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Revision Memo

- 12/03** Highlights of revisions:
- Added section describing physical differences between subsonic and transonic flows and the implications of these differences
 - Minor changes throughout body of section

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SCOPE

This subsection presents calculation methods for determining pressure drop through piping and related equipment for gas and vapor compressible flow. For general considerations other than pressure drop, see [Section XIV-A](#).

REFERENCES

DESIGN PRACTICES

(Besides other subsections of this section)

[Section XIV-B](#), Instrumentation - Flow Measurement and Control

[Section XV-C](#), Safety in Plant Design - Pressure Relief

OTHER LITERATURE

1. Powley, M. B., "Flow of Compressible Fluids," Can. J. Chem. Eng. 36, 241-245 (1958).
2. Perry, J. H., Chemical Engineer's Handbook, Seventh Edition, p. 2-183, 6-22-6-25 (1997).
3. Crane Co., Technical Paper No. 410, "Flow of Fluids Through Valves, Fittings and Pipe" (1988).
4. Lapple, C. E., Trans **AIChE** 39, 385-432 (1943).
5. Humble Pipeline Co. Report No. PLR-F-106-57, "Gas Transmission Handbook" (1967).
6. Bird, R. B., Stewart, W. E., and Lightfoot, E. N., Transport Phenomena, John Wiley and Sons, New York (1960).
7. Miller, R. W., Flow Measurement Engineering Handbook, Third Edition, McGraw Hill, New York (1996).
8. Levenspiel, O., A.I.Ch.E.J. v23, p. 402, (1977).

DEFINITIONS

See [Section XIV-A](#).

BASIC DESIGN CONSIDERATIONS

The considerations discussed below affect the basis for the calculation procedures given later in this subsection. Where indicated, Basic Design Considerations for liquid flow in [Section XIV-B](#) should be consulted.

COMPRESSIBLE FLOW CONSIDERATIONS

For incompressible fluids, the assumption of constant density simplifies the mass and momentum conservation laws as well as the resulting equations for pressure drop in piping systems. When the assumption of constant density is relaxed, the thermodynamic state of the fluid becomes an additional element of the analysis because the pressure and temperature of the flowing fluid are linked through an equation of state. In the context of these design practices, the major implication of the pressure-temperature coupling is that the Bernoulli equation must be modified to include changes in internal energy now possible as a result of compressibility effects. Recall that the Bernoulli equation was the starting point for all of the pressure drop calculation formulae presented in [Section XIV-B](#). In this section, corresponding formulae for compressible flow situations are presented based on the compressible form of the Bernoulli equation, [Equations \(1a\)](#) and [\(1b\)](#).

In addition to modifications of the pressure drop equations previously developed, the flow of compressible fluids may also introduce curious phenomena outside of the experience of the practicing engineer familiar with incompressible flows. Such phenomena include fluid acceleration due to friction, flow deceleration in a converging duct, development of abrupt changes in flow variables (shock waves), and choking flow resulting from the acceleration of gas flow through a constriction.

To understand choking, consider the one-dimensional flow of gas through a converging nozzle. A vacuum pump is attached to the nozzle outlet. For simplicity, it will also be assumed that the gas is inviscid and that the flow is isentropic. Under these conditions, the mass and energy conservation equations can be combined to yield the following relationship between cross-sectional area, A , fluid velocity, V , and Mach number, Ma . The Mach number represents the ratio of the fluid velocity to its local sonic velocity.

$$\frac{dA}{dV} = -\frac{A}{V}(1 - Ma^2)$$

When the suction of the pump is sufficiently small to ensure that the flow remains subsonic ($Ma < 1$), the above equation confirms the physical behavior already familiar for incompressible flows: the fluid is accelerated ($dV > 0$) through the converging nozzle ($dA < 0$). Larger suction pressures will increase the mass flow through the nozzle up to the sonic velocity limit ($Ma = 1$). At the sonic point, dA/dV is zero indicating the critical point corresponding to the minimum cross-sectional area in the nozzle. Attempts to increase the suction pressure further will not alter the acceleration in the converging duct. The maximum mass

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for this apparatus has been achieved and the flow is said to be choked. The choked flow condition is an important one for design and considerations specific to this flow condition are discussed in a subsequent section.

A similar phenomenon occurs for flow of gas through a long pipe of constant cross-section. Because of friction, the pressure falls in the flow direction, the gas density also decreases (equation of state), and hence the velocity increases. For sufficiently high upstream-to-downstream pressure ratios, the velocity will become sonic ($Ma = 1$) at the pipe exit. At this condition, further decreasing the downstream pressure will have no effect on the flowrate, and the flow is said to be choked. Increasing the upstream pressure while maintaining sonic flow at the exit will increase the flowrate.

GENERAL CONSIDERATIONS

In most piping designs, the primary requirement is to find an inside diameter that will permit a certain required throughput at a given pressure drop. This usually involves a trial and error procedure. A diameter is chosen and the pressure drop is calculated for the required throughput. If the calculated pressure drop is too great, a larger diameter is taken for the next trial. If the pressure drop is smaller than necessary, a smaller diameter is chosen.

Besides methods for calculating pressure drop at a given flow rate, the calculation procedures given later in this subsection also present methods for calculating flow rate at a given pressure drop. This is necessary since gas flow calculations are often complicated, especially in sonic flow.

Typical pressure drops that may be used for pipe sizing are shown in [Section XIV-B](#). In case of expensive construction materials, an economic analysis would be desirable to find the optimum line size.

PRINCIPLES OF PRESSURE DROP CALCULATION

The basic equation for calculating pressure drop for flow of gases through pipes and fittings are obtained through consideration of the steady-state energy balance:

$$K'_{1a} \Delta E + \frac{g}{g_c} \Delta z + K''_{1a} \Delta (P \underline{v}) + \frac{\Delta(V^2)}{2g_c} = K'_{1a} H - K'''_{1a} W_s \quad \text{Eq. (1a)}$$

and the differential form of the Bernoulli Theorem:

$$\frac{g}{g_c} dz + K'_{1a} \underline{v} dP + \frac{V}{g_c} dV = -K'''_{1a} dF - K'''_{1a} dW_s \quad \text{Eq. (1b)}$$

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
E	= Internal energy per unit mass	BTU/lbm	MJ/kg
F	= Frictional loss per unit mass	ft-lbf/lbm	kPa-m ³ /kg
g	= Standard acceleration of gravity	32.174 ft/s ²	9.807 m/s ²
g _c	= Gravitational conversion factor	32.174 (lbm/lbf)(ft/s ²)	1 (kg/N)(m/s ²)
H	= Heat added per unit mass	BTU/lbm	MJ/kg
P	= Absolute pressure	lbf/in ² , abs.	kPa, abs.
V	= Fluid velocity averaged over pipe cross-section	ft/s	m/s
$\underline{v}$	= Specific volume	ft ³ /lbm	m ³ /kg
W _s	= Shaft work	ft-lbf/lbm	kPa-m ³ /kg
z	= Elevation	ft	m
K'_{1a}	= Dimensional constant	778	10 ⁶
K''_{1a}	= Dimensional constant	144	10 ³
K'''_{1a}	= Dimensional constant	1	10 ³

The design methods presented in this subsection are based on these equations.

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Straight Horizontal Pipe

For the flow of gases in straight pipes, calculation of pressure drop for a given mass rate of flow is complicated by the dependence of gas density on pressure. Thus, for significant pressure drops, both the velocity and the density of the gas will change appreciably. As a result, to use the Bernoulli Theorem to develop expressions to predict pressure drop, the relationship between gas pressure and density in the pipe needs to be known, and the energy balance needs to be included in the analysis. In addition, the performance of such a line will depend on the type of flow condition existing in the line, usually assumed, for simplicity, to be either adiabatic or isothermal.

