Flow Equations for Sizing Control Valves
Preface

This preface, as well as all footnotes and annexes, is included for information purposes and is not part of ISA-75.01.01-2007 (IEC 60534-2-1 Mod).

The standards referenced within this document may contain provisions which, through reference in this text, constitute requirements of this document. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this document are encouraged to investigate the possibility of applying the most recent editions of the standards indicated within this document. Members of IEC and ISO maintain registers of currently valid International Standards. ANSI maintains registers of currently valid U.S. National Standards.

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FOREWORD

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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:

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<th>Report on voting</th>
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<td>65B/347/FDIS</td>
<td>65B/357/RVD</td>
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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, E, F, and G are for information only.

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

ISA-75.01.01-2007 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 ISA-75.01.01-2007 is valid for all valves. However, manufacturers of some valves with $x_T \geq 0.84$ have reported minor inaccuracies (see Annex G). 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 $\pm 5\%$ 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 ISA-75.01.01-2007. 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 ISA-75.01.01-2007 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 60534-1:2005, Industrial-process control valves – Part 1: Control valve terminology and general considerations


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

ANSI/ISA-75.05.01-2000 (R2005), Control Valve Terminology

3 Definitions

For the purpose of ISA-75.01.01-2007, 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.

$\ell_1 = \text{two nominal pipe diameters}$

$\ell_2 = \text{six nominal pipe diameters}$

**Figure 1 — Reference pipe section for sizing**
## 5 Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>C</td>
<td>Flow coefficient ((K_v, C_v))</td>
<td>Various (see IEC 60534-1)</td>
</tr>
<tr>
<td>C_i</td>
<td>Assumed flow coefficient for iterative purposes</td>
<td>Various (see IEC 60534-1)</td>
</tr>
<tr>
<td>d</td>
<td>Nominal valve size</td>
<td>mm (in)</td>
</tr>
<tr>
<td>D</td>
<td>Internal diameter of the piping</td>
<td>mm (in)</td>
</tr>
<tr>
<td>D_1</td>
<td>Internal diameter of upstream piping</td>
<td>mm (in)</td>
</tr>
<tr>
<td>D_2</td>
<td>Internal diameter of downstream piping</td>
<td>mm (in)</td>
</tr>
<tr>
<td>D_o</td>
<td>Orifice diameter</td>
<td>mm (in)</td>
</tr>
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<td>F_d</td>
<td>Valve style modifier (see Annex A)</td>
<td>Dimensionless</td>
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<tr>
<td>F_F</td>
<td>Liquid critical pressure ratio factor</td>
<td>Dimensionless</td>
</tr>
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<td>F_L</td>
<td>Liquid pressure recovery factor of a control valve without attached fittings</td>
<td>Dimensionless</td>
</tr>
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<td>F_LP</td>
<td>Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings</td>
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<td>F_P</td>
<td>Piping geometry factor</td>
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</tr>
<tr>
<td>F_R</td>
<td>Reynolds number factor</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>F_g</td>
<td>Specific heat ratio factor</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>G_g</td>
<td>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</td>
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<tr>
<td>M</td>
<td>Molecular mass of flowing fluid</td>
<td>kg/kg-mol (lb/lb-mol)</td>
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<tr>
<td>N</td>
<td>Numerical constants (see Table 1)</td>
<td>Various (see note 1)</td>
</tr>
<tr>
<td>P_1</td>
<td>Inlet absolute static pressure measured at point A (see Figure 1)</td>
<td>kPa or bar (psia)(see note 2)</td>
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<tr>
<td>P_2</td>
<td>Outlet absolute static pressure measured at point B (see Figure 1)</td>
<td>kPa or bar (psia)</td>
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<td>P_C</td>
<td>Absolute thermodynamic critical pressure</td>
<td>kPa or bar (psia)</td>
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<td>P_r</td>
<td>Reduced pressure ((P_1 / P_C))</td>
<td>Dimensionless</td>
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<td>P_v</td>
<td>Absolute vapor pressure of the liquid at inlet temperature</td>
<td>kPa or bar (psia)</td>
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<td>ΔP</td>
<td>Differential pressure between upstream and downstream pressure taps ((P_1 – P_2))</td>
<td>kPa or bar (psia)</td>
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<td>Q</td>
<td>Volumetric flow rate (see note 5)</td>
<td>m³/h (gpm, scfh)</td>
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<td>Re_v</td>
<td>Valve Reynolds number</td>
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<td>T_1</td>
<td>Inlet absolute temperature</td>
<td>K (R)</td>
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<tr>
<td>T_C</td>
<td>Absolute thermodynamic critical temperature</td>
<td>K (R)</td>
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<td>T_r</td>
<td>Reduced temperature ((T_1 / T_C))</td>
<td>Dimensionless</td>
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<tr>
<td>t_s</td>
<td>Absolute reference temperature for standard cubic meter</td>
<td>K (R)</td>
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<tr>
<td>W</td>
<td>Mass flow rate</td>
<td>kg/h (lbsm/h)</td>
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<td>X</td>
<td>Ratio of pressure differential to inlet absolute pressure ((ΔP / P_1))</td>
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<td>X_T</td>
<td>Pressure differential ratio factor of a control valve without attached fittings at choked flow</td>
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<td>Pressure differential ratio factor of a control valve with attached fittings at choked flow</td>
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<td>Y</td>
<td>Expansion factor</td>
<td>Dimensionless</td>
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<td>Z</td>
<td>Compressibility factor</td>
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<td>ν</td>
<td>Kinematic viscosity</td>
<td>m²/s (cS) (see note 3)</td>
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<td>ρ_1</td>
<td>Density of fluid at (P_1) and (T_1)</td>
<td>kg/m³ (lb/ft³)</td>
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<td>ρ_1/ρ_0</td>
<td>Relative density ((ρ_1/ρ_0 = 1.0) for water at 15°C)</td>
<td>Dimensionless</td>
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<td>γ</td>
<td>Specific heat ratio</td>
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<td>ζ</td>
<td>Velocity head loss coefficient of a reducer, expander or other fitting attached to a control valve or valve trim</td>
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</table>
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

\[
\left( \text{Applicable if } \Delta P < F_L^2 (P_1 - F_T P_v) \right)
\]

The flow coefficient shall be determined by

\[
C = \frac{Q}{N_1} \sqrt{\frac{\rho_1 / \rho_o}{\Delta P}}
\]

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 E.

6.1.1.2 Non-choked turbulent flow with attached fittings

\[
\left\{ \text{Applicable if } \Delta P < \left( F_{Lp} / F_p \right)^2 (P_1 - F_T P_v) \right\}
\]

The flow coefficient shall be determined as follows:

\[
C = \frac{Q}{N_1 F_p} \sqrt{\frac{\rho_1 / \rho_o}{\Delta P}}
\]
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

\[
\text{Applicable if } \Delta P \geq F_L^2 \left( P_i - F_P P_v \right)
\]

The flow coefficient shall be determined as follows:

Eq. 3

\[
C = \frac{Q}{N_i F_L} \sqrt{\frac{\rho_1 / \rho_o}{P_i - F_P P_v}}
\]

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

6.1.2.2 Choked turbulent flow with attached fittings

\[
\text{Applicable if } \Delta P \geq \left( F_{LP} / F_P \right)^2 \left( P_i - F_P P_v \right)
\]

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

Eq. 4

\[
C = \frac{Q}{N_i F_{LP}} \sqrt{\frac{\rho_1 / \rho_o}{P_i - F_P P_v}}
\]

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

\[
C = \frac{Q}{N_i F_R} \sqrt{\frac{\rho_1 / \rho_o}{\Delta P}}
\]

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

(Applicable if $x < F_1x_T$)

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

Eq. 6  \[ C = \frac{W}{N_s Y P_1 P_1} \]

Eq. 7  \[ C = \frac{W}{N_s P_1 Y} \left( \frac{T_1 Z}{x M} \right) \]

Eq. 8a  \[ C = \frac{Q}{N_s P_1 Y} \left( \frac{M T_1 Z}{x} \right) \]

Eq. 8b  \[ C = \frac{Q}{N_s P_1 Y} \left( \frac{G g T_1 Z}{x} \right) \]

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

(Applicable if $x < F_1x_{TP}$)
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  \[
C = \frac{W}{N_6 F_p Y \sqrt{\frac{T_1 Z}{x M}}}
\]

Eq. 11a  \[
C = \frac{Q}{N_6 F_p Y \sqrt{\frac{MT_1 Z}{x}}}
\]

Eq. 11b  \[
C = \frac{Q}{N_6 F_p Y \sqrt{\frac{G g T_1 Z}{x}}}
\]

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 E.

