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(JVMA)

**Industrial-process control valves—
Part 2-1 : Flow capacity—
Sizing equations for fluid flow
under installed conditions**

ICS 23.060.40; 25.040.40

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Foreword

This translation has been made based on the original Japanese Industrial Standard established by the Minister of Economy, Trade and Industry through deliberations at the Japanese Industrial Standards Committee according to the proposal of establishing a Japanese Industrial Standard from Japan Valve Manufacturers' Association (JVMA), with a draft of Industrial Standard based on the provision of Article 12 Clause 1 of the Industrial Standardization Law.

This Standard has been made based on **IEC 60534-2-1:1998** *Industrial-process control valves—Part 2-1: Flow capacity—Sizing equations for fluid flow under installed conditions* for the purposes of making it easier to compare this Standard with International Standard; to prepare Japanese Industrial Standard conforming with International Standard; and to propose a draft of an International Standard which is based on Japanese Industrial Standard.

Attention is drawn to the possibility that some parts of this Standard may conflict with a patent right, application for a patent after opening to the public, utility model right or application for registration of utility model after opening to the public which have technical properties. The relevant Minister and the Japanese Industrial Standards Committee are not responsible for identifying the patent right, application for a patent after opening to the public, utility model right or application for registration of utility model after opening to the public which have the said technical properties.

JIS B 2005 consists of the following 12 parts under the general title “*Industrial-process control valves*”

Part 1: Control valve terminology and general considerations

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

Part 2-3: Flow capacity—Test procedures

Part 2-4: Flow capacity—Section Four: Inherent flow characteristics and rangeability

Part 3-1: Dimensions—Face-to-face dimensions for flanged, two-way, globe-type, straight pattern and centre-to-face dimensions for flanged, two-way, globe-type, angle pattern control valves

Part 3-2: Dimensions—Face-to-face dimensions for rotary control valves except butterfly valves

Part 3-3: Dimensions—End-to-end dimensions for butt-weld, two-way, globe-type, straight pattern control valves

Part 5: Marking

Part 6-1: Mounting details for attachment of positioners to control valves—Section 1: Positioner mounting on linear actuators

Part 6-2: Mounting details for attachment of positioners to control valves—Positioner mounting on rotary actuators

Part 7: Control valve data sheet

Part 8-1: Noise considerations—Section One: Laboratory measurement of noise generated by aerodynamic flow through control valves

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the original JIS is to be the final authority.

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Contents

	Page
Introduction.....	1
1 Scope	1
2 Normative references	1
3 Definitions	2
3.1 valve style modifier F_d	2
4 Installation.....	2
5 Symbols	2
6 Sizing equations for incompressible fluids.....	4
6.1 Turbulent flow	4
6.2 Non-turbulent (laminar and transitional) flow	5
7 Sizing equations for compressible fluids	6
7.1 Turbulent flow	6
7.2 Non-turbulent (laminar and transitional) flow	7
8 Determination of correction factors.....	8
8.1 Piping geometry factor F_P	8
8.2 Reynolds number factor F_R	9
8.3 Liquid pressure recovery factors F_L or F_{LP}	11
8.4 Liquid critical pressure ratio factor F_F	11
8.5 Expansion factor Y	11
8.6 Pressure differential ratio factor x_T or x_{TP}	12
8.7 Specific heat ratio factor F_γ	13
8.8 Compressibility factor Z	13
Annex A (informative) Derivation of valve style modifier F_d	21
Annex B (informative) Control valve sizing flow charts.....	25
Annex C (informative) Physical constants	29
Annex D (informative) Examples of sizing calculations.....	30
Annex E (informative) Bibliography	41

Industrial-process control valves— Part 2-1 : Flow capacity—Sizing equations for fluid flow under installed conditions

Introduction This Japanese Industrial Standard has been prepared based on the first edition of **IEC 60534-2-1** *Industrial-process control valves—Part 2-1: Flow capacity—Sizing equations for fluid flow under installed conditions* published in 1998 without modifying the technical contents.

1 Scope This part of **JIS B 2005** 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 **JIS B 2005** 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 $K_v/d^2 < 0.04(C_v/d^2 < 0.047)$.

Remarks : The International Standard corresponding to this Standard is as follows.

In addition, symbols which denote the degree of correspondence in the contents between the relevant International Standard and **JIS** are IDT (identical), MOD (modified), and NEQ (not equivalent) according to **ISO/IEC Guide 21**.

IEC 60534-2-1:1998 *Industrial-process control valves—Part 2-1: Flow capacity—Sizing equations for fluid flow under installed conditions* (IDT)

2 Normative references The following standards contain provisions which, through reference in this text, constitute provisions of this Standard. The most recent editions of the standards (including amendments) indicated below shall be applied.

