HANDBOOK FOR CONTROL VALVE SIZING
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NOMENCLATURE

SIZING AND SELECTION OF CONTROL VALVES

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<th>Description</th>
<th>Units (notes)</th>
</tr>
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<tbody>
<tr>
<td>A</td>
<td>flow passage area at the actual valve stroke</td>
<td>mm²</td>
</tr>
<tr>
<td>C_v</td>
<td>flow coefficient</td>
<td>U.S. gallons/min</td>
</tr>
<tr>
<td>d</td>
<td>nominal valve size</td>
<td>mm</td>
</tr>
<tr>
<td>D</td>
<td>internal diameter of piping</td>
<td>mm</td>
</tr>
<tr>
<td>d_e</td>
<td>equivalent circular flow passage diameter</td>
<td>mm</td>
</tr>
<tr>
<td>d_h</td>
<td>hydraulic diameter of a single flow passage</td>
<td>mm</td>
</tr>
<tr>
<td>F_d</td>
<td>valve style modifier</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F_c</td>
<td>liquid critical pressure ratio factor</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F_L</td>
<td>liquid pressure recovery factor for a control valve without attached fittings</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F_LP</td>
<td>combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F_P</td>
<td>piping geometry factor</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F_R</td>
<td>Reynolds number factor</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F_T</td>
<td>specific heat ratio factor  = $\gamma / 1.4$</td>
<td>dimensionless</td>
</tr>
<tr>
<td>K_B1</td>
<td>Bernoulli coefficients for inlet and outlet of a valve with attached reducers</td>
<td>dimensionless</td>
</tr>
<tr>
<td>K_C</td>
<td>coefficient of constant cavitation</td>
<td>dimensionless</td>
</tr>
<tr>
<td>K_c</td>
<td>flow coefficient</td>
<td>m³/h</td>
</tr>
<tr>
<td>K_1</td>
<td>upstream and downstream resistance coefficients</td>
<td>dimensionless</td>
</tr>
<tr>
<td>M</td>
<td>molecular mass of the flowing fluid</td>
<td>kg/kmol</td>
</tr>
<tr>
<td>p_c</td>
<td>absolute thermodynamic critical pressure</td>
<td>bar</td>
</tr>
<tr>
<td>p_v</td>
<td>absolute vapour pressure of the liquid at inlet temperature</td>
<td>bar</td>
</tr>
<tr>
<td>p_vc</td>
<td>vena contracta absolute pressure</td>
<td>bar</td>
</tr>
<tr>
<td>p_i</td>
<td>inlet absolute pressure measured at upstream pressure tap</td>
<td>bar</td>
</tr>
<tr>
<td>p_o</td>
<td>outlet absolute pressure measured at downstream pressure tap</td>
<td>bar</td>
</tr>
<tr>
<td>Δp</td>
<td>pressure differential between upstream and downstream pressures</td>
<td>bar</td>
</tr>
<tr>
<td>Δp_max</td>
<td>maximum allowable pressure differential for control valve sizing purposes for incompressible fluids</td>
<td>bar</td>
</tr>
<tr>
<td>P_w</td>
<td>wetted perimeter of flow passage</td>
<td>mm</td>
</tr>
<tr>
<td>q_m</td>
<td>mass flow rate</td>
<td>kg/h</td>
</tr>
<tr>
<td>q_v</td>
<td>volumetric flow rate</td>
<td>m³/h</td>
</tr>
<tr>
<td>q_m(max)</td>
<td>maximum mass flow rate in choked condition</td>
<td>kg/h</td>
</tr>
<tr>
<td>q_v(max)</td>
<td>maximum volumetric flow rate in choked condition</td>
<td>m³/h</td>
</tr>
<tr>
<td>Re_v</td>
<td>valve Reynolds number</td>
<td>dimensionless</td>
</tr>
<tr>
<td>T_1</td>
<td>inlet absolute temperature</td>
<td>K</td>
</tr>
<tr>
<td>u</td>
<td>average fluid velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>v</td>
<td>specific volume</td>
<td>m³/kg</td>
</tr>
<tr>
<td>x</td>
<td>ratio of pressure differential to inlet absolute pressure</td>
<td>dimensionless</td>
</tr>
<tr>
<td>x_cr</td>
<td>ratio of pressure differential to inlet absolute pressure in critical conditions $(\Delta p / p_1)_{cr}$</td>
<td>dimensionless</td>
</tr>
<tr>
<td>x_FZ</td>
<td>coefficient of incipient cavitation</td>
<td>dimensionless</td>
</tr>
<tr>
<td>x_t</td>
<td>pressure differential ratio factor in choked flow condition for a valve without attached fittings</td>
<td>dimensionless</td>
</tr>
<tr>
<td>x_tIP</td>
<td>value of x_t for valve / fitting assembly</td>
<td>dimensionless</td>
</tr>
<tr>
<td>Y</td>
<td>expansion factor</td>
<td>dimensionless</td>
</tr>
<tr>
<td>Z</td>
<td>compressibility factor (ratio of ideal to actual inlet specific mass)</td>
<td>dimensionless</td>
</tr>
<tr>
<td>Y</td>
<td>specific heat ratio</td>
<td>dimensionless</td>
</tr>
<tr>
<td>ρ_0</td>
<td>specific mass of water at 15.5 °C i.e. 999 kg/m³</td>
<td>kg/m³</td>
</tr>
<tr>
<td>ρ_1</td>
<td>specific mass of fluid at p_1 and T_1</td>
<td>kg/m³</td>
</tr>
<tr>
<td>ρ_v</td>
<td>ratio of specific mass of fluid in upstream condition to specific mass of water at 15.5 °C $(p_1 / p_0)$</td>
<td>dimensionless</td>
</tr>
<tr>
<td>ν</td>
<td>kinematic viscosity $(ν = μ / p)$</td>
<td>centistokes = $10^{-6}$ m²/s</td>
</tr>
<tr>
<td>μ</td>
<td>dynamic viscosity</td>
<td>centipoises = $10^{-3}$ Pa·s</td>
</tr>
</tbody>
</table>
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.