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

\[\text{Applicable if } x \geq F_T x_T\]

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

Eq. 12  \[
C = \frac{W}{0.667 N_6 \sqrt{F_T x_T P_1 \rho_1}}
\]

Eq. 13  \[
C = \frac{W}{0.667 N_6 P_1 \sqrt{\frac{T_1 Z}{F_T x_T M}}}
\]

Eq. 14a  \[
C = \frac{Q}{0.667 N_6 P_1 \sqrt{\frac{MT_1 Z}{F_T x_T}}}
\]

Eq. 14b  \[
C = \frac{Q}{0.667 N_6 P_1 \sqrt{\frac{G g T_1 Z}{F_T x_T}}}
\]

7.1.2.2 Choked turbulent flow with attached fittings

\[\text{Applicable if } x \geq F_T x_{TP}\]
The flow coefficient shall be determined using one of the following equations:

**Eq. 15**

\[
C = \frac{W}{0.667N_c F_p \sqrt{F_1 x_{tp} P_1 P_1}}
\]

**Eq. 16**

\[
C = \frac{W}{0.667N_c F_p P_1 \sqrt{T_1 Z}}
\]

**Eq. 17a**

\[
C = \frac{Q}{0.667N_c F_p P_1 \sqrt{MT_1 Z}}
\]

**Eq. 17b**

\[
C = \frac{Q}{0.667N_c F_p P_1 \sqrt{GgT_1 Z}}
\]

### 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 \( (\frac{P_1 + P_2}{2}) \) 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**

\[
C = \frac{W}{N_{27} F_R \sqrt{\frac{T_1}{\Delta P(P_1 + P_2)}}}
\]

**Eq. 19**

\[
C = \frac{Q}{N_{22} F_R \sqrt{\frac{MT_1}{\Delta P(P_1 + P_2)}}}
\]

**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**  \[ F_P = \frac{1}{\sqrt{1 + \frac{\Sigma \zeta}{N_2} \left( \frac{C_1}{d^2} \right)^2}} \]

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 $\zeta_B$ 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 $\zeta_1$ and $\zeta_2$ coefficients may be approximated as follows:

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

**Eq. 24**  \[ \zeta_2 = 1.0 \left( 1 - \left( \frac{d}{D_2} \right)^2 \right)^2 \]

**Eq. 25**  \[ \zeta_1 + \zeta_2 = 1.5 \left( 1 - \left( \frac{d}{D} \right)^2 \right)^2 \]

The $F_P$ values calculated with the above $\zeta$ 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:

\[
\text{Eq. 26} \quad C_i = 1.3C
\]

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 2a or 2b. Then, determine if

\[
\text{Eq. 27} \quad \frac{C}{F_P} \leq C_i
\]

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:

\[
\text{Eq. 28} \quad Re_v = \frac{N_2 F_{d} Q}{\sqrt{C_i F_L}} \left( \frac{F_{L}^{-2} C_{i}^{2}}{N_2 D^4} + 1 \right)^{1/4}
\]

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 than 0.016 $N_{18}$.

\[
\text{Eq. 29} \quad \frac{C}{F_R} \leq C_i
\]
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 F.

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:

$$E \text{q. 30} \quad F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2 \left( \Sigma \left( \zeta_1 + \zeta_{B1} \right) \right)}{N_2 \left( \frac{C}{d^2} \right)}}}$$

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:

$$E \text{q. 31} \quad F_F = 0.96 - 0.28 \frac{P_v}{P_c}$$

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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 $\gamma$.

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 $F_\gamma$.

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:

\[
x_{TP} = \frac{x_T}{F_p^2} \left( 1 + \frac{x_T \zeta_i}{N_5 \left( \frac{C_L}{d^2} \right)^2} \right)
\]

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. \( \zeta_i \) is the sum of the inlet velocity head loss coefficients \( (\zeta + \zeta_{B1}) \) 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 \( \zeta \) may be estimated using Equation 23.

8.7 Specific heat ratio factor \( F_\gamma \)

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 \( F_\gamma \) is used to adjust \( x_T \). Use the following equation to calculate the specific heat ratio factor:

\[
F_\gamma = \frac{\gamma}{1.40}
\]

NOTE See Annex C for values of \( \gamma \) 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 \( T_c \).
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<td></td>
<td>$5.20 \times 10^3$</td>
<td>–</td>
</tr>
<tr>
<td>$N_{27}$</td>
<td>$7.75 \times 10^1$</td>
<td>$6.70 \times 10^1$</td>
</tr>
<tr>
<td></td>
<td>$7.75 \times 10^1$</td>
<td>$6.70 \times 10^1$</td>
</tr>
<tr>
<td></td>
<td>$1.37 \times 10^1$</td>
<td>lbm/h</td>
</tr>
<tr>
<td></td>
<td>$1.37 \times 10^1$</td>
<td>lbm/h</td>
</tr>
<tr>
<td>$N_{32}$</td>
<td>$1.40 \times 10^2$</td>
<td>$1.27 \times 10^2$</td>
</tr>
<tr>
<td></td>
<td>$1.70 \times 10^1$</td>
<td>–</td>
</tr>
</tbody>
</table>

**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

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

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 E 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 E for example of the use of these curves.

**Figure 2b — Piping geometry factor $F_P$ for $C_v / d^2$**
Figure 3a — Reynolds number factor $F_R$ for full-size trim valves
(reference Annex F)
Figure 3b — Reynolds number factor $F_R$ for reduced trim valves
(applicable to low flow / small $C_v$ control valves)
(reference Annex F)
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;
- $\alpha$ angular rotation of closure member (see Figure A.2), degrees;
- $\beta$ maximum angular rotation of closure member (see Figure A.2), degrees;
- $\zeta_{B1}$ velocity of approach factor, dimensionless; and
- $\mu$ 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:

$$F_d = \frac{N_{26} \sqrt{F_L} F_R^2 \left( C / d^2 \right)^2 \sqrt{CF_L}}{Q \left( F_L^2 C^2 / N_o D^4 + 1 \right)^{1/4}}$$

Eq. A.1

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

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

Eq. A.2

NOTE Values for $N_{30}$ 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:

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

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

$$d_H = \frac{4A_o}{l_w}$$

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

$$d_o = \sqrt{\frac{4N_oA_o}{\pi}}$$

$F_d$ 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:

$$A_o = \frac{N_{23}CF_L}{N_o}$$

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

Therefore, since $N_o = 1$, 

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

$$d_H = \frac{4A_o}{l_w} = \frac{4N_{23}CF_L}{\pi(D_o + d_i)}$$
From above,

\[ F_d = \frac{d_h}{d_o} \]

\[ = \frac{4N_{23} CF_L}{\pi (D_o + d_i)} \]

\[ = \sqrt{\frac{4N_{23} CF_L}{\pi}} \]

