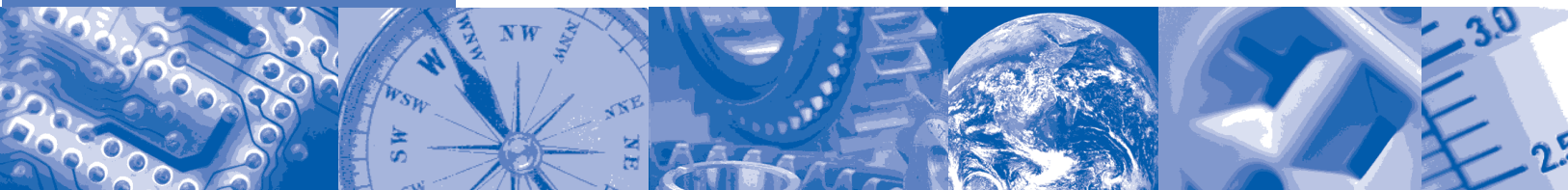


ANSI/ISA-75.01-1985 (R1995)

Formerly ANSI/ISA-S75.01-1985 (R1995)



Flow Equations for Sizing Control Valves



ISA—The Instrumentation,
Systems, and
Automation Society

Reaffirmed 18 October 1995

ANSI/ISA-75.01-1985 (R1995), Flow Equations for Sizing Control Valves

ISBN 0-87664-899-5

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1 Scope

This standard presents equations for predicting the flow of compressible and incompressible fluids through control valves. The equations are not intended for use when mixed-phase fluids, dense slurries, dry solids, or non-Newtonian liquids are encountered. In addition, the prediction of cavitation, noise, or other effects is not a part of this standard.

2 Introduction

The equations of this standard are based on the use of experimentally determined capacity factors obtained by testing control valve specimens according to the procedures of ANSI/ISA S75.02, "Control Valve Capacity Test Procedure" ([see Annex J—References](#)).

The equations are used to predict the flow rate of a fluid through a valve when all the factors, including those related to the fluid and its flowing condition, are known. When the equations are used to select a valve size, it is often necessary to use capacity factors associated with the fully open or rated condition to predict an approximate required valve flow coefficient (C_v). This procedure is further explained in [Annex A](#).

2.1 Flow variable and fluid properties

The flow rate of a fluid through a control valve is a function of the following (where applicable):

- a) Inlet and outlet conditions
 - 1) Pressure
 - 2) Temperature
 - 3) Piping geometry
- b) Liquid properties
 - 1) Composition
 - 2) Density
 - 3) Vapor pressure
 - 4) Viscosity
 - 5) Surface tension
 - 6) Thermodynamic critical pressure
- c) Gas and vapor properties
 - 1) Composition
 - 2) Density
 - 3) Ratio of specific heats
- d) Control valve properties
 - 1) Size
 - 2) Valve travel
 - 3) Flow path geometry

3 Nomenclature

Symbol	Description
C_v	Valve flow coefficient
d	Valve inlet diameter
D	Internal diameter of the pipe
F_d	Valve style modifier
F_F	Liquid critical pressure ratio factor, dimensionless
F_k	Ratio of specific heats factor, dimensionless
F_L	Liquid pressure recovery factor of a valve without attached fittings, dimensionless
F_{LP}	Product of the liquid pressure recovery factor of a valve with attached fittings (no symbol has been identified) and the piping geometry factor, dimensionless
F_P	Piping geometry factor, dimensionless
F_R	Reynolds number factor, dimensionless
F_s	Laminar, or streamline, flow factor, dimensionless
g	Local acceleration of gravity
G_f	Liquid specific gravity at upstream conditions [ratio of density of liquid at flowing temperature to density of water at 60°F (15.6°C)], dimensionless
G_g	Gas specific gravity (ratio of density of flowing gas to density of air with both at standard conditions, which is equal to the ratio of the molecular weight of gas to the molecular weight of air), dimensionless
k	Ratio of specific heats, dimensionless
K	Head loss coefficient of a device, dimensionless
K_B	Bernoulli coefficient, dimensionless
K_i	Velocity head factors for an inlet fitting, dimensionless
M	Molecular weight, atomic mass units
$N_1, N_2, \text{ etc.}$	Numerical constants for units of measurement used
p_1	Upstream absolute static pressure, measured two nominal pipe diameters upstream of valve-fitting assembly
p_2	Downstream absolute static pressure, measured six nominal pipe diameters downstream of valve-fitting assembly
Δp	Pressure differential, $p_1 - p_2$
p_c	Absolute thermodynamic critical pressure
p_r	Reduced pressure, dimensionless
p_v	Absolute vapor pressure of liquid at inlet temperature
p_{vc}	Apparent absolute pressure at vena contracta
q	Volumetric flow rate
q_{max}	Maximum flow rate (choked flow conditions) at a given upstream condition

Symbol	Description
Re_v	Valve Reynolds number, dimensionless
T_r	Reduced temperature, dimensionless
T_c	Absolute thermodynamic critical temperature
T_1	Absolute upstream temperature (in degrees K or R)
U_1	Velocity at valve inlet
w	Weight or mass flow rate
x	Ratio of pressure drop to absolute inlet pressure ($\Delta p/p_1$), dimensionless
x_T	Pressure drop ratio factor, dimensionless
x_{TP}	Value of x_T for valve-fitting assembly, dimensionless
Y	Expansion factor, ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number, dimensionless
Z	Compressibility factor, dimensionless
γ_1 (gamma)	Specific weight, upstream conditions
μ (mu)	Viscosity, absolute
ν (nu)	Kinematic viscosity, centistokes
ρ (rho)	Density
Subscripts	
1	Upstream conditions
2	Downstream conditions
s	Nonturbulent
t	Turbulent

4 Incompressible fluid — flow of nonvaporizing liquid

The flow rate of a liquid through a given control valve at a given travel is a function of the differential pressure ($p_1 - p_2$) when the liquid does not partially vaporize between the inlet and outlet of the valve. If vapor bubbles form either temporarily (cavitation) or permanently (flashing), this relationship may no longer hold. (Refer to [Section 5](#) for choked flow equations that apply when extensive vaporization occurs.) In the transitional region between nonvaporizing liquid flow and fully choked flow, the actual flow rate is less than that predicted by either the equations in this section or those in [Section 5](#). Cavitation that occurs in this transitional region can produce physical damage to the valve and/or to the downstream piping and equipment.

4.1 Equations for turbulent flow

The equations for determining the flow rate of a liquid through a valve under turbulent, nonvaporizing flow conditions are:

$$\begin{aligned}
 & q = N_1 F_p C_v \sqrt{\frac{p_1 - p_2}{G_f}} \\
 \text{or} & \\
 & C_v = \frac{q}{N_1 F_p \sqrt{p_1 - p_2}} \\
 & w = N_6 F_p C_v \sqrt{(p_1 - p_2) \gamma_1} \\
 \text{or} & \\
 & C_v = \frac{w}{N_6 F_p \sqrt{(p_1 - p_2) \gamma_1}}
 \end{aligned}
 \quad \left. \begin{array}{l} \\ \\ \\ \end{array} \right\} \begin{array}{l} \text{(Eq. 1)} \\ \\ \text{(Eq. 2)} \end{array}$$

4.2 Numerical constants N

The numerical constants N are chosen to suit the measurement units used in the equations. Values for N are listed in [Table 1](#).

Table 1 — Numerical constants for liquid flow equations

Constant		Units Used in Equations					
N		w	q	$p, \Delta p$	d, D	γ_1	v
N_1	0.0865	—	m ³ /h	kPa	—	—	—
	0.865	—	m ³ /h	bar	—	—	—
	1.00	—	gpm	psia	—	—	—
N_2	0.00214	—	—	—	mm	—	—
	890	—	—	—	in	—	—
N_4	76 000	—	m ³ /h	—	mm	—	centistokes*
	17 300	—	gpm	—	in	—	centistokes*
N_6	2.73	kg/h	—	kPa	—	kg/m ³	—
	27.3	kg/h	—	bar	—	kg/m ³	—
	63.3	lb/h	—	psia	—	lb/ft ³	—

*To convert m²/s to centistokes, multiply m²/s by 10⁶. To convert centipoises to centistokes, divide centipoises by G_f .

4.3 Piping geometry factor F_p

The piping geometry factor F_p accounts for fittings attached to either the valve inlet or the outlet that disturb the flow to the extent that valve capacity is affected. F_p is actually the ratio of the flow coefficient of a valve with attached fittings to the flow coefficient (C_v) of a valve installed in a straight pipe of the same size as the valve.

For maximum accuracy, F_p must be determined by the test procedures specified in ANSI/ISA S75.02 (see Annex J—References). Where estimated values are permissible (see Baumann reference, *Effect of Pipe Reducers on Control Valve Capacity*), F_p may be determined by using the following equation:

$$F_p = \left(\frac{\Sigma K C_v^2}{N_2 d^4} + 1 \right)^{-1/2} \quad (\text{Eq. 3})$$

(See Annex B for the mathematical derivation of F_p .)

