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Indian Standard
INDUSTRIAL-PROCESS CONTROL VALVES
PART 2 FLOW CAPACITY
Section 1 Sizing Equations for Fluid Flow Under Installed Conditions

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NATIONAL FOREWORD

This Indian Standard (Part 2/Sec 1) which is identical with IEC 60534-2-1 : 1998 'Industrial-process control valves — Part 2-1: Flow capacity — Sizing equations for fluid flow under installed conditions' issued by the International Electrotechnical Commission (IEC) was adopted by the Bureau of Indian Standards on the recommendation of the Industrial Process Measurement and Control Sectional Committee and approval of the Electrotechnical Division Council.

This standard supersedes IS 10189 (Part 2/Sec 1) : 1992 'Industrial process control valves: Part 2 Flow capacity, Section 1 Sizing equations for incompressible fluid flow under installed conditions'.

The text of IEC Standard has been approved as suitable for publication as an Indian Standard without deviations. Certain conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'.

b) Comma (,) has been used as a decimal marker, while in Indian Standards, the current practice is to use a point (.) as the decimal marker.

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards, which are to be substituted in their respective places, are listed below along with their degree of equivalence for the editions indicated:

International Standard	*Corresponding Indian Standard*	*Degree of Equivalence*
IEC 60534-1 : 1987 Industrial-process control valves — Part 1: Control valve terminology and general considerations	IS/IEC 60534-1 : 1987 Industrial-process control valves: Part 1 Control valve terminology and general considerations	Identical
IEC 60534-2-3 : 1997 Industrial-process control valves — Part 2: Flow capacity — Section 3: Test procedures	IS/IEC 60534-2-3 : 1997 Industrial-process control valves: Part 2 Flow capacity, Section 3 Test procedures	do

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be same as that of the specified value in this standard.

Indian Standard

INDUSTRIAL-PROCESS CONTROL VALVES

PART 2 FLOW CAPACITY

Section 1 Sizing Equations for Fluid Flow Under Installed Conditions

1 Scope

This part of IEC 60534 includes equations for predicting the flow 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 which require that the basic equations be modified by appropriate correction factors. The equations for compressible fluids are for use with gas or vapour and are not intended for use with multiphase streams such as gas-liquid, vapour-liquid or gas-solid mixtures.

For compressible fluid applications, this part of IEC 60534 is valid for valves with $x_T \leq 0{,}84$ (see table 2). For valves with $x_T > 0{,}84$ (e.g. some multistage valves), greater inaccuracy of flow prediction can be expected.

Reasonable accuracy can only be maintained for control valves if $K_v/d^2 < 0{,}04$ ($C_v/d^2 < 0{,}047$).

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of IEC 60534. 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 IEC 60534 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*

3 Definitions

For the purpose of this part of IEC 60534, definitions given in IEC 60534-1 apply with the addition of the following:

3.1

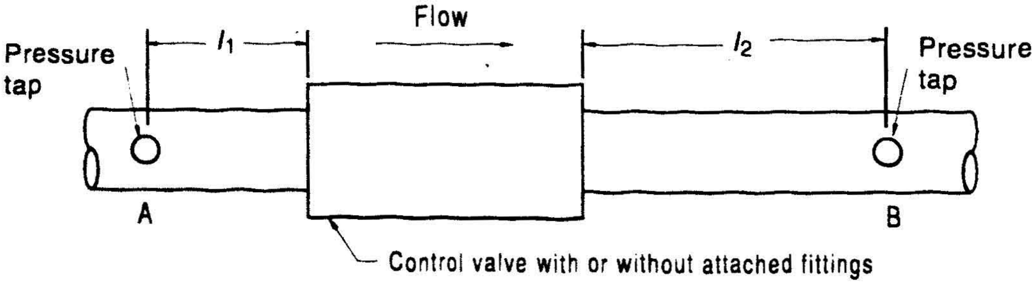
valve style modifier F_d

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
D	Internal diameter of the piping	mm
D_1	Internal diameter of upstream piping	mm
D_2	Internal diameter of downstream piping	mm
D_o	Orifice diameter	mm
F_d	Valve style modifier (see annex A)	1 (see note 4)
F_F	Liquid critical pressure ratio factor	1
F_L	Liquid pressure recovery factor of a control valve without attached fittings	1 (see note 4)
F_{LP}	Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings	1 (see note 4)
F_P	Piping geometry factor	1
F_R	Reynolds number factor	1
F_γ	Specific heat ratio factor	1
M	Molecular mass of flowing fluid	kg/kmol
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 (see note 2)
p_2	Outlet absolute static pressure measured at point B (see figure 1)	kPa or bar
p_c	Absolute thermodynamic critical pressure	kPa or bar
p_r	Reduced pressure (p_1/p_c)	1
p_v	Absolute vapour pressure of the liquid at inlet temperature	kPa or bar
Δp	Differential pressure between upstream and downstream pressure taps ($p_1 - p_2$)	kPa or bar
Q	Volumetric flow rate (see note 5)	m^3/h
Re_v	Valve Reynolds number	1
T_1	Inlet absolute temperature	K
T_c	Absolute thermodynamic critical temperature	K
T_r	Reduced temperature (T_1/T_c)	1
t_s	Absolute reference temperature for standard cubic metre	K
W	Mass flow rate	kg/h
x	Ratio of pressure differential to inlet absolute pressure ($\Delta p/p_1$)	1
x_T	Pressure differential ratio factor of a control valve without attached fittings at choked flow	1 (see note 4)
x_{TP}	Pressure differential ratio factor of a control valve with attached fittings at choked flow	1 (see note 4)
Y	Expansion factor	1
Z	Compressibility factor	1
ν	Kinematic viscosity	m^2/s (see note 3)
ρ_1	Density of fluid at p_1 and T_1	kg/m^3
ρ_1/ρ_o	Relative density (ρ_1/ρ_o = 1,0 for water at 15 °C)	1
γ	Specific heat ratio	1

ζ	Velocity head loss coefficient of a reducer, expander or other fitting attached to a control valve or valve trim	1
ζ_1	Upstream velocity head loss coefficient of fitting	1
ζ_2	Downstream velocity head loss coefficient of fitting	1
ζ_{B1}	Inlet Bernoulli coefficient	1
ζ_{B2}	Outlet Bernoulli coefficient	1

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 metres per hour, identified by the symbol *Q*, refer to standard conditions. The standard cubic metre 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 60534-1.

6.1.1 Non-choked turbulent flow

6.1.1.1 Non-choked turbulent flow without attached fittings

$$\left[\textit{Applicable if } \Delta p < F_L^2 \left(p_1 - F_F \times p_v\right)\right]$$

The flow coefficient shall be determined by

$$C = \frac{Q}{N_1}\sqrt{\frac{\rho_1/\rho_o}{\Delta p}} \tag{1}$$

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 D.

6.1.1.2 Non-choked turbulent flow with attached fittings

$$\left\{\textit{Applicable if } \Delta p < \left[\left(F_{LP}/F_P\right)^2 \left(p_1 - F_F \times p_v\right)\right]\right\}$$

The flow coefficient shall be determined as follows:

$$C = \frac{Q}{N_1 F_P}\sqrt{\frac{\rho_1/\rho_o}{\Delta p}} \tag{2}$$

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

$\left[\textit{Applicable if } \Delta p \geq F_L^2 (p_1 - F_F \times p_v)\right]$

The flow coefficient shall be determined as follows:

$$C = \frac{Q}{N_1 F_L} \sqrt{\frac{\rho_1 / \rho_0}{p_1 - F_F \times p_v}} \tag{3}$$

NOTE – An example of sizing a valve with choked flow without attached fittings is given in annex D

6.1.2.2 Choked turbulent flow with attached fittings

$\left[\textit{Applicable if } \Delta p \geq (F_{LP} / F_P)^2 (p_1 - F_F \times p_v)\right]$

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

$$C = \frac{Q}{N_1 F_{LP}} \sqrt{\frac{\rho_1 / \rho_0}{p_1 - F_F \times p_v}} \tag{4}$$

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 60534-1. 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:

$$C = \frac{Q}{N_1 F_R} \sqrt{\frac{\rho_1 / \rho_0}{\Delta p}} \tag{5}$$

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 behaviour 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.

7.1 Turbulent flow

7.1.1 Non-choked turbulent flow

7.1.1.1 Non-choked turbulent flow without attached fittings

[*Applicable if* $x < F_\gamma \, x_T$]

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

$$C = \frac{W}{N_6 \, Y \sqrt{x \, p_1 \, \rho_1}} \tag{6}$$

$$C = \frac{W}{N_8 \, p_1 \, Y} \sqrt{\frac{T_1 \, Z}{x \, M}} \tag{7}$$

$$C = \frac{Q}{N_9 \, p_1 \, Y} \sqrt{\frac{M \, T_1 \, Z}{x}} \tag{8}$$

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* $x < F_\gamma \, x_{TP}$]

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

$$C = \frac{W}{N_6 \, F_p \, Y \sqrt{x \, p_1 \, \rho_1}} \tag{9}$$

$$C = \frac{W}{N_8 \, F_p \, p_1 \, Y} \sqrt{\frac{T_1 \, Z}{x \, M}} \tag{10}$$

$$C = \frac{Q}{N_9 \, F_p \, p_1 \, Y} \sqrt{\frac{M \, T_1 \, Z}{x}} \tag{11}$$

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 D.

