

\[ \Delta P_{\text{max}}^{(2)} = F_{L2} \left( p_{2}^f - F_{pw} \right) \]

\[ \Delta P_{\text{max}}^{(1)} = F_{L1} \left( p_{2}^f - F_{pw} \right) \]

\[ F_{L1} > F_{L2} \]

\[ C_{v1} = C_{v2} \]
HANDBOOK FOR CONTROL VALVE SIZING

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<td>$F_d$</td>
<td>Valve style modifier</td>
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<td>$F_F$</td>
<td>Liquid critical pressure ratio factor</td>
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<td>Liquid pressure recovery factor for a control valve without attached fittings</td>
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<td>$F_{γ}$</td>
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<td>$K_C$</td>
<td>Coefficient of constant cavitation</td>
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<td>$K_1$</td>
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<td>$M$</td>
<td>Molecular mass of the flowing fluid</td>
<td>kg/kmole</td>
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<td>$p_c$</td>
<td>Absolute thermodynamic critical pressure</td>
<td>bar</td>
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<td>$p_v$</td>
<td>Absolute vapour pressure of the liquid at inlet temperature</td>
<td>bar</td>
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<td>$p_{vc}$</td>
<td>Vena contracta absolute pressure</td>
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<td>$p_1$</td>
<td>Inlet absolute pressure measured at upstream pressure tap</td>
<td>bar</td>
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<td>$p_2$</td>
<td>Outlet absolute pressure measured at downstream pressure tap</td>
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<td>$Δp$</td>
<td>Pressure differential between upstream and downstream pressures</td>
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<td>$Δp_{max}$</td>
<td>Maximum allowable pressure differential for control valve sizing purposes for incompressible fluids</td>
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<td>Mass flow rate</td>
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<td>Maximum volumetric flow rate in choked condition</td>
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<td>Valve Reynolds number</td>
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<td>$u$</td>
<td>Average fluid velocity</td>
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Note - Unless otherwise specified
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<td>(v)</td>
<td>Specific volume</td>
<td>m(^3)/kg</td>
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<td>(x)</td>
<td>Ratio of pressure differential to inlet absolute pressure</td>
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<td>(x_{cr})</td>
<td>Ratio of pressure differential to inlet absolute pressure in critical conditions ((\Delta p/p_1)_{cr})</td>
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<td>(x_{FZ})</td>
<td>Coefficient of incipient cavitation</td>
<td>dimensionless</td>
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<td>(x_T)</td>
<td>Pressure differential ratio factor in choked flow condition for a valve without attached fittings</td>
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<td>(x_{TP})</td>
<td>Value of (x_T) for valve/fitting assembly</td>
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<td>(Z)</td>
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<td>(\gamma)</td>
<td>Specific heat ratio</td>
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<tr>
<td>(\rho_0)</td>
<td>Specific mass of water at 15.5°C i.e. 999 kg/m(^3)</td>
<td>kg/m(^3)</td>
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<td>(\rho_1)</td>
<td>Specific mass of fluid at (p_1) and (T_1)</td>
<td>kg/m(^3)</td>
</tr>
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<td>(\rho_r)</td>
<td>Ratio of specific mass of fluid in upstream condition to specific mass of water at 15.5°C ((\rho_1/\rho_0) - for liquids is indicated as (\rho/\rho_0))</td>
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<tr>
<td>(\nu)</td>
<td>Kinematic viscosity ((\nu = \mu/\rho))</td>
<td>Centistoke = (10^{-6}) m(^2)/s</td>
</tr>
<tr>
<td>(\mu)</td>
<td>Dynamic viscosity</td>
<td>Centipoise = (10^{-3}) Pa · s</td>
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SIZING AND SELECTION OF CONTROL VALVES

The correct sizing and selection of a control valve must be based on the full knowledge of the process.

1 - PROCESS DATA

The following data should at least be known:

a - Type of fluid and its chemical-physical and thermodynamic characteristics, such as pressure \(p\), temperature \(T\), vapour pressure \(p_v\), thermodynamic critical pressure \(p_c\), specific mass \(\rho\), kinematic viscosity \(\nu\) or dynamic viscosity \(\mu\), specific heat at constant pressure \(C_p\), specific heat at constant volume \(C_v\), specific heat ratio \(\gamma\), molecular mass \(M\), compressibility factor \(Z\), ratio of vapour to its liquid, presence of solid particles, inflammability, toxicity.

b - Maximum operating range of flow rate related to pressure and temperature of fluid at valve inlet and to \(\Delta p\) across the valve.

c - Operating conditions (normal, max., min. etc.).

d - Ratio of pressure differential available across the valve to total head loss along the process line at various operating conditions.

e - Operational data, such as:
   - maximum differential pressure with closed valve
   - stroking time
   - plug position in case of supply failure
   - maximum allowable leakage of valve in closed position
   - fire resistance
   - max. outwards leakage
   - noise limitations

f - Interface information, such as:
   - sizing of downstream safety valves
   - accessibility of the valve
   - materials and type of piping connections
   - overall dimensions, including the necessary space for disassembling and maintenance
   - design pressure and temperature
   - available supplies and their characteristics

2 - VALVE SPECIFICATION

On the ground of the above data it is possible to finalise the detailed specification of the valve (data sheet), i.e. to select:

- valve rating
- body and valve type
- body size, after having calculated the maximum flow coefficient \(C_v\) with the appropriate sizing equations
- type of trim
- materials trim of different trim parts
- leakage class
- inherent flow characteristic
- packing type
- type and size of actuator
- accessories

3 - FLOW COEFFICIENT

3.1 - FLOW COEFFICIENT “\(K_v\)”

The flow coefficient \(K_v\), is the standard flow rate which flows through a valve at a given opening, i.e. referred to the following conditions:

- static pressure drop \(\Delta p(K_v)\) across the valve of 1 bar \((10^5 \text{ Pa})\)
- flowing fluid: water at a temperature from 5 to 40° C
- volumetric flow rate in m\(^3\)/h

The value of \(K_v\) can be determined from tests using the following formula:

\[
K_v = q_v \sqrt{\frac{\Delta p(K_v)}{\Delta p}} \cdot \frac{\rho}{\rho_o} \quad (1)
\]

where:

\(\Delta p(K_v)\) is the static pressure drop of \(10^5\) Pa
\(\Delta p\) is the static pressure drop from upstream to downstream in Pa
\(\rho\) is the specific mass of fluid in kg/m\(^3\)
\(\rho_o\) is the specific mass of water in kg/m\(^3\)

The equation (1) is valid at standard conditions (see point 3.3).

3.2 - FLOW COEFFICIENT “\(C_v\)”

The flow coefficient \(C_v\), is the standard flow rate which flows through a valve at a given opening,
i.e. referred to the following conditions:

- static pressure drop \((\Delta p_{(Cv)})\) across the valve of 1 psi (6895 Pa)
- flowing fluid: water at a temperature from 40 to 100° F (5 to 40° C)
- volumetric flow rate: expressed in gpm

The value of \(C_v\) can be determined from tests using the following formula:

\[
C_v = q_v \cdot \sqrt{\frac{\Delta p_{(Cv)}}{\Delta p}} \cdot \frac{\rho}{\rho_o} \quad (2)
\]

where:

\(\Delta p_{(Cv)}\) is the static pressure drop of 1 psi (see above)
\(\Delta p\) is the static pressure drop from upstream to downstream expressed in psi.
\(\rho\) is the specific mass of the fluid expressed in lb/ft\(^3\)
\(\rho_o\) is the specific mass of the water expressed in lb/ft\(^3\)

Also the above equation (2) is valid at standard conditions as specified under point 3.3.

3.3 - STANDARD TEST CONDITIONS

The standard conditions referred to in definitions of flow coefficients \((K_v, C_v)\) are the following:

- flow in turbulent condition
- no cavitation and vaporisation phenomena
- valve diameter equal to pipe diameter
- static pressure drop measured between upstream and downstream pressure taps located as in Fig. 1
- straight pipe lengths upstream and downstream the valve as per Fig. 1
- Newtonian fluid

Note: Though the flow coefficients were defined as liquid (water) flow rates nevertheless they are used for control valve sizing both for incompressible and compressible fluids.

4 - SIZING EQUATIONS

Sizing equations allow to calculate a value of the flow coefficient starting from different operating conditions (type of fluid, pressure drop, flow rate, type of flow and installation) and making them mutually comparable as well as with the standard one.

The equations outlined in sub-clauses 4.1 and 4.2 are in accordance with the standard IEC 534-2-1

4.1 - SIZING EQUATIONS FOR INCOMPRESSIBLE FLUIDS (TURBULENT FLOW)

In general actual flow rate of a incompressible fluid through a valve is plotted in Fig. 2 versus the square root of the pressure differential \((\sqrt{\Delta p})\) under constant upstream conditions.

The curve can be splitted into three regions:

- a first normal flow region (not critical), where the flow rate is exactly proportional to \(\sqrt{\Delta p}\).
  This not critical flow condition takes place until \(p_{vc} > p_v\).
- a second semi-critical flow region, where the flow rate still rises when the pressure drop is increased, but less than proportionally to \(\sqrt{\Delta p}\).
  In this region the capability of the valve to convert the pressure drop increase into flow rate is reduced, due to the fluid vapourisation and the subsequent cavitation.
- In the third limit flow or saturation region the flow rate remains constant, in spite of further increments of \(\Delta p\).

This means that the flow conditions in vena contracta have reached the maximum evaporation rate (which depends on the upstream flow conditions) and the mean velocity is close to the sound velocity, as in a compressible fluid.

The standard sizing equations ignore the hatched area of the diagram shown in Fig. 2, thus neglecting the semi-critical flow region. This
approximation is justified by simplicity purposes and by the fact that it is not practically important to predict the exact flow rate in the hatched area; on the other hand such an area should be avoided, when possible, as it always involves vibration and noise problems as well as mechanical problems due to cavitation.