In many instances, the nominal sizes for valve and pipe (d and D) may be used in Equations 3, 5, 6, and 7 without significant error.

The factor ΣK is the algebraic sum of the effective velocity head coefficients of all fittings attached to but not including the valve. For instance,

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2} \quad (\text{Eq. 4})$$

where K_1 and K_2 are the resistance coefficients of the inlet and outlet fittings, respectively, and K_{B1} and K_{B2} are the Bernoulli coefficients for the inlet and outlet fittings, respectively. The Bernoulli coefficients compensate for the changes in pressure resulting from differences in stream area and velocity.

When the diameters of the inlet and outlet fittings are identical, $K_{B1} = K_{B2}$, both factors drop out of the equation. When the diameters of the inlet and outlet fittings are different, K_B is calculated as follows:

$$K_B = 1 - \left(\frac{d}{D} \right)^4 \quad (\text{Eq. 5})$$

The fittings most commonly encountered are standard, short-pattern concentric pipe reducers. These fittings have little taper, and their pressure loss will not exceed that of an abrupt contraction with a slightly rounded entrance. On that basis, if experimental values for the resistance coefficients K_1 and K_2 are unavailable, estimated values may be computed as follows:

Inlet reducer only:

$$K_1 = 0.5 \left(1 - \frac{d^2}{D_1^2} \right)^2 \quad (\text{Eq. 6})$$

Outlet increaser only:

$$K_2 = 1.0 \left(1 - \frac{d^2}{D_2^2} \right)^2 \quad (\text{Eq. 7})$$

When the reducer and increaser are the same size:

$$K_1 + K_2 = 1.5 \left(1 - \frac{d^2}{D^2} \right)^2 \quad (\text{Eq. 8})$$

(See Annex C for a graphic representation of system head changes around a valve with attached reducers.)

4.4 Equations for nonturbulent flow

Nonturbulent flow occurs at high fluid viscosities and/or low velocities. In these circumstances, the flow rate through a valve is less than for turbulent flow, and the Reynolds number factor F_R must be introduced. F_R is the ratio of nonturbulent flow rate to the turbulent flow rate predicted by Equations 1 or 2. The corresponding nonturbulent equations then become, respectively:

$$\begin{aligned} \text{or} \quad q &= N_1 F_R C_v \sqrt{\frac{p_1 - p_2}{G_f}} \\ C_v &= \frac{q}{N_1 F_R \sqrt{\frac{G_f}{p_1 - p_2}}} \end{aligned} \quad \left. \vphantom{\begin{aligned} q &= N_1 F_R C_v \sqrt{\frac{p_1 - p_2}{G_f}} \\ C_v &= \frac{q}{N_1 F_R \sqrt{\frac{G_f}{p_1 - p_2}}} \end{aligned}} \right\} \quad (\text{Eq. 9})$$

$$\begin{aligned} \text{or} \quad w &= N_6 F_R C_v \sqrt{(p_1 - p_2) \gamma_1} \\ C_v &= \frac{w}{N_6 F_R \sqrt{(p_1 - p_2) \gamma_1}} \end{aligned} \quad \left. \vphantom{\begin{aligned} w &= N_6 F_R C_v \sqrt{(p_1 - p_2) \gamma_1} \\ C_v &= \frac{w}{N_6 F_R \sqrt{(p_1 - p_2) \gamma_1}} \end{aligned}} \right\} \quad (\text{Eq. 10})$$

Note the absence of the piping geometry factor in the above equations. For nonturbulent flow, the effect of close-coupled reducers or other flow-disturbing fittings is unknown. Thus, Equation 3 applies to turbulent flow only.

Tests (see Stiles reference, *Liquid Viscosity Effects on Control Valve Sizing*, and McCutcheon reference, *A Reynolds Number for Control Valves*) show that F_R can be found by using the valve Reynolds number and Figure 1. The shading around the central curve indicates the scatter of test data and the range of uncertainty of flow rate prediction in the nonturbulent regimes.

The valve Reynolds number is defined as:

$$\text{Re}_v = \frac{N_4 F_d q}{v F_L^{1/2} C_v^{1/2}} \left(\frac{F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{1/4} \quad (\text{Eq. 11})$$

The valve style modifier F_d in Equation 11 correlates data from tests of several valve styles with different hydraulic radii, so that a single curve represents all the styles tested. (See Annex D for representative values of F_d .) Caution must be used in applying the curve in Figure 1 to valve styles for which F_d has not been established.

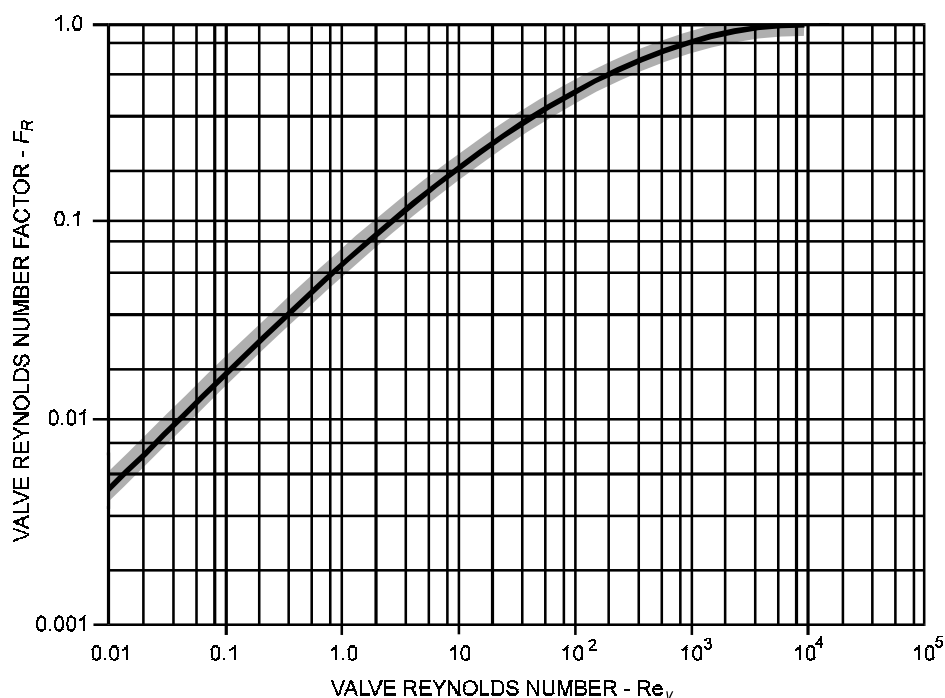


Figure 1 — Reynolds number factor

The bracketed term in Equation 11 accounts for the *velocity of approach**. Except for wide-open ball or butterfly valves, this term has only a slight effect on the Re_v calculation and can generally be omitted.

Most flow streams in process plant control valves are turbulent, with valve Reynolds numbers in excess of 10^4 , where the Reynolds number factor is 1.0. When the flow regime is questionable, Equation 11 should be used to find Re_v . For additional information on nonturbulent flow, [see Annexes E and F](#).

5 Incompressible fluid — choked flow of vaporizing liquid

Choked flow is a limiting, or maximum, flow rate. With fixed inlet (upstream) conditions, it is manifested by the failure of decreasing downstream pressure to increase the flow rate. With liquid streams, choking occurs as a result of vaporization of the liquid when the pressure within the valve falls below the vapor pressure of the liquid. Choked flow will be accompanied by either cavitation or flashing. If the downstream pressure is greater than the vapor pressure of the liquid, cavitation occurs. If the downstream pressure is equal to or less than the vapor pressure of the liquid, flashing occurs. This relationship between flow rate and pressure drop for a typical valve is shown in Figure 2.

*The flow rate through a valve is a function of the velocity of the jet stream at the vena contracta and the area of the jet at that location. This velocity is a function of the pressure drop across the valve orifice and also the valve inlet velocity, or *velocity of approach*. The velocity of approach factor is included in the valve flow coefficient C_v .

