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INTERNATIONAL STANDARD

NORME INTERNATIONALE

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 pour l'écoulement des fluides dans les conditions d'installation

IEC 60534-2-1:2011

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INDUSTRIAL-PROCESS CONTROL VALVES –

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FOREWORD

1) The International Electrotechnical Commission (IEC) is a worldwide organization for standardization comprising all national electrotechnical committees (IEC National Committees). The object of 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, IEC publishes International Standards, Technical Specifications, Technical Reports, Publicly Available Specifications (PAS) and Guides (hereafter referred to as "IEC Publication(s)"). 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. IEC collaborates closely with the International Organization for Standardization (ISO) in accordance with conditions determined by agreement between the two organizations.

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

This second edition cancels and replaces the first edition published in 1998. This edition constitutes a technical revision.

This edition includes the following significant technical changes with respect to the previous edition:

- the same fundamental flow model, but changes the equation framework to simplify the use of the standard by introducing the notion of Δp_{sizing};
- changes to the non-turbulent flow corrections and means of computing results;
- multi-stage sizing as an Annex.

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

FDIS	Report on voting
65B/783/FDIS	65B/786/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.

This publication has been drafted in accordance with the ISO/IEC Directives, Part 2.

A list of all the parts of the IEC 60534 series, under the general title *Industrial-process control valves*, can be found on the IEC website.

The committee has decided that the contents of this publication will remain unchanged until the stability date indicated on the IEC web site under "http://webstore.iec.ch" in the data related to the specific publication. At this date, the publication will be

- reconfirmed,
- withdrawn,
- replaced by a revised edition, or
- amended.

The contents of the corrigendum of April 2015 have been included in this copy.

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. The equations for incompressible flow may be used with caution for non-vaporizing multi-component liquid mixtures. Refer to Clause 6 for additional information.

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 ideal gas or vapor and are not intended for use with multiphase streams such as gas-liquid, vapor-liquid or gas-solid mixtures. Reasonable accuracy can only be maintained when the specific heat ratio, γ, is restricted to the range $1{,}08 < \gamma < 1{,}65$. Refer to Clause 7.2 for more information.

For compressible fluid applications, this standard is valid for valves with $x_T \leq 0{,}84$ (see Table D.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:

$$\frac{C}{N_{18} d^2} < 0{,}047$$

Note that while the equation structure utilized in this document departs radically from previous versions of the standard, the basic technology is relatively unchanged. The revised equation format was adopted to simplify presentation of the various equations and improve readability of the document.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

IEC 60534-1:2005, *Industrial-process control valves – Part 1: Control valve terminology and general considerations*

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

3 Terms and definitions

For the purposes of this document, the terms and definitions given in IEC 60534-1, and the following apply.

3.1
valve style modifier
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).

3.2
standard volumetric flowrates
compressible fluid volumetric flow rates in cubic metres per hour, identified by the symbol Q_S, refer to either

a) *Standard* conditions, which is an absolute pressure of 1 013,25 mbar and a temperature of 288,6 K, or

b) *Normal* conditions, which is an absolute pressure of 1 013,25 mbar and a temperature of 273 K.

Numerical constants for the flow equations are provided for both conventions (see Table 1).

4 Symbols

Symbol	Description	Unit
C	Flow coefficient (K_v, C_v)	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)	Dimensionless (see Note 4)
F_F	Liquid critical pressure ratio factor	Dimensionless
F_L	Liquid pressure recovery factor of a control valve without attached fittings	Dimensionless (see Note 4)
F_{LP}	Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings	Dimensionless
F_P	Piping geometry factor	Dimensionless
F_R	Reynolds number factor	Dimensionless
F_γ	Specific heat ratio factor	Dimensionless
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)	Dimensionless
p_v	Absolute vapour pressure of the liquid at inlet temperature	kPa or bar
Δp_{actual}	Differential pressure between upstream and downstream pressure taps ($P_1 - P_2$)	kPa or bar
Δp_{choked}	Computed value of limiting pressure differential for incompressible flow	kPa or bar
Δp_{sizing}	Value of pressure differential used in computing flow or required flow coefficient for incompressible flows	kPa or bar
Q	Actual volumetric flow rate	m^3/h
Q_S	Standard volumetric flow rate (see definition 3.2)	m^3/h
Re_v	Valve Reynolds number	Dimensionless
T_1	Inlet absolute temperature	K
T_c	Absolute thermodynamic critical temperature	K
T_r	Reduced temperature (T_1/T_c)	Dimensionless
t_s	Absolute reference temperature for standard cubic metre	K
W	Mass flow rate	kg/h
x	Ratio of actual pressure differential to inlet absolute pressure ($\Delta P/P_1$)	Dimensionless
x_{choked}	Choked pressure drop ratio for compressible flow	Dimensionless
x_{sizing}	Value of pressure drop ratio used in computing flow or required flow coefficient for compressible flows	Dimensionless