For the usual case in chemical plants and refineries of short insulated lines, heat transferred to or from the line is low, so that the flow is essentially adiabatic. Solution of the energy balance and the Bernoulli equations for the adiabatic case assuming an ideal gas yields the following equations:

$$\frac{P_2 v_2}{P_1 v_1} = \frac{T_2}{T_1} = \left[\frac{\left(1 - \left(\frac{v_2}{v_1} \right)^2 \right) G^2 v_1}{K_{2a} g_c P_1} \left(\frac{k-1}{2k} \right) + 1 \right] \quad \text{Eq. (2a)}$$

$$G = \left[\frac{K_{2a} g_c P_1}{v_1} \right]^{1/2} \left[\frac{\frac{4fL}{D} \frac{k+1}{2k} \ln \left(\frac{v_1}{v_2} \right)^2}{1 - \left(\frac{v_1}{v_2} \right)^2} \frac{k-1}{2k} \right]^{-1/2} \quad \text{Eq. (2b)}$$

where:

			Value/Dimensions	
			Customary	Metric
D	=	Diameter of pipe	ft	m
f	=	Fanning friction factor	dimensionless	dimensionless
G	=	Mass velocity	lbm/s-ft ²	kg/s-mm ²
k	=	Ratio of specific heats, C _p /C _v	dimensionless	dimensionless
L	=	Length of line	ft	m
T	=	Temperature	°R	°K
K _{2a}	=	Dimensional constant	144	10 ⁻⁹
subscripts 1, 2	=	Upstream and downstream location or condition, respectively		

Other terms are as defined for [Equations \(1a\)](#) and [\(1b\)](#).

Since the numerical evaluation of unknowns in the above equations for pressure drop is not straightforward, the assumption of adiabatic flow for refinery piping has been avoided in the past even though this is more accurate and preferred. However, in the calculation procedures that follow, a convenient graphical method of solving [Equations 2a](#) and [2b](#) is given for cases where either upstream pressure or downstream pressure is known.

These charts given in [Figures 3](#) and [4](#) are based on the methods presented by Lapple (Reference 4) and Powley (Reference 1). Perry (Reference 2) presents similar design charts taken from Levenspiel (Reference 8) which are also useful for finding allowable pipe-length for a given flow rate as well as for finding the discharge rate in a given piping system for adiabatic flow of an ideal gas. These charts may also be used for non-ideal gases, as long as the compressibility factor, Z, does not vary along the pipe. If it does, divide the line into sections, each with its average Z, and calculate the pressure drop for each section.

For long uninsulated lines, such as natural gas transmission lines, the flow will approach *isothermal* conditions. Solution of the basic equations assuming an ideal gas and isothermal flow per Bird, Stewart and Lightfoot (Reference 6) yields:

$$\frac{P_2}{P_1} = \left[1 - \frac{v_1 G^2}{K_{2a} g_c P_1} \left(\frac{4fL}{D} \right) \left[1 - \frac{D}{2fL} \ln \left(\frac{P_2}{P_1} \right) \right] \right]^{-1/2} \quad \text{Eq. (3a)}$$

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The equation can be rearranged to solve for the mass flow rate:

$$G = \left[\frac{\left(\frac{K_{2a} g_c P_1}{v_1} \right) \left[1 - \left(\frac{P_2}{P_1} \right)^2 \right]}{\frac{4fL}{D} - \ln \left(\left(\frac{P_2}{P_1} \right)^2 \right)} \right]^{1/2} \quad \text{Eq. (3b)}$$

(with all terms in [Equations \(3a\)](#) and [\(3b\)](#) as previously defined).

[Equation \(3b\)](#) is easily solved to determine the flow rate if the upstream and downstream pressures are known, but [Equation \(3a\)](#) requires a trial-and-error solution if only one of these pressures is known and the pressure drop is to be calculated. The charts given in [Figures 3A, D](#) and [Figures 4A, D](#) where $k = 1$, may be used for a graphical solution of the isothermal case. The calculation procedure is then similar to that for the adiabatic flow condition and requires calculation of the dimensionless total pipe resistance coefficient, N , defined by

$$N = 4 f \frac{L}{D} \quad \text{Eq. (3c)}$$

with all terms as previously defined.

[Equation \(3a\)](#) can be simplified under certain flow conditions. For long pipelines, the last term in the equation approaches unity (except for the unusual case of very high pressure drop) and the equation can be simplified to:

$$\frac{P_1^2 - P_2^2}{P_1 v_1} = \frac{4 f L G^2}{K_{2a} g_c D} \quad \text{Eq. (3d)}$$

with all terms as previously defined.

This is the basic form for the Weymouth formula or the Panhandle equation for gas transmission lines. The Humble Pipeline Company developed a version of this equation that is available for use in the "Gas Transmission Handbook" (Reference 5). If gas pipeline designs are to be developed, please refer to this report or consult ER&E fluid flow specialists.

For flow systems where the pressure drop is less than 10% of the upstream pressure, [Equation \(3a\)](#) can be very roughly approximated by:

$$\frac{P_1 - P_2}{L} = \frac{2f v_{avg} G^2}{K_{2a} g_c D} \propto \frac{f W^2}{D^5 \rho} \quad \text{Eq. (3e)}$$

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
v_{avg}	= Average specific volume of fluid in the pipeline	ft ³ /lbm	m ³ /kg
W	= Fluid mass flow rate	klbm/hr	kg/s
ρ	= Average fluid density in the pipeline	lbm/ft ³	kg/m ³

and all other terms are as previously defined.

This equation is the basis for the simplified method of estimating pressure drop for gas flow across single piping components and [Tables \(2A\)](#) and [\(2B\)](#) for straight pipe pressure drop calculations.

Temperature Change

As a result of adiabatic expansion, the temperature of a gas flowing through a pipe will gradually decrease. This temperature change will be substantial at high flow rates, as can be seen in [Figures 3B, C, E, F](#), and [4B, C, E, F](#) where curves for a constant ratio of downstream and upstream temperature are plotted.

Vertical and Inclined Pipe

Generally the effect of gravity head is negligible for gas flow. It should, however, not be neglected when the flow rate is small and the gas density is high. For example, gravity head should be included in the design of stacks.

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Bends, Tees, and Ys

Procedures are given in this subsection for calculating pressure drop due to compressible flow through bends, tees, or Ys either as single components or as part of a piping system.

For bends, pressure drop is calculated using [Equations \(7a-d\) of Section XIV-B](#) with the resistance factor, K, determined from [Figures 5C and 5D of that subsection](#). As part of a piping system, the bend resistance coefficients are added to the pipe resistance coefficient, N, used in [Figures 3 and 4](#).

For blanked-off tees, the same procedure should be used as for bends. For tees in which streams are split or joined, [Equations \(8a-f\) of Section XIV-B](#) should be applied. Whereas, appropriate equations from Reference 8 of [Section XIV-B](#) should be used for Ys, and Reference 9 of [Section XIV-B](#) should be applied for manifold designs.

Valves

Pressure drop due to subsonic flow through valves in a piping system is calculated in a similar manner to that described above for bends. [Figure 5 in Section XIV-B](#) can be used to determine the valve resistance coefficient, K, or the equivalent length of straight pipe. The pressure drop across the line containing the valve can be estimated using [Figures 3 and 4](#) with the valve resistance coefficient added to the pipe resistance coefficient, N. Alternatively, the calculational procedures for horizontal straight pipe can be used with the equivalent length added to the length of straight pipe for these methods.

If the valve port has a substantially smaller (<~80%) cross-sectional area than that of the line, the mass velocity in the valve port must be calculated and compared to the mass velocity in the line and sonic mass velocity to assure that the flow across the valve is subsonic.

Use of [Design Practices Section XII](#), [Section XIV-B](#), Instrumentation, Flow Measurement and Control is recommended for designing flow control or metering valves, and [Section XV](#), [Section XV-C](#), Safety in Plant Design, Pressure Relief is recommended for safety valve design or evaluation.