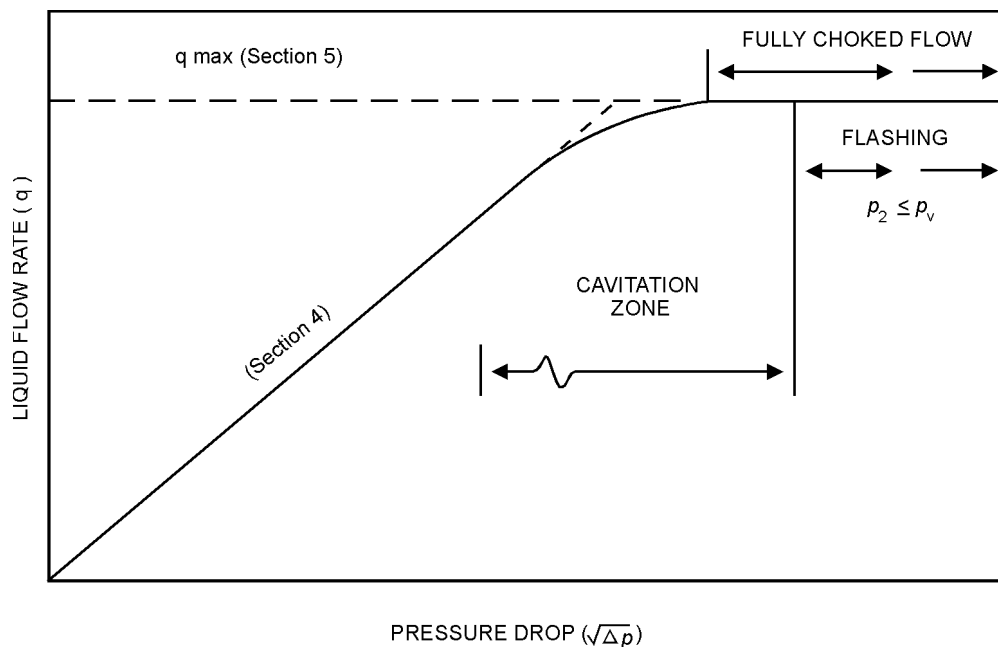


Figure 2 — Liquid flow rate versus pressure drop for a typical valve (constant upstream pressure and vapor pressure)

5.1 Liquid choked flow equations

The equations for determining the maximum flow rate of a liquid under choked conditions for valves **in straight pipes of the same size** are as follows:

$$\left. \begin{aligned} \text{or} \quad q_{\max} &= N_1 F_L C_v \sqrt{\frac{p_1 - p_{vc}}{G_f}} \\ C_v &= \frac{q_{\max}}{N_1 F_L} \sqrt{\frac{G_f}{p_1 - p_{vc}}} \end{aligned} \right\} \text{(Eq. 12a)}$$

where

$$p_{vc} = F_F p_v \quad (\text{see Annex G for } F_F) \quad \text{(Eq. 13a)}$$

giving

$$\left. \begin{aligned} \text{or} \quad q_{\max} &= N_1 F_L C_v \sqrt{\frac{p_1 - F_F p_v}{G_f}} \\ C_v &= \frac{q_{\max}}{N_1 F_L} \sqrt{\frac{G_f}{p_1 - F_F p_v}} \end{aligned} \right\} \text{(Eq. 14a)}$$

The equations for determining the maximum flow rate of a liquid under choked conditions for valves with **attached fittings** are:

$$\left. \begin{aligned} & q_{\max} = N_1 F_{LP} C_v \sqrt{\frac{p_1 - p_{vc}}{G_f}} \\ \text{or} \\ & C_v = \frac{q_{\max}}{N_1 F_{LP}} \sqrt{\frac{G_f}{p_1 - p_{vc}}} \end{aligned} \right\} \text{(Eq. 12b)}$$

where

$$p_{vc} = F_F p_v \quad (\text{see Annex G for } F_F) \quad \text{(Eq. 13b)}$$

giving

$$\left. \begin{aligned} & q_{\max} = N_1 F_{LP} C_v \sqrt{\frac{p_1 - F_F p_v}{G_f}} \\ \text{or} \\ & C_v = \frac{q_{\max}}{N_1 F_{LP}} \sqrt{\frac{G_f}{p_1 - F_F p_v}} \end{aligned} \right\} \text{(Eq. 14b)}$$

5.2 Liquid pressure recovery factor F_L

The liquid pressure recovery factor F_L applies to valves without attached fittings (see Baumann reference, *The Introduction of a Critical Flow Factor for Valve Sizing*). **This factor accounts for the influence of the internal geometry of the valve on its capacity at choked flow. Under nonvaporizing flow conditions, it is defined by the equation:**

$$F_L = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}} \quad \text{(Eq. 15a)}$$

Representative F_L values for various valve styles are listed in [Annex D](#).

5.3 Combined liquid pressure recovery factor F_{LP}

When a valve is installed with reducers or other attached fittings, the liquid pressure recovery of the valve-fitting combination is not the same as that for the valve alone. For calculations involving choked flow, it is convenient to treat the piping geometry factor (F_p) and the F_L factor for the valve-fitting combination as a single factor, F_{LP} . The value of F_L for the combination is then F_{LP}/F_p , where

$$\frac{F_{LP}}{F_p} = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}} \quad (\text{Eq. 15b})$$

(Refer to Section 4.3 and Annex B.)

For maximum accuracy, F_{LP} must be determined by using the test procedures specified in ANSI/ISA S75.02 (see Annex J—References). When estimated values are permissible, reasonable accuracy may be obtained by using the following equation to determine F_{LP} :

$$F_{LP} = F_L \left(\frac{K_i F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{-1/2} \quad (\text{Eq. 16})$$

In this equation, K_i is the head loss coefficient ($K_i + K_{bl}$) of any fitting between the upstream pressure tap and the inlet face of the valve only. (See Annex B for the mathematical derivation of F_{LP} .)

6 Compressible fluid — flow of gas and vapor

The flow rate of a compressible fluid varies as a function of the ratio of the pressure differential to the absolute inlet pressure ($\Delta p/p_1$), designated by the symbol x . At values of x near zero, the equations in this section can be traced to the basic Bernoulli equation for Newtonian incompressible fluids. However, increasing values of x result in expansion and compressibility effects that require the use of appropriate correction factors (see Buresh and Schuder reference, *The Development of a Universal Gas Sizing Equation for Control Valves*, and Driskell reference, *New Approach to Control Valve Sizing*).

6.1 Equations for turbulent flow

The flow rate of a gas or vapor through a valve may be calculated by using any of the following equivalent forms of the equation:

$$\begin{aligned} & \text{or} \quad w = N_6 F_p C_v Y \sqrt{x p_1 \gamma_1} \\ & \quad C_v = \frac{w}{N_6 F_p Y \sqrt{x p_1 \gamma_1}} \end{aligned} \quad \left. \vphantom{\begin{aligned} & \text{or} \quad w = N_6 F_p C_v Y \sqrt{x p_1 \gamma_1} \\ & \quad C_v = \frac{w}{N_6 F_p Y \sqrt{x p_1 \gamma_1}} \end{aligned}} \right\} \text{(Eq. 17)}$$

$$\begin{aligned} & \text{or} \quad q = N_7 F_p C_v p_1 Y \sqrt{\frac{x}{G_g T_1 Z}} \\ & \quad C_v = \frac{q}{N_7 F_p p_1 Y \sqrt{\frac{G_g T_1 Z}{x}}} \end{aligned} \quad \left. \vphantom{\begin{aligned} & \text{or} \quad q = N_7 F_p C_v p_1 Y \sqrt{\frac{x}{G_g T_1 Z}} \\ & \quad C_v = \frac{q}{N_7 F_p p_1 Y \sqrt{\frac{G_g T_1 Z}{x}}} \end{aligned}} \right\} \text{(Eq. 18)}$$

$$\begin{aligned} & \text{or} \quad w = N_8 F_p C_v p_1 Y \sqrt{\frac{x M}{T_1 Z}} \\ & \quad C_v = \frac{w}{N_8 F_p p_1 Y \sqrt{\frac{T_1 Z}{x M}}} \end{aligned} \quad \left. \vphantom{\begin{aligned} & \text{or} \quad w = N_8 F_p C_v p_1 Y \sqrt{\frac{x M}{T_1 Z}} \\ & \quad C_v = \frac{w}{N_8 F_p p_1 Y \sqrt{\frac{T_1 Z}{x M}}} \end{aligned}} \right\} \text{(Eq. 19)}$$

$$\begin{aligned} & \text{or} \quad q = N_9 F_p C_v p_1 Y \sqrt{\frac{x}{M T_1 Z}} \\ & \quad C_v = \frac{q}{N_9 F_p p_1 Y \sqrt{\frac{M T_1 Z}{x}}} \end{aligned} \quad \left. \vphantom{\begin{aligned} & \text{or} \quad q = N_9 F_p C_v p_1 Y \sqrt{\frac{x}{M T_1 Z}} \\ & \quad C_v = \frac{q}{N_9 F_p p_1 Y \sqrt{\frac{M T_1 Z}{x}}} \end{aligned}} \right\} \text{(Eq. 20)}$$

Note that the numerical value of x used in these equations must not exceed the choking limit ($F_K x_{Tp}$), regardless of the actual value of x . (See Section 6.4.)

6.2 Numerical constants N

The numerical constants N are chosen to suit the measurement units used in the equations. Values for N are listed in Table 2.

Table 2 — Numerical constants for gas and vapor flow equations

Constant		Units Used in Equations					
N		w	q^*	$p, \Delta p$	γ_1	T_1	d, D
N_5	0.00241	—	—	—	—	—	mm
	1000	—	—	—	—	—	in
N_6	2.73	kg/h	—	kPa	kg/m ³	—	—
	27.3	kg/h	—	bar	kg/m ³	—	—
	63.3	lb/h	—	psia	lb/ft ³	—	—
N_7	4.17	—	m ³ /h	kPa	—	K	—
	417	—	m ³ /h	bar	—	K	—
	1360	—	scfh	psia	—	°R	—
N_8	0.948	kg/h	—	kPa	—	K	—
	94.8	kg/h	—	bar	—	K	—
	19.3	lb/h	—	psia	—	°R	—
N_9	22.5	—	m ³ /h	kPa	—	K	—
	2250	—	m ³ /h	bar	—	K	—
	7320	—	scfh	psia	—	°R	—

* q is in cubic feet per hour measured at 14.73 psia and 60°F, or cubic meters per hours measured at 101.3 kPa and 15.6°C.

