$$C_v = \frac{q}{N_9 \cdot F_p \cdot p_1 \cdot Y \cdot \sqrt{\dfrac{F_k \cdot x_{TP}}{M \cdot T_1 \cdot Z}}}$$

$$= \frac{3700}{2250 \cdot 0,99 \cdot 2,7 \cdot 0,667 \cdot \sqrt{\dfrac{1,0 \cdot 0,69}{28,9 \cdot 293 \cdot 1}}} = 58,9$$

Valve relative opening h:

h = 67 %

Steam flow sizing example

L-series butterfly valve is controlling saturated steam flow. Valve size is 100 mm. What is required capacity C_V and opening of the valve with given flow data?

Flow data:

Gas mas flow rate	w = 7 t/h
Upstream temperature of steam	T_1 = 150 °C
Valve upstream pressure	p_1 = 4.75 barA
Valve downstream pressure	p_2 = 4.25 barA

Fluid properties:

Upstream density of steam	$\rho_1 = 2.54\ kg/m^3$
Ratio of specific heats	k = 1.3

Piping configuration:

Upstream pipe diameter	D_1 = 100 mm
Downstream pipe diameter	D_2 = 100 mm
Pipe schedule	sch = 40

Valve sizing coefficients, when $C_v/d^2 = 16$

Pressure drop ratio factor	x_T = 0.41

Calculation

Pressure drop ratio factor, including pipe fittings:

$x_{TP} = x_T = 0.41$ (valve and pipe diameter are the same)

Specific heat ratio factor:

$$F_k = \frac{k}{1,4} = \frac{1,3}{1,4} = 0,93$$

Check if flow is choked:

$$x = \frac{\Delta p}{p_1} = \frac{0,5}{4,75} = 0,105$$

$F_k \cdot x_{TP} = 0.93 \cdot 0.41 = 0.38$

Flow is not choked because $x < F_k \cdot x_{TP}$

Gas expansion factor:

$$Y = 1 - \frac{x}{3 \cdot F_k \cdot x_{TP}}$$

$$= 1 - \frac{0,105}{3 \cdot 0,93 \cdot 0,41}$$

$$= 0,91$$

Valve required capacity:

$$C_v = \frac{w}{N_6 \cdot F_p \cdot Y \cdot \sqrt{x \cdot p_1 \cdot \rho_1}}$$

$$= \frac{7000}{27,3 \cdot 1 \cdot 0,90 \cdot \sqrt{0,105 \cdot 4,75 \cdot 2,54}}$$

$$= 253$$

Valve relative opening h:

h = 70 %

Two phase flow of liquid and gas sizing example

Top entry rotary control valve with Q-Trim is controlling mixture of crude oil and natural gas. Valve size is 150 mm. What is required capacity C_v and opening of the valve with given flow data?

Flow data:

Liquid mass flow rate	w_f = 237 526 kg/h
Gas mass flow rate	w_g = 50 557 kg/h
Upstream temperature of fluid	T_1 = 104 °C
Valve upstream pressure	p_1 = 82.7 barA
Valve downstream pressure	p_2 = 34.7 barA

Fluid properties:

Liquid density	$\rho_f = 800\ kg/m^3$
Gas upstream density	$\rho_{g1} = 62\ kg/m^3$
Liquid critical pressure	p_c = 41.4 barA
Liquid vapour pressure in T_1	p_v = 0.37 barA
Gas ratio of specific heats	$k = 1.32$

Piping configuration:

Upstream pipe diameter	D_1 = 150 mm
Downstream pipe diameter	D_2 = 150 mm
Pipe schedule	sch = 80

Valve sizing coefficients, when $C_v/d^2 = 3.4$:

Pressure drop ratio factor	$x_T = 0.75$
Pressure recovery factor	$F_L = 0.91$

Calculation

Total mass flow rate:

$w = w_f + w_g$ = 237526 kg/h + 50557 kg/h = 288083 kg/h

Liquid portion of total mass flow rate:

$$f_f = \frac{w_f}{w} = \frac{237526}{288083} = 0{,}8245$$

Gas portion of total mass flow rate:

$$f_f = \frac{w_g}{w} = \frac{50557}{288083} = 0{,}1755$$

Pressure drop ratio factor, including pipe fittings:

$x_{TP} = x_T = 0.75$ (valve and pipe diameter are the same)

Specific heat ratio factor:

$$F_k = \frac{k}{1{,}4} = \frac{1{,}32}{1{,}4} = 0{,}943$$

Pressure recovery coefficient, including pipe fittings:

$F_{LP} = F_L = 0.91$ (valve and pipe diameter are the same)

Liquid critical pressure ratio factor:

$$F_F = 0{,}96 - 0{,}28 \cdot \sqrt{\frac{p_v}{p_c}} = 0{,}96 - 0{,}28 \cdot \sqrt{\frac{0{,}37}{41{,}4}} = 0{,}934$$

Terminal pressure drop of choked two-phase flow:

$$\Delta p_{T-2PH} = (p_1 \cdot F_k \cdot x_{TP}) \cdot f_g + \left(\frac{F_{LP}}{F_p}\right)^2 \cdot (p_1 - F_F \cdot p_v) \cdot f_f$$

$$= (82{,}7 \cdot 0{,}943 \cdot 0{,}75) \cdot 0{,}1755$$

$$+ \left(\frac{0{,}91}{1}\right)^2 \cdot (82{,}7 \cdot 0{,}934 - 0{,}37) \cdot 0{,}8245$$

$$= 66{,}5 \text{ bar}$$

Check if flow is choked:

$\Delta p < \Delta p_{T\text{-}2PH} \rightarrow$ flow is not choked

Gas expansion factor:

$$Y = 1 - \frac{x}{3 \cdot F_k \cdot x_{TP}}$$

$$= 1 - \frac{\left(\frac{82,7 - 34,7}{82,7}\right)}{3 \cdot 0,943 \cdot 0,75}$$

$$= 0,726$$

Effective density of mixture:

$$\rho_E = \left(\frac{f_g}{\rho_g 1 \cdot Y^2} + \frac{f_f}{\rho_f}\right)^{-1}$$

$$= \left(\frac{0,1755}{62 \cdot 0,726^2} + \frac{0,8245}{800}\right)$$

$$= 156,2 \text{ kg/m}^3$$

Valve required capacity:

$$C_v = \frac{w}{N_6 \cdot F_p \cdot \sqrt{\Delta p \cdot \rho_E}}$$

$$= \frac{288\ 083}{27,3 \cdot 1 \cdot \sqrt{48 \cdot 156,2}}$$

$$= 122$$

Valve relative opening h:

h = 70 %

Pulp flow sizing example

R-series segment valve is controlling flow of mechanical stock. Valve size is 80 mm. What is required capacity C_v and opening of the valve with given flow data?

Flow data:

Pulp volumetric flow rate	$q = 50\ m^3/h$
Upstream temperature of pulp	$T_1 = 40\ °C$
Valve upstream pressure	$p_1 = 2.1$ barA
Valve downstream pressure	$p_2 = 1.5$ barA

Fluid properties:

Pulp consistency	C = 8 %
Liquid critical pressure	$p_c = 221.2$ barA
Liquid vapour pressure	$p_v = 0.1$ barA

Piping configuration:

Upstream pipe diameter	$D_1 = 80$ mm
Downstream pipe diameter	$D_2 = 80$ mm
Pipe schedule	sch = 40

Calculation

Equivalent flow rate in water:

k = 0.8 (from graph in appendix J)

$$q_{water} = \frac{q_{stock}}{k} = \frac{50}{0{,}8} = 62{,}5\ m^3/h$$

Piping geometry factor:

$F_p = 1.0$ (valve and pipe diameter are the same)

Valve required capacity:

$$C_v = \frac{q_{water}}{N_1 \cdot F_P \cdot F_R \cdot \sqrt{\frac{\Delta p}{G_f}}}$$

$$= \frac{62{,}5}{0{,}865 \cdot 1 \cdot 1 \cdot \sqrt{\frac{0{,}6}{1}}}$$

$$= 93{,}3$$

Valve relative opening h:

h = 65 %

CONVERSION TABLES

LENGTH UNITS

		mm	cm	m	km	in	ft	yd	ml
millimeters	mm	1	0.1	0.001	0.000001	0.03937	0.003281	0.001094	6.21e07
centimeters	cm	10	1	0.01	0.00001	0.393701	0.032808	0.010936	0.000006
meters	m	1000	100	1	0.001	39.37008	3.28084	1.093613	0.000621
kilometers	km	1000000	100000	1000	1	39370.08	3280.84	1093.613	0.621371
inches	in	25.4	2.54	0.0254	0.000025	1	0.083333	0.027778	0.000016
feet	ft	304.8	30.48	0.3048	0.000305	12	1	0.333333	0.000189
yards	yd	914.4	91.44	0.9144	0.000914	36	3	1	0.000568
miles	ml	1609344	160934.4	1609.344	1.609344	63360	5280	1760	1

AREA UNITS

		mm^2	cm^2	m^2	in^2	ft^2	yd^2
square millimeters	mm^2	1	0.01	0.000001	0.00155	0.000011	0.000001
square centimeters	cm^2	100	1	0.0001	0.155	0.001076	0.00012
square meters	m^2	1000000	10000	1	1550.003	10.76391	1.19599
square inches	in^2	645.16	6.4516	0.000645	1	0.006944	0.000772
square feet	ft^2	92903	929.0304	0.092903	144	1	0.111111
square yards	yd^2	836127	8361.274	0.836127	1296	9	1

VOLUME UNITS

		cm^3	m^3	ltr	in^3	ft^3	US gal	Imp.gal	US brl
cubic centimeters	cm^3	1	0.000001	0.001	0.061024	0.000035	0.000264	0.00022	0.000006
cubic meters	m^3	1000000	1	1000	61024	35	264	220	6.29
liters	ltr	1000	0.001	1	61.0	0.035	0.264201	0.22	0.00629
cubic inches	in^3	16.4	0.000016	0.016387	1	0.000579	0.004329	0.003605	0.000103
cubic feet	ft^3	28317	0.028317	28.31685	1728.0	1	7.481333	6.229712	0.178127
US gallon	US gal	3785	0.003785	3.79	231	0.13	1	0.832701	0.02381
Imp gallon	Imp.gal	4545	0.004545	4.55	277	0.16	1.20	1	0.028593
US barrel for oil	US brl	158970	0.15897	159	9701	6	42	35.0	1

MASS UNITS

		g	kg	tonne	shton	Lton	lb	oz
grams	g	1	0.001	0.000001	0.000001	9.84e07	0.002205	0.035273
kilograms	kg	1000	1	0.001	0.001102	0.000984	2.204586	35.27337
metric tons	tonne	1000000	1000	1	1.102293	0.984252	2204.586	35273.37
short tons	shton	907200	907.2	0.9072	1	0.892913	2000	32000
long tons	Lton	1016000	1016	1.016	1.119929	1	2239.859	35837.74
pounds	lb	453.6	0.4536	0.000454	0.0005	0.000446	1	16
ounces	oz	28	0.02835	0.000028	0.000031	0.000028	0.0625	1

DENSITY UNITS

		g/ml	kg/m^3	lb/ft^3	lb/in^3
gram per milliliter	g/ml	1	1000	62.42197	0.036127
kilogram per cubic meter	kg/m^3	0.001	1	0.062422	0.000036
pound per cubic foot	lb/ft^3	0.01602	16.02	1	0.000579
pound per cubic inch	lb/in^3	27.68	27680	1727.84	1

Note: SPECIFIC VOLUME = 1 / DENSITY

SPECIFIC GRAVITY G_f for liquids:

G_f = ρ/ρ_0
= (density of liquid at upstream temperature) / (density of water at reference conditions)

where reference conditions according to IEC 60534 and ISA S75:
for SI units: ρ0 = 999 kg/m^3 at 15.6 °C and
at atmospheric pressure (1.013 barA)
for US units: ρ0 = 62.36 lb/ft3 at 60 °F and
at atmospheric pressure (14.73 psiA)

SPECIFIC GRAVITY G_g for gases:

G_g = ρ_{gas}/ρ_0
= (density of gas at standard conditions)/ (density of air at standard conditions)

where standard conditions according to IEC 60534 and ISA S75:
for SI units: 15.6 °C and atmospheric pressure (1.013 barA)
for US units: 60 °F and atmospheric pressure (14.73 psiA)

When gas specific gravity is calculated in standard conditions, the above equation can be simplified to: $G_g = M_w/28.96$, where M_w is molecular weight of gas.

SPEED UNITS

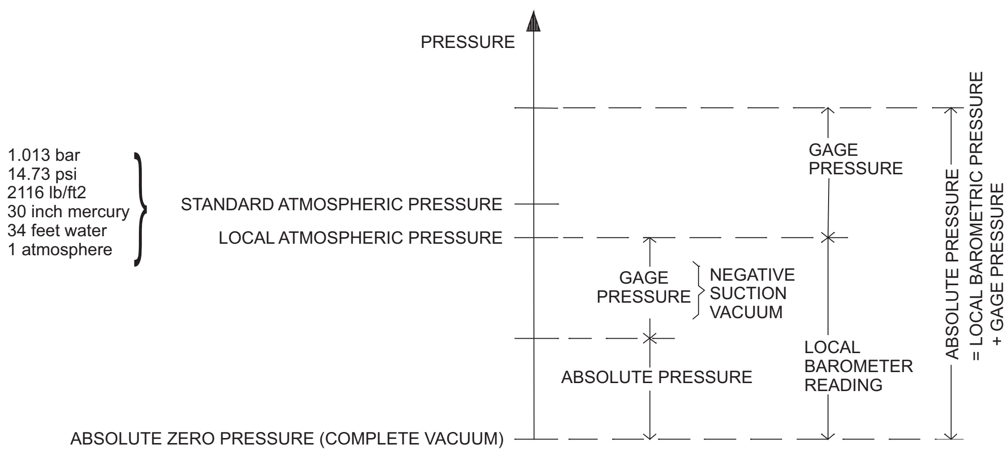
		m/s	m/min	km/h	ft/s	ft/min	mi/h
meter per second	m/s	1	59.988	3.599712	3.28084	196.8504	2.237136
meter per minute	m/min	0.01667	1	0.060007	0.054692	3.281496	0.037293
kilometer per hour	km/h	0.2778	16.66467	1	0.911417	54.68504	0.621477
feet per second	ft/s	0.3048	18.28434	1.097192	1	60	0.681879
feet per minute	ft/min	0.00508	0.304739	0.018287	0.016667	1	0.011365
miles per hour	mi/h	0.447	26.81464	1.609071	1.466535	87.99213	1

PRESSURE UNITS

		bar	psi	kPa	MPa	kgf/cm^2	mmHg	atm
bar	bar	1	14.50326	100	0.1	1.01968	750.0188	0.987167
pound per square inch	psi	0.06895	1	6.895	0.006895	0.070307	51.71379	0.068065
kilo Pascal	kPa	0.01	0.1450	1	0.001	0.01020	7.5002	0.00987
Mega Pascal	MPa	10	145.03	1000	1	10.197	7500.2	9.8717
kilogram force per cm^2	kgf/cm^2	0.9807	14.22335	98.07	0.09807	1	735.5434	0.968115
millimeter of mercury	mmHg,torr	0.001333	0.019337	0.13333	0.000133	0.00136	1	0.001316
Int.St.Atm.	atm	1.013	14.69181	101.3	0.1013	1.032936	759.769	1
meter of water	mH_2O	0.09806	1.42219	9.806	0.00981	0.09999	73.54684	0.096802
foot of water	ftH_2O	0.02989	0.433503	2.989	0.00299	0.030478	22.41806	0.029506
centimeter of mercury	cmHg	0.01333	0.193328	1.333	0.00133	0.013592	9.99775	0.013159
inch of mercury	inHg	0.03386	0.49108	3.386	0.00339	0.034526	25.39563	0.033425
inch of water	inH_2O	0.002491	0.036128	0.2491	0.000249	0.00254	1.868297	0.002459
Pascal = N/m^2	Pa	0.00001	0.000145	0.001	0.000001	0.00001	0.0075	0.00001
mmH_2O, kp/m^2	kp/m^2	9.81e05	0.001422	0.009807	0.00001	0.00099	0.073552	0.000097

		mH_2O	ftH_2O	cmHg	inHg	inH_2O	Pa	kp/m^2
bar	bar	10.19784	33.45601	75.01875	29.53337	401.4452	100000	10197.16
pound per square inch	psi	0.703141	2.306792	5.172543	2.036326	27.67965	6895	703.0943
kilo Pascal	kPa	0.10198	0.33456	0.75019	0.29533	4.01445	1000	101.972
Mega Pascal	MPa	101.978	334.560	750.187	295.334	4014.45	1000000	101972
kilogram force per cm^2	kgf/cm^2	10.00102	32.8103	73.57089	28.96338	393.6973	98070	10000.36
millimeter of mercury	mmHg,torr	0.013597	0.044607	0.100023	0.039377	0.535247	133.33	13.59588
Int.St.Atm.	atm	10.33041	33.89093	75.994	29.91731	406.664	101300	10329.73
meter of water	mH_2O	1	3.280696	7.356339	2.896043	39.36572	9806	999.9337
foot of water	ftH_2O	0.304813	1	2.242311	0.882753	11.9992	2989	304.7932
centimeter of mercury	cmHg	0.135937	0.445969	1	0.39368	5.351265	1333	135.9282
inch of mercury	inHg	0.345299	1.13282	2.540135	1	13.59293	3386	345.2759
inch of water	inH_2O	0.025403	0.083339	0.186872	0.073568	1	249.1	25.40113
Pascal = N/m^2	Pa	0.000102	0.000335	0.00075	0.000295	0.004014	1	0.101972
mmH_2O, kp/m^2	kp/m^2	0.001	0.003281	0.007357	0.002896	0.039368	9.80665	1

Note: Pressure may be expressed with reference to any arbitraty datum. The usual ones are absolute zero and local atmospheric pressure. When a pressure is expressed as a difference between its value and a complete vacuum, it is called absolute pressure. When it is expressed as a difference between its value and the local atmospheric pressure, it is called a gage pressure. The following figure shows the relationships between absolute pressure, gage pressure, and barometric pressure.



