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(IEC 60534-2-1 Mod)

Flow Equations for Sizing Control Valves

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ISA–The Instrumentation, Systems, and Automation Society

ANSI/ISA-75.01.01-2002 (60534-2-1 Mod)
Flow Equations for Sizing Control Valves

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Preface

This preface, as well as all footnotes and annexes, is included for information purposes and is not part of ANSI/ISA-75.01.01-2002 (IEC 60534-2-1 Mod).

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FOREWORD

NOTE This foreword is included for information purposes only and is identical to the foreword found in IEC 60534-2-1.

1) The IEC (International Electrotechnical Commission) is a worldwide organization for standardization comprising all national electrotechnical committees (IEC National Committees). The object of the IEC is to promote international co-operation on all questions concerning standardization in the electrical and electronic fields. To this end and in addition to other activities, the IEC publishes International Standards. Their preparation is entrusted to technical committees; any IEC National Committee interested in the subject dealt with may participate in this preparatory work. International, governmental and non-governmental organizations liaising with the IEC also participate in this preparation. The IEC collaborates closely with the International Organization for Standardization (ISO) in accordance with conditions determined by agreement between the two organizations.

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3) The documents produced have the form of recommendations for international use and are published in the form of standards, technical reports or guides and they are accepted by the National Committees in that sense.

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International Standard IEC 60534-2-1 has been prepared by subcommittee 65B: Devices, of IEC technical committee 65: Industrial-process measurement and control.

The text of this standard is based on the following documents:

FDIS	Report on voting
65B/347/FDIS	65B/357/RVD

Full information on the voting for the approval of this standard can be found in the report on voting indicated in the above table.

The current edition of IEC 60534-2-1 cancels and replaces the first edition of both IEC 60534-2 published in 1978, and IEC 60534-2-2 published in 1980, which cover incompressible and compressible fluid flow, respectively.

IEC 60534-2-1 covers sizing equations for both incompressible and compressible fluid flow.

Annexes A, B, C, D F, G, and H are for information only.

A bilingual version of this standard may be issued at a later date.

1 Scope

ANSI/ISA-75.01.01-2002 includes equations for predicting the flow coefficient of compressible and incompressible fluids through control valves.

The equations for incompressible flow are based on standard hydrodynamic equations for Newtonian incompressible fluids. They are not intended for use when non-Newtonian fluids, fluid mixtures, slurries, or liquid-solid conveyance systems are encountered.

At very low ratios of pressure differential to absolute inlet pressure ($\Delta P/P_1$), compressible fluids behave similarly to incompressible fluids. Under such conditions, the sizing equations for compressible flow can be traced to the standard hydrodynamic equations for Newtonian incompressible fluids. However, increasing values of $\Delta P/P_1$ result in compressibility effects that require that the basic equations be modified by appropriate correction factors. The equations for compressible fluids are for use with gas or vapor and are not intended for use with multiphase streams such as gas-liquid, vapor-liquid or gas-solid mixtures.

For compressible fluid applications, this part of ANSI/ISA-75.01.01-2002 is valid for all valves. However, manufacturers of some valves with $x_T \geq 0.84$ have reported minor inaccuracies (see Annex H). Caution must also be exercised when applying the equations for compressible fluids to gaseous mixtures of compounds, particularly near phase boundaries.

The accuracy of results computed with the equations in this standard will be governed by the accuracy of the constituent coefficients and the process data supplied. Methods of evaluating the coefficients used in the equations presented herein are given in ANSI/ISA-75.02-1996. The stated accuracy associated with the coefficients in that document is ±5% when $C_v/d^2 < 0.047\ N_{18}$. Reasonable accuracy can only be maintained for control valves if $C_v/d^2 < 0.047\ N_{18}$.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ANSI/ISA-75.01.01-2002. At the time of publication, the editions indicated were valid. All normative documents are subject to revision, and parties to agreements based on this part of ANSI/ISA-75.01.01-2002 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

IEC 60534-1:1987, *Industrial-process control valves – Part 1: Control valve terminology and general considerations*

IEC 60534-2-3:1997, *Industrial-process control valves – Part 2: Flow capacity – Section 3: Test procedures*

ANSI/ISA-75.02-1996, Control Valve Capacity Test Procedures

ANSI/ISA-75.05.01-2001, Control Valve Terminology

3 Definitions

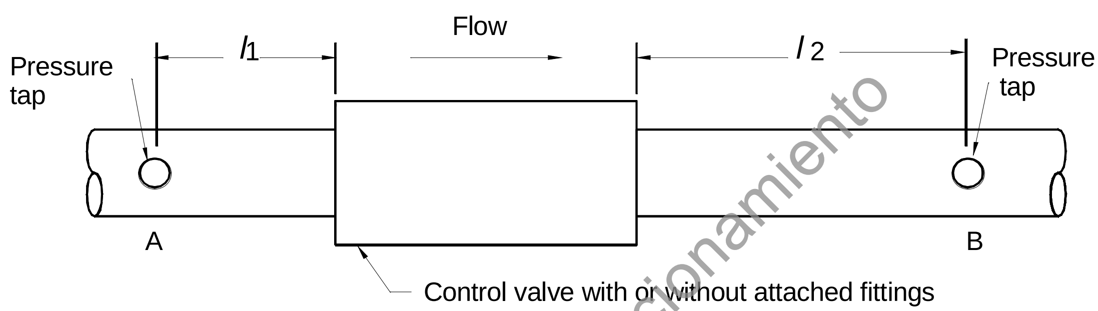
For the purpose of ANSI/ISA-75.01.01-2002, definitions given in IEC 60534-2-1 apply with the addition of the following:

3.1 valve style modifier F_d
the ratio of the hydraulic diameter of a single flow passage to the diameter of a circular orifice, the area of which is equivalent to the sum of areas of all identical flow passages at a given travel. It should be stated by the manufacturer as a function of travel (see Annex A).

4 Installation

In many industrial applications, reducers or other fittings are attached to the control valves. The effect of these types of fittings on the nominal flow coefficient of the control valve can be significant. A correction factor is introduced to account for this effect. Additional factors are introduced to take account of the fluid property characteristics that influence the flow capacity of a control valve.

In sizing control valves, using the relationships presented herein, the flow coefficients calculated are assumed to include all head losses between points A and B, as shown in Figure 1.



l_1 = two nominal pipe diameters

l_2 = six nominal pipe diameters

Figure 1 — Reference pipe section for sizing

5 Symbols

Symbol	Description	Unit
C	Flow coefficient (K_v, C_v)	Various (see IEC 60534-1) (see note 4)
C_i	Assumed flow coefficient for iterative purposes	Various (see IEC 60534-1) (see note 4)
d	Nominal valve size	mm (in)
D	Internal diameter of the piping	mm (in)
D_1	Internal diameter of upstream piping	mm (in)
D_2	Internal diameter of downstream piping	mm (in)
D_o	Orifice diameter	mm (in)
F_d	Valve style modifier (see Annex A)	Dimensionless (see note 4)
F_F	Liquid critical pressure ratio factor	Dimensionless
F_L	Liquid pressure recovery factor of a control valve without attached fittings	Dimensionless (see note 4)
F_{LP}	Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings	Dimensionless (see note 4)
F_P	Piping geometry factor	Dimensionless
F_R	Reynolds number factor	Dimensionless
F_γ	Specific heat ratio factor	Dimensionless
G_g	Gas specific gravity (ratio of density of flowing gas to density of air with both at standard conditions, which is considered in this practice to be equal to the ratio of the molecular weight of gas to molecular weight of air	Dimensionless
M	Molecular mass of flowing fluid	kg/kg-mol (lb/lb-mol)
N	Numerical constants (see Table 1)	Various (see note 1)
P_1	Inlet absolute static pressure measured at point A (see Figure 1)	kPa or bar (psia)(see note 2)
P_2	Outlet absolute static pressure measured at point B (see Figure 1)	kPa or bar (psia)
P_c	Absolute thermodynamic critical pressure	kPa or bar (psia)
P_r	Reduced pressure (P_1 / P_c)	Dimensionless
P_v	Absolute vapor pressure of the liquid at inlet temperature	kPa or bar (psia)
ΔP	Differential pressure between upstream and downstream pressure taps ($P_1 - P_2$)	kPa or bar (psi)
Q	Volumetric flow rate (see note 5)	m^3/h (gpm, scfh)
Re_v	Valve Reynolds number	Dimensionless
T_1	Inlet absolute temperature	K (R)
T_c	Absolute thermodynamic critical temperature	K (R)
T_r	Reduced temperature (T_1 / T_c)	Dimensionless
t_s	Absolute reference temperature for standard cubic meter	K (R)
W	Mass flow rate	kg/h (lbs/h)
x	Ratio of pressure differential to inlet absolute pressure ($\Delta P / P_1$)	Dimensionless
x_T	Pressure differential ratio factor of a control valve without attached fittings at choked flow	Dimensionless (see note 4)
x_{TP}	Pressure differential ratio factor of a control valve with attached fittings at choked flow	Dimensionless (see note 4)
Y	Expansion factor	Dimensionless
Z	Compressibility factor	Dimensionless
v	Kinematic viscosity	m^2/s (cS) (see note 3)
ρ_1	Density of fluid at P_1 and T_1	kg/m^3 (lb/ft^3)
ρ_1/ρ_o	Relative density (ρ_1/ρ_o = 1.0 for water at 15°C)	Dimensionless
γ	Specific heat ratio	Dimensionless
ζ	Velocity head loss coefficient of a reducer, expander or other fitting attached to a control valve or valve trim	Dimensionless