6.3 Expansion factor Y

The expansion factor Y accounts for the change in density of a fluid as it passes from the valve inlet to the vena contracta and for the change in area of the vena contracta as the pressure drop is varied (contraction coefficient). Theoretically, Y is affected by all of the following:

- 1) Ratio of port area to body inlet area
- 2) Internal geometry of the valve
- 3) Pressure drop ratio, x
- 4) Reynolds number
- 5) Ratio of specific heats, k

The influence of items 1, 2, and 3 are defined by the factor x_T . Test data (see Driskell reference, *New Approach to Control Valve Sizing*) indicate that Y may be taken as a linear function of x , as shown in the following equation for a valve without attached fittings:

$$Y = 1 - \frac{x}{3F_k x_T} \quad (\text{Limits } 1.0 \geq Y \geq 0.67) \quad (\text{Eq. 21})$$

For a valve with attached fittings, x_{TP} shall be substituted for x_T .

For all practical purposes, Reynolds number effects may be disregarded in the case of compressible fluids. The effect of the ratio of specific heats is considered in [Section 6.7](#).

6.4 Choked flow

If all inlet conditions are held constant and the differential pressure ratio (x) is increased by lowering the downstream pressure (p_2), the mass flow rate will increase to a maximum limit. Flow conditions where the value of x exceeds this limit are known as *choked flow*. **Choking occurs when the jet stream at the vena contracta attains its maximum cross-sectional area at sonic velocity.** This occurs at pressure ratios (p_1/p_{vc}) greater than about 2.0.

The value of x at the inception of choked flow conditions varies from valve to valve. It also varies with the piping geometry and with the thermodynamic properties of the flowing fluid. The factors involved are x_T ([Section 6.5](#)), x_{TP} ([Section 6.6](#)), and F_k ([Section 6.7](#)).

Choking affects the use of Equations 17 through 21 in the following manner: **The value of x used in the equations must not exceed $F_k x_T$ or $F_k x_{TP}$,** regardless of the actual value of x . The expansion factor Y at choked flow ($x \geq F_k x_{TP}$) is then at its minimum value of 2/3.

6.5 Pressure drop ratio factor x_T

For maximum accuracy, the pressure drop ratio factor x_T must be established by using the test procedures specified in ANSI/ISA S75.02 ([see Annex J—References](#)). Representative x_T values for valves are tabulated in Annex D. These representative values are not to be taken as actual. Actual values must be obtained from the valve manufacturer.

6.6 Pressure drop ratio factor with reducers or other fittings x_{TP}

When a valve is installed with reducers or other fittings, the pressure drop ratio factor of the assembly (x_{TP}) is different from that of the valve alone (x_T). For maximum accuracy, x_{TP} must be determined by test (see reference ANSI/ISA S75.02). When estimated values are permissible, the following equation may be used to determine x_{TP} :

$$x_{TP} = \frac{x_T}{F_p^2} \left(\frac{x_T K_i C_v^2}{N_5 d^4} + 1 \right)^{-1} \quad (\text{Eq. 22})$$

In this equation, x_T is the pressure drop ratio factor for a given valve installed without reducers or other fittings, and K_i is the sum of the inlet velocity head coefficients ($K_i + K_{Bi}$) of the reducer or other fitting attached to the valve inlet. This correction to x_T is usually negligible if d/D is greater than 0.5 and C_v/d^2 is less than 20, where d is in inches.

[See Annex H](#) for the mathematical derivation of x_T .

6.7 Ratio of specific heats factor F_k

The ratio of specific heats of a compressible fluid affects the flow rate through a valve. The factor F_k accounts for this effect. F_k has a value of 1.0 for air at moderate temperatures and pressures, where its specific heat ratio is about 1.40. Both theoretical and experimental evidence indicate that for valve sizing purposes, F_k may be taken as having a linear relationship to k . Therefore:

$$F_k = \frac{k}{1.40} \quad (\text{Eq. 23})$$

6.8 Compressibility factor Z

Equations 18, 19, and 20 do not contain a term for the actual specific weight of the fluid at upstream conditions. Instead, this term 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 reduced pressure and reduced temperature. For use in this section, 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 is defined similarly. That is:

$$p_r = \frac{p_1}{p_c} \quad (\text{Eq. 24})$$

$$T_r = \frac{T_1}{T_c} \quad (\text{Eq. 25})$$

Absolute thermodynamic critical pressures and temperatures for most fluids, and curves from which Z may be determined, can be found in many reference handbooks of physical data.

Annex A — Use of flow rate equations for sizing valves

Laboratory tests are conducted on actual valves in a prescribed test setup (see reference ANSI/ISA S75.02). The test fluid is usually water or air. The flow coefficient C_v and the factors F_L , x_T , etc. are determined at the rated valve travel. These data, along with factors to account for the actual fluid and the pipe configuration (F_k , F_F , F_p , etc.), are used in the equations of this standard to predict the flow rate with the valve fully open.

The principal use of the flow equations is to aid in the selection of an appropriate valve size for a specific application. In this procedure, the numbers in the equations consist of known values for the fluid and flow conditions and known values for the selected valve type at its rated opening. With these factors in the equation, the unknown (or product of unknowns, e.g., $F_p C_v$) can be computed. Although these computed numbers are often suitable for selecting a valve from a series of discrete sizes, they do not represent a true operating condition, because the factors are mutually incompatible. Some of the factors used in the equation are for the wide-open valve while others relating to the operating conditions are for the partially open valve.

Once a valve size has been selected, the remaining unknowns, such as F_p , can be computed and a judgment can be made as to whether the valve size is adequate. It is not usually necessary to carry the calculations further to predict the exact valve opening. To do this, all the pertinent sizing factors must be known at fractional valve openings.

Additional information on the use of the flow equations, along with example problems, is available in *ISA Handbook of Control Valves* and Driskell reference, *Control Valve Selection and Sizing*.

Annex B — Derivation of factors F_p and F_{LP}

If a valve is installed between reducers, the C_v of the entire assembly is different from that of the valve alone. If the inlet and outlet reducers are the same size, the only effect is the added resistance of the fittings, which creates an additional pressure drop. If there is only one reducer or if there are reducers of different sizes, there will be an additional effect on the pressure due to the difference in velocity between the inlet and outlet streams.

The velocity head expressed in feet of fluid equals $U^2/2g$, where U is the velocity of the stream and g is the acceleration of gravity. Expressed in U.S. customary units, psi, gpm, and inches, the velocity pressure becomes:

$$p = \frac{q^2 G_f}{890 d^4} \quad (\text{Eq. B-1})$$

For a resistance coefficient K , the pressure difference then becomes:

$$\Delta p = K \left(\frac{q^2 G_f}{890 d^4} \right) \quad (\text{Eq. B-2})$$

From Equations 1 and B-2, the resistance coefficient for a valve is:

$$K_{\text{valve}} = \frac{890 d^4}{C_v^2} \quad (\text{Eq. B-3})$$

The change in velocity pressure across a reducer with diameters d and D is

$$\frac{q^2 G_f}{890 d^4} - \frac{q^2 G_f}{890 D^4} = \frac{q^2 G_f}{890 d^4} \left(1 - \frac{d^4}{D^4} \right) \quad (\text{Eq. B-4})$$

From Equations B-2 and B-4, we have the factor K_B , which has been called the Bernoulli coefficient.

Here,

$$K_B = \left(1 - \frac{d^4}{D^4} \right) \quad (\text{Eq. B-5})$$

By definition:

$$(F_p C_v)^2 = \frac{q^2 G_f}{\Delta p} \quad (\text{Eq. B-6})$$

From Equations B-2 and B-6, adding all K factors:

$$(F_p C_v)^2 = \frac{890 d^4}{K_{\text{valve}} + K_1 + K_2 + K_{B1} - K_{B2}} \quad (\text{Eq. B-7})$$

Substitute K_{valve} from Equation B-3:

$$(F_P C_V)^2 = \frac{890 d^4}{\frac{890 d^4}{C_V^2} + \Sigma K} \quad (\text{Eq. B-8})$$

where

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2} \quad (\text{Eq. B-9})$$

Then, rearranging Equation B-8, we have:

$$F_P = \left(\frac{\Sigma K C_V^2}{890 d^4} + 1 \right)^{-1/2} \quad (\text{Eq. B-10})$$

It should be noted in Equation B-9 that ΣK is the sum of all the effective velocity head coefficients. If the inlet and outlet reducers are the same size, $K_{B1} = K_{B2}$, and in Equation B-9 both drop out because of the difference in their sign. For K_1 and K_2 , see Equations 6 and 7.