Venturis, Flow Nozzles, and Orifices

Fluid density is not constant for compressible flow across venturis, flow nozzles, and orifices. As the gas pressure decreases to accommodate the increase in kinetic energy, the density decreases, and the assumption of constant fluid density, as used in [Section XIV-B](#), no longer applies. A dimensionless gas expansion factor, Y, is used to correct for gas expansion due to this density change. For venturis and flow nozzles the adiabatic gas expansion factor is defined by:

$$Y = \sqrt{\frac{1 - \beta^4 \left(\frac{k}{k-1} \right) \left(\frac{P_2}{P_1} \right)^{2/k} \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \right]}{\left(1 - \beta^4 \left(\frac{P_2}{P_1} \right)^{2/k} \right) \left(1 - \left(\frac{P_2}{P_1} \right) \right)}} \quad \text{Eq. (4)}$$

where β is the ratio of throat to the line diameter and all other terms are as previously defined. [Figure 5](#) contains a graphical presentation of this relationship for flow nozzles or venturis. At flow nozzles, venturis, and other contoured inlet flow restrictions, gas expansion is assumed to be purely axial and there is excellent agreement between Y determined by [Equation \(4\)](#) or [Figure 5](#) and laboratory data.

At a square- or sharp-edged orifice, expansion is both radial and axial and empirically determined relations are used to determine the gas expansion factor. For square- or sharp-edged orifices, Miller (Reference 7) recommends that Y be calculated using the following relations:

$$Y = 1 - \left(0.41 + 0.35\beta^4 \right) \frac{\Delta P}{kP_1} \quad \text{for } \frac{\Delta P}{P_1} < 0.37 \quad \text{Eq. (5a)}$$

$$Y = Y_{0.37} - 0.35 \left(\frac{\Delta P}{P_1} - 0.37 \right) \quad \text{for } \frac{\Delta P}{P_1} > 0.37 \quad \text{Eq. (5b)}$$

where ΔP is the pressure drop across the restriction, $Y_{0.37}$ is the value of Y calculated using [Equation \(5a\)](#) with $\Delta P/P_1 = 0.37$ and all other terms are as previously defined. For restriction orifices, Miller (Reference 7) recommends using the following relation to calculate Y when $\Delta P/P_1 > 0.23$:

$$Y = Y_{0.23} - 0.364 \left(\frac{\Delta P}{P_1} - 0.23 \right) \quad \text{for Restriction Orifices } \frac{\Delta P}{P_1} > 0.23 \quad \text{Eq. (5c)}$$



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where $Y_{0.23}$ is the value of Y calculated using Equation (5a) with $\Delta P/P_1 = 0.23$. Figure 5 may also be used to estimate the value of Y for square- and sharp-edged orifices.

Pressure drop across flow nozzles, venturis, and orifices is calculated using the following relations:

$$\Delta P = K_{6a} \left[\frac{Q_1^2}{v_1 C^2 Y^2 d_o^4} \right] \quad \text{Eq. (6a)}$$

$$\Delta P = K_{6b} \left[\frac{v_1 W^2}{C^2 Y^2 d_o^4} \right] \quad \text{Eq. (6b)}$$

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
C	= Flow coefficient obtained from Section XIV-B: Flow Nozzles-Figure 8; Venturis-Equation (10); Orifices; Figures 7A and 7B	dimensionless	dimensionless
d_o	= Throat or orifice diameter	inches	mm
Q_1	= Upstream volumetric flow rate	ft ³ /min	dm ³ /s
K_{6a}	= Dimensional constant	10 ⁻³	810
K_{6b}	= Dimensional constant	0.28	8.1 x 10 ⁸

and all other terms have been previously defined.

Since the gas expansion factor, Y , is a function of the pressure drop, a trial and error procedure is required to accurately calculate the pressure drop across the restriction using Equation (6a) or (6b).

The total pressure drop due to the nozzle, venturi, or orifice includes the pressure recovery downstream of the flow restriction and is obtained by multiplying the ΔP calculated using Equation (6a) or (6b) by the pressure recovery factor, r , obtained from Figure 10 in Section XIV-B.

Expansions and Contractions

Pressure drop calculations for expansions and contractions of compressible fluids are the same as those described in Section XIV-B for incompressible fluids except that when the total pressure drop (the sum of pressure drop due to friction and that due to change in kinetic energy) is larger than 10% of the total pressure drop, or when a more accurate estimate is needed, the kinetic energy term should include the expansion factor, Y as indicated below:

$$\Delta P_k = K_{7a} \frac{Q_1^2}{v_1^2 Y^2} \left(\frac{v_2}{d_2^4} - \frac{v_1}{d_1^4} \right) \quad \text{Eq. (7a)}$$

or

$$\Delta P_k = K_{7b} \frac{W^2}{Y^2} \left(\frac{v_2}{d_2^4} - \frac{v_1}{d_1^4} \right) \quad \text{Eq. (7b)}$$

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
d_1, d_2	= Upstream and downstream inside diameters or equivalent hydraulic diameters	inches	mm
ΔP_k	= Pressure drop due to kinetic energy change	lbf/in. ²	kPa
K_{7a}	= Dimensional constant	10 ⁻³	810
K_{7b}	= Dimensional constant	0.28	8.1 x 10 ⁸

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and all other terms are as previously defined. Either [Figure 5](#) or [Equation \(4\)](#), [\(5A\)](#), or [\(5b\)](#) can be used to calculate the gas expansion factor depending on whether the entrance region is contoured or abrupt. Since Y depends on the pressure drop, a trial and error procedure is needed to determine the total pressure drop.

Perforated-Pipe Distributors

The same procedures used in [Section XIV-B](#) for incompressible fluids should be used for perforated - pipe distributor design with compressible fluids with the exception that to calculate the total hole area, the expansion factor Y from [Figure 5](#) or [Equation \(5a\)](#) or [\(5b\)](#) should be included as indicated below:

$$A_o = K_{8a} \frac{Q_1}{C Y} \sqrt{\frac{\rho_1}{\Delta P_o}} \quad \text{Eq. (8a)}$$

or

$$A_o = K_{8b} \frac{W}{C Y} \left(\frac{1}{\sqrt{\rho_1 (\Delta P_o)}} \right) \quad \text{Eq. (8b)}$$

where:

		Value/Dimensions	
		Customary	Metric
A_o	= Total required hole area in perforated-pipe distributor	in. ²	mm ²
C	= Flow coefficient determined using procedures described in Section XIV-B	dimensionless	dimensionless
ΔP_o	= Pressure drop across openings	lbf/in. ²	kPa
ρ_1	= Gas density in the distributor	lbm/ft ³	kg/m ³
K_{8a}	= Dimensional constant	2.49×10^{-2}	22.4
K_{8b}	= Dimensional constant	0.415	2.24×10^4

and all other terms are as previously defined.

SONIC VELOCITY AND CRITICAL FLOW

When a gas accelerates through a restriction, its density decreases and its velocity increases. Since the mass flow per unit area (mass velocity) is a function of both the density and velocity, a critical cross-sectional area exists at which the mass velocity is at a maximum. In this area the velocity is sonic and further decreasing the downstream pressure will not increase the mass flow. This is referred to as choked or critical flow. This condition usually occurs at a restriction in the line or at the exit of a pipe entering an area of larger cross-section. The sonic mass velocity, G_{hs} , and sonic velocity V_s , can be calculated for an ideal gas by using the following relations:

$$G_{hs} = K_{9a} \sqrt{k P_2 \rho_2} \quad \text{Eq. (9a)}$$

$$V_s = K_{9b} \sqrt{\frac{k P_2}{\rho_2}} \quad \text{Eq. (9b)}$$

where:

		Value/Dimensions	
		Customary	Metric
G_{hs}	= Sonic mass velocity	klbm/h-in. ²	kg/s-mm ²
P_2	= Restriction exit absolute pressure	lbf/in. ² , abs.	kPa, abs.
V_s	= Sonic velocity	ft/s	m/s
ρ_2	= Fluid density at exit	lbm/ft ³	kg/m ³
K_{9a}	= Dimensional constant	1.70	3.16×10^{-5}
K_{9b}	= Dimensional constant	68.1	31.6

with all other terms as previously defined.