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

\[ F_d = \frac{1.13 \sqrt{N_{23} CF_L}}{2D_o - \frac{N_{23} CF_L}{D_o}} \]

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

\[ N_{23} CF_L = \frac{0.125 \pi D_o^2 (\mu_1 + \mu_2) \left( \frac{1 - \sin \alpha}{\sin \beta} \right)}{\zeta_{B1}} \]

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

\[ A_o = \frac{0.55D_o^2 \left( \frac{1 - \sin \alpha}{\sin \beta} \right)}{N_o} \]
and since $\beta = 90^\circ$ for swing-through butterfly valves,

Eq. A.14 \quad A_o = \frac{0.55D_o^2 (1 - \sin \alpha)}{N_o}

However, since there are two equal flow areas in parallel,

Eq. A.15 \quad A_o = 0.275D_o^2 (1 - \sin \alpha)

and \quad d_o = \sqrt{\frac{4A_oN_o}{\pi}}

Eq. A.16 \quad = 0.837D_o \sqrt{1 - \sin \alpha}

\begin{align*}
 d_H &= \frac{4A_o}{0.59\pi D_o} \\
 N_{A.17} &= 0.59D_o (1 - \sin \alpha)
\end{align*}

NOTE \quad 0.59 \pi D_o is taken as the wetted perimeter $l_w$ of each semi-circle allowing for jet contraction and hub.

Eq. A.3 \quad F_d = \frac{d_h}{d_o}

which results in

Eq. A.18 \quad F_d = 0.7 \sqrt{1 - \sin \alpha}

<table>
<thead>
<tr>
<th>Table A.1 — Numerical constant $N$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant</td>
</tr>
<tr>
<td>$N_{23}$</td>
</tr>
<tr>
<td>$N_{26}$</td>
</tr>
<tr>
<td>$N_{31}$</td>
</tr>
</tbody>
</table>

NOTE \quad Use of the numerical constant provided in this table together with the practical metric units specified in the table will yield flow coefficients in the units in which they are defined.
Figure A.1 — Single seated, parabolic plug (flow tending to open)

Figure A.2 — Swing-through butterfly valve
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Annex B (informative) — Control valve sizing flow charts

B.1 Incompressible fluids

1. Start
2. Select $F_L$ using valve type and size *
3. Calculate $F_F$ using eq. 31
4. If $\Delta P < F_L^2(P_1 - F_F P_v)$:
   - Non-choked flow: Calculate $C$ using eq. 1
   - Choked flow: Calculate $C$ using eq. 3
5. Calculate $Re_v$ using eq. 28
6. Use $C$ as $C_i$
7. Use $F_d$ from table 2
8. If $Re_v > 10,000$:
   - Yes: Use calculated $C$
   - No: Calculate $F_{dp}(C_i)$ and $F_{dp}(C_i)$
9. If $\Delta P > (F_{dp}/F_P)^{2}(P_1 - F_F P_v)$:
   - Yes: Calculate $C$ using eq. 4
   - No: Calculate $C$ using eq. 2
10. If $C/C < 0.99$:
    - Yes: Use calculated $C$
    - No: $C_i = C$

* When in doubt, use inlet pipe size as valve size
B.1 Incompressible fluids (continued)

Non-turbulent flow
Establish \( C_i = 1.3C \)
Calculate \( Re_v \)
using eq. 28

Calculate \( F_R \) from Figure 3a or 3b or as the lower value of eq. F.3a & eq. F.4 from Annex F. [If \( Re_v < 10 \), use \( F_R \) from F.4 from Annex F.]

\[ \frac{C}{d} > 0.016N_{tB} \]

Yes

Calculate \( F_R \) from Figure 3a or 3b or as the lower value of eq. F.1a & eq. F.2 from Annex F. [If \( Re_v < 10 \), use \( F_R \) from F.2 from Annex F.]

\[ C/F_R \leq C_i \]

No

Increase \( C_i \) by 30 %

Yes

Use \( C_i \) as flow coefficient
B.2 Compressible fluids

Start

Select \( x_T \) using valve type and size *

Calculate \( F_\gamma \) using eq. 34

Yes

\[ x < F_\gamma x_T \]

Non-choked flow
Calculate \( Y \) using eq. 32
Calculate \( C \) using 6, 7 or 8

Calculate \( Re_v \) using eq. 28
Use \( C \) as \( C_i \)
Use \( F_{d} \) from table 2

Yes

\[ Re_v > 10,000 \]

Valve size = pipe size

No

No

Choked flow
\[ Y = 0.667 \]
Calculate \( C \) using eq. 12, 13 or 14

Calculate \( F_P \) using eq. 15, 16 or 17

Yes

\[ C_i/C < 0.99 \]

Yes

\( C_i = C \)

No

\( C_i = C \)

Use calculated \( C \)

(continued)

\* When in doubt, use inlet pipe size as valve size
B.2 Compressible fluids continued

Non-turbulent flow
Establish $C_i = 1.3 C$
Calculate $Re_v$ using eq. (28)

Calculate $F_R$ from Figure 3a or 3b or as the lower value of eq. F.3a & eq.F.4 from Annex F. [If $Re_v < 10$, use $F_R$ from F.4 from Annex F.]