JIS B 2005-1 *Industrial-process control valves—Part 1: Control valve terminology and general considerations*

NOTE : **IEC 60534-1**:1987 *Industrial-process control valves. Part 1: Control valve terminology and general considerations* is identical with the said standard.

JIS B 2005-2-3 *Industrial-process control valves—Part 2-3: Flow capacity—Test procedures*

NOTE : **IEC 60534-2-3**:1997 *Industrial-process control valves—Part 2-3: Flow capacity—Test procedures* is identical with the said standard.

3 Definitions For the purpose of this part of **JIS B 2005**, definitions given in **JIS B 2005-1** 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.

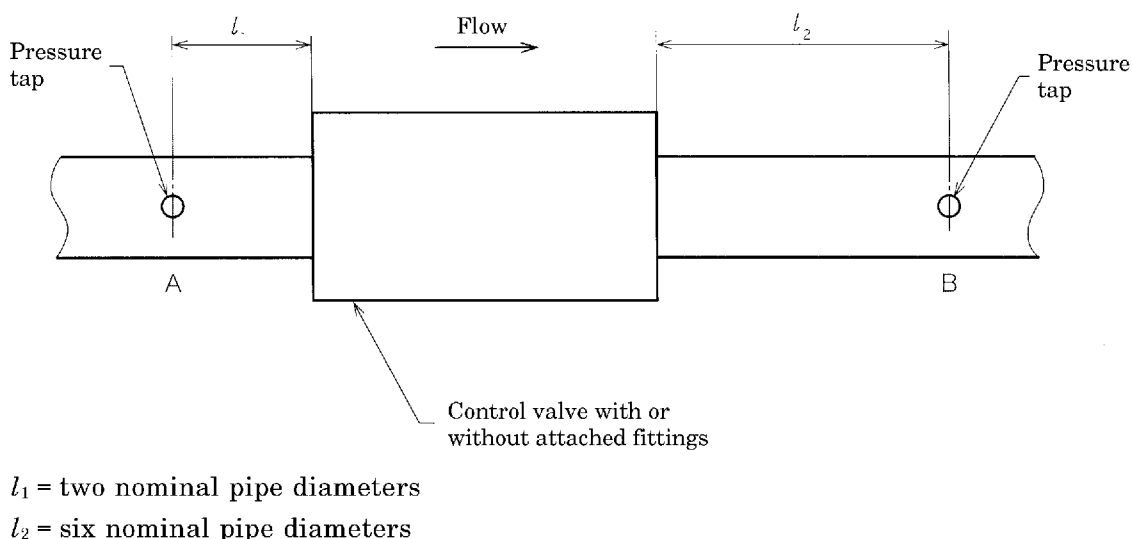


Figure 1 Reference pipe section for sizing

5 Symbols For the purpose of this part of **JIS B 2005**, the following symbols apply.

Symbol	Description	Unit
C	Flow coefficient (K_v , C_v)	Various (see JIS B 2005-1) ⁽⁴⁾
C_i	Assumed flow coefficient for iterative purposes	Various (see JIS B 2005-1) ⁽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 ⁽⁴⁾
F_F	Liquid critical pressure ratio factor	1
F_L	Liquid pressure recovery factor of a control valve without attached fittings	1 ⁽⁴⁾
F_{LP}	Combined liquid pressure recovery factor and piping geometry factor of a control valve with attached fittings	1 ⁽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 ⁽¹⁾
p_1	Inlet absolute static pressure measured at point A (see figure 1)	kPa or bar ⁽²⁾
p_2	Outlet absolute static pressure measured at point B (see figure 1)	kPa or bar ⁽²⁾
p_c	Absolute thermodynamic critical pressure	kPa or bar ⁽²⁾
p_r	Reduced pressure (p_1/p_c)	1
p_v	Absolute vapour pressure of the liquid at inlet temperature	kPa or bar ⁽²⁾
Δp	Differential pressure between upstream and downstream pressure taps ($p_1 - p_2$)	kPa or bar ⁽²⁾
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	°C
T_r	Reduced temperature (T_1/T_c)	1
t_s	Absolute reference temperature for standard cubic metre	°C
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
x_{TP}	Pressure differential ratio factor of a control valve with attached fittings at choked flow	1

Symbol	Description	Unit
Y	Expansion factor	1
Z	Compressibility factor	1
ν	Kinematic viscosity	$\text{m}^2/\text{s}^{(3)}$
ρ_1	Density of fluid at p_1 and T_1	kg/m^3
ρ_1/ρ_0	Relative density ($\rho_1/\rho_0 = 1.0$ 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

NOTES (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.

(2) $1 \text{ bar} = 10^2 \text{ kPa} = 10^5 \text{ Pa}$

(3) $1 \text{ centistoke} = 10^{-6} \text{ m}^2/\text{s}$

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

(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 101.325 kPa 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 **JIS B 2005-1**.