**Basic equation**
Valid for standard test conditions only.

\[
q_v = K_v \cdot \sqrt{\frac{\Delta p}{\rho / \rho_0}} \quad \text{with } q_v \text{ in m}^3/s \quad \Delta p \text{ in bar (10}^5 \text{ Pa)}
\]

\[
q_v = C_v \cdot \sqrt{\frac{\Delta p}{\rho / \rho_0}} \quad \text{with } q_v \text{ in gpm} \quad \Delta p \text{ in psi}
\]

**Note:** Simple conversion operations among the different units give the following relationship: \( C_v = 1.16 K_v \)

**Normal flow (not critical)**
It is individuated by the relationship:

\[
\Delta p < \Delta p_{\text{max}} = \left( \frac{F_{LP}}{F_p} \right)^2 \cdot \left( p_1 - F_r \cdot p_v \right)
\]

\[
C_v = \frac{q_m}{865 \cdot F_p \cdot \sqrt{\Delta p \cdot \rho_r}}
\]

\[
C_v = \frac{1.16 \cdot q_v}{F_p \cdot \sqrt{\frac{\Delta p}{\rho_r}}}
\]

---

Fig.2 - Flow rate diagram of an incompressible fluid flowing through a valve plotted versus downstream pressure under constant upstream conditions.
4.2 - SIZING EQUATIONS FOR COMPRESSIBLE FLUIDS (TURBULENT FLOW)

The Fig. 3 shows the flow rate diagram of a compressible fluid flowing through a valve when changing the downstream pressure under constant upstream conditions. The flow rate is no longer proportional to the square root of the pressure differential $\Delta p$ as in the case of incompressible fluids. This deviation from linearity is due to the variation of fluid density (expansion) from the valve inlet up to the vena contracta.

Due to this density reduction the gas must be accelerated up to a higher velocity than the one reached by an equivalent liquid mass flow. Under the same $\Delta p$ the mass flow rate of a compressible fluid must therefore be lower than the one of an incompressible fluid.

Limit flow

It is individuated by the relationship:

$$C_v = \frac{q_{m(\max)}}{\sqrt{\left(p_1 - F_p \cdot p_v\right) \cdot \rho_r}}$$

If the valve is without reducers $F_p = 1$ and $F_{LP} = F_L$

$$C_v = \frac{1.16 \cdot q_v(\max)}{F_{LP} \cdot \sqrt{\left(p_1 - F_p \cdot p_v\right) \cdot \rho_r}}$$

Such an effect is taken into account by means of the expansion coefficient $Y$ (see 5.6), whose value can change between 1 and 0.667.

Normal flow

It is individuated by the relationship

$$x < F_p \cdot x_T \quad \text{or} \quad 2/3 < Y \leq 1$$

$$C_v = \frac{q_m}{27.3 \cdot F_p \cdot Y \cdot \sqrt{x \cdot p_1 \cdot p_l}}$$

$$C_v = \frac{q_v}{2120 \cdot F_p \cdot p_1 \cdot Y \cdot \sqrt{M \cdot T_l \cdot Z \cdot x}}$$

Fig. 3 - Flow rate diagram of a compressible fluid flowing through a valve plotted versus differential pressure under constant upstream conditions.
4.3 - SIZING EQUATIONS FOR TWO-PHASE FLOWS

No standard formulas presently exist for the calculation of two-phase flow rates through orifices or control valves.

4.3.1- LIQUID/GAS MIXTURES

A first easy physical model for the calculation roughly considers separately the flows of the two phases through the valve orifice without mutual energy exchange.

Therefore:

\[ C_v = C_{v\text{, gas}} + C_{v\text{, liq}} \]

i.e. the flow coefficient is calculated as the sum of the one required for the gaseous phase and the other required for the liquid phase.

This method assumes that the mean velocities of the two phases in the vena contracta are considerably different.

A second physical model overcomes this limitation assuming that the two phases cross the vena contracta at the same velocity.

The mass flow rate of a gas (see above) is proportional to:

\[
Y \cdot \sqrt{x \cdot \rho_1} = Y \cdot \frac{x}{V_{g1}} = \sqrt{x / V_{eg}}
\]

where \( V_{eg} \) is the actual specific volume of the gas i.e.

\[
\frac{V_{g1}}{Y^2}
\]

In other terms this means to assume that the mass flow of a gas with specific volume \( V_{g1} \) is equivalent to the mass flow of a liquid with specific volume \( V_{eg} \) under the same operating conditions.

Assuming :

\[
V_e = f_g \cdot \frac{V_{g1}}{Y^2} + f_{liq} \cdot V_{liq1}
\]

where \( f_g \) and \( f_{liq} \) are respectively the gaseous and the liquid mass fraction of the mixture, the sizing equation becomes:

\[
q_m = 27.3 \cdot F_p \cdot C_v \cdot \sqrt{\frac{x \cdot \rho_1}{V_e}}
\]

When the mass fraction \( f_g \) is very small (under about 5%) better accuracy is reached using the first method.

For higher amounts of gas the second method is to be used.

4.3.2- LIQUID/VAPOUR MIXTURES

The calculation of the flow rate of a liquid mixed with its own vapour through a valve is very complex because of mass and energy transfer between the two phases.

No formula is presently available to calculate with sufficient accuracy the flow capacity of a valve in these conditions.

Such calculation problems are due to the following reasons:
- difficulties in assessing the actual quality of the mixture (i.e. the vapour mass percentage) at valve inlet. This is mostly true and important at low qualities, where small errors in quality evaluation involve significant errors in the calculation of the specific volume of the mixture (e.g. if $p_1 = 5$ bar, when the quality varies from 0.01 to 0.02 the mean specific volume of the mixture increases of 7.7%).

While the global transformation from upstream to downstream (practically isenthalpic) always involves a quality increase, the isenthalpic transformation of the mixture in thermodynamic balance between valve inlet and vena contracta may involve quality increase or decrease, depending on quality and pressure values (see diagram T/S at Fig. 4).

- some experimental data point out the fact that the process is not always in thermodynamic equilibrium (stratifications of metastable liquid and overheated steam).