$C/d^2 > 0.016 N_{18}$

No

Yes

$C/F_R =< C_i$

No

Increase $C_i$ by 30 %

Yes

Use $C_i$ as flow coefficient

(continued)
### Table: Physical Constants

<table>
<thead>
<tr>
<th>Gas or vapor</th>
<th>Symbol</th>
<th>$M$</th>
<th>$\gamma$</th>
<th>$F_\gamma$</th>
<th>$P_c$</th>
<th>$T_c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acetylene</td>
<td>C$_2$H$_2$</td>
<td>26.04</td>
<td>1.30</td>
<td>0.929</td>
<td>6,140</td>
<td>309</td>
</tr>
<tr>
<td>Air</td>
<td>–</td>
<td>28.97</td>
<td>1.40</td>
<td>1.000</td>
<td>3,771</td>
<td>133</td>
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<tr>
<td>Ammonia</td>
<td>NH$_3$</td>
<td>17.03</td>
<td>1.32</td>
<td>0.943</td>
<td>11,400</td>
<td>406</td>
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<td>Argon</td>
<td>A</td>
<td>39.948</td>
<td>1.67</td>
<td>1.191</td>
<td>4,870</td>
<td>151</td>
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<td>Benzene</td>
<td>C$_6$H$_6$</td>
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<td>1.12</td>
<td>0.800</td>
<td>4,924</td>
<td>562</td>
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<td>Isobutane</td>
<td>C$_4$H$_9$</td>
<td>58.12</td>
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<td>n-Butane</td>
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<td>58.12</td>
<td>1.11</td>
<td>0.793</td>
<td>3,800</td>
<td>425</td>
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<td>Isobutylene</td>
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<td>56.11</td>
<td>1.11</td>
<td>0.790</td>
<td>4,000</td>
<td>418</td>
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<td>Carbon dioxide</td>
<td>CO$_2$</td>
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<td>0.929</td>
<td>7,387</td>
<td>304</td>
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<tr>
<td>Carbon monoxide</td>
<td>CO</td>
<td>28.01</td>
<td>1.40</td>
<td>1.000</td>
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<td>133</td>
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<td>Chlorine</td>
<td>Cl$_2$</td>
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<td>7,980</td>
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<td>Ethane</td>
<td>C$_2$H$_6$</td>
<td>30.07</td>
<td>1.22</td>
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<td>4,884</td>
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<td>Ethylene</td>
<td>C$_2$H$_4$</td>
<td>28.05</td>
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<td>0.871</td>
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<td>Fluorine</td>
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<td>1.36</td>
<td>0.970</td>
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<tr>
<td>Freon 11 (trichloromonofluormethane)</td>
<td>CCl$_3$F</td>
<td>137.37</td>
<td>1.14</td>
<td>0.811</td>
<td>4,409</td>
<td>471</td>
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<tr>
<td>Freon 12 (dichlorodifluoromethane)</td>
<td>CCl$_2$F$_2$</td>
<td>120.91</td>
<td>1.13</td>
<td>0.807</td>
<td>4,114</td>
<td>385</td>
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<tr>
<td>Freon 13 (chlorotrifluoromethane)</td>
<td>CClF</td>
<td>104.46</td>
<td>1.14</td>
<td>0.814</td>
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<tr>
<td>Freon 22 (chlorodifluoromethane)</td>
<td>CHClF$_2$</td>
<td>80.47</td>
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<td>0.846</td>
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<td>369</td>
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<td>Helium</td>
<td>He</td>
<td>4.003</td>
<td>1.66</td>
<td>1.186</td>
<td>229</td>
<td>5.25</td>
</tr>
<tr>
<td>n-Heptane</td>
<td>C$<em>7$H$</em>{16}$</td>
<td>100.20</td>
<td>1.05</td>
<td>0.750</td>
<td>2,736</td>
<td>540</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>H$_2$</td>
<td>2.016</td>
<td>1.41</td>
<td>1.007</td>
<td>1,297</td>
<td>33.25</td>
</tr>
<tr>
<td>Hydrogen chloride</td>
<td>HCl</td>
<td>36.46</td>
<td>1.41</td>
<td>1.007</td>
<td>8,319</td>
<td>325</td>
</tr>
<tr>
<td>Hydrogen fluoride</td>
<td>HF</td>
<td>20.01</td>
<td>0.97</td>
<td>0.691</td>
<td>6,845</td>
<td>461</td>
</tr>
<tr>
<td>Methane</td>
<td>CH$_4$</td>
<td>16.04</td>
<td>1.32</td>
<td>0.943</td>
<td>4,600</td>
<td>191</td>
</tr>
<tr>
<td>Methyl chloride</td>
<td>CH$_3$Cl</td>
<td>50.49</td>
<td>1.24</td>
<td>0.889</td>
<td>6,677</td>
<td>417</td>
</tr>
<tr>
<td>Natural gas 4)</td>
<td>–</td>
<td>17.74</td>
<td>1.27</td>
<td>0.907</td>
<td>4,634</td>
<td>203</td>
</tr>
<tr>
<td>Neon</td>
<td>Ne</td>
<td>20.179</td>
<td>1.64</td>
<td>1.171</td>
<td>2,726</td>
<td>44.45</td>
</tr>
<tr>
<td>Nitric oxide</td>
<td>NO</td>
<td>63.01</td>
<td>1.40</td>
<td>1.000</td>
<td>6,485</td>
<td>180</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>N$_2$</td>
<td>28.013</td>
<td>1.40</td>
<td>1.000</td>
<td>3,394</td>
<td>126</td>
</tr>
<tr>
<td>Octane</td>
<td>C$<em>8$H$</em>{18}$</td>
<td>114.23</td>
<td>1.66</td>
<td>1.186</td>
<td>2,513</td>
<td>569</td>
</tr>
<tr>
<td>Oxygen</td>
<td>O$_2$</td>
<td>32.000</td>
<td>1.40</td>
<td>1.000</td>
<td>5,040</td>
<td>155</td>
</tr>
<tr>
<td>Pentane</td>
<td>C$<em>5$H$</em>{12}$</td>
<td>72.15</td>
<td>1.06</td>
<td>0.757</td>
<td>3,374</td>
<td>470</td>
</tr>
<tr>
<td>Propane</td>
<td>C$_3$H$_6$</td>
<td>44.10</td>
<td>1.15</td>
<td>0.821</td>
<td>4,256</td>
<td>370</td>
</tr>
<tr>
<td>Propylene</td>
<td>C$_3$H$_8$</td>
<td>42.08</td>
<td>1.14</td>
<td>0.814</td>
<td>4,600</td>
<td>365</td>
</tr>
<tr>
<td>Saturated steam</td>
<td>–</td>
<td>18.016</td>
<td>1.25</td>
<td>0.893</td>
<td>22,119</td>
<td>647</td>
</tr>
<tr>
<td>Sulphur dioxide</td>
<td>SO$_2$</td>
<td>64.06</td>
<td>1.26</td>
<td>0.900</td>
<td>7,822</td>
<td>430</td>
</tr>
<tr>
<td>Superheated steam</td>
<td>–</td>
<td>18.016</td>
<td>1.315</td>
<td>0.939</td>
<td>22,119</td>
<td>647</td>
</tr>
</tbody>
</table>

1) Constants are for fluids (except for steam) at ambient temperature and atmospheric pressure.

2) Pressure units are kPa (absolute).

3) Temperature units are in K.

4) Representative values; exact characteristics require knowledge of exact constituents.
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Annex D (informative) — Alternate non-turbulent flow calculation method

The following explicit calculation method may be used for liquids to compute the Reynolds Number Factor, \( F_R \), for valves where \( C_v / d^2 \) is less than 30 when \( d \) is in inches; \( Q \) = U.S. gpm; \( \Delta P \) is in psi.

D.1 Calculate a pseudo-\( \text{Re}_{vi} \)

\[
\text{Eq. D.1} \quad \text{Re}_{vi} = N_2 \frac{Q}{\sqrt{0.9 C_{vT}}} \left[ 1 + \frac{0.8 C_{vT}}{N_2 d^4} \right]^{1/4}
\]

where

\[
\text{Eq. D.2} \quad C_{vT} = \sqrt{\frac{G_f}{\Delta P}}
\]

**NOTE** This calculation assumes the valve is a globe-type valve with an \( F_L \) of 0.9.

D.2 Calculate a preliminary laminar flow coefficient, \( C_{vLi} \)

\[
\text{Eq. D.3} \quad C_{vLi} = \frac{C_{vT}}{0.019 \text{Re}_{vi}^{2/3}}
\]

If \( C_{vLi} / d^2 < 0.1 \), then go to step D.4.

D.3 Calculate exponent, \( n \):

\[
\text{Eq. D.4} \quad n_1 = \left[ \frac{29.9 d^2}{C_{vLi} \left( \frac{0.9}{F_L} \right)^{0.7}} \right]^{2}
\]

\[
\text{Eq. D.5} \quad n_2 = 1 + 1.7 \left[ \frac{C_{vLi}}{d^2 \left( \frac{0.9}{F_L} \right)^{0.7}} \right]^{2/3}
\]

\[
\text{Eq. D.6} \quad n = \begin{cases} n_1 & \text{if } n_1 \leq 9 \\ n_2 & \text{if } n_1 > 9 \end{cases}
\]

**NOTE** Neither \( n \) nor \( n_2 \) should be less than 1.0.
D.4 Compute laminar flow coefficient

If $0.1 < \frac{C \nu L_i}{d^2} < 30$ then

$$\text{Eq. D.7} \quad C_{\nu L} = \frac{0.192 \left( \frac{G \nu Q}{\Delta P F_d n} \right)^{2/3}}{F_L^{1.667}} \left[ 1 + \frac{C_{\nu L_i}^2 F_{\nu L}^2}{N_2 d^4} \right]^{1/6}$$

If $C_{\nu L_i}/d^2 < 0.1$, then

$$\text{Eq. D.8} \quad C_{\nu L} = 0.194 \left( \frac{G \nu Q}{\Delta P F_d} \right)^{2/3} F_L^{1.667}$$

For needle-type trims, where

$$F_d = 0.09 \sqrt{\frac{C_{\nu L} F_L}{D_o}}$$

then,

$$\text{Eq. D.9} \quad C_{\nu L} = 0.973 \left( Q G_{\nu L} \left( \frac{D_o}{\Delta P} \right)^{0.5} \right)$$

where $D_o$ is the orifice diameter in inches.