ζ_1	Upstream velocity head loss coefficient of fitting	Dimensionless
ζ_2	Downstream velocity head loss coefficient of fitting	Dimensionless
ζ_{B1}	Inlet Bernoulli coefficient	Dimensionless
ζ_{B2}	Outlet Bernoulli coefficient	Dimensionless

NOTE 1 To determine the units for the numerical constants, dimensional analysis may be performed on the appropriate equations using the units given in Table 1.
NOTE 2 1 bar = 10^2 kPa = 10^5 Pa
NOTE 3 1 centistoke = 10^{-6} m^2/s
NOTE 4 These values are travel-related and should be stated by the manufacturer.
NOTE 5 Volumetric flow rates in cubic meters per hour, identified by the symbol Q, refer to standard conditions. The standard cubic meter is taken at 1013.25 mbar and either 273 K or 288 K (see Table 1).

6 Sizing equations for incompressible fluids

The equations listed below identify the relationships between flow rates, flow coefficients, related installation factors, and pertinent service conditions for control valves handling incompressible fluids. Flow coefficients may be calculated using the appropriate equation selected from the ones given below. A sizing flow chart for incompressible fluids is given in Annex B.

6.1 Turbulent flow

The equations for the flow rate of a Newtonian liquid through a control valve when operating under non-choked flow conditions are derived from the basic formula as given in IEC 60534-2-1.

6.1.1 Non-choked turbulent flow

6.1.1.1 Non-choked turbulent flow without attached fittings

$$\left[\textit{Applicable if } \Delta P < F_L^{\,2}\left(P_1 - F_F P_v\right)\right]$$

The flow coefficient shall be determined by

Eq. 1 $$C = \frac{Q}{N_1}\sqrt{\frac{\rho_1/\rho_o}{\Delta P}}$$

NOTE 1 The numerical constant N_1 depends on the units used in the general sizing equation and the type of flow coefficient: K_v or C_v.

NOTE 2 An example of sizing a valve with non-choked turbulent flow without attached fittings is given in Annex F.

6.1.1.2 Non-choked turbulent flow with attached fittings

$$\left\{ \textit{Applicable if } \Delta P < \left[\left(F_{LP}/F_p\right)^2\left(P_1 - F_F P_v\right)\right]\right\}$$

The flow coefficient shall be determined as follows:

Eq. 2 $$C = \frac{Q}{N_1 F_p}\sqrt{\frac{\rho_1/\rho_o}{\Delta P}}$$

NOTE Refer to 8.1 for the piping geometry factor F_P.

6.1.2 Choked turbulent flow

The maximum rate at which flow will pass through a control valve at choked flow conditions shall be calculated from the following equations:

6.1.2.1 Choked turbulent flow without attached fittings

$$\left\lfloor \textit{Applicable if } \Delta P \geq F_L^{\,2} \left(P_1 - F_F P_v\right) \right\rfloor$$

The flow coefficient shall be determined as follows:

Eq. 3
$$C = \frac{Q}{N_1 F_L} \sqrt{\frac{\rho_1 / \rho_o}{P_1 - F_F P_v}}$$

NOTE An example of sizing a valve with choked flow without attached fittings is given in Annex F.

6.1.2.2 Choked turbulent flow with attached fittings

$$\left\lfloor \textit{Applicable if } \Delta P \geq \left(F_{LP} / F_p\right)^2 \left(P_1 - F_F P_v\right) \right\rfloor$$

The following equation shall be used to calculate the flow coefficient:

Eq. 4
$$C = \frac{Q}{N_1 F_{LP}} \sqrt{\frac{\rho_1 / \rho_o}{P_1 - F_F P_v}}$$

6.2 Non-turbulent (laminar and transitional) flow

The equations for the flow rate of a Newtonian liquid through a control valve when operating under non-turbulent flow conditions are derived from the basic formula as given in IEC 60534-2-1. This equation is applicable if $Re_v < 10{,}000$ (see Equation 28).

6.2.1 Non-turbulent flow without attached fittings

The flow coefficient shall be calculated as follows:

Eq. 5
$$C = \frac{Q}{N_1 F_R} \sqrt{\frac{\rho_1 / \rho_o}{\Delta P}}$$

6.2.2 Non-turbulent flow with attached fittings

For non-turbulent flow, the effect of close-coupled reducers or other flow disturbing fittings is unknown. While there is no information on the laminar or transitional flow behavior of control valves installed between pipe reducers, the user of such valves is advised to utilize the appropriate equations for line-sized valves in the calculation of the F_R factor. This should result in conservative flow coefficients since additional turbulence created by reducers and expanders will further delay the onset of laminar flow. Therefore, it will tend to increase the respective F_R factor for a given valve Reynolds number.

7 Sizing equations for compressible fluids

The equations listed below identify the relationships between flow rates, flow coefficients, related installation factors, and pertinent service conditions for control valves handling compressible fluids. Flow rates for compressible fluids may be encountered in either mass or volume units and thus equations are necessary to handle both situations. Flow coefficients may be calculated using the appropriate equations selected from the following. A sizing flow chart for compressible fluids is given in Annex B.

The flow rate of a compressible fluid varies as a function of the ratio of the pressure differential to the absolute inlet pressure ($\Delta P/P_1$), designated by the symbol x. At values of x near zero, the equations in this section can be traced to the basic Bernoulli equation for Newtonian incompressible fluids. However, increasing values of x result in expansion and compressibility effects that require the use of appropriate factors (see Buresh, Schuder, and Driskell references).

7.1 Turbulent flow

7.1.1 Non-choked turbulent flow

7.1.1.1 Non-choked turbulent flow without attached fittings

$\left\lfloor \textit{Applicable if } x < F_\gamma x_T \right\rfloor$

The flow coefficient shall be calculated using one of the following equations:

Eq. 6 $$C = \frac{W}{N_6 Y \sqrt{x P_1 \rho_1}}$$

Eq. 7 $$C = \frac{W}{N_8 P_1 Y} \sqrt{\frac{T_1 Z}{x M}}$$

Eq. 8a $$C = \frac{Q}{N_9 P_1 Y} \sqrt{\frac{M T_1 Z}{x}}$$

Eq. 8b $$C = \frac{Q}{N_7 P_1 Y} \sqrt{\frac{G g T_1 Z}{x}}$$

NOTE 1 Refer to 8.5 for details of the expansion factor *Y*.

NOTE 2 See Annex C for values of *M*.

7.1.1.2 Non-choked turbulent flow with attached fittings

$\left\lfloor \textit{Applicable if } x < F_\gamma x_{TP} \right\rfloor$

The flow coefficient shall be determined from one of the following equations:

Eq. 9 $$C = \frac{W}{N_6 F_p Y \sqrt{x P_1 \rho_1}}$$

Eq. 10 $$C = \frac{W}{N_8 F_p P_1 Y} \sqrt{\frac{T_1 Z}{x M}}$$

Eq. 11a $$C = \frac{Q}{N_9 F_p P_1 Y} \sqrt{\frac{M T_1 Z}{x}}$$

Eq. 11b $$C = \frac{Q}{N_7 F_p P_1 Y} \sqrt{\frac{G g T_1 Z}{x}}$$

NOTE 1 Refer to 8.1 for the piping geometry factor F_P.

