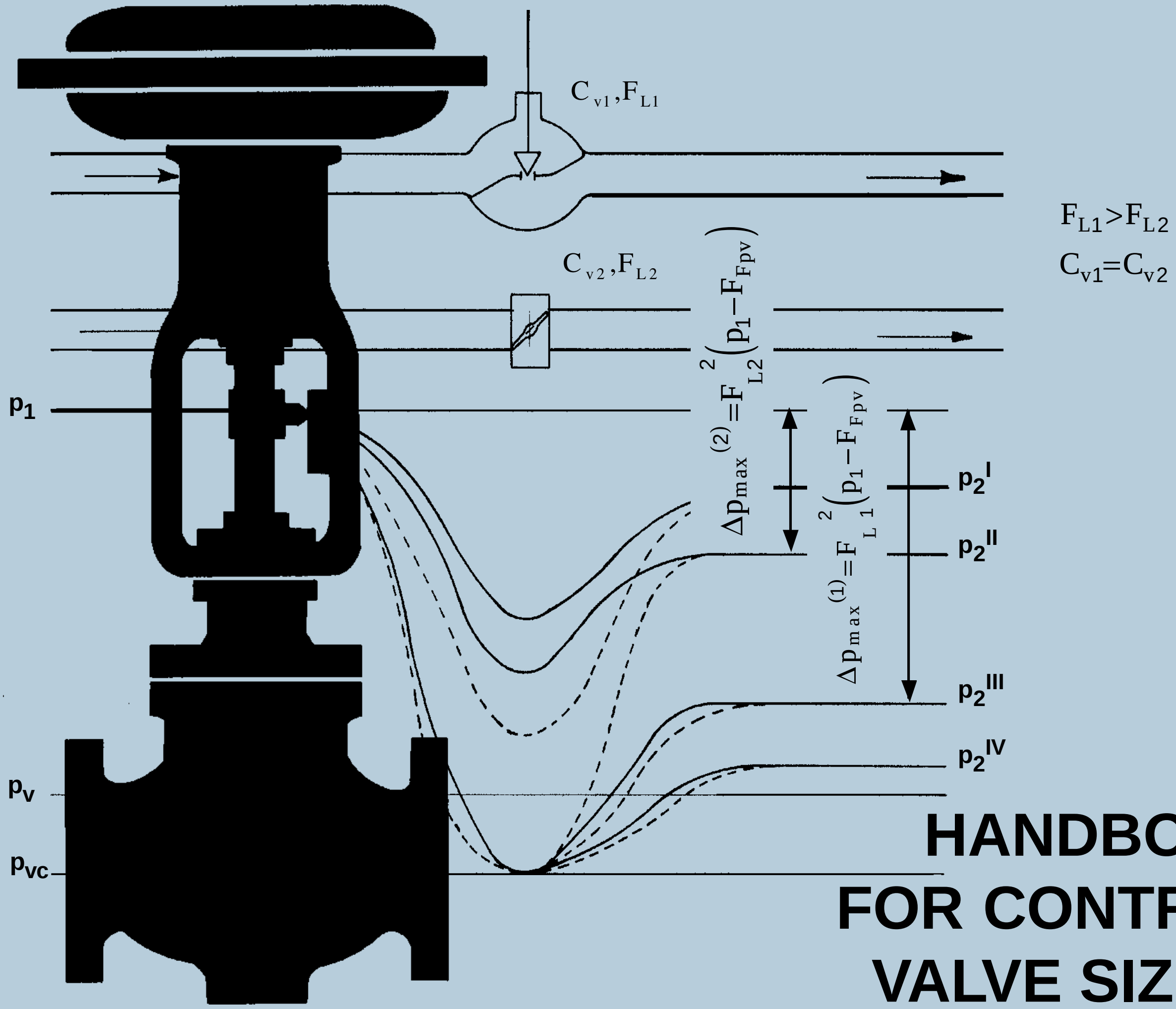
Bulletin 1-I



HANDBOOK FOR CONTROL VALVE SIZING

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CONTENTS

Symbols	Description	Units (note)
C_d	Specific flow coefficient = C_v/d^2	various
C_v	Flow coefficient	U.S. gallons/min
d	Nominal valve size	mm
D	Internal diameter of piping	mm
F_d	Valve style modifier	dimensionless
F_F	Liquid critical pressure ratio factor	dimensionless
F_L	Liquid pressure recovery factor for a control valve without attached fittings	dimensionless
F_{LP}	Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings	dimensionless
F_P	Piping geometry factor	dimensionless
F_R	Reynolds number factor	dimensionless
F_γ	Specific heat ratio factor = $\gamma/1.4$	dimensionless
K_{B1} and K_{B2}	Bernoulli coefficients for inlet and outlet of a valve with attached reducers	dimensionless
K_c	Coefficient of constant cavitation	dimensionless
K_v	Flow coefficient	m^3/h
K_1 and K_2	Upstream and downstream resistance coefficients	dimensionless
M	Molecular mass of the flowing fluid	kg/kmole
p_c	Absolute thermodynamic critical pressure	bar
p_v	Absolute vapour pressure of the liquid at inlet temperature	bar
p_{vc}	Vena contracta absolute pressure	bar
p_1	Inlet absolute pressure measured at upstream pressure tap	bar
p_2	Outlet absolute pressure measured at downstream pressure tap	bar
Δp	Pressure differential between upstream and downstream pressures	bar
Δp_{max}	Maximum allowable pressure differential for control valve sizing purposes for incompressible fluids	bar
q_m	Mass flow rate	kg/h
q_v	Volumetric flow rate	m^3/h
$q_{m(max)}$	Maximum mass flow rate in choked condition	kg/h
$q_{v(max)}$	Maximum volumetric flow rate in choked condition	m^3/h
Re_v	Valve Reynolds number	dimensionless
T_1	Inlet absolute temperature	K
u	Average fluid velocity	m/s