0. NORMATIVE REFERENCES

- IEC 60534-2-1, Industrial process control valves – Flow capacity – Sizing under installed conditions
- IEC 60534-2-3, Industrial process control valves – Flow capacity – Test procedures
- IEC 60534-7, Industrial process control valves – Control valve data sheet
- IEC 60534-8-2, Industrial process control valves – Noise considerations – Laboratory measurement of noise generated by hydrodynamic flow through control valves

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 $C_v$, or specific heat ratio $\gamma$;
   - molecular mass $M$;
   - compressibility factor $Z$;
   - ratio of vapour to its liquid (quality);
   - presence of solid particles;
   - flammability;
   - toxicity;
   - other.

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

c. Operating conditions (normal, maximum, minimum, start-up, emergency, other).

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;
   - maximum 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 basis 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

The flow coefficient is the coefficient used to calculate the flow rate of a control valve under given conditions.

3.1 Flow coefficient \( K_v \) (metric units)

The flow coefficient \( K_v \) is the standard flow rate which flows through a valve at a given opening, referred to the following conditions:
- static pressure drop \( \Delta p \) across the valve of 1 bar \( (10^5 \text{ Pa}) \);
- flowing fluid is water at a temperature from 5 to 40° C;
- the volumetric flow rate \( q_v \) is expressed in \( \text{m}^3/\text{h} \).

The value of \( K_v \) can be determined from tests according to par. 3.3 using the following formula, valid at standard conditions only (refer to par. 3.3):

\[
K_v = q_v \cdot \frac{\Delta p_{Kv}}{\Delta p} \cdot \frac{\rho_1}{\rho_0}
\]

where:
- \( \Delta p_{Kv} \) is the static pressure drop of \( 10^5 \text{ Pa} \) [Pa];
- \( \Delta p \) is the static pressure drop from upstream to downstream [Pa];
- \( \rho_1 \) is the specific mass of flowing fluid [kg/m\(^3\)];
- \( \rho_0 \) is the specific mass of water [kg/m\(^3\)].

3.2 Flow coefficient \( C_v \) (imperial units)

The flow coefficient \( C_v \) is the standard flow rate which flows through a valve at a given opening, referred to the following conditions:
- static pressure drop \( \Delta p_{Cv} \) across the valve of 1 psi \( (6.895 \text{ Pa}) \);
- flowing fluid is water at a temperature from 40 to 100° F \( (5 \text{ to } 40^\circ \text{ C}) \);
- the volumetric flow rate \( q_v \) is expressed in gpm.

The value of \( C_v \) can be determined from tests using the following formula, valid at standard conditions only (refer to par. 3.3):

\[
C_v = q_v \cdot \frac{\Delta p_{Cv}}{\Delta p} \cdot \frac{\rho_1}{\rho_0}
\]

where:
- \( \Delta p_{Cv} \) is the static pressure drop of 1 psi [psi];
- \( \Delta p \) is the static pressure drop from upstream to downstream [psi];
- \( \rho_1 \) is the specific mass of the flowing fluid [lb/ft\(^3\)];
- \( \rho_0 \) is the specific mass of the water [lb/ft\(^3\)].