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

$\left[\textit{Applicable if } x \geq F_\gamma \, x_T\right]$

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

$$C = \frac{W}{0{,}667 \, N_6 \sqrt{F_\gamma \, x_T \, p_1 \, \rho_1}} \tag{12}$$

$$C = \frac{W}{0{,}667 \, N_8 \, p_1} \sqrt{\frac{T_1 \, Z}{F_\gamma \, x_T \, M}} \tag{13}$$

$$C = \frac{Q}{0{,}667 \, N_9 \, p_1} \sqrt{\frac{M \, T_1 \, Z}{F_\gamma \, x_T}} \tag{14}$$

7.1.2.2 Choked turbulent flow with attached fittings

$\left[\textit{Applicable if } x \geq F_\gamma \, x_{TP}\right]$

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

$$C = \frac{W}{0{,}667 \, N_6 \, F_p \sqrt{F_\gamma \, x_{TP} \, p_1 \, \rho_1}} \tag{15}$$

$$C = \frac{W}{0{,}667 \, N_8 \, F_p \, p_1} \sqrt{\frac{T_1 \, Z}{F_\gamma \, x_{TP} \, M}} \tag{16}$$

$$C = \frac{Q}{0{,}667 \, N_9 \, F_p \, p_1} \sqrt{\frac{M \, T_1 \, Z}{F_\gamma \, x_{TP}}} \tag{17}$$

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 60534-1. 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

$$C = \frac{W}{N_{27} \, F_R} \sqrt{\frac{T_1}{\Delta p (p_1 + p_2) M}} \tag{18}$$

$$C = \frac{Q}{N_{22} \, F_R} \sqrt{\frac{M \, T_1}{\Delta p (p_1 + p_2)}} \tag{19}$$

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 behaviour 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 IEC 60534-2-3.

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

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

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.

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

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

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

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:

Inlet reducer:

$$\zeta_1 = 0{,}5\left[1-\left(\frac{d}{D_1}\right)^2\right]^2 \tag{23}$$

Outlet reducer (expander):

$$\zeta_2 = 1{,}0\left[1-\left(\frac{d}{D_2}\right)^2\right]^2 \tag{24}$$

Inlet and outlet reducers of equal size:

$$\zeta_1 + \zeta_2 = 1{,}5\left[1-\left(\frac{d}{D}\right)^2\right]^2 \tag{25}$$

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:

$$C_i = 1{,}3\,C \tag{26}$$

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

$$\frac{C}{F_p} \leq C_i \tag{27}$$

If the condition of equation (27) is satisfied, then use the C 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 3 using a valve Reynolds number calculated from the following equation:

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

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 from equations (30) and (31) for full size trims or equations (32) and (33) for reduced trims. In either case, using the lower of the two F_R values, determine if

$$\frac{C}{F_R} \leq C_i \tag{29}$$

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 procedure by again increasing C_i by 30 %. This may require several iterations until the condition required in equation (29) is met.

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:

$$F_R = 1 + \left(\frac{0{,}33\, F_L^{1/2}}{n_1^{1/4}} \right) \log_{10} \left(\frac{Re_v}{10\,000} \right) \tag{30}$$

for the transitional flow regime,

where

$$n_1 = \frac{N_2}{\left(\frac{C_i}{d^2} \right)^2} \tag{30a}$$

or

$$F_R = \frac{0{,}026}{F_L} \sqrt{n_1\, Re_v} \quad \text{(not to exceed } F_R = 1\text{)} \tag{31}$$

for the laminar flow regime.

NOTE 1 – Use the lower value of F_R from equations (30) and (31). If $Re_v < 10$, use only equation (31).

NOTE 2 – Equation (31) is applicable to fully developed laminar flow (straight lines in figure 3). The relationships expressed in equations (30) and (31) are based on test data with valves at rated travel and may not be fully accurate at lower valve travels.

NOTE 3 – In equations (30a) and (31), 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:

$$F_R = 1 + \left(\frac{0{,}33\, F_L^{1/2}}{n_2^{1/4}} \right) \log_{10} \left(\frac{Re_v}{10\,000} \right) \tag{32}$$

for the transitional flow regime,

where

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

or

$$F_R = \frac{0{,}026}{F_L} \sqrt{n_2\, Re_v} \quad \text{(not to exceed } F_R = 1\text{)} \tag{33}$$

for the laminar flow regime.

NOTE 1 – Select the lowest value from equations (32) and (33). If $Re_v < 10$, use only equation (33).

NOTE 2 – Equation (33) is applicable to fully developed laminar flow (straight lines in figure 3).

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 IEC 60534-2-3. 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:

$$F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2}{N_2}(\Sigma\zeta_1)\left(\frac{C}{d^2}\right)^2}} \qquad (34)$$

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 vapour pressure of the liquid at inlet temperature. At vapour 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:

$$F_F = 0{,}96 - 0{,}28\sqrt{\frac{p_v}{p_c}} \qquad (35)$$

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;

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 beyond the range of influence since turbulent flow almost always exists.

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

Y may be calculated using equation (36).

$$Y = 1 - \frac{x}{3 F_\gamma \, x_T} \qquad (36)$$

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 (36) 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 IEC 60534-2-3.

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:

$$x_{TP} = \frac{\dfrac{x_T}{F_p^2}}{1 + \dfrac{x_T \, \zeta_i}{N_5} \left(\dfrac{C_i}{d^2} \right)^2} \qquad (37)$$

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

$$F_\gamma = \frac{\gamma}{1{,}40} \tag{38}$$

NOTE – See annex C for values of γ and F_γ.

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 behaviour 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

$$p_r = \frac{p_1}{p_c} \tag{39}$$

$$T_r = \frac{T_1}{T_c} \tag{40}$$

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	$p \times \Delta p$	ρ	T	d, D	ν
N_1	1×10^{-1}	$8,65 \times 10^{-2}$	–	m^3/h	kPa	kg/m^3	–	–	–
	1	$8,65 \times 10^{-1}$	–	m^3/h	bar	kg/m^3	–	–	–
N_2	$1,60 \times 10^{-3}$	$2,14 \times 10^{-3}$	–	–	–	–	–	mm	–
N_4	$7,07 \times 10^{-2}$	$7,60 \times 10^{-2}$	–	m^3/h	–	–	–	–	m^2/s
N_5	$1,80 \times 10^{-3}$	$2,41 \times 10^{-3}$	–	–	–	–	–	mm	–
N_6	3,16	2,73	kg/h	–	kPa	kg/m^3	–	–	–
	$3,16 \times 10^{1}$	$2,73 \times 10^{1}$	kg/h	–	bar	kg/m^3	–	–	–
N_8	1,10	$9,48 \times 10^{-1}$	kg/h	–	kPa	–	K	–	–
	$1,10 \times 10^{2}$	$9,48 \times 10^{1}$	kg/h	–	bar	–	K	–	–
N_9 (t_s = 0 °C)	$2,46 \times 10^{1}$	$2,12 \times 10^{1}$	–	m^3/h	kPa	–	K	–	–
	$2,46 \times 10^{3}$	$2,12 \times 10^{3}$	–	m^3/h	bar	–	K	–	–
N_9 (t_s = 15 °C)	$2,60 \times 10^{1}$	$2,25 \times 10^{1}$	–	m^3/h	kPa	–	K	–	–
	$2,60 \times 10^{3}$	$2,25 \times 10^{3}$	–	m^3/h	bar	–	K	–	–
N_{17}	$1,05 \times 10^{-3}$	$1,21 \times 10^{-3}$	–	–	–	–	–	mm	–
N_{18}	$8,65 \times 10^{-1}$	1,00	–	–	–	–	–	mm	–
N_{19}	2,5	2,3	–	–	–	–	–	mm	–
N_{22} (t_s = 0 °C)	$1,73 \times 10^{1}$	$1,50 \times 10^{1}$	–	m^3/h	kPa	–	K	–	–
	$1,73 \times 10^{3}$	$1,50 \times 10^{3}$	–	m^3/h	bar	–	K	–	–
N_{22} (t_s = 15 °C)	$1,84 \times 10^{1}$	$1,59 \times 10^{1}$	–	m^3/h	kPa	–	K	–	–
	$1,84 \times 10^{3}$	$1,59 \times 10^{3}$	–	m^3/h	bar	–	K	–	–
N_{27}	$7,75 \times 10^{-1}$	$6,70 \times 10^{-1}$	kg/h	–	kPa	–	K	–	–
	$7,75 \times 10^{-1}$	$6,70 \times 10^{-1}$	kg/h	–	bar	–	K	–	–
N_{33}	$6,00 \times 10^{1}$	$5,58 \times 10^{1}$	–	–	–	–	–	mm	–

NOTE – Use of the numerical constants 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.