Note - Unless otherwise specified

Symbols	Description	Units
v	Specific volume	m^3/kg
x	Ratio of pressure differential to inlet absolute pressure	dimensionless
x_{cr}	Ratio of pressure differential to inlet absolute pressure in critical conditions $(\Delta p/p_1)_{cr}$	dimensionless
x_{FZ}	Coefficient of incipient cavitation	dimensionless
x_T	Pressure differential ratio factor in choked flow condition for a valve without attached fittings	dimensionless
x_{TP}	Value of x_T for valve/fitting assembly	dimensionless
Y	Expansion factor	dimensionless
Z	Compressibility factor - ratio of ideal to actual inlet specific mass	dimensionless
γ	Specific heat ratio	dimensionless
ρ_o	Specific mass of water at 15.5°C i.e. 999 kg/m^3	kg/m^3
ρ_1	Specific mass of fluid at p_1 and T_1	kg/m^3
ρ_r	Ratio of specific mass of fluid in upstream condition to specific mass of water at 15.5°C (ρ_1/ρ_o - for liquids is indicated as ρ/ρ_o)	dimensionless
ν	Kinematic viscosity ($\nu = \mu/\rho$)	Centistoke = 10^{-6} m^2/s
μ	Dynamic viscosity	Centipoise = 10^{-3} Pa · s

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 "ρ", kinematic viscosity "ν" or dynamic viscosity "μ", specific heat at constant pressure "C_p", specific heat at constant volume "C_v", specific heat ratio "γ", 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 Δp 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 Kv, is the standard flow rate which flows through a valve at a given opening, i.e. referred to the following conditions:

- static pressure drop ($\Delta p_{(Kv)}$) across the valve of 1 bar (10^5 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:

$$K_v = q_v \sqrt{\frac{\Delta p_{(Kv)}}{\Delta p} \cdot \frac{\rho}{\rho_o}} \qquad (1)$$

where:

$\Delta p_{(Kv)}$ is the static pressure drop of 10^5 Pa
Δp is the static pressure drop from upstream to downstream in Pa
ρ is the specific mass of fluid in kg/m^3
ρ_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 ÷ 40° C)
- volumetric flow rate: expressed in gpm

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

$$C_v = q_v \cdot \sqrt{\frac{\Delta p_{(Cv)}}{\Delta p} \cdot \frac{\rho}{\rho_o}} \qquad (2)$$

where:

$\Delta p_{(Cv)}$ is the static pressure drop of 1 psi (see above)
Δp is the static pressure drop from upstream to downstream expressed in psi.
ρ is the specific mass of the fluid expressed in lb/ft^3
ρ_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 vaporisation 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 Fig. 2, thus neglecting the semi-critical flow region. This

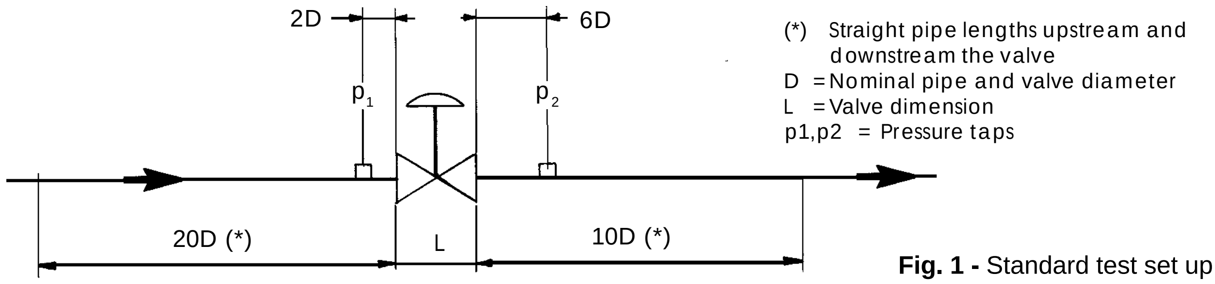


Fig. 1 - Standard test set up

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.

$$q_v = K_v \cdot \sqrt{\frac{\Delta p}{\rho / \rho_o}}$$ with q_v in m^3/s, Δp in bar (10^5 Pa)

$$q_v = C_v \cdot \sqrt{\frac{\Delta p}{\rho / \rho_o}}$$ with q_v in gpm, Δp 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:

$$\Delta p < \Delta p_{max} = \left(\frac{F_{LP}}{F_P}\right)^2 \cdot \left(p_1 - F_F \cdot p_v\right)$$