D.5 Compute transitional flow coefficient

$$\text{Eq. D.10} \quad C_{\nu T} = \frac{C_{\nu T}}{1 + \left( \frac{0.33 F_{\nu L}^{0.5} \nu^{0.25}}{n} \right) \log \left( 1.73 F_d \frac{Q}{\nu C_{\nu L}^{0.5}} \right) F_{\nu L}^{0.5}}$$

NOTE The value of $n$ in the above equation should be determined from the Equations D.4 and D.5 but using $C_{\nu L}$ as calculated from Equations D.7 or D.8 instead of $C_{\nu L_i}$. For $C_{\nu L_i} / d^2 < 0.1$ use $n = 1$

For $C_{\nu L_i} / d^2 < 0.1$ and needle-type trims:

$$\text{Eq. D.11} \quad C_{\nu T} = \frac{Q G_{\nu L}^{0.5}}{\Delta P^{0.5} \left[ 1 + 0.33 \log \left( \frac{0.156 Q}{D_o \nu} \right) \right]}$$
D.6 Determine sizing flow coefficient

The largest of the turbulent flow coefficient, \( C_{vT} \), transitional flow coefficient, \( C_{vLt} \), or laminar flow coefficient, \( C_{vL} \), should be used for selecting the correct valve size.

D.7 Calculating the flow coefficient for gases

NOTE \( q = \text{scfh}, \mu = \text{absolute viscosity of gas at inlet temperature in centipoise}, G_g = \text{specific gravity of gas.} \) Determine \( C_{vT} \) from Equation D.7 to D.9, or D.12 to D.15.

If \( C_{vT}/d^2 < 0.1 \), then:

Assuming Laminar Flow

Eq. D.12

\[
C_{vL} = \frac{C_{vT} \sqrt{\mu C_{vT}^{0.5}}}{0.0423 \sqrt{q F_d G_g}}
\]

For needle-type trims, where

\[
F_d = 0.09 \sqrt{\frac{C_v F_L}{D_o}}
\]

then,

Eq. D.13

\[
C_{vL} = \frac{C_{vT} \sqrt{\mu D_o}}{0.0127 \sqrt{q G_g}}
\]

Assuming Transitional Flow

Eq. D.14

\[
C_{vLt} = C_{vT} \left[ \frac{\mu C_{vT}^{0.5}}{2.65 \times 10^{-3} F_d q G_g} \right]^{0.18}
\]

For needle-type trims,

Eq. D.15

\[
C_{vLT} = C_{vT} \left[ \frac{\mu D_o}{2.38 \times 10^{-5} q G_g} \right]^{0.18}
\]

For sizing purposes, select the larger of the turbulent (\( C_{vT} \)), the laminar (\( C_{vL} \)), or the transitional flow Coefficient (\( C_{vLT} \)).

Example:

A 1/2" valve is required to pass 16.2 scfh gas at \( G_g = 1.34; \mu = 0.0215 \text{ cP}; P_1 = 190 \text{ psia}, \text{ and } P_2 = 170 \text{ psia}. \) \( C_{vT} \) calculated from the above is 0.005. The manufacturer stated orifice diameter \( D_o \) is 0.197 inches and a tapered, needle-type trim is used.

Here \( C_v/d^2 = 0.005/0.5^2 < 0.1 \), so we can use Equations D.13 and D.15, respectively.
Assuming Laminar Flow

\[ C_{vl} = \frac{0.005 \sqrt{0.0215 \times 0.197}}{0.0127 \sqrt{16.2 \times 1.34}} = 0.0055 \]

Assuming Transitional Flow

\[ C_{vlt} = 0.005 \left[ \frac{0.0215 \times 0.197}{2.38 \times 10^{-5} \times 16.2 \times 1.34} \right]^{0.18} = 0.0073 \]

Since \( C_{vlt} \) is larger, the selected trim size has to be 0.0073 or larger.

Example 1: Incompressible flow – non-choked turbulent flow without attached fittings

Process data:

Fluid: water
Inlet temperature: $T_1 = 363$ K
Density: $\rho_1 = 965.4$ kg/m$^3$
Vapor pressure: $P_v = 70.1$ kPa
Thermodynamic critical pressure: $P_c = 22,120$ kPa
Kinematic viscosity: $\nu = 3.26 \times 10^{-7}$ m$^2$/s
Inlet absolute pressure: $P_1 = 680$ kPa
Outlet absolute pressure: $P_2 = 220$ kPa
Flow rate: $Q = 360$ m$^3$/h
Pipe size: $D_1 = D_2 = 150$ mm

Valve data:

Valve style: globe
Trim: parabolic plug
Flow direction: flow-to-open
Valve size: $d = 150$ mm
Liquid pressure recovery factor: $F_L = 0.90$ (from Table 2)
Valve style modifier: $F_d = 0.46$ (from Table 2)

Calculations:

Eq. 31

$$F_e = 0.96 - 0.28 \sqrt[3]{\frac{P_v}{P_c}} = 0.944$$

where

$P_v = 70.1$ kPa; and
$P_c = 22,120$ kPa.
Next, determine the type of flow:

\[ F_L^2(P_F - F_P \times P_v) = 497.2 \text{ kPa} \]

which is more than the differential pressure (\( \Delta P = 460 \text{ kPa} \)); therefore, the flow is non-choked, and the flow coefficient \( C \) is calculated using Equation 1.

\[
C = \frac{Q}{N_i} \sqrt{\frac{\rho_i / \rho_o}{\Delta P}} = 165 \text{ m}^3/\text{h} \text{ for } K_v
\]

where

\( Q = 360 \text{ m}^3/\text{h} \);

\( N_i = 1 \times 10^{-1} \text{ from Table 1} \);

\( \rho_i/\rho_o = 0.965 \); and

\( \Delta P = 460 \text{ kPa} \).

Next, calculate \( \text{Re}_v \).

\[
\text{Re}_v = \frac{N_2 F_d Q}{\sqrt{C_i F_L} \left[ \frac{F_i^2 C_i^2}{N_2 D^4} + 1 \right]^{1/4}} = 2.967 \times 10^6
\]

where

\( N_2 = 1.60 \times 10^{-3} \text{ from Table 1} \);

\( N_4 = 7.07 \times 10^{-2} \text{ from Table 1} \);

\( F_d = 0.46 \);

\( Q = 360 \text{ m}^3/\text{h} \);

\( \nu = 3.26 \times 10^{-7} \text{ m}^2/\text{s} \);

\( C_i = C = K_v = 165 \text{ m}^3/\text{h} \);

\( F_L = 0.90 \); and

\( D = 150 \text{ mm} \).

Since the valve Reynolds number is greater than 10,000, the flow is turbulent, and the flow coefficient \( C \) as calculated above is correct.