COMMONLY USED UNITS FOR VOLUMETRIC LIQUID FLOW

		l/sec	l/min	m^3/hr	ft^3/min	ft^3/hr	gal/min	US brl/d
liter per second	l/sec	1	60	3.6	2.119093	127.1197	15.85037	543.4783
liter per minute	l/min	0.016666	1	0.06	0.035317	2.118577	0.264162	9.057609
cubic meters per hour	m^3/hr	0.277778	16.6667	1	0.588637	35.31102	4.40288	150.9661
cubic feet per minute	ft^3/min	0.4719	28.31513	1.69884	1	60	7.479791	256.4674
cubic feet hour	ft^3/hr	0.007867	0.472015	0.02832	0.01667	1	0.124689	4.275326
US gallon per minute	US gal/min	0.06309	3.785551	0.227124	0.133694	8.019983	1	34.28804
US barrel per day	US brl/d	0.00184	0.110404	0.006624	0.003899	0.2339	0.029165	1

COMMONLY USED UNITS FOR VOLUMETRIC GAS FLOW

		Nm^3/hr	scfh	scfm	mscfd	mmscfd	Nl/sec
Normal cub. meter per h	Nm^3/hr	1	35.31073	0.588582	0.847458	0.000847	0.277778
Stand. cub. feet per h	scfh	0.02832	1	0.016669	0.024	0.000024	0.007867
Stand. cub. feet per min.	scfm	1.699	59.99294	1	1.439831	0.00144	0.471944
10^3 St. cub. ft per day	mscfd	1.18	41.66667	0.694526	1	0.001	0.327778
10^6 St. cub. ft per day	mmscfd	1180	41666.67	694.5262	1000	1	327.7778
Normal liter per second	Nl/sec	3.6	127.1186	2.118893	3.050847	0.003051	1

Note: Standard or normal conditions refer to:
for SI units: 15.6 °C and atmospheric pressure (1.013 barA)
for US units: 60 °F and atmospheric pressure (14.73 psiA)

Typically gas flow rates are presented in normal (NTP) or standard (STD) conditions

When converting volumetric flow rate to mass flow rate, then:
MASS FLOW RATE = (VOLUME FLOW RATE)*(DENSITY)

$$\frac{kg}{hr} = \frac{Nm^3}{hr} \cdot \frac{1{,}013 \cdot 10^5 \cdot Mw}{8314 \cdot 288{,}75}$$

Note: When converting actual flow to flow in normal conditions ([Nm^3/h] or [scfh]), then:

$$q[Nm^3/hr] = q_{act}\,[m^3/hr] \cdot \frac{p_1 \cdot T_N}{p_N \cdot T_1}$$

where: p_1 = upstream pressure (SI units: [barA]; US units: [psiA])
T1 = upstream temperature (SI units: [K]; US units: [°R])
pN = standard pressure (1.013 barA or 14.73 psiA)
TN = standard temperature (288.75 K or 519.75 °R)

MASS FLOW UNITS

		kg/h	lb/hour	kg/s	t/h
kilogram per hour	kg/h	1	2.204586	0.000278	0.001
pound per hour	lb/hour	0.4536	1	0.000126	0.000454
kilogram per second	kg/s	3600	7936.508	1	3.6
ton per hour	t/h	1000	2204.586	0.277778	1

TEMPERATURE UNITS

K = Kelvin
°C = degrees in Celsius
°F = degrees in Fahrenheit
°R = degrees in Rankine

$$K = °C + 273.15 = (°F + 99.67) / 1.8 + 200 = °R / 1.8$$

$$°C = K - 273.15 = (°F - 32) / 1.8 = °R / 1.8 - 273.15$$

$$°F = (K - 200) * 1.8 - 99.67 = °C * 1.8 + 32 = °R - 459.67$$

$$°R = K * 1.8 = (°C + 273.15) * 1.8 = °F + 459.67$$

TORQUE UNITS

		Nm	kgfm	ftlb	inlb
Newton Meter	Nm	1	0.101972	0.737561	8.850732
kilogram force meter	kgfm	9.80665	1	7.233003	86.79603
foot pound	ftlb	1.35582	0.138255	1	12
inch pound	inlb	0.112985	0.011521	0.083333	1

DYNAMIC VISCOSITY UNITS

		cp	poise	lb/(ft*s)	lbf*s/ft^2	Ns/m^2
centipoise, mPa*s	cp	1	0.01	0.000672	0.000021	0.001
g/(cm*s) = dyne*s/cm^2	poise	100	1	0.067197	0.002089	0.1
pound per foot second	lb/(ft*s)	1488.16	14.8816	1	0.031081	1.48816
pound force sec. per ft^2	lbf*s/ft^2	47880	478.8	32.17396	1	47.88
Ns/m^2 = kg/(m*s) = Pl	Ns/m^2	1000	10	0.671971	0.020886	1

KINEMATIC VISCOSITY UNITS

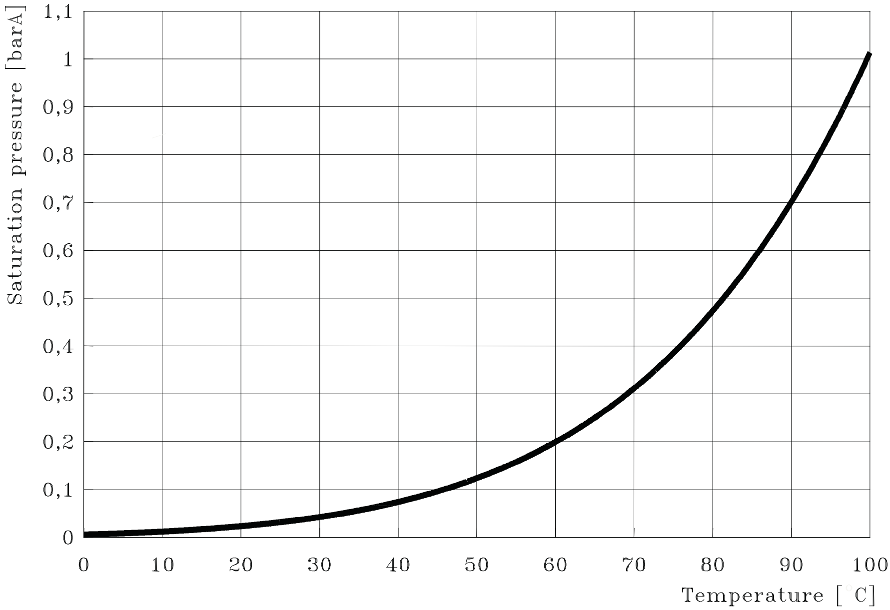
		cs,cSt	St,stoke	ft^2/s	m^2/s
centistoke	cs,cSt	1	0.01	0.000011	0.000001
stoke = cm^2/s	St,stoke	100	1	0.001076	0.0001
square feet per second	ft^2/s	92903	929.03	1	0.092903
square meter per second	m^2/s	1000000	10000	10.76392	1

Note: Conversion from dynamic viscosity to kinematic viscosity in SI units is:

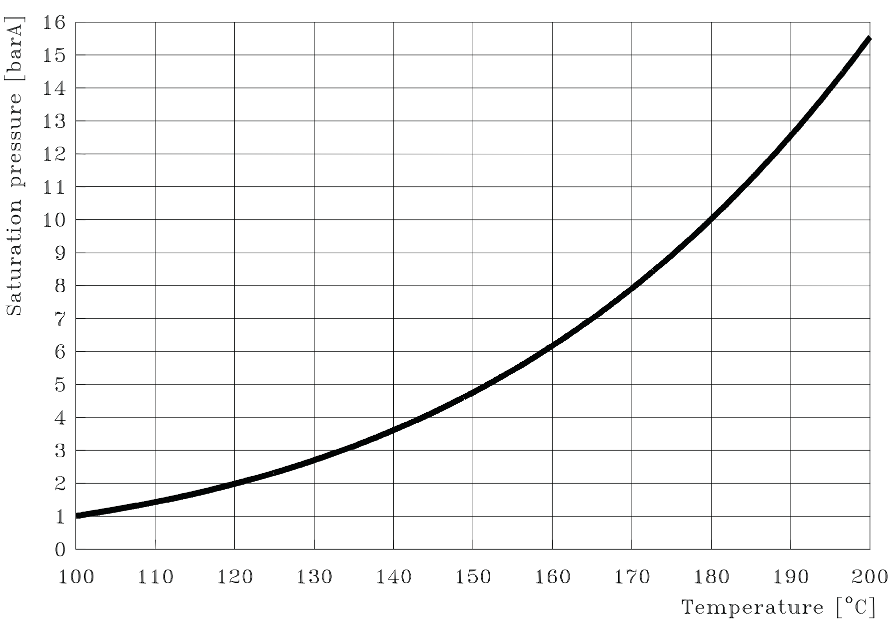
$$\text{kinematic viscosity } \eta \text{ [m}^2\text{/s]} = \frac{\text{dynamic viscosity } \mu \text{ [Ns/m}^2\text{]}}{\text{density } \rho \text{ [kg/m}^3\text{]}}$$

$$\text{kinematic viscosity } \nu \text{ [cs]} = \frac{\text{dynamic viscosity } \mu \text{ [cp]}}{\text{specific gracity } G_f}$$

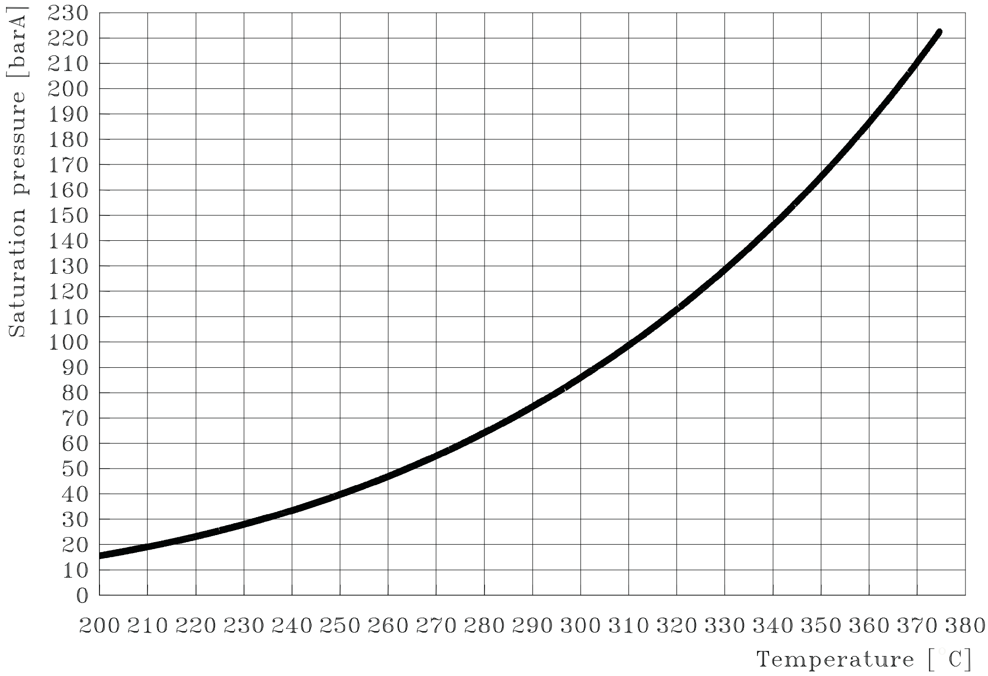
STEAM



Saturation pressure of steam between 0 °C and 100 °C



Saturation pressure of steam between 100 °C and 200 °C



Saturation pressure of steam between 200 °C and 375 °C

Density of saturated steam.

Saturation temperature		Saturation pressure	Specific volume	Density
°C	°K	[barA]	[m³/kg]	[kg/m³]
0	273.15	0.00611	206.140	0.00485
5	278.15	0.00872	147.048	0.00680
10	283.15	0.01227	106.350	0.00940
15	288.15	0.01704	77.9219	0.01283
20	293.15	0.02337	57.7989	0.01730
25	298.15	0.03166	43.3740	0.02306
30	303.15	0.04241	32.9094	0.03039
35	308.15	0.05622	25.2312	0.03963
40	313.15	0.07375	19.5366	0.05119
45	318.15	0.09582	15.2697	0.06549
50	323.15	0.12335	12.0413	0.08305
55	328.15	0.15741	9.57601	0.10443
60	333.15	0.19920	7.67676	0.13026
65	338.15	0.25009	6.20127	0.16126
70	343.15	0.31162	5.04579	0.19819
75	348.15	0.38549	4.13399	0.24190
80	353.15	0.47360	3.40923	0.29332
85	358.15	0.57803	2.82913	0.35347
90	363.15	0.70109	2.36171	0.42342
95	368.15	0.84526	1.98270	0.50436
100	373.15	1.01325	1.67351	0.59755
105	378.15	1.20800	1.41980	0.70432
110	383.15	1.43266	1.21046	0.82613
115	388.15	1.69060	1.03681	0.96450
120	393.15	1.98543	0.89203	1.12104
125	398.15	2.32098	0.77072	1.29748
130	403.15	2.70132	0.66861	1.49564
135	408.15	3.13075	0.58226	1.71743
140	413.15	3.61379	0.50894	1.96488
145	418.15	4.15520	0.44640	2.24013
150	423.15	4.75997	0.39286	2.54541
155	428.15	5.43330	0.34685	2.88311
160	433.15	6.18065	0.30715	3.25573
165	438.15	7.00766	0.27278	3.66590

Saturation temperature		Saturation pressure	Specific weight	Density
°C	°K	[barA]	[m³/kg]	[kg/m³]
170	443.15	7.92023	0.24293	4.11642
175	448.15	8.92444	0.21691	4.61024
180	453.15	10.0266	0.19416	5.15048
185	458.15	11.2333	0.17420	5.74044
190	463.15	12.5512	0.15665	6.38365
195	468.15	13.9873	0.14117	7.08386
200	473.15	15.5488	0.12747	7.84505
205	478.15	17.2430	0.11532	8.67152
210	483.15	19.0774	0.10452	9.56782
215	488.15	21.0598	0.09489	10.5389
220	493.15	23.1983	0.08628	11.5900
225	498.15	25.5009	0.07857	12.7269
230	503.15	27.9760	0.07166	13.9557
235	508.15	30.6323	0.06543	15.2832
240	513.15	33.4783	0.05982	16.7167
245	518.15	36.5232	0.05475	18.2641
250	523.15	39.7760	0.05017	19.9340
255	528.15	43.2462	0.04601	21.7361
260	533.15	46.9434	0.04223	23.6808
265	538.15	50.8773	0.03879	25.7797
270	543.15	55.0581	0.03566	28.0454
275	548.15	59.4960	0.03280	30.4923
280	553.15	64.2018	0.03018	33.1362
285	558.15	69.1863	0.02778	35.9946
290	563.15	74.4607	0.02558	39.0874
295	568.15	80.0369	0.02356	42.4372
300	573.15	85.9269	0.02171	46.0694
305	578.15	92.1435	0.01999	50.0130
310	583.15	98.7001	0.01842	54.3017
315	588.15	105.611	0.01696	58.9740
320	593.15	112.891	0.01561	64.0754
325	598.15	120.556	0.01436	69.6593
330	603.15	128.625	0.01319	75.7892
335	608.15	137.117	0.01212	82.5418

SUPERHEATED STEAM

Specific volume (V) (unit: [m^3/kg]) as a function of temperature (unit: [°C]) and pressure (unit: [barA]).