$$C_V = \frac{q_m}{865 \cdot F_P \cdot \sqrt{\Delta p \cdot \rho_r}}$$

$$C_V = \frac{1.16 \cdot q_v}{F_P \cdot \sqrt{\frac{\Delta p}{\rho_r}}}$$

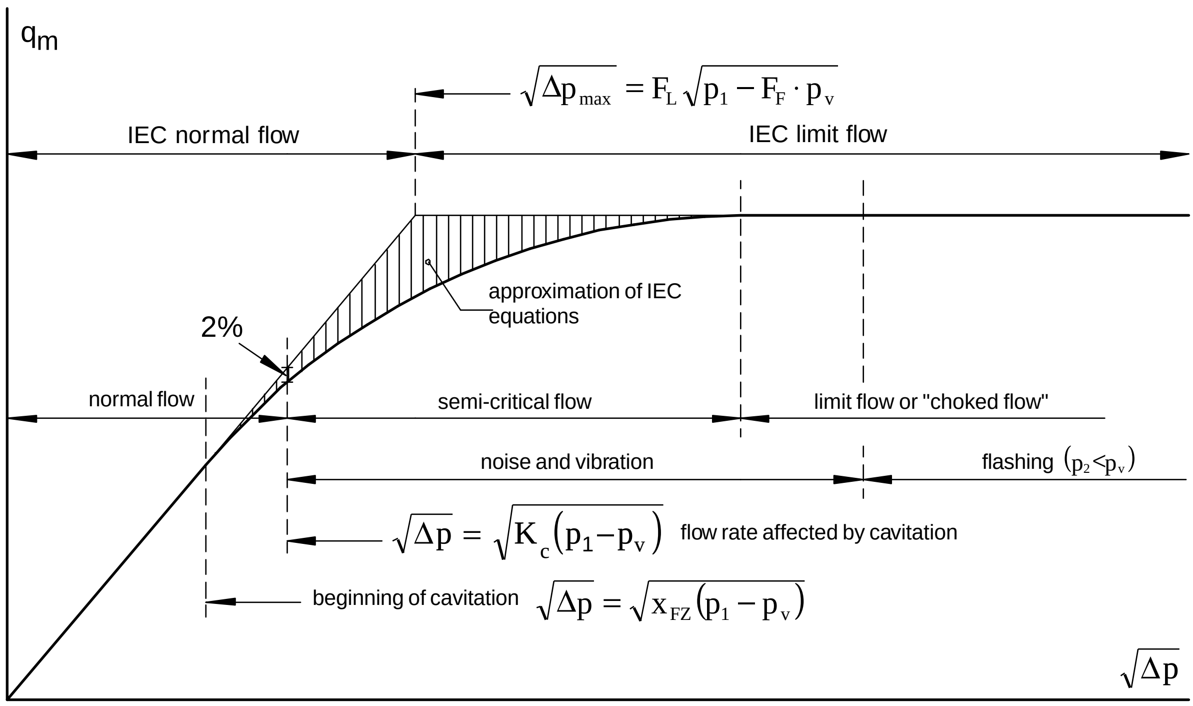


Fig.2 - Flow rate diagram of an incompressible fluid flowing through a valve plotted versus downstream pressure under constant upstream conditions.

Limit flow
It is individuated by the relationship:

$$\Delta p \geq \Delta p_{max} = \left(\frac{F_{LP}}{F_p}\right)^2 \cdot \left(p_1 - F_F \cdot p_v\right)$$

$$C_v = \frac{q_{m(max)}}{865 \cdot F_{LP} \cdot \sqrt{(p_1 - F_F p_v)\rho_r}}$$

$$C_v = \frac{1.16 \cdot q_{v(max)}}{F_{LP} \cdot \sqrt{\frac{(p_1 - F_F p_v)}{\rho_r}}}$$

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

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 $\sqrt{\Delta p}$ 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 must be accelerated up to a higher velocity than the one reached by an equivalent liquid mass flow. Under the same Δp 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 (see 5.6), whose value can change between 1 and 0.667.

Normal flow
It is individuated by the relationship

$x < F_\gamma \cdot x_T$ or $2/3 < Y \leq 1$

$$C_v = \frac{q_m}{27.3 \cdot F_p \cdot Y \cdot \sqrt{x \cdot p_1 \cdot \rho 1}}$$

$$C_v = \frac{q_v}{2120 \cdot F_p \cdot p_1 \cdot Y} \cdot \sqrt{\frac{M \cdot T_1 \cdot Z}{x}}$$

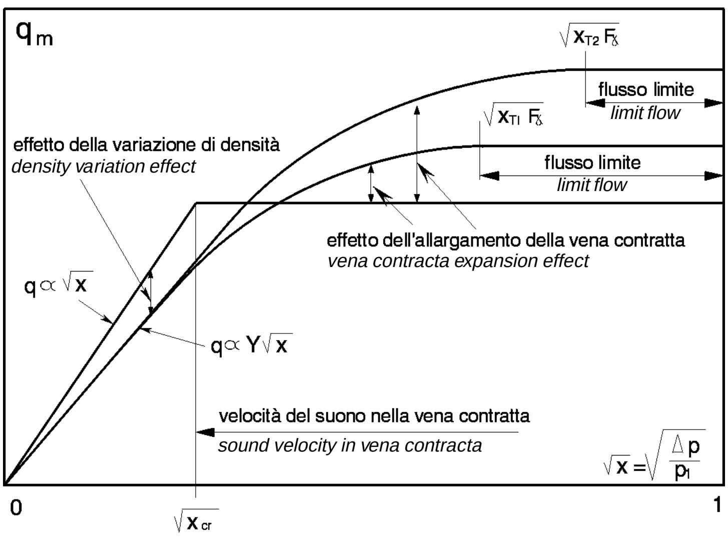


Fig.3 - Flow rate diagram of a compressible fluid flowing through a valve plotted versus differential pressure under constant upstream conditions.

Limit flow
It is individuated by the relationship

$x \geq F_\gamma \cdot x_{TP}$ and/or $Y = 2/3 = 0.667$

$$C_v = \frac{q_{m(max)}}{18.2 \cdot F_p \cdot \sqrt{F_\gamma \cdot x_{TP} \cdot p_1 \cdot \rho_1}}$$

$$C_v = \frac{q_{v(max)}}{1414 \cdot F_p \cdot p_1} \cdot \sqrt{\frac{M \cdot T_1 \cdot Z}{F_\gamma \cdot x_{TP}}}$$

where: q_v is expressed in Nm^3/h

If valve is without reducers $F_p = 1$
and x_{TP} becomes x_T

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:

$$C_v = C_{v\ g} + C_{v\ liq}$$

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:

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

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

$$V_{g1} / Y^2$$

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 :

$$V_e = f_g \frac{V_{gl}}{Y^2} + f_{liq} \cdot V_{liq1}$$

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

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

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 isoenthalpic) always involves a quality increase, the isoenthropic 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÷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).

$$C_v = C_{v\ liq} + C_{v\ vap}$$

- *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.