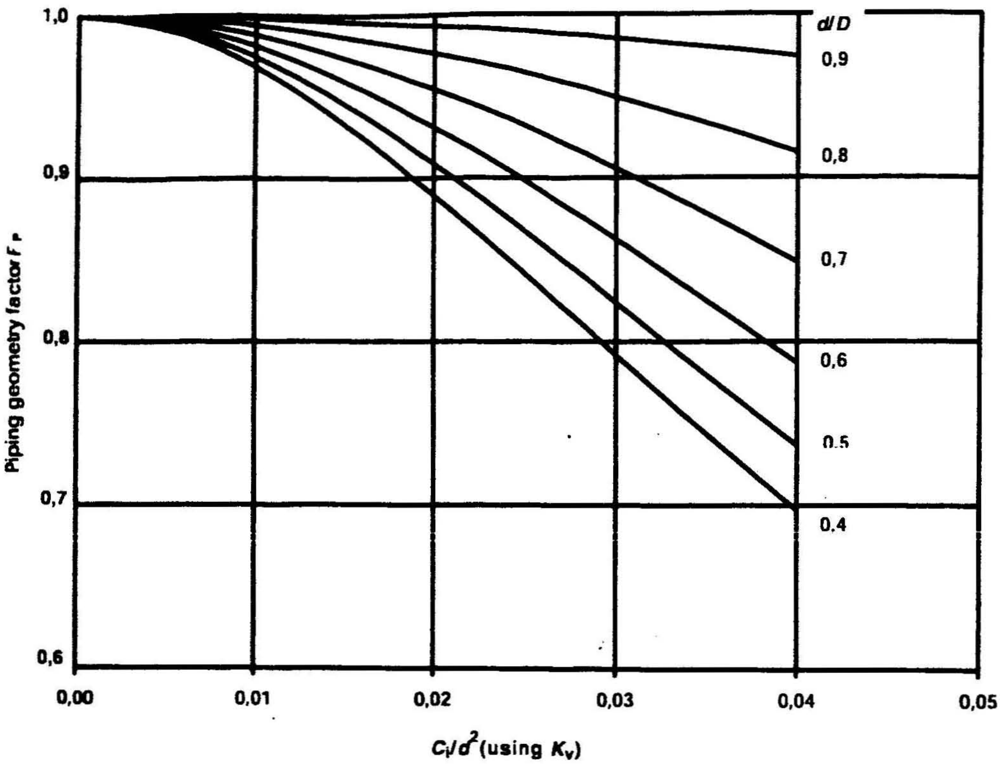
Table 2 – Typical values of valve style modifier F_d, liquid pressure recovery factor F_L, and pressure differential ratio factor x_T at full rated travel [1)]

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	$\frac{N_{19}\sqrt{C \times F_L}}{D_o}$
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 (centred 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
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 manufacturer.

2) Flow tends to open or close the valve, i.e. push the closure member away from or towards the seat.

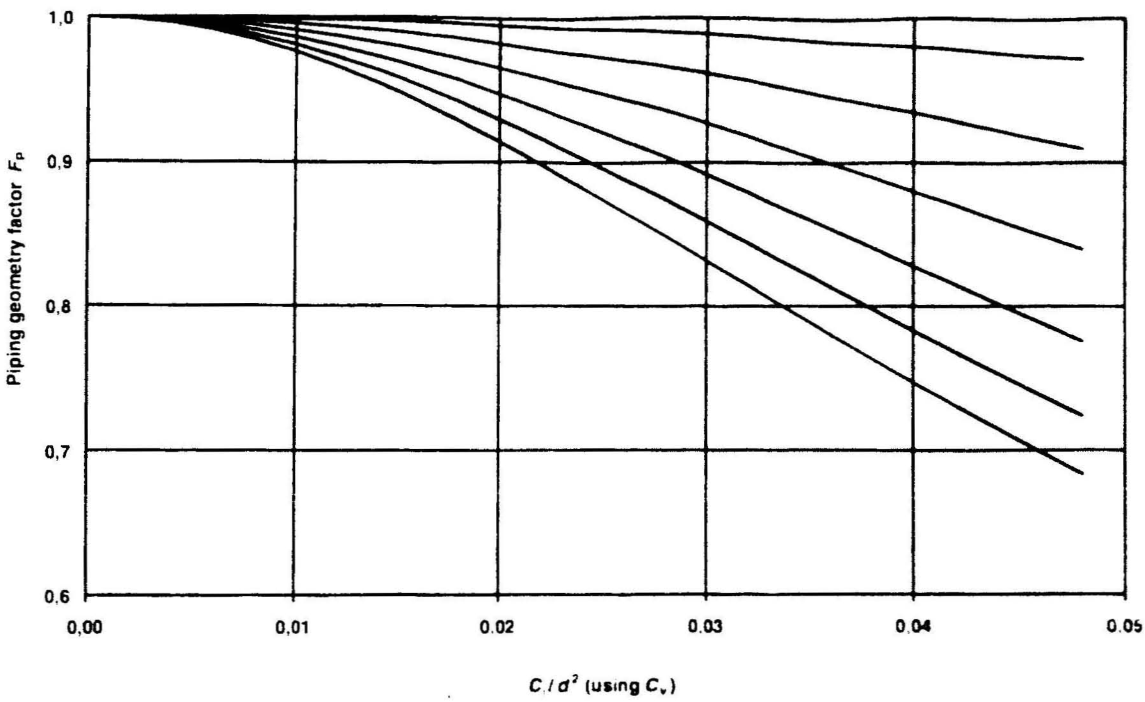
3) Outward means flow from centre of cage to outside, and inward means flow from outside of cage to centre.



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.

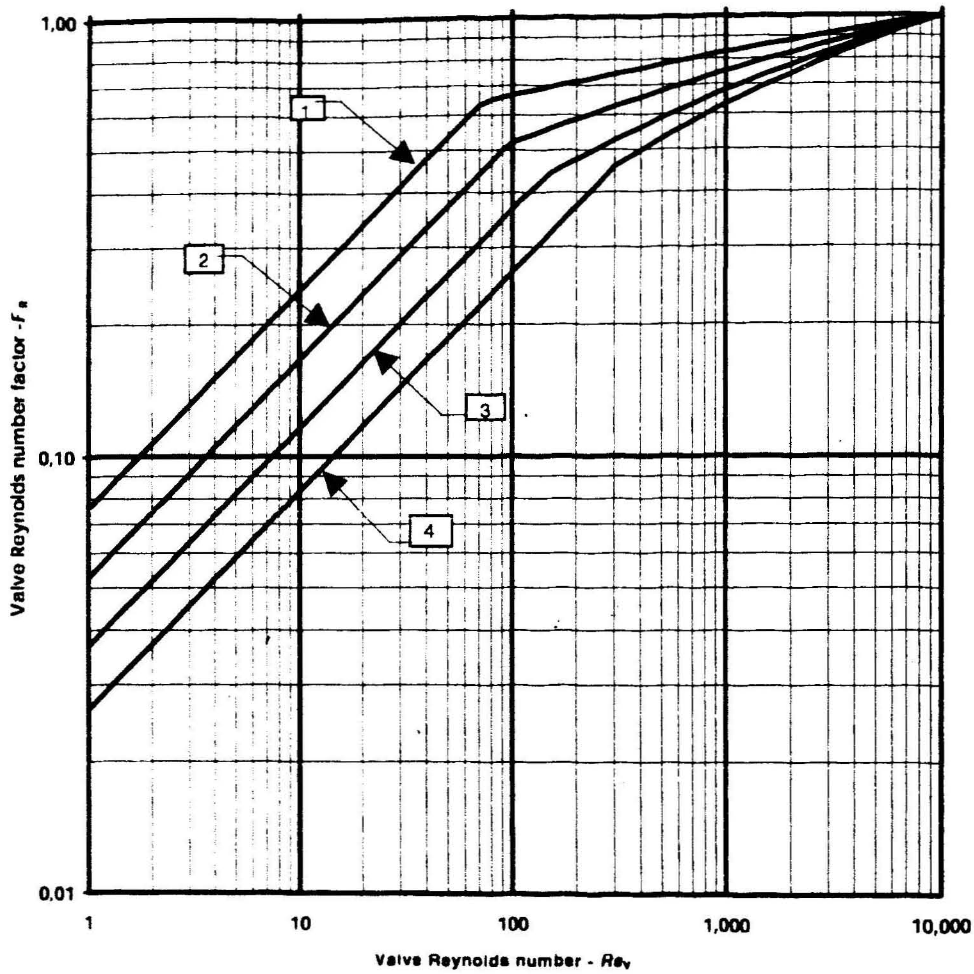
Figure 2a – Piping geometry factor F_P for K_v/d^2



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



Curve 1 is for $C/d^2 = 0,016\ N_{18}$

Curve 2 is for $C/d^2 = 0,023\ N_{18}$

Curve 3 is for $C/d^2 = 0,033\ N_{18}$

Curve 4 is for $C/d^2 = 0,047\ N_{18}$

NOTE – Curves are based on F_L being approximately 1,0.