- experimental data are available on liquid-vapour mixtures flowing through orifices at flow rates $10 \div 12$ times higher than the ones resulting from calculation when considering the fluid as compressible with a specific mass equal to the one at the valve inlet.

The most reliable explanation of such results is that the two phases flow at quite different velocities, though mutually exchanging mass and energy.

On the ground of the above considerations it is possible to state that:

- for low vapour quality (less than about three percent vapour by mass) at valve inlet the most suitable equation is the one obtained from the sum of the flow capacities of the two phases (at different flow velocities).

$$C_v = C_v^{\text{liq}} + C_v^{\text{vap}}$$

- for high vapour quality at valve inlet the most suitable equation is the one obtained from the hypothesis of equal velocities of the two phases, i.e. of the equivalent specific volume.

$$C_v = \frac{q_m}{27.3 \cdot F_p \cdot \sqrt{x \cdot p_1 / V_e}}$$

**Fig. 4** - Thermodynamic transformations of a water / vapour mixture inside a valve.

In the transformation shown at left side of the diagram (isenthalpic between inlet and vena contracta $V_c$) the vapour quality increases.

In the transformation at right side the quality decreases, moving from 1 to $V_c$.

In both cases the point 2 are on the same isenthalpic curve passing through the point 1, but with a higher quality.
4.4 - SIZING EQUATIONS FOR NON TURBULENT FLOW

Sizing equations of subclauses 4.1 and 4.2 are applicable in turbulent flow conditions, i.e. when the Reynolds number calculated inside the valve is higher than about 30,000.

The well-known Reynolds number:

\[ \text{Re} = \frac{\rho \cdot u \cdot d}{\mu} \]

is the dimensionless ratio between mass forces and viscous forces. When the first prevails the flow is turbulent; otherwise it is laminar.

Should the fluid be very viscous or the flow rate very low, or the valve very small, or a combination of the above conditions, a laminar type flow (or transitional flow) takes place in the valve and the \( C_v \) coefficient calculated in turbulent flow condition must be corrected by \( F_{R} \) coefficient.

Due to that above, factor \( F_{R} \) becomes a fundamental parameter to properly size the low flow control valves i.e. the valves having flow coefficients \( C_v \) from approximately 1.0 down to the microflows range.

In such valves non turbulent flow conditions do commonly exist with conventional fluids too (air, water, steam etc.) and standard sizing equations become unsuitable if proper coefficients are not used.

The currently used equations are the following:

\[
\begin{align*}
C_v & = \frac{q_m}{865 \cdot F_R \cdot \sqrt{\Delta p \cdot \rho_r}} & \text{incompressible fluid} \\
C_v & = \frac{1.16 \cdot q_v}{F_R \cdot \sqrt{\Delta p \cdot \rho_r}} & \text{compressible fluid}
\end{align*}
\]

The above equations are the same outlined in subclauses 4.1 and 4.2 for non limit flow condition and modified with the correction factor \( F_{R} \). The choked flow condition was ignored not being consistent with laminar flow.

Note the absence of piping factors \( F_p \) and \( Y \) which were defined in turbulent regime.

The effect of fittings attached to the valve is probably negligible in laminar flow condition and it is presently unknown.

In equations applicable to compressible fluid the correcting factor \( \frac{p_1 + p_2}{2} \) was introduced to account for the fluid density change.

5 - PARAMETERS OF SIZING EQUATIONS

In addition to the flow coefficient some other parameters occur in sizing equations with the purpose to identify the different flow types (normal, semi-critical, critical, limit); such parameters only depend on the flow pattern inside the valve body. In many cases such parameters are of primary importance for the selection of the right valve for a given service. It is therefore necessary to know the values of such parameters for the different valve types at full opening as well as at other stroke percentages.

Such parameters are:

- \( F_L \) - liquid pressure recovery factor for incompressible fluids
- \( K_c \) - coefficient of constant cavitation
- \( F_p \) - piping factor
- \( F_{LP} \) - combined coefficient of \( F_L \) with \( F_p \)
- \( F_F \) - liquid critical pressure ratio factor
- \( Y \) - expansion factor
- \( x_{FZ} \) - coefficient of incipient cavitation
- \( x_T \) - pressure differential ratio factor in choked condition
- \( x_{TP} \) - combined coefficient of \( F_p \) with \( x_T \)
- \( F_R \) - Reynolds number factor

5.1 - RECOVERY FACTOR \( F_L \)

The recovery factor of a valve only depends on the shape of the body and the trim. It shows the valve capability to transform the kinetic energy of the fluid in the vena contracta into pressure energy; it is so defined:

\[
F_L = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}}
\]

Since \( p_{vc} \) (pressure in vena contracta) is always lower than \( p_2 \), it is always \( F_L \leq 1 \). Moreover it is important to remark that the lower is this coefficient the higher is the valve capability to transform the kinetic energy into pressure energy (high recovery valve).

The higher this coefficient is (close to 1) the higher is the valve attitude to dissipate energy by friction rather than in vortices, with conse-
quently lower reconversion of kinetic energy into pressure energy (low recovery valve). In practice the sizing equations simply refer to the pressure drop \((p_1 - p_2)\) between valve inlet and outlet and until the pressure \(p_{vc}\) in vena contracta is higher than the saturation pressure \(p_v\) of the fluid at valve inlet, then the influence of the recovery factor is practically negligible and it does not matter whether the valve dissipates pressures energy by friction rather than in whirlpools.