NOTE 2 An example of sizing a valve with non-choked turbulent flow with attached fittings is given in Annex F.

7.1.2 Choked turbulent flow

The maximum rate at which flow will pass through a control valve at choked flow conditions shall be calculated as follows:

7.1.2.1 Choked turbulent flow without attached fittings

$\left\lfloor \textit{Applicable if } x \geq F_\gamma x_T \right\rfloor$

The flow coefficient shall be calculated from one of the following equations:

Eq. 12 $$C = \frac{W}{0.667 N_6 \sqrt{F_\gamma x_T P_1 \rho_1}}$$

Eq. 13 $$C = \frac{W}{0.667 N_8 P_1} \sqrt{\frac{T_1 Z}{F_\gamma x_T M}}$$

Eq. 14a $$C = \frac{Q}{0.667 N_9 P_1} \sqrt{\frac{M T_1 Z}{F_\gamma x_T}}$$

Eq. 14b $$C = \frac{Q}{0.667 N_7 P_1} \sqrt{\frac{G g T_1 Z}{F_\gamma x_T}}$$

7.1.2.2 Choked turbulent flow with attached fittings

$\left\lfloor \textit{Applicable if } x \geq F_\gamma x_{TP} \right\rfloor$

The flow coefficient shall be determined using one of the following equations:

Eq. 15 $$C = \frac{W}{0.667 N_6 F_p \sqrt{F_\gamma x_{TP} P_1 \rho_1}}$$

Eq. 16 $$C = \frac{W}{0.667 N_8 F_p P_1} \sqrt{\frac{T_1 Z}{F_\gamma x_{TP} M}}$$

Eq. 17a $$C = \frac{Q}{0.667 N_9 F_p P_1} \sqrt{\frac{M T_1 Z}{F_\gamma x_{TP}}}$$

Eq. 17b $$C = \frac{Q}{0.667 N_7 F_p P_1} \sqrt{\frac{G g T_1 Z}{F_\gamma x_{TP}}}$$

7.2 Non-turbulent (laminar and transitional) flow

The equations for the flow rate of a Newtonian fluid through a control valve when operating under non-turbulent flow conditions are derived from the basic formula as given in IEC 60534-2-1. These equations are applicable if $Re_v < 10{,}000$ (see Equation 28). In this subclause, density correction of the gas is given by $(P_1 + P_2) / 2$ due to non-isentropic expansion.

7.2.1 Non-turbulent flow without attached fittings

The flow coefficient shall be calculated from one of the following equations:

Eq. 18 $$C = \frac{W}{N_{27} F_R} \sqrt{\frac{T_1}{\Delta P \left(P_1 + P_2\right) M}}$$

Eq. 19 $$C = \frac{Q}{N_{22} F_R} \sqrt{\frac{M T_1}{\Delta P \left(P_1 + P_2\right)}}$$

NOTE An example of sizing a valve with small flow trim is given in Annex D.

7.2.2 Non-turbulent flow with attached fittings

For non-turbulent flow, the effect of close-coupled reducers or other flow-disturbing fittings is unknown. While there is no information on the laminar or transitional flow behavior of control valves installed between pipe reducers, the user of such valves is advised to utilize the appropriate equations for line-sized valves in the calculation of the F_R factor. This should result in conservative flow coefficients since additional turbulence created by reducers and expanders will further delay the onset of laminar flow. Therefore, it will tend to increase the respective F_R factor for a given valve Reynolds number.

8 Determination of correction factors

8.1 Piping geometry factor F_P

The piping geometry factor F_P is necessary to account for fittings attached upstream and/or downstream to a control valve body. The F_P factor is the ratio of the flow rate through a control valve installed with attached fittings to the flow rate that would result if the control valve was installed without attached fittings and tested under identical conditions which will not produce choked flow in either installation (see Figure 1). To meet the accuracy of the F_P factor of ±5%, the F_P factor shall be determined by test in accordance with ANSI/ISA-75.02-1996.

When estimated values are permissible, the following equation shall be used:

Eq. 20 $$F_p = \frac{1}{\sqrt{1 + \frac{\Sigma\zeta}{N_2}\left(\frac{C_i}{d^2}\right)^2}}$$

In this equation, the factor $\Sigma\zeta$ is the algebraic sum of all of the effective velocity head loss coefficients of all fittings attached to the control valve. The velocity head loss coefficient of the control valve itself is not included.

Eq. 21 $$\Sigma\zeta = \zeta_1 + \zeta_2 + \zeta_{B1} - \zeta_{B2}$$

In cases where the piping diameters approaching and leaving the control valve are different, the ζ_B coefficients are calculated as follows:

Eq. 22 $$\zeta_B = 1 - \left(\frac{d}{D}\right)^4$$

If the inlet and outlet fittings are short-length, commercially available, concentric reducers, the ζ_1 and ζ_2 coefficients may be approximated as follows:

Eq. 23 Inlet reducer: $$\zeta_1 = 0.5\left[1 - \left(\frac{d}{D_1}\right)^2\right]^2$$

Eq. 24 Outlet reducer (expander): $$\zeta_2 = 1.0\left[1 - \left(\frac{d}{D_2}\right)^2\right]^2$$

Eq. 25 Inlet and outlet reducers of equal size: $$\zeta_1 + \zeta_2 = 1.5\left[1 - \left(\frac{d}{D}\right)^2\right]^2$$

The F_P values calculated with the above ζ factors generally lead to the selection of valve capacities slightly larger than required. This calculation requires iteration. Proceed by calculating the flow coefficient C for non-choked turbulent flow.

NOTE Choked flow equations and equations involving F_P are not applicable.

Next, establish C_i as follows:

Eq. 26 $$C_i = 1.3C$$

Using C_i from Equation 26, determine F_P from Equation 20. If both ends of the valve are the same size, F_P may instead be determined from Figure 2. Then, determine if

Eq. 27 $$\frac{C}{F_p} \le C_i$$

If the condition of Equation 27 is satisfied, then use the C_i established from Equation 26. If the condition of Equation 27 is not met, then repeat the above procedure by again increasing C_i by 30%. This may require several iterations until the condition required in Equation 27 is met. An iteration method more suitable for computers can be found in Annex B.

For graphical approximations of F_P, refer to Figures 2a and 2b.

8.2 Reynolds number factor F_R

The Reynolds number factor F_R is required when non-turbulent flow conditions are established through a control valve because of a low pressure differential, a high viscosity, a very small flow coefficient, or a combination thereof.

The F_R factor is determined by dividing the flow rate when non-turbulent flow conditions exist by the flow rate measured in the same installation under turbulent conditions.

Tests show that F_R can be determined from the curves given in Figure 3a or 3b using a valve Reynolds number calculated from the following equation:

Eq. 28 $$Re_v = \frac{N_4 F_d Q}{\nu \sqrt{C_i F_L}} \left(\frac{F_L^2 C_i^2}{N_2 D^4} + 1 \right)^{1/4}$$

This calculation will require iteration. Proceed by calculating the flow coefficient C for turbulent flow. The valve style modifier F_d converts the geometry of the orifice(s) to an equivalent circular single flow passage. See Table 2 for typical values and Annex A for details. To meet a deviation of ±5% for F_d, the F_d factor shall be determined by test in accordance with IEC 60534-2-3.

NOTE Equations involving F_P are not applicable.

Next, establish C_i as per Equation 26.

Apply C_i as per Equation 26 and determine F_R. F_R is determined from Figure 3a for full-size trim valves.

F_R is determined from Figure 3b for reduced trim valves where C_i/d^2 at rated travel is less then 0.016 N_{18}.

Eq. 29 $$\frac{C}{F_R} \le C_i$$

If the condition of Equation 29 is satisfied, then use the C_i established from Equation 26. If the condition of Equation 29 is not met, then repeat the above equation by again increasing C_i by 30 percent. This may require several iterations until the conditions required in Equation 29 are met.

The equations defining nonturbulent flow for full size and reduced trim valves are stated in Annex G.