Figure 3 – Reynolds number factor F_R

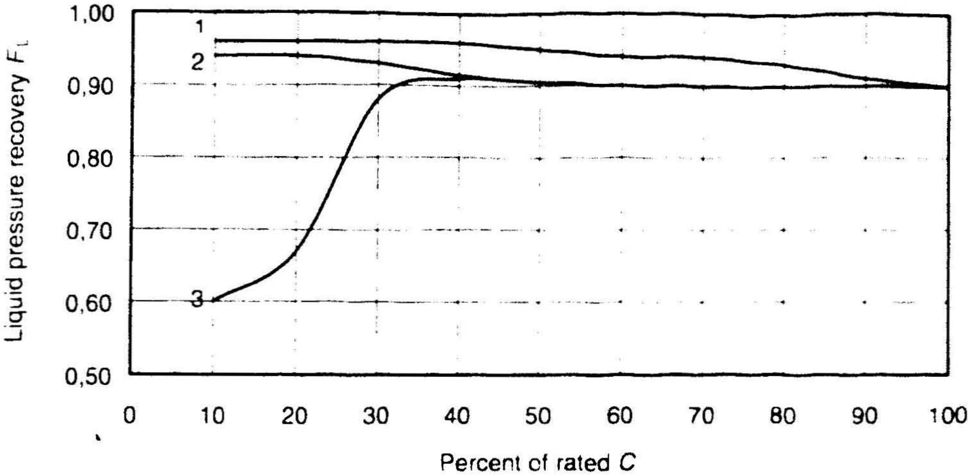


Figure 4a – Double seated globe valves and cage guide globe valves (see legend)

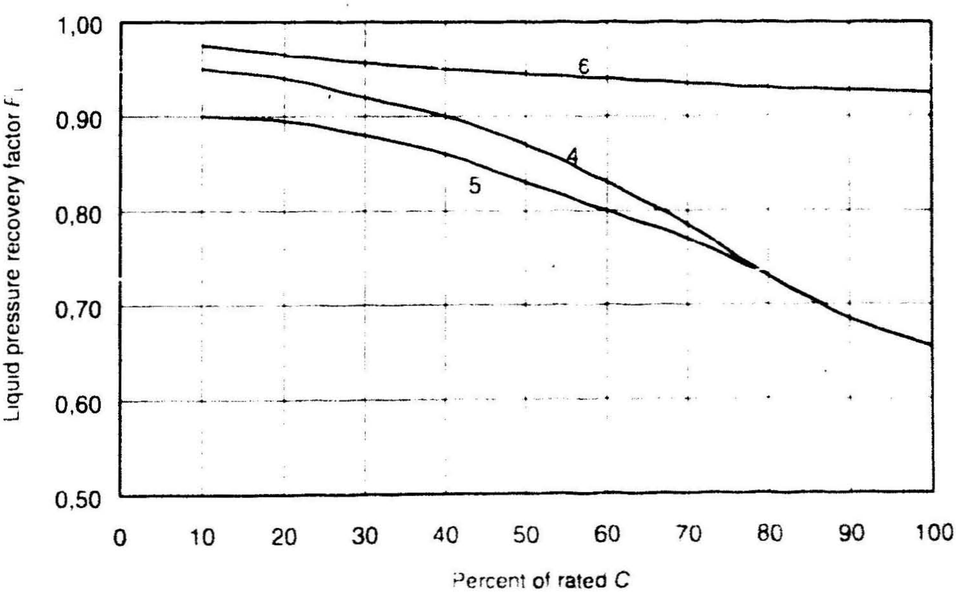


Figure 4b – Butterfly valves and contoured small flow valve (see legend)

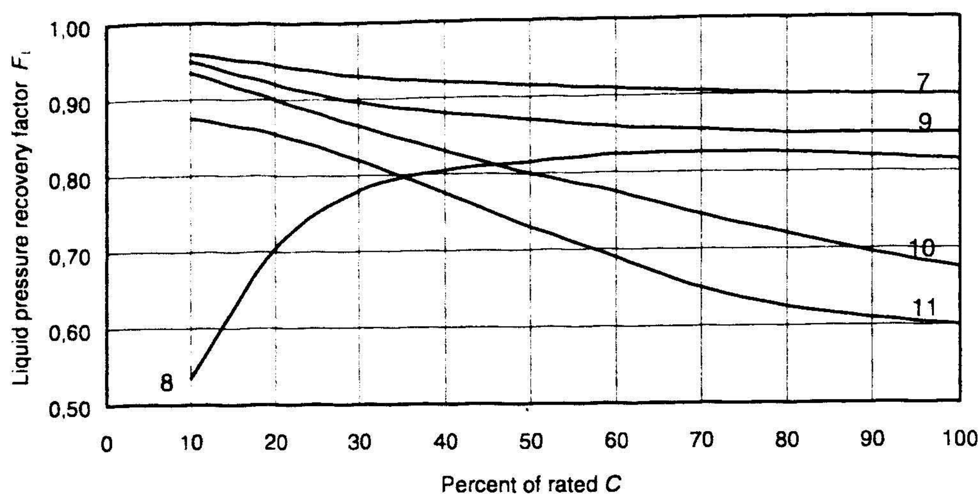


Figure 4c – Contoured globe valves, eccentric spherical plug valves, and segmented ball valve (see legend)

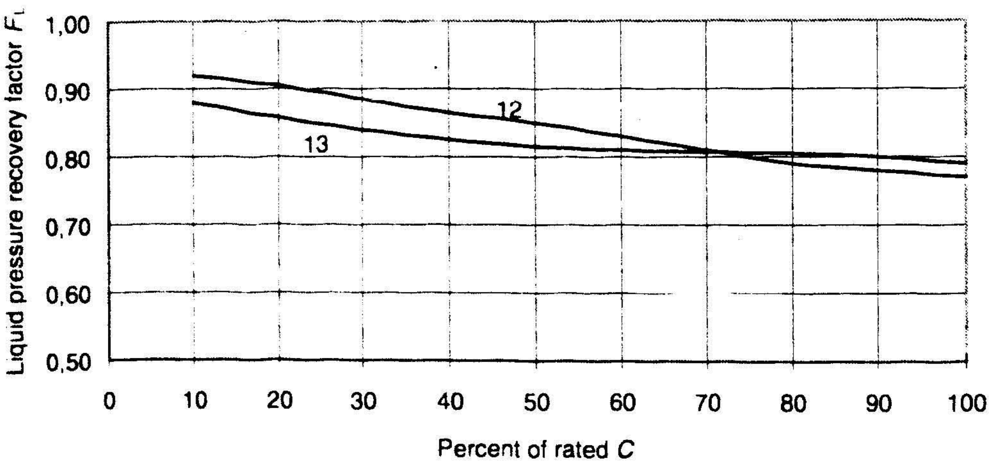


Figure 4d – Eccentric conical plug valves (see legend)

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 small 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

12 Eccentric conical plug valve, flow-to-close

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

Figure 4 – Variation of F_L with percent of rated C

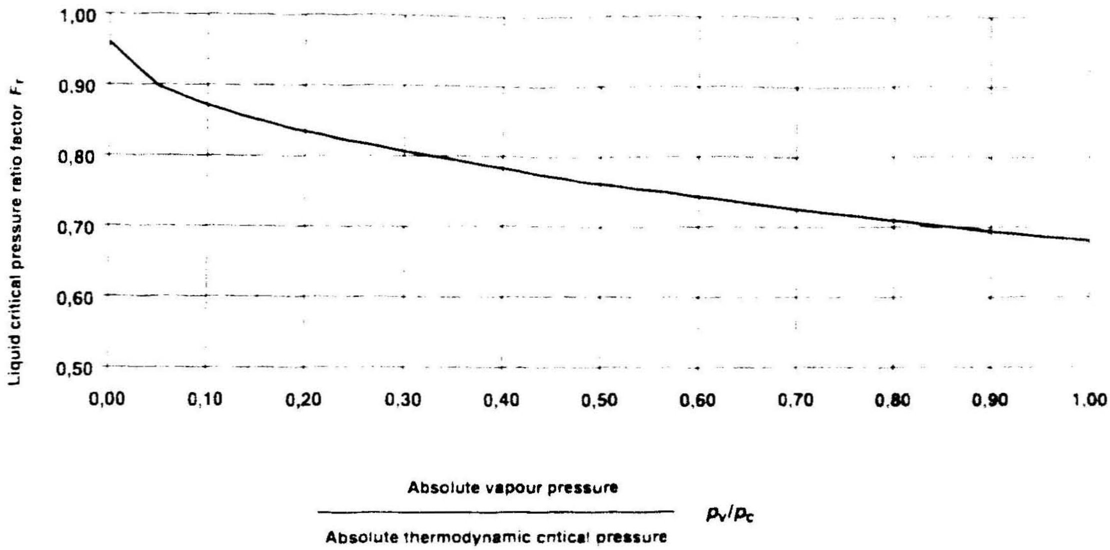


Figure 5 – Liquid critical pressure ratio factor F_F

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, millimetres squared;
- d_H hydraulic diameter of a single flow passage, millimetres;
- d_i inside diameter of annular flow passage (see figure A.1), millimetres;
- d_o equivalent circular diameter of the total flow area, millimetres;
- D_o diameter of seat orifice (see figures A.1 and A.2), millimetres;
- l_w wetted perimeter of a single flow passage, millimetres;
- 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;
- 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:

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

For valves having $C_i/d^2 \le 0{,}016\ N_{18}$, F_d is calculated as follows:

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

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:

$$F_d = \frac{d_H}{d_o} \qquad \text{(A.3)}$$

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

$$d_H = \frac{4\ A_o}{l_w} \qquad \text{(A.4)}$$

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

$$d_o = \sqrt{\frac{4\, N_o\, A_o}{\pi}} \qquad \text{(A.5)}$$

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:

$$A_o = \frac{N_{23}\, C\, F_L}{N_o} \qquad \text{(A.6)}$$

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

Therefore, since $N_o = 1$,

$$d_o = \sqrt{\frac{4\, A_o}{\pi}}$$

$$= \sqrt{\frac{4\, N_{23}\, C\, F_L}{\pi}} \qquad \text{(A.7)}$$

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

$$= \frac{4\, N_{23}\, C\, F_L}{\pi (D_o + d_i)} \qquad \text{(A.8)}$$

From above,

$$F_d = \frac{d_H}{d_o} \qquad \text{(A.3)}$$

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

$$= \frac{1{,}13\sqrt{N_{23}\, C\, F_L}}{D_o + d_i} \qquad \text{(A.9)}$$

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

$$d_i = D_o - \frac{N_{23}\, C\, F_L}{D_o} \qquad \text{(A.10)}$$

$$F_d = \frac{1{,}13\sqrt{N_{23}\, C\, F_L}}{2\, D_o - \dfrac{N_{23}\, C\, F_L}{D_o}} \qquad \text{(A.11)}$$

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:

$$N_{23}\, C F_L = \frac{0{,}125\pi\ D_o^{\,2}(\mu_1+\mu_2)\left(\dfrac{1-\sin\alpha}{\sin\beta}\right)}{\zeta_{B1}} \qquad \text{(A.12)}$$

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).

$$A_o = \frac{0{,}55\, D_o^{\,2}\left(\dfrac{1-\sin\alpha}{\sin\beta}\right)}{N_o} \qquad \text{(A.13)}$$

and since $\beta = 90°$ for swing-through butterfly valves,

$$A_o = \frac{0{,}55\, D_o^{\,2}\,(1-\sin\alpha)}{N_o} \qquad \text{(A.14)}$$

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

$$A_o = 0{,}275\, D_o^{\,2}\,(1-\sin\alpha) \qquad \text{(A.15)}$$

and

$$d_o = \sqrt{\frac{4\, A_o\, N_o}{\pi}}$$

$$= 0{,}837\, D_o \sqrt{1-\sin\alpha} \qquad \text{(A.16)}$$

$$d_H = \frac{4\, A_o}{0{,}59\,\pi\, D_o}$$

$$= 0{,}59\, D_o\,(1-\sin\alpha) \qquad \text{(A.17)}$$

NOTE – $0{,}59\ \pi\ D_o$ is taken as the wetted perimeter l_w of each semi-circle allowing for jet contraction and hub.

$$F_d = \frac{d_H}{d_o} \qquad \text{(A.3)}$$

which results in

$$F_d = 0{,}7\sqrt{1-\sin\alpha} \qquad \text{(A.18)}$$

Table A.1 – Numerical constant N

Constant	Flow coefficient *C*		Formulae unit		
	K_v	C_v	Q	d	ν
N_{23}	$1,96 \times 10^1$	$1,70 \times 10^1$	–	mm	–
N_{26}	$1,28 \times 10^7$	$9,00 \times 10^6$	m³/h	mm	m²/s
N_{31}	$2,1 \times 10^4$	$1,9 \times 10^4$	m³/h	–	m²/s
N_{32}	$1,4 \times 10^2$	$1,27 \times 10^2$	–	mm	–

NOTE – 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.

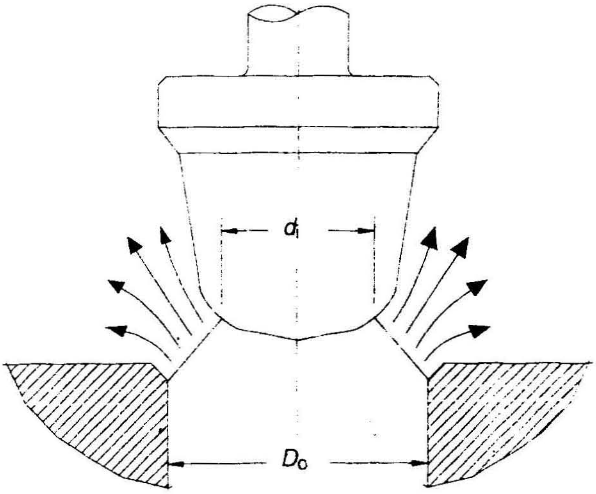


Figure A.1 – Single seated, parabolic plug (flow tending to open)

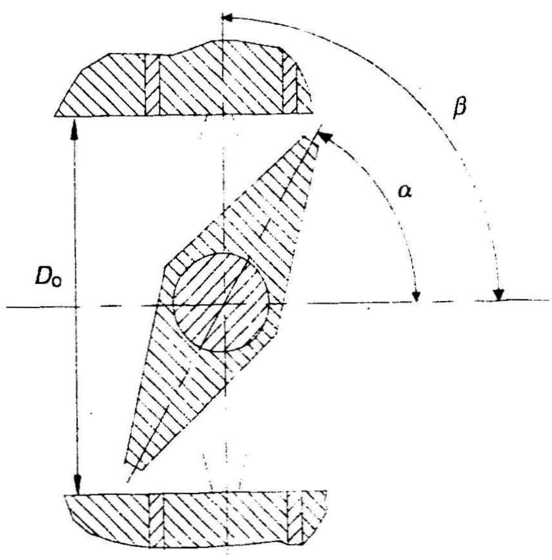
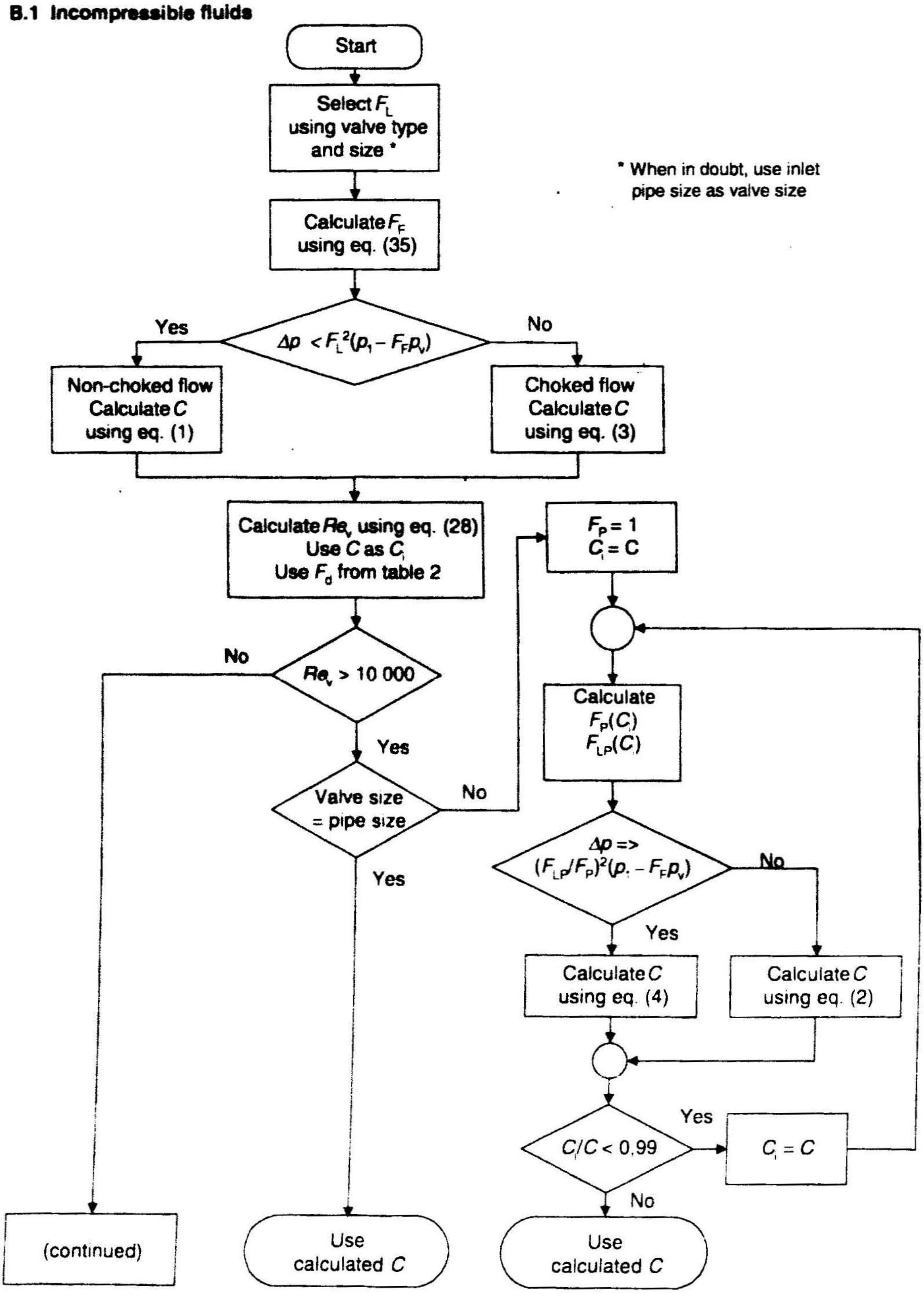