By definition, from Equation 15:

$$F_L^2 = \frac{p_1 - p_2}{p_1 - p_{vc}} = \frac{\Delta p_a}{\Delta p_{vc}} \quad (\text{Eq. B-11})$$

where Δp_a is the pressure drop across the valve, and Δp_{vc} is the drop to the vena contracta.

Also, from Equation 1:

$$q^2 = (F_P C_V)^2 \frac{\Delta p_b}{G_f} = C_V^2 \frac{\Delta p_a}{G_f} \quad (\text{Eq. B-12})$$

where Δp_b is the drop across the valve with reducers.

From Equation B-12:

$$\Delta p_a = F_P^2 \Delta p_b \quad (\text{Eq. B-13})$$

Substituting this expression into Equation B-11, we have:

$$F_L^2 = F_P^2 \frac{\Delta p_b}{\Delta p_{vc}} \quad (\text{Eq. B-14})$$

By definition:

$$(F_L)_p^2 = \frac{\Delta p_b}{\Delta p_{vc} + \Delta p_i} \quad (\text{Eq. B-15})$$

where $(F_L)_p$ is the pressure recovery factor for the valve with reducers, and Δp_i is the drop across the inlet reducer.

From Equation B-2:

$$\Delta p_i = \frac{K_i q^2 G_f}{890 d^4} \quad (\text{Eq. B-16})$$

where $K_i = K_1 + K_{B1}$.

Substituting the expression for q^2 from Equation B-12 into Equation B-16, we have:

$$\Delta p_i = \frac{K_i F_p^2 C_v^2 \Delta p_b}{890 d^4} \quad (\text{Eq. B-17})$$

Substituting Equations B-14 and B-17 into B-15, we have the following development:

$$\begin{aligned} (F_L)_p^2 &= \frac{\Delta p_b}{\frac{F_p^2 \Delta p_b}{F_L^2} + \frac{K_i F_p^2 C_v^2 \Delta p_b}{890 d^4}} \\ (F_L)_p &= \frac{1}{F_p} \left(\frac{1}{F_L^2} + \frac{K_i C_v^2}{890 d^4} \right)^{-1/2} \\ F_{LP} = (F_L)_p F_p &= \left[\frac{1}{F_L^2} + \frac{K_i}{890} \left(\frac{C_v}{d^2} \right)^2 \right]^{-1/2} \\ F_{LP} &= F_L \left[\frac{F_L^2 K_i}{N_2} \left(\frac{C_v}{d^2} \right)^2 + 1 \right]^{-1/2} \quad (\text{Eq. B-18}) \end{aligned}$$

Annex C — Control valve-piping system head changes

An understanding of the various loss mechanisms involved in a control valve-piping system can be obtained by looking at the energy grade lines and the hydraulic grade lines for a liquid flow system containing abrupt contractions and expansions in the form of concentric reducers. These are shown schematically in [Figure C-1](#). For ease of comprehension, the curves are displayed as straight line segments. The energy grade line includes only the available energy and excludes internal energy. Each point of pressure change associated with this figure is defined below. Some of the pressure drops are nonrecoverable and some are recoverable, as shown in the hydraulic grade line. The terms below also define the various coefficients associated with the system. The Bernoulli coefficients, K_{B1} and K_{B2} , account for the change in the velocity pressure of the stream and relate the total kinetic energy to that calculated with the valve inlet velocity U_1 .

Table C-1 — Definitions of head terms (Refer to Figure C-1)

Reference Letter (See Fig. C-1)	Head Terms*	U.S. Units	SI Units
A	Inlet pressure head	p_1 / γ	$p_1 / \rho g$
B	Inlet velocity head	$(d/D_1)^4 (U_1^2 / 2g)$	$(d/D_1)^4 (U_1^2 / 2g)$
C	Reducer drop	$(K_1 + K_{B1}) (U_1^2 / 2g)$	$(K_1 + K_{B1}) (U_1^2 / 2g)$
D	Differential to vena contracta	$(E) / (1 - F_L^2)$	$(E) / (1 - F_L^2)$
E	Pressure recovery at valve	$(D) - (H)$	$(D) - (H)$
F	Increaser recovery	$(K_{B2} - K_2) (U_1^2 / 2g)$	$(K_{B2} - K_2) (U_1^2 / 2g)$
G	Reducer loss	$K_1 (U_1^2 / 2g)$	$K_1 (U_1^2 / 2g)$
H	Valve loss	$N_2 (d^4 / C_v^2) (U_1^2 / 2g)$	$N_2 (d^4 / C_v^2) (U_1^2 / 2g)$
I	Increaser loss	$K_2 (U_1^2 / 2g)$	$K_2 (U_1^2 / 2g)$
J	Outlet pressure head	p_2 / γ	$p_2 / \rho g$
K	Outlet velocity head	$(d/D_2)^4 (U_1^2 / 2g)$	$(d/D_2)^4 (U_1^2 / 2g)$
L	Total head loss	$(p_1 - p_2) / \gamma$	$(p_1 - p_2) / \rho g$

* All units are absolute and consistent: pound, foot, and second in U.S. customary units; SI for metric units.



Annex D — Representative values of valve capacity factors

The values in Table D-1 are typical only for the types of valves shown at their rated travel for full-size trim. Significant variations in value may occur because of any of the following reasons: reduced travel, trim type, reduced port size, and valve manufacturer.

Table D-1 — Representative values of valve capacity factors

Valve Type	Trim Type	Flow Direction*	x_T	F_L	F_s	F_d^{**}	C_v/d^{2t}	
GLOBE	Single port	Ported plug	Either	0.75	0.9	1.0	1.0	9.5
		Contoured plug	Open	0.72	0.9	1.1	1.0	11
	Close		0.55	0.8	1.1	1.0	11	
	Characterized cage	Open	0.75	0.9	1.1	1.0	14	
		Close	0.70	0.85	1.1	1.0	16	
	Wing guided	Either	0.75	0.9	1.1	1.0	11	
	Double port	Ported plug	Either	0.75	0.9	0.84	0.7	12.5
		Contoured plug	Either	0.70	0.85	0.85	0.7	13
		Wing guided	Either	0.75	0.9	0.84	0.7	14
	Rotary	Eccentric spherical plug	Open	0.61	0.85	1.1	1.0	12
			Close	0.40	0.68	1.2	1.0	13.5
ANGLE	Contoured plug	Open	0.72	0.9	1.1	1.0	17	
		Close	0.65	0.8	1.1	1.0	20	
	Characterized cage	Open	0.65	0.85	1.1	1.0	12	
		Close	0.60	0.8	1.1	1.0	12	
	Venturi	Close	0.20	0.5	1.3	1.0	22	
BALL	Segmented	Open	0.25	0.6	1.2	1.0	25	
	Standard port (diameter $\cong 0.8d$)	Either	0.15	0.55	1.3	1.0	30	
BUTTERFLY	60-Degree aligned	Either	0.38	0.68	0.95	0.7	17.5	
	Fluted vane	Either	0.41	0.7	0.93	0.7	25	
	90-Degree offset seat	Either	0.35	0.60	0.98	0.7	29	

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

** In general, an F_d value of 1.0 can be used for valves with a single flow passage. An F_d value of 0.7 can be used for valves with two flow passages, such as double-ported globe valves and butterfly valves.

† In this table, d may be taken as the nominal valve size, in inches.

Annex E — Reynolds number factor F_R

The information contained in this annex is an elaboration of the discussion presented in [Section 4.4](#). It presents a method used for resolving laminar and transitional flow problems.

Figure E-1 shows the relationships between F_R and the valve Reynolds number Re_v for the three types of problems that may be encountered with viscous flow. These are:

- a) Determining the required flow coefficient when selecting a control valve size
- b) Predicting the flow rate that a selected valve will pass
- c) Predicting the pressure differential that a selected valve will exhibit

In [Figure E-1](#), the straight diagonal lines extending downward at an F_R value of approximately 0.3 indicate conditions under which laminar flow exists. At a valve Reynolds number of 40 000, all three curves in [Figure E-1](#) reach an F_R value of 1.0. At this number and at all higher Re_v values, fully turbulent flow conditions exist. Between the laminar region, indicated by the straight diagonal lines of [Figure E-1](#), and the turbulent region, where $F_R = 1.0$, the flow regime is transitional (i.e., neither laminar nor turbulent).

Equation 11 for determining the valve Reynolds number Re_v is:

$$Re_v = \frac{N_4 F_d q}{v F_L^{1/2} C_v^{1/2}} \left(\frac{F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{1/4} \quad (\text{Eq. 11})$$

F_R values and the solutions to the three classes of problems may be obtained by using the following procedures.