8.3 Liquid pressure recovery factors F_L or F_{LP}

8.3.1 Liquid pressure recovery factor without attached fittings F_L

F_L is the liquid pressure recovery factor of the valve without attached fittings. This factor accounts for the influence of the valve internal geometry on the valve capacity at choked flow. It is defined as the ratio of the actual maximum flow rate under choked flow conditions to a theoretical, non-choked flow rate which would be calculated if the pressure differential used was the difference between the valve inlet pressure and the apparent *vena contracta* pressure at choked flow conditions. The factor F_L may be determined from tests in accordance with ANSI/ISA-75.02-1996. Typical values of F_L versus percent of rated flow coefficient are shown in Figure 4.

8.3.2 Combined liquid pressure recovery factor and piping geometry factor with attached fittings F_{LP}

F_{LP} is the combined liquid pressure recovery factor and piping geometry factor for a control valve with attached fittings. It is obtained in the same manner as F_L.

To meet a deviation of ±5% for F_{LP}, F_{LP} shall be determined by testing. When estimated values are permissible, the following equation shall be used:

Eq. 30 $$F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2}{N_2}\left(\Sigma\zeta_1\right)\left(\frac{C}{d^2}\right)^2}}$$

Here $\Sigma\zeta_1$ is the velocity head loss coefficient, $\zeta_1 + \zeta_{B1}$, of the fitting attached upstream of the valve as measured between the upstream pressure tap and the control valve body inlet.

8.4 Liquid critical pressure ratio factor F_F

F_F is the liquid critical pressure ratio factor. This factor is the ratio of the apparent *vena contracta* pressure at choked flow conditions to the vapor pressure of the liquid at inlet temperature. At vapor pressures near zero, this factor is 0.96.

Values of F_F may be determined from the curve in Figure 5 or approximated from the following equation:

Eq. 31 $$F_F = 0.96 - 0.28\sqrt{\frac{P_v}{P_c}}$$

8.5 Expansion factor Y

The expansion factor Y accounts for the change in density as the fluid passes from the valve inlet to the *vena contracta* (the location just downstream of the orifice where the jet stream area is a minimum). It also accounts for the change in the *vena contracta* area as the pressure differential is varied.

Theoretically, Y is affected by all of the following:

a) ratio of port area to body inlet area;

b) shape of the flow path;

c) pressure differential ratio x ;

d) Reynolds number; and

e) specific heat ratio γ.

The influence of items a), b), c), and e) is accounted for by the pressure differential ratio factor x_T, which may be established by air test and which is discussed in 8.6.1.

The Reynolds number is the ratio of inertial to viscous forces at the control valve orifice. In the case of compressible flow, its value is generally beyond the range of influence, except where the flow rate or the C_v is very low or a combination of both exist (see 7.2 and 8.2).

The pressure differential ratio x_T is influenced by the specific heat ratio of the fluid.

Y may be calculated using Equation 32.

Eq. 32 $$Y = 1 - \frac{x}{3F_\gamma x_T}$$

The value of x for calculation purposes shall not exceed $F_\gamma x_T$. If $x > F_\gamma x_T$, then the flow becomes choked and $Y = 0.667$. See 8.6 and 8.7 for information on x, x_T and F_γ.

8.6 Pressure differential ratio factor x_T or x_{TP}

8.6.1 Pressure differential ratio factor without fittings x_T

x_T is the pressure differential ratio factor of a control valve installed without reducers or other fittings. If the inlet pressure P_1 is held constant and the outlet pressure P_2 is progressively lowered, the mass flow rate through a valve will increase to a maximum limit, a condition referred to as choked flow. Further reductions in P_2 will produce no further increase in flow rate.

This limit is reached when the pressure differential x reaches a value of $F_\gamma x_T$. The limiting value of x is defined as the critical differential pressure ratio. The value of x used in any of the sizing equations and in the relationship for Y (Equation 32) shall be held to this limit even though the actual pressure differential ratio is greater. Thus, the numerical value of Y may range from 0.667, when $x = F_\gamma x_T$, to 1.0 for very low differential pressures.

The values of x_T may be established by air test. The test procedure for this determination is covered in ANSI/ISA-75.02-1996.

NOTE Representative values of x_T for several types of control valves with full size trim and at full rated openings are given in Table 2. Caution should be exercised in the use of this information. When precise values are required, they should be obtained by test.

8.6.2 Pressure differential ratio factor with attached fittings x_{TP}

If a control valve is installed with attached fittings, the value of x_T will be affected.

To meet a deviation of ±5% for x_{TP}, the valve and attached fittings shall be tested as a unit. When estimated values are permissible, the following equation shall be used:

Eq. 33
$$x_{TP} = \frac{\dfrac{x_T}{F_p^{\,2}}}{1 + \dfrac{x_T \zeta_i}{N_5}\left(\dfrac{C_i}{d^2}\right)^2}$$

NOTE Values for N_5 are given in Table 1.

In the above relationship, x_T is the pressure differential ratio factor for a control valve installed without reducers or other fittings. ζ_i is the sum of the inlet velocity head loss coefficients ($\zeta_1 + \zeta_{B1}$) of the reducer or other fitting attached to the inlet face of the valve.

If the inlet fitting is a short-length, commercially available reducer, the value of ζ may be estimated using Equation 23.

8.7 Specific heat ratio factor F_γ

The factor x_T is based on air near atmospheric pressure as the flowing fluid with a specific heat ratio of 1.40. If the specific heat ratio for the flowing fluid is not 1.40, the factor F_γ is used to adjust x_T. Use the following equation to calculate the specific heat ratio factor:

Eq. 34
$$F_\gamma = \frac{\gamma}{1.40}$$

NOTE See Annex C for values of γ and $F\gamma$.

8.8 Compressibility factor Z

Several of the sizing equations do not contain a term for the actual density of the fluid at upstream conditions. Instead, the density is inferred from the inlet pressure and temperature based on the laws of ideal gases. Under some conditions, real gas behavior can deviate markedly from the ideal. In these cases, the compressibility factor Z shall be introduced to compensate for the discrepancy. Z is a function of both the reduced pressure and reduced temperature (see appropriate reference books to determine Z). Reduced pressure P_r is defined as the ratio of the actual inlet absolute pressure to the absolute thermodynamic critical pressure for the fluid in question. The reduced temperature T_r is defined similarly. That is

Eq. 35 $$P_r = \frac{P_1}{P_c}$$

Eq. 36 $$T_r = \frac{T_1}{T_c}$$

NOTE See Annex C for values of P_c and T_c.

Table 1 — Numerical constants *N*

Constant	Flow coefficient C		Formulae unit						
	K_v	C_v	W	Q	P, ΔP	ρ	T	d, D	ν
N_1	1×10^{-1} 1	8.65×10^{-2} 8.65×10^{-1} 1	– – –	m³/h m³/h gpm	kPa bar psia	kg/m³ kg/m³ lbm/ft³	– – –	– – –	– – –
N_2	1.60×10^{-3}	2.14×10^{-3} 8.90×10^{2}	– –	– –	– –	– –	– –	mm in	– –
N_4	7.07×10^{-2}	7.60×10^{-2} 8.73×10^{4} 2.153×10^{4}	– – –	m³/h gpm scfh	– – –	– – –	– – –	– – –	m²/s cS cS
N_5	1.80×10^{-3}	2.41×10^{-3} 1.00×10^{3}	– –	– –	– –	– –	– –	mm in	– –
N_6	3.16 3.16×10^{1}	2.73 2.73×10^{1} 6.33×10^{1}	kg/h kg/h lbm/h	– – –	kPa bar psia	kg/m³ kg/m³ lbm/ft³	– – –	– – –	– – –
N_7	4.82 4.82×10^{2}	4.17 4.17×10^{2} 1.36×10^{3}	– – –	m³/h m³/h scfh	kPa bar psia	– – –	– – –	– – –	– – –
N_8	1.10 1.10×10^{2}	9.48×10^{-1} 9.48×10^{1} 1.93×10^{1}	kg/h kg/h lbm/h	– – –	kPa bar psia	– – –	K K R	– – –	– – –
N_9 (t_s = 0°C)	2.46×10^{1} 2.46×10^{3}	2.12×10^{1} 2.12×10^{3} 6.94×10^{3}	– – –	m³/h m³/h scfh	kPa bar psia	– – –	K K R	– – –	– – –
N_9 (t_s = 15°C)	2.60×10^{1} 2.60×10^{3}	2.25×10^{1} 2.25×10^{3} 7.32×10^{3}	– – –	m³/h m³/h scfh	kPa bar psia	– – –	K K R	– – –	– – –
N_{18}	8.65×10^{-1}	1.00 6.45×10^{2}	– –	– –	– –	– –	– –	mm in	– –
N_{19}	2.5	2.3 9.06×10^{-2}	– –	– –	– –	– –	– –	mm in	– –
N_{22} (t_s = 0°C)	1.73×10^{1} 1.73×10^{3}	1.50×10^{1} 1.50×10^{3} 4.92×10^{3}	– – –	m³/h m³/h scfh	kPa bar psia	– – –	K K R	– – –	– – –
N_{22} (t_s = 15°C)	1.84×10^{1} 1.84×10^{3}	1.59×10^{1} 1.59×10^{3} 5.20×10^{3}	– – –	m³/h m³/h scfh	kPa bar psia	– – –	K K R	– – –	– – –
N_{27} (t_s = 0°C)	7.75×10^{-1} 7.75×10^{-1}	6.70×10^{-1} 6.70×10^{-1} 1.37×10^{1}	kg/h kg/h lbm/h	– – –	kPa bar psia	– – –	K K R	– – –	– – –
N_{32}	1.40×10^{2}	1.27×10^{2} 1.70	– –	– –	– –	– –	– –	mm in	– –