Pres-sure [barA]	Temperature [°C]										
	100	120	140	160	180	200	220	240	260	280	300
1	1.70	1.79	1.89	1.98	2.08	2.17	2.26	2.36	2.45	2.54	2.64
1.2	---	1.49	1.57	1.65	1.73	1.81	1.88	1.96	2.04	2.12	2.20
1.4	---	1.27	1.34	1.41	1.48	1.55	1.61	1.68	1.75	1.81	1.88
1.6	---	1.11	1.17	1.23	1.29	1.35	1.41	1.47	1.53	1.59	1.65
1.8	---	0.986	1.04	1.09	1.15	1.20	1.25	1.31	1.36	1.41	1.46
2	---	---	0.935	0.983	1.03	1.08	1.13	1.17	1.22	1.27	1.31
3	---	---	0.617	0.650	0.683	0.715	0.748	0.780	0.811	0.843	0.874
4	---	---	---	0.484	0.509	0.534	0.558	0.582	0.606	0.630	0.654
5	---	---	---	0.384	0.404	0.425	0.444	0.464	0.484	0.503	0.522
6	---	---	---	0.317	0.335	0.352	0.369	0.385	0.402	0.418	0.434
7	---	---	---	---	0.285	0.300	0.314	0.329	0.343	0.357	0.371
8	---	---	---	---	0.247	0.261	0.274	0.287	0.299	0.312	0.324
9	---	---	---	---	0.218	0.230	0.242	0.254	0.265	0.276	0.287
10	---	---	---	---	0.195	0.206	0.217	0.227	0.238	0.248	0.258
12	---	---	---	---	---	0.169	0.179	0.188	0.197	0.205	0.214
14	---	---	---	---	---	0.143	0.152	0.160	0.167	0.175	0.182
16	---	---	---	---	---	---	0.131	0.138	0.145	0.152	0.159
18	---	---	---	---	---	---	0.115	0.122	0.128	0.134	0.140
20	---	---	---	---	---	---	0.102	0.109	0.114	0.120	0.126
22	---	---	---	---	---	---	0.092	0.098	0.103	0.108	0.113
24	---	---	---	---	---	---	---	0.089	0.094	0.099	0.103
26	---	---	---	---	---	---	---	0.081	0.086	0.090	0.095
28	---	---	---	---	---	---	---	0.074	0.079	0.083	0.088
30	---	---	---	---	---	---	---	0.068	0.073	0.077	0.081
32	---	---	---	---	---	---	---	0.063	0.068	0.072	0.076
34	---	---	---	---	---	---	---	---	0.063	0.067	0.071
36	---	---	---	---	---	---	---	---	0.059	0.063	0.066
38	---	---	---	---	---	---	---	---	0.055	0.059	0.063
40	---	---	---	---	---	---	---	---	0.052	0.056	0.059
42	---	---	---	---	---	---	---	---	0.049	0.053	0.056
44	---	---	---	---	---	---	---	---	0.046	0.050	0.053
46	---	---	---	---	---	---	---	---	0.043	0.047	0.050
48	---	---	---	---	---	---	---	---	---	0.045	0.048
50	---	---	---	---	---	---	---	---	---	0.042	0.046
55	---	---	---	---	---	---	---	---	---	0.037	0.041
60	---	---	---	---	---	---	---	---	---	0.033	0.036
65	---	---	---	---	---	---	---	---	---	---	0.033
70	---	---	---	---	---	---	---	---	---	---	0.030
75	---	---	---	---	---	---	---	---	---	---	0.027
80	---	---	---	---	---	---	---	---	---	---	0.024

Pressure [barA]	Temperature [°C]									
	320	340	360	380	400	420	440	460	480	500
1	2.73	2.82	2.91	3.01	3.10	3.19	3.29	3.38	3.47	3.56
1.2	2.27	2.35	2.43	2.51	2.58	2.66	2.74	2.81	2.89	2.97
1.4	1.95	2.01	2.08	2.15	2.21	2.28	2.35	2.41	2.48	2.54
1.6	1.70	1.76	1.82	1.88	1.94	1.99	2.05	2.11	2.17	2.23
1.8	1.51	1.57	1.62	1.67	1.72	1.77	1.82	1.88	1.93	1.98
2	1.36	1.41	1.45	1.50	1.55	1.59	1.64	1.69	1.73	1.78
3	0.906	0.937	0.968	1.000	1.031	1.062	1.093	1.124	1.155	1.186
4	0.678	0.702	0.725	0.749	0.772	0.795	0.819	0.842	0.865	0.889
5	0.541	0.560	0.579	0.598	0.617	0.636	0.654	0.673	0.692	0.710
6	0.450	0.466	0.482	0.498	0.513	0.529	0.545	0.560	0.576	0.592
7	0.385	0.399	0.412	0.426	0.439	0.453	0.466	0.480	0.493	0.507
8	0.336	0.348	0.360	0.372	0.384	0.396	0.408	0.420	0.431	0.443
9	0.298	0.309	0.320	0.330	0.341	0.352	0.362	0.373	0.383	0.394
10	0.268	0.277	0.287	0.297	0.306	0.316	0.326	0.335	0.345	0.354
12	0.222	0.230	0.239	0.247	0.255	0.263	0.271	0.279	0.287	0.295
14	0.189	0.197	0.204	0.211	0.218	0.225	0.232	0.238	0.245	0.252
16	0.165	0.171	0.178	0.184	0.190	0.196	0.202	0.208	0.214	0.220
18	0.146	0.152	0.157	0.163	0.168	0.174	0.179	0.185	0.190	0.195
20	0.131	0.136	0.141	0.146	0.151	0.156	0.161	0.166	0.171	0.176
22	0.118	0.123	0.128	0.133	0.137	0.142	0.146	0.151	0.155	0.159
24	0.108	0.112	0.117	0.121	0.125	0.130	0.134	0.138	0.142	0.146
26	0.099	0.103	0.107	0.111	0.115	0.119	0.123	0.127	0.131	0.135
28	0.092	0.096	0.099	0.103	0.107	0.111	0.114	0.118	0.121	0.125
30	0.085	0.089	0.092	0.096	0.099	0.103	0.106	0.110	0.113	0.116
32	0.079	0.083	0.086	0.090	0.093	0.096	0.099	0.103	0.106	0.109
34	0.074	0.078	0.081	0.084	0.087	0.090	0.093	0.096	0.099	0.102
36	0.070	0.073	0.076	0.079	0.082	0.085	0.088	0.091	0.094	0.096
38	0.066	0.069	0.072	0.075	0.078	0.080	0.083	0.086	0.089	0.091
40	0.062	0.065	0.068	0.071	0.074	0.076	0.079	0.081	0.084	0.087
42	0.059	0.062	0.065	0.067	0.070	0.072	0.075	0.077	0.080	0.082
44	0.056	0.059	0.061	0.064	0.067	0.069	0.071	0.074	0.076	0.078
46	0.053	0.056	0.059	0.061	0.063	0.066	0.068	0.070	0.073	0.075
48	0.051	0.053	0.056	0.058	0.061	0.063	0.065	0.067	0.070	0.072
50	0.048	0.051	0.053	0.056	0.058	0.060	0.062	0.065	0.067	0.069
55	0.043	0.046	0.048	0.050	0.052	0.054	0.056	0.058	0.060	0.062
60	0.039	0.041	0.044	0.046	0.048	0.050	0.051	0.053	0.055	0.057
65	0.035	0.038	0.040	0.042	0.044	0.045	0.047	0.049	0.051	0.052
70	0.032	0.034	0.037	0.038	0.040	0.042	0.044	0.045	0.047	0.048
75	0.029	0.032	0.034	0.036	0.037	0.039	0.040	0.042	0.043	0.045
80	0.027	0.029	0.031	0.033	0.035	0.036	0.038	0.039	0.041	0.042

Note: DENSITY = (SPECIFIC VOLUME)$^{-1}$

PHYSICAL CONSTANTS

Gas phase

Gas with the molecular formula		Molecular weight M g/mol	Weight of std. cubic metre ρ_n [kg/m^3]	Specific gravity G_g	Critical Pressure p_c [bar]	Critical temp. t_c [°C]	Ratio of specific heats k *)	
Acetylene	C_2H_2	26.04	1.11	0.91	62.4	35.7	1.23	
Air		28.98	1.23	1.00	37.7	-140.7	1.40	
Ammonia	NH_3	17.08	0.73	0.60	113.0	132.4	1.32	
Argon	Ar	39.94	1.69	1.38	48.6	-122.4	1.67	
Bentzene	C_6H_6	78.11	3.30	2.70	48.5	288.5	1.10	(90°C)
Butane (n)	C_4H_{10}	58.12	2.59	2.09	36.5	153.2		
Butane (i)	C_4H_{10}	58.12	2.51	2.05	36.5	135.1	1.11	(15°C)
Butylene	C_4H_8	56.11	2.37	1.94	39.3	146.6		
Carbon dioxide	CO_2	44.01	1.87	1.53	74.0	31.0	1.31	
Carbon monoxide	CO	28.01	1.18	0.97	35.0	-140.2	1.40	
Chlorine	Cl_2	70.91	3.05	2.49	77.0	144	1.34	
Chlorine dioxide	ClO_2	67.46	2.94	2.40				
Cyanogen	C_2N_2	52.04	2.23	1.78	60.8	128.3	1.26	
Dichordi fluor	CF_2Cl_2	120.92	4.82	3.93	40.1	111.5	1.14	
Ethane	C_2H_6	30.07	1.29	1.05	48.8	32.2	1.22	
Ethylene	C_2H_4	28.05	1.19	0.98	51.2	9.7	1.25	
Fluorine	F_2	38.00	1.61	1.31	55.7	-128.7		
Formaldehyde	CH_2O	30.03	1.32					
Helium	He	4.00	0.17	0.14	2.30	-267.9	1.66	
Hydrogen	H_2	2.02	0.08	0.07	13.0	-239.9	1.41	
Hydrogen bromide	HBr	80.92	3.45	2.82	85.6	90.00	1.36	
Hydrogen chloride	HCl	36.47	1.55	1.27	84.1	51.40	1.41	
Hydrogen iodide	HJ	127.93	5.49	4.48	83.1	150.80	1.40	
Hydrogen sulfide	H_2S	34.08	1.46	1.19	90.1	100.50	1.33	
Methane, Natural gas	CH_4	16.04	0.68	0.55	46.3	82.50	1.30	
Methyl chloride	CH_3Cl	50.49	2.19	1.78	66.8	143.10	1.28	
Methyl ether	C_2H_6O	46.07	1.98	1.62	53.3	126.90	1.11	(20°C)
Methyl mercaptan	CH_4S	48.10	1.06	0.87	72.3	196.80		
Neon	Ne	20.18	0.85	0.70	27.2	-228.7	1.67	
Nitric oxide	NO	30.01	1.27	1.04	65.9	94.00	1.40	
Nitrogen	N_2	28.02	1.18	0.97	33.9	-147.10	1.40	
Nitrous oxide	N_2O	44.02	1.88	1.53	72.7	36.60	1.28	
Oxygen	O_2	32.00	1.35	1.11	50.4	-118.8	1.40	
Ozone	O_3	48.00	2.03	1.66	93.5	5.00	1.29	
Propane (n)	C_3H_8	44.09	1.90	1.56	42.6	96.80	1.14	
Propylene	C_3H_6	42.08	1.81	1.48	45.9	92.00	1.14	
Sulphur dioxide	SO_2	64.06	2.77	2.26	78.8	157.30	1.29	
Steam	H_2O	18.02	(0.76)	(0.622)	220.4	374	1.3	(std. cond.)

*) k is in temperature 0 °C unless otherwise specified

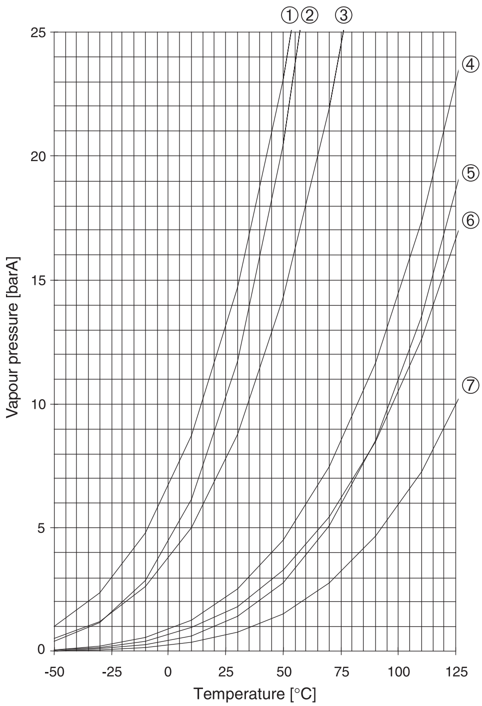
Liquid phase

Liquid with the molecular formula		Molecular weight M g/mol	Specific Gravity G_f	Critical Pressure p_c [barA]	Critical Temp. T_c [K]	Vapour Pressure Curve Page
Acetaldehyde	C_2H_40	44.05	0.783	55.7	461	141
Acetic acid	$C_2H_4O_2$	60.05	1.049	57.9	594	143
Acetone	C_3H_6O	58.08	0.792	47.0	508	145
Air		28.98	0.559	37.7	132	150
Ammonia	NH_3	17.08	0.639	113.0	406	141
Aniline	$C_6H_5NH_2$	93.12	1.022	53.1	426	142
Argon	A	39.94	1.373	48.6	151	150
Benzene	C_6H_6	78.11	0.879	48.5	562	145
Benzoic acid	$C_7H_6O_2$	122.12	1.075	45.6	752	142
Benzyl alcohol	C_7H_8O	108.14	1.041	46.6	677	142
Boron trichloride	BCl_3	117.17	1.35	38.7	452	141
Bromine	Br_2	159.81	3.12	103.4	584	145
Bromobenzene	C_6H_5Br	157.01	1.495	45.2	670	143
Butyl ether	$C_8H_{18}O$	130.23	0.768	25.3	580	143
Carbon dioxide	COS	44.01	0.777	74.0	304	150
Carbon disulfide	CS_2	76.13	1.263	74.0	546	145
Carbon monoxide	CO	28.01	0.803	35.0	132	150
Carbon tetrachloride	CCl_4	153.82	1.594	45.6	556	145
Carbon tetrafluoride	CF_4	88.01	1.33	37.9	227	150
Carbonyl sulfide	COS	60.07	1.274	64.3	378	141
Chlorine	Cl_2	70.91	1.563	77.0	417	141
Chlorobenzene	C_6H_5Cl	112.56	1.107	45.2	632	143
Chloroform	$CHCl_3$	119.39	1.489	54.7	536	145
Cis-2-butene	C_4H_8	56.11	0.621	42.0	435	141
Cis-2-hexene	C_6H_{12}	84.16	0.687	32.8	518	145
Cis-2-pentene	C_5H_{10}	70.14	0.656	36.5	476	141
Cis-3-hexene	C_6H_{12}	84.16	0.68	32.8	517	145
Cyclohexane	C_6H_{12}	84.16	0.779	40.3	553	145
Cyclohexanone	$C_6H_{10}O$	98.15	0.951	38.5	629	143
Cyclohexene	C_6H_{10}	82.15	0.816	43.5	560	145
Cyclopentane	C_5H_{10}	70.14	0.745	45.1	512	145
Cyclopentanone	C_5H_8O	84.12	0.95	53.7	626	143
Deuterium oxide	D_2O	20.03	1.105	219.5	644	145
Dichloromethane	CH_2Cl_2	84.93	1.317	60.8	510	148
Diethyl amine	$C_4H_{11}N$	73.14	0.707	37.1	497	145
Diethyl ketone	$C_5H_{10}O$	86.13	0.814	37.4	561	146
Diethyl sulfide	$C_4H_{10}S$	90.18	0.837	39.6	557	146
Diisopropyl ether	$C_6H_{14}O$	102.18	0.724	28.8	500	146
Diphenyl	$C_{12}H_{10}$	154.21	0.99	38.5	789	142
Dipropylamine	$C_6H_{15}N$	101.19	0.738	31.4	550	143
Ethane	C_2H_6	30.07	0.548	48.8	305	150
Ethanol	C_2H_6O	46.07	0.79	63.9	516	146
Ethyl acetate	$CH_3COOC_2H_5$	88.10	0.9	38.3	523	146
Ethyl acrylate	$C_5H_8O_2$	100.12	0.921	37.5	552	146
Ethyl amine	C_2H_7N	45.09	0.683	56.2	456	148
Ethyl benzoate	$C_9H_{10}O_2$	150.18	1.046	32.4	697	142
Ethyl bromide	C_2H_5Br	108.97	1.451	62.3	504	148
Ethyl butyl ether	$C_6H_{14}O$	102.18	0.749	30.4	531	146
Ethyl chloride	C_2H_5Cl	64.52	0.896	52.7	460	148