The \(F_L\) coefficient is crucial when approaching to cavitation, which can be avoided selecting a lower recovery valve.

**a - Determination of \(F_L\)**

Since it is not easy to measure the pressure in the vena contracta with the necessary accuracy, the recovery factor is determined in critical conditions:

\[
F_L = \frac{1.16 q_{v(max)}}{C_v \sqrt{p_1 - 0.96 p_v}}
\]

Critical conditions are reached with a relatively high inlet pressure and reducing the outlet pressure \(p_2\) until the flow rate does not increase any longer and this flow rate is assumed as \(q_{v(max)}\).

\(F_L\) can be determined measuring only the pressure \(p_1\) and \(q_{v(max)}\).

**b - Accuracy in determination of \(F_L\)**

It is relatively easier determining the critical flow rate \(q_{v(max)}\) for high recovery valves (low \(F_L\)) than for low recovery valves (high \(F_L\)). The accuracy in the determination of \(F_L\) for values higher than 0.9 is not so important for the calculation of the flow capacity as to enable to correctly predict the cavitation phenomenon for services with high differential pressure.

**c - Variation of \(F_L\) versus valve opening and flow direction**

The recovery factor depends on the profile of velocities which takes place inside the valve body. Since this last changes with the valve opening, the \(F_L\) coefficient considerably varies along the stroke and, for the same reason, is often strongly affected by the flow direction. The Fig. 6 shows the values of the recovery factor versus the plug stroke for different valve types and the two flow directions.

---

**Fig. 5 - Comparison between two valves with equal flow coefficient but with different recovery factor, under the same inlet fluid condition, when varying the downstream pressure.** At the same values of \(C_v, p_1\) and \(p_2\) valves with higher \(F_L\) can accept higher flow rates of fluid.
5.2 - COEFFICIENT OF INCIPIENT CAVITATION $X_{FZ}$ AND COEFFICIENT OF CONSTANT CAVITATION $K_c$

When in the vena contracta a pressure lower than the saturation pressure is reached then the liquid evaporates, forming vapour bubbles. If, due to pressure recovery, the downstream pressure (which only depends on the downstream piping layout) is higher than the critical pressure in the vena contracta, then vapour bubbles totally or partially implode, instantly collapsing. This phenomenon is called cavitation and causes well known damages due to high local pressures generated by the vapour bubble implosion. Metal surface damaged by the cavitation show a typical pitted look with many micro- and macro-pits. The higher is the number of imploding bubbles, the higher are damaging speed and magnitude; these depend on the elasticity of the media where the implosion takes place (i.e. on the fluid temperature) as well as on the hardness of the metal surface (see table at Fig. 7).
Critical conditions are obviously reached gradually. Moreover the velocity profile in the vena contracta is not completely uniform, hence may be that a part only of the flow reaches the vaporization pressure. The FL recovery factor is determined in proximity of fully critical conditions, so it is not suitable to predict an absolute absence of vaporization. In order to detect the beginning of the constant bubble formation, i.e. the constant cavitation, the coefficient \( K_c \) was defined. This coefficient is defined as the ratio \( \Delta p / (p_1 - p_v) \) at which cavitation begins to appear in a water flow through the valve with such an intensity that, under constant upstream conditions, the flow rate deviation from the linearity versus \( \sqrt{\Delta p} \) exceeds 2%. Usually the beginning of cavitation is identified by the coefficient of incipient cavitation \( x_{FZ} \). The \( x_{FZ} \) coefficient can be determined by test using sound level meters or accelerometers connected to the pipe and relating noise and vibration increase with the beginning of bubble formation. Some informations on this regard are given by standard IEC 534-8-2 “Laboratory measurement of the noise generated by a liquid flow through a control valve", which the Fig. 8 was drawn from. A simple calculation rule uses the formula \( K_c = 0.8 F_L^2 \). Such a simplification is however only acceptable when the diagram of the actual flow rate versus \( \sqrt{\Delta p} \), under constant upstream conditions, shows a sharp break point between the linear/proportional zone and the horizontal one. If on the contrary the break point radius is larger (i.e. if the \( \Delta p \) at which the deviation from the linearity takes place is different from the \( \Delta p \) at which the limit flow rate is reached) then the coefficient of proportionality between \( K_c \) and \( F_L^2 \) can come down to 0.65. Since the coefficient of constant cavitation changes with the valve opening, it is usually referred to a 75% opening.

5.3 - PIPING FACTOR \( F_p \)

As already explained characteristic coefficients of a given valve type are determined in standard conditions of installation. The actual piping geometry will obviously differ from the standard one. The coefficient \( F_p \) takes into account the way that a reducer, an expander, a Y or T branch, a bend or a shut-off valve affect the value of \( C_v \) of a control valve. A calculation can only be carried out for pressure and velocity changes caused by reducers and expanders directly connected to the valve. Other effects, such as the ones caused by a change in velocity profile at valve inlet due to reducers or other fittings like a short radius bend close to the valve, can only be evaluated by specific tests. Moreover such perturbations could involve undesired effects, such as plug instability due to asymmetrical and unbalancing fluidodynamic forces. When the flow coefficient must be determined within ± 5 % tolerance the \( F_p \) coefficient must be determined by test. When estimated values are permissible the following equation may be used:

\[
F_p = \frac{1}{\sqrt{1 + \frac{\Sigma K}{0.00214 \left( \frac{C_v}{d^2} \right)^2}}}
\]

being:

\[\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}\]