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 Figure 1;
- straight pipe lengths upstream and downstream the valve as per Figure 1;
- Newtonian fluid.

Note: Simple conversion operations among the different units give the following relationship: \( C_v \approx 1.16 \cdot K_v \).

Note: Although the flow coefficients were defined as liquid (water) flow rates, nevertheless they are used for control valve sizing both for incompressible and compressible fluids. Refer to par. 5.6 and 5.9 for more information.

Figure 1 – Standard test set up.
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 this chapter are in accordance with the standards IEC 60534-2-1 and IEC 60534-2-3.

4.1 Sizing equations for incompressible fluids (turbulent flow)

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

The curve can be split 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 $\sqrt{\Delta p}$.

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 Figure 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 vibrations and noise problems as well as mechanical problems due to cavitation.

Refer to Figure 4 for sizing equations in normal and limit flow.

4.2 Sizing equations for compressible fluids (turbulent flow)

The Figure 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 $\sqrt{\Delta p}$ 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 is 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.

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

Refer to Figure 4 for sizing equations in normal and limit flow.
Figure 2 – Flow rate diagram of an **incompressible fluid** flowing through a valve plotted versus downstream pressure under constant upstream conditions.

Figure 3 – Flow rate diagram of a **compressible fluid** flowing through a valve plotted versus differential pressure under constant upstream conditions.
**Basic equations** (valid for standard test conditions only, par. 3.3)

<table>
<thead>
<tr>
<th>Condition</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical conditions</td>
<td>[ \Delta p = p_1 - p_2 \geq \Delta p_{\text{max}} = \left( \frac{F_{LP}}{F_p} \right)^2 \left( p_1 - F_p \cdot p_v \right) ]</td>
</tr>
<tr>
<td>Normal flow (not critical) ( \Delta p &lt; \Delta p_{\text{max}} )</td>
<td>[ C_v = \frac{q_m}{865 \cdot F_p \cdot \sqrt{\Delta p} \cdot \rho_r} ]</td>
</tr>
<tr>
<td>Limit flow (critical or choked flow) ( \Delta p \geq \Delta p_{\text{max}} )</td>
<td>[ C_v = \frac{1.16 \cdot q_{v(\text{max})}}{F_{LP}} \cdot \sqrt{\frac{\rho_r}{\rho_1 - F_p \cdot p_v}} ]</td>
</tr>
</tbody>
</table>

**Sizing equations for incompressible fluids** (1)

<table>
<thead>
<tr>
<th>Condition</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical conditions</td>
<td>[ x = \frac{p_1 - p_2}{p_1} \geq F_r \cdot x_T \text{ and/or } Y = 2/3 = 0.667 ]</td>
</tr>
<tr>
<td>Normal flow (not critical) ( x &lt; F_r \cdot x_T \text{ or } 2/3 &lt; Y \leq 1 )</td>
<td>[ C_v = \frac{q_m}{27.3 \cdot F_p \cdot Y \cdot \sqrt{x \cdot p_1 \cdot \rho_1}} ]</td>
</tr>
<tr>
<td>Limit flow (critical or choked flow) ( x \geq F_r \cdot x_T \text{ and/or } Y = 2/3 = 0.667 )</td>
<td>[ C_v = \frac{q_{v(\text{max})}}{1414 \cdot F_p \cdot \rho_1} \cdot \sqrt{\frac{M \cdot T_1 \cdot \rho_r}{F_r \cdot x_T}} ]</td>
</tr>
</tbody>
</table>

**Units**

<table>
<thead>
<tr>
<th>Unit</th>
<th>Symbol</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_v )</td>
<td>( \text{m}^3/\text{h} )</td>
<td>( p_1 )</td>
</tr>
<tr>
<td>( C_v )</td>
<td>( \text{gpm} )</td>
<td>( p_2 )</td>
</tr>
<tr>
<td>( q_m )</td>
<td>( \text{kg/h} )</td>
<td>( p_3 )</td>
</tr>
<tr>
<td>( q_{v(\text{max})} )</td>
<td>( \text{m}^3/\text{h} )</td>
<td>( p_4 )</td>
</tr>
<tr>
<td>( q_{v(\text{max})} )</td>
<td>( \text{m}^3/\text{h} )</td>
<td>( p_5 )</td>
</tr>
<tr>
<td>( T )</td>
<td>( \text{K} )</td>
<td>( \rho_0 )</td>
</tr>
<tr>
<td>( M )</td>
<td>( \text{kg/kmol} )</td>
<td>( \rho_1 )</td>
</tr>
<tr>
<td>( \Delta p )</td>
<td>( \text{bar} )</td>
<td>( \rho_r )</td>
</tr>
</tbody>
</table>

Notes

1) For valve without reducers: \( F_p = 1 \) and \( F_{LP} = F_L \)
2) For valve without reducers: \( F_p = 1 \) and \( x_T = x_T \)
3) Formula with volumetric flow rate \( q_v \) [\( \text{m}^3/\text{h} \)] refers to normal conditions (1 013.25 mbar absolute and 273 K).