NOTE Use of the numerical constants provided in this table together with the practical metric and US units specified in the table will yield flow coefficients in the units in which they are defined.

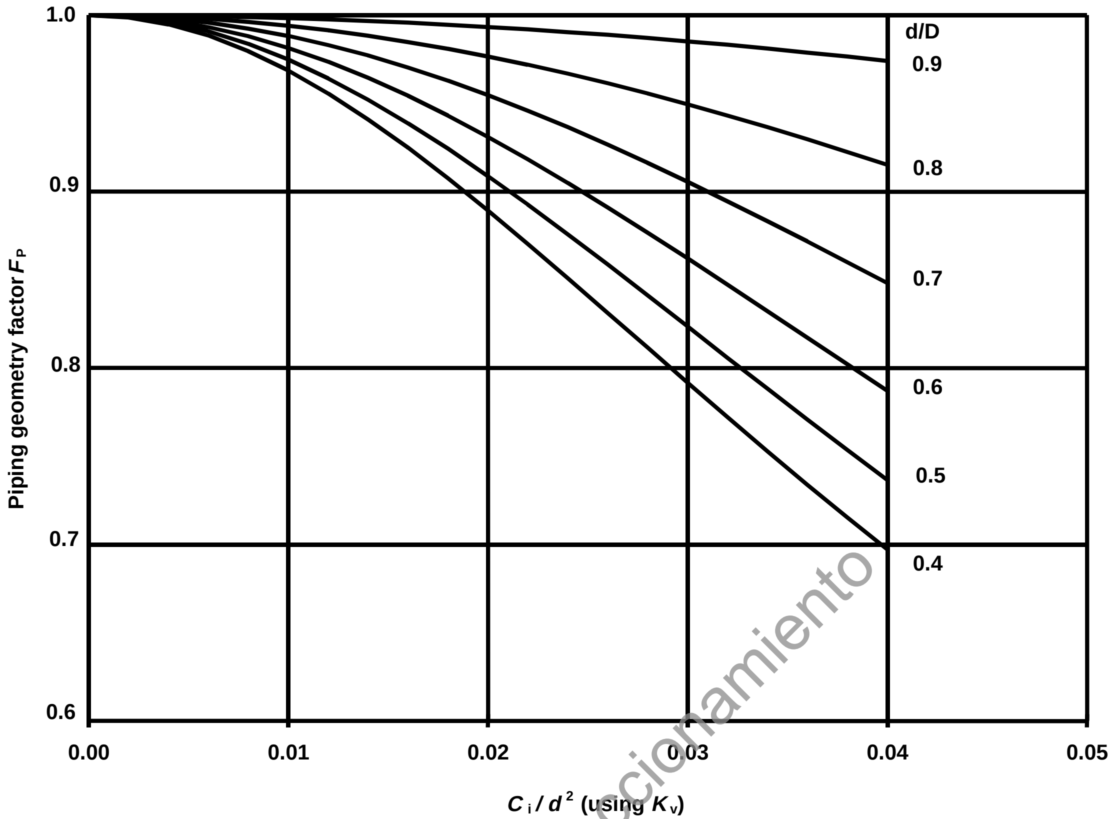
Table 2 — Typical values of valve style modifier F_d, liquid pressure recovery factor F_L, and pressure differential ratio factor x_T at full rated travel [1)]

Valve type	Trim type	Flow direction [2)]	F_L	x_T	F_d
Globe, single port	3 V-port plug	Open or close	0.9	0.70	0.48
	4 V-port plug	Open or close	0.9	0.70	0.41
	6 V-port plug	Open or close	0.9	0.70	0.30
	Contoured plug (linear and equal percentage)	Open Close	0.9 0.8	0.72 0.55	0.46 1.00
	60 equal diameter hole drilled cage	Outward [3)] or inward [3)]	0.9	0.68	0.13
	120 equal diameter hole drilled cage	Outward [3)] or inward [3)]	0.9	0.68	0.09
	Characterized cage, 4-port	Outward [3)] Inward [3)]	0.9 0.85	0.75 0.70	0.41 0.41
Globe, double port	Ported plug	Inlet between seats	0.9	0.75	0.28
	Contoured plug	Either direction	0.85	0.70	0.32
Globe, angle	Contoured plug (linear and equal percentage)	Open Close	0.9 0.8	0.72 0.65	0.46 1.00
	Characterized cage, 4-port	Outward [3)] Inward [3)]	0.9 0.85	0.65 0.60	0.41 0.41
	Venturi	Close	0.5	0.20	1.00
Globe, small flow trim	V-notch	Open	0.98	0.84	0.70
	Flat seat (short travel)	Close	0.85	0.70	0.30
	Tapered needle	Open	0.95	0.84	$N_{19}\dfrac{(CF_L)^{0.5}}{D_o}$
Rotary	Eccentric spherical plug	Open Close	0.85 0.68	0.60 0.40	0.42 0.42
	Eccentric conical plug	Open Close	0.77 0.79	0.54 0.55	0.44 0.44
Butterfly (centered shaft)	Swing-through (70°)	Either	0.62	0.35	0.57
	Swing-through (60°)	Either	0.70	0.42	0.50
	Fluted vane (70°)	Either	0.67	0.38	0.30
High Performance Butterfly (eccentric shaft)	Offset seat (70°)	Either	0.67	0.35	0.57
Ball	Full bore (70°)	Either	0.74	0.42	0.99
	Segmented ball	Either	0.60	0.30	0.98

1) These values are typical only; actual values shall be stated by the valve manufacturer.

2) Flow tends to open or close the valve, i.e. push the closure device (plug, ball, or disc) away from or towards the seat.