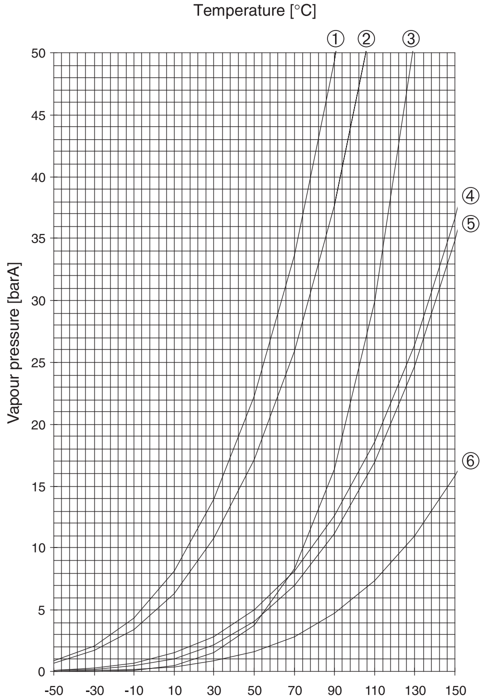
Liquid with the molecular formula		Molecular weight M g/mol	Specific Gravity G_f	Critical Pressure p_c [barA]	Critical Temp. T_c [K]	Vapour Pressure Curve Page
Ethyl ether	$C_4H_{10}O$	74.12	0.714	36.5	467	148
Ethyl formate	$C_3H_6O_2$	74.08	0.927	47.4	508	146
Ethyl propionate	$C_5H_{10}O_2$	102.13	0.895	33.6	546	146
Ethylbenzene	C_8H_{10}	106.17	0.867	36.1	617	143
Ethylcyclopentane	C_7H_{14}	98.19	0.771	33.9	570	143
Ethylene	C_2H_4	28.05	0.577	51.2	282	150
Ethylene glycol	$C_2H_6O_2$	62.07	1.155	77.0	645	142
Ethylene oxide	C_2H_4O	44.05	0.899	71.9	469	148
Fluorine	F_2	38.00	1.51	52.2	144	150
Fluorobenzene	C_6H_5F	96.10	1.024	45.5	560	146
Formaldehyde	CH_2O	30.03	0.815	65.9	408	148
Formic acid	CH_2O_2	46.03	1.226	85.1	580	146
Furan	C_4H_4O	68.08	0.938	55.0	490	148
Heptane	C_7H_{16}	100.20	0.684	27.2	540	146
Hexane	C_6H_{14}	86.17	0.659	30.0	508	146
Hydrazine	H_4N_2	32.05	1.008	146.9	653	143
Hydrogen bromide	HBr	80.92	2.16	85.6	363	150
Hydrogen chloride	HCl	36.47	1.193	84.1	325	150
Hydrogen fluoride	HF	20.01	0.967	64.8	461	148
Hydrogen iodide	HI	127.93	2.803	83.1	424	148
Hydrogen sulfide	H_2S	34.08	0.993	90.1	373	150
Iodide	I_2	253.81	3.74	116.5	819	142
Isobutane	C_4H_{10}	58.12	0.557	36.5	408	148
Isobutanol	$C_4H_{10}O$	74.12	0.802	43.0	548	146
Isobutyl acetate	$C_6H_{12}O$	116.16	0.875	30.4	561	143
Isobutyric acid	$C_4H_8O_2$	88.11	0.968	40.5	609	143
Isopropyl amine	C_3H_9N	59.11	0.688	50.7	476	148
Isopropyl chloride	C_3H_7Cl	78.54	0.862	47.2	485	148
M-cresol	C_7H_8O	108.14	1.034	45.6	706	142
M-xylene	C_8H_{10}	106.17	0.864	35.5	617	144
Methanol	CH_4O	32.04	0.792	79.7	513	149
Methyl acetate	$C_3H_6O_2$	74.08	0.934	46.9	507	147
Methyl acrylate	$C_4H_6O_2$	86.09	0.956	42.6	536	147
Methylamine	CH_5N	31.06	0.703	74.6	430	149
Methyl bromide	CH_3Br	94.94	1.737	86.1	464	149
Methyl chloride	CH_3Cl	50.49	0.915	66.8	416	149
Methyl ethyl ether	C_3H_8O	60.10	0.7	44.0	438	149
Methyl ethyl ketone	C_4H_8O	72.11	0.805	41.5	536	147
Methylcyclohexane	C_7H_{14}	98.19	0.774	34.8	572	147
Methylcyclopentane	C_6H_{12}	84.16	0.754	37.9	533	147
Morpholine	C_4H_9NO	87.12	1	54.7	618	144
N-butane	C_4H_{10}	58.12	0.579	38.0	425	141
N-butanol	$C_4H_{10}O$	74.12	0.81	44.2	563	144
N-butylaniline	$C_{10}H_{15}N$	149.24	0.932	28.4	721	142
N-butyric acid	$C_4H_8O_2$	88.11	0.958	52.7	628	144
N-hexane	C_6H_{14}	86.18	0.659	29.7	507	147
N-octane	C_8H_{18}	114.23	0.703	24.8	569	144
Nitric oxide	NO	30.01	1.28	65.9	180	151
Nitro methane	CH_3NO_2	61.04	1.138	63.1	588	147
Nitrogen	N_2	28.02	0.804	33.9	126	151

Liquid with the molecular formula		Molecular weight M g/mol	Specific Gravity G_f	Critical Pressure p_c [barA]	Critical Temp. T_c [K]	Vapour Pressure Curve Page
Nitrogen dioxide	NO_2	46.01	1.447	101.3	431	141
Nitrous oxide	N_2O	44.02	1.226	72.7	310	151
Octane	C_8H_{18}	114.22	0.702	25.0	569	144
O-cresol	C_7H_8O	108.14	1.028	50.1	698	142
O-xylene	C_8H_{10}	106.17	0.88	37.3	630	144
Oxygen	O_2	32.00	1.149	50.4	155	151
Ozone	O_3	48.00	1.356	93.5	261	151
P-cresol	C_7H_8O	108.14	1.019	51.5	705	142
P-xylene	C_8H_{10}	106.17	0.861	35.2	616	144
Pentane	C_5H_{12}	72.15	0.626	33.4	470	141
Perchloryl fluoride	$ClFO_3$	102.45	2.003	54.4	368	141
Phenol	C_6H_6O	94.11	1.059	61.3	694	142
Phosgene	CCl_2O	98.92	1.381	57.5	455	141
Propane	C_3H_8	44.09	0.582	42.6	370	141
Propionic acid	$C_3H_6O_2$	74.08	0.993	53.7	612	144
Propyl chloride	C_3H_7Cl	78.54	0.891	45.8	503	147
Propylene	C_3H_6	42.08	0.612	45.9	365	149
Propylene oxide	C_3H_6O	58.08	0.829	49.2	482	149
Pyridine	C_5H_5N	79.10	0.983	60.8	617	144
Styrene	C_8H_8	104.15	0.906	39.9	647	144
Sulphur dioxide	SO_2	64.06	1.455	78.8	431	149
Toluene	C_7H_8	92.13	0.866	42.2	594	144
Trans-2-butene	C_4H_8	56.11	0.604	41.0	429	149
Trans-2-hexene	C_6H_{12}	84.16	0.678	32.7	516	147
Trans-2-pentene	C_5H_{10}	70.14	0.649	36.6	475	149
Trans-3-hexene	C_6H_{12}	84.16	0.677	32.5	520	147
Trichloroethylene	C_2HCl_3	131.39	1.462	49.2	571	147
Triethylamine	$C_6H_{15}N$	101.19	0.728	30.4	535	144
Valeraldehyde	$C_5H_{10}O$	86.13	0.81	35.5	554	147
Vinyl acetate	$C_4H_6O_2$	86.09	0.932	43.6	525	147
Vinyl chloride	C_2H_3Cl	62.50	0.969	56.0	430	149
Water	H_2O	18.02	0.998	220.4	647	132-133

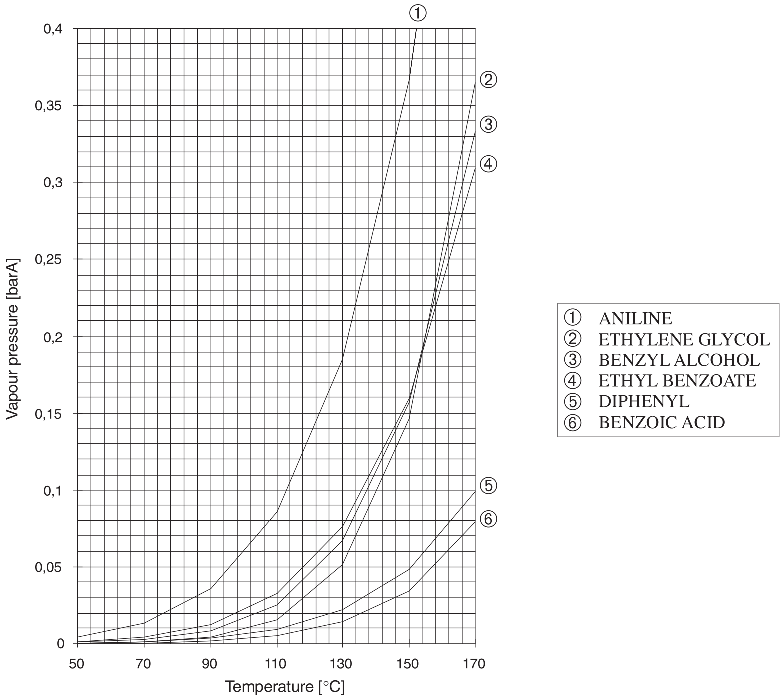
VAPOUR PRESSURE CURVES

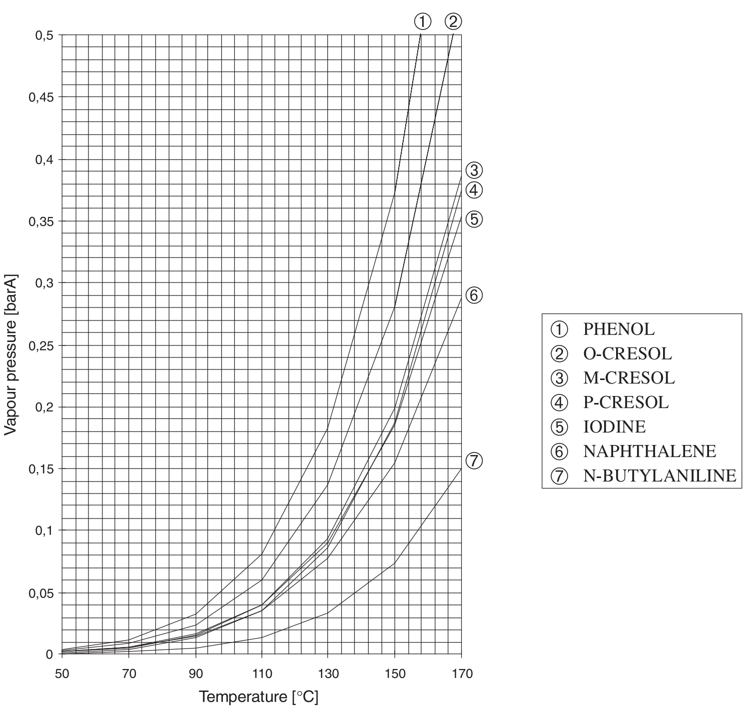


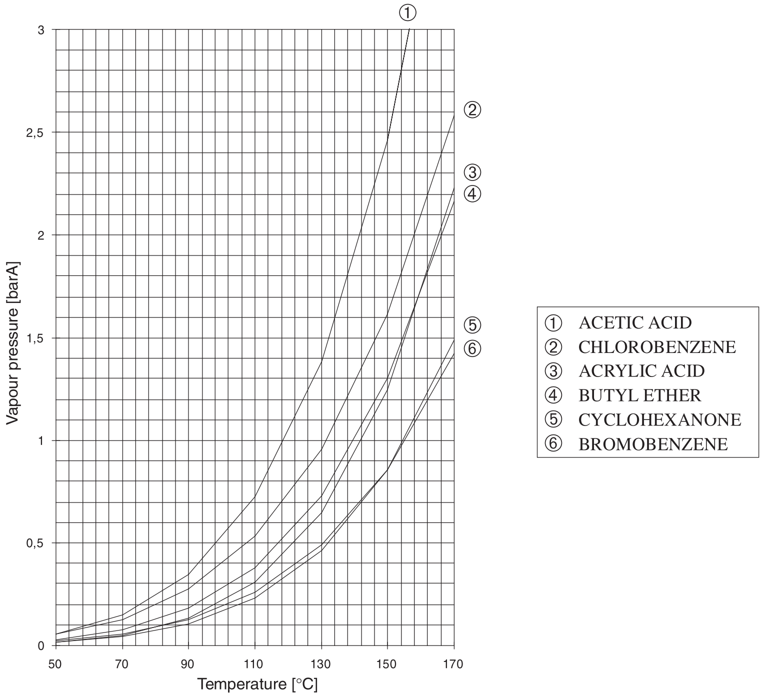
① CARBONYL SULFIDE
② AMMONIA
③ CHLORINE
④ CIS-2-BUTENE
⑤ ACETALDEHYDE
⑥ BORON TRICHLORIDE
⑦ CIS-2-PENTENE



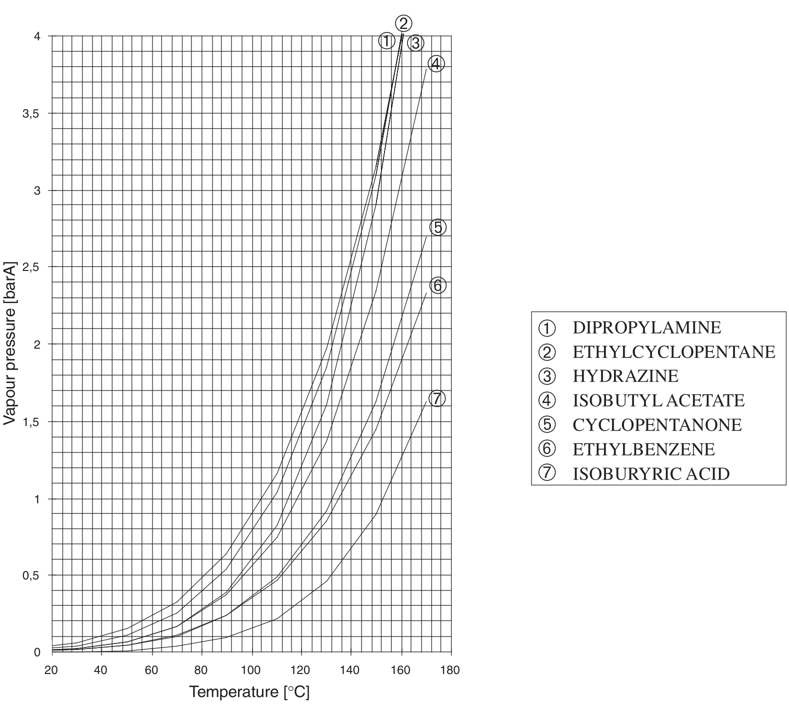
① PERCHLORYL FLUORIDE
② PROPANE
③ NITROGEN DIOXIDE
④ N-BUTANE
⑤ PHOSGENE
⑥ PENTANE


Vapour pressure [barA]
0,4
0,35
0,3
0,25
0,2
0,15
0,1
0,05
0
50
70
90
110
130
150
170
Temperature [°C]
① ANILINE
② ETHYLENE GLYCOL
③ BENZYL ALCOHOL
④ ETHYL BENZOATE
⑤ DIPHENYL
⑥ BENZOIC ACID

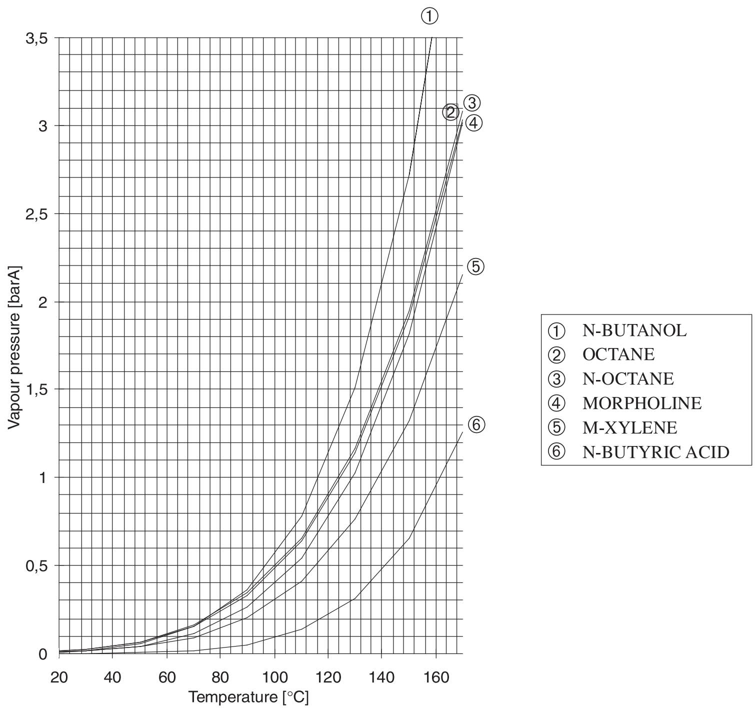

Vapour pressure [barA]
0,5
0,45
0,4
0,35
0,3
0,25
0,2
0,15
0,1
0,05
0
50
70
90
110
130
150
170
Temperature [°C]
① PHENOL
② O-CRESOL
③ M-CRESOL
④ P-CRESOL
⑤ IODINE
⑥ NAPHTHALENE
⑦ N-BUTYLANILINE

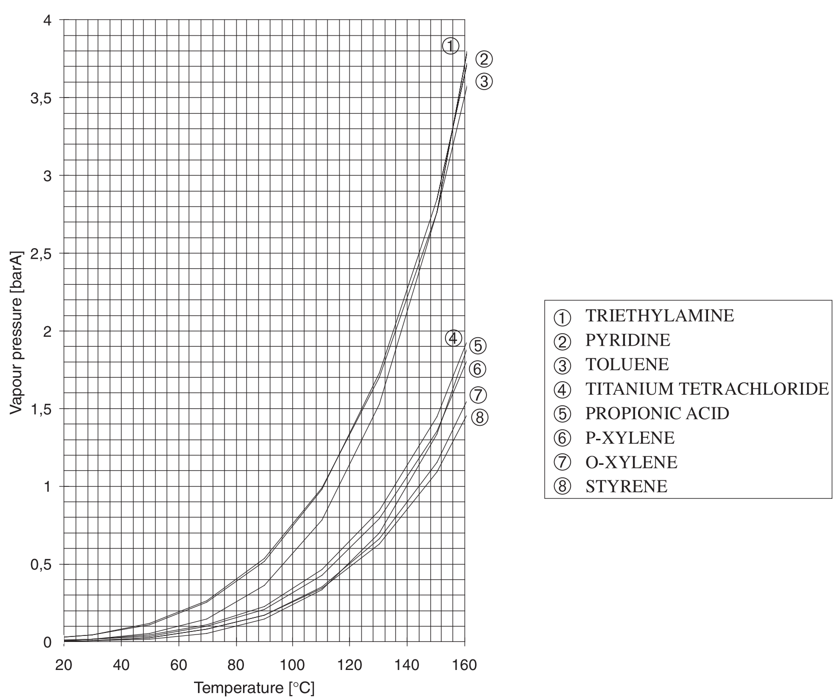


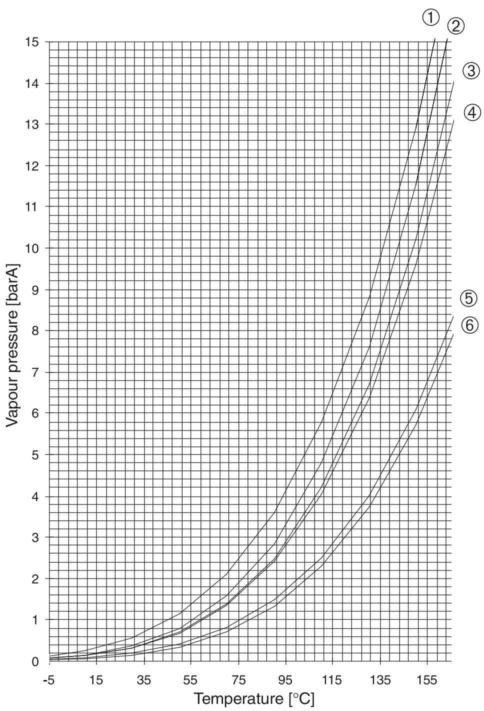
Vapour pressure [barA]
Temperature [°C]
① ACETIC ACID
② CHLOROBENZENE
③ ACRYLIC ACID
④ BUTYL ETHER
⑤ CYCLOHEXANONE
⑥ BROMOBENZENE



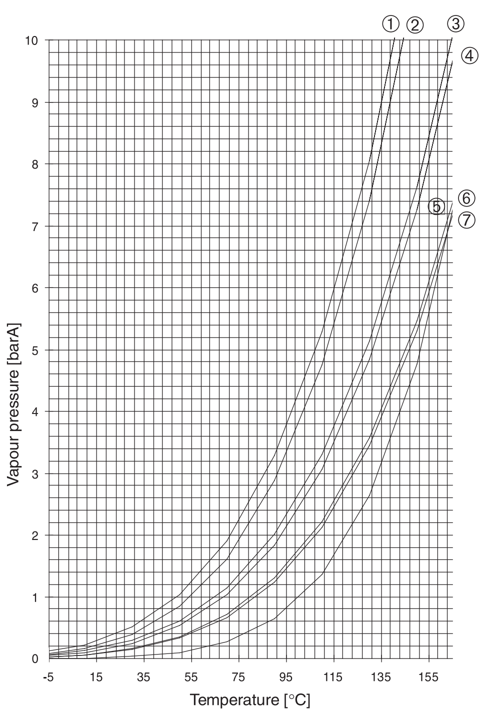
Vapour pressure [barA]
Temperature [°C]
① DIPROPYLAMINE
② ETHYLCYCLOPENTANE
③ HYDRAZINE
④ ISOBUTYL ACETATE
⑤ CYCLOPENTANONE
⑥ ETHYLBENZENE
⑦ ISOBURYRIC ACID


Vapour pressure [barA]
Temperature [°C]
① N-BUTANOL
② OCTANE
③ N-OCTANE
④ MORPHOLINE
⑤ M-XYLENE
⑥ N-BUTYRIC ACID

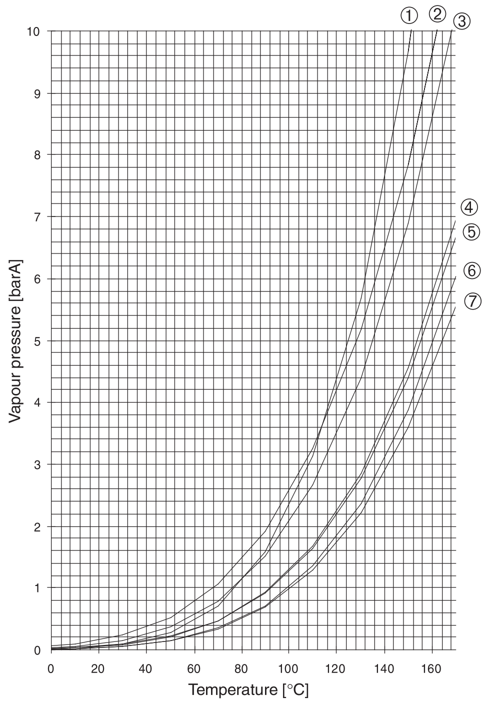

Vapour pressure [barA]
Temperature [°C]
① TRIETHYLAMINE
② PYRIDINE
③ TOLUENE
④ TITANIUM TETRACHLORIDE
⑤ PROPIONIC ACID
⑥ P-XYLENE
⑦ O-XYLENE
⑧ STYRENE



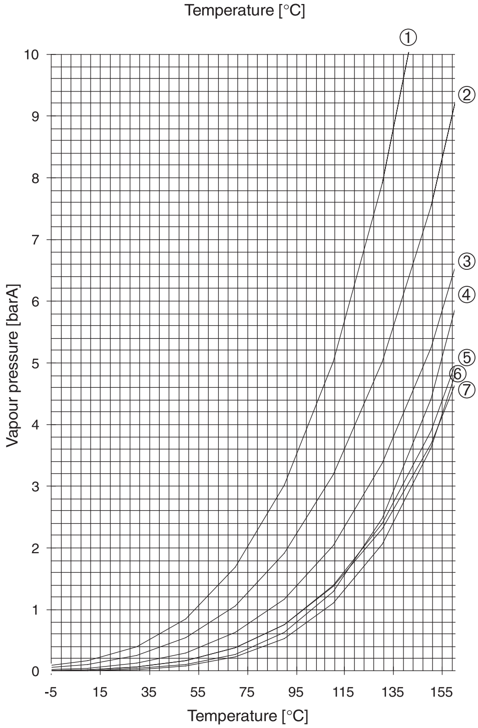
1. CARBON DISULFIDE
2. ACETONE
3. BROMINE
4. CHLOROFORM
5. CARBON TETRACHLORIDE
6. BENZENE



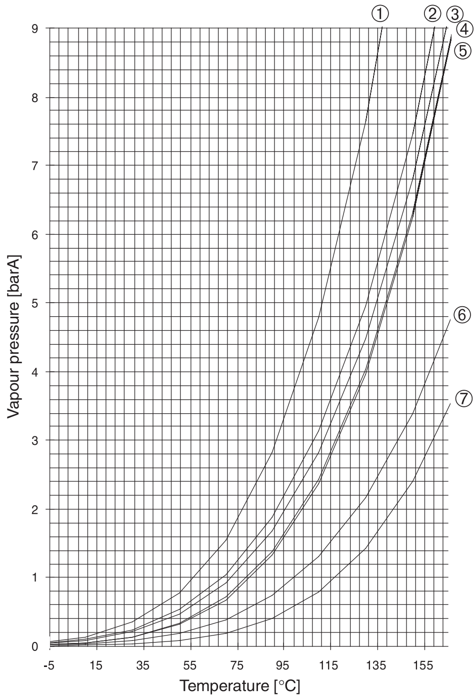
1. CYCLOPENTANE
2. DIETHYL AMINE
3. CIS-3-HEXENE
4. CIS-2-HEXENE
5. DEUTERIUM OXIDE
6. CYCLOHEXANE
7. CYCLOHEXENE



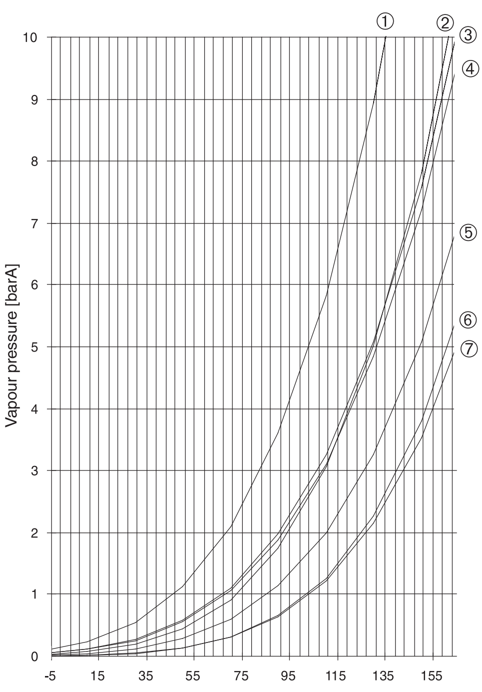
① ETHANOL
② DIISOPROPYL ETHER
③ ETHYL ACETATE
④ ETHYL BUTYL ETHER
⑤ DIETHYL SULFIDE
⑥ ETHYL ACRYLATE
⑦ DIETHYL KETONE



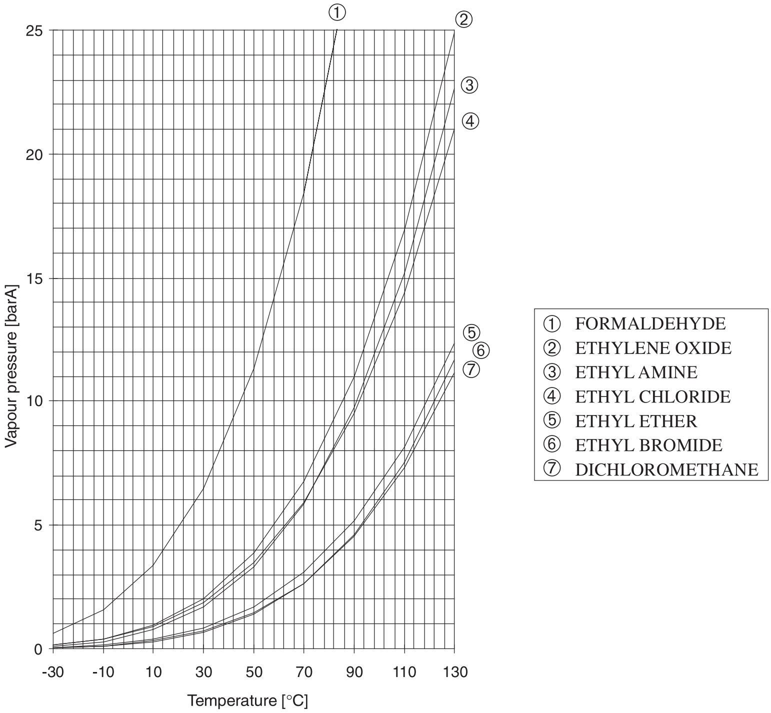
① ETHYL FORMATE
② HEXANE
③ FLUOROBENZENE
④ FORMIC ACID
⑤ ETHYL PROPIONATE
⑥ ISOBUTANOL
⑦ HEPTANE

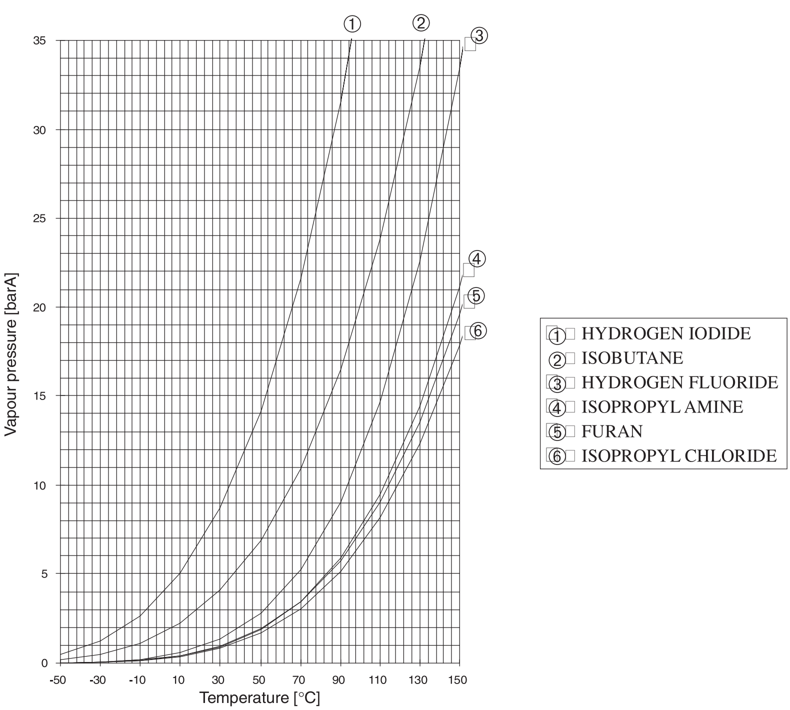


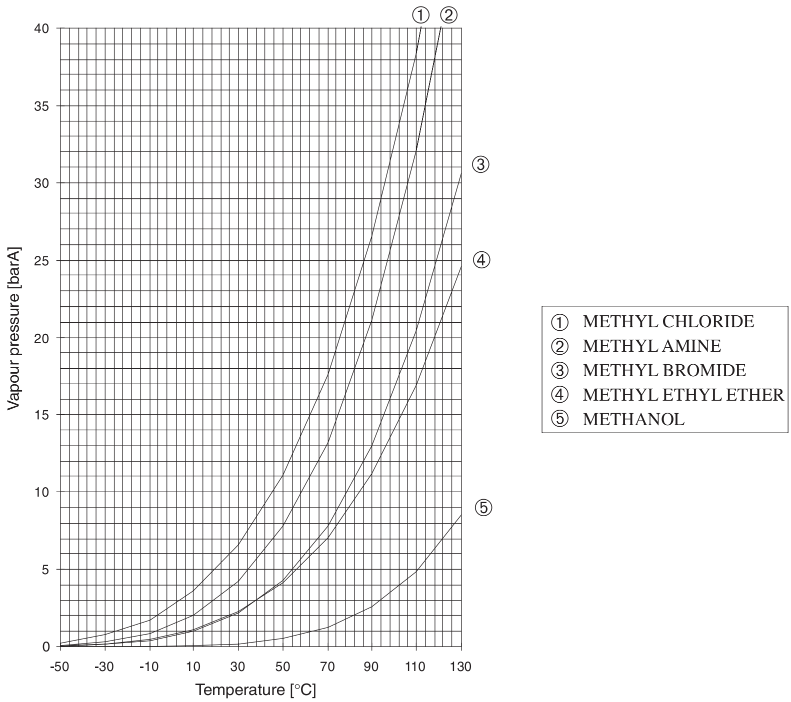
① METHYL ACETATE
② N-HEXANE
③ METHYLCYCLOPENTANE
④ METHYL ACRYLATE
⑤ METHYL ETHYL KETONE
⑥ METHYLCYCLOHEXANE
⑦ METHYL ISOBUTYL KETONE

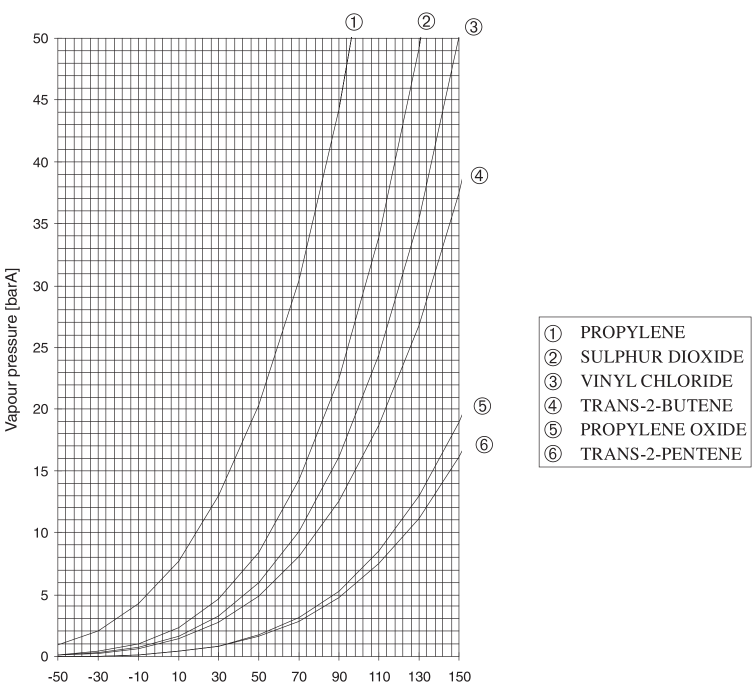


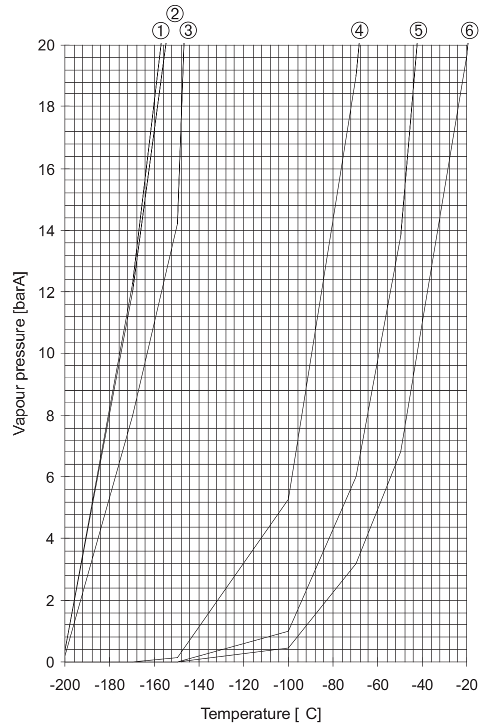
① PROPYL CHLORIDE
② VINYL ACETATE
③ TRANS-2-HEXENE
④ TRANS-3-HEXENE
⑤ TRICHLOROETHYLENE
⑥ NITRO METHANE
⑦ VALERALDEHYDE