6.1.1 Non-choked turbulent flow

6.1.1.1 Non-choked turbulent flow without attached fittings

[Applicable if $\Delta p < F_L^2(p_1 - F_F \times p_v)$]

The flow coefficient shall be determined by

$$C = \frac{Q}{N_1} \sqrt{\frac{\rho_1/\rho_0}{\Delta p}} \dots\dots\dots (1)$$

NOTES 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 .

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

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

The flow coefficient shall be determined as follows:

$$C = \frac{Q}{N_1 F_P} \sqrt{\frac{\rho_1/\rho_0}{\Delta p}} \dots\dots\dots (2)$$

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

[Applicable if $\Delta p \geq F_L^2(p_1 - F_F \times p_v)$]

The flow coefficient shall be determined as follows:

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

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

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

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

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

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 **JIS B 2005-1**. This equation is applicable if $Re_v < 10\,000$ [see equation (28)].

6.2.1 Non-turbulent flow without attached fittings The flow coefficient shall be calculated as follows:

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

6.2.2 Non-turbulent flow with attached fittings For non-turbulent flow, the effect of close-coupled reducers or other flow-disturbing fittings is unknown. While there is no information on the laminar or transitional flow behaviour of control valves installed between pipe reducers, the user of such valves is advised to utilize the appropriate equations for line-sized valves in the calculation of the F_R factor. This should result in conservative flow coefficients, since additional turbulence created by reducers and expanders will further delay the onset of laminar flow. Therefore, it will tend to increase the respective F_R factor for a given valve Reynolds number.

7 Sizing equations for compressible fluids The equations listed below identify the relationships between flow rates, flow coefficients, related installation factors and pertinent service conditions for control valves handling compressible fluids. Flow rates for compressible fluids may be encountered in either mass or volume units and thus equations are necessary to handle both situations. Flow coefficients may be calculated using the appropriate equations selected from the following. A sizing flow chart for compressible fluids is given in annex B.

7.1 Turbulent flow

7.1.1 Non-choked turbulent flow

7.1.1.1 Non-choked turbulent flow without attached fittings

(Applicable if $x < F_T x_T$)

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

$$C = \frac{W}{N_6 Y \sqrt{x p_1 \rho_1}} \dots\dots\dots (6)$$

$$C = \frac{W}{N_8 p_1 Y} \sqrt{\frac{T_1 Z}{x M}} \dots\dots\dots (7)$$

$$C = \frac{Q}{N_9 p_1 Y} \sqrt{\frac{M T_1 Z}{x}} \dots\dots\dots (8)$$

NOTES 1 Refer to **8.5** for details of the expansion factor Y .

2 See annex C for values of M .

7.1.1.2 Non-choked turbulent flow with attached fittings

(Applicable if $x < F_T x_{TP}$)

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

$$C = \frac{W}{N_6 F_P Y \sqrt{x p_1 \rho_1}} \dots\dots\dots (9)$$

$$C = \frac{W}{N_8 F_P p_1 Y} \sqrt{\frac{T_1 Z}{x M}} \dots\dots\dots (10)$$

$$C = \frac{Q}{N_9 F_P p_1 Y} \sqrt{\frac{M T_1 Z}{x}} \dots\dots\dots (11)$$

NOTES 1 Refer to **8.1** for the piping geometry factor F_P .

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

(Applicable if $x \geq F_\gamma x_T$)

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

$$C = \frac{W}{0.667 N_6 \sqrt{F_\gamma x_T p_1 \rho_1}} \dots\dots\dots (12)$$

$$C = \frac{W}{0.667 N_8 p_1} \sqrt{\frac{T_1 Z}{F_\gamma x_T M}} \dots\dots\dots (13)$$

$$C = \frac{Q}{0.667 N_9 p_1} \sqrt{\frac{M T_1 Z}{F_\gamma x_T}} \dots\dots\dots (14)$$

7.1.2.2 Choked turbulent flow with attached fittings

(Applicable if $x \geq F_\gamma x_{TP}$)

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

$$C = \frac{W}{0.667 N_6 F_P \sqrt{F_\gamma x_{TP} p_1 \rho_1}} \dots\dots\dots (15)$$

$$C = \frac{W}{0.667 N_8 F_P p_1} \sqrt{\frac{T_1 Z}{F_\gamma x_{TP} M}} \dots\dots\dots (16)$$

$$C = \frac{Q}{0.667 N_9 F_P p_1} \sqrt{\frac{M T_1 Z}{F_\gamma x_{TP}}} \dots\dots\dots (17)$$

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 **JIS B 2005-1**. These equations are applicable if $Re_v < 10\,000$ [see equation (28)]. In this subclause, the 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:

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

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

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 $\pm 5\%$, the F_P factor shall be determined by test in accordance with **JIS B 2005-2-3**.