Where \( C_v \) is the selected flow coefficient, \( K_1 \) and \( K_2 \) are resistance coefficient which take into account head losses due to turbulences and frictions at valve inlet and outlet, \( K_{B1} \) and/or \( K_{B2} = 1 - (d / D)^4 \) are the so called Bernoulli coefficients, which account for the pressure changes due to velocity changes due to reducers or expanders.
In case of reducers:

\[ K_1 = 0.5 \left[ 1 - \left( \frac{d}{D} \right)^2 \right] \]

In case of expanders:

\[ K_2 = 1.0 \left[ 1 - \left( \frac{d}{D} \right)^2 \right] \]

In case of the same ratio \( d/D \) for reducers and expanders:

\[ K_1 + K_2 = 1.5 \left[ 1 - \left( \frac{d}{D} \right)^2 \right] \]

5.4 - RECOVERY FACTOR WITH REDUCERS \( F_{LP} \)

Reducers, expanders, fittings and, generally speaking, any installation not according to the standard test manifold not only affect the standard coefficient (changing the actual inlet and outlet pressures), but also modify the transition point between normal and choked flow, so that \( \Delta p_{\text{max}} \) is no longer equal to \( F_L^2 (p_1 - F_F p_v) \), but it becomes:

\[ \left( \frac{F_{LP}}{F_p} \right)^2 (p_1 - F_F p_v) \]

(see Fig. 9)

It is determined by test, like for the recovery factor \( F_L \) (see point 5.1).

\[ F_{LP} = \frac{1.16 \cdot q_{v(\text{max}),LP} C_v}{\sqrt{p_1 - 0.96 p_v}} \]

When \( F_L \) is known it also can be determined by the following relationship:

\[ F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2}{0.00214 (\Sigma K)} \left( \frac{C_v}{d^2} \right)^2}} \]

Where: \( (\Sigma K)_1 = K_1 + K_{B1} \)

5.5 - LIQUID CRITICAL PRESSURE RATIO FACTOR \( F_F \)

The coefficient \( F_F \) is the ratio between the apparent pressure in vena contracta in choked con-

\[ q \]

\[ \sqrt{\Delta p} \]

\[ \Delta p_{\text{max}} = F_{LP} \sqrt{p_1 - F_F p_v} \]

\[ q_{\text{max}} = F_L C_v \]

\[ q_{\text{max}} = F_{LP} C_v \]

\[ \sqrt{\Delta p_{\text{max}}} = \frac{F_{LP}}{F_p} \sqrt{p_1 - F_F p_v} \]

Fig. 9 - Effect of reducers on the diagram of \( q \) versus \( \sqrt{\Delta p} \) when varying the downstream pressure at constant upstream pressure.
dition and the vapour pressure of the liquid at inlet temperature:

\[ F_F = \frac{p_{vc}}{p_v} \]

When the flow is at limit conditions (saturation) the flow rate equation must no longer be expressed as a function of \( \Delta p = p_1 - p_2 \) but of \( \Delta p_{vc} = p_1 - p_{vc} \) (differential pressure in vena contracta). Starting from the basic equation (at point 4.1):

\[ q_v = C_v \cdot \sqrt{\frac{p_1 - p_2}{\rho_t}} \]

and from:

\[ F_L = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}} \]

the following equation is obtained:

\[ q_v = F_L \cdot C_v \cdot \sqrt{\frac{p_1 - p_{vc}}{\rho_t}} \]

Since \( p_{vc} \) depends on the vapour pressure \( p_{vc} = F_F \cdot p_v \) therefore:

\[ q_v = F_L \cdot C_v \cdot \sqrt{\frac{p_v}{\rho_v}} \]

Supposing that at saturation conditions the fluid is a homogeneous mixture of liquid and its vapour with the two phases at the same velocity and in thermodynamic equilibrium, the following equation may be used:

\[ F_F = 0.96 - 0.28 \sqrt{\frac{p_L}{p_c}} \]

where \( p_c \) is the critical thermodynamic pressure.

5.6 - EXPANSION FACTOR \( Y \)

This coefficient allows to use for compressible fluids the same equation structure valid for incompressible fluids. It has the same nature of the expansion factor utilized in the equations of the throttling type devices (orifices, nozzles or Venturi) for the measure of the flow rate. The \( Y \)'s equation is obtained from the theory on the basis of the following hypothesis (experimentally confirmed):

- \( Y \) is a function of the fluid type, namely the exponent of the adiabatic transformation \( \gamma = \frac{c_p}{c_v} \)
- \( Y \) is function of the geometry (i.e. type) of the valve

From the first hypothesis: \( Y = 1 - ax \), therefore:

\[ q_m \alpha Y \sqrt{x} \]

A mathematic procedure allows to calculate the value of \( Y \) which makes maximum the above function (that means finding the point where the rate \( dq_m/\sqrt{x} \) becomes zero.

\[ q_m \alpha (1 - ax)^{\sqrt{x}} = \sqrt{x} - a\sqrt{x}^3 \]

By setting

\[ \frac{dq_m}{dx} = \frac{1}{2\sqrt{x}} - \frac{3a\sqrt{x}}{2} = 0 \]

\[ \frac{1}{\sqrt{x}} = 3a\sqrt{x} \quad \text{hence:} \quad x = \frac{1}{3a} \]

i.e.: \( Y = 1 - \frac{1}{3a} \cdot a = \frac{2}{3} \)

As \( Y = 1 \) when \( x = 0 \) and \( Y = 2/3 \), when the flow rate is maximum (i.e. \( x = x_T \)) the equation of \( Y \) becomes the following:

\[ Y = 1 - \frac{x}{3x_T} \]

thus taking into account also the third hypothesis. As a matter of fact \( x_T \) is an experimental value to be determined for each valve type. Finally the second hypothesis will be taken into account with an appropriate correction factor:

\[ F_Y = \gamma /1.4, \] which is the ratio between the exponent of the adiabatic transformation for the actual gas and the one for air.