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

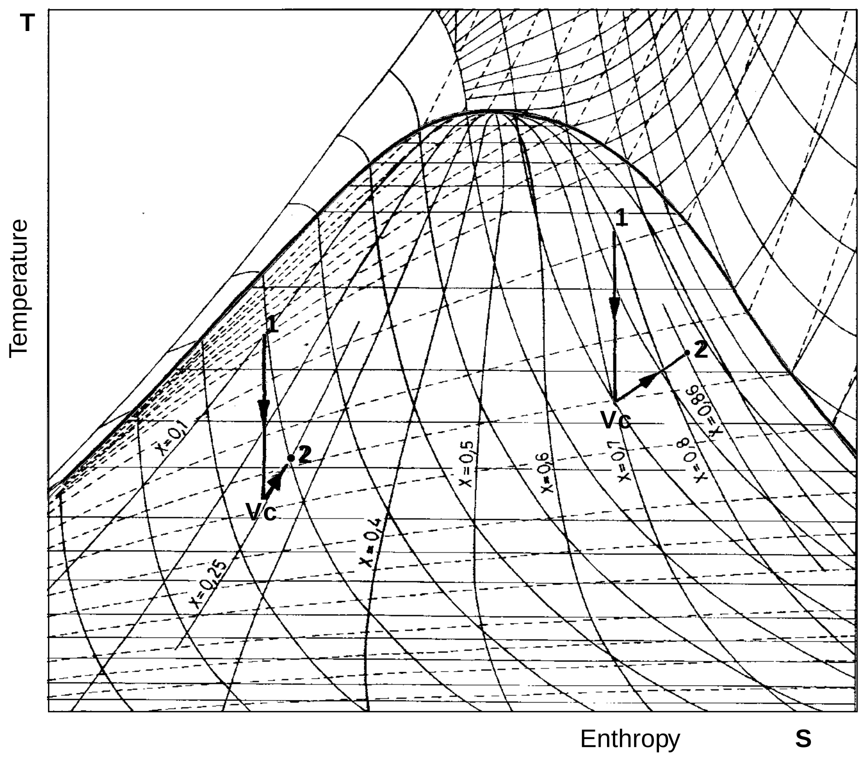


Fig. 4 - Thermodynamic transformations of a water / vapour mixture inside a valve.

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

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

In both cases the point 2 are on the same isoenthalpic 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:

$$Re = \frac{\rho \cdot u \cdot d}{\mu}$$

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 Cv 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:

incompressible fluid

$$C_V = \frac{q_m}{865 \cdot F_R \cdot \sqrt{\Delta p \cdot \rho_r}}$$

$$C_V = \frac{1.16 \cdot q_v}{F_R \cdot \sqrt{\frac{\Delta p}{\rho_r}}}$$

compressible fluid

$$C_V = \frac{q_m}{67 \cdot F_R} \cdot \sqrt{\frac{T_1}{\Delta p \cdot (p_1 + p_2) \cdot M}}$$

$$C_V = \frac{q_v}{1500 \cdot F_R} \cdot \sqrt{\frac{M \cdot T_1}{\Delta p \cdot (p_1 + p_2)}}$$

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 Fp 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 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 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_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:

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

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.

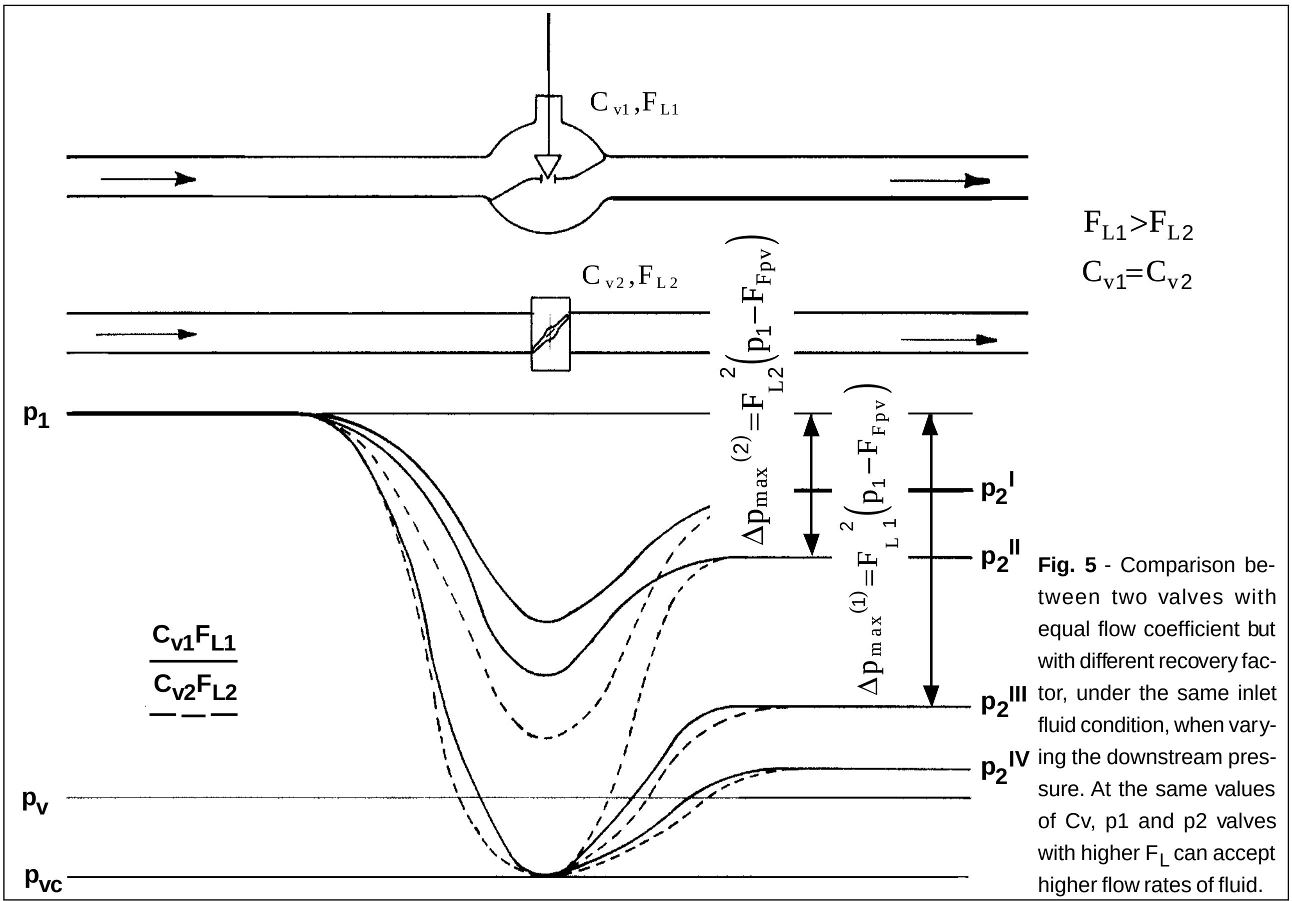


Fig. 5 - Comparison between two valves with equal flow coefficient but with different recovery factor, under the same inlet fluid condition, when varying the downstream pressure. At the same values of Cv, p1 and p2 valves with higher F_L can accept higher flow rates of fluid.