Figure A.2 – Swing-through butterfly valve

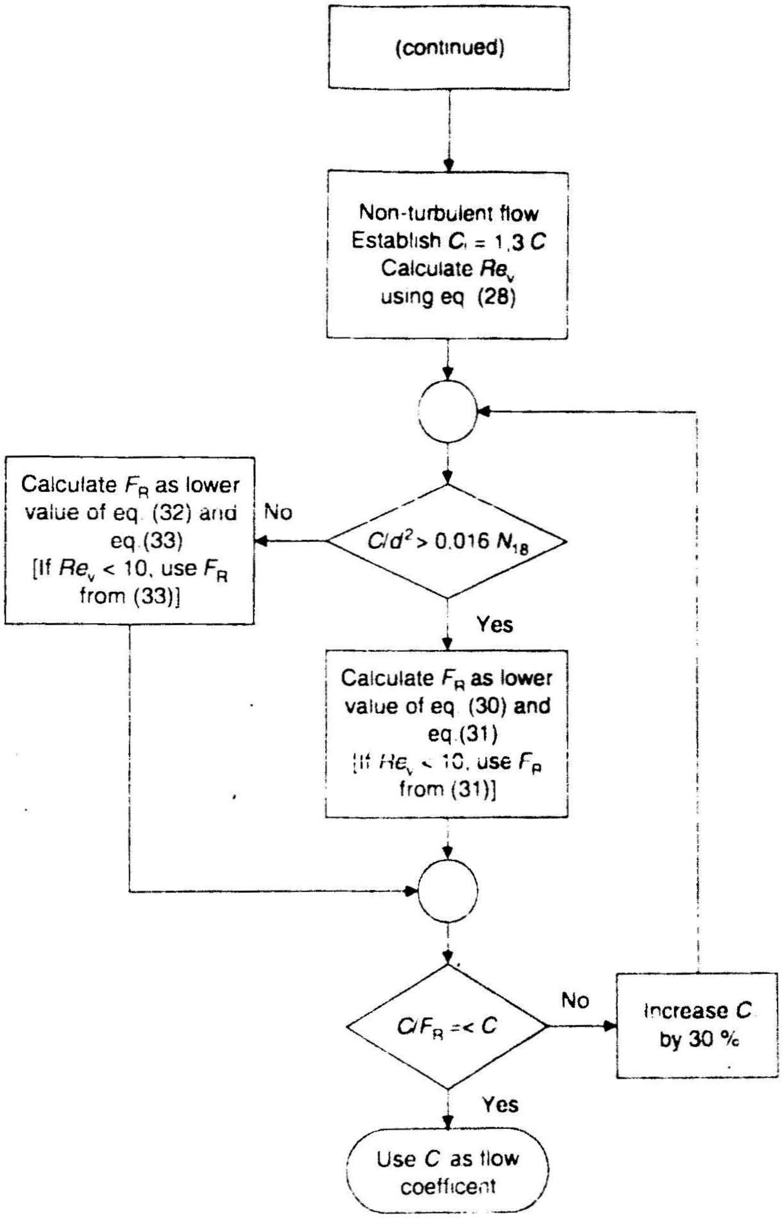
Annex B
(informative)

Control valve sizing flow charts

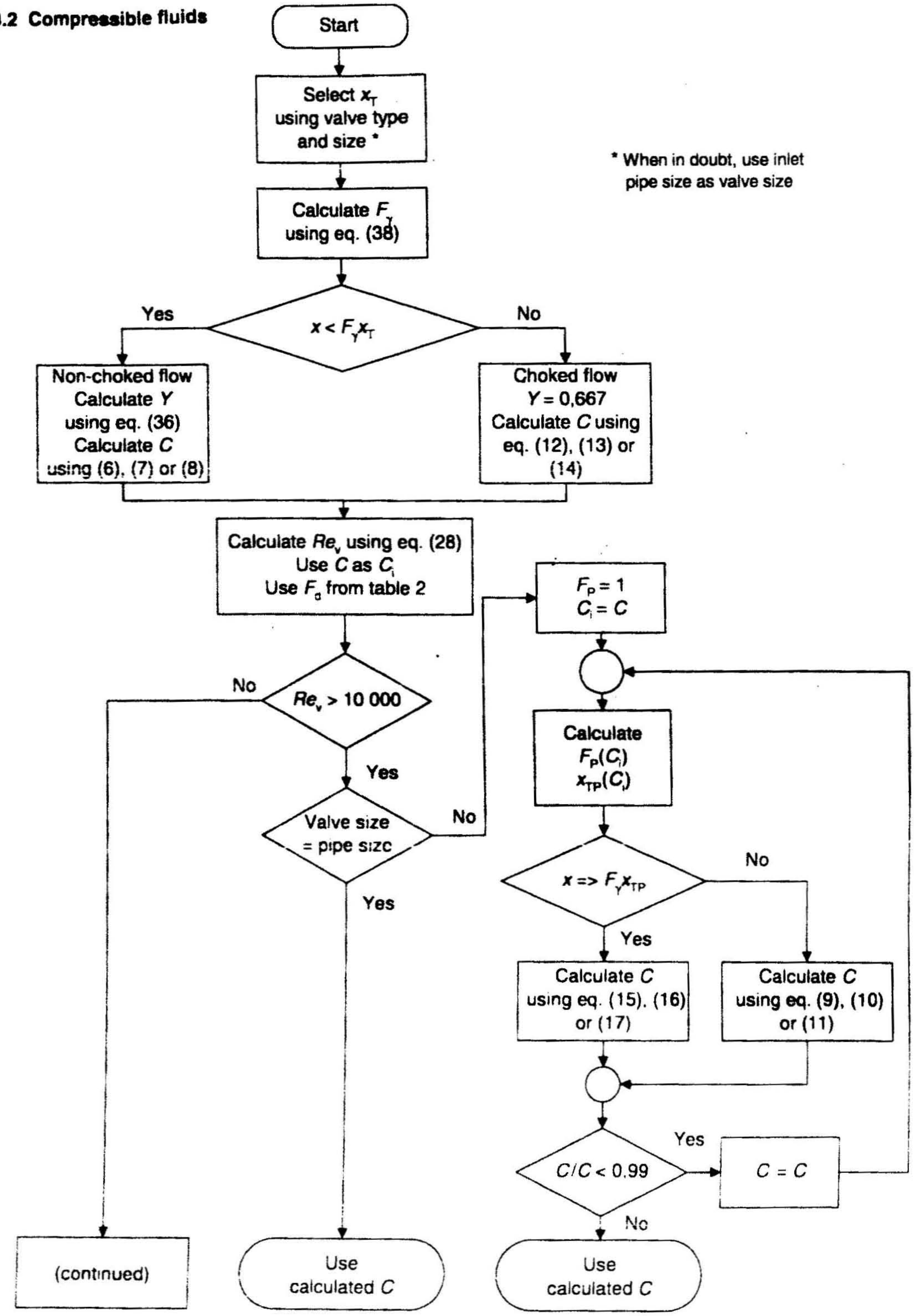
B.1 Incompressible fluids



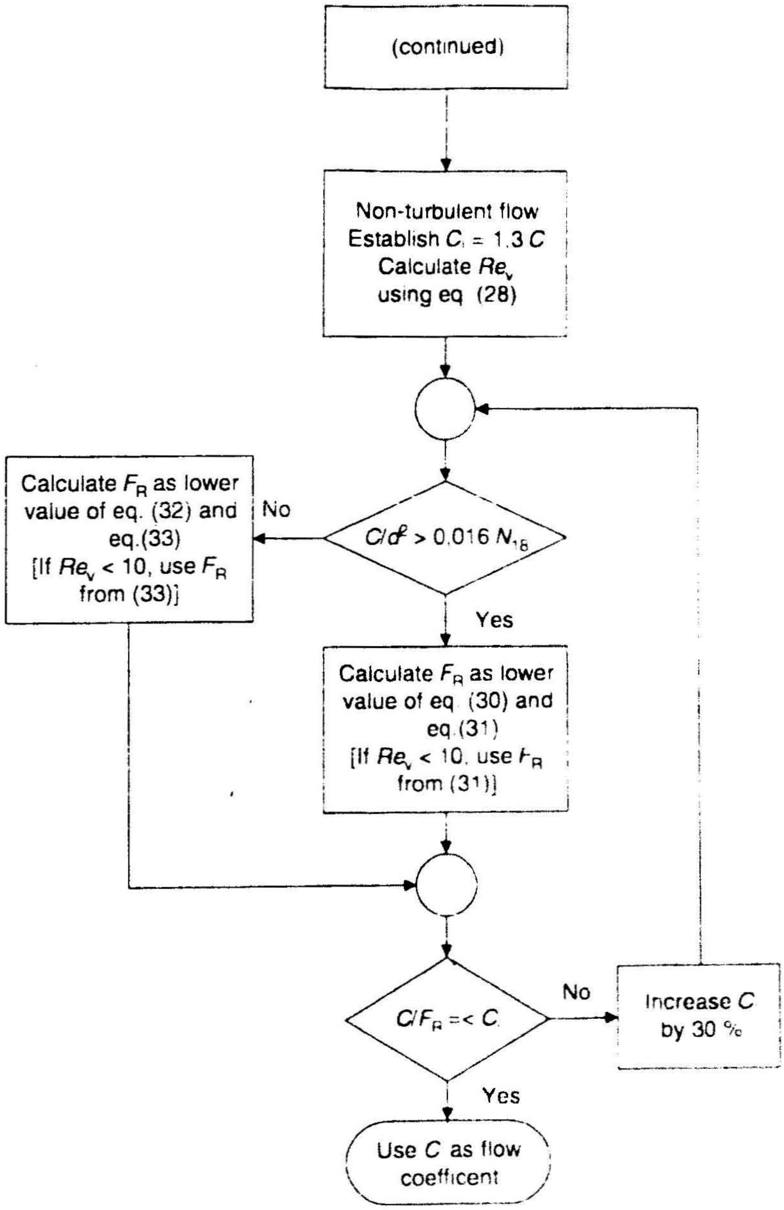
B.1 Incompressible fluids (continued)