Vapour pressure [barA]
0
5
10
15
20
25
-30
-10
10
30
50
70
90
110
130
Temperature [°C]
① FORMALDEHYDE
② ETHYLENE OXIDE
③ ETHYL AMINE
④ ETHYL CHLORIDE
⑤ ETHYL ETHER
⑥ ETHYL BROMIDE
⑦ DICHLOROMETHANE

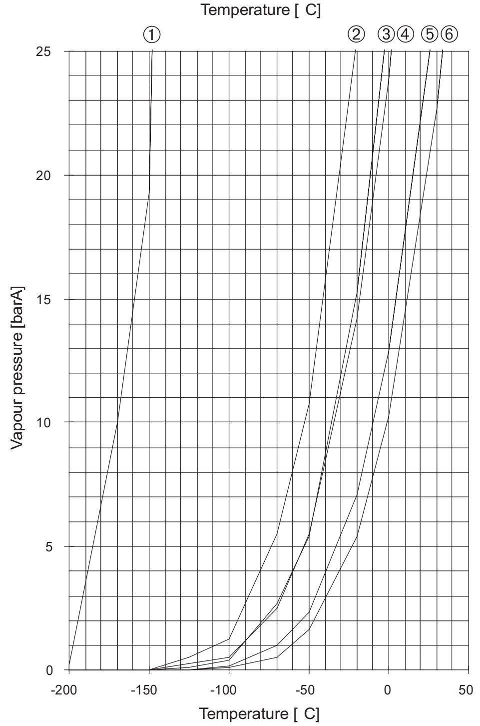

Vapour pressure [barA]
0
5
10
15
20
25
30
35
-50
-30
-10
10
30
50
70
90
110
130
150
Temperature [°C]
① HYDROGEN IODIDE
② ISOBUTANE
③ HYDROGEN FLUORIDE
④ ISOPROPYL AMINE
⑤ FURAN
⑥ ISOPROPYL CHLORIDE


Vapour pressure [barA]
Temperature [°C]
① METHYL CHLORIDE
② METHYL AMINE
③ METHYL BROMIDE
④ METHYL ETHYL ETHER
⑤ METHANOL

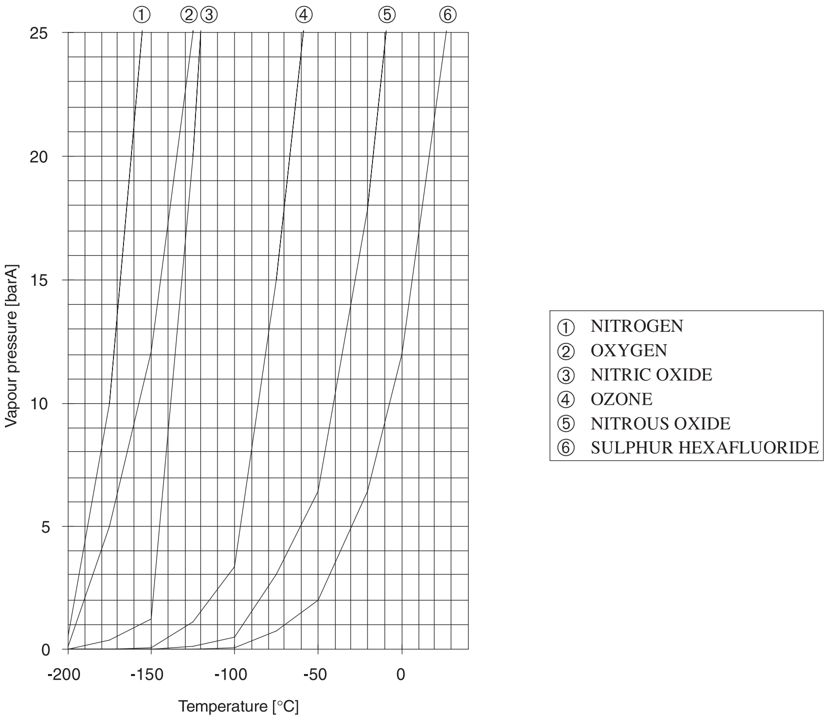

Vapour pressure [barA]
Temperature [°C]
① PROPYLENE
② SULPHUR DIOXIDE
③ VINYL CHLORIDE
④ TRANS-2-BUTENE
⑤ PROPYLENE OXIDE
⑥ TRANS-2-PENTENE



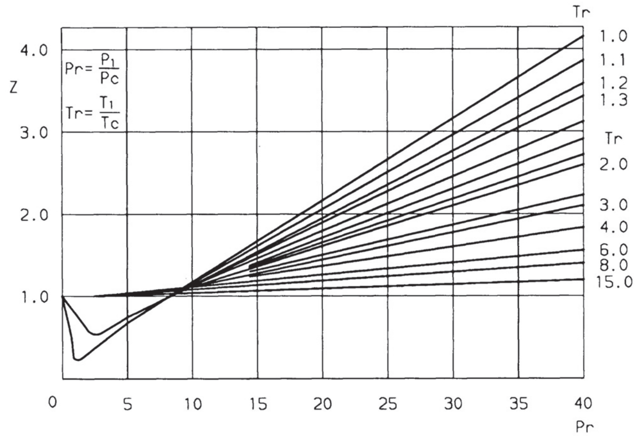
① AIR
② CARBON MONOXIDE
③ ARGON
④ CARBON TETRAFLUORIDE
⑤ BORON TRIFLUORIDE
⑥ CARBON DIOXIDE

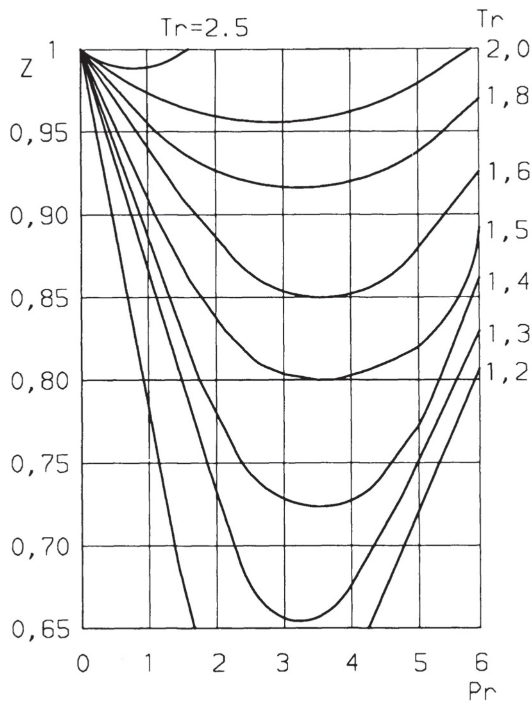


① FLUORINE
② ETHYLENE
③ HYDROGEN CHLORIDE
④ ETHANE
⑤ HYDROGEN BROMIDE
⑥ HYDROGEN SULFIDE


①
②
③
④
⑤
⑥
25
20
15
10
5
0
Vapour pressure [barA]
-200
-150
-100
-50
0
Temperature [°C]
① NITROGEN
② OXYGEN
③ NITRIC OXIDE
④ OZONE
⑤ NITROUS OXIDE
⑥ SULPHUR HEXAFLUORIDE

COMPRESSIBILITY FACTOR Z





Compressibility factor Z as function of Pr and Tr.

PIPE TABLES

Dimensions and weights of most common welded and seamless wrought steel pipe based on ANSI/ASME B36.10M standard.

Nominal size [in]	US UNITS Outside diameter [in]	Wall thickness [in]	Plain end weight [lb/ft]	IDENTIFICATION API Spec.	Strength standard *)	Schedule number	SI UNITS Outside diameter [mm]	Wall thick-ness [mm]	Plain end weight [kg/m]
1/2	0.840	0.109	0.85	L5	STD	40	21.3	2.77	1.27
1/2	0.840	0.147	1.09	L5	XS	80	21.3	3.73	1.62
1/2	0.840	0.188	1.31	...	...	160	21.3	4.78	1.95
1/2	0.840	0.294	1.71	L5	XXS	...	21.3	7.47	2.55
3/4	1.050	0.113	1.13	L5	STD	40	26.7	2.87	1.68
3/4	1.050	0.154	1.47	L5	XS	80	26.7	3.91	2.20
3/4	1.050	0.219	1.94	...	...	160	26.7	5.56	2.90
3/4	1.050	0.308	2.44	L5	XXS	...	26.7	7.82	3.64
1	1.315	0.133	1.68	L5	STD	40	33.4	3.38	2.50
1	1.315	0.179	2.17	L5	XS	80	33.4	4.55	3.24
1	1.315	0.250	2.84	...	...	160	33.4	6.35	4.24
1	1.315	0.358	3.66	L5	XXS	...	33.4	9.09	5.45
1 1/2	1.900	0.145	2.72	L5	STD	40	48.3	3.68	4.05
1 1/2	1.900	0.200	3.63	L5	XS	80	48.3	5.08	5.41
1 1/2	1.900	0.281	4.86	...	...	160	48.3	7.14	7.24
1 1/2	1.900	0.400	6.41	L5	XXS	...	48.3	10.16	9.55
2	2.375	0.154	3.65	L5	STD	40	60.3	3.91	5.44
2	2.375	0.218	5.02	L5	XS	80	60.3	5.54	7.48
2	2.375	0.344	7.46	...	...	160	60.3	8.74	11.12
2	2.375	0.436	9.03	...	XXS	...	60.3	11.07	13.45
3	3.500	0.216	7.58	L5	STD	40	88.9	5.49	11.29
3	3.500	0.300	10.25	L5	XS	80	88.9	7.62	15.27
3	3.500	0.438	14.32	...	...	160	88.9	11.13	21.34
3	3.500	0.600	18.58	L5	XXS	...	88.9	15.24	27.68
4	4.500	0.237	10.79	L5	STD	40	114.3	6.02	16.07
4	4.500	0.337	14.98	L5	XS	80	114.3	8.56	22.32
4	4.500	0.438	19.00	L5	...	120	114.3	11.13	28.31
4	4.500	0.531	22.51	L5	...	160	114.3	13.49	33.53
4	4.500	0.674	27.54	L5	XXS	...	114.3	17.12	41.03
6	6.625	0.280	18.97	L5	STD	40	168.3	7.11	28.27
6	6.625	0.432	28.57	L5	XS	80	168.3	10.97	42.56
6	6.625	0.562	36.39	L5	...	120	168.3	14.27	54.21
6	6.625	0.719	45.35	L5	...	160	168.3	18.26	67.56
6	6.625	0.864	53.16	...	XXS	...	168.3	21.95	79.19

the table continues on the next page

Nominal size [in]	US UNITS			IDENTIFICATION			SI UNITS		
	Outside diameter [in]	Wall thickness [in]	Plain end weight [lb/ft]	API Spec.	Strength standard *)	Schedule number	Outside diameter [mm]	Wall thick-ness [mm]	Plain end weight [kg/m]
8	8.625	0.250	22.36	L5	...	20	219.1	6.35	33.31
8	8.625	0.277	24.70	L5	...	30	219.1	7.04	36.79
8	8.625	0.322	28.55	L5	STD	40	219.1	8.18	42.54
8	8.625	0.406	35.64	...	...	60	219.1	10.31	53.09
8	8.625	0.500	43.39	L5	XS	80	219.1	12.70	64.63
8	8.625	0.594	50.95	...	...	100	219.1	15.09	75.90
8	8.625	0.719	60.71	L5	...	120	219.1	18.26	90.44
8	8.625	0.812	67.76	L5	...	140	219.1	20.62	100.93
8	8.625	0.875	72.42	L5	XXS	...	219.1	22.23	107.89
8	8.625	0.806	67.31	...	...	160	219.1	20.47	100.26
10	10.750	0.250	28.04	L5	...	20	273.1	6.35	41.76
10	10.750	0.307	34.24	L5	...	30	273.1	7.80	51.01
10	10.750	0.365	40.48	L5	STD	40	273.1	9.27	60.31
10	10.750	0.500	54.74	L5	XS	60	273.1	12.70	81.54
10	10.750	0.594	64.43	...	...	80	273.1	15.09	95.98
10	10.750	0.719	77.03	L5	...	100	273.1	18.26	114.74
10	10.750	0.844	89.29	...	...	120	273.1	21.44	133.02
10	10.750	1.000	104.13	L5	XXS	140	273.1	25.40	155.12
10	10.750	1.125	115.64	...	...	160	273.1	28.58	172.27
12	12.750	0.250	33.38	L5	...	20	323.9	6.35	49.72
12	12.750	0.330	43.77	L5	...	30	323.9	8.38	65.21
12	12.750	0.375	49.56	L5	STD	...	323.9	9.53	73.83
12	12.750	0.406	53.52	L5	...	40	323.9	10.31	79.73
12	12.750	0.500	65.42	L5	XS	...	323.9	12.70	97.45
12	12.750	0.562	73.15	L5	...	60	323.9	14.27	108.98
12	12.750	0.688	88.63	L5	...	80	323.9	17.48	132.03
12	12.750	0.844	107.32	...	...	100	323.9	21.44	159.87
12	12.750	1.000	125.49	L5	XXS	120	323.9	25.40	186.94
12	12.750	1.125	139.67	L5	...	140	323.9	28.58	208.07
12	12.750	1.312	160.27	L5	...	160	323.9	33.32	238.75
14	14.000	0.250	36.71	L5	...	10	355.6	6.35	54.69
14	14.000	0.312	45.61	L5	...	20	355.6	7.92	67.94
14	14.000	0.375	54.57	L5	STD	30	355.6	9.53	81.29
14	14.000	0.438	63.44	L5	...	40	355.6	11.13	94.51
14	14.000	0.500	72.09	L5	XS	...	355.6	12.70	107.39
14	14.000	0.594	85.05	...	...	60	355.6	15.09	126.69
14	14.000	0.750	106.13	L5	...	80	355.6	19.05	158.10
14	14.000	0.938	130.85	L5	...	100	355.6	23.83	194.93
14	14.000	1.094	150.79	...	...	120	355.6	27.79	224.63
14	14.000	1.250	170.21	L5	...	140	355.6	31.75	253.56
14	14.000	1.406	189.11	...	...	160	355.6	35.71	281.71

the table continues on the next page

Nominal size [in]	US UNITS			IDENTIFICATION			SI UNITS		
	Outside diameter [in]	Wall thickness [in]	Plain end weight [lb/ft]	API Spec.	Strength standard *)	Schedule number	Outside diameter [mm]	Wall thick-ness [mm]	Plain end weight [kg/m]
16	16.000	0.250	42.05	L5	...	10	406.4	6.35	62.64
16	16.000	0.312	52.27	L5	...	20	406.4	7.92	77.87
16	16.000	0.375	62.58	L5	STD	30	406.4	9.53	93.22
16	16.000	0.500	82.77	L5	XS	40	406.4	12.70	123.30
16	16.000	0.656	107.50	...	...	60	406.4	16.66	160.14
16	16.000	0.844	136.61	...	...	80	406.4	21.44	203.51
16	16.000	1.031	164.82	...	...	100	406.4	26.19	245.53
16	16.000	1.219	192.43	...	...	120	406.4	30.96	286.66
16	16.000	1.438	223.64	...	...	140	406.4	36.53	333.15
16	16.000	1.594	245.25	...	...	160	406.4	40.49	365.34
18	18.000	0.250	47.39	L5	...	10	457	6.35	70.60
18	18.000	0.312	58.94	L5	...	20	457	7.92	87.80
18	18.000	0.375	70.59	L5	STD	...	457	9.53	105.15
18	18.000	0.438	82.15	L5	...	30	457	11.13	122.38
18	18.000	0.500	93.45	L5	XS	...	457	12.70	139.21
18	18.000	0.562	104.67	L5	...	40	457	14.27	155.92
18	18.000	0.750	138.17	L5	...	60	457	19.05	205.83
18	18.000	0.938	170.92	L5	...	80	457	23.83	254.62
18	18.000	1.156	207.96	...	...	100	457	29.36	309.79
18	18.000	1.475	260.32	...	...	120	457	37.47	387.79
18	18.000	1.562	274.22	...	...	140	457	39.67	408.50
18	18.000	1.781	308.50	...	...	160	457	45.24	459.57
20	20.000	0.250	52.73	L5	...	10	508	6.35	78.55
20	20.000	0.375	78.60	L5	STD	20	508	9.53	117.09
20	20.000	0.500	104.13	L5	XS	30	508	12.70	155.12
20	20.000	0.594	123.11	...	...	40	508	15.09	183.39
20	20.000	0.812	166.40	L5	...	60	508	20.62	247.88
20	20.000	1.031	208.87	...	...	80	508	26.19	311.15
20	20.000	1.281	256.10	...	...	100	508	32.54	381.50
20	20.000	1.500	296.37	...	...	120	508	38.10	441.49
20	20.000	1.750	341.09	...	...	140	508	44.45	508.11
20	20.000	1.969	379.17	...	...	160	508	50.01	564.84

*) STD = standard
XS = extra strong
XXS = double extra strong

PRESSURE DROP IN STEEL PIPES

Pressure drop (Δp) of 20 °C water per 10 m length of new commericial steel pipe with varios pipe internal diameter (DN) and velocity (v) or flow rate (q).