**Figure 4** – Basic and sizing equations both for incompressible and for compressible fluids for **turbulent flow regime** (source: IEC 60534-2-1 and IEC 60534-2-3).
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.

The following methods are based on Parcol experience: in case of doubt or lack of data, size the valve using both methods and, conservatively, assume the higher flow coefficient which results from calculation.

4.3.1 Liquid/gas mixtures

In case of valve sizing with liquid/gas mixtures without mass and energy transfer between the phases, two physical models are indicated.

The first model is applicable for volume fractions of the gas phase lower than approximately 30% and usual average fluid velocities inside the downstream pipe. It consists in the sum of the flow coefficient required for the gaseous phase and the flow coefficient required for the liquid phase:

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

This model roughly considers separately the flows of the two phases through the valve orifice without mutual energy exchange, assuming that the mean velocities of the two phases in the vena contracta are considerably different.

The second physical model overcomes the above limitation assuming that the two phases cross the vena contracta at the same velocity. It is usually applicable for volume fractions of the gas phase higher than approximately 80%.

According to formulas in Figure 4, the mass flow rate of a gas is proportional to the term:

\[ q_m + Y \cdot \sqrt{x \cdot \rho_1} \]

Defining the actual specific volume of the gas \( v_{eg} \) as:

\[ v_{eg} = \frac{v_{g1}}{\sqrt{2}} \]

the above relation can be rewritten as:

\[ q_m + Y \cdot \sqrt{\frac{x}{v_{g1}}} = \frac{x}{\sqrt{v_{eg}}} \]

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:

\[ v_e = f_g \cdot \frac{v_{g1}}{\sqrt{2}} + f_{liq} \cdot v_{liq} \]

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

\[ C_v = \frac{q_m}{27.3 \cdot F_p \cdot \sqrt{\frac{x \cdot \rho_1}{v_e}}} = \frac{q_m}{27.3 \cdot F_p \cdot \sqrt{\Delta p}} \]

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 the mass and energy transfer between the two phases.

No formulas are 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 (refer to diagram T/S in Figure 5).

- 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 10-12 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 basis of the above considerations it is possible to state that:

- for low vapour quality 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):

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

- 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 \( v_e \), as shown in par. 4.3.1.
Figure 5 – Thermodynamic transformations of a water/vapour mixture inside a valve.

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

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

In both cases the point 2 are on the same isenthalpic curve passing through point 1, but with a higher quality.

4.4 Sizing equations for non-turbulent flow

Sizing equations of par. 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 10 000 (refer to par. 5.9).

The well-known Reynolds number:

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

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 gpm down to the micro-flows 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 equations for non-turbulent flow are derived from those outlined in Figure 4 for non limit flow conditions and modified with the correction factors \( F_R \) and \( Y_R \), respectively the Reynolds number factor and the expansion factor in non-turbulent conditions.

The sizing equations for non-turbulent flow are listed in Figure 6.

The choked flow condition was ignored not being consistent with laminar flow.

Note the absence of piping factor \( F_p \) defined for turbulent flow. This because the effect of fittings attached to the valve is probably negligible in laminar flow condition and it is actually unknown.
### Sizing equations for incompressible fluids

\[
C_v = \frac{q_m}{865 \cdot F_R \cdot \sqrt{\Delta p \cdot \rho_L}}
\]

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

### Sizing equations for compressible fluids \(^{(1)}\)

\[
C_v = \frac{q_m}{67 \cdot F_R \cdot Y_R \cdot \sqrt{\Delta p \cdot (p_1 + p_2) \cdot M}}
\]

\[
C_v = \frac{q_v}{1500 \cdot F_R \cdot Y_R \cdot \sqrt{\Delta p \cdot (p_1 + p_2)}}
\]

### Expansion factor \(Y_R\)

\[
Re_v < 1000
\]
\[
Y_R = \sqrt{1 - \frac{x}{2}}
\]
\[
1000 \leq Re_v < 10000
\]
\[
Y_R = \frac{Re_v - 1000}{9000} \cdot \left(1 - \frac{x}{3 \cdot F_R \cdot x_T} - \frac{1}{2} \sqrt{1 - \frac{x}{2}}\right) + \frac{1}{2}
\]