**Example 2: Incompressible flow – choked flow without attached fittings**

**Process data:**
Fluid: water

Inlet temperature: $T_1 = 363$ K

Density: $\rho_1 = 965.4$ kg/m$^3$

Vapor pressure: $P_v = 70.1$ kPa

Thermodynamic critical pressure: $P_c = 22,120$ kPa

Kinematic viscosity: $\nu = 3.26 \times 10^{-7}$ m$^2$/s

Inlet absolute pressure: $P_1 = 680$ kPa

Outlet absolute pressure: $P_2 = 220$ kPa

Flow rate: $Q = 360$ m$^3$/h

Pipe size: $D_1 = D_2 = 100$ mm

Valve data:

Valve style: ball valve

Trim: segmented ball

Flow direction: flow-to-open

Valve size: $d = 100$ mm

Liquid pressure recovery factor: $F_L = 0.60$ (from Table 2)

Valve style modifier: $F_d = 0.98$ (from Table 2)

Calculations:

Eq. 31 $F_r = 0.96 - 0.28 \left( \frac{P_v}{P_c} \right) = 0.944$
where

\( P_v = 70.1 \text{ kPa} \); and

\( P_c = 22,120 \text{ kPa} \).

Next, determine the type of flow.

\[
F_L^2 \left( P_1 - F_F \times P_v \right) = 221 \text{ kPa}
\]

which is less than the differential pressure (\( \Delta P = 460 \text{ kPa} \)); therefore, the flow is choked and the flow coefficient \( C \) is calculated using Equation 3.

\[
\text{Eq. 3} \quad C = \frac{Q}{N_1 F_L} \sqrt{\frac{\rho_1}{\rho_0} \frac{P_1}{P_c P_v}} = 238 \text{ m}^3/\text{h for } K_v
\]

where

\( Q = 360 \text{ m}^3/\text{h} \);

\( N_1 = 1 \times 10^{-1} \) from Table 1;

\( F_L = 0.60 \);

\( \rho_1/\rho_0 = 0.965 \);

\( P_1 = 680 \text{ kPa} \),

\( F_F = 0.944 \); and

\( P_v = 70.1 \text{ kPa} \).

Next, calculate \( Re_v \).

\[
\text{Eq. 28} \quad Re_v = \frac{N_4 F_d Q}{\nu \sqrt{C_i F_L}} \left[ \frac{F_L^2 C_i^2}{N_2 D^4} + 1 \right]^{1/4} = 6.598 \times 10^6
\]

where

\( N_2 = 1.60 \times 10^{-3} \) from Table 1;

\( N_4 = 7.07 \times 10^{-2} \) from Table 1;

\( F_d = 0.98 \);

\( Q = 360 \text{ m}^3/\text{h} \);

\( \nu = 3.26 \times 10^{-7} \text{ m}^2/\text{s} \);

\( C_i = C = K_v = 238 \text{ m}^3/\text{h} \);
Since the valve Reynolds number is greater than 10,000, the flow is turbulent and no more correction is necessary.

**Example 3: Compressible flow – non-choked flow with attached fittings**

Process data:

**Fluid:** carbon dioxide

**Inlet temperature:** \( T_1 = 433 \text{ K} \)

**Molecular mass:** \( M = 44.01 \text{ kg/kmol} \)

**Kinematic viscosity:** \( \nu = 1.743 \times 10^{-5} \text{ m}^2/\text{s} \)

**Specific heat ratio:** \( \gamma = 1.30 \)

**Compressibility factor:** \( Z = 0.988 \)

**Inlet absolute pressure:** \( P_1 = 680 \text{ kPa} \)

**Outlet absolute pressure:** \( P_2 = 310 \text{ kPa} \)

**Flow rate:** \( Q = 3,800 \text{ standard m}^3/\text{h at 101.325 kPa and 0°C} \)

**Inlet pipe size:** \( D_1 = 80 \text{ mm} \)

**Outlet pipe size:** \( D_2 = 100 \text{ mm} \)

**Reducers:** short length, concentric

**Valve data:**

**Valve style:** rotary

**Trim:** eccentric rotary plug

**Flow direction:** flow-to-open

**Valve size:** \( d = 50 \text{ mm} \)

**Pressure differential ratio factor:** \( x_T = 0.60 \) (from Table 2)

**Liquid pressure recovery factor:** \( F_L = 0.85 \) (from Table 2)

**Valve style modifier:** \( F_d = 0.42 \) (from Table 2)

**Calculations:**
Eq. 34 \[ F_{\gamma} = \frac{\gamma}{1.40} = 0.929 \]

where

\[ \gamma = 1.30. \]

and with this

\[ x = \frac{\Delta P}{P_1} = 0.544 \]

which is less than \( F_{\gamma} x_T = 0.557 \); therefore, the flow is non-choked and the flow coefficient is calculated from Equation 11. Next, \( Y \) is calculated from Equation 32;

Eq. 32 \[ Y = 1 - \frac{x}{3F_{\gamma} x_T} = 0.674 \]

where

\[ x = 0.544; \]
\[ F_{\gamma} = 0.929; \] and
\[ x_T = 0.60 \]

Eq. 11 \[ C = \frac{Q}{N_9 L_x P_1 Y} \sqrt{\frac{MT_x Z}{x}} = 62.7 \text{ m}^3/\text{h} \text{ for } K_v \]

where

\[ Q = 3,800 \text{ m}^3/\text{h}; \]
\[ N_9 = 2.46 \times 10^1 \text{ for } t_s = 0^\circ \text{C from Table 1}; \]
assume \( F_P = 1; \)
\[ P_1 = 680 \text{ kPa}; \]
\[ Y = 0.674; \]
\[ M = 44.01 \text{ kg/kmol}; \]
Now, calculate $Re_v$ using Equation 28.

$$\text{Eq. 28} \quad Re_v = \frac{N_4 F_d Q}{\nu \sqrt{C_i F_L}} \left[ \frac{F_L^2 C_i^2}{N_2 D^4} + 1 \right]^{1/4} = 8.96 \times 10^5$$

where

$N_2 = 1.60 \times 10^{-3}$ from Table 1;

$N_4 = 7.07 \times 10^{-2}$ from Table 1;

$F_d = 0.42$;

$Q = 3,800 \text{ m}^3/\text{h}$;

$\nu = 1.743 \times 10^{-5} \text{ m}^2/\text{s}$;

$C_i = C = K_V = 62.7 \text{ m}^3/\text{h}$;

$F_L = 0.85$; and

$D = 80 \text{ mm}$.

Since the valve Reynolds number is greater than 10,000, the flow is turbulent.

Now, calculate the effect of the inlet and outlet reducers on $C$.

