ASME STANDARD

Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi

ASME MFC-3M-1985

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Foreword

Until now there has been no U. S. Standard covering all of the measurements of fluid flows through closed conduits and pipes and using differential pressure devices (primary elements). Most people have used for guidance the ASME "Fluid Meters". The International Standards Organization (ISO) developed a general use standard ISO 5167, but incorporating expressions for discharge coefficients that the A.S.M.E. Fluid Meters Research committee determined covers a considerably broader span of fluids and flowing conditions and reduced the uncertainty in the prediction of the discharge coefficient.

This Standard has been prepared by the ASME Standards Committee for the Measurement of Fluid Flow in Closed Conduits (MFFCC) and incorporates the ASME-ISO orifice coefficient equations in both US and SI units. It is intended to cover the broader requirements of flow measurements found throughout industry using differential producing flowmeters. This Standard is intended to be a practical working document, with representative calculations for some of the equations given in Appendix B.

In order to assist the U. S. user to participate better in international trade, this Standard has been made as consistent and technically equivalent as practical with the ISO 5167 and it contains the same SI units and also customary US units, in brackets, [ ]. There have been some technical and many editorial changes made in consideration of U.S. practice and some new technical insights.

This document was approved as an ASME Standard on September 24, 1985.

The change in scope was approved by the Board on Standardization and accepted by the Council on Codes and Standards.

ERRATA

to

ASME MFC-3M-1985

MEASUREMENT OF FLUID FLOW IN PIPES USING ORIFICE, NOZZLE, AND VENTURI

p 2	Uniform equivalent roughness, delete $[\epsilon]$, use K for both US and SI
p 5	2.4.3, 4th line, change '32' to '31'
p 6	line 16, Change '32' to '31'
p 7	line above Eq 9, add at end 'See 6.3.2.2
p 10	3.3 c), line 6, delete 'and therefore......temperature'
	line 11, delete 'If there is.......coefficients'
p 12	line 46 (line above 5.1.5), add at end 'for orifice plates and Table 6 for venturis'
p 14	(Eq 19), change 0.015 to 0.0025
p 15	5.3, 3), change to 'Calibrate in-situ or in a calibration installation as nearly as possible identical to the planned flow measurement'
	1), add 'and' at end of sentence
	3), change 'calibrated' to 'calibrate'
p 15	5.4, all paragraphs, change 'table 2' to 'tables 2 & 6'
p 16	Table 2, change 'orifice plates' to 'nozzles and orifice plates'
	Add 3) in caption of columns 2 nd 3
	Add at bottom of page 'Note 3) The insertion of 5 to 10D straight lengths between the bends is sufficient to make the combined effect the same as the single bends in the left column.'
p 17	1), 2), 3), rearrange to be same as 5.3
p 19	Eq 20, 21, 22 change 'ρ' to 'ρ_f' and 'u' to 'U'
p 25	Fig 4, add to upper part 'D, D/2 taps', to lower 'Flange Taps'
p 29	Delete all references to ϵ and use K.
p 31	6.3.3.1, line 4, delete '$[\epsilon/D]$'
p 33	7.1.2, line 3, change '90' to '45'
p 39	8.2.1, line 6, change '90' to '45'
	8.2.3, line 9, change '90' to '45'
p 42	Fig 8, title, add after 'ASME' 'Rough Cast'
	add notation '------- uncertainty band'
p 45	line 2, delete $[\epsilon/D]$
	Table 6, add as note 6) the same note as 3) in Table 2.
p 47	Eq 38a & 38b, 4th term in brackets, denominator, change 'D' to 'd'
	final term in brackets, denominator, change 'ρ' to '2ρ'
p 49	Heading, delete $[\epsilon]$
	US column heading, change 'ϵ' to 'K'
p 50	Table B2, change 'k=1.4' to 'Isentropic exponent κ=1.4'
	'k=1.3' to 'Isentropic exponent κ=1.3'
p 53	C.4, delete '$[\epsilon/D]$

CONTENTS

ASME STANDARD

MEASUREMENT OF FLUID FLOW IN PIPES USING ORIFICE, NOZZLE, AND VENTURI

1. SCOPE AND FIELD OF APPLICATION

This Standard specifies the geometry and method of use (installation and flowing conditions) for orifice plates, nozzles and venturi tubes when they are inserted in a conduit running full, to determine the rate of the fluid flowing. It also gives necessary information for calculating flow rate and its associated uncertainty.

It applies only to pressure difference devices in which the flow remains turbulent and subsonic throughout the measuring section, is steady or varies only slowly with time and the fluid is considered single-phased. In addition, the uncertainties are given in the appropriate sections of this Standard for each of these devices, within the pipe size and Reynolds number limits which are specified.

It deals with devices for which sufficient calibrations have been made to enable the specification of coherent systems of application and to enable calculations to be made with certain predictable limits of uncertainty.

The devices introduced into the pipe are called "primary devices." The term primary device also includes the pressure taps and the associated upstream and downstream piping. All other instruments or devices required for the measurement or transmission of the differential pressures are known as "secondary elements", and in combination are referred to as the secondary devices. This Standard covers the primary devices; secondary devices[1] will be mentioned only occasionally.
The different primary devices covered in this Standard are as follows:

- orifice plates, which can be used with the various following arrangements of pressure taps:
  - flange pressure taps.
  - D and D/2 pressure taps,[2]
  - corner pressure taps

- nozzles:
  - ASME long radius nozzles.

- venturi tubes
  - classical venturi tubes [3],

Please note that this Standard does not cover pipe or conduit sizes under 50 mm [2 in](nominal).

This Standard does not apply to ASME Performance Test Code Measurements.

This Standard is applicable to measurement of flow of any fluid, (liquid, vapor, or gas)

[1] See ISO 2186, 'Fluid flow in closed conduits - Connections for pressure signal transmission between primary and secondary devices' and API RP550.
[2] Orifice plates with vena contracta taps are not considered in this Standard.
[3] In the US the classical venturi tube is sometimes called the Herschel venturi tube.

2. SYMBOLS AND DEFINITIONS

The vocabulary and symbols used in this Standard are defined in ANSI/ASME MFC-1M 'Glossary of the Terms Used in the Measurement of Fluid Flow in Pipes' and in ISO 4006, 'Measurement of fluid flow in closed conduits - Vocabulary and symbols'. SI and US measurement units are used throughout, the SI units first and the US units following in brackets, [ ].

Table 1 reproduces the symbols which are used in this Standard.

The definitions in the following sections are given only for terms used in some special sense or for terms the meaning of which it seems useful to emphasize.

2.1 Symbols

TABLE 1 - Symbols

Symbols — Represented quantity (US symbols are given in [] when different from SI)	Dimensions: M: mass, L: length, T: time, θ: temperature	SI Unit	US Unit (see Note) Customary
C Discharge Coefficient	dimensionless		
C_p Specific heat at constant pressure	$L^2T^{-2}\theta^{-1}$	$\frac{J}{kg \cdot mole \cdot K}$	$\frac{BTU}{lb_m \cdot mole \cdot {}^\circ R}$
d Diameter of orifice or throat of primary device at flowing conditions	L	m	in
d_{meas} Diameter at a specified measured temperature	L	m	in
D Upstream internal pipe diameter (or upstream diameter of a classical venturi tube) at flowing conditions	L	m	in
e Relative uncertainty	dimensionless		
E Orifice plate thickness	L	m	in
e_x Orifice eccentricity	dimensionless		
F_a Thermal Expansion Correction Factor	dimensionless		
G_i (gas) $Mw_{(gas)}/Mw_{(air)}$ [Ideal Specific Gravity]	dimensionless		
$Mw_{(gas)}$ = Molecular weight of the gas	M	kg/mole	lb_m/mole
$Mw_{(air)}$ = Molecular weight of air=28.9624	M	kg/mole	lb_m/mole
G (liquids) Relative density [Specific gravity]	dimensionless		
g_c Conversion constant	dimensionless	$\frac{kg \cdot m}{Ns^2}$	$\frac{32.17405\ lb_m \cdot ft}{lb_f \cdot s^2}$
g_o Standard acceleration due to gravity, 9.806650 m/s^2 [32.17405 ft/s^2]	LT^{-2}	m/s^2	ft/s^2
h Pressure loss (See $\Delta\omega$ [h])			
h_w Differential Pressure (See $\Delta p[h_w]$)			
K Uniform equivalent roughness (see Appendix A)	L	m	in
ℓ Pressure tap spacing from orifice plate	L	m	in
L Ratio of pressure tap spacing to D, $L = \ell/D$	dimensionless		
n Polytropic exponent	dimensionless		
p Static pressure of the fluid	$ML^{-1}T^{-2}$	Pa	lb_f/in^2
p_c Critical absolute pressure of a substance	$ML^{-1}T^{-2}$	Pa	lb_f/in^2
p_{f_1} Static pressure of flowing fluid at upstream pressure tap	$ML^{-1}T^{-2}$	Pa	lb_f/in^2
p_{f_2} Static pressure of flowing fluid at downstream pressure tap	$ML^{-1}T^{-2}$	Pa	lb_f/in^2
q_m Mass rate of flow $q_m = \rho_f q_v = \rho_b q_V$	MT^{-1}	kg/s	lb_m/s
q_v Volume rate of flow at flowing conditions	L^3T^{-1}	m^3/s	ft^3/s
q_V Volume rate of flow at base conditions $q_V = q_v \cdot \rho_f/\rho_b$	L^3T^{-1}	m^3/s	ft^3/s

R Radius	L	m	in
R_a *Arithmetical mean deviation from the mean line of the profile (see ISO/R 468)*	L	m	in
R_D, R_d Reynolds number referred to D or d	dimensionless		
t Temperature of the flowing fluid	θ	°C	°F
T Absolute temperature of the flowing fluid	θ	K	°R
U Mean axial velocity of the fluid in the pipe	LT^{-1}	m/s	ft/s
Y Expansion Factor (See ϵ [Y])	dimensionless		
Z Gas (vapor) compressibility factor	dimensionless		
α_P Thermal expansion factor of the pipe	θ^{-1}	m/m/°C	in/in/°F
α_{PE} Thermal expansion factor of the primary device	θ^{-1}	m/m/°C	in/in/°F
β Diameter ratio, $\beta = d/D$	dimensionless		
$\Delta p[h_w]$ Differential pressure (see note 1 below)	$ML^{-1}T^{-2}$	Pa	$(inH_2O)_{g_0,T}$
$\Delta\omega[h]$ Pressure loss	$ML^{-1}T^{-2}$	Pa	lb_f/in^2
ϵ[Y] Expansion factor	dimensionless		
$\epsilon_1[Y_1]$ Expansion factor based on upstream pressure	dimensionless		
$\epsilon_2[Y_2]$ Expansion factor based on downstream pressure	dimensionless		
κ Isentropic exponent	dimensionless		
κ_1 Isentropic exponent based on upstream measurements	dimensionless		
κ_2 Isentropic exponent based on downstream measurements	dimensionless		
κ_m Mean isentropic exponent	dimensionless		
μ Absolute viscosity of the fluid	$ML^{-1}T^{-1}$	Pa·s	$g\cdot cm/s^2$ (see Note[2])
ν Kinematic viscosity of the fluid, $\nu = \mu/\rho$	L^2T^{-1}	m^2/s	ft^2/s
ξ Relative pressure loss	dimensionless		
ρ_f *Density of flowing fluid*	ML^{-3}	kg/m^3	lb_m/ft^3
ρ_b Density of fluid at base conditions	ML^{-3}	kg/m^3	lb_m/ft^3
$\rho_{w,68°}$ Density of water at 68°F and 14.696 psia, 62.31572 lb_m/ft^3	ML^{-3}	kg/m^3	lb_m/ft^3
τ Pressure ratio, $\tau = p_2/p_1$	dimensionless		
ϕ Total angle of the divergent	dimensionless	radian	degree
Subscript $_1$ refers to the upstream conditions			
Subscript $_2$ refers to the downstream conditions			

Note [1]: In the US system of units the inH_2O is a pressure unit and is equal to the difference between the pressure at the bottom of a column of water one inch high, at a temperature of 68°F, at a standard gravity of g_0 = 32.17405, and the standard atmospheric pressure (14.696 psi) on top of the water. One inH_2O = 0.24864107 KPa.

Note [2]: In this Standard for US practice the centipoise is used for absolute viscosity and replaces the previous US unit, $lb_m/ft\ s$; $\mu_{cP} = (lb_m/ft\ s)\ (1.488164\ 10^3)$

$$\text{Also, } \mu_{cP} = \frac{(lb_f\cdot s)}{ft^2}\ \frac{(g_c\cdot 1.488164\ 10^3)}{32.17405}$$

Note [3]: In this Standard customary U.S. units (in, psia, etc) are given in the equations for the convenience of the user. They are often given in brackets [] after the SI units.

2.2 Pressure Measurement: Definitions

2.2.1 pressure tap: Hole or annular slot in a flange, fitting or the wall of a pipe, or throat of a primary device which is flush with the inside surface.

2.2.2 static pressure of a fluid flowing through a primary device: Pressure measured by connecting a pressure measuring device to a pressure tap in the plane of the differential pressure taps. (Only absolute static pressure is used in the equations presented in this Standard.)

2.2.3 differential pressure: Difference between the static pressure measured on the upstream side and on the downstream side of a primary device (or in the throat for a venturi tube or nozzle). For installations other than horizontal the lead lines must be installed in accordance with ISO 2186 or API RP550 to eliminate errors due to elevation differences between the taps.

The term "differential pressure" is applicable only if the pressure taps are in the positions specified by this Standard for each standard primary device.

2.2.4 pressure ratio: the absolute static pressure at the downstream pressure tap, divided by the upstream tap pressure ; p_{f_2}/p_{f_1}

2.3 Primary devices: Definitions

2.3.1 orifice or throat: Opening of minimum cross-sectional area in a primary element.

Standard primary element orifices are circular and coaxial with the meter run.

2.3.2 orifice plate: Thin plate in which a circular concentric hole has been machined.

Standard orifice plates are described as "thin plate" and "with sharp square edge," because the thickness of the plate is small compared with the hole diameter (bore) and because the upstream edge of the orifice is sharp and square.

2.3.3 nozzle: Primary element which consists of a convergent inlet connected to a cylindrical section generally called the "throat."

2.3.4 venturi tube: Element which consists of a cylindrical entrance section, followed by a convergent inlet connected to a cylindrical section called the "throat" an(a conical expanding section called the "divergent".

If the convergent inlet is conical, the element is called a "classical venturi tube."

2.3.5 diameter ratio of a primary element in a given pipe: The diameter of the orifice or throat of the primary element divided by the internal diameter of the measuring pipe upstream of the primary element, or of the lead-in section of a venturi.

However, when the primary element has a cylindrical section upstream, equivalent in diameter to that of the pipe (as in the case of the classical venturi tube), the diameter ratio (. ratio) is the quotient of the throat diameter divided by the diameter of this cylindrical section at the plane of the upstream pressure taps.

2.4 Flow: Definitions

2.4.1 rate of flow of fluid passing through a primary device: Mass or volume of fluid passing through the orifice or throat per unit time; in all cases it is necessary to state explicitly whether the mass rate of flow expressed in mass per time unit, or the volume rate of flow, expressed in volume per time unit, is being used.

2.4.2 Reynolds number

A dimensionless parameter used to define the flow profile condition--it expresses a ratio between inertia and viscous forces. In this Standard it is referred to:

- either the upstream condition of the fluid and the upstream diameter of the pipe, i.e.

SI units — US units

$$R_D = \frac{U_1 D}{\nu_1} = \frac{q_m}{\frac{\pi}{4}\,\mu\, D} \qquad\qquad R_D = \frac{U_1 D}{12\nu} = \frac{22738\ q_m}{\mu\ D} \qquad \text{(Eq 1)}$$

- or the orifice or throat diameter of the primary device, i.e.

$$R_d = R_D \cdot \beta^{-1} \qquad \text{(Eq 2)}$$

The value of the volume rate of flow may be substituted at flowing or base conditions to obtain the Reynolds number since:

$$q_m = q_v\, \rho_f \qquad \text{(Eq 3)}$$

$$q_m = q^{V}\, \rho_b \qquad \text{(Eq 4)}$$

2.4.3 Isentropic Exponent κ:

In the expansion of a gas or vapor through a differential producer the variation in the local fluid density introduces a 'compressible flow effect' on the flow measurement which is taken into account by the expansion factor; e.g., equations 27, 28 and 31.

The relationship between pressure and density for the expansion is assumed to be:

$$(P/\rho)^n = \text{constant} \qquad \text{(Eq a)}$$

where n is the polytropic exponent.

The isentropic exponent is a thermodynamic state property defined by:

$$\kappa = \frac{\rho}{p}\left[\frac{\delta p}{\delta\rho}\right]_s = \frac{\rho c^2}{p} = \left[\frac{c_p}{c_v}\right]\frac{\rho}{p}\left[\frac{\delta p}{\delta\rho}\right]_T \qquad \text{(Eq b)}$$

The isentropic exponent is, in general, a function of the fluid and its pressure and temperature, and can be considered a 'normalized' slope of the isentrope in the P-ρ plane or a normalized speed of sound, c. For an ideal gas (at zero pressure), eq. (b) reduces to the ratio of ideal-gas specific heats:

$$\kappa = c_p / c_v \qquad \text{(Eq c)}$$

In practice, it is sufficiently accurate to substitute the ratio of ideal-gas specific heats for the isentropic exponent, if the pressure is less than 0.25 times the critical pressure.

A comparison of equations (b) and (c) shows that the polytropic exponent n is equal to the isentropic exponent κ only if κ is constant along the isentrope. In practice, an average or 'effective' mean value of the isentropic exponent is used as the polytropic exponent.

The average, or effective, value is the sum of the value of the isentropic exponent at the high pressure and the value at the low pressure divided by 2.

It should be noted that equation (a) forms the basis for the theoretical derivation of *the expansion factor for flow nozzles and venturis [eq 32]. For low pressure* differentials ($\Delta p/p < 0.04$ $[h_w/p < 1]$) the mean isentropic exponent κ_m approaches κ_1 *at the upstream state (1). Thus, $n = \kappa_1$ can be used. For larger pressure* differentials, the arithmetic average $\kappa_m = (\kappa_1+\kappa_2)/2$ should be used to determine the polytropic exponent to be used in equations 27, 28 and 31.

Provided the ratio of $\Delta p/p \le 0.04$ ($h_w/p \le 1.0$) is maintained the real gas effect on the expansion factor may be assumed negligible and the isentropic exponent calculated by:

SI	US	
$\kappa = C_p/(C_p - 8314)$	$\kappa = C_p/(C_p - 1.986)$	(Eq 5)

2.4.4 Discharge coefficients

Calibration of standard primary devices by means of nominally incompressible fluids (liquids) shows that C, called the discharge coefficient, a dimensionless number defined by the following relation, is dependent only on the Reynolds number for a given primary device in a given installation.

SI

$$C = \frac{q_m}{\frac{\pi}{4} d^2 \sqrt{\frac{2\,\Delta p \cdot \rho_f}{1-\beta^4}}}$$

US

$$C = \frac{q_m}{0.9970190\ d^2 \sqrt{\frac{h_w \cdot \rho_f}{1-\beta^4}}}$$

where for US customary units the constant is: $\frac{\pi}{4}\left[\frac{1}{12}\right]^2\left[\frac{2g_c\ \rho_{w,68°F}}{12}\right]^{0.5} = 0.09970190$

The numerical value of C is the same for different installations whenever such installations are geometrically similar and flows are characterized by identical *Reynolds numbers. Note: $\rho_f = \rho_{f_1} = \rho_{f_2}$ for liquids.*

The equations for the numerical values of C given in this Standard were based on data determined experimentally. See the appropriate sections.