### Reynolds number factor \(F_R\)

**laminar flow** \(Re_v < 10\)

\[
F_R = \min \left[ \frac{0.026}{F_L} \cdot \sqrt{n \cdot Re_v} \right]_{1.00}
\]

**transitional flow** \(10 \leq Re_v < 10000\)

\[
F_R = \min \left[ 1 + \left( \frac{0.33 \cdot F_L^{\frac{1}{3}}}{n^{\frac{1}{3}}} \right) \cdot \log \left( \frac{Re_v}{10000} \right) \right]_{1.00}
\]

### Trim style constant \(n\)

**full size trim**

\[
\left( \frac{C_v}{d^2} \right) \geq 0.016 \quad n = \frac{467.3}{\left( \frac{C_v}{d^2} \right)^{\frac{3}{2}}}
\]

**reduced trim**

\[
\left( \frac{C_v}{d^2} \right) < 0.016 \quad n = 1 + 127 \cdot \left( \frac{C_v}{d^2} \right)^{\frac{3}{2}}
\]

### Units

- \(C_v\) [gpm]
- \(q_m\) [kg/h]
- \(q_v\) [m³/h] for incompressible fluids
- \(p_1\) [bar a]
- \(p_2\) [bar a]
- \(q_v\) [Nm³/h] for compressible fluids
- \(p_1\) [-] refer to par. 5.9
- \(p_2\) [-] refer to par. 5.9
- \(F_R\) [-] refer to par. 5.9
- \(Y_R\) [-] refer to par. 5.9
- \(M\) [kg/kmol]
- \(T\) [K]
- \(\Delta p\) [bar]
- \(d\) [mm]

### Notes

1) Formula with volumetric flow rate \(q_v\) [Nm³/h] refers to normal conditions \((1.013.25 \text{ mbar absolute and } 273 \text{ K})\).

For use with volumetric flow rate \(q_v\) [Sm³/h] in standard conditions \((1.013.25 \text{ mbar absolute and } 288.6 \text{ K})\), replace constant 1500 with 1590.

---

**Figure 6** – Sizing equations both for incompressible and for compressible fluids for **non-turbulent flow regime** (source: IEC 60534-2-1).
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);
- $x_F/z$ coefficient of incipient cavitation;
- $K_c$ coefficient of constant cavitation;
- $F_p$ piping geometry factor;
- $F_{Lp}$ combined coefficient of $F_L$ with $F_p$;
- $F_v$ liquid critical pressure ratio factor;
- $Y$ expansion factor (for compressible fluids);
- $x_T$ pressure differential ratio factor in choked condition;
- $x_{TP}$ combined coefficient of $F_P$ with $x_T$;
- $F_R$ Reynolds number factor;
- $F_d$ valve style modifier.

5.1 Liquid pressure 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 defined as follows:

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

Since the pressure in vena contracta $p_{vc}$ 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 consequently 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 $p_s$ 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:

$$F_L = \frac{1.16 \cdot q_{v(max)}}{C_v \cdot \sqrt{p_1 - 0.96 \cdot p_y}}$$

The above formula is valid using water as test fluid. 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 Figure 7 shows the values of the recovery factor versus the plug stroke for different valve types and the two flow directions.

![Figure 7 – Typical $F_L$ values versus $C_v$, % and flow direction for different PARCOL valve types.](image)
Figure 8 – 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.

Figure 9 – Pressure drop comparison between single stage (venturi nozzle) and multistage multipath ($Limiphon^{TM}$ trim) on liquid service using CFD analysis.
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 know damages due to high local pressures generated by the vapour bubbly 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 ad on the hardness of the metal surface (see table in Figure 10).

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 $F_L$ 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 $K_c$ is defined as:

$$ K_c = \frac{\Delta p}{p_i - p_v} $$

It identifies where the 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%. A simple calculation rule uses the formula:

$$ K_c = 0.80 \cdot F_L^2 $$

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.

<table>
<thead>
<tr>
<th>Index of resistance to cavitation</th>
</tr>
</thead>
<tbody>
<tr>
<td>stellite gr.6</td>
</tr>
<tr>
<td>chrome plating</td>
</tr>
<tr>
<td>17-4 PH H900</td>
</tr>
<tr>
<td>AISI 316/304</td>
</tr>
<tr>
<td>monel 400</td>
</tr>
<tr>
<td>gray cast iron</td>
</tr>
<tr>
<td>chrome-molybdenum alloyed steels (5% chrome)</td>
</tr>
<tr>
<td>carbon steels (WCB)</td>
</tr>
<tr>
<td>bronze (B16)</td>
</tr>
<tr>
<td>nickel plating</td>
</tr>
<tr>
<td>pure aluminium</td>
</tr>
</tbody>
</table>