Since both reducers are concentric, short length, the velocity head loss coefficients can be calculated as follows:

$$\text{Eq. 23} \quad \zeta_1 = 0.5 \left[ 1 - \left( \frac{d}{D_1} \right)^2 \right]^2 = 0.186$$

where

$d = 50 \text{ mm}$;

$D_1 = 80 \text{ mm}$; and

$$\text{Eq. 24} \quad \zeta_1 = 1.0 \left[ 1 - \left( \frac{d}{D_2} \right)^2 \right]^2 = 0.563$$
where
\[ d = 50 \text{ mm}; \]
\[ D_2 = 100 \text{ mm}; \]

and the Bernoulli coefficients are
\[ \zeta_{B1} = 1 - \left( \frac{d}{D_1} \right)^2 = 0.847 \]

where
\[ d = 50 \text{ mm}; \]
\[ D_1 = 80 \text{ mm}; \]

and
\[ \zeta_{B2} = 1 - \left( \frac{d}{D_2} \right)^2 = 0.938 \]

where
\[ d = 50 \text{ mm}; \]
\[ D_2 = 100 \text{ mm}. \]

The effective head loss coefficient of the inlet and outlet reducers is
\[ \Sigma \zeta = \zeta_1 + \zeta_2 + \zeta_{B1} - \zeta_{B2} = 0.658 \]

where
\[ \zeta_1 = 0.186 \]
\[ \zeta_2 = 0.563 \]
\[ \zeta_{B1} = 0.847 \]
\[ \zeta_{B2} = 0.938 \]

Now, the effect of the reducers is calculated by iteration, starting with \( C_i = C \) and \( F_{P(1)} = 1. \)

\[ F_{p(2)} = \frac{1}{\sqrt{1 + \frac{\sum \zeta \left( \frac{C_i}{N_2} \right)^2}{d^2}}} = 0.891 \]
where
\[ \Sigma \zeta = 0.658; \]
\[ N_2 = 1.60 \times 10^{-3} \text{ from Table 1;} \]
\[ C_1 = 62.7 \text{ m}^3/\text{h}; \text{ and} \]
\[ d = 50 \text{ mm}. \]

Since \( F_{P(2)}/F_{P(1)} = 0.891/1 < 0.97 \), further iterative steps are required.

\[ C_2 = \frac{C}{F_{p(2)}} = \frac{62.7}{0.891} = 70.4 \text{ m}^3/\text{h} \]

Eq. 20
\[
F_{p(3)} = \frac{1}{\sqrt{1 + \frac{\Sigma \zeta}{N_2} \left( \frac{C_2}{d^2} \right)^2}} = 0.868
\]

Since, \( F_{P(3)}/F_{P(2)} = 0.868/0.891 < 0.99 \) one more iterative step will be required

where
\[ \Sigma \zeta = 0.658; \]
\[ N_2 = 1.60 \times 10^{-3} \text{ from Table 1;} \]
\[ C_2 = \frac{C}{F_{p(3)}} = \frac{62.7}{0.868} = 70.23 \text{ m}^3/\text{h} \]
\[ d = 50 \text{ mm}. \]

\[
F_{p(4)} = \frac{1}{\sqrt{1 + \frac{\Sigma \zeta}{N_2} \left( \frac{C_2}{d^2} \right)^2}} = 0.869
\]

Now, \( F_{P(3)}/F_{P(4)} = 0.868/0.869 > 0.99 \), \( F_P = 0.869 \) will be used

Eq. 33
\[
X_{TP} = \frac{\frac{x_T}{F_p^2}}{1 + \frac{X_T \zeta}{N_5} \left( \frac{C_2}{d^2} \right)^2} = 0.625
\]
where
\[ x_T = 0.60; \]
\[ F_P = 0.869; \]
\[ \zeta_1 = \zeta_1 + \zeta_B = 1.033; \]
\[ N_5 = 1.80 \times 10^{-3} \text{ from Table 1}; \]
\[ C_2 = 70.23 \text{ m}^3/\text{h}; \]
\[ d = 50 \text{ mm}; \]

and with this \( F_\gamma x_T P = 0.582 \), which is greater than \( x = 0.544 \).

Finally, \( C \) results from Equation 11 as follows:
\[
Eq. \ 11 \quad C = \frac{Q}{N_9 F_P P_1 Y} \sqrt{\frac{MT_1 Z}{x}} = 72.1 \text{ m}^3/\text{h} \quad \text{for } K_c
\]

where
\[ Q = 3,800 \text{ m}^3/\text{h}; \]
\[ N_9 = 2.46 \times 10^1 \text{ for } t_0 = 0^\circ \text{C from Table 1}; \]
\[ F_P = 0.869; \]
\[ P_1 = 680 \text{ kPa}; \]
\[ Y = 0.674; \]
\[ M = 44.01 \text{ kg/kmol}; \]
\[ T_1 = 433 \text{ K}; \]
\[ Z = 0.988; \text{ and } \]
\[ x = 0.544 \]

**Example 4: Compressible flow – small flow trim sized for gas flow**

**Process data:**

**Fluid:** argon gas

**Inlet temperature:** \( T_1 = 320 \text{ K} \)

**Inlet absolute pressure:** \( P_1 = 2.8 \text{ bar (absolute)} \)
Outlet absolute pressure: \( P_2 = 1.3 \text{ bar (absolute)} \)

Flow rate: \( Q = 0.46 \text{ standard m}^3/\text{h at 1,013.25 mbar and 47°C} \)

Molecular mass: \( M = 39.95 \)

Kinematic viscosity: \( \nu = 1.338 \times 10^{-5} \text{ m}^2/\text{s at 1 bar (absolute) and 15°C} \)

Specific heat ratio: \( \gamma = 1.67 \)

Specific heat ratio factor: \( F_\gamma = 1.19 \)

Valve data:

Trim: tapered needle plug

Liquid pressure recovery factor: \( F_L = 0.98 \)

Pressure differential ratio factor: \( x_T = 0.8 \)

Orifice diameter: \( D_o = 5 \text{ mm} \)

Valve size: \( d = 15 \text{ mm} \)

Internal diameter of piping: \( D = 15 \text{ mm} \)

Calculation:

The first step is to check the Reynolds number \( Re_v \).

\[
\text{Eq. 28 } \quad \text{Re}_v = \frac{N_d F_o Q}{\sqrt{C_i F_L}} \left[ \frac{F_L^2 C_i^2}{N_d D^4 T} + 1 \right]^{1/4}
\]

This requires input of \( C_i \), which has to be determined. Since \( x < F_\gamma x_T \), the flow coefficient can be estimated by first using the nonchoked flow Equation 8 to calculate \( C \), then multiplying \( C \) by 1.3 in accordance with the iteration procedure of 8.1.

\[
\text{Eq. 32 } \quad Y = 1 - \frac{x}{3 F_\gamma x_T} = 0.812
\]

\[
\text{Eq. 8 } \quad C = \frac{Q}{N_9^2 P_1 T Y \sqrt{\frac{MT^2 Z}{x}}} = 0.0138 \text{ for } C_v
\]

where

\( Q = 0.46 \text{ m}^3/\text{h}; \)

\( N_9 = 2.25 \times 10^3 \text{ for } t_9 = 15°C \text{ from Table 1}; \)
\[ P_1 = 2.8 \text{ bar}; \]
\[ M = 39.95 \text{ kg/kmol}; \]
\[ T_1 = 320 \text{ K}; \]
\[ Z = 1; \]
\[ F_\gamma = 1.19; \]
\[ x = 0.536; \text{ and} \]
\[ Y = 0.812. \]

Eq. 26 \[ C_i = 1.3 \, C = 0.0179 \text{ for } C_v \]

where
\[ C = 0.0138 \text{ for } C_v. \]

Next, estimate \( F_d \) from the equation in Table 2.

\[ F_d = \frac{N_{19} \sqrt{C F_L}}{D_o} = 0.061 \]

where
\[ C = C_i = 0.0179 \text{ for } C_v; \]
\[ F_L = 0.98; \]
\[ N_{19} = 2.3 \text{ from Table 1}; \text{ and} \]
\[ D_o = 5 \text{ mm}. \]