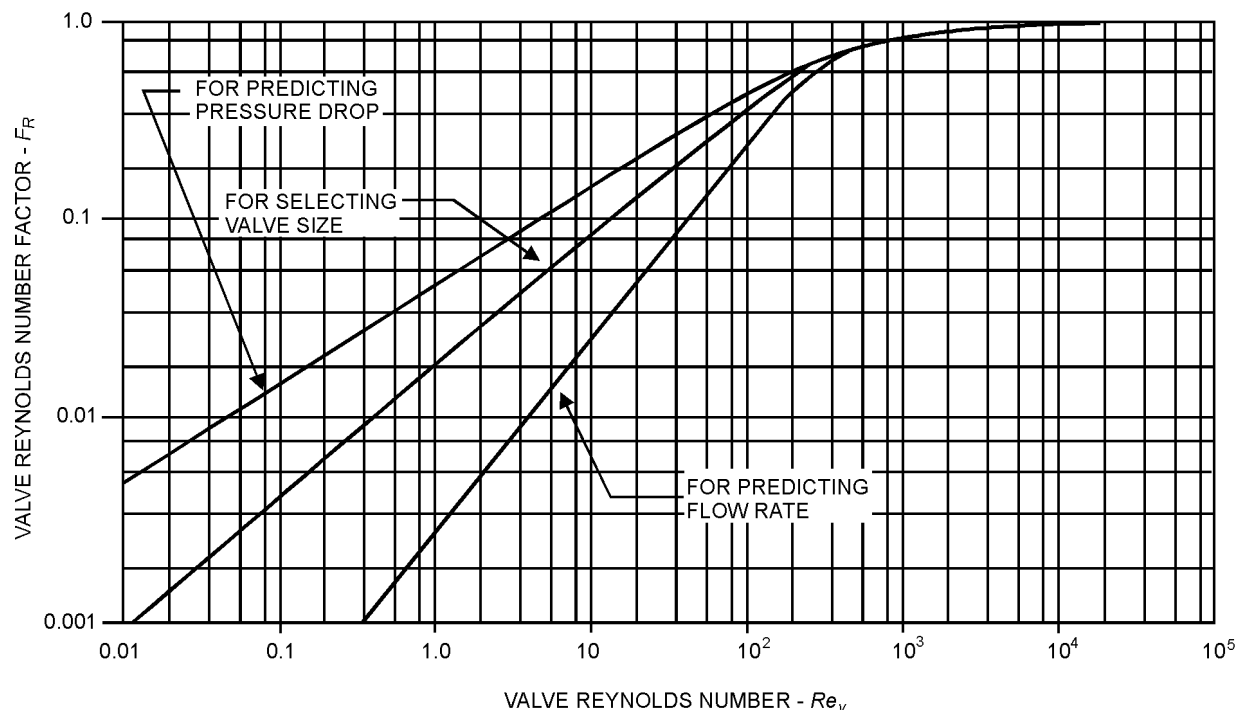


Figure E-1 — Reynolds number factor for valve sizing
(See [Figure 1](#) for the range of uncertainty.)

Determining required flow coefficient (selecting valve size)

The following treatment is based on valves without attached fittings; therefore, $F_p = 1.0$.

- a) Calculate a pseudo valve flow coefficient C_{vt} , assuming turbulent flow, using:

$$C_{vt} = \frac{q}{N_1 \sqrt{\frac{p_1 - p_2}{G_f}}} \quad (\text{Eq. E-1})$$

- b) Calculate Re_v by using Equation 11, substituting C_{vt} from Step 1 for C_v . For F_L , select a representative value for the valve style desired.

- c) Find F_R as follows:

- 1) If Re_v is less than 56, the flow is laminar, and F_R may be found by using either the curve in [Figure E-1](#) labeled "For Selecting Valve Size" or by using the equation

$$F_R = 0.019 (Re_v)^{0.67} \quad (\text{Eq. E-2})$$

- 2) If Re_v is greater than 40 000, the flow may be taken as turbulent, and $F_R = 1.0$.
- 3) If Re_v lies between 56 and 40 000, the flow is transitional, and F_R may be found by using either the curve in [Figure E-1](#) or [Table E-1](#) in the column headed "Valve Size Selection."

d) Obtain the required C_v from:

$$C_v = \frac{C_{vt}}{F_R} \quad (\text{Eq. E-3})$$

e) After determining C_v , check the F_L value for the selected valve size and style. If this value is significantly different from the value selected in Step 2, use the new value and repeat Steps 1 through 4.

Predicting flow rate

a) Calculate q_t , assuming turbulent flow, using:

$$q_t = N_1 C_v \sqrt{\frac{p_1 - p_2}{G_f}} \quad (\text{Eq. E-4})$$

b) Calculate Re_v by using Equation 11, substituting q_t for q from Step 1.

c) Find F_R as follows:

1) If Re_v is less than 106, the flow is laminar, and F_R may be found by using the curve in [Figure E-1](#) labeled "For Predicting Flow Rate" or by using the equation

$$F_R = 0.0027 Re_v \quad (\text{Eq. E-5})$$

2) If Re_v is greater than 40 000, the flow may be taken as turbulent, and $F_R = 1.0$.

3) If Re_v lies between 106 and 40 000, the flow is transitional, and F_R may be found by using the curve in [Figure E-1](#) or [Table E-1](#) in the column headed "Flow Rate Prediction."

d) Obtain the predicted flow rate from:

$$q = F_R q_t \quad (\text{Eq. E-6})$$

Predicting pressure drop

a) Calculate Re_v according to Equation 11.

b) Find F_R as follows:

1) If Re_v is less than 30, the flow is laminar, and F_R may be found by using the curve in [Figure E-1](#) labeled "For Predicting Pressure Drop" or by using the equation

$$F_R = 0.052 (Re_v)^{0.5} \quad (\text{Eq. E-7})$$

2) If Re_v is greater than 40 000, the flow may be taken as turbulent, and $F_R = 1.0$.

3) If Re_v lies between 30 and 40 000, the flow is transitional, and F_R may be found by using the curve in [Figure E-1](#) or [Table E-1](#) in the column headed "Pressure Drop Prediction."

c) Obtain the predicted pressure drop from:

$$\Delta p = G_f \left(\frac{q}{N_1 F_R C_v} \right)^2 \quad (\text{Eq. E-8})$$

Table E-1 — Reynolds number factor F_R for transitional flow

F_R^*	Valve Reynolds Number, Re_v^*		
	Valve Size Selection	Flow Rate Prediction	Pressure Drop Prediction
0.284	56	106	30
0.32	66	117	38
0.36	79	132	48
0.40	94	149	59
0.44	110	167	74
0.48	130	188	90
0.52	154	215	113
0.56	188	253	142
0.60	230	298	179
0.64	278	351	224
0.68	340	416	280
0.72	471	556	400
0.76	620	720	540
0.80	980	1100	870
0.84	1560	1690	1430
0.88	2470	2660	2300
0.92	4600	4800	4400
0.96	10 200	10 400	10 000
1.00	40 000	40 000	40 000

* Linear interpolation between listed values is satisfactory.

Annex F — Equations for nonturbulent liquid flow

The following method for handling liquid nonturbulent flow permits a direct solution for the unknown — flow rate, C_v , or Δp — without using tables or curves and without first computing a Reynolds number. It is especially useful with programmable calculators or computers. The results are in conformance with [Section 4.4](#).

Figure 1 in [Section 4.4](#) has the following features:

- a) A straight horizontal line at $F_R = 1.0$, representing the turbulent flow region. Here, the flow rate varies as the square root of differential pressure (Equation 1).
- b) A straight diagonal line, representing the laminar flow region. Here, the flow rate varies directly with the differential pressure.
- c) A curved portion, representing the transitional flow region.
- d) A shaded envelope to indicate the scatter of the test data (see Stiles reference, *Liquid Viscosity Effects on Control Valve Sizing*, and McCutcheon reference, *A Reynolds Number for Control Valves*) and the uncertainty to be expected in the nonturbulent flow region.

From Equation 9:

$$q = N_1 F_R C_v \sqrt{\frac{p_1 - p_2}{G_f}} \quad (\text{Eq. 9})$$

and Equation 11:

$$\text{Re}_v = \frac{N_4 F_d q}{\nu F_L^{1/2} C_v^{1/2} \left(\frac{F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{1/4}} \quad (\text{Eq. 11})$$

For the laminar flow region, an equation can be written for the straight line found in [Figure 1](#), such that:

$$F_R = \left(\frac{\text{Re}_v}{370} \right)^{1/2} \quad (\text{Eq. F-1})$$

Combining these three equations, we obtain:

$$\left. \begin{aligned} q &= N_s (F_s C_v)^{3/2} \frac{\Delta p}{\mu} \\ \text{or} \quad C_v &= \frac{1}{F_s} \left(\frac{q \mu}{N_s \Delta p} \right)^{2/3} \end{aligned} \right\} \quad (\text{Eq. F-2})$$

where,

$$F_s = \frac{F_d^{2/3}}{F_L^{1/3}} \left(\frac{F_L^2 C_v^2}{N_s^2 d^4} + 1 \right)^{1/6} \quad (\text{Eq. F-3})$$

and

μ = absolute viscosity, centipoises

N_s = a constant that depends on the units used, i.e.,

N_s	q	Δp
47	gpm	psi
1.5	m ³ /hr	kPa
15	m ³ /hr	bar

F_s is generally a function of a specific manufacturer's valve style and varies little from size to size. This variation is usually no greater than the uncertainty in the value of the factor F_d that accounts for the hydraulic radius. Representative values of F_s are listed in [Annex D](#). Once a particular valve has been selected, the actual values of F_d , F_L , and C_v/d^2 may be used to compute F_s .