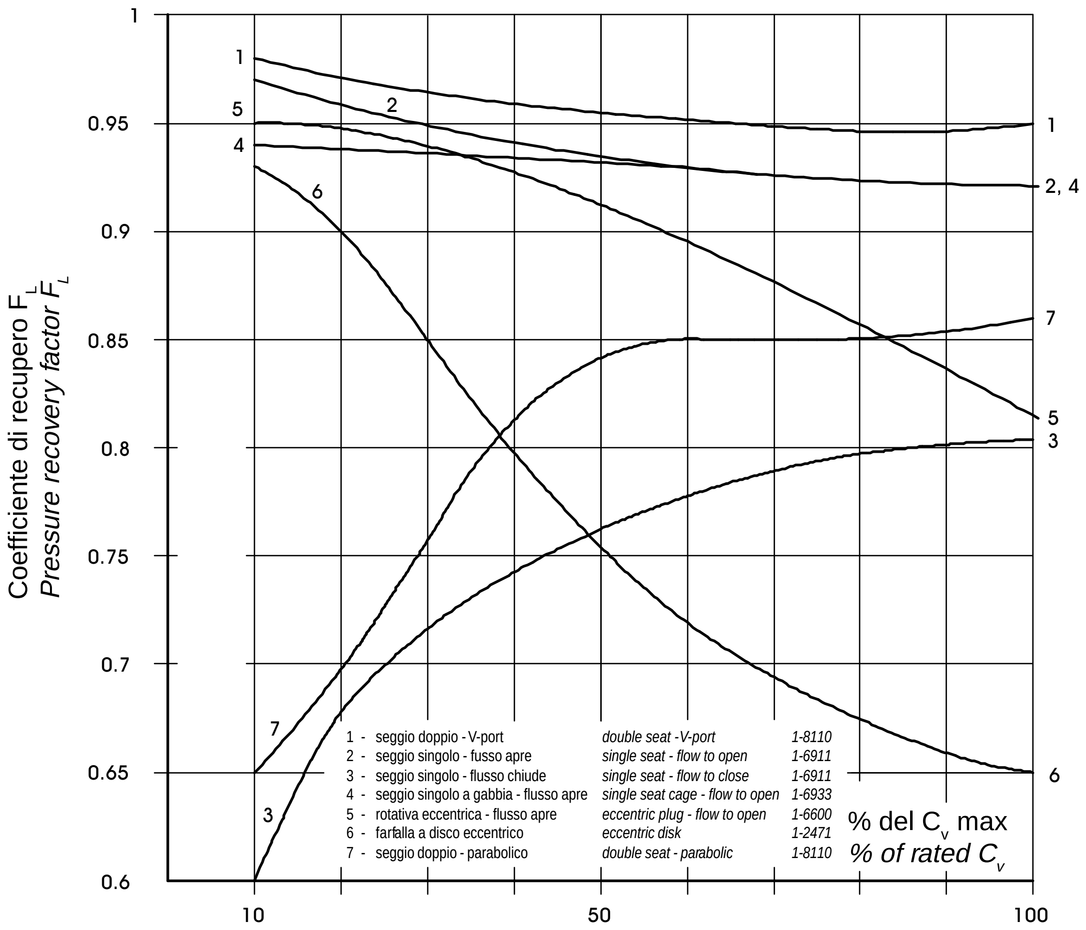


Fig. 6 - Typical F_L values versus % value C_V and flow direction for different PARCOL valve types.

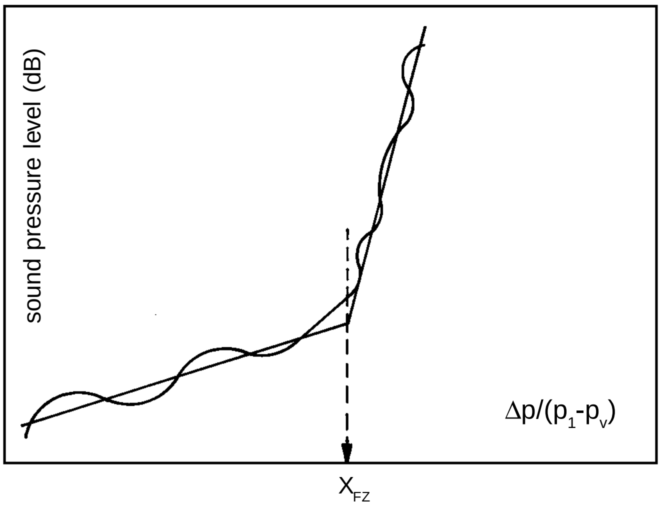
Index of resistance to cavitation	
stellite gr. 6	20
chrome plating	(5)
17-4-PH H900	2
AISI 316/304	1
monel 400	(0.8)
gray cast iron	0.75
chrome-molybdenum alloyed steels (5% chrome)	0.67
carbon steels (WCB)	0.38
bronze (B16)	0.08
nickel plating	(0.07)
pure aluminium	0.006

Fig. 7 - Cavitation resistance of some metallic materials referred to stainless steels AISI 304/316. Values between brackets only for qualitative comparison.

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 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 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



$$x_{FZ} = \frac{\Delta p_{tr}}{p_1 - p_v}$$

where Δp_{tr} is the value of Δp at which the transition takes place from not cavitating to cavitating flow.

Fig. 8 - Determination of the coefficient of incipient cavitation by means of phonometric analysis.
(Drawn from IEC Standard 534-8-2)

the formula $Kc = 0.8\ F_L^2$. 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 Δp at which the deviation from the linearity takes place is different from the Δp 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:

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

In case of expanders:

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

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

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

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 Δp_{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).