DN 25		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	0.177	0.00060
0.2	0.353	0.00304
0.5	0.884	0.01500
1	1.767	0.05000
2	3.534	0.17600
3	5.301	0.37800
5	8.836	1.05000
10	17.671	3.80000

DN 40		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	0.452	0.00050
0.2	0.905	0.00170
0.5	2.262	0.00844
1	4.524	0.02875
2	9.048	0.10000
3	13.572	0.22500
5	22.619	0.59375
10	45.239	2.31250

DN 50		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	0.707	0.00040
0.2	1.414	0.00128
0.5	3.534	0.00625
1	7.069	0.02200
2	14.137	0.08000
3	21.206	0.18000
5	35.343	0.47500
10	70.686	1.80000

DN 80		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	1.810	0.00021
0.2	3.619	0.00070
0.5	9.048	0.00344
1	18.096	0.01250
2	36.191	0.04500
3	54.287	0.10125
5	90.478	0.26563
10	180.96	1.03125

DN 100		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	2.827	0.00016
0.2	5.655	0.00056
0.5	14.137	0.00275
1	28.274	0.01000
2	56.549	0.03500
3	84.823	0.07650
5	141.37	0.20000
10	282.74	0.77500

DN 150		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	6.362	0.00010
0.2	12.723	0.00032
0.5	31.809	0.00183
1	63.617	0.00600
2	127.23	0.02200
3	190.85	0.04650
5	318.09	0.12500
10	636.17	0.48333

DN 200		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	11.310	0.00007
0.2	22.619	0.00022
0.5	56.549	0.00119
1	113.10	0.00425
2	226.19	0.01550
3	339.29	0.03263
5	565.49	0.08750
10	1131.0	0.33750

DN 250		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	17.671	0.00005
0.2	35.343	0.00018
0.5	88.357	0.00093
1	176.71	0.00330
2	353.43	0.01200
3	530.14	0.02610
5	883.57	0.07000
10	1767.1	0.27000

DN 300		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	25.4	0.00004
0.2	50.9	0.00015
0.5	127	0.00073
1	254	0.00258
2	509	0.00967
3	763	0.02100
5	1272	0.05417
10	2545	0.20833

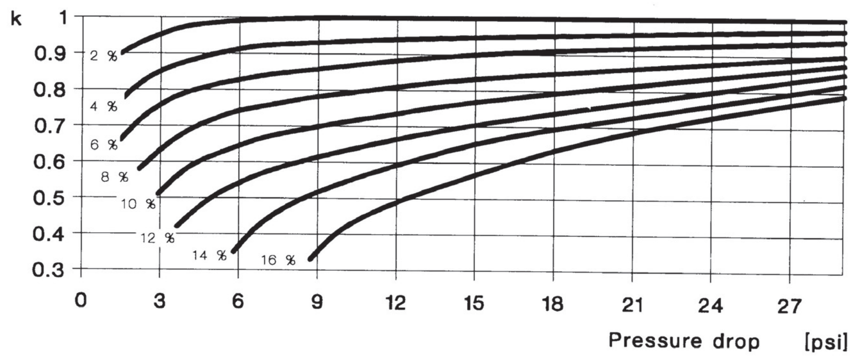
DN 400		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	45.239	0.00003
0.2	90.478	0.00010
0.5	226.19	0.00053
1	452.39	0.00188
2	904.78	0.00675
3	1357.2	0.01406
5	2261.9	0.03750
10	4523.9	0.15000

DN 500		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	70.686	0.00002
0.2	141.37	0.00008
0.5	353.43	0.00040
1	706.86	0.00140
2	1413.7	0.00520
3	2120.6	0.01080
5	3534.3	0.03000
10	7068.6	0.11500

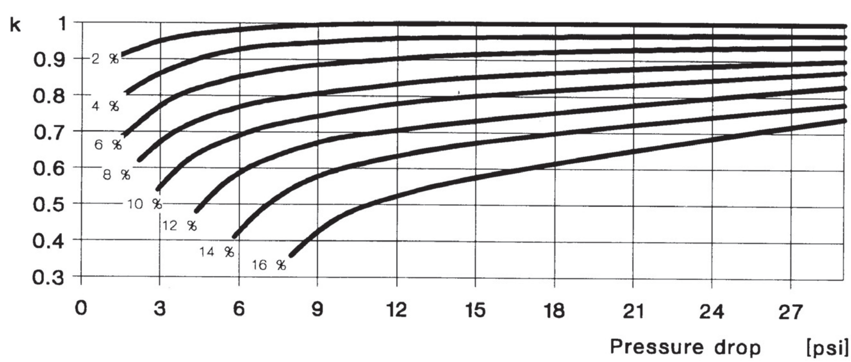
DN 600		
v [m/s]	q [m^3/h]	Δp [bar]
0.1	101.79	0.00002
0.2	203.58	0.00006
0.5	508.94	0.00032
1	1017.9	0.00113
2	2035.8	0.00400
3	3053.6	0.00900
5	5089.4	0.02396
10	10178.8	0.09167

CORRECTION COEFFICIENT k FOR PULP FLOW

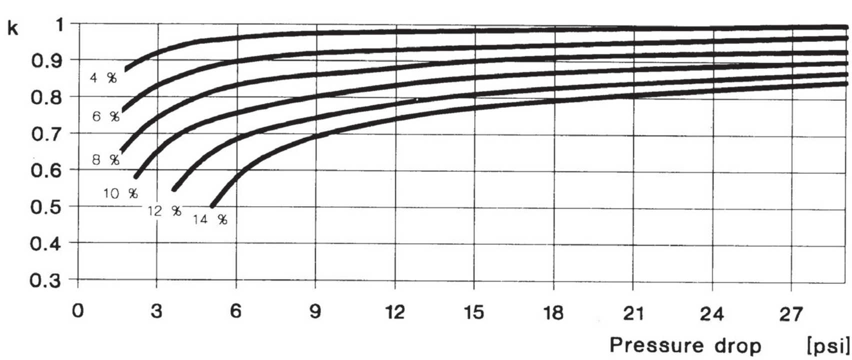
Kraft pulp



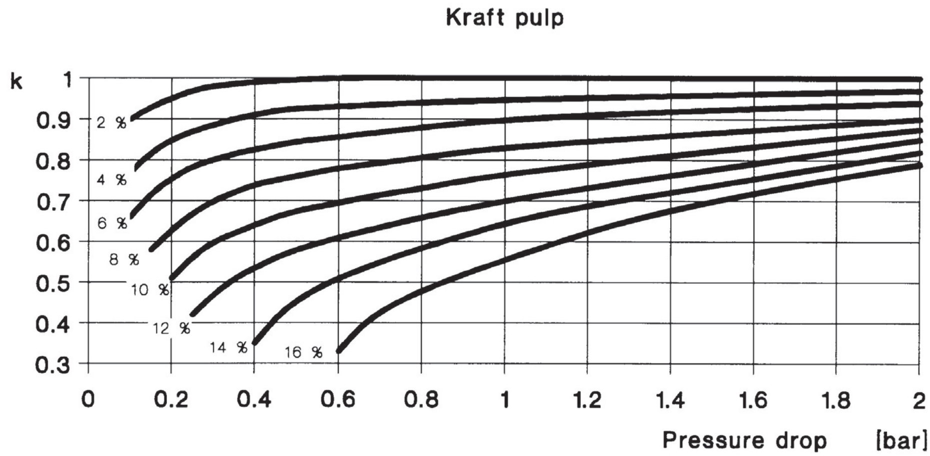
Mechanical pulp

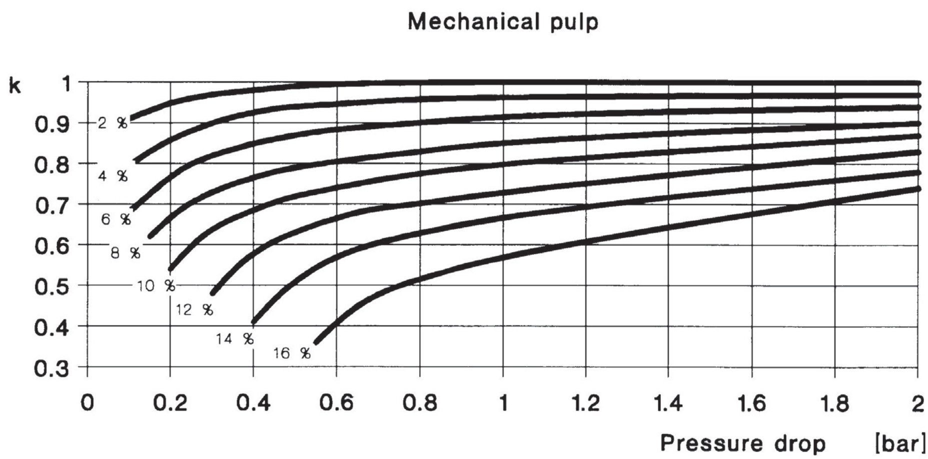


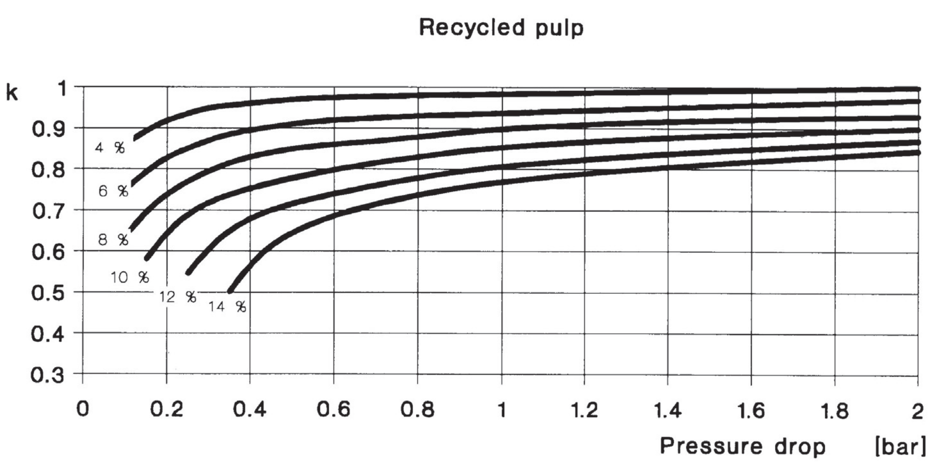
Recycled pulp



Correction coefficient in SI units.


Kraft pulp
k
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
2 %
4 %
6 %
8 %
10 %
12 %
14 %
16 %
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2
Pressure drop [bar]


Mechanical pulp
k
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
2 %
4 %
6 %
8 %
10 %
12 %
14 %
16 %
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2
Pressure drop [bar]

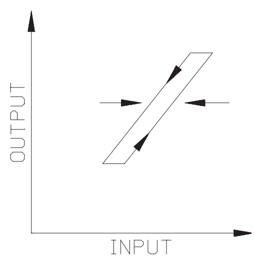

Recycled pulp
k
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
4 %
6 %
8 %
10 %
12 %
14 %
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2
Pressure drop [bar]

NOISE PREDICTION ASPECTS

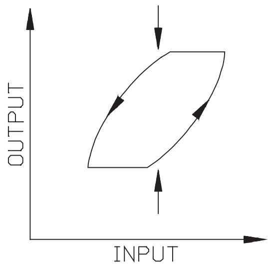
In control valve sizing calculations in this flow control manual, the practice is to use IEC/ISA sizing methods. The valve noise level calculation method presented, however, is based mainly on the VDMA 24422 (version 1979) and IEC 60534-8 standards. The calculation procedure presented here covers only the noise generated by the dynamics of flow within a closed piping system, whereas mechanical rattling, transmitted noise amplification caused by reflections and/or resonances are excluded. The calculation procedure is based on sound pressure level measurements carried out using a standardized test arrangement. The microphone position is 1 m downstream from the valve outlet, and 1 m from pipe surface. The calculations are valid for a straight uninsulated pipe in an open environment. The noise calculation equations are verified by experiments. Therefore, owing to the large number of factors influencing noise level, the accuracy is within tolerance band of up to 10 dB(A).

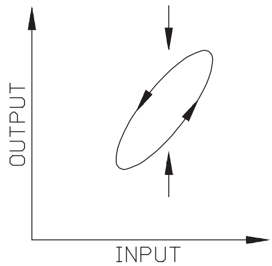
Control valve performance terminology

backlash	A relative movement between mechanical parts when direction of motion is reversed
dead band	The range through which an input signal can be varied without response in output.



dead time	The interval of time between input change and the start of output response.
deviation	Any departure from a desired or expected value.
error	The difference between the indication and the ideal value of the measured signal.
hunting	An undesirable oscillation of output with constant input.
hysteresis plus dead band	Maximum deviation between measured output with increasing signal compared to output measured with decreasing signal. Hysteresis can be achieved by subtracting dead band from measured value.





linearity	The closeness to which a curve approximates a straight line.
rangeability	Installed rangeability is the ratio of maximum to minimum flow where flow is within some tolerance. Inherent rangeability is similarly ratio of maximum to minimum flow coefficients.
repeatability	The closeness of consecutive measurements of the output for the same input approaching from the same direction.
resolution	The least interval between two adjacent values which can be distinguished one from the other.
response time	The time required for an output to change from an initial value to a specified percentage of the final value as a result of input change.
stiction	Static friction resistance to the start of motion.
turndown	The ratio of plant maximum designed flow rate to minimum designed flow rate.

MANIFOLD DESIGN GUIDELINES FOR VALVES

Different components in process piping cause disturbances in the flow pattern. Pumps, joints, elbows etc. are all components that may affect the performance of a control valve. These disturbances must be taken into account when control valves are selected. If maximum capacity is required, it is recommended that valves are installed in the pipe line with enough straight pipe upstream and downstream of the valve to ensure constant velocity profile distribution. Some examples of possible disturbances to the flow pattern are given below.

Since every case is individual there are no universal rules for manifold design, but the following pages contain some information about the subject. The designer should always remember that control valve applications are usually designed individually, using detailed data for each case.

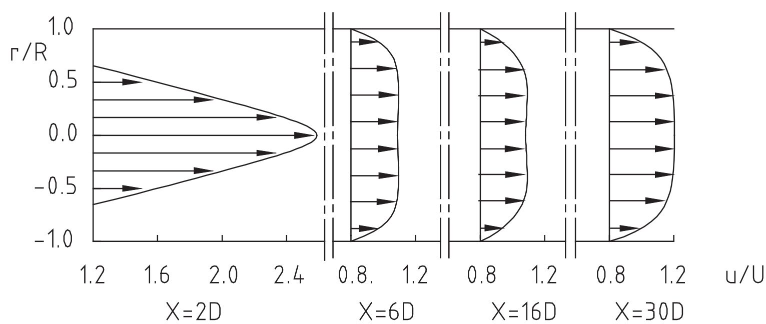
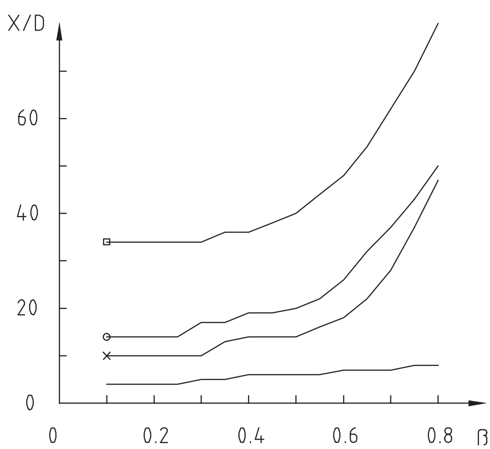


Figure 1. Velocity profile distribution after an orifice at different distances



where
x= 90 deg bent or an T-joint
O= two or more 90 deg bend in same plane
= two or more 90 deg bend in different plane
• = straight distance after the apparat in question
ß = flow port area / pipe area

Figure 2. Straight pipeline needed with orifices to achieve fully formed velocity distribution

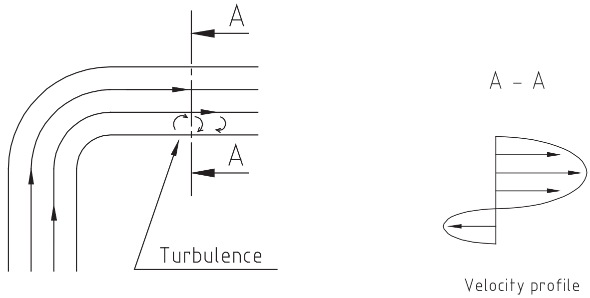


Figure 3. Flow pattern in a 90º bend. Where the turbulence is heavy cavitation may occur.

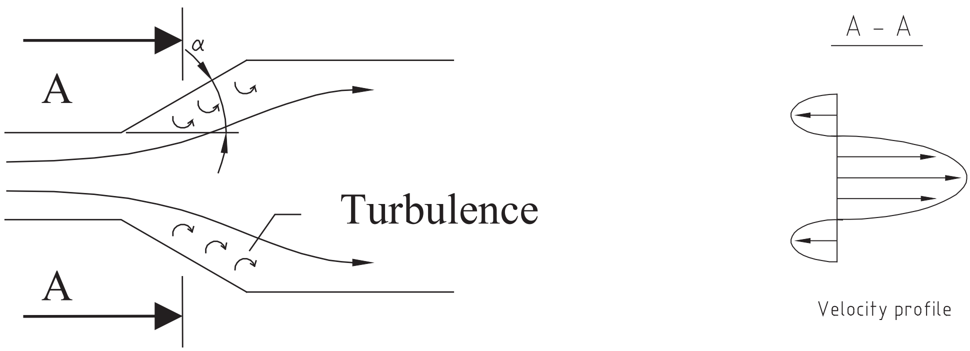


Figure 4. Flow pattern in an expansion piece.