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Critical Flow in Piping

The critical pressure drop in a piping system is indicated in [Figure 3](#) by the intersection of the dashed line curves with curves for constant total pipe resistance coefficient, N . The mass velocity for critical or choked flow can also be calculated using these figures. There is no pressure drop recovery for critical flow operations.

Critical Flow in Flow Nozzles and Venturis

For a convergent-divergent passage through which a compressible fluid accelerates, as the downstream pressure decreases, the throat velocity increases until a critical pressure ratio is reached at which the throat velocity is sonic. Further decreases in the downstream pressure will not increase the mass flow rate. The flow through the passage is referred to as subsonic down to the critical pressure ratio and critical below the ratio. In critical flow the throat velocity is always sonic, but the velocity increases in the diffuser section, where a normal shock front occurs. Across the shock front there is a discontinuous pressure increase and velocity decrease, together with discontinuous changes in temperature, density and entropy. The minimum downstream pressure at which critical flow begins is dependent on geometry and the specific heat ratio, k of the fluid.

For flow without a diffuser section, such as through long radius ASME flow nozzles or thick, square-edged orifices, the critical downstream pressure, P_{CF} , can be calculated for an ideal gas from the relation:

$$P_{CF} = \frac{P_1 \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}}}{\left(1 - \frac{k}{2} \left(\frac{2}{k+1} \right)^{\frac{k+1}{k-1}} \beta^4 \right)} \quad \text{Eq. (10)}$$

where all other terms in the expression are as previously defined. These sonic conditions are also indicated by the end points on the curves for flow nozzles or venturi meters shown in [Figure 5](#). These relations can be used to determine the choking pressure drop for compressible flow through a nozzle at a known upstream pressure.

For a venturi nozzle the ratio of the downstream to upstream pressure at choking (the choking pressure ratio) may vary from 0.8 to 0.95 depending on the diffuser area ratio.

Flow coefficients for critical flow nozzles and venturis depend on geometry and the Reynolds number in the throat Re_o . Some applicable relations for different geometries where β , d_o/d , is less than 0.25 have been suggested by Miller (Reference 9) and are presented below.

ASME long radius flow nozzles and ASME-ISO cylindrical throat venturis:

$$C = \left(1.00 - \frac{7.21}{Re_o^{0.5}} \right) (1 - \beta^4)^{-0.5} \quad \text{for } 10^4 < Re_o < 3.5 \times 10^5 \quad \text{Eq. (11)}$$

$$C = 0.9887 (1 - \beta^4)^{-0.5} \quad \text{for } 3.5 \times 10^5 < Re_o < 2.5 \times 10^6 \quad \text{Eq. (12)}$$

$$C = \left(1.00 - \frac{0.222}{Re_o} \right) (1 - \beta^4)^{-0.5} \quad \text{for } 2.5 \times 10^6 < Re_o < 2 \times 10^7 \quad \text{Eq. (13)}$$

ASME-ISO toroidal throat venturi:

$$C = \left(0.99354 - \frac{1.525}{Re_o^{0.5}} \right) (1 - \beta^4)^{-0.5} \quad \text{for } 10^5 < Re_o < 10^7 \quad \text{Eq. (14)}$$

where all terms in these relations have been defined previously.

Critical Flow Through Orifices and Valves

Choked or critical flow does not occur across a standard, thin, square-edged orifice. For choking to occur, the discharge coefficient must approach unity. However, choked flow is achieved when the plate thickness is increased. For plate thickness exceeding the diameter of the orifice, the choking pressure drop for isentropic flow across the orifice, ΔP_{CF} , can be calculated by rearranging [Equation \(10\)](#) as:

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$$\Delta P_{CF} = P_1 \left[1 - \left(1 - \frac{k}{2} \left(\frac{2}{k+1} \right)^{\frac{k+1}{k-1}} \beta^4 \right)^{-1} \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \right] \quad \text{Eq. (15)}$$

For sharp, square-edged orifices with a ratio of plate thickness to bore diameter between 1 and 6, Miller (Reference 7) suggests that the critical flow coefficient can be determined from the relation:

$$C = 0.83932 (1 - \beta^4)^{-0.5} \quad \text{Eq. (16)}$$

where all terms are as previously defined.

Restriction orifices are generally thick, square-edged orifices. Valves having a port area of less than about 80% of the line cross-sectional area can also be considered as thick, square-edged orifices for critical flow calculations.

CALCULATION PROCEDURES

The following design methods, equations, and guidelines must be used together with the material given above under “**BASIC DESIGN CONSIDERATIONS.**” The first section below presents procedures for calculating pressure drop due to single components. The second section should be used for calculating pressure drop in flow systems containing more than one component.

PRESSURE DROP ACROSS SINGLE PIPING COMPONENTS

For calculating pressure drop across or flow rate through single piping components, use the following procedures (Note that absolute pressures are used for all calculations under this heading):

Straight Horizontal Pipe

Use the following procedures for straight pipe not containing any fittings.

- 1) Simplified Method - If the calculated pressure drop ($P_1 - P_2$) is less than about 10% of the inlet pressure, P_1 , reasonable accuracy will be obtained using [Figures 1A or 1B](#) or the following simplified form of the Fanning pressure drop equation:

$$\Delta P_L = K_{17} (C_2 W^2 \underline{v}) = K_{17} \left(C_2 \frac{W^2}{\rho} \right) \quad \text{Eq. (17)}$$

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
C_2	= Function of f and d ; ($C_2 \propto f/d^5$) C_2 is given as a function of pipe size in Tables 2A and 2B	-	-
ΔP_L	= Pressure drop per length of line	psi/100 ft	kPa/m
K_{17}	= Dimensional constant	10^{-3}	0.23

and all other terms are as previously defined. The specific volume of the gas $\underline{v}$, can be evaluated at either upstream or downstream conditions, whichever are known. For steam, values of $\underline{v}$ may be obtained from [Figures 2A or 2B](#).

The procedure is as follows:

- a) For a given flow rate and pipe size -- obtain the value of C_2 from [Table 2A or 2B](#) for the pipe size specified and calculate ΔP_L using [Equation \(17\)](#).
 - b) For a given pressure drop and pipe size -- obtain the value of C_2 from [Table 2A or 2B](#) for the pipe size specified and calculate W from [Equation \(17\)](#) by substitution.
- 2) Modified Simplified Method - If the calculated pressure drop ($P_1 - P_2$) is greater than 10% but less than about 40% of the inlet pressure, P_1 , [Equation \(17\)](#) or [Figures 1A or 1B](#) can still be used with reasonable accuracy provided that the specific volume



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used in the calculation is based on the average of the upstream and downstream conditions. A trial and error solution should be made until the $\bar{v}$ used in the trial is equal to the average of $\bar{v}_1$ and $\bar{v}_2$.

- 3) Detailed Graphical and Computational Methods - A more accurate pressure drop can be determined by use of the methods outlined below, but this is usually not necessary if the pressure drop range is between 10% and 40% of the inlet pressure. If in doubt, however, use one of the detailed methods presented below.

a) Flow rate and upstream pressure are known.

- (1) For non-circular ducts calculate the equivalent hydraulic diameter, d_{eq} .

$$d_{eq} = 4 \times \left(\frac{\text{Cross-sectional Area}}{\text{Duct Perimeter}} \right), \text{ in consistent units} \quad \text{Eq. (18)}$$

Note: As discussed in [Section XIV-B](#), no geometric analogy exists between d_{eq} and the circular pipe diameter, d other than the fact that each represents a measure of the size of the cross section. Therefore, d_{eq} **cannot** be used to calculate velocity, V , from the flow rate, Q . However, d_{eq} can be used to calculate the Reynolds number from Eq. (19a) and, for fully-turbulent flows, the frictional pressure drop.