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

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

$$F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2}{0.00214}(\Sigma K)_1 \left(\frac{C_v}{d^2}\right)^2}}$$

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-

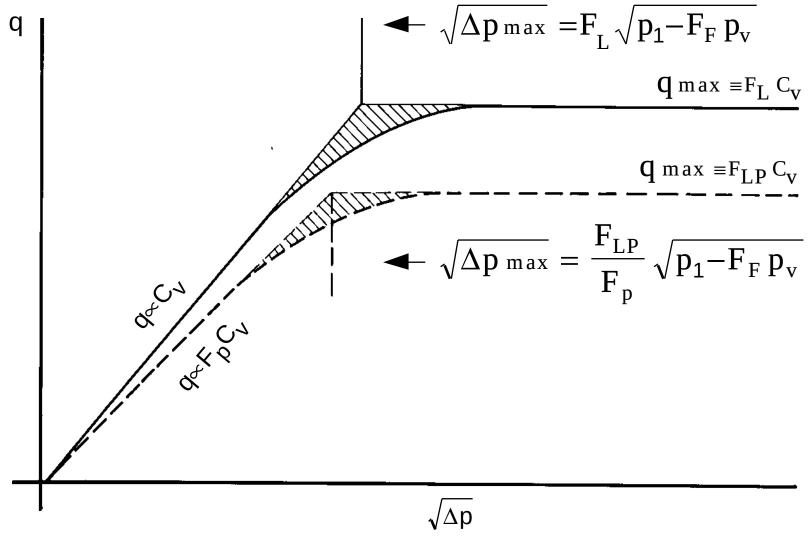


Fig. 9 - Effect of reducers on the diagram of q versus $\sqrt{\Delta p}$ when varying the downstream pressure at constant upstream pressure.

dition and the vapour pressure of the liquid at inlet temperature:

$$F_F = p_{vc} / p_v$$

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):

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

and from:

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

the following equation is obtained:

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

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

$$q_v = F_L \cdot C_v \cdot \sqrt{\frac{p_v}{p_c}}$$

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:

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

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 linear function of $x = \Delta p / p_1$
- Y is a function of the fluid type, namely the exponent of the adiabatic transformation $\gamma = c_p / c_v$
- Y is function of the geometry (i.e. type) of the valve

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

$$q_m \propto Y\sqrt{x}$$

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 / dx becomes zero.

$$q_m \propto (1 - ax)\sqrt{x} = \sqrt{x} - a\sqrt{x^3}$$

By setting

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

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

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:

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

The final equation becomes:

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

Cv/d^2 (d in mm)	15×10^{-3}	20×10^{-3}	25×10^{-3}	30×10^{-3}	35×10^{-3}	40×10^{-3}
F_L	.5 .6 .7 .8 .9	.5 .6 .7 .8 .9	.5 .6 .7 .8 .9	.5 .6 .7 .8 .9	.5 .6 .7 .8 .9	.5 .6 .7 .8 .9
d/D	F_{LP}	F_{LP}	F_{LP}	F_{LP}	F_{LP}	F_{LP}
.25	.49 .58 .67 .77 .85	.48 .57 .66 .74 .81	.47 .56 .64 .71 .78	.47 .54 .61 .68 .74	.45 .53 .59 .65 .70	.44 .51 .57 .62 .66
.33	.49 .58 .68 .76 .85	.48 .57 .66 .74 .82	.48 .56 .64 .71 .78	.47 .54 .62 .68 .74	.46 .53 .59 .65 .70	.44 .51 .57 .62 .66
.40	.49 .58 .68 .77 .85	.48 .57 .66 .74 .82	.48 .56 .64 .72 .78	.47 .55 .62 .69 .75	.46 .53 .60 .66 .71	.45 .51 .57 .62 .67
.50	.49 .59 .68 .77 .86	.49 .58 .66 .75 .83	.48 .56 .65 .72 .79	.47 .55 .62 .69 .76	.46 .54 .60 .66 .72	.45 .52 .58 .63 .68
.66	.49 .59 .68 .77 .86	.49 .58 .67 .76 .84	.48 .57 .66 .74 .81	.48 .56 .64 .71 .78	.47 .55 .62 .69 .74	.46 .53 .60 .66 .71
.75	.49 .59 .69 .78 .87	.49 .58 .68 .76 .85	.49 .58 .66 .75 .83	.48 .57 .65 .73 .80	.47 .56 .63 .70 .77	.47 .54 .62 .68 .74

Fig. 10 - Values of F_{LP} for valves with short type reducer at the inlet with abrupt section variation