The final equation becomes:

\[ Y = 1 - \frac{x}{3F_Y x_T} \]
**Fig. 10** - Values of $F_{LP}$ for valves with short type reducer at the inlet with abrupt section variation

<table>
<thead>
<tr>
<th>$Cv/d^2$ (d in mm)</th>
<th>$15 \times 10^{-3}$</th>
<th>$20 \times 10^{-3}$</th>
<th>$25 \times 10^{-3}$</th>
<th>$30 \times 10^{-3}$</th>
<th>$35 \times 10^{-3}$</th>
<th>$40 \times 10^{-3}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$d/D$</td>
<td>$F_{LP}$</td>
<td>$F_{LP}$</td>
<td>$F_{LP}$</td>
<td>$F_{LP}$</td>
<td>$F_{LP}$</td>
<td>$F_{LP}$</td>
</tr>
<tr>
<td>.25</td>
<td>.49 .58 .66 .77 .85</td>
<td>.48 .57 .66 .74 .81</td>
<td>.47 .56 .64 .71 .78</td>
<td>.47 .54 .61 .68 .74</td>
<td>.45 .53 .59 .65 .70</td>
<td>.44 .51 .57 .62 .66</td>
</tr>
<tr>
<td>.33</td>
<td>.49 .58 .66 .76 .85</td>
<td>.48 .57 .66 .74 .82</td>
<td>.48 .56 .64 .71 .78</td>
<td>.47 .54 .62 .68 .74</td>
<td>.46 .53 .59 .65 .70</td>
<td>.44 .51 .57 .62 .66</td>
</tr>
<tr>
<td>.40</td>
<td>.49 .58 .68 .77 .85</td>
<td>.48 .57 .66 .74 .82</td>
<td>.48 .56 .64 .72 .78</td>
<td>.47 .55 .62 .69 .75</td>
<td>.46 .53 .60 .66 .71</td>
<td>.45 .51 .57 .62 .67</td>
</tr>
<tr>
<td>.50</td>
<td>.49 .59 .68 .77 .86</td>
<td>.48 .58 .66 .75 .83</td>
<td>.48 .56 .65 .72 .79</td>
<td>.47 .55 .62 .69 .76</td>
<td>.46 .54 .60 .66 .72</td>
<td>.45 .52 .58 .63 .68</td>
</tr>
<tr>
<td>.66</td>
<td>.49 .59 .68 .77 .86</td>
<td>.48 .58 .66 .76 .84</td>
<td>.48 .57 .66 .74 .81</td>
<td>.48 .56 .64 .71 .78</td>
<td>.47 .55 .62 .69 .74</td>
<td>.46 .53 .60 .66 .71</td>
</tr>
<tr>
<td>.75</td>
<td>.49 .59 .69 .78 .87</td>
<td>.49 .58 .68 .76 .85</td>
<td>.49 .58 .66 .75 .83</td>
<td>.48 .57 .65 .73 .80</td>
<td>.47 .56 .63 .70 .77</td>
<td>.47 .54 .62 .68 .74</td>
</tr>
</tbody>
</table>

**Fig. 11** - Liquid critical pressure ratio factor

\[ F_c = 0.96 - 0.28 \sqrt{\frac{p_v}{p_c}} \]

$p_v = \text{Vapour pressure (bar abs.)}$

$p_c = \text{Critical pressure (bar abs.)}$

$p_v = \text{Vapour pressure (bar abs.)}$

$p_c = \text{Critical pressure (bar abs.)}$

**Fig. 12** - Critical pressure ratio factor for water

\[ F_c = 0.96 - 0.28 \sqrt{\frac{p_v}{221.2}} \]

$p_v = \text{Vapour pressure (bar abs.)}$

$p_c = \text{Critical pressure (bar abs.)}$

**Fig. 13** - Expansion factor $Y$.

The diagram is valid for a given of $F_Y$ value.
Therefore the maximum flow rate is reached when \( x = F_T \cdot x_T \) (or \( F_T \cdot x_{TP} \) if the valve is supplied with reducers); correspondently the expansion factor reaches the minimum value of 0.667.

5.7 - PRESSURE DIFFERENTIAL RATIO FACTOR IN CHOKE FLOW CONDITION \( x_T \)

As already seen the recovery factor does not occur in sizing equations for compressible fluids. Its use is unsuitable for gas and vapours because of the following physical phenomenon. Let us suppose that in a given section of the valve, under a given value of the downstream pressure \( p_2 \), the sound velocity is reached. The critical differential ratio

\[
x_{cr} = \left( \frac{\Delta p}{p_1} \right)_{cr}
\]

is reached as well, being

\[
x_{cr} = F_T^2 \left[ 1 - \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma-1}} \right]
\]

If the downstream pressure \( p_2 \) is further reduced, the flow rate still increases, as, due to the specific internal geometry of the valve, the section of the vena contracta widens transversally (it is not physically confined into solid walls). A confined vena contracta can be got for instance in a Venturi meter to measure flow rate: for such a geometry, once the sound velocity is reached for a given value of \( p_2 \), the relevant flow rate remains constant, even reducing further \( p_2 \). Nevertheless the flow rate does not unlimitedly increase, but only up to a given value of \( \Delta p / p_1 \) (to be determined by test), the so called pressure differential ratio factor in choked flow condition, \( x_T \).