Note: In these equations the orifice or throat of the primary device is at the flowing fluid temperature.

2.4.5 Expansion factor

Calibration of a given primary device by means of a compressible fluid (gas), shows that the ratio:

SI

$$\frac{q_m}{\frac{\pi}{4} d^2 \sqrt{\frac{2 \Delta p \cdot \rho_{f1}}{1-\beta^4}}}$$

US

$$\frac{q_m}{0.9970190\ d^2 \sqrt{\frac{h_w \cdot \rho_{f1}}{1-\beta^4}}}$$

is dependent on the value of the Reynolds number as well as on the values of the differential pressure and variations in the isentropic exponent of the gas.

The method adopted for representing these variations consists in multiplying the discharge coefficient of the primary device as determined by direct liquid calibration for the same value of Reynolds number, by the expansion factor defined by the relation, for upstream taps:

SI

$$\epsilon_1 = \frac{q_m}{C \frac{\pi}{4} d^2 \sqrt{\frac{2 \Delta p \cdot \rho_{f1}}{1-\beta^4}}}$$

US

$$Y_1 = \frac{q_m}{0.9970190\ C\ d^2 \sqrt{\frac{h_w \cdot \rho_{f1}}{1-\beta^4}}} \qquad \text{(Eq 7)}$$

e_1 [Y_1] is equal to unity when the fluid is incompressible (liquid) and less than unity when the fluid is compressible (gas or vapor). It essentially corrects for density differences between pressure taps due to expansion to the lower pressure. If a downstream pressure tap is used to obtain the density, the downstream expansion factor is defined by:

SI

$$\epsilon_2 = \frac{q_m}{C \frac{\pi}{4} d^2 \sqrt{\frac{2 \Delta p \cdot \rho_{f2}}{1-\beta^4}}}$$

US

$$Y_2 = \frac{q_m}{0.9970190\ C\ d^2 \sqrt{\frac{h_w \cdot \rho_{f2}}{1-\beta^4}}} \qquad \text{(Eq 8)}$$

This method is possible because experiments show that ϵ_1[Y_1] is practically independent of Reynolds number and, for a given diameter ratio of a given primary device, only depends on the differential pressure ratio and the isentropic exponent.

The numerical values of ϵ_1[Y_1] have, for orifices, been based on experimental data and, for nozzles and venturis, been based on the thermodynamic general energy equation. (See 6.3.2.2)

$$\epsilon_2 = \epsilon_1 \sqrt{1 + \frac{\Delta p}{p_{f2}}} \qquad Y_2 = Y_1 \sqrt{1 + \frac{h_w}{27.73\ p_{f2}}} \qquad \text{(Eq 9)}$$

where ϵ_1[Y_1] is to be calculated, using:

$$\frac{\Delta p}{p_{f1}} = \frac{\Delta p}{p_{f2} + \Delta p} \qquad \frac{h_w}{27.73\ p_{f1}} = \frac{h_w}{27.73\ p_{f2} + h_w} \qquad \text{(Eq 10)}$$

2.4.6 Pipe internal roughness criterion

For this Standard a maximum roughness of 9μm [350μin] is considered adequate for all except corner taps. See table 5 for corner taps.

In practice roughness can be measured with standard equipment for machined surfaces but can only be estimated for the rougher surfaces of pipes.

3. PRINCIPLE OF MEASUREMENT AND THE METHOD OF COMPUTATION

3.1 Principle of measurement

The principle of measurement is based on the installation of a primary element (such as an orifice plate, a nozzle or a venturi tube) into a pipe line in which a flowing fluid fills the pipe. The primary element causes a static pressure difference between upstream and throat or downstream side of the element. The flow rate can be determined from the measured value of this pressure difference and from a knowledge of the characteristics of the flowing fluid as well as the circumstances under which the element is being used. It is assumed that the element is geometrically and fluid dynamically similar to one on which calibration has been made and that the conditions of use are the same; i.e., that it is in accordance with this Standard. To ensure the uncertainty the element must be calibrated if geometric and fluid dynamic similarity do not exist.

Calibration of flow meters can be done using a range of techniques for both liquid and gas flows. These include:

- gravimetric and timing, (for example, see ISO 4185)
- volumetric and timing, (for example, see ISO 4373, ISO/DIS 8316)
- comparison to master meters or other transfer standards.

These can be carried out in laboratory testing facilities, or more preferably, via "in-situ" tests. (The uncertainty statements of this Standard do not apply.)

The mass rate of flow can be determined since it is related to the pressure differential within the uncertainty stated in this Standard, by the following formulas:

SI	US	
$q_m = \frac{\pi}{4} C\varepsilon_1 d^2 \sqrt{\frac{2\Delta p \cdot \rho_{f1}}{1-\beta^4}}$	$q_m = 0.09970190\ CY_1 d^2 \sqrt{\frac{h_w \cdot \rho_{f1}}{1-\beta^4}}$	(Eq 11)

or for a downstream measurement:

$q_m = \frac{\pi}{4} C\varepsilon_2 d^2 \sqrt{\frac{2\Delta p \cdot \rho_{f2}}{1-\beta^4}}$	$q_m = 0.09970190\ CY_2 d^2 \sqrt{\frac{h_w \cdot \rho_{f2}}{1-\beta^4}}$	(Eq 12)

Note: For liquids ε_1[Y_1] and ε_2[Y_2] are 1.0, when d is at the flowing temperature. The value of the volume rate of flow at flowing or base conditions (a selected base pressure and temperature), can be calculated since:

$$q_v = q_m / \rho_f \qquad \text{(Eq 13)}$$

$$q_V = q_m / \rho_b \qquad \text{(Eq 14)}$$

3.2 Method of sizing the bore of the selected Primary Element

If it is necessary to have a closely specified ΔP [h_w] for a given flow rate, the bore *diameter*, d, must be calculated carefully, using Eq 15(b) as shown below. If, on the other hand, flow calculation need only have accurate values for d and D, *under* flowing conditions a ±1% value for β (and therefore d) can be obtained by using the value of 1.0 for both F_a and ϵ[Y] and *0.6 for C for orifices* in Eq 15(b) with n=1. Measured values for d and D can then be used, corrected as in Eq 16a and 16b, for the flow calculation.

In the accurate sizing of the bore of any primary element it is necessary to use an iterative procedure because the discharge *coefficient*, C, the expansion factor ϵ [Y], and the effect of temperature (thermal expansion) on D and d are not initially known. These values are all β (d/D) ratio dependent. It is therefore necessary to iterate for β and then solve for the measured bore at a reference temperature (*20°C or 68°F*). By substituting the relationship $\beta^2 D^2 = d^2$, Eq 11 may be rearranged to equate known design factors to the β-dependent *functions* for iteration as:

SI

$$F_a C \epsilon_1 \frac{\beta^2}{\sqrt{1-\beta^4}} = \frac{q_m}{\frac{\pi}{4} D^2_{meas} \sqrt{2\Delta p \cdot \rho_{f1}}}$$

US

$$F_a C Y_1 \frac{\beta^2}{\sqrt{1-\beta^4}} = \frac{q_m}{0.09970190\, D^2_{meas} \sqrt{h_w \cdot \rho_{f1}}} \qquad \text{(Eq 15a)}$$

where all of the terms on *the right hand* side of the equation are known and constant at the design conditions.

F_a is the thermal expansion correction factor:

$$F_a = \left\{ 1 + \frac{2}{1-\beta^4_{meas}} \left[\alpha_{pE} - \beta^4_{meas}\, \alpha_p \right] \left[t - t_{meas} \right] \right\}$$

The Miller (iterative) form of Eq 15a is:

$$\beta_n = \left\{ 1 + \left[\frac{F_{a(n-1)}\ \epsilon_{1(n-1)} \left[Y_{1(n-1)} \right] C_{(n-1)}}{[\text{right side of Eq 15a}]} \right]^2 \right\}^{-0.25} \qquad \text{(Eq 15b)}$$

The bore is then $d_{meas} = \beta_{calc}\, D_{meas}$

Once d has been determined the actual measured *dimensions of* d and D should be used to calculate the flow rate as shown in clause 3.3.

3.3 Computation of flow rate

a) Except for Venturi tubes, within the limits of this standard, C is dependent on R_D, which is itself dependent on q_m. In such cases, the final value of C, and hence of q_m, is to be obtained by iteration from an initial chosen value of C (or R_D). Generally it may be convenient to adopt *the value* of C at a Reynolds number selected at 80% of the maximum flow of the system being considered.

b) Δp represents the differential pressure, as defined under 2.2.3.

c) d,D, and β in the formulae, are the values at flowing conditions; measurements taken at other conditions shall be corrected for any possible expansion or contraction of the primary device and the pipe due to the values o fluid temperature during the measurement.

It must be assumed that the primary device is at the same temperature as the pipe. The D and d values at any flowing temperature can be calculated with the equations:

$$d = [1 + \alpha_{pE} (t - t_{meas})] d_{meas} \qquad \text{(Eq 16a)}$$

$$D = [1 + \alpha_{p} (t - t_{meas})] D_{meas} \qquad \text{(Eq 16b)}$$

The true β can be calculated by the ratio of d/D from these equations.

These values are then used to calculate the discharge coefficient C, and the gas expansion factor ϵ [Y] for use in the equations 11 or 12 to calculate the flow rate for any ΔP, knowing the density, ρ, and viscosity, ν, at the flowing conditions.

d) If, however the flow rate (Eq 11 or 12) is to be calculated using the values of d_{meas} and D_{meas}, a single combined correction factor may be calculated as:

$$d^2 = F_a d^2_{meas} = \left\{1 + \frac{2}{1-\beta^4} \left[\alpha_{pE} - \beta^4 \alpha_p\right]\left[t - t_{meas}\right]\right\} d^2_{meas} \qquad \text{(Eq 17)}$$

Normally the temperature t_{meas} is assumed to be 20°C [68°F]

e) It is necessary to know the density and the viscosity of the fluid under the conditions of the flow measurement.

3.4 Determination of gas (vapor) density

The density of the gas (vapor) is required to be known at either the plane of the upstream pressure tap or the plane of the downstream tap; it can either be measured directly with a densitometer or calculated from the fluid properties and equations of state. A useful relation using ideal specific gravity is:

SI	US	
$\rho = 3.483407 \cdot 10^{-3} \frac{pG_i}{ZT}$	$\rho = 2.698825 \frac{pG_i}{ZT}$	(Eq 18)

where compressability Z is found in tables, calculated from one of the equations of state, or determined through generalized diagrams.

For calculating the density of a gas or vapor at base conditions (ρ_b) the base temperature, pressure and the compressibility factors are substituted into the above equations.

3.4.1 The static pressure of the fluid shall be measured in the radial plane of the upstream or the downstream pressure tap, by means of a separate pressure tap or by connecting in common with the differential pressure measurement (see 6.2.1 for description of tap holes) or by means of a carrier ring taps (as described in 6.2.4). Flow in or out of the pressure measurement line may cause an error in the differential pressure measurement. Separate taps may reduce this error if it is occurring.

3.4.2 Although the temperature of the fluid from which the density and viscosity can be determined is preferably the one in the upstream pressure tap plane, a well or protrusion located there would introduce errors. It may be assumed that the downstream and upstream temperatures are the same providing, for gas, that $p_2/p_1 \geq 0.75$, and therefore the temperature of the fluid shall be measured downstream of the primary device. The thermometer well shall restrict the flow a minimum while still providing an accurate measure of the temperature of the flowing fluid. The distance between it and the primary device shall be at least equal to 5 D and a maximum of 15 D.

3.4.3 Any method of determining reliable values of the pressure, temperature, viscosity and density of the fluid is acceptable providing the locations of pockets, wells, protrusions, etc. are within the requirements of this Standard, and do not interfere with the distribution of the flow. (See Section 6.).

4. GENERAL MEASUREMENT REQUIREMENTS

4.1 Primary device

4.1.1 The primary device shall be manufactured, installed and used in accordance with this Standard.

When the manufacturing characteristics and conditions of use of the primary devices are outside the limits given in this Standard, it is necessary to calibrate the primary device under, as nearly as practical, the actual conditions of use. After the calibration, additional uncertainties may be calculated only insofar as this Standard is followed. If this Standard is not followed no guidance can be given.

4.1.2 In order to avoid greater uncertainties than those given in this Standard, it is recommended that a primary device used for continuous flow measurement be visually checked periodically, more often if inspection shows the edge sharpness, surface roughness or plate flatness has changed enough to lack conformity with this Standard.

4.1.3 The coefficient of thermal expansion of the material used in the primary device (α_{PE}) and of the pipe (α_P) must be known if flowing temperature is different from that at which the diameters were measured.(See 3.3c and d)

4.2 Type of fluid

4.2.1 The fluid may be either compressible (gas) or considered as incompressible (liquid).

4.2.2 The fluid shall for all practical purposes be physically and thermally homogeneous and of single phase through the primary device.

4.2.3 The density and viscosity of the fluid at the flowing conditions must be known; see ¶ 3.4 for determination of density, knowing the pressure and temperature.

4.3 Flow conditions

4.3.1 The rate of flow shall be constant or, in practice, vary only slightly and slowly with time. This Standard does not provide for the measurement of pulsating flow.

Note: See ISO Technical Report 3313 "Measurement of pulsating fluid flow by means of orifice plates, nozzles or venturi tubes, in particular in the case of sinusoidal or square wave intermittent periodic type fluctuation."

4.3.2 The uncertainties specified in this standard are valid only when there is no change of phase through the primary device. If liquid vaporization is experienced in the primary element, it may be corrected by increasing the static pressure or by reducing the temperature. Relocating the flow control valve to downstream of the device may be a possible solution. If condensation is occurring with compressible fluid flow, the static pressure should be reduced or the temperature increased, or both. To predict whether or not there is a phase change, the flow computation shall be carried out on the assumption that the expansion is isothermal if the fluid is a liquid, or isentropic if the fluid is a gas (because the temperature of the transition is so critical).

4.3.3 If the fluid is a gas, the pressure ratio P_{f_2}/P_{f_1} as defined in § 2.2.4 shall be equal to or greater than 0.75.

5. INSTALLATION REQUIREMENTS

5.1 General

5.1.1 The method of measurement applies only to fluids flowing through a pipe line of circular cross-section.

5.1.2 The pipe shall run full at the measuring section.

5.1.3 The primary device shall be installed in the pipe line at a position such that the flow conditions immediately upstream approach those of a fully developed profile and are free from swirl (see 5.6). Such conditions may be expected to exist if the installation conforms to requirements given in this section 5.

5.1.4 The primary device shall be installed between two sections of straight cylindrical pipe in which there is no obstruction or branch connection (whether or not there is flow into or out of such connections during measurement) other than those specified in this Standard.

The pipe is considered straight when it appears to be reasonably so by visual inspection. (See ANSI B36.10 and ASTM A530). The required minimum straight lengths of pipe, which conform to the description above, vary according to the piping arrangement, the type of primary device and the diameter ratio. They are specified in table 2 for orifice plates and Table 6 for venturies..

5.1.5 The value for the pipe diameter D to be used in the computation of the diameter ratio shall be the arithmetic mean of measurements made on at least four equally separated diameters in the plane of the upstream tap. (See 3.3c) The upstream pipe is said to be circular and cylindrical when no diameter in any plane differs by more than 0.25% from the arithmetic mean of measurements made on at least four diameters, distributed in each of at least three cross-sections themselves distributed over a length of 0.5 D. Two of these cross sections shall be at distances 0D and 0.5 D from the upstream tap, and one being in the plane of the weld in the case of a welded neck construction even when the weld is farther upstream than 0.5D. Care should be taken not to make a D measurement at a ring gap or a gasket.

No diameter of the downstream straight length measured over a length of at least 2 D downstream of the primary element, shall differ from the mean diameter of the upstream straight length by more than ±0.5%.

5.2.2 Location of primary device and rings

5.2.2.1 The primary device shall be placed in the pipe in such a way that the fluid flows from the upstream face towards the *downstream face* (see the arrow "flow direction" on the appropriate figure).

5.2.2.2 The orifice plate shall be perpendicular to the center-line of the pipe to within ± 1°.

5.2.3.3 Eccentricity The eccentricity of the orifice bore d to the upstream pipe bore D can result in a positive bias error in the discharge coefficient. In line sizes *greater than nominal 100mm [4in] the following equation for maximum eccentricity* towards the measuring taps can be used to maintain the coefficient uncertainty given in this Standard:

$$e_x \le \frac{0.0025\ D}{0.1 + 2.3\beta^4} \qquad \text{(Eq 19)}$$

In line sizes of nominal *75mm [3in] or less, an eccentricity* towards the taps should be no greater than 0.8mm [0.03in]. For eccentricities in other directions, away from the taps, an eccentricity of 1.5% D may be allowed.

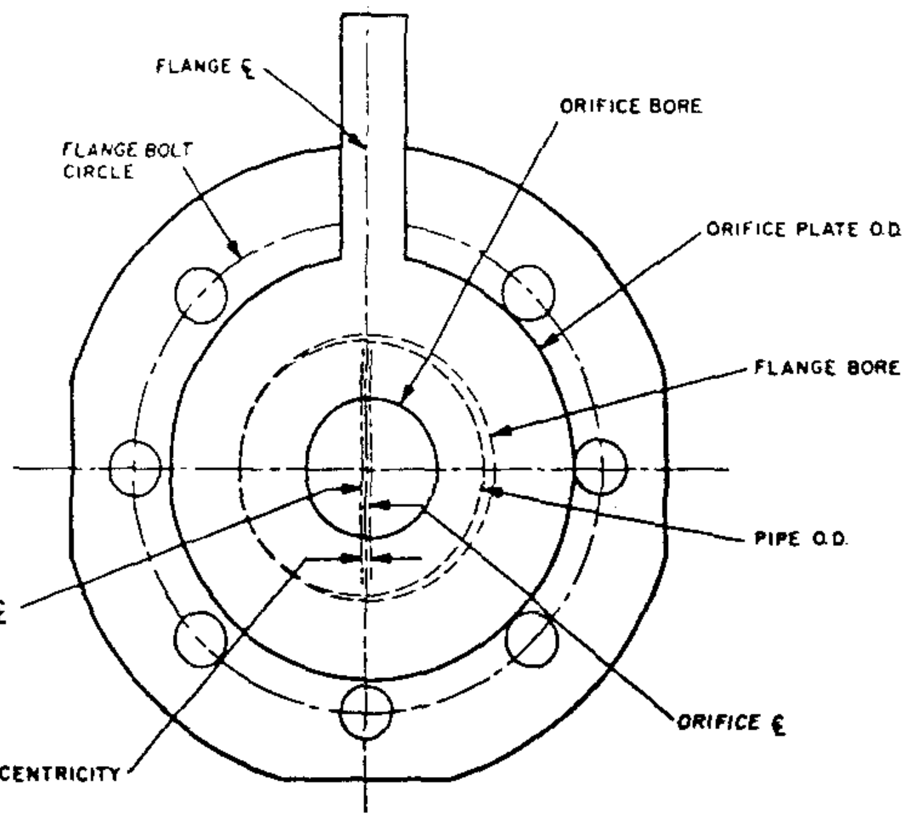


Figure 1, Eccentricity of installed orifice

5.2.4 Assembly, rings, and gaskets

5.2.4.1 When seal rings for orifices or carrier rings for corner tap orifices are used, they shall be so centered that at no point do they protrude into the pipe.