**Figure 10** – Cavitation resistance of some metallic materials referred to stainless steels AISI 304/316. Values between brackets are listed for qualitative comparison only.
5.3 Piping geometry factor $F_p$

According to par. 3.3, the flow coefficients of a given valve type are determined under 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 fluid dynamic 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:

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

where:
- $C_v$ is the selected flow coefficient [gpm];
- $d$ is the nominal valve size [mm];
- $\Sigma K$ is defined as $\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}$, with:
  - $K_1$ and $K_2$ are resistance coefficient which take into account head losses due to turbulences and frictions respectively at valve inlet and outlet (see Figure 12);
  - $K_{B1}$ and $K_{B2}$ are the so called Bernoulli coefficients, which account for the pressure changes due to velocity changes due to reducers or expanders, respectively at valve inlet and outlet (see Figure 12); in case of the same ratio $d/D$ for reducer and expander, their sum is null;
- $D$ is the internal diameter of the piping [mm].

### Table

<table>
<thead>
<tr>
<th>Type of Fitting</th>
<th>Coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet reducer</td>
<td>$K_1 = 0.5 \cdot \left[1 - \left( \frac{d}{D_1} \right)^2 \right]^2$</td>
</tr>
<tr>
<td>Outlet expander</td>
<td>$K_2 = 1.0 \cdot \left[1 - \left( \frac{d}{D_2} \right)^2 \right]^2$</td>
</tr>
<tr>
<td>In case of the same ratio $d/D$</td>
<td>$K_1 + K_2 = 1.5 \cdot \left[1 - \left( \frac{d}{D} \right)^2 \right]^2$</td>
</tr>
<tr>
<td>In case of different ratio $d/D$</td>
<td>$K_{B1} = 1 - \left( \frac{d}{D_1} \right)^4$</td>
</tr>
<tr>
<td></td>
<td>$K_{B2} = 1 - \left( \frac{d}{D_2} \right)^4$</td>
</tr>
</tbody>
</table>

5.4 Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings $F_{LP}$

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 $\Delta P_{max}$ is no longer equal to $F_p (p_1 - F_F p_\nu)$, but it becomes:

$$ \Delta P_{max} = \left( \frac{F_{LP}}{F_p} \right)^2 (\rho_i - F_F \cdot p_\nu) $$

As for the recovery factor $F_L$, the coefficient $F_{LP}$ is determined by test (refer to par. 5.1.a):

$$ F_{LP} = \frac{1.16 \cdot q_{v(max)}/L}{C_v \cdot \sqrt{\rho_i - 0.96 \cdot p_\nu}} $$

The above formula is valid using water as test fluid. When $F_L$ is known, it can also be determined by the following relationship:

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

where $(\Sigma K)_h = K_1 + K_{B1}$ is the velocity head loss coefficient of the fitting upstream the valve, as measured between the upstream pressure tap and the control valve body inlet. For detail of terms refer to par. 5.3 and Figure 12.

Figure 12 – Resistance and Bernoulli coefficients.
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 condition 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 the differential pressure across the valve ($\Delta p = p_1 - p_2$), but as function of the differential pressure in vena contracta ($\Delta p_{vc} = p_1 - p_{vc}$).

Starting from the basic equation (refer to par. 4.1):

$$q_v = C_v \cdot \sqrt{\frac{p_1 - p_2}{\rho_r}}$$

And from:

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

The following equation is obtained:

$$q_v = F_L \cdot C_v \cdot \sqrt{\frac{p_1 - p_{vc}}{\rho_r}}$$

Expressing the differential pressure in vena contracta $p_{vc}$ as function of the vapour pressure ($p_{vc} = F_F \cdot p_v$), the flow rate can be calculated as:

$$q_v = F_L \cdot C_v \cdot \sqrt{\frac{p_1 - F_F \cdot p_v}{p_{vc}}}$$

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:

$$F_F = 0.96 - 0.28 \cdot \frac{p_v}{p_c}$$

where $p_c$ is the fluid critical thermodynamic pressure. Refer to Figure 14 for plotted curves of generic liquid and for water.

![Figure 13 - Effect of reducers on the diagram of q versus $\sqrt{\Delta p}$ when varying the downstream pressure at constant upstream pressure.](image-url)
5.6 Expansion factor $Y$ and specific heat ratio factor $F_\gamma$

The expansion factor $Y$ 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):

1. $Y$ is a linear function of $x = \frac{\Delta p}{p_1}$;
2. $Y$ is function of the geometry (i.e. type) of the valve;
3. $Y$ is a function of the fluid type, namely the exponent of the adiabatic transformation $\gamma = \frac{c_p}{c_v}$.