Symbol	Description	Unit
x_T	Pressure differential ratio factor of a control valve without attached fittings at choked flow	Dimensionless (see Note 4)
x_{TP}	Pressure differential ratio factor of a control valve with attached fittings at choked flow	Dimensionless
Y	Expansion factor	Dimensionless
Z_1	Compressibility factor at inlet conditions	Dimensionless
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)	Dimensionless
γ	Specific heat ratio	Dimensionless
ζ	Velocity head loss coefficient of a reducer, expander or other fitting attached to a control valve or valve trim	Dimensionless
ζ_1	Upstream velocity head loss coefficient of fitting	Dimensionless
ζ_2	Downstream velocity head loss coefficient of fitting	Dimensionless
ζ_{B1}	Inlet Bernoulli coefficient	Dimensionless
ζ_{B2}	Outlet Bernoulli coefficient	Dimensionless

NOTE 1 To determine the units for the numerical constants, dimensional analysis may be performed on the appropriate equations using the units given in Table 1.

NOTE 2 1 bar = 10^2 kPa = 10^5 Pa

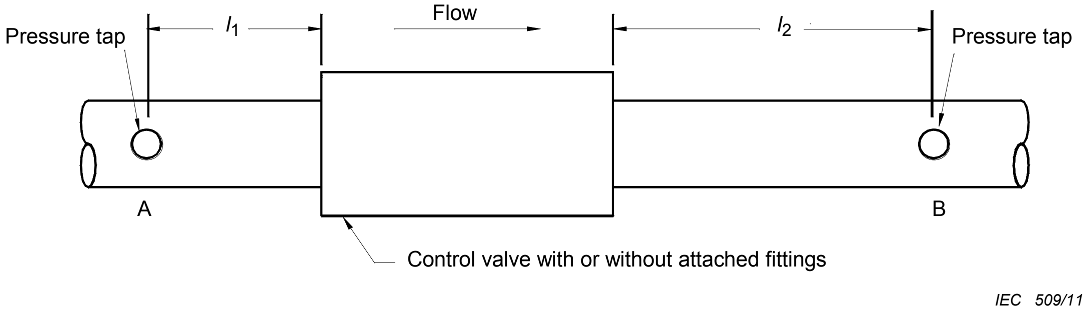
NOTE 3 1 centistoke = 10^{-6} m^2/s

NOTE 4 These values are travel-related and should be stated by the manufacturer.

5 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

6 Sizing equations for incompressible fluids

6.1 Turbulent flow

The fundamental flow model for incompressible fluids in the turbulent flow regime is given as:

$$Q = CN_1F_P\sqrt{\frac{\Delta p_{sizing}}{\rho_1/\rho_o}} \qquad (1)$$

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 The piping geometry factor, F_P, reduces to unity when the valve size and adjoining pipe sizes are identical. Refer to 8.1 for evaluation and additional information.

This model establishes the relationship between flow rate, flow coefficient, fluid properties, related installation factors, and pertinent service conditions for control valves handling incompressible fluids. Equation (1) may be used to compute the required flow coefficient, the flow rate or applied pressure differential given any two of the three quantities.

This model rigorously applies only to single component, single phase fluids (i.e., no multi-phase mixtures, no multi-component mixtures). However, this model may be used with caution under certain conditions for multi-component mixtures in the liquid phase. The underlying assumptions of the flow model would be satisfied for liquid-liquid fluid mixtures subject to the following restrictions:

- the mixture is homogenous;
- the mixture is in chemical and thermodynamic equilibrium;
- the entire throttling process occurs well away from the multiphase region.

When these conditions are satisfied, the mixture density should be substituted for the fluid density ρ_1 in Equation (1).