5.2.4.2 Gaskets or sealing rings, if used, shall be made and inserted in such a way that they do not protrude at any point inside the pipe or across the pressure tap or slot when corner taps are used. Their compressed thickness shall be used in determining the location of the pressure taps for orifice installations.

5.2.4.3 If gaskets are used between the primary device and the annular chamber rings, they shall not protrude inside the annular chamber.

5.2.4.4 The maximum allowable recess or gap preceding or following the orifice plate, in the orifice flange, ring type joints, carrier rings, or between the end of the pipe and the plate, the length of which, measured parallel to the axis of the pipe, is 13mm [0.5in] for flange or D&D/2 taps.

5.3 The upstream lengths of pipe given in this Standard for given uncertainties are based on data taken in 1927 and analyzed in the 1930's. These uncertainties are to be treated as bias error limits as outlined in MFC-2. Some additional data taken in recent years have indicated that these lengths may not be long enough. Additional data will improve the confidence in the required lengths and will be used in a revision as soon as it is available. Until then, for important flow measurements it is recommended to:

1) Always use lengths as much longer than specified as practical, and
2) Use flow conditioners, or
3) Calibrated in-situ or in a calibration installation as nearly as possible identical to the planned flow measurement

5.4 Upstream and downstream straight lengths for installation between various fittings and the primary device

5.4.1 The straight lengths listed in Table 2 &6 are the minimum acceptable lengths and may be subject to an additional uncertainty of up to 0.5% on the discharge coefficient, except as noted.

5.4.2 When the straight lengths are equal to or longer than twice the values given in Table 2 &6 no additional uncertainty is indicated.

5.4.3 When either the upstream or the downstream straight lengths are shorter than the values given in table 2 &6 this Standard gives no information by which to predict the value of any further uncertainty to be taken into account.

5.4.4 The valves mentioned in table 2 &6 shall be fully open. It is recommended that control of the rate of flow be effected by valves located downstream of the primary device. Isolating valves located upstream shall be preferably of the gate or ball type, full bore, and shall be fully open.

5.4.5 After a single change of direction (bend or tee), it is recommended that if pairs of single taps are used they be installed so that their axes will be perpendicular to the plane of the bend or tee.

5.1.6 The pipe bore shall be circular over the entire minimum length of straight pipe required. Beyond D length upstream, the bore is taken to be circular if it appears to be reasonably so by visual inspection. (See ANSI B36.10 and ASTM A530). The circularity of the outside of the pipe may be taken as a guide, except in the immediate vicinity of the primary element where special requirements shall apply according to the type of primary element used (see 5.2.1 and 5.3.1).

5.1.7 The internal diameter D of the measuring pipe shall comply with the values given for each type of primary device.

5.1.8 The inside surface of the measuring pipe shall visually appear to be clean, free from pitting, deposit and encrustations for at least a length of 10 D upstream and 4 D downstream of the primary element. The maximum roughness shall be 9µm [350µin].

Grooves, scoring, pits, ridges resulting from seams, distortion caused by welding, offsets etc. (regardless of the size of such irregularities) which effect the inside diameter at such points by more than the tolerance,

$$\frac{\delta D}{D} = \frac{\pm 0.0021}{0.1 + 2.3\beta^4}$$

shall not be permitted. When this value is exceeded, any method may be used to correct the irregularities.

5.1.9 To ensure the proper operation of the primary device, the pipe flanges and the measuring section shall be insulated when large temperature gradients are present.

5.2 Specific installation requirements for orifice plates

5.2.1 Circularity of the pipe

In the immediate vicinity of the primary device the following requirements shall apply:

5.2.1.1 The length of the upstream pipe adjacent to the primary element (or to the carrier ring if there is one) shall be at least 2 D and circular and cylindrical.

Beyond 2 D from the primary element, the upstream pipe between the primary device and the first upstream fitting or disturbance can be made up of one or more sections of pipe.

No additional uncertainty in the discharge coefficient is involved provided that the step between any two sections within 10 D upstream does not exceed the requirement for cylindricality of 0.25% as defined in 5.1.5.

If the step is greater than the limit given in the paragraph above, the installation is not in accordance with this Standard.

TABLE 2 - Recommended straight lengths for nozzles and orifice plates For 0.5% additional uncertainty

Note the limitations given above in ¶ 5.4.1, 5.4.2, 5.4.3, 5.4.4, and 5.4.5.

All straight lengths are expressed as multiples of the diameter D. They shall be measured from the upstream face of the primary device.

Interpolation for intermediate β values is to be used.

The pipe roughness, at least over the length indicated in table 2 shall not exceed maximum roughness of 9µm [350µin].

Upstream (inlet) of the primary device Columns C and C′ are for flow conditioners. See 5.5 and Fig. 2.												Down-stream
β	Single 90° bend or tee (flow from one branch only)	Two or more 90° bends in the same plane 3)			Two or more 90° bends in different planes 3)			Reducer (2D to D over a length of *1.5 D* to 3 D)	Expander (0.5D to D over a length of *1D to 2D*)	Globe Valve Fully *Open* 2)	Full Bore Ball or Gate *Valve* Fully Open	All Fittings Included In this *Table*
			C′	C		C′	C					
0.20	6	7	3.5	5	17	4.5	5	5[1)]	8	6	6	2
0.25	6	7	3.5	5	17	4.5	5	5[1)]	8	6	6	2
0.30	6	8	3.5	5	17	4.5	5	5[1)]	8	6	6	2.5
0.35	6	8	3.5	5	18	4.5	5	5[1)]	8	6	6	2.5
0.40	7	9	3.5	5	18	4.5	5	5[1)]	8	10	6	3
0.45	7	9	4	5	19	5	5	5[1)]	9	12	6	3
0.50	7	10	4	5	20	5	5	5	9	15	6	3
0.55	8	11	4	5	22	5.5	5.5	5	10	18	7	3
0.60	9	13	4.5	5.5	24	6	5.5	5	11	22	7	3.5
0.65	11	16	5	6	27	6.5	6	6	13	25	8	3.5
0.70	14	18	5.5	6.5	31	7	6.5	7	15	25	10	3.5
0.75	18	21	6	7	35	8	7	11	19	25	12	4

	Fittings	Minimum upstream straight length required
For all β values	Abrupt symmetrical reduction having a diameter ratio ≥ 0.5	15

Note 1) *These lengths require no additional uncertainty, but the uncertainties* **for shorter lengths are not well enough known to be given in this Standard.**

Note 2) These lengths require no additional uncertainty.

Note 3) The insertion of 5 to 10D straight lengths between the two bends is sufficient to make the combined effect the same as the single bends in the left column

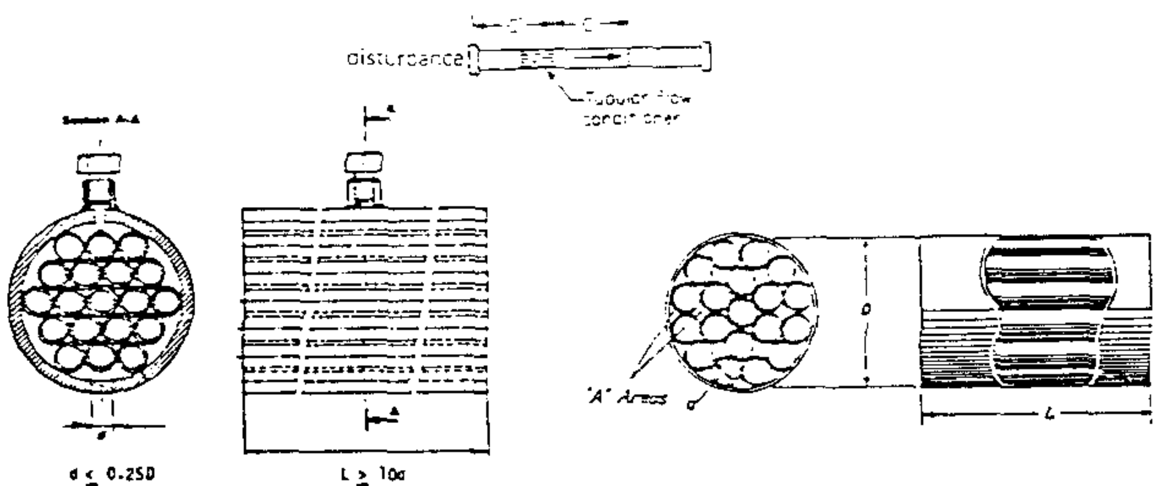
5.5 Flow conditioning devices

NOTE: Since many upstream piping and fitting configurations will result in a swirl, and swirl may take 50 to 100 D straight lengths to die out by itself, it is recommended that a flow conditioner be used whenever possible on important flow measuring installations. Flow conditioners are used to enable the use of shorter upstream lengths.

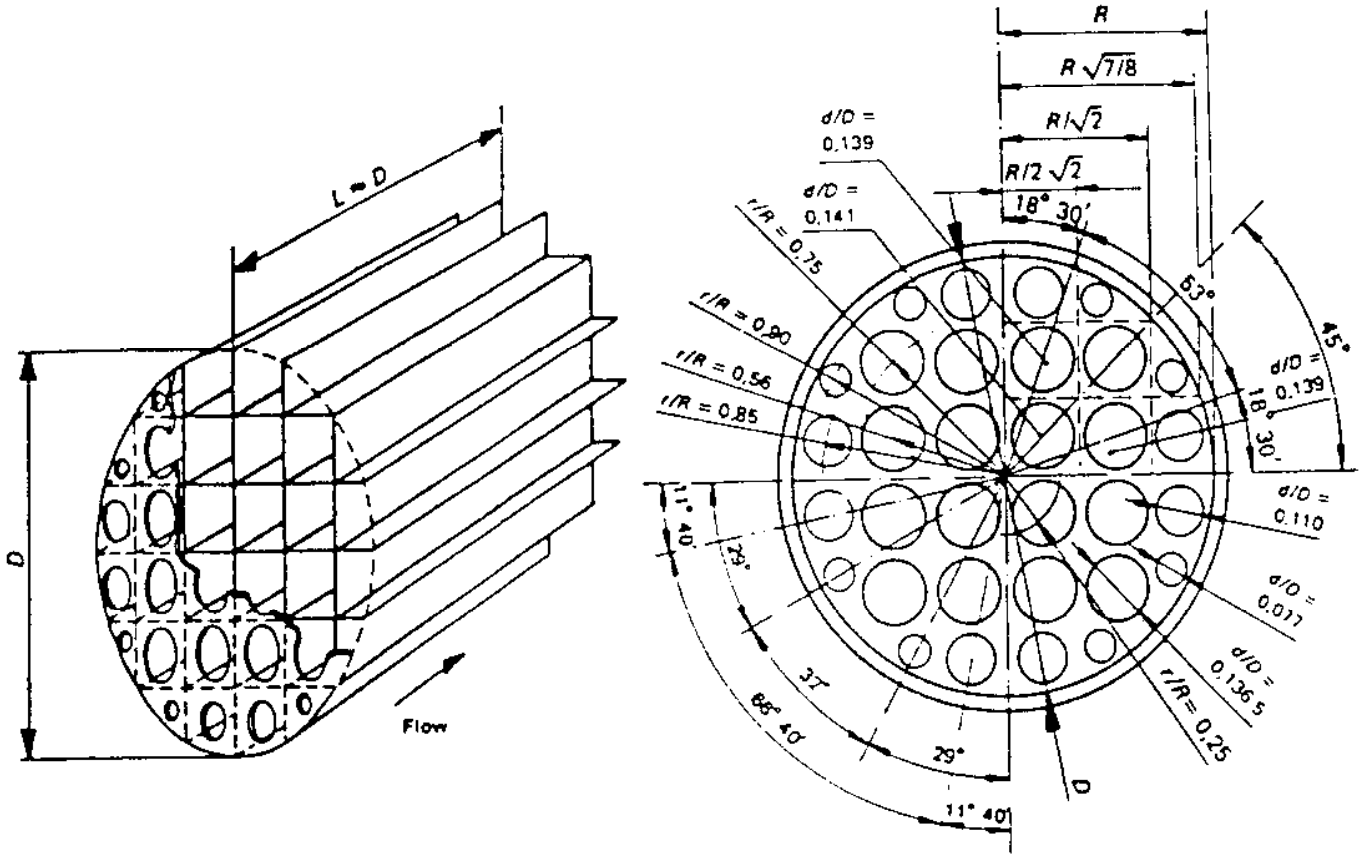
Flow conditioning devices of the types described below will provide installations with zero additional uncertainty providing the straight pipe length between the disturbance and the conditioner, and the straight pipe length between the conditioner and the primary element, are equal to at least the lengths shown in Table 2. See ¶ 5.1.3. and ¶ 5.6. The upstream lengths (C') and downstream lengths (C) are given for the configuration for which the effects have been sufficiently documented. The effectiveness of flow conditioners, for the reduction of required upstream lengths of pipe, given in this Standard for given uncertainties, are based on data taken in the 1920's and 1930's. (These uncertainties are to be treated as bias limit errors as covered by ANSI/ASME MFC-2M). Some additional data taken in recent years have indicated that these lengths may not be long enough. Additional data will improve the *confidence in the lengths required with conditioners. Until then, for important flow* measurements, it is recommended to:

1) Always use lengths as much longer than specified as practical, and
2) Use flow conditioners, or
   3) Calibrated in-situ or in a calibration installation as nearly as possible identical to the planned flow measurement

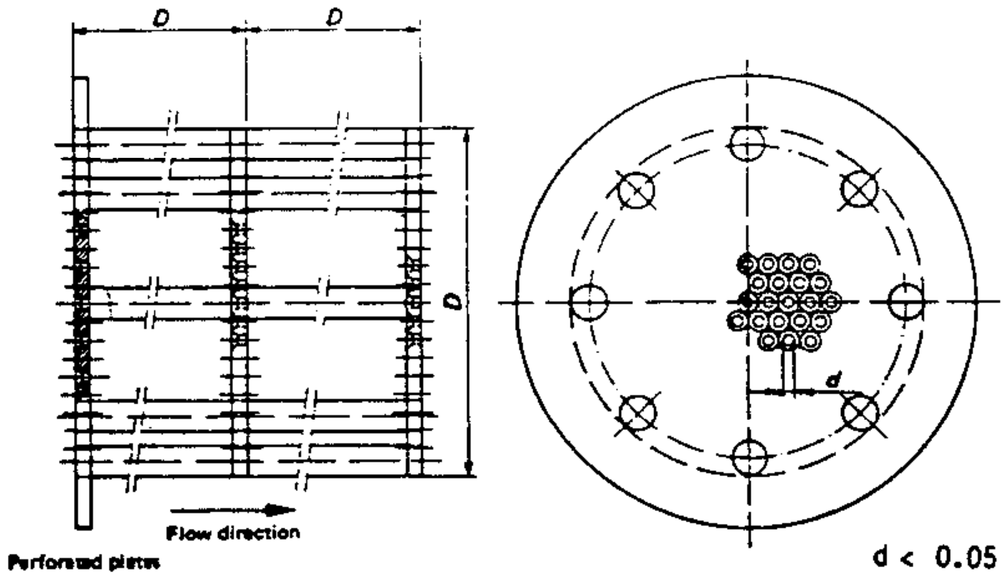
FIGURE 2-Conditioners



Type A Tube Bundle Flow Conditioner



Type B: Zanker "Straightener"



d < 0.05 D

Type C: Sprenkle "Straightener"

5.5.1 Types of conditioning devices

The three standardized types of conditioners, A, B and C are shown in figure 2. It should be noted that these devices create approximately the following pressure losses:

SI | US

In the case of Types A & B:

$$\Delta\omega = 5 \frac{\rho_f U^2}{2} = 4.053 \frac{q_m^2}{D^4 \rho_f} \qquad h = 0.002996\, \rho_f U^2 = 100.7 \frac{q_m^2}{D^4 \rho_f} \qquad \text{(Eq 20)}$$

In the case of Type C:
with unbevelled upstream holes,

$$\Delta\omega = 14 \frac{\rho_f U^2}{2} = 11.35 \frac{q_m^2}{D^4 \rho_f} \qquad h = 0.00839\, \rho_f U^2 = 281.96 \frac{q_m^2}{D^4 \rho_f} \qquad \text{(Eq 21)}$$

with beveled upstream holes,

$$\Delta\omega = 11 \frac{\rho_f U^2}{2} = 8.917 \frac{q_m^2}{D^4 \rho_f} \qquad h = 0.006591\, \rho_f U^2 = 221.5 \frac{q_m^2}{D^4 \rho_f} \qquad \text{(Eq 22)}$$

5.5.1.1 *Type A: Tube Bundle Conditioner*

This type of conditioner consists of 19 parallel tubes fixed together and held rigidly in the pipe. It is important in this case to ensure that the various tubes are parallel with each other and with the pipe axis since, if this requirement is not complied with, the conditioner itself might introduce disturbances into the flow. Both ends of all tubes shall be machined with an internal taper that makes the end wall as thin as practical.

In the construction of the assembly the maximum internal diameter of a 'vane' tube shall not exceed 1/4 the pipe diameter and the free area outside the tubes shall not exceed 1/16 the cross sectional area of the pipe. The overall length of the conditioner shall be at least 10 times the tube internal diameter. See Fig 2.

5.5.1.2 Type B: ZANKER "Straightener"

This conditioner consists of a perforated plate with holes of certain specified sizes followed by a number of channels (one for each hole) formed by the intersection of a number of plates. The important dimensions are given in figure 2.

The various plates should be chosen of minimum thickness and yet provide adequate strength.

5.5.1.3 Type C: SPRENKLE "Straightener"

This conditioner consists of three perforated plates in series with a length equal to one pipe diameter between successive plates. The perforations should preferably be chamfered on the upstream side, and the total area of the holes in each plate should be greater than 40% of the cross sectional area of the pipe. The ratio of plate thickness to hole diameter shall be at least 1.0 and the diameter of the holes shall be less than 1/20 of the pipe diameter.

The three plates shall be held together by bars or studs, which shall be located around the periphery of the pipe bore, and which shall be of as small a diameter as possible, consistent with providing the required strength.

5.5.1.4 Other types of conditioners are subject to agreement between the buyer and seller.

5.6 General requirement for flow conditions at the primary device.

If the prescribed installation conditions given in table 2 or in 5.5 cannot be met this Standard still remains applicable if the flow conditions immediately upstream of the primary device conform to 5.1.3.

Swirl free conditions can be taken to exist when the swirl angle over the pipe is less than 2° when measured with a directional detecting pitot tube or other suitable way.

Acceptable velocity profile conditions can be presumed to exist when at each point across the pipe cross-section the ratio of the local axial velocity to the maximum axial velocity at the cross-section agrees to within ±5% with that which would exist in swirl-free flow at the same radial position at a cross-section located at the end of a very long straight length (over 100 D) of similar pipe.

6. ORIFICE PLATES AND PRESSURE TAPS

The various types of standard orifice plate installations are similar and are defined by the arrangement of the pressure taps.

Pressure tap locations shall be as described in 6.2.3

6.1 Description

The axial plane cross-section of the plate is shown in figure 3.