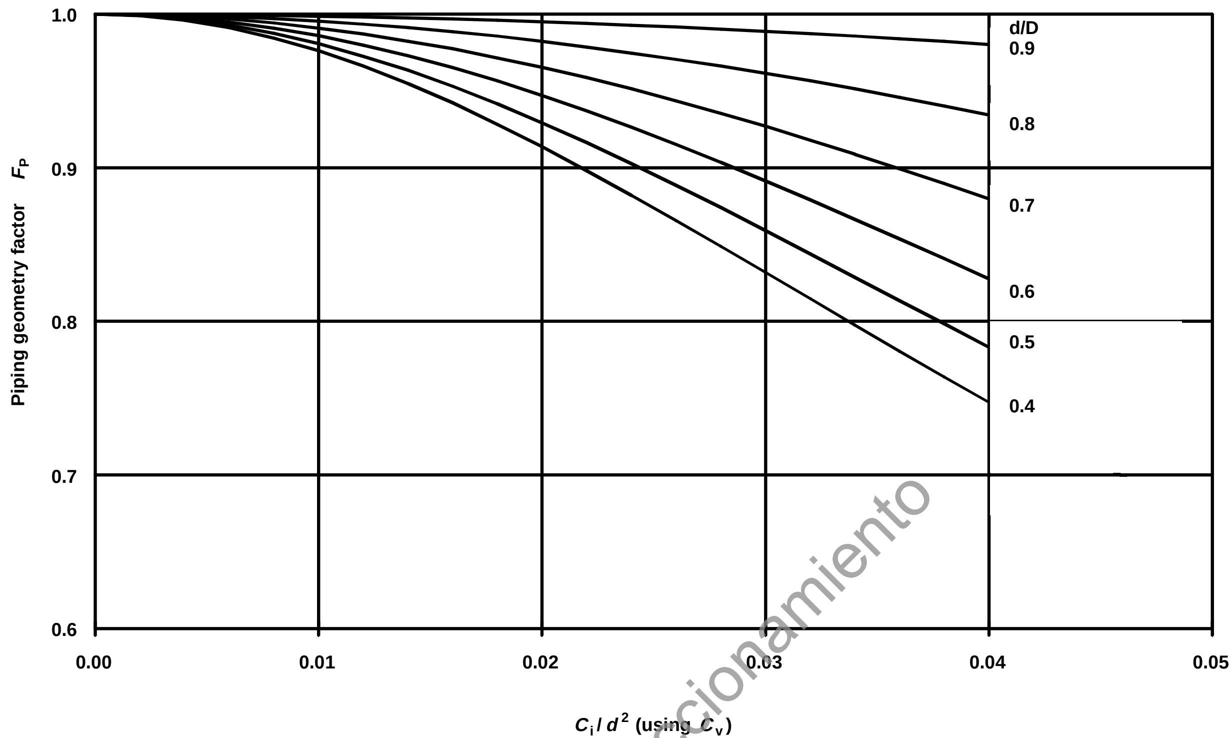
3) Outward means flow from center of cage to outside, and inward means flow from outside of cage to center.



NOTE 1 Pipe diameter D is the same size at both ends of the valve (see Equation 25).

NOTE 2 Refer to Annex F for example of the use of these curves.

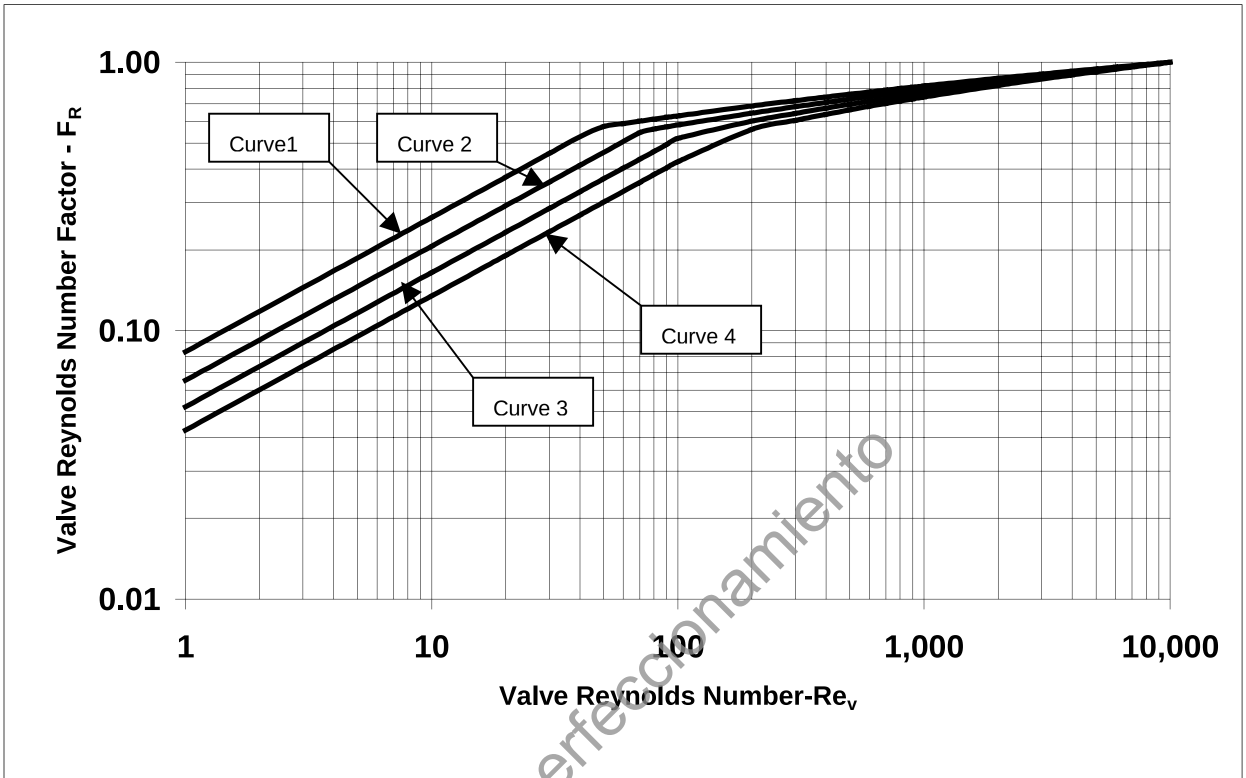
Figure 2a — Piping geometry factor F_P for K_v / d^2



NOTE 1 Pipe diameter D is the same size at both ends of the valve (see Equation 25).

NOTE 2 Refer to Annex F for example of the use of these curves.

Figure 2b — Piping geometry factor F_P for C_v / d^2



Curve 1 is for C_i/d^2 = 0.016 N_{18} and F_L= 0.9

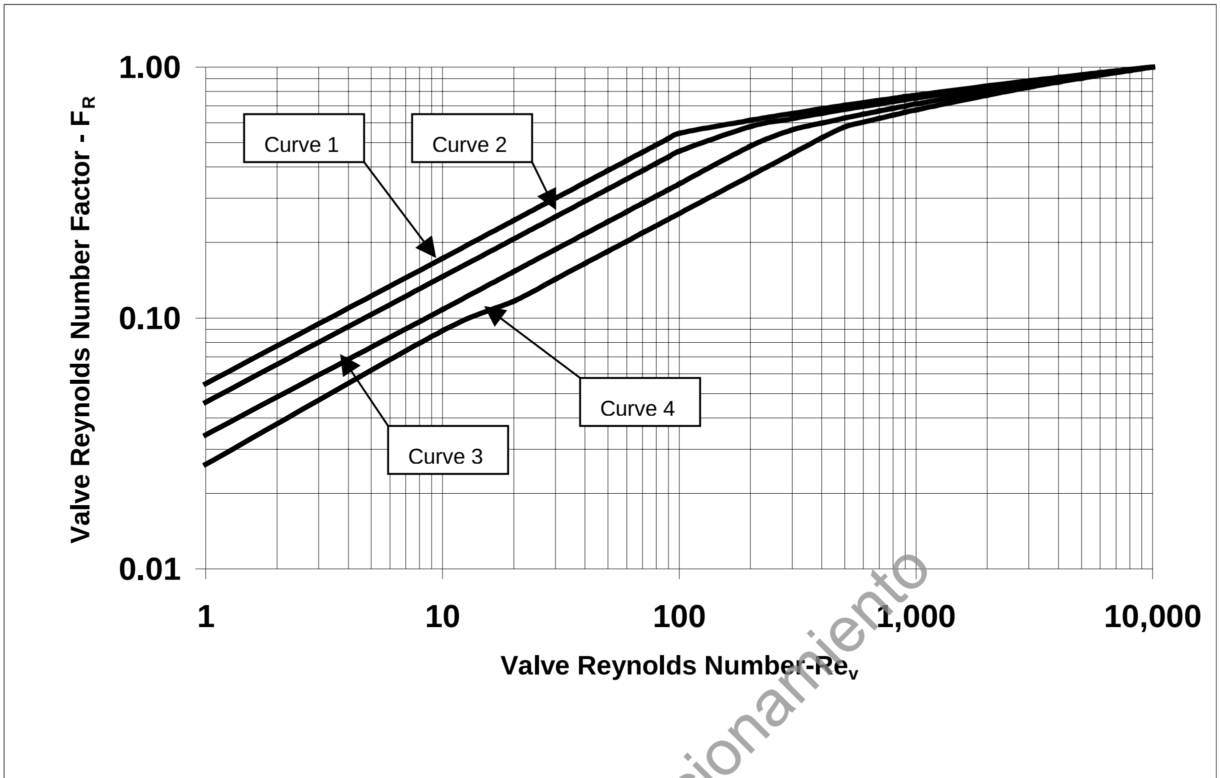
Curve 2 is for C_i/d^2 = 0.023 N_{18} and F_L= 0.8

Curve 3 is for C_i/d^2 = 0.033 N_{18} and F_L= 0.7

Curve 4 is for C_i/d^2 = 0.047 N_{18} and F_L= 0.6

The F_L values shown are considered typical for the respective C_i/d^2 ratios.