- (2) Calculate the pipe Reynolds number, Re :

$$Re = \frac{DV\rho}{\mu} = K_{19a} \left(\frac{DV\rho}{\mu'} \right) \quad \text{Eq. (19a)}$$

$$= K_{19b} \left(\frac{q' S_g}{d \mu'} \right); \text{ valid only for circular cross-sections} \quad \text{Eq. (19b)}$$

$$= K_{19c} \left(\frac{W}{d \mu'} \right); \text{ valid only for circular cross-sections} \quad \text{Eq. (19c)}$$

where:

		Value/Dimensions	
		Customary	Metric
d	= Inside diameter of pipe or equivalent diameter	inches	mm
D	= Inside diameter of pipe or equivalent diameter	ft	m
q'	= Volumetric (molar) flow rate	std. ft ³ /h @60°F and 14.7 psia	std. dm ³ /s @15°C and 101.3 kPa
Re	= Pipe Reynolds number	dimensionless	dimensionless
S_g	= Specific gravity of gas relative to air both at std. conditions (ratio of gas molecular weight to molecular weight of air)	dimensionless	dimensionless
V	= Linear gas velocity averaged over pipe cross-section	ft/s	m/s
μ	= Dynamic viscosity of gas	lbm/ft-s	Pa-s
μ'	= Dynamic viscosity of gas	cP	Pa-s
K_{19a}	= Dimensional constant	1488	1
K_{19b}	= Dimensional constant	0.482	1.56
K_{19c}	= Dimensional constant	6.31×10^3	1.27×10^3

and all other terms are as previously defined.

- (3) Find the friction factor, f , from [Figure 1 or 2 of Section XIV-B](#).
 (4) Find k , the ratio of specific heats of the gas, from [Table 1](#) or the isentropic exponent from the QUEST/DISTEX program or PEGASYS.

Either a graphical or computational approach can be used for the remainder of the procedure.

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For the graphical approach:

- (5) Calculate the pipe resistance coefficient, N , using Equation (3c). If $N > 400$, divide the line into shorter sections and calculate the pressure drop of individual sections, starting from the upstream end.
- (6) Calculate the mass velocity, G_h , and then the term $G_h^2/P_1\rho_1$.

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
G_h	=	Mass velocity of gas	
		lbm/h-in ²	kg/s-mm ²

and all other terms are as previously defined.

- (7) Use value of k determined in Step (4) for adiabatic flow, which is the usual situation in refinery or chemical plant piping. For isothermal flow, as in gas transmission lines, use $k = 1$.
- (8) Find $\Delta P/P_1$ from Figure 3. For values of k and N falling between those given in the charts, use linear interpolation where the curves are rather straight and graphical interpolation where they run upward. Remember that the pressures in these charts are absolute pressures. When the value of $G_h^2/P_1\rho_1$ falls below the range covered by the charts, the gas may be treated as an incompressible fluid. In this case use the procedure for liquid flow given in Section XIV-B.

When the value of $G_h^2/P_1\rho_1$ falls on the vertical portion of the N -curve in Figure 3, the gas velocity at the pipe end will be sonic. The pressure drop then consists of two parts: pressure drop across the pipe, given by the point where the N -curve in Figure 3 crosses the dashed curve marking the sonic flow region boundary, and pressure drop across the shock wave at the pipe outlet. The latter is determined by the pressure in the downstream equipment.

When the calculated value of $G_h^2/P_1\rho_1$ falls beyond the vertical portion of the N -curve in Figure 3, this represents a physically impossible situation which requires either raising P_1 or increasing the pipe diameter to make possible the desired flow rate.

- (9) Finally, calculate ΔP from P_1 and the value obtained for $\Delta P/P_1$.

For the alternative computational method:

- (5) Adiabatic flow typical in refinery and chemical piping requires a trial-and-error solution of Equations (2b) and (2a), while isothermal operations like gas transmissions lines, require a trial-and-error solution of Equation (3b).
 - (6) For the adiabatic flow condition, determine by trial-and-error the value of v_2 required in Equation (2b) to obtain a value G equal to that desired. This value of v_2 should be used in Equation (2a) to determine P_2 . Calculate ΔP by determining $P_1 - P_2$.
 - (7) For isothermal conditions, use of trial-and-error solution of Equation (3a) with various values of P_2 on the right hand side of the equation is required until the calculated value of P_2/P_1 is such that when it is multiplied by P_1 , the resultant value of P_2 agrees with the trial value. Calculate ΔP as $P_1 - P_2$.
 - (8) Check for sonic flow. Once the values of P_2 and v_2 have been determined G_{hs} should be calculated using Equation (9a) and the result compared with the value of G_h . If G_h equals G_{hs} , sonic flow occurs at the end of the pipeline, and any additional reduction in downstream pressure will not increase the mass flow rate. Therefore, G_h cannot be greater than G_{hs} , and if the calculations shows that G_h is greater than G_{hs} , this is an impossible condition indicating that either P_1 or the pipe line size need to be increased to achieve the desired mass flow rate.
- b) If flow rate and downstream pressure are known, use the following procedure to find the pressure drop:
- (1) For non-circular ducts, calculate the equivalent hydraulic diameter from Equation (18).
 - (2) Calculate the pipe Reynolds number, Re , from Equation (19a, b, or c), using the values of density and dynamic viscosity at the upstream temperature and known downstream pressure.
 - (3) Find the friction factor, f , from Figure 1 or 2 of Section XIV-B.
 - (4) Find k , the ratio of specific heat capacities of the gas from Table 1 or the isentropic exponent from the QUEST/DISTEX program or PEGASYS.

Using the graphical method:

- (5) Calculate the pipe resistance coefficient, N from Equation (3c). If $N > 400$, divide the line into shorter sections and calculate the pressure drop of the individual sections, starting from the downstream end.
- (6) Calculate the mass velocity, G_h , and the term $G_h^2/P_2\rho_2$, where P_2 is the downstream absolute pressure and ρ_2 is the downstream density calculated at the upstream temperature.

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- (7) Find $\Delta P/P_1$ using [Figure 4](#). For values of k and N falling between those given in the charts, use linear interpolation where the curves are rather straight, and graphical interpolation where they are curved. Remember that the pressures used on these charts are absolute pressures.

When the value of $G_h^2/P_2\rho_2$ falls below the range covered by the charts, treat the gas as an incompressible fluid and use the procedure for liquid flow given in [Section XIV-C](#).

When the value of $G_h^2/P_2\rho_2$ falls on or beyond the right hand side of any of the charts in [Figure 4](#), the gas velocity is sonic, and a high pressure drop will exist at the pipe outlet. To calculate the pressure drop in this case, use the following procedure:

- i) Find the value of $G_h^2/P_1\rho_1$ from the vertical portion of the curve in [Figure 3](#) corresponding to the values of k and N calculated above.
- ii) Calculate $P_1\rho_1$ from this value of $G_h^2/P_1\rho_1$ and the given value of G_h .
- iii) Find P_1/ρ_1 from the following equation, which is derived from the equation of state ($P_v = ZnRT$):

$$\frac{P_1}{\rho_1} = R \left(\frac{Z T_1}{M} \right) \quad \text{Eq. (20)}$$

where:

		Value/Dimensions	
		<u>Customary</u>	<u>Metric</u>
M	= Molecular weight	-	-
P_1	= Upstream pressure	psia, abs.	kPa, abs.
R	= Gas constant	10.73	8.314
		psia-ft ³ /lb mol-°R	kJ/kg mol-°K
T_1	= Upstream temperature	°R	°K
Z	= Compressibility factor	dimensionless	dimensionless
ρ_1	= Upstream density	lbm/ft ³	kg/m ³

- iv) Calculate P_1 from the following equation:

$$P_1 = \sqrt{\left(\frac{P_1}{\rho_1} \right) (\rho_1 P_1)} \quad \text{Eq. (21)}$$

- v) Calculate $\Delta P = P_1 - P_2$.