The letters shown in figure 3 are for reference purposes in the following text.

6.1.1 General shape

6.1.1.1 Unless otherwise stated, the remainder of 6.1 applies only to that part of the plate intended to be located within the pipe.

6.1.1.2 Orifice plates with vent or drain holes through them are not covered by this Standard.

6.1.2 Upstream and downstream faces (Fig 3.)

6.1.2.1 The upstream face of the plate A shall be flat. It is considered as such when the maximum gap between it and a straight edge of length D laid across it anywhere is less than 0.01 (D-d)/2. It is assumed that the orifice plate mounting does not significantly distort the plate.

6.1.2.2 The upstream face of the orifice plate shall have a maximum roughness of 1.3μm [50μin] within a circle whose diameter is not less than D and is concentric with the bore.

6.1.2.3 It is useful to provide a distinctive mark which is visible even when the orifice plate is installed to show that the upstream face of the orifice plate is correctly installed relative to the direction of flow.

6.1.2.4 The surface flatness and smoothness of the downstream face can be judged by visual inspection.

6.1.2.5 It is unnecessary to provide the same high quality of surface finish for the downstream face as for the upstream face.

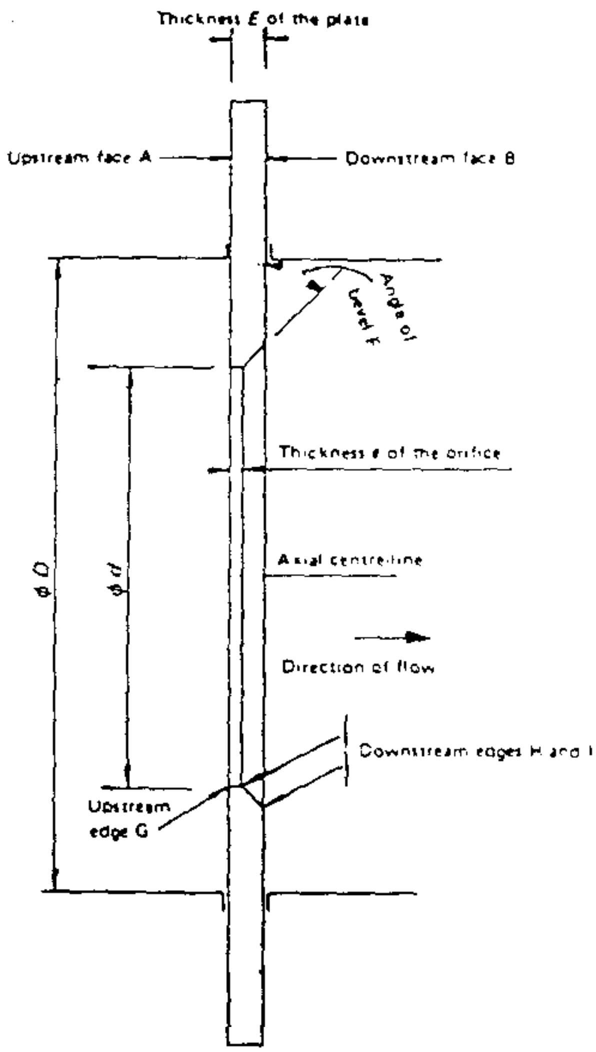


Figure 3, Standard Orifice Plate

6.1.3 Plate thickness E and edge thickness e

6.1.3.1 The minimum edge thickness e of the orifice shall be equal to or greater than 0.01d but not less than 0.125mm[0.005in]. The maximum shall be equal to or less than 0.02D or equal to or less than 0.125d whichever is smaller, but not greater than E.

6.1.3.2 The values of e measured at any point on the orifice shall not differ among themselves by more than 0.001 D.

TABLE 3 MINIMUM[1] ORIFICE PLATE THICKNESS

Δp $[h_w]$	Nominal pipe sizes 50mm≤D≤150mm 2in ≤D≤ 6in	150mm≤D≤250mm 6in ≤D≤ 10in	250mm≤D≤500mm 10in ≤D≤ 20in	500mm≤D≤900mm 20in ≤D≤ 36in
$\beta \le 0.5$				
250 kPa 1005 inH_2O	3 mm 0.120 in	5 mm 0.183 in	10 mm 0.370 in	13 mm 0.495 in
50kPa 201 inH_2O	3 mm 0.120 in	3 mm 0.120 in	6 mm 0.245 in	10 mm 0.370 in
25 kPa 100 inH_2O	3 mm 0.120 in	3 mm 0.120 in	6 mm 0.245 in	10 mm 0.370 in
$\beta > 0.5$				
250 kPa 1005 inH_2O	3 mm 0.120 in	5 mm 0.183 in	10 mm 0.370 in	13 mm 0.495 in
50kPa 201 inH_2O	3 mm 0.120 in	3 mm 0.120 in	5 mm 0.183 in	10 mm 0.370 in
25 kPa 100 inH_2O	3 mm 0.120 in	3 mm 0.120 in	5 mm 0.183 in	6 mm 0.245 in

Note(1): See ¶ 6.1.3.3 for maximum thicknesses.

6.1.3.3 The thickness E of the plate shall be the minimum given in Table 3, and a maximum of 1.5 times that value, but not greater than 13 mm [0.5 in].

6.1.3.4 The values of E measured at any point of the plate shall not differ among themselves by more than 0.001 D.

6.1.4 Angle of bevel F

6.1.4.1 If the thickness E of the plate exceeds the thickness e of the orifice, the plate shall be bevelled on the downstream side. The bevelled surface shall be smooth.

6.1.4.2 The angle of bevel F shall be approximately 45°. (see Fig 3).

6.1.5 Edges G, H and I

6.1.5.1 The upstream edge G and the downstream edges H and I shall have neither wire-edges, nor burrs, nor, in general, any peculiarities visible to the unaided eye.

6.1.5.2 The upstream edge G shall be sharp. If d ≥ 25mm [1 in] this requirement may be considered as satisfied by visual inspection, checking that the edge does not seem to reflect a beam of light when viewed with an unaided eye.

It is considered so if the edge radius is not greater than 0.0004 d up to d = 75mm [3in], and 0.025mm [0.001in] above that, as measured by the lead foil or other suitable method.

If $d \leq$ 25mm [1 in] visual inspection may not be sufficient.

If there is any doubt as to whether this requirement is satisfied, the edge radius must actually be measured.

6.1.6 Diameter of orifice d

6.1.6.1 Because of the uncertainty of the discharge coefficient, tighter restrictions on eccentricity, edge sharpness effects, and increased upstream straight pipe lengths, the user is advised to remain below a β of 0.7 and above a β of 0.2. (See clause 6.3.3.1)

6.1.6.2 The value d of the diameter of the orifice shall be taken as the mean of the measurements of at least four diameters at approximately equal angular spacing, correcting for thermal expansion. (See 3.3c).

No diameter shall differ by more than 0.05% from the mean for d = 25mm [1in] and larger and 0.01mm [0.0004in] for d down to 10mm [0.40 in].

6.1.6.3 The orifice shall be cylindrical and perpendicular to the upstream face.

6.1.7 BI-DIRECTIONAL PLATES

6.1.7.1 If the orifice plate is intended to be used for measuring reverse flows,
the plate shall not be bevelled;
the two faces shall be as specified for the upstream face in 6.1.2;
the thickness E of the plate shall be equal to the thickness e of the orifice as described in 6.1.3 (between 0.005D and 0.02D);
the two edges of the orifice shall be as specified for the upstream edge in 6.1.5.

6.1.7.2 For orifice plates with D and D/2 taps, at least two sets of upstream and downstream pressure taps must be provided and used appropriately for the direction of flow, (see 6.2).

6.1.8 MATERIAL AND MANUFACTURE

6.1.8.1 The plate can be manufactured of any material and in any way, provided it is and remains in accordance with the foregoing description during the flow measurements.

6.2 DIFFERENTIAL PRESSURE TAPS

At least one upstream pressure tap and one downstream pressure tap shall be provided for each primary device, installed in one of the recommended standard positions.

NOTE: Although there is not enough data to make quantitative statements, there is good evidence that connecting two or more taps equally spaced around the periphery can materially reduce the effects of eccentricity, non-uniform flow profile, pulsating flow, etc. Annular chambers are often used for the inter-connection. Care must be taken to avoid vapor condensation or liquid vaporization in the external lead lines.

A single plate can be used with several sets of pressure taps suitable for different types of standard orifice plates, but to avoid mutual interference, several taps on the one side of the orifice plate shall not be in the same axial plane.

6.2.1 Shape and diameters of pressure taps for flange and D & D/2 orifices.

6.2.1.1 The centerline of the taps shall meet the pipe centerline and be at right angles to it ($\pm 2^{\circ}$).

6.2.1.2 At the point of break-through the hole shall be circular. The edges shall be flush with the internal surface of the pipe wall and be as sharp as possible. To ensure the elimination of all burrs or wire edges at the inner edge, rounding shall be permitted but shall be kept as small as possible and where it can be measured its radius shall be less than 0.0625d. No irregularity shall appear inside the connecting hole, on the edges of the hole drilled in the pipe wall, or in the pipe wall close to the pressure tap.

6.2.1.3 Conformity of the pressure taps with the foregoing 2 subsections can be judged by visual inspection.

6.2.1.4 The recommended maximum diameter of the tap holes through the pipe wall or flange are given in table 4. Interpolate for sizes in between. Upstream and downstream tap holes must be the same diameter.

Table 4 Recommended Maximum Diameters of Pressure Tap Holes for flange and D&D/2 taps

Nominal Inside Pipe Diameter D	Maximum Diameter of tap holes
50 to 75mm [2 to 3in]	10mm [0.375in]
D >100mm [4in]	13mm [0.5in]

The minimum size of tap holes shall be 6mm [0.25in]

6.2.1.5 The pressure tap holes shall be circular and cylindrical. These holes may increase in diameter at any location away from the inner wall. If, however, they are decreased, this decrease may not occur for at least 2.5 hole diameters away from the pipe inner wall.

6.2.2 Angular position of pressure taps for flange and D & D/2 orifices

6.2.2.1 The axis of the upstream tap and that of the downstream tap may be located in different axial planes. (see 5.4.5).

6.2.2.2 However, attention is called to the fact that, in all cases, the reading of differential pressure obtained by these pressure taps shall be in accordance with the definition of 2.2.3.

6.2.3 <u>SPACING OF PRESSURE TAPS</u>

6.2.3.1 The spacing (ℓ) of a pressure tap is the distance between the centerline of the pressure tap and the plane of one specified face of the orifice plate. When installing the pressure taps, due account has to be taken of the thickness of the gaskets and/or sealing material, which are to be used.

6.2.3.2 The location of the pressure taps with respect to the orifice plate defines the type of standard installation; flange, D and D/2, corner.

6.2.3.3 The location of corner taps is described in 6.2.4.

6.2.3.4 <u>ORIFICE PLATES WITH FLANGE TAPS (see fig 4)</u>

The spacing ℓ_1 of the upstream tap is nominally 25.4 mm [1 in] and is measured from the UPSTREAM face of the orifice plate. The spacing ℓ_2' of the downstream pressure tap is nominally 25.4 mm [1 in] and is measured from the DOWNSTREAM face of the orifice plate.

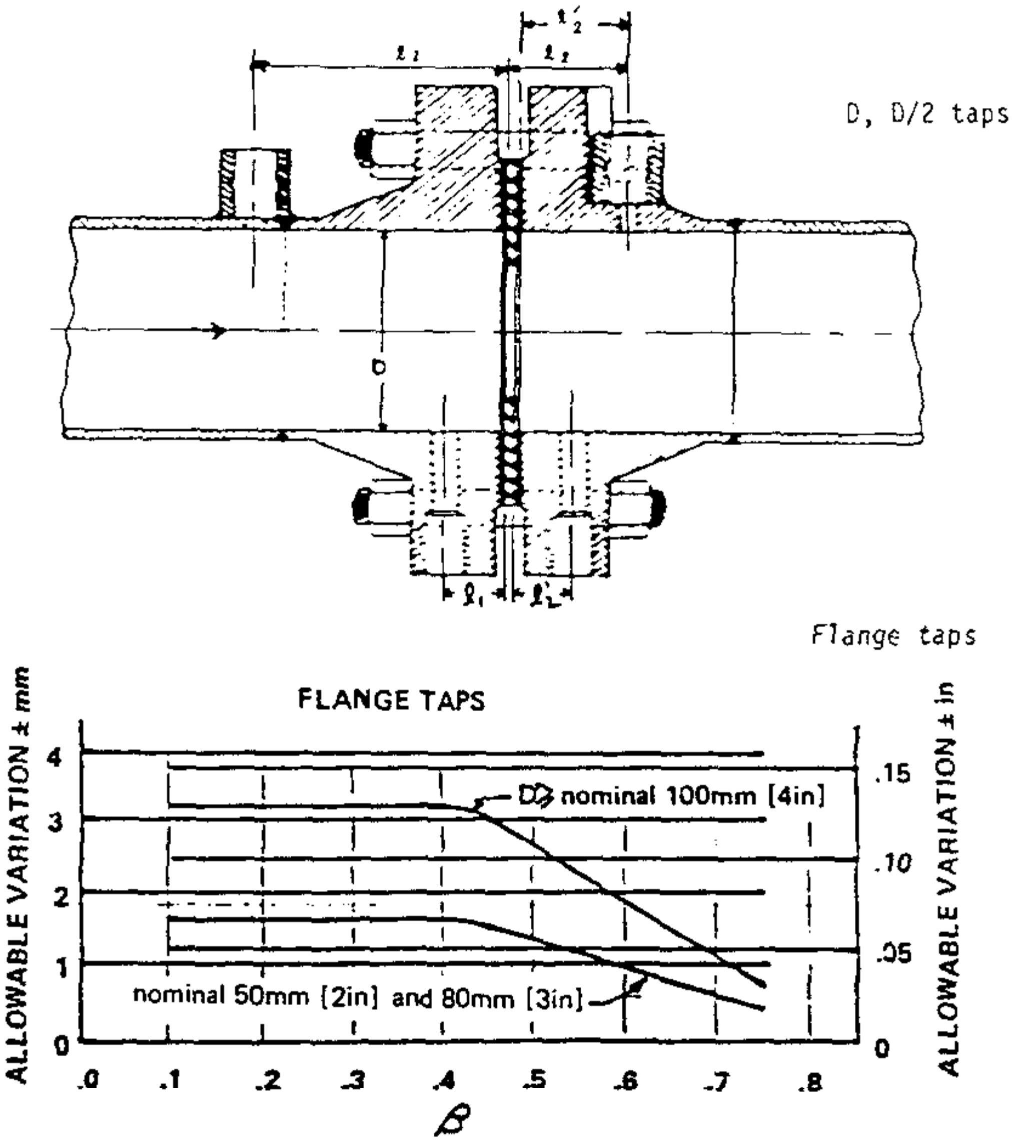


Figure 4: Location for Orifice Flange and D & D/2 Pressure Taps

6.2.3.5 ORIFICE PLATES WITH D AND D/2 TAPS (see fig 4)

Both ℓ_1 and ℓ_2 spacings are measured from the UPSTREAM face of the orifice plate.

The spacing ℓ_1 of the upstream pressure tap is nominally equal to D, but may be D±5% without modification of the flow coefficient.

The spacing ℓ_2 of the downstream pressure tap is nominally equal to 0.5D, but may be 0.5 D ± 1mm [0.040in] without modification of the discharge coefficient:

[for the purposes of this Standard $\ell_2'=(\ell_2-E)=0.47D$, where ℓ_2' is the distance from the downstream face of the plate to the center of the tap hole].

6.2.4 *ORIFICE PLATE WITH CORNER TAPS* (see fig 5)

6.2.4.1 The spacing between the centerlines of the taps and the respective faces of the plate is to be selected so that the tap holes break through the wall flush with the faces of the plate(see 6.2.4.5).

6.2.4.2 The taps may be either single taps or annular slots. Both types of taps can be located either in the pipe, its flanges, or in carrier rings as shown in figures 4 and 5.

6.2.4.3 The diameter (a) of single taps or the width (j) of annular slots are given below. The minimum diameter is determined in practice by the likelihood of accidental blockage and satisfactory performance.

Clean fluids and steam

For $\beta \le 0.65$: $0.005\ D \le a$ or $j \le 0.03\ D$

For $\beta > 0.65$: $0.01\ D \le a$ or $j \le 0.02\ D$

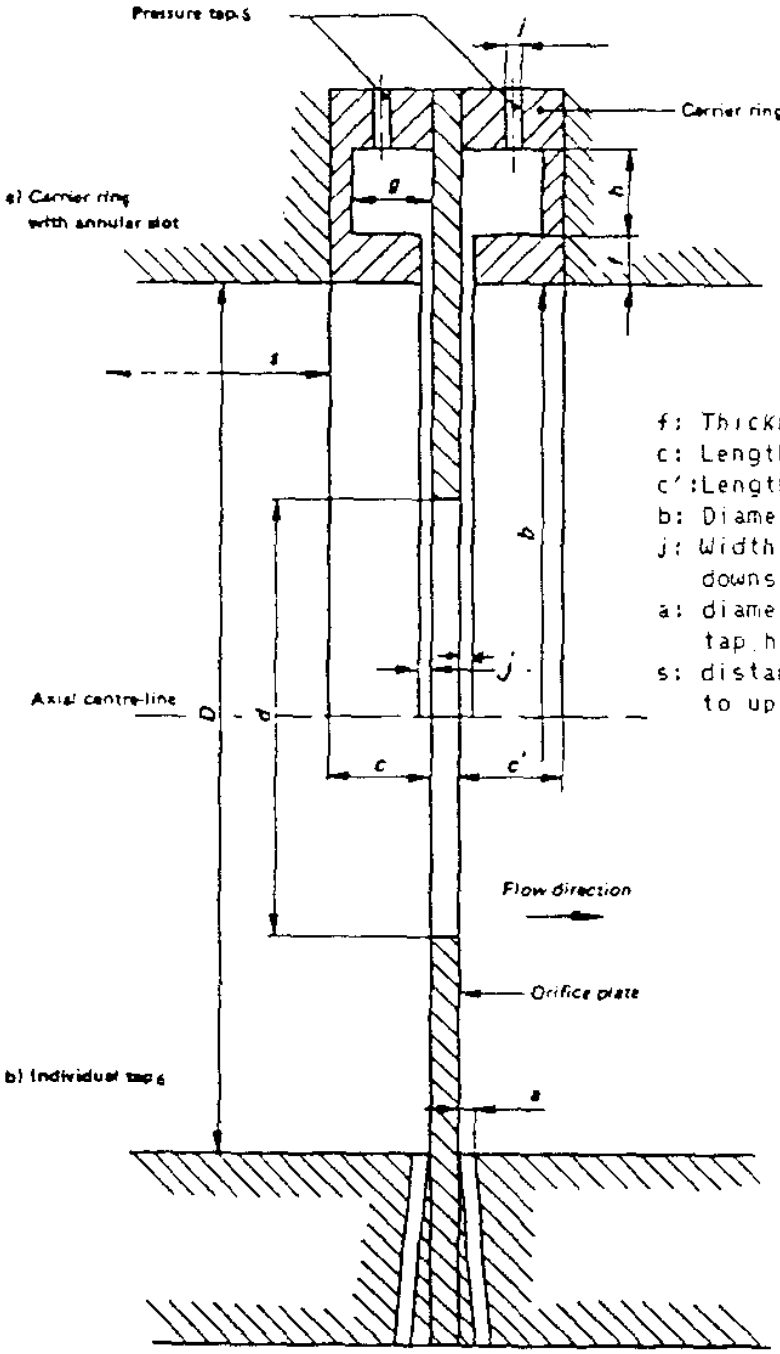
For any values of β

For clean fluids: 1mm [0.05in] ≤ a or j ≤ 10mm [0.5 in]

For steam with annular chambers: 1mm [0.05in] ≤ a or j ≤ 10mm [0.5in]

For steam and for liquefied gases with single taps:
4mm [0.16in] ≤ a or j ≤ 10mm [0.5in]

6.2.4.4 The annular slots usually break through the pipe over the entire perimeter, with no break in continuity. If not, each chamber shall connect with the inside of the pipe by at least four openings, the axes of which are at equal angles to one another and the individual opening area of each is at least $12mm^2$ [$0.02in^2$].



f: Thickness of the slot
c: Length of upstream ring
c':Length of downstream ring
b: Diameter of the ring
j: Width of the upstream and downstream annular slots
a: diameter of individual tap holes.
s: distance from carrier ring to upstream step

FIGURE 5, Corner Taps

6.2.4.5 If individual pressure taps, as shown in figure 5b, lower portion, are used, the center-line of the taps shall cross the center-line of the pipe at as near a right angle (90°) as possible.