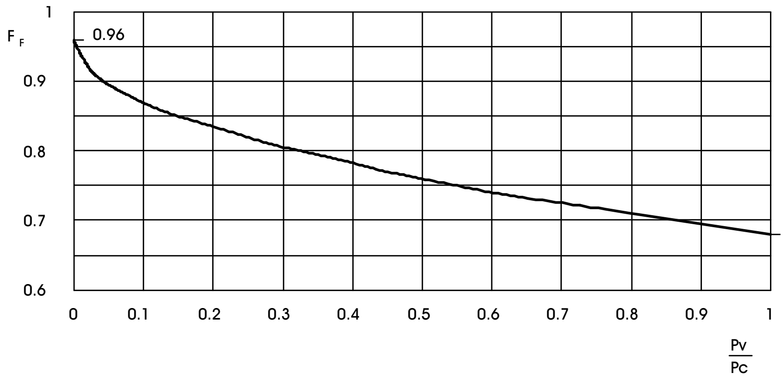


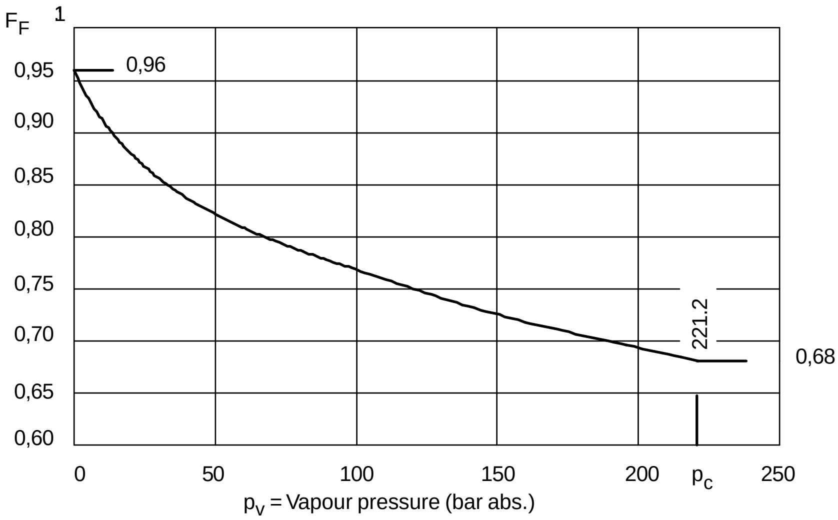
Fig. 11 -Liquid critical pressure ratio factor

p_V = Vapour pressure (bar abs.)

p_C = Critical pressure (bar abs.)

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

Fig. 12 -Critical pressure ratio factor for water



$$F_F = 0.96 - 0.28\sqrt{\frac{p_v}{221.2}}$$

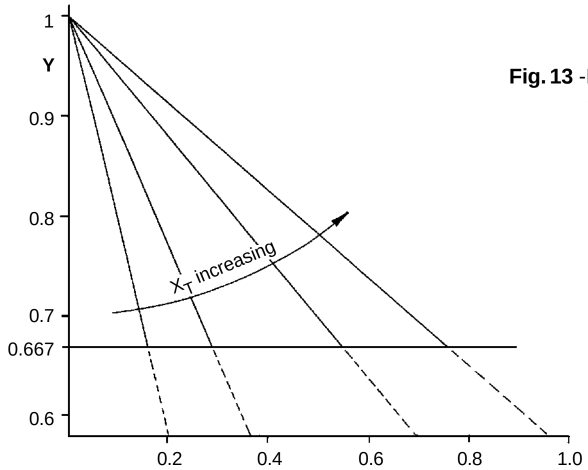


Fig. 13 -Expansion factor Y.
The diagram is valid for a given of Fγ value.

$$x = \frac{\Delta p}{p_1}$$

Therefore the maximum flow rate is reached when $x = F_\gamma \cdot x_T$ (or $F\gamma \cdot x_{TP}$ 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 CHOKED 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

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

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 CHOKED 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.

$$x_{TP} = \frac{x_T}{(F_p)^2} \cdot \frac{1}{1 + \frac{x_T (K_1 + K_{B1})}{0.0024} \cdot \left(\frac{C_v}{d^2}\right)^2}$$

Cd	10		15		20		25		30	
x_T	.40 .50 .60 .70 .80		.40 .50 .60 .70 .80		.40 .50 .60 .70		.20 .30 .40 .50		.15 .20 .25	
d/D	x_{TP}	F_p	x_{TP}	F_p	x_{TP}	F_p	x_{TP}	F_p	x_{TP}	F_p
.80	.40 .49 .59 .69 .78	.99	.40 .49 .58 .67 .75	.98	.39 .48 .56 .64	.96	.21 .30 .39 .47	.94	.17 .21 .26	.91
.75	.40 .50 .59 .69 .78	.98	.40 .49 .58 .67 .75	.97	.40 .49 .57 .65	.94	.22 .31 .40 .48	.91	.18 .23 .27	.88
.67	.40 .50 .60 .69 .78	.98	.41 .50 .59 .68 .76	.95	.42 .51 .59 .67	.91	.24 .33 .43 .51	.87	.19 .25 .30	.83
.60	.41 .51 .60 .70 .79	.97	.42 .52 .61 .69 .78	.93	.43 .53 .61 .69	.89	.25 .36 .45 .54	.84	.21 .27 .32	.79
.50	.41 .52 .61 .70 .80	.96	.44 .53 .63 .71 .79	.91	.46 .55 .64 .72	.85	.28 .39 .49 .58	.79	.24 .30 .36	.73
.40	.42 .52 .62 .71 .80	.95	.44 .55 .65 .74 .82	.89	.49 .58 .67 .75	.82	.30 .42 .53 .62	.76	.26 .33 .40	.70
.33	.43 .53 .62 .72 .81	.94	.46 .56 .66 .75 .83	.88	.50 .60 .69 .78	.81	.31 .44 .55 .64	.74	.27 .34 .40	.69
.25	.44 .53 .63 .73 .83	.93	.48 .58 .67 .76 .85	.87	.52 .62 .71 .79	.79	.33 .46 .57 .67	.72	.27 .37 .44	.65

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.
$C_d = C_v / d^2 = 20$ d / D = 2/3 = 0.67
A linear interpolation between $x_T = 0.6$ and $x_T = 0.7$ results in $x_{TP} = 0.63$

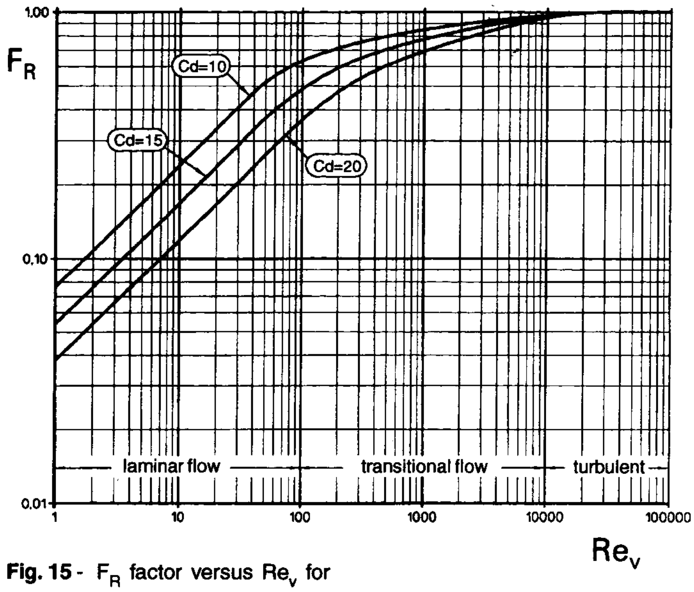


Fig. 15 - F_R factor versus Re_v for some Cd values

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:

$$Re_v = \frac{7{,}6 \cdot 10^4 \cdot Fd \cdot q_v}{\nu \sqrt{F_L \cdot C_v}} \cdot \sqrt[4]{\frac{467 \cdot F_L^{\,2} \cdot C_v^{\,2}}{D^4} + 1}$$

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.
Fd 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.

VALVE STYLE MODIFIER Fd				
Valve type		Flow direction	Relative flow coefficient	
			0.10	1.00
Globe, parabolic plug (1-6911, 1-6951, 1-6921, 1-6981 e 1-4411)		Flow-to-open Flow-to-close	0.10 0.20	0.46 1.00
Butterfly valve 1-2471, 1-2512, 1-2311	Max. opening 90° 60°	Whatever	0.20 0.20	0.7 0.5
Cage valve 1-6933, 1-4433, 1-6971, 1-4471	Number of holes 50 100 200	Whatever	0.45 0.32 0.22	0.14 0.10 0.07
Double seat 1-8110	Parabolic V-port	Between seats	0.10 0.10	0.32 0.28

Fig. 16 -Typical Fd values for PARCOL control valves. More accurate values on request

(Company)	CONTROL VALVE DATA SHEET	TAG N° ______
REV. / POS. / (PLANT)		MFR SERIAL N° ______
		QUANTITY______ SHEET ______ OF______
		ORDERING SPEC. ITEM______

No.	Section	Item	I (min.)	II (norm.)	III (max.)	unit
1	PROCESS DATA RELEVANT FOR CONTROL VALVE SELECTION	Location				
2		Service				
3		Process fluid				
4		Ambient temp. min. max.				
5		Allowable sound pressure level dB(A)				
6		Upstream pipe | Thickness				
7		Downstream pipe | Thickness				
8		Pipe material				
9		Pipe insulation ☐ thermal ☐ acoustic				
10		Design pressure bar | Temper. °C				
11		Pipe connection / rating				
12		Pipe identification n°				
13		Upstream condition ☐ liquid ☐ vapour ☐ gas				
14						
15			I (min.)	II (norm.)	III (max.)	unit
16		Flow rate				
17		Inlet press. r1				
18		Outlet press. P2				
19		Temperature T1				
20		Inlet density ρ 1 or M				
21		Vapour pressure Pv				
22		Critical pressure Pc				
23		Viscosity				
24		γ =cp/cv				—
25		Compressibility factor Z1				—
26						
27		Shutoff pressure P1 P2				
28		Air supply min. max.				
29		Power failure position ☐ open ☐ closed ☐ hold				
30	Cv / SPL	F_L | x_T | F_{LP} | x_{TP}				—
31		Calculated max. Cv				
32		Calculated min. Cv				
33		Selected Cv				
34		Predicted SPL				dB(A)
35	VALVE BODY ASSEMBLY	MFR Model				
36		Body type				
37		Flow direction				
38		Pressure rating				
39		Nominal size Port				
40		End conn. ☐ Flgd. ☐ flgless ☐ welded ☐				
41						
42		End extensions				
43		Bonnet style ☐ standard ☐ extension ☐ bellows				
44						
45		Body / bonnet materials				
46		Trim ☐ standard ☐ low noise ☐ anticavit.				
47		Characteristic ☐ linear ☐ eq. Percent. ☐				
48		Plug (disk) / matl.				
49		Seat matl.				
50		Stem (shaft) / matl. | Guide (cage) / matl.				
51		Seat style ☐ metallic ☐ soft				
52		Trim coating				
53						
54		Leakage specification				
55		Packing ☐ teflon ☐ graphite ☐				
56						

No.	Section	Item
57	ACTUATOR	MFR. Model
58		Pneumatic ☐ diaphragm ☐ piston ☐
59		Style ☐ Sprg. opposed ☐ double act. ☐ lr spr.
60		Size Effective area
61		Travel / angle
62		Supply pressure min. max.
63		Bench range
64		
65		Air connection
66		Other actuator ☐ electric ☐ hydraulic ☐ manual
67		
68		Handwheel ☐ top ☐ side
69		
70	POSITIONER	MFR. Model
71		Input signal ☐ electric ☐ pneumatic
72		Valve open at
73		Valve closed at
74		Style ☐ single action ☐ double action
75		Characteristic ☐ linear ☐
76		Air connection
77		Accessories ☐ by-pass ☐ gauges
78		Reduction of haz. ☐ Ex-d ☐
79		
80	POS. IND. SWITCH	MFR. Model
81		Switch type ☐ mench. ☐ Proximity ☐ pneumatic
82		Switching position ☐ closed ☐ % travel ☐ open
83		Switch acting ☐ make ☐ break
84		Reduction of haz. ☐ Ex-d ☐
85		
86	SOLENOID VALVE	MFR. Model
87		Valve style ☐ two way ☐ three way ☐ four way
88		De-energ.: control valve ☐ open ☐ closed ☐ hold
89		
90		Air connection port size
91		Electrical data : V Hz W
92		Reduction of haz. ☐ Ex-d ☐
93		
94	OTHER ACCESSORIES	☐ Air set MFR. Model
95		☐ With filter ☐ with gauge
96		☐ Transducer MFR. Model
97		
98		Booster MFR. Model
99		
100		Lockup MFR. Model
101		
102		Air tubing material
103		
104	SPECIAL REQUIREMENTS	Inspection plan:
105		
106		Part to be tested ☐ body / bonnet
107		☐ Bolts / nuts ☐ trim
108		
109		
110		
111		
112		

NOTES :

This data sheet was derived from IEC 60534-7 with some improvements not affecting the numbering of the original items.

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