From the first hypothesis:

$$Y = 1 - a \cdot x$$

therefore:

$$q_m \div Y \cdot \sqrt{x}$$

A mathematic procedure allows to calculate the value of $Y$ which makes maximum the above function (this means finding the point where the rate $dq_m/dx$ becomes zero):

$$q_m \div (1 - a \cdot x) \cdot \sqrt{x} = \sqrt{x} - a \cdot x^3$$

By setting:

$$\frac{dq_m}{dx} = \frac{1}{2} \sqrt{x} - \frac{3 \cdot a \cdot \sqrt{x}}{2} = 0$$

$$\frac{1}{\sqrt{x}} = 3 \cdot a \cdot \sqrt{x}$$

hence: $$x = \frac{1}{3 \cdot a}$$

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

As $Y = 1$ when $x = 0$ and $Y = 2/3 = 0.667$, when the flow rate is maximum (i.e. $x = x_T$) the equation of $Y$ becomes the following:

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

thus taking into account also the second hypothesis. In fact, $x_T$ is an experimental value to be determined for each valve type.

Finally the third hypothesis will be taken into account with an appropriate correction factor, the specific heat ratio factor $F_\gamma$, which is the ratio between the exponent of the adiabatic transformation for the actual gas and the one for air:

$$F_\gamma = \frac{\gamma}{1.4}$$

The final equation becomes:

$$Y = 1 - \frac{x}{3 \cdot F_\gamma \cdot x_T}$$

Therefore the maximum flow rate is reached when:

$$x = F_\gamma \cdot x_T$$

(or $x = F_\gamma \cdot x_{Tp}$ if the valve is supplied with reducers).

Correspondently the expansion factor reaches the minimum value of 0.667.
Figure 14 – Liquid critical pressure ratio factor $F_F$ (for a generic liquid above, for water under).

Figure 15 – Expansion factor $Y$. The diagram is valid for a given $F_Y$ value.
5.7 Pressure differential ratio factor in choked flow condition $x_T$

The recovery factor $F_L$ does not occur in sizing equations for compressible fluids. Its use is unsuitable for gas and vapours because of the following physical phenomenon.

Assume 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:

$$x_{cr} = F_L^2 \cdot \left[ 1 - \left( \frac{2}{\gamma + 1} \right)^{\gamma \over \gamma-1} \right]$$

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, $x_T$.

Although some relationships between $x_T$ and $F_L$ are available, reliable values of $x_T$ must be obtained only by tests, as the internal geometry of body governs either the head losses inside the body and the expansion mode of vena contracta. If vena contracta is free to expand the relationship between $x_T$ and $F_L$ may be approximately the following:

$$x_T \simeq 0.85 \cdot F_L^2$$

On the contrary, if vena contracta is fully confined by the inner body walls (Venturi shape) and the pressure losses inside the body are negligible, $x_T$ tends to be coincident with $x_{cr}$.

<table>
<thead>
<tr>
<th>Valve type</th>
<th>Trim type</th>
<th>Flow direction</th>
<th>$F_L$</th>
<th>$x_T$</th>
<th>$F_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Globe, single port</td>
<td>Contoured plug (linear and equal percentage)</td>
<td>Open</td>
<td>0.90</td>
<td>0.72</td>
<td>0.46</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Close</td>
<td>0.80</td>
<td>0.55</td>
<td>1.00</td>
</tr>
<tr>
<td>Globe, angle</td>
<td>Contoured plug (linear and equal percentage)</td>
<td>Open</td>
<td>0.90</td>
<td>0.72</td>
<td>0.46</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Close</td>
<td>0.80</td>
<td>0.65</td>
<td>1.00</td>
</tr>
<tr>
<td>Butterfly, eccentric shaft</td>
<td>Offset seat ($70^\circ$)</td>
<td>Either</td>
<td>0.67</td>
<td>0.35</td>
<td>0.57</td>
</tr>
<tr>
<td>Globe and angle</td>
<td>Multistage, multipath</td>
<td>2</td>
<td>0.97</td>
<td>0.812</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>0.99</td>
<td>0.888</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4</td>
<td>0.99</td>
<td>0.925</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>0.99</td>
<td>0.950</td>
<td>-</td>
</tr>
</tbody>
</table>

$F_d$ at full rated stroke (source: IEC 60534-2-1).
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.

$$F_R = \frac{C_v_{\text{non-turbulent}}}{C_v_{\text{turbulent}}}$$

The $F_R$ factor is determined by tests and can be calculated with the formulas listed in table of Figure 6. It is function of the valve Reynolds number $Re_v$ which can be determined by the following relationship:

$$Re_v = \frac{0.076 \cdot F_d \cdot q_v}{\sqrt{F_L \cdot C_v}} \cdot \sqrt[4]{\frac{F_L^2 \cdot C_v^2}{0.00214 \cdot D^4} + 1}$$

The term under root takes into account for the valve inlet velocity (the so called "velocity of approach"). Except for wide open ball and butterfly valves, it can be neglected in the enthalpic balance and taken as unity.