6.2 Pressure differentials

6.2.1 Sizing pressure differential, Δp_{sizing}

The value of the pressure differential used in Equation (1) to predict flow rate or compute a required flow coefficient is the lesser of the actual pressure differential or the choked pressure differential:

$$\Delta p_{sizing} = \begin{cases} \Delta p & if\ \Delta p < \Delta p_{choked} \\ \Delta p_{choked} & if\ \Delta p \geq \Delta p_{choked} \end{cases} \quad (2)$$

6.2.2 Choked pressure differential, Δp_{choked}

The condition where further increase in pressure differential at constant upstream pressure no longer produces a corresponding increase in flow through the control valve is designated "choked flow". The pressure drop at which this occurs is known as the choked pressure differential and is given by the following equation:

$$\Delta p_{choked} = \left(\frac{F_{LP}}{F_P}\right)^2 (p_1 - F_F p_v) \quad (3)$$

NOTE The expression $\left(\frac{F_{LP}}{F_P}\right)^2$ reduces to $F_L{}^2$ when the valve size and adjoining pipe sizes are identical. Refer to 8.1 for more information.

6.2.3 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 vapor pressures near zero, this factor is 0,96.

Values of F_F may be supplied by the user if known. For single component fluids it may be determined from the curve in Figure D.3 or approximated from the following equation:

$$F_F = 0{,}96 - 0{,}28\sqrt{\frac{p_v}{p_c}} \quad (4)$$

Use of Equation (4) to describe the onset of choking of multi-component mixtures is subject to the applicability of appropriate corresponding states parameters in the flashing model.

6.3 Non-turbulent (laminar and transitional) flow

The flow model embodied in Equation (1) is for fully developed, turbulent flow only. Non-turbulent conditions may be encountered, especially when flow rates are quite low or fluid viscosity is appreciable. To affirm the applicability of Equation (1), the value of the valve Reynolds Number (see Equation (23)) should be computed. Equation (1) is applicable if $Re_V \geq 10\,000$.

7 Sizing equations for compressible fluids

7.1 General

The fundamental flow model for compressible fluids in the turbulent flow regime is given as:

$$W = CN_6 F_P Y \sqrt{x_{sizing}\, p_1 \rho_1} \quad (5)$$

This model establishes the relationship between flow rates, flow coefficients, fluid properties, related installation factors and pertinent service conditions for control valves handling compressible fluids.

Two equivalent forms of Equation (5) are presented to accommodate conventional available data formats:

$$W = CN_8 F_P p_1 Y \sqrt{\frac{x_{sizing} M}{T_1 Z_1}} \quad (6)$$

$$Q_s = CN_9 F_P p_1 Y \sqrt{\frac{x_{sizing}}{M T_1 Z_1}} \quad (7)$$

NOTE See Annex D for values of *M*.

Equation (6) is derived by substituting the fluid density as computed from the ideal gas equation-of-state into Equation (5). Equation (7) expresses the flow rate in standard volumetric units. Equations (5) through (7) may be used to compute the required flow coefficient, the flow rate or applied pressure differential given any two of the three quantities.

7.2 Pressure differentials

7.2.1 Sizing pressure drop ratio, x_{sizing}

The value of the pressure drop ratio used in Equations (5) through (7) to predict flow rate or compute a required flow coefficient is the lesser of the actual pressure drop ratio or the choked pressure drop ratio:

$$x_{sizing} = \begin{cases} x & if\ x < x_{choked} \\ x_{choked} & if\ x \geq x_{choked} \end{cases} \quad (8)$$

where

$$x = \frac{\Delta p}{p_1} \quad (9)$$

7.2.2 Choked pressure drop ratio, x_{choked}

The pressure drop ratio at which flow no longer increases with increased value in pressure drop ratio, is the choked pressure drop ratio, given by the following equation:

$$x_{choked} = F_\gamma x_{TP} \quad (10)$$

NOTE The expression x_{TP} reduces to x_T when the valve size and adjoining pipe sizes are identical. Refer to 8.1 for more information.

7.3 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:

$$F_\gamma = \frac{\gamma}{1,4} \quad (11)$$

NOTE See Annex D for values of γ and F_γ.

Equation (11) evolved from assumption of perfect gas behaviour and extension of an orifice plate model based on air and steam testing to control valves. Analysis of that model over a range of 1,08 < γ < 1,65 leads to adoption of the current linear model embodied in Equation (11). The difference between the original orifice model, other theoretical models and Equation (11) is small within this range. However, the differences become significant outside of the indicated range. For maximum accuracy, flow calculations based on this model should be restricted to a specific heat ratio within this range and to ideal gas behaviour.

7.4 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 8.4.

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 shall be calculated using Equation (12).

$$Y = 1 - \frac{x_{sizing}}{3x_{choked}} \tag{12}$$

NOTE The expansion factor, *Y*, has a limiting value of $\frac{2}{3}$ under choked flow conditions.

7.5 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. 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:

$$p_r = \frac{p_1}{p_c} \tag{13}$$

$$T_r = \frac{T_1}{T_c} \tag{14}$$