5.8 - PRESSURE DIFFERENTIAL RATIO FACTOR IN CHOKE FLOW CONDITION FOR A VALVE WITH REDUCERS \( x_{TP} \)

\( x_{TP} \) is the same coefficients \( x_T \) however determined on valves supplied with reducers or installed not in according to the standard set up.

\[
x_{TP} = \frac{x_T}{\left( F_T \right)^2 \left[ 1 + \frac{x_T}{x_{TP}} \left( K_{D1} + K_{BI} \right) \left( \frac{C_d}{d^2} \right)^2 \right]}
\]

<table>
<thead>
<tr>
<th>Cd</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>25</th>
<th>30</th>
</tr>
</thead>
<tbody>
<tr>
<td>x_T</td>
<td>.40</td>
<td>.50</td>
<td>.60</td>
<td>.70</td>
<td>.80</td>
</tr>
<tr>
<td>d/D</td>
<td>.80</td>
<td>.75</td>
<td>.67</td>
<td>.60</td>
<td>.50</td>
</tr>
<tr>
<td>x_{TP}</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>F_p</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>x_{TP}</td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>F_p</td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>x_{TP}</td>
<td></td>
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<td></td>
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<tr>
<td>F_p</td>
<td></td>
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</tr>
<tr>
<td>x_{TP}</td>
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<td>F_p</td>
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<td>x_{TP}</td>
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<tr>
<td>F_p</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

Fig. 14 - Calculated values of \( x_{TP} \) and \( F_p \) for valves installed between two commercial concentric reducers (with abrupt section variation)

C_d = C_v / d^2 (d expressed in inches).

Example: For a 2” valve is: \( C_v = 80 \) and \( x_T = 0.65 \) The valve is installed in a 3” pipe between two short type reducers.

C_d = C_v / d^2 = 20 \( \qquad d / D = 2 / 3 \) = 0.67

A linear interpolation between \( x_T = 0.6 \) and \( x_T = 0.7 \) results in \( x_{TP} = 0.63 \)
Some practical values of \( x_{TP} \) versus some piping parameters and the specific flow coefficient \( C_d \) are listed in the table at Fig. 14.

### 5.9 - REYNOLDS NUMBER FACTOR \( F_R \)

The \( F_R \) factor is defined as the ratio between the flow coefficient \( C_v \) for not turbulent flow, and the corresponding coefficient calculated for turbulent flow under the same conditions of installation. If experimental data are not available, \( F_R \) can be derived by the diagrams of Fig. 15 versus the valve Reynolds number \( Re_v \) which can be determined by the following relationship:

### VALVE STYLE MODIFIER \( F_d \)

<table>
<thead>
<tr>
<th>Valve type</th>
<th>Flow direction</th>
<th>( F_d ) factor</th>
<th>( C_v ) coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>Globe, parabolic plug</td>
<td>Flow-to-open</td>
<td>0.10</td>
<td>1.00</td>
</tr>
<tr>
<td>(1-6911, 1-6951, 1-6921, 1-6981 e 1-4411)</td>
<td>Flow-to-close</td>
<td>0.20</td>
<td>1.00</td>
</tr>
<tr>
<td>Butterfly valve</td>
<td>Max. opening</td>
<td>0.20</td>
<td>0.7</td>
</tr>
<tr>
<td>1-2471, 1-2512, 1-2311</td>
<td>90°</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>60°</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cage valve</td>
<td>Number of holes</td>
<td>0.45</td>
<td>0.14</td>
</tr>
<tr>
<td>1-6933, 1-4433, 1-6971, 1-4471</td>
<td>50</td>
<td>0.45</td>
<td>0.14</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.32</td>
<td>0.10</td>
</tr>
<tr>
<td></td>
<td>200</td>
<td>0.22</td>
<td>0.07</td>
</tr>
<tr>
<td>Double seat</td>
<td>Parabolic</td>
<td>0.10</td>
<td>0.32</td>
</tr>
<tr>
<td>1-8110</td>
<td>V-port</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Between seats</td>
<td>0.10</td>
<td>0.32</td>
</tr>
<tr>
<td></td>
<td>seats</td>
<td>0.10</td>
<td>0.28</td>
</tr>
</tbody>
</table>

**Fig. 16** - Typical \( F_d \) values for PARCOL control valves. More accurate values on request.

The term under root accounts for the valve inlet velocity (velocity of approach) which, except for wide-open ball and butterfly valves, can be neglected in the enthalpic balance and taken as unity.

\( F_d \) factor ("the valve style modifier") has been introduced to account for the geometry of trim in the throttling section.

Being the \( C_v \) in \( Re_v \) equation the flow coefficient calculated by assuming turbulent flow conditions, the actual value of \( C_v \) must be found by an iterative calculation.
This data sheet was derived from IEC 60534-7 with some improvements not affecting the numbering of the original items.