INTERNATIONAL STANDARD

IEC 60534-2-1

First edition
1998-09

Industrial-process control valves –

Part 2-1:
Flow capacity –
Sizing equations for fluid flow under installed conditions

Vannes de régulation des processus industriels –

Partie 2-1:
Capacité d'écoulement –
Equations de dimensionnement des vannes de régulation pour l'écoulement des fluides dans les conditions d'installation

Reference number
IEC 60534-2-1:1998(E)

Numbering

As from 1 January 1997 all IEC publications are issued with a designation in the 60000 series.

Consolidated publications

Consolidated versions of some IEC publications including amendments are available. For example, edition numbers 1.0, 1.1 and 1.2 refer, respectively, to the base publication, the base publication incorporating amendment 1 and the base publication incorporating amendments 1 and 2.

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Commission Electrotechnique Internationale
International Electrotechnical Commission
Международная Электротехническая Комиссия

PRICE CODE X

For price, see current catalogue

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INTERNATIONAL ELECTROTECHNICAL COMMISSION

INDUSTRIAL-PROCESS CONTROL VALVES –

Part 2-1: Flow capacity – Sizing equations for fluid flow under installed conditions

FOREWORD

1) The IEC (International Electrotechnical Commission) is a worldwide organization for standardization comprising all national electrotechnical committees (IEC National Committees). The object of the IEC is to promote international co-operation on all questions concerning standardization in the electrical and electronic fields. To this end and in addition to other activities, the IEC publishes International Standards. Their preparation is entrusted to technical committees; any IEC National Committee interested in the subject dealt with may participate in this preparatory work. International, governmental and non-governmental organizations liaising with the IEC also participate in this preparation. The IEC collaborates closely with the International Organization for Standardization (ISO) in accordance with conditions determined by agreement between the two organizations.
2) The formal decisions or agreements of the IEC on technical matters express, as nearly as possible, an international consensus of opinion on the relevant subjects since each technical committee has representation from all interested National Committees.
3) The documents produced have the form of recommendations for international use and are published in the form of standards, technical reports or guides and they are accepted by the National Committees in that sense.
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International Standard IEC 60534-2-1 has been prepared by subcommittee 65B: Devices, of IEC technical committee 65: Industrial-process measurement and control.

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

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

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

The 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 and E are for information only.

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

INDUSTRIAL-PROCESS CONTROL VALVES –

Part 2-1: Flow capacity – 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

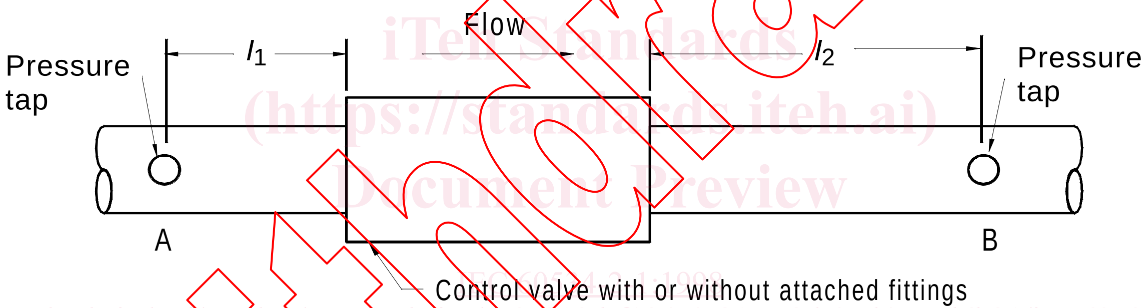
For the purpose of this part of IEC 60534, definitions given in IEC 60534-1 apply with the addition of the following:

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

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

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:

$$C = \frac{Q}{N_1 F_p} \sqrt{\frac{\rho_1 / \rho_o}{\Delta p}} \qquad (2)$$

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

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

The flow coefficient shall be determined as follows:

$$C = \frac{Q}{N_1 F_L} \sqrt{\frac{\rho_1 / \rho_o}{p_1 - F_F \times p_v}} \qquad (3)$$

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

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

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

$$C = \frac{Q}{N_1 F_{LP}} \sqrt{\frac{\rho_1 / \rho_o}{p_1 - F_F \times p_v}} \qquad (4)$$

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:

$$C = \frac{Q}{N_1 F_R} \sqrt{\frac{\rho_1 / \rho_o}{\Delta p}} \qquad (5)$$

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

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

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

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

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:

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

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.

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

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

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

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:

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

Outlet reducer (expander):

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

Inlet and outlet reducers of equal size:

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

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:

$$C_i = 1{,}3\,C \qquad (26)$$

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

$$\frac{C}{F_p} \leq C_i \qquad (27)$$

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 3 using a valve Reynolds number calculated from the following equation:

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

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

$$\frac{C}{F_R} \leq C_i \qquad (29)$$