Figure 3a — Reynolds number factor F_R for full-size trim valves (reference Annex G)



Curve 1 is for C_i/d^2 = 0.00444 N_{18}

Curve 2 is for C_i/d^2 = 0.00222 N_{18}

Curve 3 is for C_i/d^2 = 0.00044 N_{18}

Curve 4 is for $C_i/d^2 \leq$ 0.0000004 N_{18}

Curves are based on F_L being approximately 1.0.

Figure 3b — Reynolds number factor F_R for reduced trim valves (applicable to low flow / small C_v control valves) (reference Annex G)

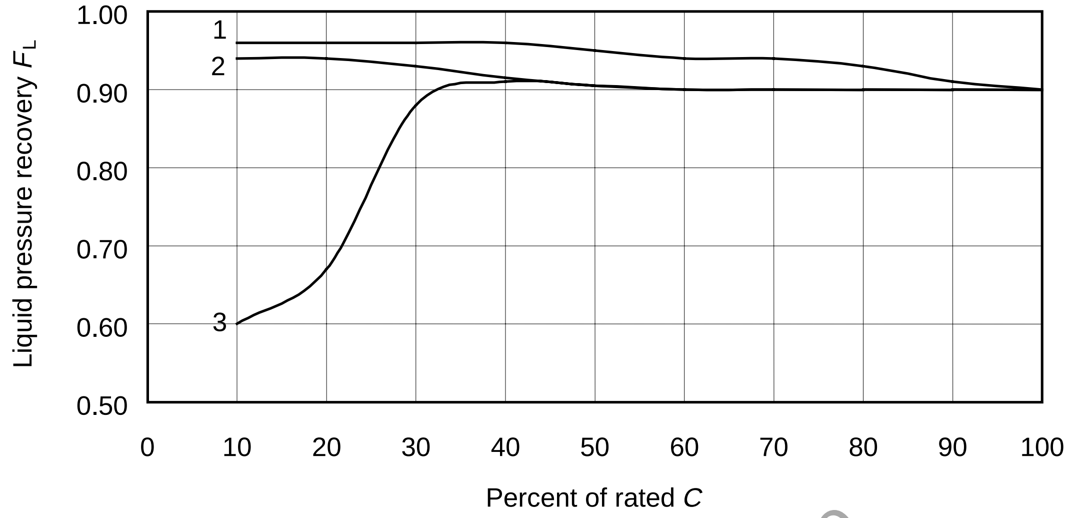


Figure 4a — Double seated globe valves and cage guided globe valves (see legend)

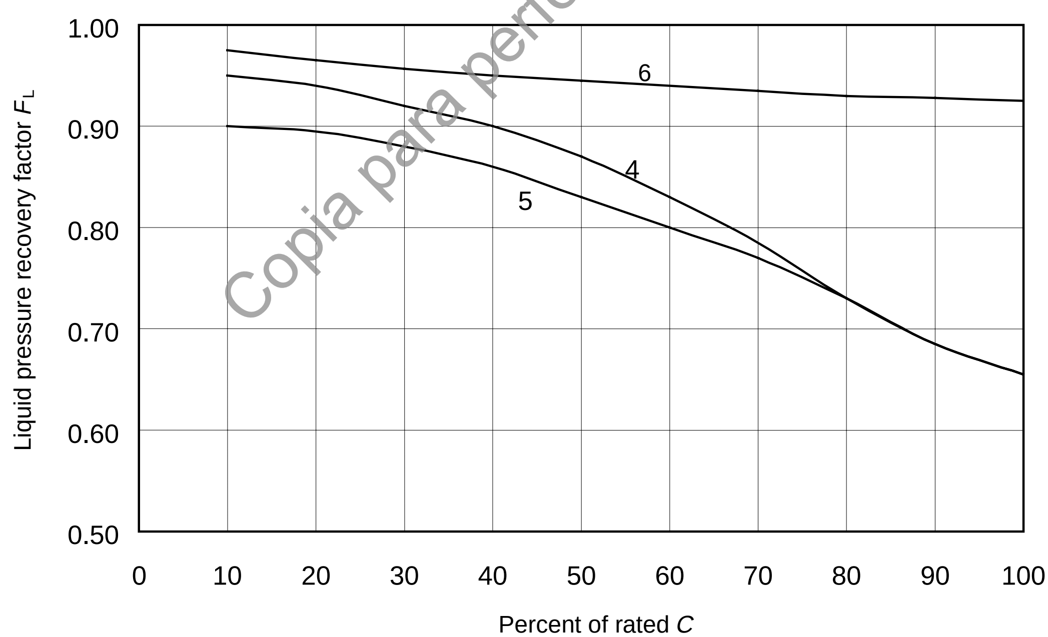


Figure 4b — Butterfly valves and contoured plug, low flow valves (see legend)

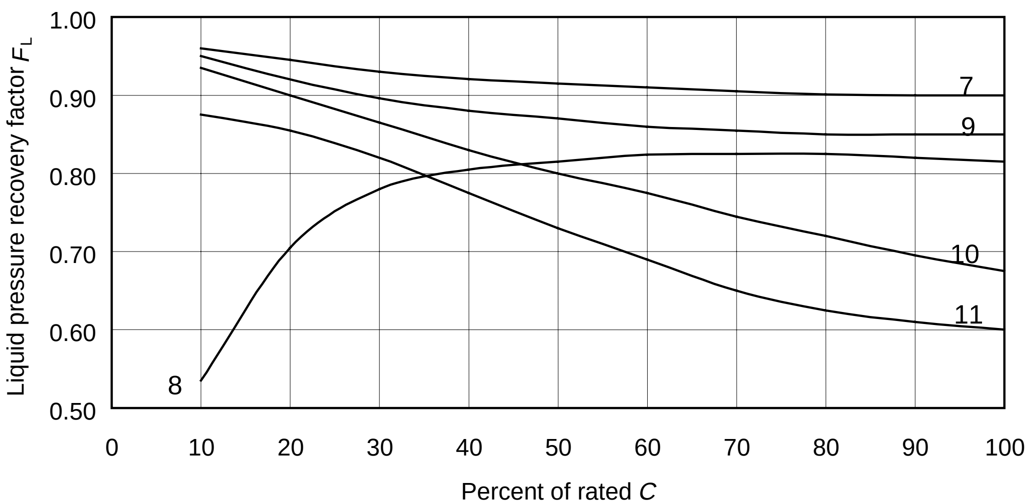


Figure 4c — Contoured globe valves, eccentric spherical plug valves, and segmented ball valves (see legend)

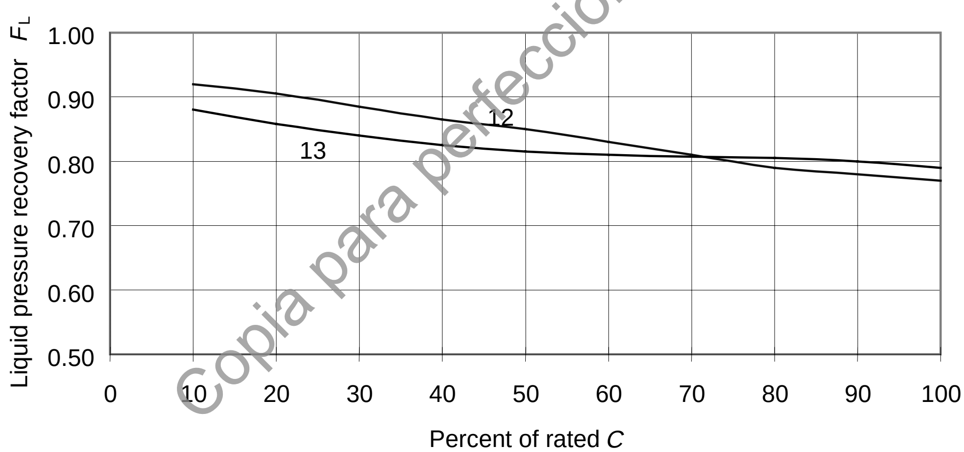


Figure 4d — Eccentric conical plug valves (see legend)

Legend

1 Double seated globe valve, V-port plug

2 Ported cage guided globe valve (flow-to-open and flow-to-close)

3 Double seated globe valve, contoured plug

4 Offset seat butterfly valve

5 Swing-through butterfly valve

6 Contoured plug, low flow valve

7 Single port, equal percentage, contoured globe valve, flow-to-open

8 Single port, equal percentage, contoured globe valve, flow-to-close

9 Eccentric spherical plug valve, flow-to-open

10 Eccentric spherical plug valve, flow-to-close

11 Segmented ball valve

12 Eccentric conical plug valve, flow-to-open

13 Eccentric conical plug valve, flow-to-close

NOTE These values are typical only; actual values shall be stated by the manufacturer.