If there are several individual pressure taps for the same upstream or downstream axial plane, their center-lines shall form equal angles with each other, around the pipe. The diameters of individual pressure taps are given in 6.2.4.3.

The pressure taps shall be circular and cylindrical over a length at least 2.5 times the diameter of the taps, measured from the inner wall of the pipe.

6.2.4.6 The inner diameter b of the carrier rings must be equal to or greater than the diameter D of the pipe to ensure that they do not protrude into the pipe. The inner diameter must not be greater than 1.0025D.

See ¶ 5.1.5, ¶ 5.2.1.2 and ¶ 5.2.1.3 for allowable step.

The lengths c and c′ of the upstream and downstream rings (figure 5) shall not be greater than 0.5 D.

The length f shall be greater than or equal to twice the width a of the annular slot. The area of the cross section of the annular chamber g x h shall be greater than or equal to half the total area of the opening connecting this chamber to the inside of the pipe.

6.2.4.7 All surfaces of the ring which can be in contact with the measured fluid shall be clean and have a good machined finish.

6.2.4.8 The pressure taps connecting the annular chambers to the secondary devices are pipe-wall taps, circular at the point of breakthrough and with diameters (j) between 4 and 10mm [0.15 and 0.5in] (See ¶ 6.2.1.5)

6.2.4.9 The upstream and downstream carrier rings are not necessarily symmetrical in relation to each other, but they shall both comply with the foregoing specifications.

6.2.4.10 The diameter of the pipe to be used for the calculation of the diameter ratio, and hence the flow-rate, is to be measured as defined in ¶ 5.1.5, the carrier ring being regarded as part of the primary device. The mean diameter of the carrier ring b shall be used in the calculation. This also applies to the length requirement given in ¶ 5.2.1.1 so that the length s is to be taken from the upstream edge of the recess formed by the carrier ring.

6.3 Coefficients and corresponding uncertainties for orifice plates.

6.3.1 Limits of use, roughness (for corner taps) shall conform to the values in table 5.

TABLE 5 - Upper limits of relative roughness of the upstream pipe for Corner tap orifice plates

β	≤0.3	0.32	0.34	0.36	0.38	0.4	0.45	0.5	0.6	0.7
Corner taps 10^4 k/D	25	18.1	12.9	10.0	8.3	7.1	5.6	4.9	4.2	4.0

The value of K, uniform equivalent roughness, expressed in length units, depends on several factors such as height, distribution, angularity and other geometric aspects of the roughness elements at the pipe wall.

A full scale pressure loss test of a sample length of the particular pipe should be carried out to determine the value of K satisfactorily.

However, approximate values of k for different materials can be obtained from the various tables given in reference literature, and appendix A gives values of k for a variety of materials, as derived from the Colebrook formula.

Most of the experiments on which the values of C given for corner taps in the present Standard are based, were carried out in pipes with a relative roughness $k/D \leq 3.8 \times 10^{-4}$

Pipes with higher relative roughness may be used if the roughness for at least 10 D upstream of the orifice plate is within a maximum roughness of 9µm [350µin] (or the limits given in table 5 for corner taps).

6.3.2 Coefficients

6.3.2.1 DISCHARGE COEFFICIENT

The discharge coefficient C is given by the equation: (Eq 23)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 0.0900 L_1\,\beta^{4}(1-\beta^{4})^{-1} - 0.0337\,L_2'\beta^{3} + 91.71\beta^{2.5}\,R_D^{-0.75}$$

where:

L_1 = dimensionless correction for upstream tap location = $\ell_1 D^{-1}$, measured from upstream face.

L_2 = " " " downstream " " = $\ell_2 D^{-1}$, " " " " .

L_2' = " " " " " " = $(\ell_2 - E)D^{-1}$, " " downstream " .

FOR SI UNITS, d′ and D′ in millimeters

For Corner Taps: $L_1 = L_2' = 0$ (Eq 24)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{3} + 91.71\,\beta^{2.5}\,R_D^{-0.75}$$

For Flange Taps: $(D' \geq = 58.6\ \text{mm})\ L_1 = L_2' = 25.4\ D^{-1}$ (Eq 25a)

$$C = 0.5959 + 0.0312\beta^{2.1} - 0.1840\beta^{8} + 2.2860(D')^{-1}\beta^{4}(1-\beta^{4})^{-1} - 0.8560(D')^{-1}\beta^{3} + 91.71\beta^{2.5}R_D^{-0.75}$$

For Flange Taps: $(50.8 \leq D' \leq 58.6)$mm, $L_1 = 0.4333$, $L_2' = 25.4\ (D')^{-1}$ (Eq 25b)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 0.0390\,\beta^{4}(1-\beta^{4})^{-1} - 0.8560(D')^{-1}\beta^{3} + 91.71\beta^{2.5}\,R_D^{-0.75}$$

For D & D/2 Taps: $L_1 = 0.4333$, $L_2' = 0.47$ (Eq 26)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 0.0390\,\beta^{4}(1-\beta^{4})^{-1} - 0.01584\,\beta 3 + 91.71\,\beta^{2.5}\,R_D^{-0.75}$$

FOR US UNITS, d and D in inches.

For Corner Taps: $L_1 = L_2' = 0$ (Eq 24)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 91.71\,\beta^{2.5}\,R_D^{-0.75}$$

For Flange Taps: $(D \geq 2.3$ inches$)$, $L_1 = L_2'$ (Eq 25a)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 0.0900\,D^{-1}\beta^{4}(1-\beta^{4})^{-1} - 0.0337\,D^{-1}\beta^{3} + 91.71\,\beta^{2.5}\,R_D^{-0.75}$$

For Flange Taps: $(2 < D < 2.3)$ inches $L_1 = 0.4333$, $L_2' = D^{-1}$ (Eq 25b)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 0.0390\,\beta^{4}(1-\beta^{4})^{-1} - 0.0337\,D^{-1}\beta^{3} + 91.71\,\beta^{2.5}\,R_D^{-0.75}$$

For D & D/2 Taps: $L_1 = 0.4333$, $L_2' = 0.47$ (Eq 26)

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^{8} + 0.0390\,\beta^{4}(1-\beta^{4})^{-1} - 0.01584\,\beta^{3} + 91.71\,\beta^{2.5}\,R_D^{-0.75}$$

This formula is to be used only for tap arrangements defined in sub-sections 6.2.3.4, 6.2.3.5, or 6.2.4. In particular, it is not permitted to enter into the equation pairs of values of ℓ_1 and ℓ_2' which do not match one of the three standardized tap arrangements.

This formula is valid, and so are the uncertainties, as given hereunder, when the measurement matches all of the limitations stated in 6.3.1, and the general installation requirements as stated in section 5.

6.3.2.2 EXPANSION FACTOR

For the three tap arrangements, the empirical formulas for computing the expansion factors are as follows:

$$\epsilon_1 = 1 - (0.41 + 0.35\beta^4)\ \Delta p/\kappa p_1 \qquad Y_1 = 1-(0.41 + 0.35\beta^4)\ \frac{h_w}{27.73\kappa p_1} \qquad \text{(Eq 27)}$$

$$\epsilon_2=\sqrt{1+\frac{\Delta p}{p_2}} -(0.41+0.35\beta^4)\frac{\Delta p}{\kappa p_2\sqrt{1+\Delta p/p_2}} \qquad Y_2=\sqrt{1+\frac{h_w}{27.73p_2}} -(0.41+0.35\beta^4)\frac{h_w}{27.73\kappa p_2\sqrt{1+\frac{h_w}{27.73p_2}}} \qquad \text{(Eq 28)}$$

where subscript 1 indicates upstream and subscript 2 indicates downstream.

This formula is applicable only within the range of the limits of use given in clauses *6.3.1* and *6.3.3.1*.

Test results for the determination of ϵ [Y] are known for air, steam, and natural gas only. However, there is no known objection to using the same formula for other gases and vapours for which the isentropic exponent is known, or can be calculated.

However the formula is applicable only if $p_2/p_1 \geq 0.75$

6.3.3 Uncertainties

6.3.3.1 UNCERTAINTY OF DISCHARGE COEFFICIENT

Any equations may be used for calculating the discharge coefficient providing they give results that agree with the equations in 6.3.2.1 above within the uncertainties given below.

The percentage uncertainty (Bias limits) of the value of C (beyond any known error in β, D, R_D, and κ/D) is equal to:

For 50mm ≤ D ≤ 900mm [2in ≤ D ≤ 36in] (Nominal sizes)

β =	0.2	0.6	0.75	
$10000 < R_D \leq 10^8$	←(0.6)%→		←(β)%→	See example below.
$2000 \leq R_D \leq 10000$	←(0.6 + β)%→			

The uncertainties given here are only valid for the Equations 23 through 26.

Example: The uncertainty, for R_D = 12 000 and β = 0.53, is 0.6%,
and for β = 0.74, is 0.74%

NOTE: See 5.4.1, 5.4.2, and 5.4.3 for additional information.

6.3.3.2 UNCERTAINTY OF EXPANSION FACTOR

When β, $\Delta p/p$ [h_w/p], and κ are assumed to be known without error, the percentage uncertainty of the value of ϵ [Y] is equal to:

SI	US
$4\ \Delta p/p$ %	$0.144\ h_w/p$ %

6.4 Pressure loss $\Delta\omega$ [h]

The pressure loss, $\Delta\omega$ [h], for the orifice plates described in this Standard is approximately related to the differential pressure Δp [h_w] by the equation,

$$\Delta\omega = \frac{\sqrt{1-\beta^4} - C\beta^2}{\sqrt{1-\beta^4} + C\beta^2}\,\Delta p \qquad\qquad h = \frac{\sqrt{1-\beta^4} - C\beta^2}{\sqrt{1-\beta^4} + C\beta^2}\,h_w \quad \text{(Eq 29)}$$

This pressure loss is the difference in static pressure between a wall pressure measured on the upstream side of the primary device, where the influence of the approach impact pressure adjacent to the plate becomes negligible (approximately 1D upstream of the primary device) and that measured on the downstream side of the device where the static pressure recovery by expansion of the jet may be considered as just completed (approximately 6D downstream of the primary device).

7. ASME FLOW NOZZLES

There are three (3) types of long radius style ASME Flow Nozzles covered by the Standard. These are:

- ASME High beta ratio nozzle ($0.50 \le \beta \le 0.80$)
- ASME Low beta ratio nozzle ($0.20 \le \beta < 0.50$)
- ASME Throat Tap flow nozzle ($0.25 \le \beta < 0.50$)

ISA (now International Standards Organization ISO) 1932 flow nozzles and Venturi *Nozzles generally are not manufactured* in the United States and have not been included in this Standard. Information on these designs are provided in ISO 5167 or ASME Fluid Meters.

While this Standard covers details of the ASME throat tap nozzle, the user is directed to ANSI/ASME PTC 6 for additional information as to construction, use and inspection of the ASME throat tap nozzle.

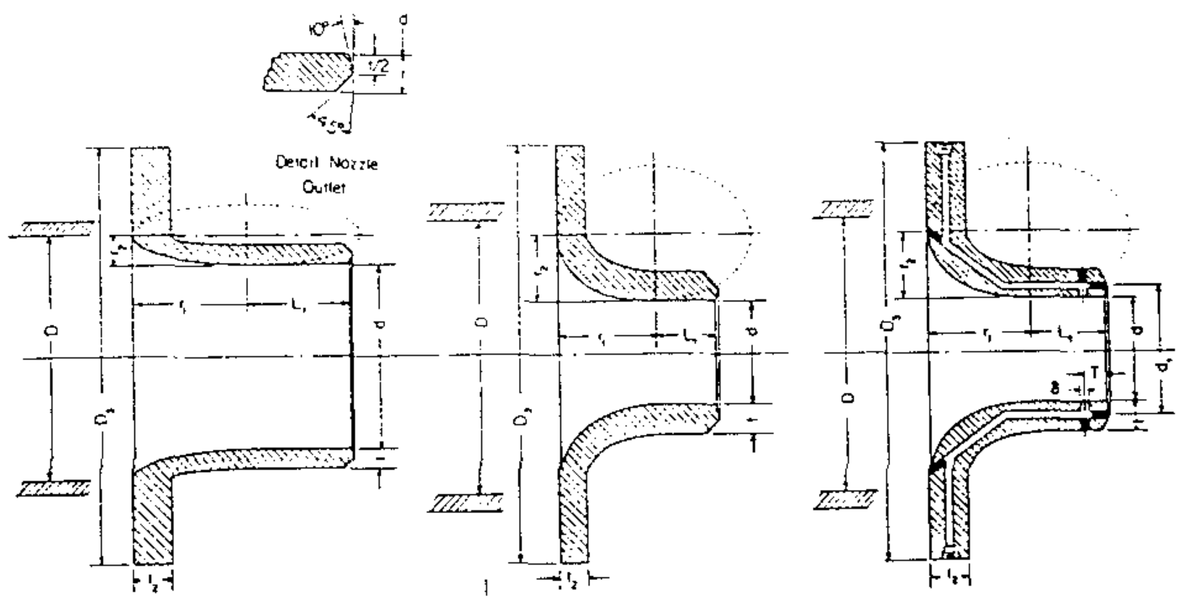
7.1 RECOMMENDED PROPORTIONS FOR ASME NOZZLES

Figure 6 illustrates the proportions of each of the three types of ASME flow nozzles with respect to throat and pipe inside diameter.

7.1.1 Entrance Section - All ASME flow nozzles covered by this Standard are long radius style nozzles which have the shape of a quarter ellipse in the entrance section. The value of the major axis and the minor axis of the ellipse are shown in Figure 6 for each type of flow nozzle. The major center line of the ellipse shall be parallel to the centerline of the nozzle within one tenth (0.1%) percent. The ellipse shall terminate at a point no greater than D regardless of the value of the minor axis. The profile of the ellipse may be checked by means of a template.

7.1.2 Throat Section - The throat section shall have a diameter (d) and a length as shown in Figure 6. The value (d) shall be the average of four equally spaced (approximately 45 degree) measurements of the throat diameter taken in each of three equally spaced intervals along the length of the throat section and covering at least 3/4 of the throat length. The throat shall be as cylindrical as possible. No diameter shall differ by more than 0.05% from the average diameter d. Under no circumstances shall the throat diameter increase toward the nozzle exit end. A decrease in diameter (d) toward the exit end is acceptable if within the 0.05% variation allowed from the average diameter (d).

7.1.3 Exit End Section - The exit end section shall be as shown in Figure 6.



High β Nozzle

$0.50 \le \beta \le 0.80$

$r_1 = D/2$

$r_2 = (D-d)/2$

$L_1 \le 0.6d$ or $\le D/3$

$2t \le D-(d+6mm[1/4in])$

$3mm[1/8in] \le t_2 \le 0.15D$

Low β Nozzle

$0.20 \le \beta < 0.50$

$r_1 = d$

$5/8d \le r_2 \le 2/3d$

$0.6d \le L_1 \le 3/4d$

$3mm[1/8in] \le t \le 12mm[1/2in]$

$3mm[1/8in] \le t_2 \le 0.15D$

Low β Nozzle, with throat taps

$.25 \le \beta < 0.50$

$r_1 = d$

$5/8d \le r_2 \le 2/3d$

$L_t = 3/4d$

$d_t = 5/4d$

$t = 1/4d$

$t_2 = 38mm[3/2in]$

$3mm[1/8in] \le \delta \le 6mm[1/4in]$

$T = 1/4d$

Figure 6, ASME NOZZLES

7.1.4 General Requirements for the ASME flow nozzles -

7.1.4.1 The distance from the pipe inside diameter and the outside diameter of the nozzle throat shall be greater than or equal to 3mm[0.125in].

7.1.4.2 It is recommended that a shoulder for centering of the nozzle assembly in the pipe be provided. If this shoulder is provided, it should be no larger in outside diameter than D-0.060D and should be no longer than t_2 x 2. In no case shall the centering shoulder cover any part of the downstream tap.

7.1.4.3 The thickness 't' shall be sufficient to prevent distortion of the nozzle throat from the strains of machining or installation.

7.1.4.4 The surface of the inner face of the nozzle shall be polished or machined smooth and shall have a maximum roughness of 0.8μm.[32μin]. The exit end must not have rounding or burrs.

7.1.4.5 The downstream (outside) face of the nozzle shall be cylindrical and machined smooth or otherwise constructed so as to eliminate any pockets or pits which might retain debris or matter which may be found in the flowing media.

7.1.4.6 ASME long radius nozzles may be made from any suitable material provided that the material does not wear easily and the nozzle remains dimensionally stable.

7.2 PRESSURE TAP REQUIREMENTS

7.2.1 General Requirements - ASME long radius flow nozzles shall use taps which conform to the requirements of ¶ 6.2.1 and ¶ 6.2.2 herein.

7.2.2 Upstream Tap - The upstream tap(s) shall be located in the pipe wall at a distance D (+0.2D,-0.1D) from the plane of the inlet face of the nozzle.

7.2.3 Downstream Tap - Throat tap nozzles shall have tap(s) as shown in Figure 6. Nozzles without throat tap(s) shall be used with wall tap(s) located at 0.5D (± 0.01D) from the plane of the inlet face of the nozzle. (Under certain installation geometries this specification places the tap downstream of the nozzle, which is not permitted.) Under no circumstances may any part of the downstream tap be located downstream of the plane of the nozzle exit end.

7.3 INSTALLATION REQUIREMENTS

7.3.1 Pipe - ASME flow nozzles which are used in accordance with this Standard shall be used with pipe conforming to 5.2.0. In addition to the requirements of 5.2.0, the pipe internal surface roughness should not exceed 8µm[300µin] over an area of 4 D preceding and 2 D following the plane of the inlet face of the nozzle. If boring and/or honing (machining) is required, such machining should extend for a distance of at least 4 D upstream and 2 D downstream of the plane of the inlet face of the nozzle. The machined portion shall be tapered into the unmachined portion of the pipe at an included angle of less than 30 degrees. The depth of the machining should be the minimum required to obtain the surface finish. The machined inside diameter (D) of the pipe should be uniform throughout the machined length ±0.25%. All machining should be accomplished after all necessary welding of flanges, pressure taps or other welded attachments has been accomplished.