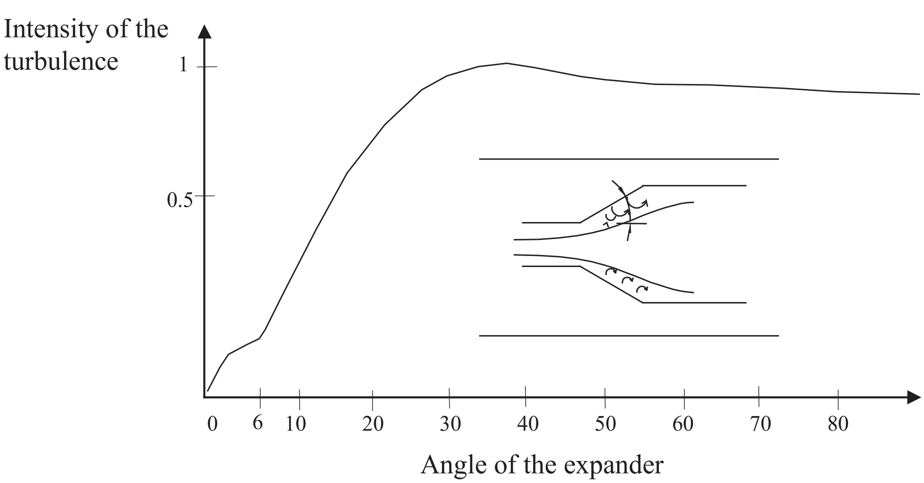


Figure 5. The intensity of the turbulence in an expansion piece depending on the angle. (As the angle becomes greater, the intensity of the turbulence increases rapidly.)

As can be seen from the examples above, it is important in pipeline design to try to avoid an assembly that combines, for instance, a 90° bend just before a butterfly valve. In the case of liquid flow there could be problems with cavitation in the bend, causing noise and damage. Disturbances from a bend can also affect the performance of the control valve. Usually the effect on the maximum capacity is insignificant, especially when the control valve is installed with pipe reducing and expansion pieces as is usually the case. The cavitation features of the valve can also change.

Particular points that should be taken into account in manifold design for butterfly valves:

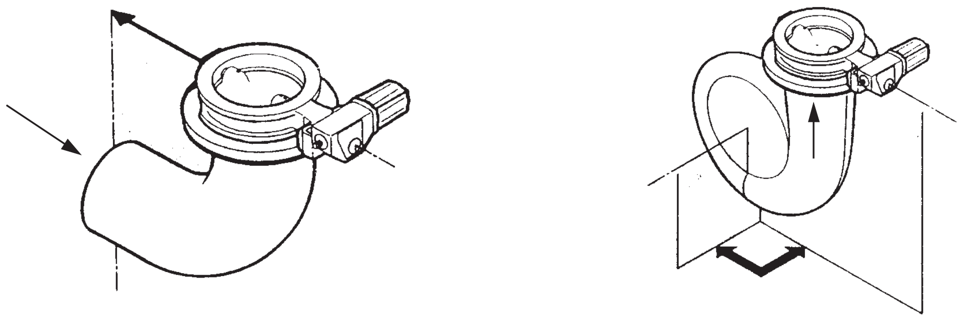


Figure 6. The installation direction of a butterfly valve after a bend or a centrifugal pump.

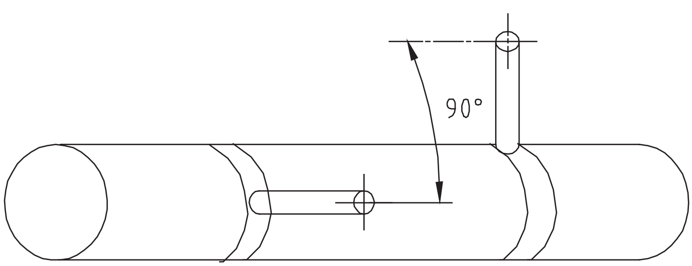


Figure 7. If two butterfly valves are installed in series, their shafts should be at a 90° angle to each other.

With fixed resistors the following points should be noted:

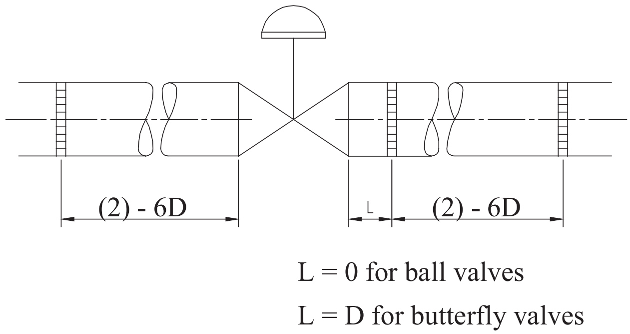
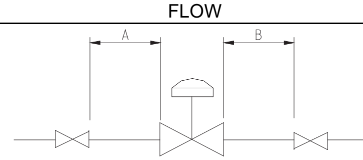
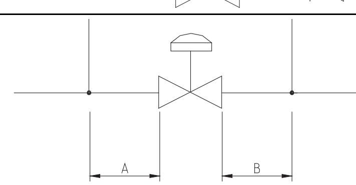
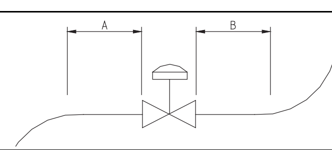
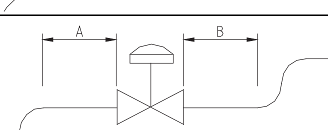


Figure 8. Installation of baffle plates, orifice plates and attenuator plates.

To ensure optimum performance from the valve, the following guidelines should be followed. This will ensure the valve behaves as predicted. In practice, however, millions of valves have been installed in all kinds of piping configurations without any problems. This is because, in most cases, the accuracy of the flow data actually contributes more to the control valve sizing accuracy than any of the issues described above. Thus the table below can be used as a guideline for building accurate laboratory equipment, but for a typical flow loop in the mill, these instructions are not critical.

SOURCE OF DISTURBANCE	LENGTHS USED IN LABORATORY TESTS (PIPE DIAMETERS)	LENGTHS FOR LESS CRITICAL FLOW CONDITIONS, BASED ON EXPERIENCE	TYPICAL SYSTEM CONFIGURATION → FLOW
Control or on/ off- valve	A=15 B=15	Ball valve: A=1 , B=1 Butterfly valve: A=2, B=2	
L's Tube bend	A=18 B=10	Ball valve: A=2 , B=2 Butterfly valve: A=3, B=3	
Long radius bends	A=18 B=10	Ball valve: A=1, B=1 Butterfly valve: A=2, B=2	
T's or Y's used as spool pieces	A=10 B=10	Ball valve: A=1, B=1 Butterfly valve: A=2, B=2	

The affect of the reducing pieces just before and after the valve are taken into account when the valve is sized with, for instance, the control valve sizing and selection program 'Nelprof'.

Sooner or later, the designer faces a situation where he must compromise on manifold design. It may not be possible to have enough straight pipeline. In such cases the designer has to consider the complete installation. Are the velocities high or low, is the pressure high or low? With high velocities and low pressure it is more important to try to avoid complicated manifold designs. If the control valve is very close to cavitation point, it is a natural solution to extend the straight pipeline downstream of the valve. It is always worth investigating whether the valve could be relocated etc. If the capacity of the control valve is at its upper limit, it is important to ensure a constant velocity profile at the valve inlet. This can be achieved with sufficient length of straight pipe upstream from the valve. These are the kind of things that the designer should consider before the final layout is decided. As a rule of thumb, it can be said that the easier the case, the greater the compromise that can be made with manifold design and still maintain control valve performance at the desired level.

List of symbols

ν	kinematic viscosity cSt
ζ	pressure head loss coefficient
ε	loss of material
ρ	density (SI units: [kg/m^3]; US units: [lb/ft^3])
α	geometry factor 0.70 for baffle plates 0.00762 * (c_v/d^2) + 0.55 for orifice plates, when (C_v/d^2) < 21 0.71 for orifice plates, when (C_v/d^2) ≥ 21
ψ	valve turning angle
ξ	pipe friction factor
δ	sliding materials constant
λ	stick-slip parameter
φ(h)	relative flow coefficient at relative opening h
ρ_0	density of water at 15.6 °C or at 60 °F 999 (SI units: [kg/m^3]) 62.36 (US units: [lb/ft^3])
ρ_1	upstream density (SI units: [kg/m^3]; US units: [lb/ft^3])
ρ_2	downstream density (SI units: [kg/m^3]; US units: [lb/ft^3])
ρ_E	effective density (SI units: [kg/m^3]; US units: [lb/ft^3])
ρ_f	density of the liquid-phase on the valve inlet side (SI units: [kg/m^3]; US units: [lb/ft^3])
ρ_g	density of the gas-phase on the valve inlet side (SI units: [kg/m^3]; US units: [lb/ft^3])
ρ_n	gas density (SI units: [kg/m^3]; US units: [lb/ft^3]) at normal conditions; pressure of 1.013 barA at 15.6°C or pressure of 14.7 psiA at 60°F
Δh_ε	relative valve travel error
Σ	K_1+K_2+K_{B1}-K_{B2}
Σ_i	head loss coefficient for an inlet fitting = K_1 + K_{B1}
ΔL_d	change in sound pressure level [dBA]
ΔL_G	valve-specific correction coefficient; usually presented in the manufacturer's noise prediction graphs along the C_v tables
Δp	pressure drop (SI units: [bar]; US units: [psi])
Δp_0	pressure difference across a closed valve
Δp_f	pressure difference across a fully open valve
Δp_h	pressure lost
Δp_m	differential pressure across the valve at the maximum designed flow rate
Δp_T	terminal pressure drop (SI units: [bar]; US units [psi])
ΔQ_ε	relative flow rate error
Δq_ε	flow rate error
ΔSPL	difference in sound pressure level [db(A)]
μ	friction coefficient
μ_0	static friction coefficient for decelerating motion
μ_{d1}	contact friction coefficient for accelerating motion
μ_{d2}	contact friction coefficient for decelerating motion
μ_{dmin}	minimum value for dynamic friction coefficient
μ_s	static friction coefficient for accelerating motion
A	flow area
a	speed of pressure pulse
A_B	plate cross-section flow area (mm^2 or in^2)
A_P	pipe area at the plate (mm^2 or in^2)
b	actuator coefficient
C_v	flow coefficient of the valve/baffle plate
Cv(1.0)	valve flow coefficient when valve is fully open
Cv(h)	valve flow coefficient at relative opening h
C_{vf}	maximum capacity coefficient of valve
d	inlet diameter of a valve (SI units: [mm]; US units: [in])
D	pipe inlet diameter (SI units: [mm]; US units: [in])
D_{in}	external diameter of inner tube (SI units: [mm]; US units [in])
D_j	diameter of jet
D_{LHD}	large hole diameter
D_{out}	internal diameter of outer shell (SI units: [mm]; US units [in])
dQ_p	the change in flow rate
D_{SHD}	small hole diameter
f	frequency
F	force

F_d	valve style modifier 1 for ball, segment, eccentric rotary plug, and single seated globe valves 0.7 for butterfly and double seated globe valves 0.5 for Q-trims and other noise attenuating trims
F_F	liquid critical pressure ratio factor
f_f	w_f/w ; the fraction of liquid mass flow rate of the total mass flow rate
f_g	w_g/w ; the fraction of gas mass flow rate of the total mass flow rate
F_k	specific heat ratio factor
F_L	valve pressure recovery factor
F_{LP}	product of the liquid pressure recovery factor of a valve with attached fittings and the piping geometry factor
f_m	actuator damping coefficient
F_m	piston force
F_P	piping geometry factor
F_R	Reynold's number factor
f_v	valve damping coefficient
g	acceleration of gravity
G	gain
G_{AC}	actuator gain
G_f	liquid specific gravity ($G_f = \rho/\rho_0 = 1$ for water at 15.6°C or at 60°F)
G_{FB}	constant feedback gain
G_g	gas specific gravity
G_{pc}	total position control gain
G_{pv}	pilot valve gain
H	relative height
h	pressure head
h_{f1}	enthalpy of saturated liquid at inlet temperature conditions
h_{f2}	enthalpy of saturated liquid at outlet pressure conditions
h_{fg2}	evaporation enthalpy at outlet pressure conditions
I	acoustic intensity
I_0	10^{-12}W/m^2
J	combined inertia moment of actuator and valve
k	1. constant 2. ratio of specific heats (dimensionless) 3. correction coefficient (see appendix J)
K_1	upstream resistance coefficient
K_2	downstream resistance coefficient
K_{B1}	inlet Bernoulli coefficient
K_{B2}	outlet Bernoulli coefficient
K_i	sum of inlet velocity head coefficients: $K_1 + K_{B1}$
L	pipe length
L_G	valve specific correction coefficient; usually presented in the manufacturer's noise prediction graphs along the C_v tables
L_P	correction coefficient for pipe wall thickness
L_{P2}	high downstream pressure correction 0 if p_2 < 30 barA (435 psiA) - (p_2 - 30 bar)/2.5 if p_2 is between 30 and 55 barA, or - (p_2 - 435 psi)/36.3 if p_2 is between 435 and 798 psiA - 10 if p_2 > 55 barA (798 psiA)
L_{pA}	calculated noise level of the attenuator plate
L_{pB}	noise level for baffle [dB(A)]
L_{pd}	predicted sound pressure level of a diffuser [dBA]
L_{PTOT}	total noise level
L_{pV}	calculated noise level of the valve[dBA]
M	1. molecular weight (dimensionless) 2. torque
Ma	Mach number
M_{air}	gas molecular weight for air = 28.96
M_d	dynamic torque
m_{red}	reduced mass for actuator piston and lever mechanism
$M_{v\mu}$	valve friction torque
n	wear exponent
N_1	0.865 (SI units: [m^3/h],[barA]) 1.0 (US units: [gpm],[psiA])
N_2	0.00214 (SI units: [mm]) 890 (US units: [in])
N_4	76 000 (SI units: [m^3/h], [mm], [centistoke]) 17 300 (US units: [gpm], [in], [centistoke])

N_5	0.00241 for SI units ([mm]) 1000 for US units ([in])
N6	27.3 for SI units 63.3 for US units
N_7	417 for SI units 1360 for US units
N_8	94.8 for SI units 19.3 for US units
N_9	2250 for SI units 7320 for US units
N_{10}	6.6 for SI units [kg/h] 1.0 for US units [lb/h]
N_{11}	0.0509 (SI units: [m^3/h], [bar], [mm^2]) 38 (US units: [gpm], [psi], [in^2])
N_{12}	12.03 for SI units 0.09325 for US units
N_{13}	353.9 for SI units 0.05103 for US units
N_{14}	91 for SI units 223 for US units
N_{15}	$1.414*10^{-3}$ for SI units ([kg/h], [m/s], [kg/m^3], [mm]) 9.821 for US units ([lb/h], [ft/s], [lb/ft^3], [in])
N_{16}	1.23 (SI units) 0.00144 (US units)
p	static pressure
p_1	upstream pressure (SI units: [barA]; US units: [psiA])
p_2	valve downstream pressure (SI units: [barA]; US units: [psiA])
p_A,p_B	pressures on piston
p_c	thermodynamical critical pressure (SI units: [barA]; US units: [psiA])
p_i - p_2	pressure drop across the baffle plate (SI units: [bar]; US units: [psi])
p_v	liquid vapour pressure at inlet temperature (SI units: [barA]; US units [psiA])
p_{VC}	vena contracta pressure
q	1. liquid volumetric flow rate (SI units: [m^3/h]; US units: [gpm]) 2. gas volumetric flow rate (SI units:[Nm^3/h], in which a cubic metre is taken at 1.013 barA and 15(C; US units: [scfh], in which a cubic foot is taken at 14.73 psiA and 60(F)
q_m	maximum flow rate required by the process
Q_p	relative flow rate
q_{stock}	measured flow rate of pulpstock flow
q_{water}	equivalent flow rate of water
r	distance from the reference point to the observation point
r_0	distance from the centreline of the vent stack to the reference point; usually r_0 = 1 m
Re	Reynolds number
S	Strouhal number
T	absolute temperature (SI units: [(K]; US units [(R])
T_1	upstream temperature (SI units: [(K]; US units: [(R])
T_2	downstream temperature; usually assumed to be same as upstream temperature (SI units: [(K]; US units: [(R])
T_c	thermodynamic critical temperature (SI units: [(K]; US units: [(R])
v	velocity (SI units: [m/s]; US units [ft/s])
v_{fg2}	outlet velocity of the liquid and vapour mixture (SI units [m/s]; US units [ft/s])
v_r	relative slide velocity
v_s	speed of sound (SI units: [m/s]; US units [ft/s]), given by equation (49)
w	1. mass flow rate (SI units: [kg/h]; US units: [lb/h]) 2. w_g + w_f ; total mass flow rate of the mixture (SI units: [kg/h]; US units: [lb/h])
x	distance covered by actuator piston
x_v	fraction of liquid flashed to vapour
Y	gas expansion factor
Z	compressibility

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