Since the $C_v$ in $Re_v$ equation is the flow coefficient calculated by assuming turbulent flow conditions, the actual value of $C_v$ must be found by an iterative calculation.

5.10 Valve style modifier $F_d$

The $F_d$ factor is the valve style modifier and takes into account for the geometry of trim in the throttling section. It can determined by tests or, in first approximation, by means of its definition:

$$F_d = \frac{d_H}{d_o}$$

where:
- $d_H$ is the hydraulic diameter of a single flow passage [mm];
- $d_o$ is equivalent circular flow passage diameter [mm].

In detail, the hydraulic diameter is defined as four times the "hydraulic radius" of the flow passage at the actual valve stroke:

$$d_H = \frac{4 \cdot A}{P_w}$$

while the equivalent circular flow passage diameter is:

$$d_o = \sqrt{\frac{4 \cdot A}{\pi}}$$

where:
- $A$ is the flow passage area at the actual valve stroke [mm$^2$];
- $P_w$ is the wetted perimeter of flow passage (it is equal to 1 for circular holes) [mm].

The simpler is the geometry of flow pattern in the throttling section the more reliable is the theoretical evaluation of $F_d$ factor with the above formulas. Knowing the value of the $F_d$ factor is especially important in the following cases:

- **micro-flow valves**: where it is frequent the presence of laminar flow, and then the use of the $F_R$ factor. In these valves, characterised by flute, needle or other type of plug, it is important to keep in mind that the theoretical evaluation of $F_d$ factor is highly dependent by the annular gap between plug and seat. In these cases, the theoretical evaluation of $F_d$ factor is reliable only for flow coefficient $C_v$ higher than 0.1.

- **low-noise valves**: the $F_d$ factor defines, in particular formulations, the flow diameter and then the predominating frequency of the acoustic spectrum produced by the valve. Its knowledge is then very important the estimation of the noise produced by the valve during operation. As an example, the valves with multi-drilled cage trims have a $F_d$ factor equal to:

$$F_d = \frac{1}{\sqrt{N_o}}$$

where $N_o$ is the number of drilled holes in parallel.

It follows that, higher is the value of $N_o$, smaller are the holes at same flow coefficient $C_v$ and lower is the $F_d$ factor, which means lower generated noise.

For more information about noise in control valves, refer to Parcol Technical Bulletin “Noise Manual”.


## Control Valve Datasheet (source: IEC 60534-7)

<table>
<thead>
<tr>
<th>SERIAL NO.</th>
<th>REV. NO.</th>
<th>T/F</th>
<th>N等于2</th>
<th>N等于4</th>
<th>T/F</th>
<th>N等于2</th>
<th>N等于4</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Location</td>
<td>57</td>
<td>MFR</td>
<td>Model</td>
<td></td>
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</tr>
<tr>
<td>2</td>
<td>Service</td>
<td>58</td>
<td>Pneumatic</td>
<td>diaphragm</td>
<td>Piston</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Haz. area class</td>
<td>59</td>
<td>Style</td>
<td>sprg. return</td>
<td>double act.</td>
<td></td>
<td></td>
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<td>4</td>
<td>Ambient temp min</td>
<td>60</td>
<td>Size</td>
<td>min</td>
<td>max</td>
<td>cm²</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Allowable sound pressure level</td>
<td>61</td>
<td>Travel/angle</td>
<td></td>
<td></td>
<td></td>
<td></td>
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**Remarks:**

- Project
- Dig. ref. No.
- Plant
- Mt. req. No.
- Rev. Date
- Name
- Date
- Name
- P.O. No.
- Item No.
- Qty

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Figure 17 – Control valve datasheet (source: IEC 60534-7).
Parcol test facility for Cv testing with water in accordance with IEC 60534-2-3
(max DN 12”, max flow rate 650 m³/h @ 6.5 bar)

Flow coefficient tests on DN 12” 1-2483 series drilled wings butterfly control valve

Flow coefficient tests on DN 8” 1-6943 series VeGA globe control valve

Flow simulations on different Parcol control valves using CFD analysis

From left: 1-4800 series, specialty control valves for urea service; 1-5700 series, pressure reducing and desuperheating stations; 1-7000 series, tandem plug control valves for high pressure applications