7.3.2 Flanged installation - ASME nozzles shown in Figure 6 are designed to be installed between raised face pipe flanges. Nozzles may also be used with other style of flanges if such use does not interfere with the flowing media.

7.3.3 Installation without flanges - ASME nozzles may also be installed directly in pipe conforming to this Standard by welding or pinning the nozzle to the pipe inside diameter. If such a method is used, care should be taken to ensure against any protrusions into the flowing media upstream or downstream of the nozzle.

7.3.4 Centering - The nozzle shall be manufactured so that the clearance between the nozzle shoulder and the pipe inside diameter shall be uniformly greater than 0.8mm[0.030in] of the pipe into which it is installed.

7.3.5 Straight Piping Lengths - The upstream and downstream straight piping requirements for ASME nozzles are the same as given in 5.4 herein for orifice plates.

7.3.6 Flow Conditioners - The flow conditioners as described in 5.5 herein may be used with ASME flow nozzles within the limitations and expected results as given.

7.4 ASME FLOW NOZZLE COEFFICIENTS

7.4.1 Wall Tap Nozzle Discharge Coefficients - When used in accordance with this Standard, the coefficient of discharge of ASME flow nozzles with wall taps is as given by the following formula:

$$C = 0.9975 - 0.00653\ (10^{6}/R_d)^{0.5} \qquad \text{(Eq. 30)}$$

Theory suggests that the exponent should go to 0.2 for R_d greater than 10^6, but the difference is so small that it is difficult to confirm experimentally.

Table B1 in Appendix B gives the results of sample calculations for coefficients of discharge at $R_D = 10^5$ using Eq. 30, for checking purposes.

7.4.2 Throat Tap Nozzle Discharge Coefficients - When used in accordance with this Standard, the uncalibrated coefficient of discharge of ASME flow nozzles with throat taps is as given in the ASME Performance Test Codes 6 and 19.5 (latest edition).

7.4.3 Expansion Factor $\varepsilon[Y]$ - The expansion factor, ε [Y], is calculated, when using upstream pressure values, by means of the following formula:

$$e_1 [Y_1] = \left[\left(\frac{\kappa \tau^{\frac{2}{\kappa}}}{\kappa - 1} \right) \left(\frac{1-\beta^4}{1-\beta^4 \tau^{\frac{2}{\kappa}}} \right) \left(\frac{1-\tau^{\frac{\kappa-1}{\kappa}}}{1-\tau} \right) \right]^{0.5} \qquad \text{(Eq 31)}$$

For ε_2 $[Y_2]$ using downstream pressure values, substitute:

$$e_2[Y_2] = (1 + \Delta p/p_2)^{0.5} \ \varepsilon_1[Y_1] \qquad \text{(Eq. 32)}$$

Eq. 31 and 32 are only applicable for values of β, D and R_D as given in 7.4.4 herein. Eq. 31 and 32 have been verified by test results for air, steam and natural gas only; however, there is no known objection to using the same formula for other gases and vapors for which the isentropic exponent is known.

Appendix E shows the results of sample calculations for the expansion factor, ε [Y], for a range of pressure ratios and beta ratios for use in assisting the user in verifying his calculation procedures.

7.4.4 Limits of use for the coefficients listed in 7.4.1 and 7.4.2 shall only be in accordance with this Standard when:

100 mm [4in] $\leq$ D $\leq$ 750 mm [30in]

β limits according to Clause 7

$10^4 \leq R_D \leq 6 \times 10^6$

$P_2/P_1 \geq 0.75$

7.5 UNCERTAINTIES

7.5.1 Uncertainty of discharge coefficient (C) - When β and R_D are assumed to be known without error, the percentage uncertainty of the value of C for ASME flow nozzles is as shown below, when the nozzle is made and installed in accordance with this Standard and the nozzle is used within the limits of 7.4.4

TABLE 7 - UNCERTAINTY OF DISCHARGE COEFFICIENTS

NOZZLE TYPE	UNCERTAINTY OF C
- ASME High beta ratio nozzle with wall taps	2.0%
- ASME Low beta ratio nozzle with wall taps	2.0%
- ASME Throat Tap flow nozzle	as stated in PTC 6 or 19.5.

7.5.2 Uncertainty of the expansion factor ϵ [Y] - The uncertainty of the expansion factor can be calculated as follows:

S I Units

$$\frac{\Delta\epsilon}{\epsilon} \cdot 100 = \frac{2\Delta p}{p_1} \%$$

U. S. Units

$$\frac{\Delta Y}{Y} \cdot 100 = 0.072 \frac{h_w}{p_1} \% \qquad \text{(Eq 33)}$$

7.6 UNRECOVERED PRESSURE LOSS - For ASME flow nozzles the fraction of the differential pressure which is unrecovered can be estimated by the following equation:

S I Units

$$\Delta\omega = (1 + 0.014\beta - 2.06\beta^2 + 1.18\beta^3)\Delta p$$

U. S. Units

$$h = (1 + 0.014\beta - 2.06\beta^2 + 1.18\beta^3)h_w \qquad \text{(Eq. 34)}$$

The definition given in the last paragraph of 6.4 also applies to the permanent pressure loss of ASME nozzles.

8. ASME VENTURI TUBES

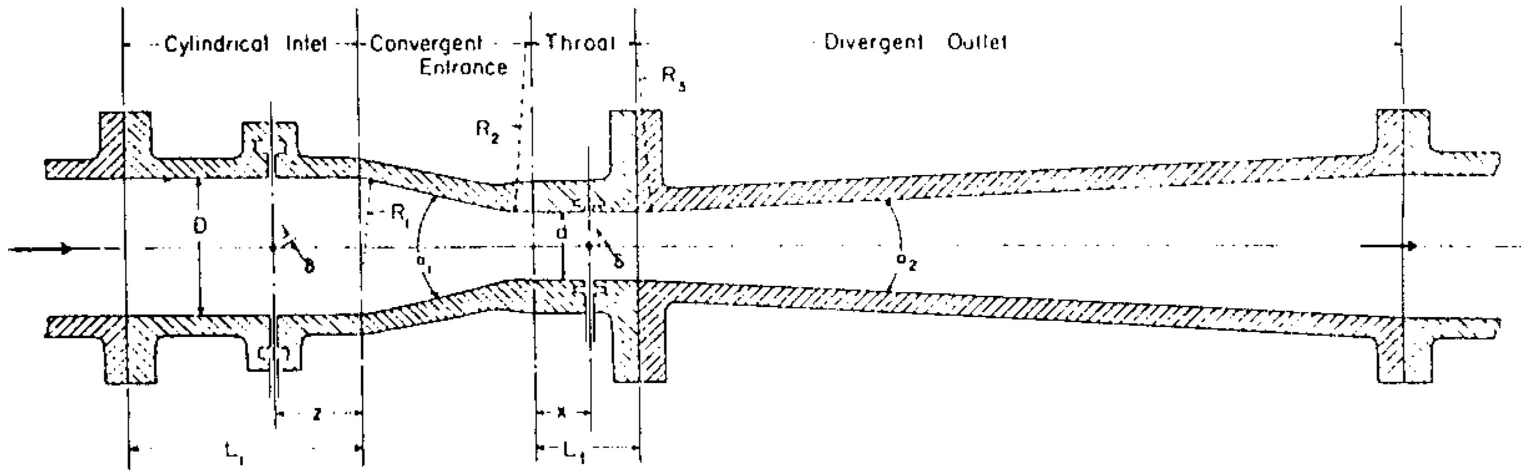
There are three types of ASME venturi tubes covered by this Standard. Each type is identified by the method of manufacturing, the internal surface of the entrance cone and the profile at the intersection of the entrance cone and the throat. These three types are described in ¶ 8.1.1 to ¶ 8.1.3.

ISO Venturi Nozzles generally are not manufactured in the United States and have not been included in this Standard. Information on these designs are provided in ISO 5167 or ASME Fluid Meters.

8.1.1 ASME VENTURI WITH A ROUGH CAST CONVERGENT - this venturi tube is manufactured from castings. The throat is machined and the junctions between the throat, the entrance section and the convergent and divergent sections are rounded.

8.1.2 ASME VENTURI WITH A MACHINED CONVERGENT - This venturi tube is manufactured as in ¶ 8.1.1, except that the entrance section, convergent and throat are machined as one assembly from a bar, forging or other suitable material. The junctions between the convergent and the entrance sections and throat may not be rounded.

8.1.3 ASME VENTURI WITH A FABRICATED CONVERGENT - This venturi tube is manufactured from formed metal sheet and/or other suitable materials and is usually joined together by welding. The junctions between the convergent and the entrance section, convergent and throat are not rounded. The throat section is machined if necessary to secure the degree of roundness or roughness required herein.



$L_1 \geq D$ or $L_1 \geq (D/4 + 250mm[10in]$

$z \leq D/2 \pm D/4$ for $100mm[4in] \leq D \leq 150mm[6in]$

$D/4 \leq z \leq D/2$ for $150mm[6in] \leq D \leq 810mm[32in]$

$L_t \geq d/3$

$y \geq d/6$

$4mm[5/32in] \leq \delta \leq 10mm[25/64in]$ and $\delta \leq 0.1D$ or $0.13d$

$R1 = 1.375\ D \pm 20\%$

$R2 = 3.625d \pm 0.125d$

$5d \leq R3 \leq 15d$

$a_1 = 21^\circ \pm 1^\circ$

$7^\circ \leq a_2 \leq 15^\circ$

Figure 7

PROFILE OF THE ASME VENTURI TUBE

8.2 GEOMETRIC PROFILES FOR ASME VENTURI TUBE

Figure 7 illustrates a cross sectional view of an ASME Classical Venturi Tube. The letters used in Figure 7 are for reference purposes. The venturi is composed of an entrance cylinder (A), connected to a conical convergent section (B), a cylindrical throat section (C), and a conical divergent section (E). All sections are concentric with the center line of the venturi tube. This may be checked by visual examination.

8.2.1 ENTRANCE SECTION - The entrance section shall have an inside diameter (D) and it shall be at least one inside diameter long. The inside diameter of the entrance section shall not vary from the matching pipe inside diameter by more than 0.01D and it shall be concentric with the matching upstream pipe when examined visually. The inside diameter of the entrance section shall be measured in the plane of the pressure taps at a minimum of four (4) equally spaced (approximately 45 degree) measurements passing through the center line of the section. These measurements must be made so that at least one measurement is taken at or near each pressure tap. No inside diameter measurement shall vary from the average of these measurements by more than ±0.5%.

8.2.2 CONVERGENT SECTION - The convergent section (B) shall be conical with an included angle of 21 ±1 degrees. The convergent section begins at the plane of the radius R_1, ends at the plane of the radius R_2, and is approximately [2.7(D-d)] long when measured on the center line of the venturi. The profile of the convergent section may be checked with a straight template and shall not deviate from the template by more than ±0.005D.

8.2.3 THROAT - The throat (C) shall have an inside diameter (d) which shall be round to within ±0.1% of the average inside diameter. The throat shall be parallel and cylindrical with the center line of the venturi tube when examined visually. The throat begins at the radius R_2 and ends at the radius R_3 and has a length of 1.0d ±0.05d. The radii at each end of the throat shall be as provided in ¶ 8.2.7, ¶ 8.2.8 or ¶ 8.2.9 herein and when compared with a template shall not deviate from it by more than 0.02d.

The inside diameter (d) shall be measured in the plane of the pressure taps at four (4) equally spaced (approximately 45°) measurements passing through the center line of the throat. The location of these measurements may be made beginning at any point on the internal circumference as long as at least one (1) measurement is taken at or near each pressure tap. No inside diameter measurement shall vary from the average of these measurements by more than ±0.1%.

8.2.4 DIVERGENT SECTION - the divergent section (E) shall be conical and shall have an included angle between 7° and 15°. It is recommended than an angle of 7° be chosen for minimum permanent pressure loss. The smallest diameter of the divergent section shall be not less than the inside diameter (d). The larger end of the divergent section shall have an inside diameter (D) and shall terminate at the matching pipe inside diameter, unless truncated as allowed by ¶ 8.2.5. When furnishing Venturi tubes without flanged ends, the venturi may be supplied with an exit cylinder section attached to the divergent to accommodate installation to the matching downstream pipe.

8.2.5 TRUNCATION - A Venturi tube may be shortened by up to 35% of the divergent section length by truncation. A venturi tube is truncated when the inside diameter of the venturi outlet end is less than the diameter (D). Such truncation may slightly increase the permanent pressure loss.

8.2.6 ROUGHNESS - The throat and the radii of all curvatures shall have a maximum roughness of less than 1.3μm [50μin]. The roughness of the other parts of the Venturi shall depend on its method of manufacture (See ¶8.2.7, ¶8.2.8, and ¶8.2.9); however, the internal surfaces shall be clean and free from all protrusions, encrustations, and/or welding deposits.

8.2.7 CHARACTERISTICS OF AN ASME VENTURI WITH A ROUGH CAST CONVERGENT - The minimum length of the entrance section (A) shall be equal to the smaller of 1D or (0.25 D + 250mm[10 in.]). The internal surface of the entrance section (A) and the convergent section (B) may be left rough cast provided that they are free from cracks, fissures, depressions, irregularities and impurities, and the maximum roughness is 8μm[300μin]. The radius of curvature R_1 shall be equal to 1.375D (±20%). The radius of curvature R_2 shall be between 3.625d and 3.750d. The radius of curvature R_3 shall be 10d ± 5d. The value of R_3 shall increase as the divergent angle decreases.

8.2.8 CHARACTERISTICS OF AN ASME VENTURI WITH A MACHINED CONVERGENT - The entrance section (A), convergent section (B), and divergent section (E) shall have a maximum roughness of 0.5μm[20μin]. The radii of curvature of R_1, R_2, and R_3 shall be less than 0.25d and preferably equal to zero.

8.2.9 CHARACTERISTICS OF AN ASME VENTURI WITH A FABRICATED CONVERGENT - The entrance section (A), convergent section (B), and divergent section (E) shall have a maximum roughness of 6.5mm[250min]. The radii of curvature of R_1, R_2, and R_3 shall be less than 0.25d and preferably equal to zero. All internal welded seams shall be flush with the surrounding surfaces.

8.3 MATERIALS AND MANUFACTURE OF ASME VENTURI TUBES

8.3.1 MATERIALS - Venturi tubes covered by this Standard may be manufactured from any material provided that the material used does not wear excessively and remains dimensionally stable in continued use.

8.3.2 JOINING THE CONVERGENT SECTION TO THE THROAT - It is recommended that the convergent section and the throat be manufactured from one piece of material. If this is not possible, as in fabricated construction, it is recommended that either:

(1) the throat section be machined after it is joined to the convergent section; or

(2) the throat section be of sufficient length to allow for the machining of the radius R_2 and a portion of the convergent angle, requiring the joining of the convergent section to the throat at a diameter greater than d.

8.3.3 JOINING THE DIVERGENT SECTION TO THE THROAT - Care shall be taken to ensure that the divergent section is centered with the throat. There shall be no steps between the inside diameters of the two parts.

8.3.4 THROAT LINING - Venturi throats may be lined with any material conforming to clause 8.3.1.

8.4 PRESSURE TAPS

8.4.1 NUMBER OF TAPS - If individual pressure taps are desired, a minimum of two (2) upstream and two (2) throat taps shall be provided. It is recommended that four (4) upstream and four (4) throat taps be provided and that they be interconnected by means of annular chambers conforming to §8.4.6.

8.4.2 TAP LOCATION - Upstream taps shall be located on the entrance section at a distance of 0.5D (+0.0D, -0.25D) from the beginning of the convergent section. Throat taps shall be located at (0.5 ± 0.02)d. Both upstream and throat taps shall be located on equal spacings (i.e. 180° or 90°) apart.

8.4.3 TAP HOLE EDGE - The edge of each pressure tap hole shall be square, sharp, and free of burrs or nicks at the inner surface of the venturi tube.

8.4.4 TAP LENGTH - The pressure tap hole shall be circular and cylindrical for a length of at least 2.5 times the diameter of the hole measured from the inside diameter of the Venturi.

8.4.5 TAP SIZE - The recommended size of the tap hole is between 4mm [0.15in] and 10mm[0.4 in] inclusive, but not greater than 0.1D for upstream taps and 0.13d for throat taps. It is additionally recommended that pressure taps be as small as possible while still considering the possibility of tap hole plugging by contamination.

8.4.6 TAPS WITH ANNULAR CHAMBERS - The cross sectional area of the annular chamber, if used, should be greater than half the sum of the pressure tap hole areas. It is recommended that the annular chamber be doubled in cross sectional area if the venturi is to be installed with insufficient upstream piping from a disturbance which may cause swirls or vortices in the measured fluid.

8.5 ASME VENTURI TUBE DISCHARGE COEFFICIENTS

8.5.1 LIMITS OF USE - The effects of D, β, and R_D on the value of the discharge coefficients of the Venturi tube outside the limits given in this section are not sufficiently well established to allow a reliable statement of uncertainty other than that given in Appendix C.

Flow Measurement at the limits of the values of D, β, and R_D simultaneously should be avoided because the probable uncertainty of the coefficients is greater.

8.5.2 COEFFICIENT OF DISCHARGE (C) FOR ASME VENTURI TUBES WITH A ROUGH CAST OR FABRICATED CONVERGENT - The value of the coefficient of discharge (C) for a venturi tube with a rough cast or fabricated convergent section is

$$C = 0.984 \qquad \text{(Eq. 35)}$$

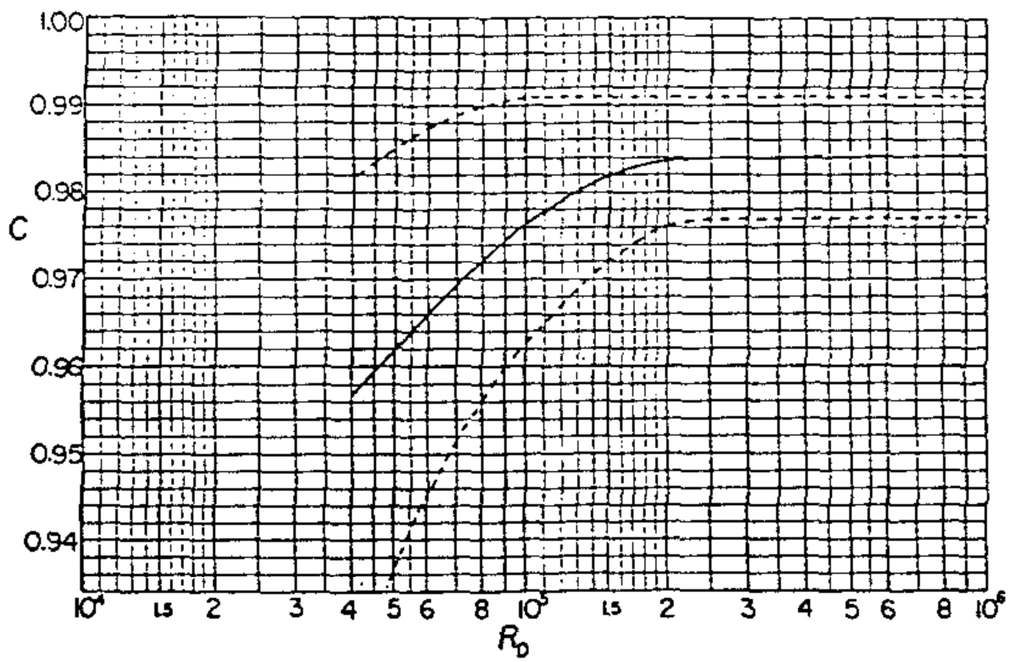
When the venturi is manufactured and used in accordance with this Standard and the following limitations are observed:

S I	U S
$100\text{mm} \le D \le 1200\text{mm}$	$[4\text{in} \le D \le 48\text{in}]$

$$0.30 \le \beta \le 0.75$$

$$2 \times 10^5 \le R_D \le 6 \times 10^6$$

Note: Upper limitation of 1200mm [48in] is recommended as a maximum; however, current data indicates that Venturis may be used up to 2100mm [84in] with reliability.