Equation F-2 may be solved directly for the unknown if the flow is fully laminar. In the transitional region, to avoid using a curve or table, the following equations have been established for determining F_R :

$$F_R = 1.044 - 0.358 \left(\frac{C_{vs}}{C_{vt}} \right)^{0.655} \quad (\text{Eq. F-4})$$

$$F_R = 1.084 - 0.375 \left(\frac{\Delta p_s}{\Delta p_t} \right)^{0.336} \quad (\text{Eq. F-5})$$

$$F_R = 1.004 - 0.358 \left(\frac{q_t}{q_s} \right)^{0.588} \quad (\text{Eq. F-6})$$

In these equations, the subscript s denotes a value computed from Equation F-2 assuming laminar flow conditions, and the subscript t denotes a value computed from Equation 9 assuming turbulent flow conditions ($F_R = 1.0$).

When the value F_R calculated by the above equations is less than 0.48, the flow may be taken as laminar, and Equation F-2 governs. When F_R is greater than 0.98, the flow may be taken as turbulent, and Equation 9 governs (F_R is $\cong 1.0$). The piping geometry factor F_p should not be used in either Equation 9 or Equation F-2, because the effect that close-coupled fittings have on nonturbulent flow through control valves has not been established. Also, the equation used in this standard for F_p is based on turbulent flow only. For maximum accuracy, a valve must be installed with a straight inlet pipe the same size as the valve. The length of the straight pipe should be sufficient for the stream to attain its normal velocity profile, a condition upon which the research data are based.

The following examples demonstrate how problems may be solved.

PROBLEM 1. Find the valve size.

Given: $q = 500$ gpm, $G_f = 0.9$, $\Delta p = 20$ psi, $\mu = 20\,000$ cp

Selected valve: Butterfly, $C_v / d^2 = 19$, $F_s = 0.93$ (from a manufacturer's catalog or [Annex D](#))

Using Equation 9 for turbulent flow:

$$q = N_1 F_R C_{vt} \left(\frac{\Delta p}{G_f} \right)^{1/2}$$

$$500 = (1.0)(1.0) C_{vt} \left(\frac{20}{0.90} \right)^{1/2}$$

$$C_{vt} = 106$$

Using Equation F-2 for laminar flow:

$$C_{vs} = \frac{1}{F_s} \left(\frac{q\mu}{N_s \Delta p} \right)^{2/3}$$

$$C_{vs} = \frac{1}{0.93} \left(\frac{500(20\,000)}{47(20)} \right)^{2/3} = 520$$

Using Equation F-4 for transitional flow,

$$F_R = 1.044 - 0.358 \left(\frac{520}{106} \right)^{0.655} = 0.03$$

This value for F_R is less than the 0.48 limit for transitional flow, so the flow is laminar. The C_v required is 520. To meet this requirement, a representative 6-inch valve has a $C_v = 19d^2 = 684$, or as listed in the manufacturer's catalog.

PROBLEM 2. Find the differential pressure.

Given: $q = 1070$ gpm, $G_f = 0.84$, $\mu = 5900$ cp, $C_v = 400$, $F_s = 1.25$

Using Equation 9 assuming turbulent flow:

$$q = N_1 (1.0) C_v \left(\frac{\Delta p_t}{G_f} \right)^{1/2}$$

$$1070 = (1.0)(1.0) 400 \left(\frac{\Delta p_t}{0.84} \right)^{1/2}$$

$$\Delta p_t = 601 \text{ psi}$$

Using Equation F-2 assuming laminar flow:

$$q = N_s (F_s C_v)^{3/2} \frac{\Delta p_s}{\mu}$$

$$1070 = 47 [1.25(400)]^{3/2} \frac{\Delta p_s}{5900}$$

$$\Delta p_s = 12.0 \text{ psi}$$

Using Equation F-5 for transitional flow:

$$F_R = 1.084 - 0.375 \left(\frac{12.0}{6.01} \right)^{0.336} = 0.61$$

Because F_R is between 0.48 and 0.98, the flow is transitional.

Find the pressure drop using Equation 9:

$$q = N_1 F_R C_v \left(\frac{\Delta p}{G_f} \right)^{1/2}$$

$$1070 = 1.0(0.61)(400) \left(\frac{\Delta p}{0.84} \right)^{1/2}$$

$$\Delta p = 16 \text{ psi}$$

Note that the pseudo values of Δp , assuming turbulent (6 psi) or laminar flow (12 psi), are not applicable, because the flow is actually transitional.

PROBLEM 3. Find the valve size.

Given: $q = 17 \text{ m}^3/\text{h}$, $\rho = 1100 \text{ kg/m}^3$, $\Delta p = 69 \text{ kPa}$, $\mu = 1000 \text{ N}\cdot\text{s/m}^2$ (or 10^6 cp)

Selected valve: Ball, $C_v / d^2 = 30$, $F_s = 1.3$

Using Equation 9 for turbulent flow:

$$q = N_1 F_R C_{vt} \left(\frac{\Delta p}{G_f} \right)^{1/2}$$

$$17 = 0.0865(1.0) C_{vt} \left(\frac{69}{1.1} \right)^{1/2}$$

$$C_{vt} = 24.8$$

Using Equation F-2 for laminar flow:

$$C_{vs} = \frac{1}{F_s} \left(\frac{q\mu}{N_s \Delta p} \right)^{2/3}$$

$$C_{vs} = \frac{1}{1.3} \left[\frac{17(10^6)}{1.5(69)} \right]^{2/3}$$

$$C_{vs} = 2310$$

For transitional flow:

$$F_R = 1.044 - 0.358 \left(\frac{2310}{24.8} \right)^{0.655} = -5.9$$

A value less than 0.48 indicates laminar flow. Therefore, the required C_v is 2310. To meet this requirement, a 250-mm (10-in) valve has a $C_v = 30(10)^2 = 3000$.

Annex G — Liquid critical pressure ratio factor F_F

Flow rate is a function of the pressure drop from the valve inlet to the vena contracta. Under nonvaporizing liquid flow conditions, the apparent vena contracta pressure (p_{vc}) can be predicted from the downstream pressure (p_2), because the pressure recovery is a consistent fraction of the pressure drop to the vena contracta. The effect of this pressure recovery is recognized in the valve flow coefficient (C_v).

Under choked flow conditions, there is no relationship between p_2 and p_{vc} because vaporization affects pressure recovery. The liquid critical pressure ratio factor is used to predict p_{vc} . It is the ratio of the apparent vena contracta pressure under choked flow conditions to the vapor pressure of the liquid at its inlet temperature.

An equation for predicting F_F has been published in previous standards. A theoretical equation based on the assumption (see Allen reference, *Flow of a Flashing Mixture of Water and Steam through Pipes and Valves*) that the fluid is always in thermodynamic equilibrium states that:

$$F_F = 0.96 - 0.28 \left(\frac{p_v}{p_c} \right)^{1/2} \quad (\text{Eq. G-1})$$

Because a liquid does not remain in thermodynamic equilibrium as it flashes across a valve (see Bailey reference, *Metastable Flow of Saturated Water*), the actual flow rate will be greater than that predicted by the use of Equation G-1.

In experiments with nonvalve restrictions (see Burnell reference, *Flow of Boiling Water through Nozzles, Orifices and Pipes*), the following equation for F_F was derived:

$$F_F = 1 - \frac{\sigma}{F_o} \quad (\text{Eq. G-2})$$

where σ is the surface tension of the liquid in N/m and F_o is an experimentally determined orifice factor for the restriction or valve in the same units. This equation allows for the fact that liquids vaporizing across a restriction are not in thermodynamic equilibrium, but become metastable and choke at a critical vena contracta pressure. The equation has been tested only for deaerated water. Limited data indicate that values of F_o for values at rated travel range from around 0.2 N/m for a streamlined angle valve to nearly 1.0 for a more tortuous double-ported globe valve. The surface tension of water in N/m can be approximated based on the Othmer equation:

$$\sigma = \left[\frac{(374 - ^\circ\text{C})}{4080} \right]^{1.05} \quad (\text{Eq. G-3})$$

or

$$\sigma = \left[\frac{(705 - ^\circ\text{F})}{7340} \right]^{1.05} \quad (\text{Eq. G-4})$$

Annex H — Derivation of factor x_{TP}

The slope of the Y versus x curve for any specific valve is determined using air or gas as the test fluid, and is designated by the value of x at $Y = 2/3$. This value, known as x_T , is the pressure drop ratio factor. For most valves, it is less than 1.0, but it may be greater for some valve styles.

If a valve is installed with a fitting at the inlet and/or outlet, the pressure drop ratio factor for the combination of the valve plus the fitting (x_{TP}) usually differs from that of the valve alone.