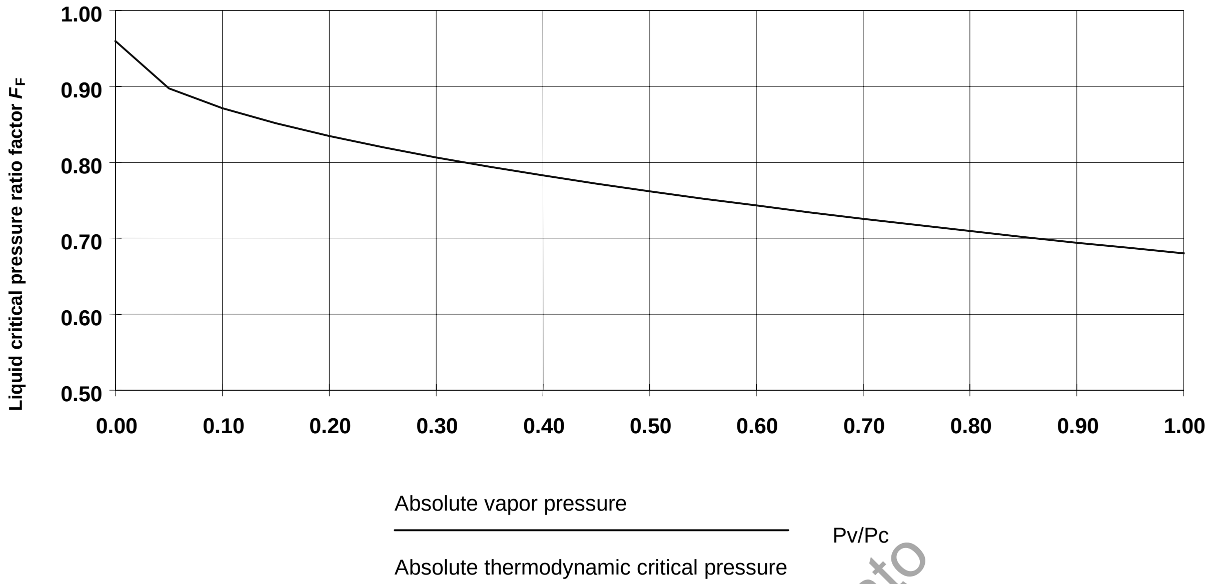


Figure 5 — Liquid critical pressure ratio factor F_F

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Annex A(informative) — Derivation of valve style modifier F_d

All variables in this annex have been defined in this part except for the following:

A_o area of *vena contracta* of a single flow passage, millimeters squared;

d_H hydraulic diameter of a single flow passage, millimeters;

d_i inside diameter of annular flow passage (see Figure A.1), millimeters;

d_o equivalent circular diameter of the total flow area, millimeters;

D_o diameter of seat orifice (see Figures A.1 and A.2), millimeters;

l_w wetted perimeter of a single flow passage, millimeters;

N_o number of independent and identical flow passages of a trim, dimensionless;

α angular rotation of closure member (see Figure A.2), degrees;

β maximum angular rotation of closure member (see Figure A.2), degrees;

ζ_{B1} velocity of approach factor, dimensionless; and

μ discharge coefficient, dimensionless.

The valve style modifier F_d, defined as the ratio d_H / d_o at rated travel and where $C_i / d^2 > 0.016\ N_{18}$, may be derived from flow tests using the following equation:

Eq. A.1
$$F_d = \frac{N_{26}\,\nu\,F_L^{\,2} F_R^{\,2} \left(C / d^2\right)^2 \sqrt{C F_L}}{Q\left(\dfrac{F_L^{\,2} C^2}{N_2 D^4} + 1\right)^{1/4}}$$

For valves having $C_i / d^2 \leq 0.016\ N_{18}$, F_d is calculated as follows:

Eq. A.2
$$F_d = \frac{N_{31}\,\nu\,F_L^{\,2} F_R^{\,2} \sqrt{C F_L}}{Q\left[1 + N_{32}\left(\dfrac{C}{d^2}\right)^{2/3}\right]}$$

NOTE Values for N_{26} and N_{32} are listed in Table A.1.

The test for determining F_d is covered in IEC 60534-2-3.

Alternatively, F_d can be calculated by the following equation:

Eq. A.3 $$F_d = \frac{d_H}{d_o}$$

The hydraulic diameter d_H of a single flow passage is determined as follows:

Eq. A.4 $$d_H = \frac{4\,A_o}{l_w}$$

The equivalent circular diameter d_o of the total flow area is given by the following equation:

Eq. A.5 $$d_o = \sqrt{\frac{4N_o A_o}{\pi}}$$

F_d may be estimated with sufficient accuracy from dimensions given in manufacturers' drawings.

The valve style modifier for a single-seated, parabolic valve plug (flow tending to open) (see Figure A.1) may be calculated from Equation A.3.

From Darcey's equation, the area A_o is calculated from the following equation:

Eq. A.6 $$A_o = \frac{N_{23} C F_L}{N_o}$$

NOTE Values for N_{23} are listed in Table A.1.

Therefore, since $N_o = 1$,

Eq. A.7 $$d_o = \sqrt{\frac{4A_o}{\pi}}$$

$$= \sqrt{\frac{4N_{23}\ CF_L}{\pi}}$$

Eq. A.8 $$d_H = \frac{4A_o}{l_w}$$

$$= \frac{4N_{23} CF_L}{\pi\left(D_o + d_i\right)}$$

From above,

Eq. A.3 $$F_d = \frac{d_H}{d_o}$$

$$= \frac{\left[\dfrac{4N_{23}CF_L}{\pi\left(D_o + d_i\right)}\right]}{\sqrt{\dfrac{4N_{23}CF_L}{\pi}}}$$

Eq. A.9 $$= \frac{1.13\sqrt{N_{23}CF_L}}{D_o + d_i}$$

where d_i varies with the flow coefficient. The diameter d_i is assumed to be equal to zero when $N_{23}CF_L = D_o^2$. At low C values, $d_i \approx D_o$; therefore,

Eq A.10 $$d_i = D_o - \frac{N_{23}CF_L}{D_o}$$

Eq. A.11 $$F_d = \frac{1.13\sqrt{N_{23}CF_L}}{2D_o - \dfrac{N_{23}CF_L}{D_o}}$$

The maximum F_d is 1.0.

For swing-through butterfly valves, see Figure A.2.

The effective orifice diameter is assumed to be the hydraulic diameter of one of the two jets emanating from the flow areas between the disk and valve body bore; hence $N_o = 2$.

The flow coefficient C at choked or sonic flow conditions is given as

Eq. A.12 $$N_{23}CF_L = \frac{0.125\pi D_o^{\,2}\left(\mu_1 + \mu_2\right)\left(\dfrac{1-\sin\alpha}{\sin\beta}\right)}{\zeta_{B1}}$$

Assuming the velocity of approach factor $\zeta_{B1} = 1$, making $\mu_1 = 0.7$ and $\mu_2 = 0.7$, and substituting equation A.6 into Equation A.12 yields Equation A.13.

Eq. A.13 $$A_o = \frac{0.55D_o^{\,2}\left(\dfrac{1-\sin\alpha}{\sin\beta}\right)}{N_o}$$