- (8) In all cases except where $G_h^2/P_2\rho_2$ falls beyond the right-hand side of [Figure 4](#), calculate ΔP from the following equation (all terms as previously defined):

$$\Delta P = P_2 \left[\frac{\Delta P / P_1}{1 - \Delta P / P_1} \right] \quad \text{Eq. (22)}$$

Using the computational method:

- (5) First test for a choked flow or critical flow condition by using [Equation \(9a\)](#) to determine G_{hs} . If G_h is greater than or equal to G_{hs} , there is a critical flow condition in the pipeline. The graphical method outlined in Step (7) above is then recommended for determining the upstream pressure for adiabatic flow. The computational method is complicated and involves determination of the choked flow pressure and density using [Equations \(9a\)](#) and [Equation \(20\)](#) and substituting these values as the downstream pressure and density into [Equations \(2a\)](#) and [\(2b\)](#). The upstream pressure and density would then be determined by a complex trial-and-error procedure or other computational solution method. For a choked isothermal flow condition, the computational method is more easily applied. The critical flow pressure and density are determined by simultaneous solution of [Equations \(9a\)](#) and [\(20\)](#). The resultant pressure is substituted for the downstream pressure, P_2 , in [Equation \(3a\)](#) and a trial-and-error solution for P_1 is performed until the calculated and assumed value of the upstream pressure converge.
- (6) For subsonic adiabatic flow, the computational method is difficult to apply because it involves simultaneous solution of two complex equations [[Equations \(2a\)](#) and [\(2b\)](#)] for both P_1 and v_1 -- unless both the downstream and upstream temperatures are known. If both temperatures are known, then [Equation \(2a\)](#) can be rearranged as:

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$$P_1 = P_2 \frac{v_2}{v_1} \left(\frac{T_2}{T_1} \right) \quad \text{Eq. (23)}$$

This relation can be substituted for P_1 into Equation (2b) and a trial-and-error procedure can be used to determine the value of v_1 needed to match the given value of G for the condition. The determined value of v_1 can be substituted in to Equation (23) to determine P_1 .

- (7) For subsonic isothermal flow, a trial-and-error solution of Equation (3a) can be used to calculate P_1 . In this procedure, Equation (20) is rearranged to express v_1 as a function of P_1 :

$$v_1 = \frac{R}{P_1} \left(\frac{Z T}{M} \right) \quad \text{Eq. (24)}$$

which is substituted into Equation (3a). Trial values of P_1 are substituted into the right hand side of the equation and the calculated value of P_2 from the left hand side is compared to the actual value of P_2 . The value of P_1 is the trial value that results in calculation of the actual value of P_2 .

- c) If **pressure drop** is given and an accurate estimate of flow rate is wanted, either graphical or computational methods may be used as follows:
- (1) When $\Delta P/P_1 < 0.10$, treat the gas as an incompressible fluid and use the simplified form of the Fanning friction pressure drop equation, Equation (17). When $\Delta P/P_1$ is greater than or equal to 0.10, proceed as described below.
 - (2) For non-circular ducts, calculate the equivalent hydraulic diameter from Equation (5).
 - (3) Find k , the ratio of specific heat capacities of the gas, from Table 1 or the isentropic exponent from the QUEST/DISTEX program or PEGASYS.
 - (4) For a first trial, use Figure 2 in Section XIV-B to obtain the friction factor, f , for complete turbulence for the given pipe size; the complete turbulence region of the plots in Figure 2 corresponds to the region where the friction factor curves become horizontal at high Reynolds number. The required pipe diameter, also required to determine the friction factor, is obtained from the smaller plots found on the right side of Figure 2, Section XIV-B.

Using the graphical approach:

- (5) Calculate the pipe resistance coefficient, N , from Equation (3c). If $N > 400$, divide the line into shorter sections, with estimated pressure drops for each, and calculate the flow rate in each section following the instructions given below. Check if the flow rates in the various sections are equal. If not, modify the pressure drop estimates, and try again.
- (6) Calculate $\Delta P/P_1$ and find $G_h^2/P_1 \rho_1$ from Figure 3 (Note: use $k = 1$ for isothermal flow). For values of k and N falling between those given in the charts, use linear interpolation where the curves are rather straight and graphical interpolation where they turn upward.

When the value of $G_h^2/P_1 \rho_1$ falls below the range covered by the charts, treat the gas as an incompressible fluid and use the procedure for liquid flow given in Section XIV-B.

- (7) Calculate G from the known values of P_1 and ρ_1 and the value obtained for $G_h^2/P_1 \rho_1$.
- (8) Calculate the mass flow rate, W , from G and the pipe cross-sectional area.
- (9) Calculate the Reynolds number, Re , from Equation (19c) and check Figures 1 or 2 of Section XIV-B to see if the assumed value for the friction factor was correct. If the difference is more than 10%, repeat the above calculation Steps (4) - (9) with the new value of the friction factor.

Using the computational method:

- (5) For adiabatic flow, solve for G and v_2 using Equations (2a) and (2b).
- (6) For isothermal flow solve for G using Equation (3b).
- (7) Check that G_h calculated from the value of G determined above is less than G_{hs} calculated from Equation (9A). If the value of G_h is equal to or greater than the value of G_{hs} , a critical or choked flow condition exists in the line and G_h must therefore equal G_{hs} .
- (8) Calculate the mass flow rate, W , from G and the pipe cross-sectional area.
- (9) Calculate the Reynolds number, Re , from Equation (19c) and check Figures 1 or 2 of Section XIV-B to see if the assumed value for the friction factor was correct. If the difference is more than 10%, repeat the above calculation Steps (4) - (9) with the new value of the friction factor.

Straight Inclined or Vertical Pipe

When pressure drop by friction and acceleration as calculated above for horizontal pipe is low, (e.g., in stacks), a pressure drop due to change in elevation must be included. Calculate this pressure drop from Equation (6) of Section XIV-B. When flow rate

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must be calculated at a known pressure drop, first subtract the elevation pressure drop from the given pressure drop. Then find the flow rate using the remaining pressure drop in Procedure 3c, above.

Bends

Use the following procedure:

- (1) Find the resistance coefficient K from [Figures 5C or 5D in Section XIV-B](#).
- (2) Calculate the pressure drop (or flow rate) from [Equation \(7\) in Section XIV-B](#).
- (3) If the pressure drop (ΔP) is larger than 10% of the absolute pressure, or if an accurate estimate is needed, proceed as if the bend were a straight piece of horizontal pipe with a pipe resistance coefficient N equal to K or an equivalent length of straight pipe, L_e , defined by

$$L_e = \frac{d K}{48 f} = \frac{D K}{4 f} \quad \text{Eq. (25)}$$

For this purpose, use the procedures given above for straight horizontal pipe. Note that steps for calculating the pipe Reynolds number, Re , or the Fanning friction factor, f , are not required for calculating single component bend pressure drop.

Tees and Y's

For blanked-off tees, use the same procedure as for bends. For tees in which streams are split or joined, use [Equations \(8a - f\) of Section XIV-B](#). For Y's and manifolds, see [References 8 & 9 of Section XIV-B](#).

Valves

Use the same procedure as for bends. If the valve port has a substantially smaller (say, < 80%) cross-sectional area than that of the line, calculate the mass velocity G_h in the valve port and compare this with the sonic mass G_{hs} , calculated from [Equation \(9a\)](#).

If G_h turns out be larger than G_{hs} , use the procedure for sonic flow in thick, square-edged orifices presented below. Assume an orifice diameter, d_o , giving the same cross-sectional area as that of the valve port, and (provided that it is not known from other sources) use flow coefficient given by [Equation \(16\)](#).