B.2 Compressible fluids



B.2 Compressible fluids (continued)



Annex C
(informative)

Physical constants [1)]

Gas or vapour	Symbol	M	γ	F_γ	p_c [2)]	T_c [3)]
Acetylene	C_2H_2	26,04	1,30	0,929	6 140	309
Air	–	28,97	1,40	1,000	3 771	133
Ammonia	NH_3	17,03	1,32	0,943	11 400	406
Argon	A	39,948	1,67	1,191	4 870	151
Benzene	C_6H_6	78,11	1,12	0,800	4 924	562
Isobutane	C_4H_9	58,12	1,10	0,784	3 638	408
n-Butane	C_4H_{10}	58,12	1,11	0,793	3 800	425
Isobutylene	C_4H_8	56,11	1,11	0,790	4 000	418
Carbon dioxide	CO_2	44,01	1,30	0,929	7 387	304
Carbon monoxide	CO	28,01	1,40	1,000	3 496	133
Chlorine	Cl_2	70,906	1,31	0,934	7 980	417
Ethane	C_2H_6	30,07	1,22	0,871	4 884	305
Ethylene	C_2H_4	28,05	1,22	0,871	5 040	283
Fluorine	F_2	18,998	1,36	0,970	5 215	144
Freon 11 (trichloromonofluormethane)	CCl_3F	137,37	1,14	0,811	4 409	471
Freon 12 (dichlorodifluoromethane)	CCl_2F_2	120,91	1,13	0,807	4 114	385
Freon 13 (chlorotrifluoromethane)	CClF	104,46	1,14	0,814	3 869	302
Freon 22 (chlorodifluoromethane)	$CHClF_2$	80,47	1,18	0,846	4 977	369
Helium	He	4,003	1,66	1,186	229	5,25
n-Heptane	C_7H_{16}	100,20	1,05	0,750	2 736	540
Hydrogen	H_2	2,016	1,41	1,007	1 297	33,25
Hydrogen chloride	HCl	36,46	1,41	1,007	8 319	325
Hydrogen fluoride	HF	20,01	0,97	0,691	6 485	461
Methane	CH_4	16,04	1,32	0,943	4 600	191
Methyl chloride	CH_3Cl	50,49	1,24	0,889	6 677	417
Natural gas [4)]	–	17,74	1,27	0,907	4 634	203
Neon	Ne	20,179	1,64	1,171	2 726	44,45
Nitric oxide	NO	63,01	1,40	1,000	6 485	180
Nitrogen	N_2	28,013	1,40	1,000	3 394	126
Octane	C_8H_{18}	114,23	1,66	1,186	2 513	569
Oxygen	O_2	32,000	1,40	1,000	5 040	155
Pentane	C_5H_{12}	72,15	1,06	0,757	3 374	470
Propane	C_3H_8	44,10	1,15	0,821	4 256	370
Propylene	C_3H_6	42,08	1,14	0,814	4 600	365
Saturated steam	–	18,016	1,25 – 1,32 [14)]	0,893 – 0,943 [4)]	22 119	647
Sulphur dioxide	SO_2	64,06	1,26	0,900	7 822	430
Superheated steam	–	18,016	1,315	0,939	22 119	647

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.

Annex D
(informative)

Examples of sizing calculations

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/m³
Vapour pressure:	$p_v = 70{,}1$ kPa
Thermodynamic critical pressure:	$p_c = 22\ 120$ kPa
Kinematic viscosity:	$\nu = 3{,}26 \times 10^{-7}$ m²/s
Inlet absolute pressure:	$p_1 = 680$ kPa
Outlet absolute pressure:	$p_2 = 220$ kPa
Flow rate:	$Q = 360$ m³/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:

$$F_F = 0{,}96 - 0{,}28\sqrt{\frac{p_v}{p_c}} = 0{,}944 \qquad (35)$$

where

$p_v = 70{,}1$ kPa;

$p_c = 22\ 120$ kPa.

Next, determine the type of flow:

$$F_L^2 (p_1 - F_F \times p_v) = 497{,}2\,\text{kPa}$$

which is more than the differential pressure (Δp = 460 kPa); therefore, the flow is non-choked, and the flow coefficient C is calculated using equation (2):

$$C=\frac{Q}{N_1}\sqrt{\frac{\rho_1/\rho_o}{\Delta p}}=165\ \text{m}^3/\text{h for } K_v \tag{1}$$

where

Q = 360 m^3/h;

N_1 = 1 × 10^{-1} from table 1;

ρ_1/ρ_o = 0,965;

Δp = 460 kPa.

Next, calculate Re_v:

$$Re_v=\frac{N_4\,F_d\,Q}{\nu\sqrt{C_i\,F_L}}\left[\frac{F_L^2\,C_i^2}{N_2\,D^4}+1\right]^{1/4}=2{,}967\times10^6 \tag{28}$$

where

N_2 = 1,60 × 10^{-3} from table 1;

N_4 = 7,07 × 10^{-2} from table 1;

F_d = 0,46;

Q = 360 m^3/h;

ν = 3,26 × 10^{-7} m^2/s;

C_i = C = K_v = 165 m^3/h;

F_L = 0,90;

D = 150 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:	ρ_1 = 965,4 kg/m^3
Vapour pressure:	p_v = 70,1 kPa
Thermodynamic critical pressure:	p_c = 22 120 kPa
Kinematic viscosity:	ν = 3,26 × 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:

$$F_F = 0{,}96 - 0{,}28\sqrt{\frac{p_v}{p_c}} = 0{,}944 \tag{35}$$

where

p_v = 70,1 kPa;

p_c = 22 120 kPa.

Next, determine the type of flow:

$$F_L^2(p_1 - F_F \times p_v) = 221 \text{ kPa}$$

which is less than the differential pressure (Δp = 460 kPa); therefore, the flow is choked and the flow coefficient C is calculated using equation (3):

$$C = \frac{Q}{N_1 F_L}\sqrt{\frac{\rho_1/\rho_o}{p_1 - F_F\, p_v}} = 238 \text{ m}^3/\text{h for } K_v \tag{3}$$

where

Q = 360 m^3/h;

$N_1 = 1 \times 10^{-1}$ from table 1;

F_L = 0,60;

ρ_1/ρ_o = 0,965;

p_1 = 680 kPa,

F_F = 0,944;

p_v = 70,1 kPa.

Next, calculate Re_v:

$$Re_v = \frac{N_4 F_d Q}{v\sqrt{C_i F_L}}\left[\frac{F_L^2 C_i^2}{N_2 D^4} + 1\right]^{1/4} = 6{,}598 \times 10^6 \tag{28}$$

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 m^3/h;

$v = 3{,}26 \times 10^{-7}$ m^2/s;

$C_i = C = K_v$ = 238 m^3/h;

F_L = 0,60;

D = 100 mm.

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 K
Molecular mass:	M = 44,01 kg/kmol
Kinematic viscosity:	ν = 1,743 × 10^{-5} m^2/s
Specific heat ratio:	γ = 1,30
Compressibility factor:	Z = 0,988
Inlet absolute pressure:	p_1 = 680 kPa
Outlet absolute pressure:	p_2 = 310 kPa
Flow rate:	Q = 3 800 standard m^3/h at 101,325 kPa and 0 °C
Inlet pipe size:	D_1 = 80 mm
Outlet pipe size:	D_2 = 100 mm
Reducers:	short length, concentric

Valve data:

Valve style:	rotary
Trim:	eccentric rotary plug
Flow direction:	flow-to-open
Valve size:	d = 50 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:

$$F_\gamma = \frac{\gamma}{1{,}40} = 0{,}929 \qquad (38)$$

where

γ = 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 (36):

$$Y = 1 - \frac{x}{3\,F_\gamma\, x_T} = 0{,}674 \qquad (36)$$

where

x = 0,544;

F_γ = 0.929;

x_T = 0,60.