Uncertainty Band

Fig. 8, Discharge Coefficients of ASME Rough Cast Venturi tubes as a function of RD

8.5.3 COEFFICIENT OF DISCHARGE (C) FOR ASME VENTURI TUBES WITH A MACHINED CONVERGENT - The value of the coefficient of discharge (C) for a venturi tube with a machined convergent section is:

$$C = 0.995 \qquad \text{(Eq. 36)}$$

When the venturi is manufactured and used in accordance with this Standard and the following limitations are observed:

$$50\text{mm} \le D \le 250\text{mm} \qquad [2\text{in} \le D \le 10\text{in}]$$

$$0.30 \le \beta \le 0.75$$

$$2 \times 10^5 \le R_D \le 2 \times 10^6$$

8.5.4 EXPANSION FACTOR ε [Y]

The information given in ¶7.4.3 also applies to all venturi tubes covered by Section 8, provided the limits of use as given in ¶8.5.2 or ¶8.5.3 are observed.

8.6.1 UNCERTAINTY OF COEFFICIENT OF DISCHARGE (C) - The percentage of uncertainty of the coefficient of discharge as given in ¶8.5.2 and ¶8.5.3 is equal to 1.0% regardless of the method of manufacture.

8.6.2 UNCERTAINTY OF THE EXPANSION FACTOR ε [Y] - The percentage of uncertainty of the expansion factor for ASME venturi tubes as given in 8.5.4, is equal to the following expression:

SI

$$\frac{\delta\varepsilon}{\varepsilon} \cdot 100 = [4+100\beta^8] \frac{\Delta p}{p_1} \%$$

US

$$\frac{\delta Y}{Y} \cdot 100 = [0.144 + 3.61\beta^9] \frac{h_w}{p_1} \%$$

8.7 <u>PRESSURE LOSS OF VENTURI TUBES</u> - ξ [h_1]

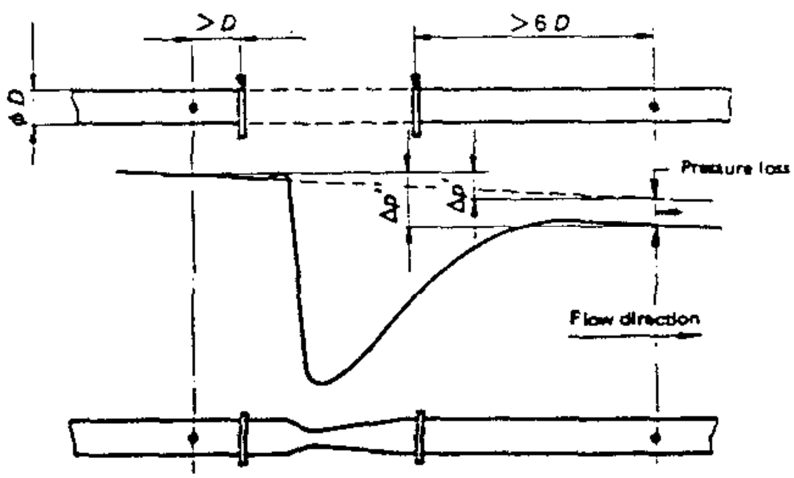


Figure 9, PRESSURE LOSS ACROSS AN ASME VENTURI TUBE

8.7.1 DEFINITION OF PRESSURE LOSS (Figure 9) - the loss of pressure caused by a Venturi tube, at a specified flow rate, may be measured by determining the pressure in the pipe prior to and subsequent to the installation of the Venturi. As illustrated by Figure 9, $\Delta p'$ is the difference in pressure measured between a point 1 D upstream from the upstream end of the Venturi and a point 6 D downstream from the downstream end of the Venturi prior to installation of the venturi. $\Delta p''$ is the difference in pressure after installation at the same pressure measuring locations. The pressure loss is equal to $\Delta p'' - \Delta p'$.

8.7.2 VALUE OF THE PRESSURE LOSS - The value, $\xi = \frac{\Delta p'' - \Delta p'}{\Delta p}$, the pressure loss:

(1) Decreases as diameter ratio increases.

(2) Decreases as Reynolds number increases.

(3) Decreases as divergent angle decreases.

(4) Roughness of Internal Surface - Pressure loss decreases as roughness decreases.

(5) Decreases as alignment with matching pipe is improved.

Pressure losses are generally within 5% to 20% of the developed differential pressure as shown in the charts in Appendix D. The largest influence on pressure loss is the divergent cone angle and β. Utilizing only these two factors, the formulae shown below provides estimates for loss as a percentage of developed differential pressure.

For 15° included angle divergent -
$\xi = 0.436 - 0.86\beta + 0.59\beta^2$

For 7° included angle divergent -
$\xi = 0.218 - 0.42\beta + 0.38\beta^2$

8.8 INSTALLATION REQUIREMENTS FOR ASME VENTURI TUBES

8.8.1 CIRCULARITY OF PIPE - *In the immediate vicinity of the ASME venturi tube the* following shall apply:

(1) The average upstream pipe diameter (D′) shall be within ±1.0% of the average diameter (D) for at least 2D measured upstream from the entrance section.

(2) No single measurement of the upstream pipe diameter (D′) shall differ by more than ±2.0% from the average diameter (D) of the ASME venturi entrance section for at least 2D preceding the entrance section.

(3) The average downstream pipe diameter must not be less than 90% of inside diameter of the downstream end of the diffuser section of the ASME Venturi.

8.8.2 ROUGHNESS OF UPSTREAM PIPE - The upstream pipe shall not have a roughness of *greater than $k/D \times 10^{-3}$ for at least 2D from the upstream end of the entrance* section of the venturi tube.

8.8.3 ALIGNMENT OF THE ASME VENTURI - The offset between the center lines of the upstream pipe and the Venturi shall be less than 0.005D and shall be aligned with the upstream piping to within 1°.

8.8.4 RECOMMENDED UPSTREAM STRAIGHT LENGTHS FOR ASME VENTURIS - Table 6 gives the minimum straight upstream lengths recommended by this Standard for which there is an additional 0.5% uncertainty on discharge coefficient.

8.8.4.1 LIMITATIONS - *The limitations given in* §5.4.1 thru §5.4.4 apply to the values given in Table 6. All straight lengths are expressed in multiples of inside diameter (D) as measured from the upstream tap of the Venturi tube. The pipe roughness for the entire length of recommended straight run should not exceed that of new, commercially available pipe.

TABLE 6 - RECOMMENDED STRAIGHT LENGTHS (4),(5)
(For 0.5% Additional Uncertainty)

Diameter Ratio	Single 90° short Radius Bend(1)	Two or more 90° bends in the same plane(1) (6)	Two or more 90° bends in different planes(1,3) (6)	Reducer 3D to D over a length of of 3.5D	Expander 0.75D to D over a length of D	Ball or Gate Valve Fully Open
0.30	0.5[2]	0.5	0.5	0.5[2]	0.5	0.5
0.35	0.5[2]	0.5	0.5	0.5	0.5	0.5
0.40	0.5[2]	0.5	0.5	0.5	0.5	1.5
0.45	0.5	0.5	0.5	0.5	1.0	1.5
0.50	0.5	1.5	8.5	0.5	1.5	1.5
0.55	0.5	1.5	12.5	0.5	1.5	2.5
0.60	1.0	2.5	17.5	0.5	1.5	2.5
0.65	1.5	2.5	23.5	1.5	2.5	2.5
0.70	2.0	2.5	27.5	2.5	3.5	3.5
0.75	3.0	3.5	29.5	3.5	4.5	3.5

Notes:
1) - The radius of curvature of the bend shall be equal to or greater than the pipe diameter.
2) - These lengths require no additional uncertainty, but the shorter lengths are not proven sufficiently to be published in this Standard.
3) - Data has been published which would suggest that, after two elbows not in the same plane, less error in coefficient would be found by eliminating all straight upstream pipe.
4) - The minimum lengths of straight upstream pipe for ASME Venturis are less than those in Table 2 for orifice plates and flow nozzles because they are derived from different experimental results. The convergent portion of the ASME Venturi is designed to obtain a more uniform velocity profile at the throat.
5) - The statements in 5.4.5 regarding Table 2 are also applicable regarding Table 6.
6) - The insertion of 5 to 10D straight lengths between the two bends is sufficient to make the combined effect the same as the single bends in the left column.

8.8.5 RECOMMENDED DOWNSTREAM STRAIGHT LENGTHS - Fittings and other disturbances as indicated by Table 6, situated at least four (4) throat diameters downstream of the throat tap, do not affect the accuracy of the measurement.

9. UNCERTAINTIES IN THE MEASUREMENT OF FLOW RATE

Reference shall be made to the ASME MFFCC Standard and ISO 5168, (Calculation of the uncertainty of a measurement of flow rate), which give useful general information on this subject.

9.1 Definition of uncertainty

9.1.1 For the purpose of this Standard the uncertainty is defined as a range of values within which the true value of the measurement is estimated to lie at the 95% probability.

In some cases, the confidence level which can be attached to this range of values will be greater than 95%, but this will be so only where the value of a quantity used in the calculation of flow rate is known with a confidence level in excess of 95%. In such a case, reference shall be made to the ASME MFC - 1M Standard and ISO 5168.

9.1.2 The uncertainty on the measurement of the flow rate shall be calculated and given under this name whenever a measurement is claimed to be in conformity with this Standard.

9.1.3 The uncertainty can be expressed in absolute or relative terms and the result of the flow measurement can then be given in any of the following forms:

rate of flow = $q \pm \delta q$

or rate of flow = $q\ (1 \pm e)$

or rate of flow = q within $(100\ e)\%$

where the uncertainty δq shall have the same dimensions as q while $e = \delta q/q$ and is non-dimensional.

9.1.4 The uncertainty of the flow measurement so defined is equivalent to twice the standard deviation of statistical terminology. Like the latter, it is obtained by combining the uncertainties of the individual quantities used in the flow rate calculation, assuming them to be small and independent of each other. For one single measuring device used in one test, some of these uncertainties may be systematic, of which only an estimate is known. Their combination is permitted as if they were random errors having a distribution conforming to the Laplace-Gauss normal law.

9.1.5 For convenience a distinction is made between the uncertainties associated wit measurement equipment of the user and those quantities specified in this Standard. The latter uncertainties are on the discharge coefficient and the expansion factor an indicate the uncertainty over which the user has no control. They occur because smal variations in geometry are allowed and because investigations on which the values are based could not be made under "ideal" conditions.

9.2 Practical computation of the uncertainty

9.2.1 The basic formula of computation of the mass rate of flow q_m is:

SI

$$q_m = \frac{\pi}{4} \; C \varepsilon_1 d^2 \sqrt{\frac{2\Delta p \bullet \rho_{1}}{1-\beta^4}}$$

US

$$q_m = 0.099\ 701\ 90\ C Y_1 d^2 \sqrt{\frac{2 h_w \bullet \rho_{1}}{1-\beta^4}} \qquad \text{(Eq 37)}$$

In fact, the various quantities which appear on the right-hand side of this formula are not independent, so that it is not correct to compute the uncertainty of q_m directly from the uncertainties of these quantities.

For example, C is a function of d, D, κ, U_1, ν_1

ε [Y] is a function of d, D, Δp [h_w], p_1, κ

However, it is sufficient to calculate the uncertainties of ε [Y], Δp [h_w], and ρ_1 as if independent of each other and also independent of the uncertainties of C and d.

9.2.1.2 A practical working formula for δq_m may then be derived, which takes account of the interdependence of C, d and D which enters into the calculation as a consequence of the dependence of C on β. It shall be noted that C may also be dependent on the pipe diameter D, as well as on the Reynolds number R_D. However, the deviations of C due to these influences are of a second order and are included in the uncertainty on C.

Similarly, the deviations of ε [Y] which are due to uncertainties in the value of β, the pressure ratio and the isentropic exponent are also of a second order and are included in the uncertainty on ε [Y].

9.2.1.3 The uncertainties which shall be included in a practical working formula for δqm are therefore those of the quantities C, ε [Y], D, Δp [h_w], and ρ_1.

9.2.2 The practical working formula for the uncertainty, δq_m, of the mass rate of flow is as follows:

For SI units:

$$e = \frac{\delta q_m}{q_m} = \left\{\left[\frac{\delta C}{C}\right]^2 + \left[\frac{\delta \varepsilon}{\varepsilon}\right]^2 + \left[\frac{2\beta^4}{1-\beta^4}\right]^2\left[\frac{\delta D}{D}\right]^2 + \left[\frac{2}{1-\beta^4}\right]^2\left[\frac{\delta d}{d}\right]^2 + \left[\frac{\delta \Delta p}{2\Delta p}\right]^2 + \left[\frac{\delta \rho}{2\rho}\right]^2\right\}^{0.5} \qquad \text{(Eq 38a)}$$

For US units:

$$e = \frac{\delta q_m}{q_m} = \left\{\left[\frac{\delta C}{C}\right]^2 + \left[\frac{\delta Y}{Y}\right]^2 + \left[\frac{2\beta^4}{1-\beta^4}\right]^2\left[\frac{\delta D}{D}\right]^2 + \left[\frac{2}{1-\beta^4}\right]^2\left[\frac{\delta d}{d}\right]^2 + \left[\frac{\delta h_w}{2h_w}\right]^2 + \left[\frac{\delta \rho}{2\rho}\right]^2\right\}^{0.5} \qquad \text{(Eq 38b)}$$

In the formula above some of the uncertainties, like those on the flow and expansion coefficients, are given in this Standard (see ¶9.2.2.1 and ¶9.2.2.2) while others must be determined by the user (see ¶9.2.2.3 and ¶9.2.2.4).

9.2.2.1 In the formulas above, the values of $\delta C/C$ and of $\delta \varepsilon/\varepsilon$ shall be taken from the appropriate sections of this Standard.

9.2.2.2 When the straight lengths are such that an additional uncertainty of 0.5% must be included, this additional uncertainty shall be added simply according to ¶5.4.1 and ¶5.4.2, but not quadratically combined with the other uncertainties in the formula given in ¶9.2.2. Other additional uncertainties must be added in the same way.

9.2.2.3 In the formula above the maximum values of $\delta D/D$ and of $\delta d/d$, which can be derived from the specifications given in section 5, and in ¶6.1.6, ¶7.1 and ¶8.2, can be adopted or alternatively the smaller actual values can be computed by the user. (The maximum values of $\delta D/D$ may be taken as 0.4% while the maximum value for $\delta d/d$ may be taken as 0.07%).

9.2.2.4 The values of $\delta \Delta p/\Delta p$ [$\delta h_w/h_w$] and $\delta p_{f_1}/p_{f_1}$ shall be determined by the user because this Standard does not specify in detail the method of measurement of the quantities Δp [h_w] and p_{f_1}.

APPENDIX A

Typical values of the pipe wall roughness k

Material	Condition	SI k, mm	US k , in•10^{-3}
brass, copper, aluminum, plastics, glass	smooth, without sediment	<0.03	<1
steel	new, seamless cold drawn new, seamless hot drawn new, seamless rolled new, welded longitudinally new, welded spirally slightly rusted rusty encrusted with heavy encrustations bituminized, new bituminized, normal galvanized	<0.03 0.05 to 0.10 0.10 0.10 to 0.20 0.20 to 0.30 0.50 to 2 >2 0.03 to 0.05 0.01 to 0.02 0.13	<1 2 to 4 10 4 to 10 10 to 15 20 to 80 >80 1to2 0.5 to 1 5
cast iron	new rusty encrusted bituminized, new	0.25 1.0 to 1.5 >1.5 0.03 to 0.05	10 40 to 60 >60 1 to 2
asbestos cement	new; coated and not coated normal: not coated	<0.03 0.05	<1 2

APPENDIX B

TABLE B1
RESULTS OF SAMPLE CALCULATIONS FOR DISCHARGE COEFFICIENTS AT $R_D = 10^5$

ORIFICES (¶6.3.2.1 EQUATIONS)

Beta β	Corner taps	D and D/2 taps	Flange taps 50mm (2in)	Flange taps 100mm (4in)
0.30	0.5992	0.5991	0.5990	0.5991
0.50	0.6053	0.6060	0.6058	0.6058
0.70	0.6067	0.6136	0.6132	0.6110

ASME NOZZLE (EQ.30)

0.30	0.9862	
0.50	0.9829	
0.70	0.9802	

TABLE B2
RESULTS OF SAMPLE CALCULATIONS FOR EXPANSION FACTOR ϵ [Y]

FOR ASME NOZZLES

p_2/p_1	0.95	0.80
β Isentropic Exponent κ = 1.4		
0.30	0.9726	0.8855
0.50	0.9706	0.8785
0.70	0.9622	0.8506
β Isentropic Exponent κ =1.3		
0.30	0.9705	0.8773
0.50	0.9683	0.8700
0.70	0.9594	0.8406

APPENDIX C

CLASSICAL VENTURI TUBES USED OUTSIDE THE SCOPE COVERED BY THIS STANDARD

C.1 GENERAL

As indicated in §8.5.1 the effects of R_D, K/D and β on C are not yet well enough known to allow standardization outside the limits specified in this Standard.

This appendix summarizes the data from all the available results which can be used. The values or the direction of variation of discharge coefficients and uncertainties are given in terms of the parameters β, R_D and K/D in order to allow an estimate of the rate of flow. These various effects will be treated separately though some results show that they are not independent.

The number of tests available on this subject is small and these tests were mostly carried out on Venturi tubes whose geometry was not strictly in accordance with this Standard. As a result the confidence in the discharge coefficients and the uncertainties is relatively low.

C.2 EFFECTS OF THE DIAMETER RATIO β

From an examination of the results available for Venturi tubes with diameter ratios around and above β = 0.75* it has been noted that the spread of measured discharge coefficients is wider than for smaller diameter ratios. Hence the uncertainty on the coefficient should be increased.

It is recommended to double the uncertainty on C when β is above the maximum permissible value.

*This recommendation is based on tests carried out on Venturi tubes of diameter ratio β up to 0.8.

C.3 INFLUENCE OF THE REYNOLDS NUMBER R_D

C.3.1 General

The effect of the Reynolds number R_D vary according to the type of classical Venturi tube and causes a variation in the discharge coefficient and an increase in the uncertainty.

These variations are more important when R_D is less than the specified minimum of R_D than when R_D is greater than the specified maximum of R_D.

C.3.2 Classical venturi tube with a rough-cast convergent

The effects of Reynolds number are as described below.

When R_D decreases below 2×10^5 the discharge coefficient C decreases steadily and the uncertainty increases.