Orifices

Use the following procedure:

- 1) Calculate pressure drop (or flow rate) using the procedure for liquid flow in [Section XIV-B](#).
- 2) If the pressure drop turns out to be larger than 10% of the upstream absolute pressure, or if an accurate estimate is needed, proceed as follows:
 - a) If the upstream pressure, P_1 , upstream density, ρ_1 , mass flow rate, W , orifice diameter, d_o , and upstream pipe diameter, d_1 , are given, and the pressure drop, ΔP , is to be determined, use the following procedure:
 - (1) Calculate the Reynolds number, Re , in the upstream piping from [Equation \(19c\)](#). Calculate d_o/d_1 , and find the flow coefficient C from [Figure 7A or 7B of Section XIV-B](#).
 - (2) Calculate the pressure drop from [Equation \(6b\)](#) using $Y = 1$.
 - (3) Calculate $\Delta P/P_1$, find the ratio of specific heat capacities, $k = C_p/C_v$ from [Table 1](#) or the isentropic exponent from the QUEST/DISTEX program or PEGASYS, and find the expansion factor Y from [Figure 5](#) or [Equation \(5a\)](#) or (5b).
 - (4) Calculate a new value of ΔP from [Equation \(6b\)](#), recalculate $\Delta P/P_1$ obtain a new value for Y , and calculate a new ΔP . Repeat, if needed, until convergence is obtained on ΔP .
 - (5) Check for choked or sonic flow in thick square-edged orifices. Calculated G_h at the orifice exit and compare with G_{hs} calculated from [Equation \(9a\)](#). If G_h is greater than or equal to G_{hs} , a choked flow condition exists in the orifice, and the maximum mass velocity possible at the exit is G_{hs} . If higher mass velocity is required, either an increase in the upstream pressure, P_1 , or the orifice diameter, d_o , would be required.
 - (6) For subsonic flow or for thin, sharp-edged orifices, the overall pressure drop is obtained by multiplying the ΔP value obtained in Step (4) by the pressure recovery factor, r , obtained from [Figure 10 of Section XIV-B](#).
 - b) If the upstream temperature, T_1 , downstream pressure, P_2 , mass flow rate, W , orifice diameter, d_o , and upstream pipe diameter, d_1 , are given, and the pressure drop, ΔP , is to be determined, use the following procedure:
 - (1) For a first trial, calculate a preliminary value for the upstream density, ρ_1 , based on T_1 and P_2 .
 - (2) Calculate the Reynolds number Re , in the upstream piping from [Equation \(19c\)](#). Calculate d_o/d_1 , and obtain the flow coefficient, C , from [Figure 7A or 7B of Section XIV-B](#).
 - (3) Calculate ΔP from [Equation \(6b\)](#), using $Y = 1$.

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- (4) Obtain the pressure recovery factor, r from [Figure 10 of Section XIV-B](#), and calculate the upstream pressure, P_1 , from the following equation:

$$P_1 = P_2 + r \Delta P \quad \text{Eq. (26)}$$

- (5) Find a new value for ρ_1 , using P_1 and T_1 .
- (6) Calculate $\Delta P/P_1$, find the ratio of specific heat capacities, $k = C_p/C_v$ from [Table 1](#) or the isentropic exponent from the QUEST/DISTEX program or PEGASYS, and obtain the expansion factor Y from [Figure 5](#) or [Equation 5\(a\)](#) or [5\(b\)](#).
- (7) Calculate a new value for ΔP from [Equation \(6b\)](#). If this differs by more than 10% from the one calculated above, retain the present value of ΔP and repeat the calculation from step 4 until convergence is obtained on ΔP .
- (8) Check for choked or sonic flow in thick square-edged orifices. Calculate G_h at the orifice exit and compare with G_{hs} calculated from [Equation \(9a\)](#). If G_h is greater than or equal to G_{hs} , a choked flow condition exists in the orifice, and the maximum mass velocity possible at the exit is G_{hs} . If higher mass velocity is required, either an increase in the upstream pressure, P_1 , or the orifice diameter, d_o , would be required.
- (9) For subsonic flow or for thin, sharp-edged orifices, the overall pressure drop is obtained by multiplying the ΔP value obtained in Step (7) by the pressure recovery factor, r .
- c) If the upstream density, ρ_1 , upstream pressure, P_1 , downstream pressure, P_2 , orifice diameter, d_o , and upstream pipe diameter, d_1 , are given, and the mass flow rate W through the orifice is to be determined, use the following procedure:
- (1) Calculate $\Delta P = (P_1 - P_2)/r$.
 - (2) Calculate $\Delta P/P_1$ and d_o/d_1 , find the ratio of specific heat capacities, $k = C_p/C_v$ from [Table 1](#) or the isentropic exponent from the QUEST/DISTEX program or PEGASYS, and obtain the expansion factor Y from [Figure 5](#) or [Equation \(5a\)](#) or [5\(b\)](#).
 - (3) Calculate W from [Equation \(6b\)](#), using $C = 0.60$.
 - (4) Calculate the Reynolds number, Re , in the upstream piping from [Equation \(19c\)](#), and obtain a new value for the flow coefficient, C , from [Figure 7A or 7B of Section XIV-B](#).
 - (5) Recalculate W from [Equation \(6b\)](#), using the new value for C and repeat the above procedure, if necessary, until convergence is obtained on W .
 - (6) For thick, square-edged orifices, check for choked or sonic flow. Calculate G_h at the orifice exit and compare with G_{hs} calculated from [Equation \(9a\)](#). If G_h is greater than or equal to G_{hs} , a choked flow condition exists in the orifice, and the maximum mass velocity possible at the exit is G_{hs} . In this case W is calculated from the value of G_{hs} .
- d) If the upstream density, ρ_1 , upstream pressure, P_1 , downstream pressure, P_2 , mass flow rate, W , and upstream pipe diameter, d_1 are given, and the orifice diameter, d_o , is to be determined, use the following procedure:
- (1) Assume $d_o/d_1 = 0.6$.
 - (2) Calculate the Reynolds number, Re , in the upstream piping from [Equation \(19c\)](#) and obtain the flow coefficient C from [Figure 7A or 7B of Section XIV-B](#).
 - (3) Obtain the pressure recovery factor, r from [Figure 10 of Section XIV-B](#) and calculate $\Delta P = (P_1 - P_2)/r$.
 - (4) Calculate $\Delta P/P_1$ and d_o/d_1 , find the ratio of specific heat ratio capacities, $k = C_p/C_v$ from [Table 1](#) or the isentropic exponent from the QUEST/DISTEX program or PEGASYS, and obtain the expansion factor Y from [Figure 5](#) or [Equation \(5a\)](#) or [5\(b\)](#).
 - (5) Calculate d_o from [Equation \(6b\)](#).
 - (6) Calculate a new value for d_o/d_1 , and repeat the above steps, if necessary, until convergence is obtained on d_o/d_1 .
 - (7) For thick, square-edged orifices, check for choked or sonic flow. Calculate G_h at the orifice exit and compare with G_{hs} calculated from [Equation \(9a\)](#). If G_h is greater than or equal to G_{hs} , a choked flow condition exists in the orifice, and the maximum mass velocity possible at the exits is G_{hs} so that a larger orifice diameter is required for the required mass flow rate.

Flow Nozzles

Use the procedure for orifices with the flow coefficient from [Figure 8 of Section XIV-B](#). If, during the calculation procedure, $\Delta P/P_1$ turns out to be greater than indicated by the end points of the flow nozzles and venturi curves in [Figure 5](#), or the downstream pressure is below P_{CF} calculated from [Equation \(10\)](#), then sonic conditions apply; therefore, use the following procedures, using either the discharge coefficients from [Figure 8 of Section XIV-B](#) or those given by [Equations \(10\) - \(13\)](#) for ASME long radius flow nozzles.