Calculate \( Re_v \) as follows:

\[ \text{Eq. 28} \quad Re_v = \frac{N_4 F_d Q}{\sqrt{C F_L}} \left[ \frac{F_L^2}{N_2 D^4} + 1 \right]^{1/4} = 1202 \]

where
\[ N_2 = 2.14 \times 10^{-3} \text{ from Table 1}; \]
\[ N_4 = 7.6 \times 10^{-2} \text{ from Table 1}; \]
\[ F_d = 0.061; \]
\[ Q = 0.46 \text{ m}^3/\text{h}; \]
\[ \nu = 1.338 \times 10^{-5} \text{ m}^2/\text{s}; \]
\( F_L = 0.98; \)
\( C_i = 0.0179 \text{ for } C_v; \) and
\( D = 15 \text{ mm.} \)

Determine if \( C/d^2 < 0.016 \ N_{18} \).

\[
C/d^2 = 7.97 \times 10^{-5}
\]

\[
0.016 \ N_{18} = 0.016
\]

\( C/d^2 < 0.016 \ N_{18} \)

where
\( N_{18} = 1.00 \text{ from Table 1; } \)
\( C = 0.0179; \) and
\( d = 15 \text{ mm.} \)

Since the Reynolds number is below 10,000, the flow is non-turbulent; hence flow coefficient Equation F.3b has to be used. Since \( C/d^2 < 0.016 \ N_{18} \) and \( Re_v > 10 \), calculate \( F_R \) from both Equations F.3a and F.4 and use the lower value (reference Annex F).

**Eq. F.3b**

\[
n_2 = 1 + N_{32} \left( \frac{C_i}{d^2} \right)^{2/3} = 1.235
\]

where
\( N_{32} = 1.27 \times 10^2 \text{ from Table 1; } \)
\( C_i = 0.0179 \text{ for } C_v; \)
\( Re_v = 1,202; \) and
\( d = 15 \text{ mm.} \)

**Eq. G.3a**

\[
F_R = 1 + \left( \frac{0.33 F_L^{1/2}}{n_2^{1/4}} \right) \log_{10} \left( \frac{Re_v}{10,000} \right) = 0.715
\]

where
\( F_L = 0.98; \)
\( n_2 = 1.235; \) and
\( Re_v = 1,202. \)
Eq. F.4  \[ F_R = \frac{0.026}{F_L} \sqrt{n_2 R \nu} = 1.022 \]

NOTE  \( F_R \) is limited to 1.

where

\[ F_L = 0.98; \]
\[ n_2 = 1.235; \] and
\[ Re_\nu = 1,202. \]

Use \( F_R = 0.715 \), the lower of the two calculated values.

Eq. 19  \[ C = \frac{Q}{N_{22} F_R} \sqrt{\frac{M T_1}{\Delta P (P_1 + P_2)}} = 0.0184 \text{ for } C_\nu \]

where

\[ Q = 0.46 \text{ m}^3/\text{h}; \]
\[ N_{22} = 1.59 \times 10^3 \text{ for } t_5 = 15^\circ \text{C from Table 1}; \]
\[ F_R = 0.715; \]
\[ M = 39.95 \text{ kg/kmol}; \]
\[ T_1 = 320 \text{ K}; \]
\[ \Delta P = 1.5 \text{ bar}; \]
\[ P_1 = 2.8 \text{ bar}; \] and
\[ P_2 = 1.3 \text{ bar}. \]

Check:

Eq. 29  \[ \frac{C}{F_R} < C_i \]

\[ \frac{0.0184}{0.715} = 0.0257 > 0.0179 \]

Since \( C/F_R \) is not less than \( C_i \), repeat the iteration process by increasing \( C_i \) by 30%.

New \( C_i = 1.3 \) \( C_i = 0.0233 \]

where
\( C_l = 0.0179; \)

\[
F_d = \frac{N_{19} \sqrt{C_F}}{D_o} = 0.070
\]

where

\( C = C_l = 0.0233 \) for \( C_v; \)

\( F_L = 0.98; \)

\( N_{19} = 2.3 \) from Table 1; and

\( D_o = 5 \text{ mm}. \)

Calculate \( R_{e_v}. \)

\[
\text{Eq. 28} \quad R_{e_v} = \frac{N_2 \cdot F_d \cdot Q \left[ \frac{F_L^2 C_v^2}{N_2 D^4} + 1 \right]^{1/4}}{\sqrt{C_l \cdot F_L}} = 1,202
\]

where

\( N_2 = 2.14 \times 10^{-3} \) from Table 1;

\( N_4 = 7.6 \times 10^{-2} \) from Table 1;

\( F_d = 0.070; \)

\( Q = 0.46 \text{ m}^3/\text{h}; \)

\( \nu = 1.338 \times 10^{-5} \text{ m}^2/\text{s}; \)

\( F_L = 0.98; \)

\( C_l = 0.0233; \) and

\( D = 15 \text{ mm}. \)

Since the value of \( R_{e_v} \) remains the same as previously calculated, \( F_R \) remains at 0.715. Therefore, the calculated \( C \) will remain at 0.0184 and any trim with a rated \( C \) of 0.0184 or higher for \( C_v \) is appropriate.
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Annex F (informative) — Equations for Reynolds number factor, $F_R$

For full size trim where $C_i/d^2 \geq 0.016 \ N_{18}$ and $Re_v \geq 10$, calculate $F_R$ from the following equations:

Eq. F.1a

$$ F_R = 1 + 0.0010 \log_{10} \left( \frac{Re_v}{10,000} \right) $$

for the transitional flow regime,

where

Eq. F.1b

$$ n_1 = \frac{N_2}{\left( \frac{C_i}{d^2} \right)^{1/2}} $$

or

Eq. F.2

$$ F_R = \frac{0.026}{F_L} \sqrt{n_1 Re_v} \quad \text{(not to exceed } F_R=1) $$

for the laminar flow regime.

NOTE 1 Use the lower value of $F_R$ from Equations F.1a and F.2. If $Re_v<10$, use only Equation F.2.

NOTE 2 Equation F.2 is applicable to fully developed laminar flow (straight lines in Figure 3a). The relationships expressed in Equations F.1a and F.2 are based on test data with valves at rated travel and may not be fully accurate at lower valve travels.

NOTE 3 In Equations F.1b and F.2, $C_i/d^2$ must not exceed 0.04 when $K_v$ is used or 0.047 when $C_v$ is used.

For reduced trim valves where $C_i/d^2$ at rated travel is less than 0.016 $N_{18}$ and $Re_v \geq 10$, calculate $F_R$ from the following equations:

Eq. F.3a

$$ F_R = 1 + \left( \frac{0.33 F_L^{1/2}}{n_2^{1/4}} \right) \log_{10} \left( \frac{Re_v}{10,000} \right) $$

for the transitional flow regime,

where

Eq. F.3b

$$ n_2 = 1 + N_{32} \left( \frac{C_i}{d^2} \right)^{2/3} $$

or
Eq. F.4 \[ F_R = \frac{0.026}{F_L} \sqrt{n_2 R_{e_v}} \] (not to exceed $F_R=1$)

for the laminar flow regime.

NOTE 1 Select the lowest value from Equations F.3a and F.4. If $Re_v<10$, use only Equation F.4.

NOTE 2 Equation F.4 is applicable to fully developed laminar flow (straight lines in Figure 3b).
Annex G (informative) — Bibliography

INTERNATIONAL ELECTROTECHNICAL COMMISSION (IEC)

IEC 60534-1 Part 1: Control Valve Terminology and General Considerations, 2005

Available from: ANSI
25 West 43rd Street
Fourth Floor
New York, NY 10036

ISA

ANSI/ISA-75.02-1996 Control Valve Capacity Test Procedures
ANSI/ISA-75.05.01-2000 (R2005) Control Valve Terminology


Driskell, L.R., Control Valve Selection and Sizing, 1983.


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