When R_D increases above 2×10^6 the discharge coefficient does not appear to change with Reynolds number nor does the uncertainty.

For estimation of the rate of flow, the following values of the discharge coefficient C and the uncertainty, given as guidance, may be used:

R_D	C	Uncertainty %
4×10^4	0.957	2.5
6×10^4	0.966	2
1×10^5	0.976	1.5
1.5×10^5	0.982	1

C.3.3 Classical venturi tube with a machined convergent

The effects of Reynolds number are:

When R_D decreases below 2×10^5 it is often found that there is a small increase in the discharge coefficient C before there is a steady decrease with decreasing R_D. The uncertainty on C increases slowly at first and then rapidly.

When R_D increases above 2×10^6 it is found occasionally that there is a slight increase of C with R_D; the uncertainty on C also increases slightly.

There is sufficient evidence available to show that better correlation is achieved in terms of R_d than in terms of R_D.

The maximum occurs between 2×10^5 and 4×10^5.

In order to estimate the rate of flow the following values of the discharge coefficient and the uncertainty may be used:

R_d	C	Uncertainty %*
5×10^4	0.970	3
1×10^5	0.977	2.5
2×10^5	0.992	2.5
3×10^5	0.998	1.5
5×10^5	0.995	1

* For low Reynolds number, experimental results are not a Gaussian distribution. The mean deviation of C is skewed toward lower values.

C.3.4 Classical venturi tube with a rough-welded sheet-iron convergent

When R_D decreases below 2×10^5 the discharge coefficient C decreases slightly while the uncertainty on C increases.

The following values of the discharge coefficient and the uncertainty may be used to obtain an estimate of the rate of flow:

R_d	C	Uncertainty %
4×10^4	0.96	2.3
6×10^4	0.97	2.5
1×10^5	0.98	2.5
2×10^6	0.985	2.0

C.4 EFFECTS OF THE RELATIVE ROUGHNESS K/D

C.4.1 Roughness of the classical venturi tube

An increase of the convergent roughness reduces the discharge coefficient C. Considering the present stage of knowledge, it is not possible to predict the value of that reduction, but it is probably less than 2%.

Classical Venturi tubes with a machined convergent seem to be more sensitive to roughness than the others.

The pressure loss of the Venturi tube increases with the roughness.

C.4.2 Roughness of the upstream pipe

An increase of the upstream pipe roughness causes an increase of the discharge coefficient C for the classical Venturi tube. This effect seems more marked as becomes greater.

There are insufficient satisfactory data to provide quantitative results on this subject.

In order to allow an estimation of the discharge coefficient and the uncertainty, it may be noted that transfer from a hydraulically smooth pipe to a pipe in the relative roughness of which is 5×10^{-4} may involve increases of the discharge coefficient ranging from 0.2 to 0.7% for $\beta = 0.5$.

It is recommended to increase the uncertainty on C by at least half the correction made on C.

APPENDIX D

PRESSURE LOSS IN A CLASSICAL VENTURI TUBE

This appendix is given for guidance only.

D.1 For a classical venturi tube with a total angle of the divergent equal to 7° and a pipe Reynolds number R_D greater than 10^6, the relative loss,

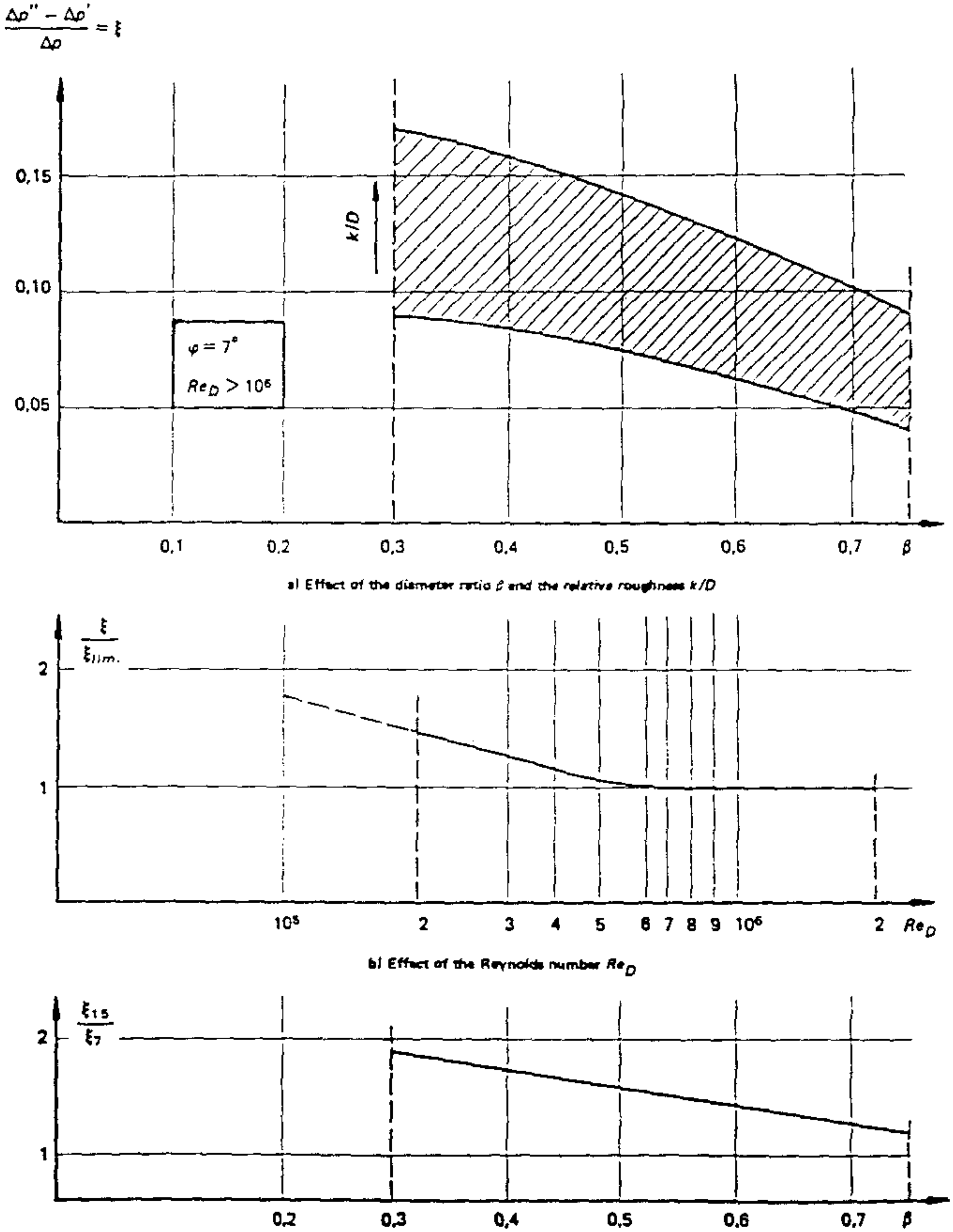
$$\xi = \frac{\Delta p'' - \Delta p'}{\Delta p} \qquad \text{(Eq D1)}$$

probably is in the hatched area shown on Figure D-a.

D.2 For a given Venturi tube, ξ decreases when R_D increases and it seems to reach a limiting value above $R_D = 10^6$. Figure D-b is an approximation on the ratio of ξ to its limiting values.

D.3 The relative pressure loss increases with the angle of the divergent, as shown in Figure D-c.

D.4 No precise indication is available on the pressure loss of a truncated Venturi tube. The length of the divergent may be reduced by about 35% without a significant increase of the permanent pressure loss.



a) Effect of the diameter ratio β and the relative roughness k/D

b) Effect of the Reynolds number Re_D

c) Effect of the angle of the divergent φ

Figure D, Pressure loss across an ASME Venturi Tube

APPENDIX E

THERMAL EXPANSION FACTOR

I. Calculation of F_a, thermal expansion correction factor, Example

Assume:
- A. 316SS Orifice Plate
- B. Carbon steel pipe
- C. $\beta = 0.60$
- D. $t_{meas} = 68\ ^\circ F$
- E. $t = 110\ ^\circ F$
- F. $\alpha_{pE} = 0.0000096$ (for 316 plate)
- G. $\alpha_p = 0.0000067$ (for carbon steel pipe)

then:

$$F_a = 1 + \frac{2}{1 - \beta^4} (\alpha_{pE} - \beta^4 \alpha_p)(t - t_{meas})$$

and: $Fa = 1.000843$

THERMAL EXPANSION DATA (SI)

Mean Coefficient of Thermal Expansion = $\frac{A}{10^6}$ (mm/mm/°C) } in Going From 21°C to Indicated Temperature

Linear Thermal Expansion = B (mm/m)

Material	Coef-ficient	Temp. Range 21°C to −198	−101	−46	21	93	149	204	260	316	371	427	482	538	593	649	704	760
Carbon steel; carbon-moly steel	A	9.00	9.90	10.44	10.93	11.48	11.88	12.28	12.64	13.01	13.39	13.77	14.11	14.35	14.62	14.74	14.90	15.05
low-chrome steels (through 3Cr)	B	1.97	1.20	0.70	0	0.82	1.52	2.25	3.02	3.83	4.69	5.58	6.51	7.41	8.37	9.25	10.18	11.11
Intermediate alloy steels;	A	8.46	9.36	9.81	10.31	10.87	11.14	11.41	11.70	11.99	12.24	12.53	12.78	13.00	13.18	13.34	13.48	13.59
(5Cr-Mo through 9Cr-Mo)	B	1.85	1.14	0.66	0	0.78	1.42	2.08	2.79	3.53	4.28	5.08	5.89	6.71	7.51	8.33	9.21	10.04
Austenitic stainless steels	A	14.67	15.48	16.02	16.40	16.81	17.05	17.26	17.46	17.68	17.86	18.09	18.29	18.52	18.70	18.86	18.97	19.08
	B	3.21	1.89	...	0	1.22	2.17	3.16	4.17	5.20	6.25	7.33	8.43	9.56	10.70	11.83	12.96	14.09
Straight chromium stainless steels;	A	7.74	8.46	9.00	9.43	9.90	10.19	10.46	10.73	11.03	11.27	11.50	11.74	11.93	12.10	12.20	12.33	12.42
(12Cr, 17Cr, and 27Cr)	B	1.70	1.03	0.60	0	0.71	1.30	1.92	2.57	3.25	3.94	4.66	5.40	6.16	6.92	7.66	8.42	9.17
25Cr-20Ni	A	11.43	12.33	12.96	13.46	13.97	14.26	14.54	14.80	15.08	15.34	15.62	15.86	16.06	16.20	16.34	16.42	16.52
	B	2.50	1.51	0.82	0	1.01	1.82	2.67	3.53	4.44	5.36	6.33	7.33	8.29	9.26	10.25	11.21	12.20
Monel	A	9.99	12.15	12.87	13.46	14.11	14.44	14.76	15.12	15.44	15.80	16.13	16.49	16.81	17.14	17.46	17.78	18.07
(67Ni-30Cu)	B	2.17	1.49	...	0	1.02	1.84	2.71	3.61	4.55	5.53	6.54	7.60	8.68	9.80	10.95	12.11	13.34
Monel	A	9.63	11.61	12.24	12.82	13.46	13.82	14.22	14.56	14.94	15.30	15.66	16.02	16.38	16.74	17.10	17.46	17.80
(66Ni-29Cu-Al)	B	2.11	1.42	0.82	0	0.97	1.77	2.61	3.47	4.40	5.36	6.35	7.38	8.46	9.58	10.83	11.93	13.14
Aluminum	A	17.82	19.62	20.88	22.05	23.31	23.90	24.48	25.02	25.56	...	...	...	...	...	...	...	...
	B	3.90	2.40	1.39	0	1.67	3.05	4.49	5.97	7.52	...	...	...	...	...	...	...	...
Gray cast iron	A	...	...	...	...	10.35	10.67	10.98	11.30	11.65	11.97	12.29	12.60	12.94	...	...	...	...
	B	...	...	...	0	0.75	1.37	2.02	2.70	3.42	4.19	4.98	5.81	6.68	...	...	...	...
Bronze	A	15.12	15.75	16.47	17.23	18.05	18.22	18.41	18.58	18.79	18.94	19.12	19.30	19.44	19.62	19.80	...	...
	B	3.32	1.92	1.10	0	1.30	2.32	3.37	4.44	5.53	6.62	7.75	8.90	10.04	11.22	12.43	...	...
Brass	A	14.76	15.30	16.11	16.81	17.57	18.00	18.41	18.85	19.24	19.66	20.09	20.52	20.93	21.33	21.76	...	...
	B	3.23	1.87	1.07	0	1.27	2.30	3.37	4.50	5.66	6.88	8.15	9.45	10.81	12.20	13.65	...	...
Wrought iron	A	10.26	11.34	11.97	12.55	13.18	13.46	13.70	13.91	14.18	14.42	14.63	14.92	15.10	...	...	...	...
	B	2.25	1.39	0.80	0	0.95	1.72	2.51	3.32	4.17	5.05	5.93	6.88	7.80	...	...	...	...
Copper-nickel	A	11.97	13.32	14.04	14.69	15.37	15.68	16.02	...	...	...	...	...	...	...	...	...	...
(70Cu-30Ni)	B	2.62	1.62	0.94	0	1.11	2.00	2.93	...	...	...	...	...	...	...	...	...	...

GENERAL NOTE:
These data are for information and it is not to be implied that materials are suitable for all the temperature ranges shown.

THERMAL EXPANSION DATA

Mean Coefficient of Thermal Expansion $= \frac{A}{10^6}$ (in./in./°F)
Linear Thermal Expansion $= B$ (in./100 ft)

in Going From 70°F to Indicated Temperature

Material	Coefficient	Temp. Range 70°F to −325	−150	−50	70	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400
Carbon steel; carbon-moly steel	A	5.00	5.50	5.80	6.07	6.38	6.60	6.82	7.02	7.23	7.44	7.65	7.84	7.97	8.12	8.19	8.28	8.36
low-chrome steels (through 3Cr)	B	2.37	1.45	0.84	0	0.99	1.82	2.70	3.62	4.60	5.63	6.70	7.81	8.89	10.04	11.10	12.22	13.34
Intermediate alloy steels;	A	4.70	5.20	5.45	5.73	6.04	6.19	6.34	6.50	6.66	6.80	6.96	7.10	7.22	7.32	7.41	7.49	7.55
(5Cr-Mo through 9Cr-Mo)	B	2.22	1.37	0.79	0	0.94	1.71	2.50	3.35	4.24	5.14	6.10	7.07	8.06	9.05	10.00	11.06	12.05
Austenitic stainless steels	A	8.15	8.60	8.90	9.11	9.34	9.47	9.59	9.70	9.82	9.92	10.05	10.16	10.29	10.39	10.48	10.54	10.60
	B	3.85	2.27	...	0	1.46	2.61	3.80	5.01	6.24	7.50	8.80	10.12	11.48	12.84	14.20	15.56	16.92
Straight chromium stainless steels;	A	4.30	4.70	5.00	5.24	5.50	5.66	5.81	5.96	6.13	6.26	6.39	6.52	6.63	6.72	6.78	6.85	6.90
(12Cr, 17Cr, and 27Cr)	B	2.04	1.24	0.72	0	0.86	1.56	2.30	3.08	3.90	4.73	5.60	6.49	7.40	8.31	9.20	10.11	11.01
25Cr-20Ni	A	6.35	6.85	7.20	7.48	7.76	7.92	8.08	8.22	8.30	8.52	8.68	8.81	8.92	9.00	9.08	9.12	9.18
	B	3.00	1.81	0.98	0	1.21	2.18	3.20	4.24	5.33	6.44	7.60	8.78	9.95	11.12	12.31	13.46	14.65
Monel	A	5.55	6.75	7.15	7.48	7.84	8.02	8.20	8.40	8.58	8.78	8.96	9.16	9.34	9.52	9.70	9.88	10.04
(67Ni-30Cu)	B	2.62	1.79	...	0	1.22	2.21	3.25	4.33	5.46	6.64	7.85	9.12	10.42	11.77	13.15	14.58	16.02
Monel	A	5.35	6.45	6.80	7.12	7.48	7.68	7.90	8.09	8.30	8.50	8.70	8.90	9.10	9.30	9.50	9.70	9.89
(66Ni-29Cu-Al)	B	2.53	1.70	0.98	0	1.17	2.12	3.13	4.17	5.28	6.43	7.62	8.86	10.16	11.50	13.00	14.32	15.78
Aluminum	A	9.90	10.90	11.60	12.25	12.95	13.28	13.60	13.90	14.20	...	...	...	...	...	...	...	...
	B	4.68	2.88	1.67	0	2.00	3.66	5.39	7.17	9.03	...	...	...	...	...	...	...	...
Gray cast iron	A	...	...	...	...	5.75	5.93	6.10	6.28	6.47	6.65	6.83	7.00	7.19	...	...	...	...
	B	...	...	...	0	0.90	1.64	2.42	3.24	4.11	5.03	5.98	6.97	8.02	...	...	...	...
Bronze	A	8.40	8.75	9.15	9.57	10.03	10.12	10.23	10.32	10.44	10.52	10.62	10.72	10.80	10.90	11.00	...	...
	B	3.98	2.31	1.32	0	1.56	2.79	4.05	5.33	6.64	7.95	9.30	10.68	12.05	13.47	14.92	...	...
Brass	A	8.20	8.50	8.95	9.34	9.76	10.00	10.23	10.47	10.69	10.92	11.16	11.40	11.63	11.85	12.09	...	...
	B	3.88	2.24	1.29	0	1.52	2.76	4.05	5.40	6.80	8.26	9.78	11.35	12.98	14.65	16.39	...	...
Wrought iron	A	5.70	6.30	6.65	6.97	7.32	7.48	7.61	7.73	7.88	8.01	8.13	8.29	8.39	...	...	...	...
	B	2.70	1.67	0.96	0	1.14	2.06	3.01	3.99	5.01	6.06	7.12	8.26	9.36	...	...	...	...
Copper-nickel[1]	A	6.65	7.40	7.80	8.16	8.54	8.71	8.90	...	...	...	...	...	...	...	...	...	...
(70Cu-30Ni)	B	3.15	1.95	1.13	0	1.33	2.40	3.52	...	...	...	...	...	...	...	...	...	...

GENERAL NOTE:
These data are for information and it is not to be implied that materials are suitable for all the temperature ranges shown.

II. THERMAL EXPANSION OF VARIOUS MATERIALS

Material, temperature	Coefficient of Thermal Expansion at 93°C [200°F] mm/mm/deg C	in/in/deg F
Hastelloy B, 0 to 100°C [32 to 212°F]	0.0000101	0.0000056
Hastelloy C	0.0000113	0.0000063
Inconel X annealed	0.0000120	0.0000067
Haynes Stellite 25 (L605)	0.0000137	0.0000076
Copper (ASTM B152, B124, B133)	0.0000167	0.0000093
Beryllium copper 25 (ASTM B194)	0.0000167	0.0000093
Titanium, 20 to 100°C [70 to 212°F]	0.0000085	0.0000047
Tantalum, 20 to 100°C [70 to 212°F]	0.0000065	0.0000036

CAUTION: These values are for temperatures around 93°C [200°F]. Coefficients will vary as much as 100% in going from -200°C [-330°F] to 750°C [1400°F]. Refer to tables in handbooks for those temperatures.

INDEX