- 1) The upstream pressure, P_1 , upstream density, ρ_1 , mass flow rate, W , nozzle diameter, d_o , and upstream pipe diameter, d_1 , are given and the pressure drop, ΔP , is to be determined. The mass flow rate, W , is limited due to sonic flow in the nozzle. The required value of W could therefore only be reached if d_o or P_1 were increased. If, however, d_o and P_1 are fixed, the

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same, the flow rate, W , and the minimum ΔP required for this flow rate are determined either graphically or computationally as follows:

Using the graphical approach:

- a) Obtain $\Delta P/P_1$ and the Y corresponding with the end point of the applicable curve from [Figure 5](#).
- b) Calculate ΔP from $\Delta P/P_1$ and the given P_1 , and use this value to calculate W from [Equation \(6b\)](#). This flow rate, W , will occur for any pressure drop $\geq \Delta P$ calculated.

Using the computational approach:

- a) Calculate P_{CF} using [Equation \(10\)](#) and determine ΔP by subtracting P_{CF} from P_1 .
 - b) Calculate W from [Equation \(6b\)](#) using a value of Y calculated from [Equation \(4\)](#) with the calculated value of P_{CF} used as P_2 .
- 2) The upstream temperature, T_1 , downstream pressure, P_2 , mass flow rate, W , nozzle diameter d_o , and upstream pipe diameter, d_1 , are given, and the pressure drop, ΔP , is to be determined. Calculate the required upstream pressure, P_1 , and ΔP either graphically or computationally as follows:

Using the graphical approach:

- a) Divide [Equation \(6b\)](#) by P_1 and insert in the equation the values for $\Delta P/P_1$ and Y from the endpoint of the applicable curve in [Figure 5](#). Then solve for P_1/P_2 .
- b) Find P_1/ρ_1 from [Equation \(20\)](#).
- c) Calculate P_1 from [Equation \(21\)](#).
- d) Calculate $\Delta P = P_1 - P_2$.

Using the computational approach:

- a) Calculate P_1 by rearrangement of [Equation \(10\)](#) and substituting P_2 for P_{CF} .
 - b) Calculate $\Delta P = P_1 - P_2$.
- 3) The upstream density, ρ_1 , upstream pressure, P_1 , downstream pressure, P_2 , nozzle diameter, d_o , and upstream pipe diameter, d_1 , are given, and the mass flow rate, W , through the nozzle is to be determined. The mass flow rate, W , can be calculated by graphical and computational methods as follows:

Using the graphical approach:

- a) Find $\Delta P/P_1$ and Y from the end point of the applicable curve in [Figure 5](#).
- b) Calculate ΔP from $\Delta P/P_1$ and P_1 .
- c) Calculate W from [Equation \(6b\)](#), using ΔP and Y .

Using the computational approach:

- a) Calculate P_{CF} from [Equation \(10\)](#).
 - b) Determine $\Delta P = P_1 - P_{CF}$.
 - c) Use $P_{CF} = P_2$, calculate Y from [Equation \(4\)](#) and then calculate W from [Equation \(6b\)](#) using calculated values P_2 and Y .
- 4) The upstream density, ρ_1 , upstream pressure, P_1 , downstream pressure, P_2 , mass flow rate, W , and upstream pipe diameter, d_1 , are given, and the nozzle diameter, d_o , is to be determined. The required nozzle diameter, d_o , can be calculated using graphical or computational approaches as follows:

Using the graphical method:

- a) Assume $d_o/d_1 = 0.2$.
- b) Calculate the Reynolds number, Re , in the upstream piping from [Equation \(19c\)](#), and obtain the flow coefficient C from [Figure 8 of Section XIV-B](#).
- c) Calculate d_o from [Equation \(6b\)](#), using $\Delta P/P_1$ and Y corresponding with the end point of the curve for $d_o/d_1 = 0.2$ in [Figure 5](#).
- d) Calculate d_o/d_1 and compare this with the assumed value. Repeat the above procedure with newly calculated values of d_o/d_1 , if necessary, until convergence is obtained on d_o/d_1 .

Using the computational approach:

- a) Assume $d_o/d_1 = 0.2$.
- b) Calculate the Reynolds number, Re , in the upstream piping from [Equation \(19c\)](#), and obtain the flow coefficient C either from [Figure 8 of Section XIV-B](#) or from [Equations \(10\) - \(13\)](#) for ASME long radius flow nozzles.
- c) Calculate Y from [Equation \(4\)](#) and ΔP from [Equation \(10\)](#) where $\Delta P = 1 - P_{CF}/P_1$.

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- d) Calculate d_o from [Equation \(6b\)](#) using the ΔP and Y calculated in Step c.
- e) Calculate d_o/d_1 and compare this with the assumed value. Repeat the above procedure with newly calculated values of d_o/d_1 , if necessary, until convergence is obtained on d_o/d_1 .

Venturis

For calculations on venturis, use the same procedures as for flow nozzles, except calculate the flow coefficient C from [Equation \(10\) of Section XIV-B](#) or [Equations \(11\) - \(14\)](#) for ASME - ISO cylindrical or toroidal throat venturis.

Expansions and Contractions

Use the following procedures:

- 1) Calculate the pressure drop as for liquid flow, following the procedure given in [Section XIV-B](#). For the density, ρ , use the upstream or downstream value, whichever is available.
- 2) If the calculated pressure drop is more than 10% of the upstream absolute pressure, or if an accurate estimate is needed, proceed as follows:
- 3) Find the ratio of specific heat capacity $k = C_p/C_v$ from [Table 1](#) or the isentropic exponent from the QUEST/DISTEX program or PEGASYS.
- 4) Calculate $(\Delta P)_t/P_1$ and find Y from [Figure 5](#), using $(\Delta P)_t/P_1$ for $\Delta P/P_1$ and the ratio of the smaller pipe diameter to the larger pipe diameter for d_o/d_1 .
- 5) Calculate $(\Delta P)_k$ from [Equation \(7b\)](#).
- 6) Calculate a new value for $(\Delta P)_t = (\Delta P)_k + (\Delta P)_f$, where $(\Delta P)_f$ is the frictional pressure drop obtained from [Equation \(7\) of Section XIV-B](#).
- 7) Repeat the last three steps, if necessary, until convergence is obtained on $(\Delta P)_f$.

Perforated-Pipe Distributors

Use the same procedure as in [Section XIV-B](#) for liquid flow perforated-pipe distributors, except calculate the required total area of the outlet holes from [Equation \(8b\)](#) instead of [Equation \(14\) in Section XIV-B](#).

INTEGRATED PRESSURE DROP CALCULATIONS FOR PIPING SYSTEMS

Use the procedures given below for calculating pressure drop in any flow system containing more than one single piping component.

Rough Estimate - For all gases, a rough estimate of pressure drop in commercial steel pipe can be obtained from [Equation \(17\)](#) in combination with the procedure for liquid flow in [Section XIV-B](#). For pressure drops less than about 40% of the inlet pressure, P_1 , use [Equation \(17\)](#) as outlined previously; for higher pressure drops, use the procedure presented below.

Accurate Estimate - For all gases, an accurate estimate of pressure drop in a piping system is obtained as follows:

- 1) Divide the system in question into sections of constant mass flow rate and constant nominal diameter. Divide each section which contains an orifice, nozzle or venturi into an upstream section, the orifice, nozzle or venturi, and a downstream section.
- 2) Calculate the pressure drops in the individual sections, starting at the end where the pressure is known.
 - a) The pressure drop in any section containing piping, valves and bends is calculated following the procedures given above for straight pipe. Valves and bends may be accounted for by adding their resistance coefficients, K (from [Figure 5A or 5B of Section XIV-B](#)) to the pipe resistance coefficient, N , or by adding equivalent lengths from [Equation \(25\)](#) to the length of straight pipe. If there is significant acceleration in the flow, or for greater accuracy, rather than lumping all the resistances together (adding equivalent lengths), perform a step-by-step calculation for each component, thereby accounting for the actual gas velocity in each. With [Equation \(9a\)](#) check if sonic velocity occurs in any valve. If so, treat the valve as a thick, square-edged orifice.
 - b) The pressure drop in any expansion, contraction, orifice, nozzle, venturi or flow junction (tees and Y's) is calculated as shown above for single piping components.
- 3) Combine the various pressure drops to obtain the pressure distribution in the entire piping system.