$$C = \frac{Q}{N_9\, F_p\, p_1\, Y}\sqrt{\frac{M\, T_1\, Z}{x}} = 62.7\ \text{m}^3/\text{h for } K_v \qquad (11)$$

where

Q = 3 800 m^3/h

N_9 = 2,46 × 10^1 for t_s = 0 °C from table 1

assume F_P = 1

p_1 = 680 kPa

Y = 0,674

M = 44.01 kg/kmol

T_1 = 433 K

Z = 0,988

x = 0,544

Now, calculate Re_v using equation (28):

$$Re_v = \frac{N_4\, F_d\, Q}{\nu\sqrt{C_i\, F_L}}\left[\frac{F_L^2\, C_i^2}{N_2\, D^4} + 1\right]^{1/4} = 8{,}96\times 10^5 \qquad (28)$$

where

N_2 = 1,60 × 10^{-3} from table 1

N_4 = 7.07 × 10^{-2} from table 1

F_d = 0,42

Q = 3 800 m^3/h

ν = 1,743 × 10^{-5} m^2/s

C_i = C = K_v = 62,7 m^3/h

F_L = 0.85

D = 80 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:

$$\zeta_1 = 0{,}5\left[1-(d/D_1)^2\right]^2 = 0{,}186 \qquad (23)$$

where

d = 50 mm

D_1 = 80 mm

$$\zeta_2 = 1{,}0\left[1-(d/D_2)^2\right]^2 = 0{,}563 \qquad (24)$$

where

d = 50 mm

D_2 = 100 mm

and the Bernoulli coefficients are:

$$\zeta_{B1} = 1-(d/D_1)^4 = 0{,}847 \qquad (22)$$

where

d = 50 mm

D_1 = 80 mm

$$\zeta_{B2} = 1-(d/D_2)^4 = 0{,}938 \qquad (22)$$

where

d = 50 mm

D_2 = 100 mm

The effective head loss coefficient of the inlet and outlet reducers is:

$$\Sigma\zeta = \zeta_1 + \zeta_2 + \zeta_{B1} - \zeta_{B2} = 0{,}658 \qquad (21)$$

where

ζ_1 = 0,186

ζ_2 = 0,563

ζ_{B1} = 0,847

ζ_{B2} = 0,938

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

$$F_{p(2)} = \frac{1}{\sqrt{1+\frac{\Sigma\zeta}{N_2}\left(\frac{C_1}{d^2}\right)^2}} = 0{,}891 \qquad (20)$$

where

$\Sigma\zeta$ = 0,658

N_2 = 1,60 × 10^{-3} from table 1

C_1 = 62,7 m^3/h

d = 50 mm

Since $F_{P(2)}/F_{P(1)}$ = 0,891/1 < 0,99, one more iteration step shall be done.

$$C_2 = \frac{C}{F_{p(2)}} = \frac{62,7}{0,891} = 70,4\ \text{m}^3/\text{h}$$

$$F_{p(3)} = \frac{1}{\sqrt{1 + \frac{\Sigma\zeta}{N_2}\left(\frac{C_2}{d^2}\right)^2}} = 0,868 \qquad (20)$$

where

$\Sigma\zeta$ = 0,658

N_2 = 1,60 × 10^{-3} from table 1

C_2 = 70,4 m^3/h

d = 50 mm

Now, $F_{P(3)}/F_{P(2)}$ = 0,868/0,891 > 0,99 so $F_{P(3)}$ will be used as F_P for the final calculation.

$$x_{TP} = \frac{\frac{x_T}{F_p^2}}{1 + \frac{x_T \zeta_i}{N_5}\left(\frac{C_2}{d^2}\right)^2} = 0,626 \qquad (37)$$

where

x_T = 0,60

F_p = 0,868

$\zeta_i = \zeta_1 + \zeta_{B1}$ = 1,033

N_5 = 1,80 × 10^{-3} from table 1

C_2 = 70,4 m^3/h

d = 50 mm

and with this $F_\gamma x_{TP}$ = 0,582, which is greater than x = 0,544.

Finally, C results from equation (11) as follows:

$$C = \frac{Q}{N_9 F_p p_1 Y}\sqrt{\frac{M T_1 Z}{x}} = 72,2\ \text{m}^3/\text{h for } K_v \qquad (11)$$

where

Q = 3 800 m^3/h

N_9 = 2,46 × 10^1 for t_s = 0 °C from table 1

F_p = 0,868

p_1 = 680 kPa

Y = 0,674

M = 44,01 kg/kmol

T_1 = 433 K

Z = 0,988

x = 0,544

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

Process data:

Fluid:	argon gas
Inlet temperature:	$T_1 = 320$ K
Inlet absolute pressure:	$p_1 = 2,8$ bar (absolute)
Outlet absolute pressure:	$p_2 = 1,3$ bar (absolute)
Flow rate:	$Q = 0,46$ standard m^3/h at 1 013,25 mbar and 15 °C
Molecular mass:	$M = 39,95$
Kinematic viscosity:	$\nu = 1,338 \times 10^{-5}$ m^2/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:	$D_o = 5$ mm
Valve size:	$d = 15$ mm
Internal diameter of piping:	$D = 15$ mm

Calculation:

The first step is to check the Reynolds number Re_v:

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

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 choked flow equation (14) to calculate C, then multiplying C by 1,3 in accordance with the iteration procedure of 8.1.

$$C = \frac{Q}{0,667 N_9 p_1} \sqrt{\frac{M T_1 Z}{F_\gamma x_T}} = 0,0127 \text{ for } C_v \quad (14)$$

where

$Q = 0,46$ m^3/h

$N_9 = 2,25 \times 10^3$ for $t_s = 15$ °C from table 1

$p_1 = 2,8$ bar

$M = 39,95$ kg/kmol

$T_1 = 320$ K

$Z = 1$

$F_\gamma = 1,19$

$x_T = 0,8$

$$C_i = 1{,}3\,C = 0{,}0165 \text{ for } C_v \tag{26}$$

where

$C = 0{,}0127$ for C_v

Next, estimate F_d from the equation in table 2:

$$F_d = \frac{N_{19}\sqrt{C F_L}}{D_o} = 0.058$$

where

$C = C_i = 0{,}016\,5$ for C_v

$F_L = 0{,}98$

$N_{19} = 2{,}3$ from table 1

$D_o = 5$ mm

Calculate Re_v as follows:

$$Re_v = \frac{N_4 F_d Q}{\nu\sqrt{C_i F_L}}\left[\frac{F_L^2 C_i^2}{N_2 D^4} + 1\right]^{1/4} = 1202 \tag{28}$$

where

$N_2 = 2{,}12 \times 10^{-3}$ from table 1

$N_4 = 7{,}6 \times 10^{-2}$ from table 1

$F_d = 0{,}058$

$Q = 0{,}46$ m^3/h

$\nu = 1{,}338 \times 10^{-5}$ m^2/s

$F_L = 0{,}98$

$C_i = 0{,}0165$ for C_v

D = 15 mm

Determine if $C/d^2 < 0{,}016\,N_{18}$:

$$C/d^2 = 7{,}333 \times 10^{-5}$$

$$0.016\,N_{18} = 0{,}016$$

$$C/d^2 < 0{,}016\,N_{18}$$

where

$N_{18} = 1{,}00$ from table 1

$C = 0{,}016\,5$

$d = 15$ mm

Since the Reynolds number is below 10 000, the flow is non-turbulent; hence flow coefficient equation (19) has to be used. Since $C/d^2 < 0{,}016\,N_{18}$ and $Re_v > 10$, calculate F_R from both equations (32) and (33) and use the lower value.

$$n_2 = 1 + N_{33}\left(\frac{C_i}{d^2}\right)^{1/2} = 1{,}476 \tag{32a}$$

where

$N_{33} = 55{,}8$ from table 1

$C_i = 0{,}016\,5$ for C_v

$Re_v = 1\,202$

$d = 15$ mm

$$F_R = 1 + \left(\frac{0{,}33\,F_L^{1/2}}{n_2^{1/4}}\right)\log_{10}\left(\frac{Re_v}{10\,000}\right) = 0{,}727 \quad (32)$$

$$F_R = 1 + (0{,}296)\,(-0{,}920) = 0{,}727$$

where

F_L = 0,98

n_2 = 1,478

Re_v = 1 202

$$F_R = \frac{0{,}026}{F_L}\sqrt{n_2\,Re_v} = 1{,}12 \quad (33)$$

NOTE – F_R is limited to 1.

where

F_L = 0,98

n_2 = 1,478

Re_v = 1 202

Use F_R = 0,727, the lower of the two calculated values.

$$C = \frac{Q}{N_{22}\,F_R}\sqrt{\frac{M\,T_1}{\Delta p\,(p_1 + p_2)}} = 0{,}018 \text{ for } C_v \quad (19)$$

where

Q = 0,46 m³/h

N_{22} = 1,59 × 10³ for t_s = 15 °C from table 1

F_R = 0,73

M = 39,95 kg/kmol

T_1 = 320 K

Δp = 1,5 bar

p_1 = 2,8 bar

p_2 = 1,3 bar

Check:

$$\frac{C}{F_R} < C_i \quad (29)$$

$$\frac{0{,}0127}{0{,}727} = 0{,}018 > 0{,}0165$$

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 = 0,021 4

where

C_i = 0,016 5

$$F_d = \frac{N_{19}\sqrt{C F_L}}{D_o} = 0{,}067$$

where

$C = C_i = 0{,}0214$ for C_v

$F_L = 0{,}98$

$N_{19} = 2{,}3$ from table 1

$D_o = 5$ mm

Calculate Re_v:

$$Re_v = \frac{N_4 F_d Q}{\nu\sqrt{C_i F_L}}\left[\frac{F_L^2 C^2}{N_2 D^4} + 1\right]^{1/4} = 1\,202 \qquad (28)$$

where

$N_2 = 2{,}14 \times 10^{-3}$ from table 1

$N_4 = 7{,}6 \times 10^{-2}$ from table 1

$F_d = 0{,}067$

$Q = 0{,}46$ $m^3/$;

$\nu = 1{,}338 \times 10^{-5}$ m^2/s

$F_L = 0{,}98$

$C_i = 0{,}0214$

$D = 15$ mm

Since the value of Re_v remains the same as previously calculated, F_R remains at 0,[illegible]. Therefore, the calculated C will remain at 0,018 and any trim with a rated C of 0,018 or higher for C_v is appropriate

Annex E
(informative)

Bibliography

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