Let us consider a valve with reducers operating at choked flow

$$[x = x_{TP}, Y = Y_T \text{ for an ideal gas } (Z = 1)]$$

From Equation 18, the volumetric valve flow equation (in U.S. customary units) is:

$$q_T = 1360 F_P C_v p_1 Y_T \sqrt{\frac{x_{TP}}{G_g T_1}} \quad (\text{Eq. H-1})$$

where the subscript T indicates the terminal or choked condition. For the valve alone at choked flow, the equation is:

$$q_T = 1360 C_v p_i Y_T \sqrt{\frac{x_T}{G_g T_1}} \quad (\text{Eq. H-2})$$

where p_i is the valve inlet pressure. From Equations H-1 and H-2, we have:

$$p_i = F_P p_1 \sqrt{\frac{x_{TP}}{x_T}} \quad (\text{Eq. H-3})$$

From the gas laws, the mean specific weight across the inlet reducer is:

$$\gamma_1 = \left(\frac{p_1 + p_i}{2} \right) \frac{M}{RT_1} = \frac{144(p_1 + p_i)}{2} \left(\frac{28.97 G_g}{1545 T_1} \right)$$
$$\gamma_1 = 1.350(p_1 + p_i) \frac{G_g}{T_1} \quad (\text{Eq. H-4})$$

Since the pressure drop, expressed in feet of head, is $K (U^2/2g)$,

$$\frac{144(p_1 - p_i)}{\gamma} = \frac{K}{2g} U^2 \quad \text{or} \quad \frac{144(p_1 - p_i)}{1.350(p_1 + p_i) \frac{G_g}{T_1}}$$

$$= \frac{K}{2g} \left[\frac{q}{3600} \frac{14.73}{0.5(p_1 + p_i)} \frac{T_1}{519.69} \frac{4(144)}{\pi d^2} \right]^2$$

Simplifying:

$$p_1^2 - p_i^2 = 1.214(10^{-9}) K G_g T_1 q^2 d^{-4} \quad (\text{Eq. H-5})$$

Substituting the expression for p_i from Equation H-3, we have:

$$p_1^2 - F_p^2 p_1^2 \left(\frac{x_{TP}}{x_T} \right) = 1.214(10^{-9}) K G_g T_1 q^2 d^{-4} \quad (\text{Eq. H-6})$$

From Equation H-1:

$$q_T^2 G_g \frac{T_1}{p_1^2} = (1360 F_p C_v Y_T)^2 x_{TP} \quad (\text{Eq. H-7})$$

Substituting this into Equation H-6, with $q = q_T$ and $K = K_i$, we have:

$$1.214(10^9)(1360 F_p C_v Y_T)^2 K_i \frac{x_{TP}}{d^4} = 1 - F_p^2 \frac{x_{TP}}{x_T} \quad (\text{Eq. H-8})$$

Solving for x_{TP} , with $Y_T = 2/3$, we have:

$$x_{TP} = \frac{x_T}{F_p^2} \left(\frac{K_i x_T}{1000} \frac{C_v^2}{d^4} + 1 \right)^{-1} \quad (\text{Eq. H-9})$$

Annex I — Control valve flow equations — SI Notation (International System of Units)

The valve flow coefficient that is compatible with SI units is A_v (see IEC reference, *Industrial Process Control Valves*). At the present time, A_v does not have wide acceptance by the technical community. This annex has been included for the benefit of those who wish to use pure, coherent SI units.

In the following equations, certain symbols commonly associated with SI practice differ from those listed in [Section 3](#). These are:

A_v Valve flow coefficient, $m^2 [A_v = C_v \times 24(10^{-6})]$

ζ (Zeta) Head loss coefficient ($\zeta = K$), dimensionless

ρ (Rho) Density, kg/m^3

Liquid Equations

Turbulent flow:

$$q = F_p A_v \left(\frac{\Delta p}{\rho} \right)^{1/2} \quad (\text{Eq. I-1})$$

$$w = F_p A_v (\Delta p \rho)^{1/2} \quad (\text{Eq. I-2})$$

$$F_p = \left[\left(\frac{\sum \zeta A_v^2}{1.23 d^4} \right) + 1 \right]^{-1/2} \quad (\text{Eq. I-3})$$

Choked flow:

$$q = F_{LP} A_v \left[\frac{p_1 - p_{vc}}{\rho} \right]^{1/2} \quad (\text{Eq. I-4})$$

$$w = F_{LP} A_v [p(p_1 - p_{vc})]^{1/2} \quad (\text{Eq. I-5})$$

$$F_{LP} = F_L \left[\left(\frac{\zeta_1 F_L^2 A_v^2}{1.23 d^4} \right) + 1 \right]^{-1/2} \quad (\text{Eq. I-6})$$

where $\zeta_i = \zeta_{s1} + \zeta_{B1}$

$$p_{vc} = F_F p_v \quad (\text{See Annex G for } F_F) \quad (\text{Eq. I-7})$$

Laminar flow ([see Annex F](#)):

$$q_s = (F_s A_v)^{3/2} \frac{\Delta p}{280 \mu} \quad (\text{Eq. I-8})$$

$$w_s = (F_s A_v)^{3/2} \frac{\Delta p \rho}{280 \mu} \quad (\text{Eq. I-9})$$

$$F_s = \frac{F_d^{2/3}}{F_L^{1/3}} \left(\frac{F_L^2 A_v^2}{1.23 d^4} + 1 \right)^{1/6} \quad (\text{Eq. I-10})$$

Transitional flow:

$$q = F_R A_v \left(\frac{\Delta p}{\rho} \right)^{1/2} \quad (\text{Eq. I-11})$$

$$w = F_R A_v (\Delta p \rho)^{1/2} \quad (\text{Eq. I-12})$$

$$F_R = 1.044 - 0.358 \left(\frac{A_{vs}}{A_{vt}} \right)^{0.655} \quad (\text{Eq. I-13})$$

$$F_R = 1.084 - 0.375 \left(\frac{\Delta p_s}{\Delta p_t} \right)^{0.336} \quad (\text{Eq. I-14})$$

$$F_R = 1.004 - 0.358 \left(\frac{q_t}{q_s} \right)^{0.588} \quad (\text{Eq. I-15})$$

Limits for $F_R = 0.48$ to 1.0 .

Gas and Vapor Equations

Turbulent flow:

$$w = F_p A_v Y (x p_1 \rho_1)^{1/2} \quad (\text{Eq. I-16})$$

$$q = 0.246 F_p A_v p_1 Y \left(\frac{x}{M T_1 Z} \right)^{1/2} \quad (\text{Eq. I-17})$$

(Normal m³ at 0°C and 101.3 kPa)

Limit: $x \leq F_k x_{TP}$ (in equation only)

$$x_{TP} = \frac{x_T}{F_p^2} \left[\left(0.72 x_T \zeta_i \frac{A_v^2}{d^4} + 1 \right) \right]^{-1} \quad (\text{Eq. I-18})$$

where

$$\zeta_i = \zeta_1 + \zeta_{B1} \quad (\text{Eq. I-19})$$

Annex J — References

INTERNATIONAL ELECTROTECHNICAL COMMISSION (IEC)

IEC/534-1	Part 1: General Considerations, 1976
IEC/534-2	Part 2: Flow Capacity. Section One — Sizing Equations for Incompressible Fluid Flow under Installed Conditions, 1978
IEC/534-2-2	Part 2: Flow Capacity. Section Two — Sizing Equations for Compressible Fluid Flow under Installed Conditions, 1980
IEC/534-2-3	Part 2: Flow Capacity. Section Three — Test Procedures, 1983

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ISA

ANSI/ISA-S75.02-1988 Control Valve Capacity Test Procedure

Baumann, H. D. "Effect of Pipe Reducers on Control Valve Capacity," *Instruments and Control Systems*, December 1968, pp. 92–102.

Baumann, H. D. "The Introduction of a Critical Flow Factor for Valve Sizing," *ISA Transactions*, Vol. 2, 1963, pp. 107–111.

Buresh, J. F., and C. B. Schuder, "The Development of a Universal Gas Sizing Equation for Control Valves," *ISA Transactions*, Vol. 3, 1964, pp. 322–328.

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

ISA Handbook of Control Valves, 2d ed., 1976

McCutcheon, E. B. "A Reynolds Number for Control Valves," *Symposium on Flow, Its Measurement and Control in Science and Industry*, Vol. I, Part 3, 1974, pp. 1087–1090.

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MISCELLANEOUS

Allen, W. F., Jr., "Flow of a Flashing Mixture of Water and Steam through Pipes and Valves," *Journal of Basic Engineering*, April 1951, pp. 357–365.

Bailey, J. F., "Metastable Flow of Saturated Water," *Journal of Basic Engineering*, November 1951, pp. 1109–1116.

Burnell, J.G., "Flow of Boiling Water through Nozzles, Orifices and Pipes," *Engineering*, December 12, 1947, pp. 572–576.

Driskell, L.R., "New Approach to Control Valve Sizing," *Hydrocarbon Processing*, July 1969, pp. 111–114.

Stiles, G.F., "Liquid Viscosity Effects on Control Valve Sizing," Technical Manual TM 17A, October 1967, Fisher Governor Company, Marshalltown, IA.

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ISBN: 0-87664-899-5