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

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

In this equation, the factor $\sum \zeta$ 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.

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

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

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

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:

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

Outlet reducer (expander):

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

Inlet and outlet reducers of equal size:

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

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 low 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:

$$C_i = 1.3 C \quad \text{..... (26)}$$

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 2a or figure 2b. Then, determine if

$$\frac{C}{F_P} \leq C_i \quad \text{..... (27)}$$

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 integration 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:

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

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 $\pm 5 \%$ for F_d , the F_d factor shall be determined by test in accordance with **JIS B 2005-2-3**.

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

$$\frac{C}{F_R} \leq C_i \quad \text{..... (29)}$$

If the condition of equation (29) is satisfied, then use the C_i established from equation (26). If the condition of equation (29) is not met, then repeat the above procedure by again increasing C_i by 30 %. This may require several iterations until the condition required in equation (29) is met.

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

$$F_R = 1 + \left(\frac{0.33 F_L^{1/2}}{n_1^{1/4}} \right) \log_{10} \left(\frac{Re_v}{10\,000} \right) \dots\dots\dots (30)$$

for the transitional flow regime,

where

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

or

$$F_R = \frac{0.026}{F_L} \sqrt{n_1 Re_v} \quad (\text{not to exceed } F_R = 1) \dots\dots\dots (31)$$

for the laminar flow regime.

- NOTES 1 Use the lower value of F_R from equations (30) and (31). If $Re_v < 10$, use only equation (31).
- 2 Equation (31) is applicable to fully developed laminar flow (straight lines in figure 3). The relationships expressed in equations (30) and (31) are based on test data with valves at rated travel and may not be fully accurate at lower valve travels.
- 3 In equations (30a) and (31), C_i/d^2 must not exceed 0.04 when K_v is used or 0.047 when C_v is used.

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

$$F_R = 1 + \left(\frac{0.33 F_L^{1/2}}{n_2^{1/4}} \right) \log_{10} \left(\frac{Re_v}{10\,000} \right) \dots\dots\dots (32)$$

for the transitional flow regime,

where

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

or

$$F_R = \frac{0.026}{F_L} \sqrt{n_2 Re_v} \quad (\text{not to exceed } F_R = 1) \dots\dots\dots (33)$$

for the laminar flow regime.

- NOTES 1 Select the lower value from equations (32) and (33). If $Re_v < 10$, use only equation (33).
- 2 Equation (33) is applicable to fully developed laminar flow (straight lines in figure 3).

8.3 Liquid pressure recovery factors F_L or F_{LP}

8.3.1 Liquid pressure recovery factor without attached fittings F_L F_L is the liquid pressure recovery factor of the valve without attached fittings. This factor accounts for the influence of the valve internal geometry on the valve capacity at choked flow. It is defined as the ratio of the actual maximum flow rate under choked flow conditions to a theoretical, non-choked flow rate which would be calculated if the pressure differential used was the difference between the valve inlet pressure and the apparent *vena contracta* pressure at choked flow conditions. The factor F_L may be determined from tests in accordance with **JIS B 2005-2-3**. Typical values of F_L versus percent of rated flow coefficient are shown in figure 4.

8.3.2 Combined liquid pressure recovery factor and piping geometry factor with attached fittings F_{LP} F_{LP} is the combined liquid pressure recovery factor and piping geometry factor for a control valve with attached fittings. It is obtained in the same manner as F_L .

To meet a deviation of $\pm 5\%$ for F_{LP} , F_{LP} shall be determined by testing in accordance with **JIS B 2005-2-3**. When estimated values are permissible, the following equation shall be used:

$$F_{LP} = \frac{F_L}{\sqrt{1 + \frac{F_L^2}{N_2} (\sum \xi_1) \left(\frac{C}{d^2}\right)^2}} \dots\dots\dots (34)$$

Here $\sum \xi_1$ is the velocity head loss coefficient, $\xi_1 + \xi_{B1}$, of the fitting attached upstream of the valve as measured between the upstream pressure tap and the control valve body inlet.

8.4 Liquid critical pressure ratio factor F_F F_F is the liquid critical pressure ratio factor. This factor is the ratio of the apparent *vena contracta* pressure at choked flow conditions to the vapour pressure of the liquid at inlet temperature. At vapour pressures near zero, this factor is 0.96.

Values of F_F may be determined from the curve in figure 5 or approximated from the following equation:

$$F_F = 0.96 - 0.28 \sqrt{\frac{P_v}{p_c}} \dots\dots\dots (35)$$

8.5 Expansion factor Y The expansion factor Y accounts for the change in density as the fluid passes from the valve inlet to the *vena contracta* (the location just downstream of the orifice where the jet stream area is a minimum). It also accounts for the change in the *vena contracta* area as the pressure differential is varied.

Theoretically, Y is affected by all of the following: