MEASUREMENT OF FLOW

10.1. MEASUREMENT OF FLOW

GENERAL REFERENCES: ASME, *Performance Test Code on Compressors and Exhausters*, PTC 10-1997, American Society of Mechanical Engineers (ASME), New York, 1997. Norman A. Anderson, *Instrumentation for Process Measurement and Control*, 3d ed., CRC Press, Boca Raton, Fla., 1997. Roger C. Baker, *Flow Measurement Handbook: Industrial Designs, Operating Principles, Performance, and Applications*, Cambridge University Press, Cambridge, United Kingdom, 2000. Roger C. Baker, *An Introductory Guide to Flow Measurement*, ASME, New York, 2003. Howard S. Bean, ed., *Fluid Meters—Their Theory and Application—Report of the ASME Research Committee on Fluid Meters,* 6th ed., ASME, New York, 1971. Douglas M. Considine, Editor-in-Chief, *Process/Industrial Instruments and Controls Handbook*, 4th ed., McGraw-Hill, New York, 1993. Bela G. Liptak, Editor-in-Chief, *Process Measurement and Analysis*, 4th ed., CRC Press, Boca Raton, Fla., 2003. Richard W. Miller, *Flow Measurement Engineering Handbook*, 3d ed., McGraw-Hill, New York, 1996. Ower and Pankhurst, *The Measurement of Air Flow,* Pergamon, Oxford, United Kingdom, 1966. Brian Price et al., *Engineering Data Book*, 12th ed., Gas Processors Suppliers Association, Tulsa, Okla., 2004. David W. Spitzer, *Flow Measurement*, 2d ed., Instrument Society of America, Research Triangle Park, N.C., 2001. David W. Spitzer, *Industrial Flow Measurement*, 3d ed., Instrument Society of America, Research Triangle Park, N.C., 2005.

10.1.1. INTRODUCTION

The flow rate of fluids is a critical variable in most chemical engineering applications, ranging from flows in the process industries to environmental flows and to flows within the human body. *Flow* is defined as mass flow or volume flow per unit of time at specified temperature and pressure conditions for a given fluid. This subsection deals with the techniques of measuring pressure, temperature, velocities, and flow rates of flowing fluids. For more detailed discussion of these variables, consult Sec. 8. Section 8 introduces methods of measuring flow rate, temperature, and pressure. This subsection builds on the coverage in Sec. 8 with emphasis on measurement of the flow of fluids.

10.1.2. PROPERTIES AND BEHAVIOR OF FLUIDS

Transportation and the storage of fluids (gases and liquids) involves the understanding of the properties and behavior of fluids. The study of fluid dynamics is the study of fluids and their motion in a force field.

Flows can be classified into two major categories: (a) incompressible and (b) compressible flow. Most liquids fall into the incompressible-flow category, while most gases are compressible in nature. A perfect fluid can be defined as a fluid that is nonviscous and nonconducting. Fluid flow, compressible or incompressible, can be classified by the ratio of the inertial forces to the viscous forces. This ratio is represented by the Reynolds number (N_{Re}). At a low Reynolds number, the flow is considered to be laminar, and at high Reynolds numbers, the flow is considered to be turbulent. The limiting types of flow are the inertialess flow, sometimes called Stokes flow, and the inviscid flow that occurs at an infinitely large Reynolds number. Reynolds numbers (dimensionless) for flow in a pipe is given as:

$$N_{Re} = \frac{\rho V D}{\mu}$$

(10-1)

where ρ is the density of the fluid, V the velocity, D the diameter, and μ the viscosity of the fluid. In fluid motion where the frictional forces interact with the inertia forces, it is important to consider the ratio of the viscosity μ to the density ρ. This ratio is known as the kinematic viscosity (ν). Tables 10-1 and 10-2 give the kinematic viscosity for several fluids. A flow is considered to be *adiabatic* when there is no transfer of heat between the fluid and its surroundings. An isentropic flow is one in which the entropy of each fluid element remains constant.

To fully understand the mechanics of flow, the following definitions explain the behavior of various types of fluids in both their static and flowing states.

A perfect fluid is a nonviscous, nonconducting fluid. An example of this type of fluid would be a fluid that has a very small viscosity and conductivity and is at a high Reynolds number. An ideal gas is one that obeys the equation of state:

$$\frac{P}{\rho} = RT$$

(10-2)

where P = pressure, ρ=density, R is the gas constant per unit mass, and T = temperature.

A flowing fluid is acted upon by many forces that result in changes in pressure, temperature, stress, and strain. A fluid is said to be isotropic when the relations between the components of stress and those of the rate of strain are the same in all directions. The fluid is said to be Newtonian when this relationship is linear. These pressures and temperatures must be fully understood so that the entire flow picture can be described.

The *static pressure* in a fluid has the same value in all directions and can be considered as a scalar point function. It is the pressure of a flowing fluid. It is normal to the surface on which it acts and at any given point has the same magnitude irrespective of the orientation of the surface. The static pressure arises because of the random motion in the fluid of the molecules that make up the fluid. In a diffuser or nozzle, there is an increase or decrease in the static pressure due to the change in velocity of the moving fluid.

Total pressure is the pressure that would occur if the fluid were brought to rest in a reversible adiabatic process. Many texts and engineers use the words *total* and *stagnation* to describe the flow characteristics interchangeably. To be accurate, the stagnation pressure is the pressure that would occur if the fluid were brought to rest adiabatically or diabatically.

Table 10-1. Density, Viscosity, and Kinematic Viscosity of Water and Air in Terms of Temperature

Temperature		Water			Air at a pressure of 760 mm Hg (14.696 lbf/in^2)		
		Density ρ (lbf sec^2/ft^4)	μ×10^6 Viscosity (lbf sec/ft^2)	Kinematic ν×10^6 viscosity (ft^2/sec)	Density ρ (lbf sec^2/ft^4)	μ×10^6 Viscosity (lbf sec/ft^2)	Kinematic ν×10^{6}viscosity (ft^2/sec)
(°C)	(°F)						
−20	−4	—	—	—	0.00270	0.326	122
−10	14	—	—	—	0.00261	0.338	130
0	32	1.939	37.5	19.4	0.00251	0.350	140
10	50	1.939	27.2	14.0	0.00242	0.362	150
20	68	1.935	21.1	10.9	0.00234	0.375	160
40	104	1.924	13.68	7.11	0.00217	0.399	183
60	140	1.907	9.89	5.19	0.00205	0.424	207
80	176	1.886	7.45	3.96	0.00192	0.449	234
100	212	1.861	5.92	3.19	0.00183	0.477	264

Conversion factors: 1 kp sec^2/m^4 = 0.01903 lbf sec^2/ft^4 (= slug/ft^3)
1 lbf sec^2/ft^4 = 32.1719 lb/ft^3 (lb = lb mass; lbf = lb force)
1 kp sec^2/m^4 = 9.80665 kg/m^3 (kg = kg mass; kp = kg force)
1 kg/m^3 = 16.02 lb/ft^3

Table 10-2. Kinematic Viscosity

Liquid	(Temperature) °C	(Temperature) °F	$\nu \times 10^6$ (ft^2/s)
Glycerine	20	68	7319
Mercury	0	32	1.35
Mercury	100	212	0.980
Lubricating oil	20	68	4306
Lubricating oil	40	104	1076
Lubricating oil	60	140	323

Total pressure will only change in a fluid if shaft work or work of extraneous forces are introduced. Therefore, total pressure would increase in the impeller of a compressor or pump; it would remain constant in the diffuser. Similarly, total pressure would decrease in the turbine impeller but would remain constant in the nozzles.

Static temperature is the temperature of the flowing fluid. Like static pressure, it arises because of the random motion of the fluid molecules. Static temperature is in most practical installations impossible to measure since it can be measured only by a thermometer or thermocouple at rest relative to the flowing fluid that is moving with the fluid. Static temperature will increase in a diffuser and decrease in a nozzle.

Total temperature is the temperature that would occur when the fluid is brought to rest in a reversible adiabatic manner. Just like its counterpart *total pressure, total* and *stagnation temperatures* are used interchangeably by many test engineers.

Dynamic temperature and pressure are the difference between the total and static conditions.

$$P_d = P_T - P_s$$

(10-3)

$$T_d = T_T - T_s$$

(10-4)

where subscript *d* refers to dynamic, *T* to total, and *s* to static.

Another helpful formula is:

$$P_K = \frac{1}{2}\rho V^2$$

(10-5)

For incompressible fluids, $P_K = P_d$.

10.1.3. TOTAL TEMPERATURE

For most points requiring temperature monitoring, either thermocouples or resistive thermal detectors (RTDs) can be used. Each type of temperature transducer has its own advantages and disadvantages, and both should be considered when temperature is to be measured. Since there is considerable confusion in this area, a short discussion of the two types of transducers is necessary.

Thermocouples The various types of thermocouples provide transducers suitable for measuring temperatures from −330 to 5000°F (−201 to 2760°C). Thermocouples function by producing a voltage proportional to the temperature differences between two junctions of dissimilar metals. By measuring this voltage, the temperature difference can be determined. It is assumed that the temperature is known at one of the junctions; therefore, the temperature at the other junction can be determined. Since the thermocouples produce a voltage, no external power supply is required to the test junction; however, for accurate measurement, a reference junction is required. For a temperature monitoring system, reference junctions must be placed at each thermocouple or similar thermocouple wire installed from the thermocouple to the monitor where there is a reference junction. Properly designed thermocouple systems can be accurate to approximately ±2°F (±1°C).

Resistive Thermal Detectors (RTDs) RTDs determine temperature by measuring the change in resistance of an element due to temperature. Platinum is generally utilized in RTDs because it remains mechanically and electrically stable, resists contaminations, and can be highly refined. The useful range of platinum RTDs is −454–1832°F (− 270− 1000°C). Since the temperature is determined by the resistance in the element, any type of electrical conductor can be utilized to connect the RTD to the indicator; however, an electrical current must be provided to the RTD. A properly designed temperature monitoring system utilizing RTDs can be accurate ±0.02°F (±0.01°C).

10.1.4. STATIC TEMPERATURE

Since this temperature requires the thermometer or thermocouple to be at rest relative to the flowing fluid, it is impractical to measure. It can be, however, calculated from the measurement of total temperature and total and static pressure.

$$T_s = \frac{T_O}{\left(\frac{P_O}{P_s}\right)^{(k-1)/k}}$$

(10-6)

10.1.5. DRY- AND WET-BULB TEMPERATURES

The moisture content or humidity of air has an important effect on the properties of the gaseous mixture. Steam in air at any relative humidity less than 100 percent must exist in a superheated condition. The saturation temperature corresponding to the actual partial pressure of the steam in air is called the dew point. This term arose from the fact that when air at less than 100 percent relative humidity is cooled to the temperature at which it becomes saturated, the air has reached the minimum temperature to which it can be cooled without precipitation of the moisture (dew). Dew point can also be defined as that temperature at which the weight of steam associated with a certain weight of dry air is adequate to saturate that weight of air.

The dry-bulb temperature of air is the temperature that is indicated by an ordinary thermometer. When an air temperature is stated without any modifying term, it is always taken to be the dry-bulb temperature. In contrast to dry-bulb, or air, temperature, the term *wet-bulb temperature of the air*, or simply *wet-bulb temperature*, is employed. When a thermometer, with its bulb covered by a wick wetted with water, is moved through air unsaturated with water vapor, the water evaporates in proportion to the capacity of the air to absorb the evaporated moisture, and the temperature indicated by the thermometer drops below the dry-bulb, or air, temperature. The equilibrium temperature finally reached by the thermometer is known as the wet-bulb temperature. The purpose in measuring both the dry-bulb and wet-bulb temperature of the air is to find the exact humidity characteristics of the air from the readings obtained, either by calculation or by use of a psychrometric chart. Instruments for measuring wet-bulb and dry-bulb temperatures are known as psychrometers. A sling psychrometer consists of two thermometers mounted side by side on a holder, with provision for whirling the whole device through the air. The dry-bulb thermometer is bare, and the wet bulb is covered by a wick which is kept wetted with clean water. After being whirled a sufficient amount of time, the wet-bulb thermometer reaches its equilibrium point, and both the wet-bulb and dry-bulb thermometers are then quickly read. Rapid relative movement of the air past the wet-bulb thermometer is necessary to get dependable readings.

For other methods of measuring the moisture content of gases, see Sec. 8.

10.1.6. PRESSURE MEASUREMENTS

Pressure is defined as the force per unit area. Pressure devices measure with respect to the ambient atmospheric pressure: The absolute pressure P_a is the pressure of the fluid (gauge pressure) plus the atmospheric pressure.

Process pressure-measuring devices may be divided into three groups:

1. Those that are based on the height of a liquid column (manometers)

2 . Those that are based on the measurement of the distortion of an elastic pressure chamber (mechanical pressure gauges such as Bourdon-tube gauges and diaphragm gauges)

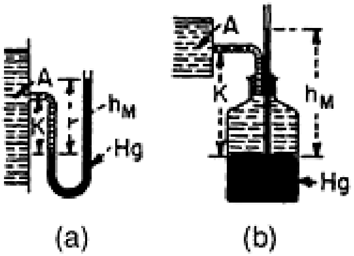

Figure 10-1. Open manometers.

3. Electric sensing devices (strain gauges, piezoresistive transducers, and piezoelectric transducers)

This subsection contains an expanded discussion of manometric methods. See Sec. 8 for other methods.

Liquid-Column Manometers The **height,** or **head,** $p_n = \rho hg/g_c$ to which a fluid rises in an open vertical tube attached to an apparatus containing a liquid is a direct measure of the pressure at the point of attachment and is frequently used to show the level of liquids in tanks and vessels. This same principle can be applied with U-tube gauges (Fig. 10-1*a*) and equivalent devices (such as that shown in Fig. 10-1*b*) to measure pressure in terms of the head of a fluid other than the one under test. Most of these gauges may be used either as **open** or as **differential manometers.** The manometric fluid that constitutes the measured liquid column of these gauges may be any liquid immiscible with the fluid under pressure. For high vacuums or for high pressures and large pressure differences, the gauge liquid is a high-density liquid, generally mercury; for low pressures and small pressure differences, a low-density liquid (e.g., alcohol, water, or carbon tetrachloride) is used.

The **open U tube** (Fig. 10-1*a*) and the **open gauge** (Fig. 10-1*b*) each show a reading h_M m (ft) of manometric fluid. If the interface of the manometric fluid and the fluid of which the pressure is wanted is K m (ft) below the point of attachment, A, ρ_A is the density of the latter fluid at A, and ρ_M is that of the manometric fluid, then gauge pressure p_A (lbf/ft^2) at A is

$$p_A = (h_M\rho_M - K\rho_A)(g/g_c)$$

(10-7)[1]

where g = local acceleration due to gravity and g_c = dimensional constant. The head H_A at A as meters (feet) of the fluid at that point is

$$h_A = h_M(\rho_M/\rho_A) - K$$

(10-8)[2]

When a gas pressure is measured, unless it is very high, ρ_A is so much smaller than ρ_M that the terms involving K in these formulas are negligible.

The **differential U tube** (Fig. 10-2) shows the pressure difference between taps A and B to be

$$p_A - p_B = [h_M(\rho_M - \rho_A) + K_A\rho_A - K_B\rho_B](g/g_c)$$

(10-9)[3]

where h_M is the difference in height of the manometric fluid in the U tube; K_A and K_B are the vertical distances of the upper surface of the manometric fluid above A and B, respectively; ρ_A and ρ_B are the densities of the fluids at A and B, respectively; and ρ_M is the density of the manometric fluid. If either pressure tap is above the higher level of manometric fluid, the corresponding K is taken to be negative. Valve *D,* which is kept closed when the gauge is in use, is used to vent off gas which may accumulate at these high points.

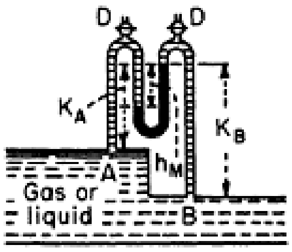

Figure 10-2. Differential U tube.

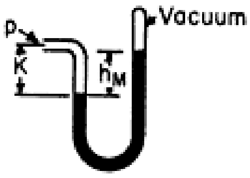

Figure 10-3. Closed U tube.

The **inverted differential U tube,** in which the manometric fluid may be a gas or a light liquid, can be used to measure liquid pressure differentials, especially for the flow of slurries where solids tend to settle out.

Closed U tubes (Fig. 10-3) using mercury as the manometric fluid serve to measure directly the absolute pressure p of a fluid, provided that the space between the closed end and the mercury is substantially a perfect vacuum.

The **mercury barometer** (Fig. 10-4) indicates directly the absolute pressure of the atmosphere in terms of height of the mercury column. Normal (standard) barometric pressure is 101.325 kPa by definition. Equivalents of this pressure in other units are 760 mm mercury (at 0°C), 29.921 inHg (at 0°C), 14.696 lbf/in^2, and 1 atm. For cases in which barometer readings, when expressed by the height of a mercury column, must be corrected to standard temperature (usually 0°C), appropriate temperature correction factors are given in ASME PTC, op. cit., pp. 23–26; and Weast, *Handbook of Chemistry and Physics,* 62d ed., Chemical Rubber, Cleveland, 1984, pp. E36–E37.

Tube Size for Manometers To avoid capillary error, tube diameter should be sufficiently large and the manometric fluids of such densities that the effect of capillarity is negligible in comparison with the gauge reading. The effect of capillarity is practically negligible for tubes with inside diameters 12.7 mm (1/2 in) or larger (see ASME PTC, op. cit., p. 15). Small diameters are generally permissible for U tubes because the capillary displacement in one leg tends to cancel that in the other.

The capillary rise in a small vertical open tube of circular cross section dipping into a pool of liquid is given by

$$h = \frac{4\sigma g_c \cos\theta}{gD(\rho_1 - \rho_2)}$$
(10-10)

Here σ = surface tension, D = inside diameter, ρ_1 and ρ_2 are the densities of the liquid and gas (or light liquid) respectively, g = local acceleration due to gravity, g_c = dimensional constant, and θ is the contact angle subtended by the heavier fluid. For most organic liquids and water, the contact angle θ is zero against glass, provided the glass is wet with a film of the liquid; for mercury against glass, θ = 140° (*International Critical Tables,* vol. IV, McGraw-Hill, New York, 1928, pp. 434–435). For further discussion of capillarity, see Schwartz, *Ind. Eng. Chem.,* **61**(1), 10–21 (1969).

Multiplying Gauges To attain the requisite precision in measurement of small pressure differences by liquid-column manometers,

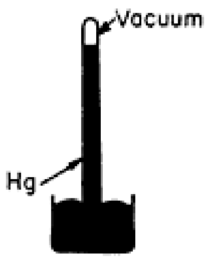

Figure 10-4. Mercury barometer.

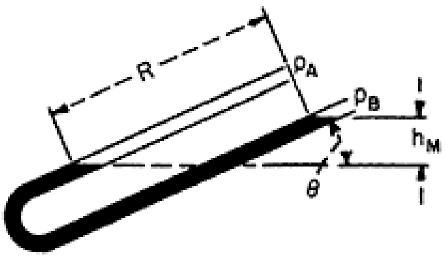

Figure 10-5. Inclined U tube.

means must often be devised to magnify the readings. Of the schemes that follow, the second and third may give tenfold multiplication; the fourth, as much as thirtyfold. In general, the greater the multiplication, the more elaborate must be the precautions in the use of the gauge if the gain in precision is not to be illusory.

1. *Change of manometric fluid.* In open manometers, choose a fluid of lower density. In differential manometers, choose a fluid such that the difference between its density and that of the fluid being measured is as small as possible.

2. *Inclined U tube* (Fig. 10-5). If the reading R m (ft) is taken as shown and R_0 m (ft) is the zero reading, by making the substitution $h_M = (R - R_0) \sin\theta$, the formulas of preceding paragraphs give $(p_A - p_B)$ when the corresponding upright U tube is replaced by one inclined. For precise work, the gauge should be calibrated because of possible variations in tube diameter and slope.

3. *The draft gauge* (Fig. 10-6). Commonly used for low gas heads, this gauge has for one leg of the U a reservoir of much larger bore than the tubing that forms the inclined leg. Hence variations of level in the inclined tube produce little change in level in the reservoir. Although h_M may be readily computed in terms of reading R and the dimensions of the tube, calibration of the gauge is preferable; often the changes of level in the reservoir are not negligible, and also variations in tube diameter may introduce serious error into the computation. Commercial gauges are often provided with a scale giving h_M directly in height of water column, provided a particular liquid (often not water) fills the tube; failure to appreciate that the scale is incorrect unless the gauge is filled with the specified liquid is a frequent source of error. If the scale reads correctly when the density of the gauge liquid is ρ_0, then the reading must be multiplied by ρ/ρ_0 if the density of the fluid actually in use is ρ.

4. *Two-fluid U tube* (Fig. 10-7). This is a highly sensitive device for measuring small gas heads. Let A be the cross-sectional area of each of the reservoirs and a that of the tube forming the U; let ρ_1 be the density of the lighter fluid and ρ_2 that of the heavier fluid; and if R is the reading and R_0 its value with zero pressure difference, then the pressure difference is

$$p_A - p_B = (R - R_0)\left(\rho_2 - \rho_1 + \frac{a}{A}\rho_1\right)\frac{g}{g_c}$$

(10-11)

where g = local acceleration due to gravity and g_c = dimensional constant.

When A/a is sufficiently large, the term $(a/A)\,\rho_1$ in Eq. (10-11) becomes negligible in comparison with the difference $(\rho_2 - \rho_1)$. However, this term should not be omitted without due consideration. In applying Eq. (10-11), the densities of the gauge liquids may not be taken from tables without the possibility of introducing serious error,

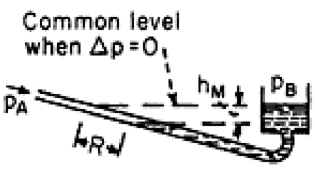

Figure 10-6. Draft gauge.

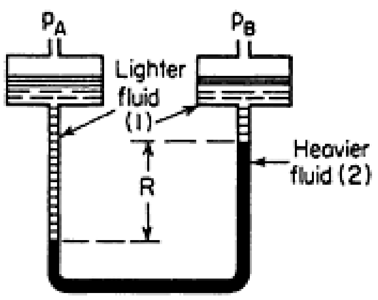

Figure 10-7. Two-fluid U tube.

for each liquid may dissolve appreciable quantities of the other. Before the gauge is filled, the liquids should be shaken together, and the actual densities of the two layers should be measured for the temperature at which the gauge is to be used. When high magnification is being sought, the U tube may have to be enclosed in a constant-temperature bath so that ($\rho_2 - \rho_1$) may be accurately known. In general, if highest accuracy is desired, the gauge should be calibrated.

Several **micromanometers,** based on the liquid-column principle and possessing extreme precision and sensitivity, have been developed for measuring minute gas-pressure differences and for calibrating low-range gauges. Some of these micromanometers are available commercially. These micromanometers are free from errors due to capillarity and, aside from checking the micrometer scale, require no calibration.

Mechanical Pressure Gauges The **Bourdon-tube gauge** indicates pressure by the amount of flection under internal pressure of an oval tube bent in an arc of a circle and closed at one end. These gauges are commercially available for all pressures below atmospheric and for pressures up to 700 MPa (about 100,000 lbf/in^2) above atmospheric. Details on Bourdon-type gauges are given by Harland [*Mach. Des.,* **40**(22), 69–74 (Sept. 19, 1968)].

A **diaphragm gauge** depends for its indication on the deflection of a diaphragm, usually metallic, when subjected to a difference of pressure between the two faces. These gauges are available for the same general purposes as Bourdon gauges but are not usually employed for high pressures. The aneroid barometer is a type of diaphragm gauge.

Small **pressure transducers with flush-mounted diaphragms** are commercially available for the measurement of either steady or fluctuating pressures up to 100 MPa (about 15,000 lbf/in^2). The metallic diaphragms are as small as 4.8 mm (3/16 in) in diameter. The transducer is mounted on the apparatus containing the fluid whose pressure is to be measured so that the diaphragm is flush with the inner surface of the apparatus. Deflection of the diaphragm is measured by unbonded strain gauges and recorded electrically.

With nonnewtonian fluids the pressure measured at the wall with non-flush-mounted pressure gauges may be in error (see subsection "Static Pressure").

Bourdon and diaphragm gauges that show both pressure and vacuum indications on the same dial are called **compound gauges.**

Conditions of Use Bourdon tubes should not be exposed to temperatures over about 65°C (about 150°F) unless the tubes are specifically designed for such operation. When the pressure of a hotter fluid is to be measured, some type of liquid seal should be used to keep the hot fluid from the tube. In using either a Bourdon or a diaphragm gauge to measure gas pressure, if the gauge is below the pressure tap of the apparatus so that liquid can collect in the lead, the gauge reading will be too high by an amount equal to the hydrostatic head of the accumulated liquid.

For measuring pressures of corrosive fluids, slurries, and similar process fluids which may foul Bourdon tubes, a **chemical gauge,** consisting of a Bourdon gauge equipped with an appropriate flexible diaphragm to seal off the process fluid, may be used. The combined volume of the tube and the connection between the diaphragm and the tube is filled with an inert liquid. These gauges are available commercially.

Further details on pressure-measuring devices are found in Sec. 8.

Calibration of Gauges Simple **liquid-column manometers** do not require calibration if they are so constructed as to minimize errors due to capillarity (see subsection "Liquid-Column Manometers"). If the scales used to measure the readings have been checked against a standard, the accuracy of the gauges depends solely upon the precision of determining the position of the liquid surfaces. Hence liquid-column manometers are primary standards used to calibrate other gauges.

For **high pressures** and, with commercial mechanical gauges, even for quite moderate pressures, a deadweight gauge (see ASME PTC, op. cit., pp. 36–41) is commonly used as the primary standard because it is safer and more convenient than use of manometers. When manometers are used as high-pressure standards, an extremely high mercury column may be avoided by connecting a number of the usual U tubes in series. Multiplying gauges are standardized by comparing them with a micromanometer. Procedure in the calibration of a gauge consists merely of connecting it, in parallel with a standard gauge, to a reservoir wherein constant pressure may be maintained. Readings of the unknown gauge are then made for various reservoir pressures as determined by the standard.

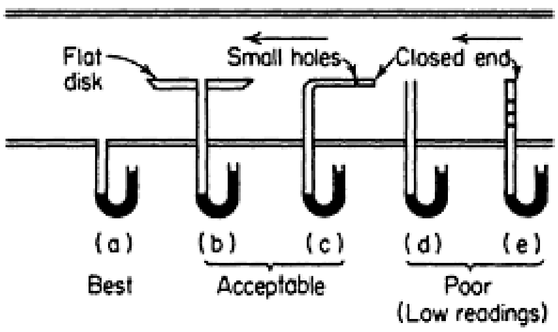

Figure 10-8. Measurement of static pressure.

Calibration of **high-vacuum gauges** is described by Sellenger [*Vacuum,* **18**(12), 645–650 (1968)].

10.1.7. STATIC PRESSURE

Local Static Pressure In a moving fluid, the local static pressure is equal to the pressure on a surface which moves with the fluid or to the normal pressure (for newtonian fluids) on a stationary surface which parallels the flow. The pressure on such a surface is measured by making a small hole perpendicular to the surface and connecting the opening to a pressure-sensing element (Fig. 10-8*a*). The hole is known as a piezometer opening or pressure tap.

Measurement of local static pressure is frequently difficult or impractical. If the channel is so small that introduction of any solid object disturbs the flow pattern and increases the velocity, there will be a reduction and redistribution of the static pressure. If the flow is in straight parallel lines, aside from the fluctuations of normal turbulence, the flat disk (Fig. 10-8*b*) and the bent tube (Fig. 10-8*c*) give satisfactory results when properly aligned with the stream. Slight misalignments can cause serious errors. Diameter of the disk should be 20 times its thickness and 40 times the static opening; the face must be flat and smooth, with the knife edges made by beveling the underside. The piezometer tube, such as that in Fig. 10-8*c*, should have openings with size and spacing as specified for a pitot-static tube (Fig. 10-12).

Readings given by open straight tubes (Fig. 10-8*d* and 10-8*e* are too low due to flow separation. Readings of closed tubes oriented perpendicularly to the axis of the stream and provided with side openings (Fig. 10-8*e*) may be low by as much as two velocity heads.

Average Static Pressure In most cases, the object of a static-pressure measurement is to obtain a suitable average value for substitution in Bernoulli's theorem or in an equivalent flow formula. This can be done simply only when the flow is in straight lines parallel to the confining walls, such as in straight ducts at sufficient distance downstream from bends (2 diameters) or other disturbances. For such streams, the sum of static head and gravitational potential head is the same at all points in a cross section taken perpendicularly to the axis of flow. Thus the exact location of a piezometer opening about the periphery of such a cross section is immaterial provided its elevation is known. However, in stating the static pressure, the custom is to give the value at the elevation corresponding to the centerline of the stream.

With flow in curved passages or with swirling flow, determination of a true average static pressure is, in general, impractical. In metering, straightening vanes are often placed upstream of the pressure tap to eliminate swirl. Figure 10-9 shows various flow equalizers and straighteners.

Specifications for Piezometer Taps The size of a static opening should be small compared with the diameter of the pipe and yet large compared with the scale of surface irregularities. For reliable results, it is essential that (1) the surface in which the hole is made be substantially smooth and parallel to the flow for some distance on either side of the opening, and (2) the opening be flush with the surface and possess no "burr" or other irregularity around its edge. Rounding of the edge is often employed to ensure absence of a burr. Pressure readings will be high if the tap is inclined upstream, is rounded excessively on the upstream side, has a burr on the downstream side, or has an excessive countersink or recess. Pressure readings will be low if the tap is inclined downstream, is rounded excessively on the downstream side, has a burr on the upstream side, or protrudes into the flow stream. Errors resulting from these faults can be large.

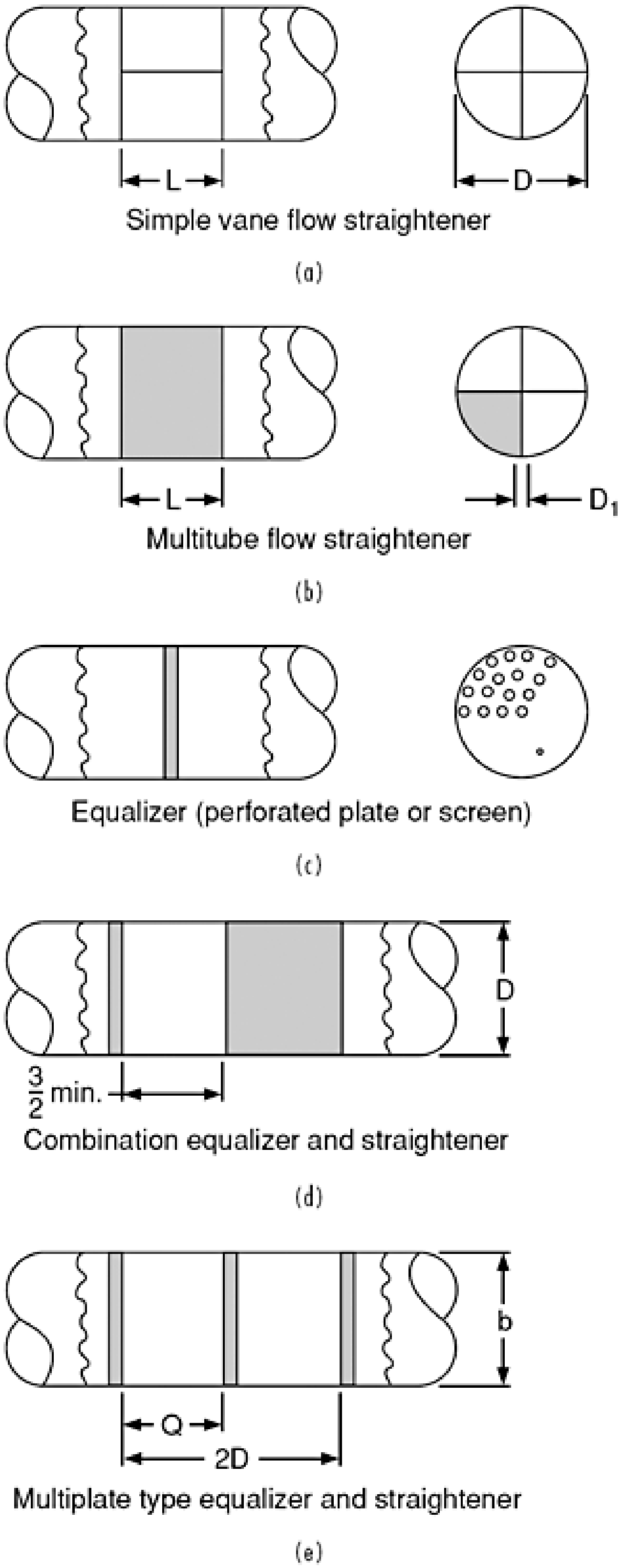

Figure 10-9. Flow equalizers and straighteners [Power Test Code 10, Compressors and Exhausters, Amer. Soc. of Mechanical Engineers, 1997].

Recommendations for **pressure-tap dimensions** are summarized in Table 10-3. Data from several references were used in arriving at these composite values. The length of a pressure-tap opening prior to any enlargement in the tap channel should be at least two tap diameters, preferably three or more.

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Table 10-3. Pressure-Tap Holes

Nominal inside pipe diameter, in	Maximum diameter of pressure tap, mm (in)	Radius of hole-edge rounding, mm (in)
1	3.18 (1/8)	<0.40 (1/64)
2	6.35 (1/4)	0.40 (1/64)
3	9.53 (3/8)	0.40–0.79 (1/64–1/32)
4	12.7 (1/2)	0.79 (1/32)
8	12.7 (1/2)	0.79–1.59 (1/32–1/16)
16	19.1 (3/4)	0.79–1.59 (1/32–1/16)

A **piezometer ring** is a toroidal manifold into which are connected several sidewall static taps located around the perimeter of a common cross section. Its intent is to give an average pressure if differences in pressure other than those due to static head exist around the perimeter. However, there is generally no assurance that a true average is provided thereby. The principal advantage of the ring is that use of several holes in place of a single hole reduces the possibility of completely plugging the static openings.

For information on prediction of static-hole error, see Shaw, *J. Fluid Mech.,* **7,** 550–564 (1960); Livesey, Jackson, and Southern, *Aircr. Eng.,* **34,** 43–47 (February 1962).

For nonnewtonian fluids, pressure readings with taps may also be low because of fluid-elasticity effects. This error can be largely eliminated by using flush-mounted diaphragms.

For information on the pressure-hole error for nonnewtonian fluids, see Han and Kim, *Trans. Soc. Rheol.,* **17,** 151–174 (1973); Novotny and Eckert, *Trans. Soc. Rheol.,* **17,** 227–241 (1973); and Higashitani and Lodge, *Trans. Soc. Rheol.,* **19,** 307–336 (1975).

10.1.8. VELOCITY MEASUREMENTS

Measurement of flow can be based on the measurement of velocity in ducts or pipes by using devices such as pitot tubes and hot wire anemometers. The local velocity is measured at various sections o f a conduit and then averaged for the area under consideration.

$$\frac{w}{\rho} = A \times V = Q$$

(10-12)

where w = mass flow rate, lb_m/s, kg/s

ρ = density, lb_m/ft^3, kg/m^3

A = area, ft/s, m/s

V = velocity, ft/s, m/s

Q = volumetric flow rate, ft^3/s, m^3/s

Equation (10-12) shows that the fluid density directly affects the relationship between mass flow rate and both velocity and volumetric flow rate. Liquid temperature affects liquid density and hence volumetric flow rate at a constant mass flow rate. Liquid density is relatively insensitive to pressure. Both temperature and pressure affect gas density and thus volumetric flow rate.

Variables Affecting Measurement Flow measurement methods may sense local fluid velocity, volumetric flow rate, total or cumulative volumetric flow (the integral of volumetric flow rate with respect to elapsed time), mass flow rate, and total mass flow.

Velocity Profile Effects Many variables can influence the accuracy of specific flow measurement methods. For example, the velocity profile in a closed conduit affects many types of flow-measuring devices. The velocity of a fluid varies from zero at the wall and at other stationary solid objects in the flow channel to a maximum at a distance from the wall. In the entry region of a conduit, the velocity field may approach plug flow and a constant velocity across the conduit, dropping to zero only at the wall. As a newtonian fluid progresses down a pipe, a velocity profile develops that is parabolic for laminar flow [Eq. (6-41)] and that approaches plug flow for highly turbulent flow. Once a steady flow profile has developed, the flow is said to be fully developed; the length of conduit necessary to achieve fully developed flow is called the entrance region. For long cylindrical, horizontal pipe ($L < 40D$, where D is the inside diameter of the pipe and L is the upstream length of pipe), the velocity profile becomes fully developed. Velocity profiles in flowing fluids are discussed in greater detail in Sec. 6 (p. 6-11).

For steady-state, isothermal, single-phase, uniform, fully developed newtonian flow in straight pipes, the velocity is greatest at the center of the channel and symmetric about the axis of the pipe. Of those flowmeters that are dependent on the velocity profile, they are usually calibrated for this type of flow. Thus any disturbances in flow conditions can affect flowmeter readings.

Upstream and downstream disturbances in the flow field are caused by valves, elbows, and other types of fittings. Two upstream elbows in two perpendicular planes will impart swirl in the fluid downstream. Swirl, similar to atypical velocity profiles, can lead to erroneous flow measurements. Although the effect is not as great as in upstream flow disturbances, downstream flow disturbances can also lead to erroneous flow measurements.

Other Flow Disturbances Other examples of deviations from fully developed, single-phase newtonian flow include nonnewtonian flow, pulsating flow, cavitation, multiphase flow, boundary layer flows, and nonisothermal flows. See Sec. 6.

Pitot Tubes The combination of pitot tubes in conjunction with sidewall static taps measures local or point velocities by measuring the difference between the total pressure and the static pressure. The pitot tube shown in Fig. 10-10 consists of an impact tube whose opening faces directly into the stream to measure impact pressure, plus one or more sidewall taps to measure local static pressure.

Dynamic pressure may be measured by use of a pitot tube that is a simple impact tube. These tubes measure the pressure at a point where the velocity of the fluid is brought to zero. Pitot tubes must be parallel to the flow. The pitot tube is sensitive to yaw or angle attack. In general angles of attack over 10° should be avoided. In cases where the flow direction is unknown, it is recommended to use a Kiel probe. Figure 10-11 shows a Kiel probe. This probe will read accurately to an angle of about 22° with the flow.

The combined pitot-static tube shown in Fig. 10-12 consists of a jacketed impact tube with one or more rows of holes, 0.51 to 1.02 mm (0.02 to 0.04 in) in diameter, in the jacket to measure the static pressure. Velocity V_0 m/s (ft/s) at the point where the tip is located is given by

$$V_0 = C\sqrt{2g_c \Delta h} = C\sqrt{2g_c(P_T - P_S)/\rho_0}$$

(10-13)

where C = coefficient, dimensionless; g_c = dimensional constant; Δh = dynamic pressure ($\Delta h_s g/g_c$), expressed in (N·m)/kg[(ft·lbf)/lb or ft of fluid flowing]; Δh_s = differential height of static liquid column corresponding to Δh; g = local acceleration due to gravity; g_c = dimensional constant; p_i = impact pressure; p_0 = local static pressure; and ρ_0 = fluid density measured at pressure p_0 and the local temperature. With gases at velocities above 60 m/s (about 200 ft/s), compressibility becomes important, and the following equation should be used:

$$V_0 = C\sqrt{\frac{2g_c k}{k-1}\left(\frac{p_0}{\rho_0}\right)\left[\left(\frac{p_i}{p_0}\right)^{(k-1)/k} - 1\right]}$$

(10-14)

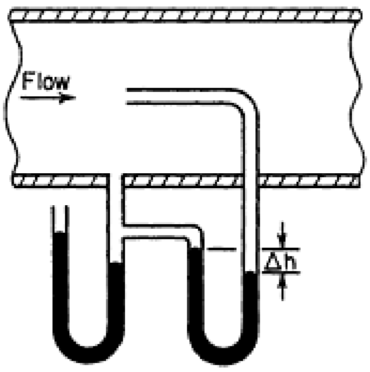

Figure 10-10. Pitot tube with sidewall static tap.

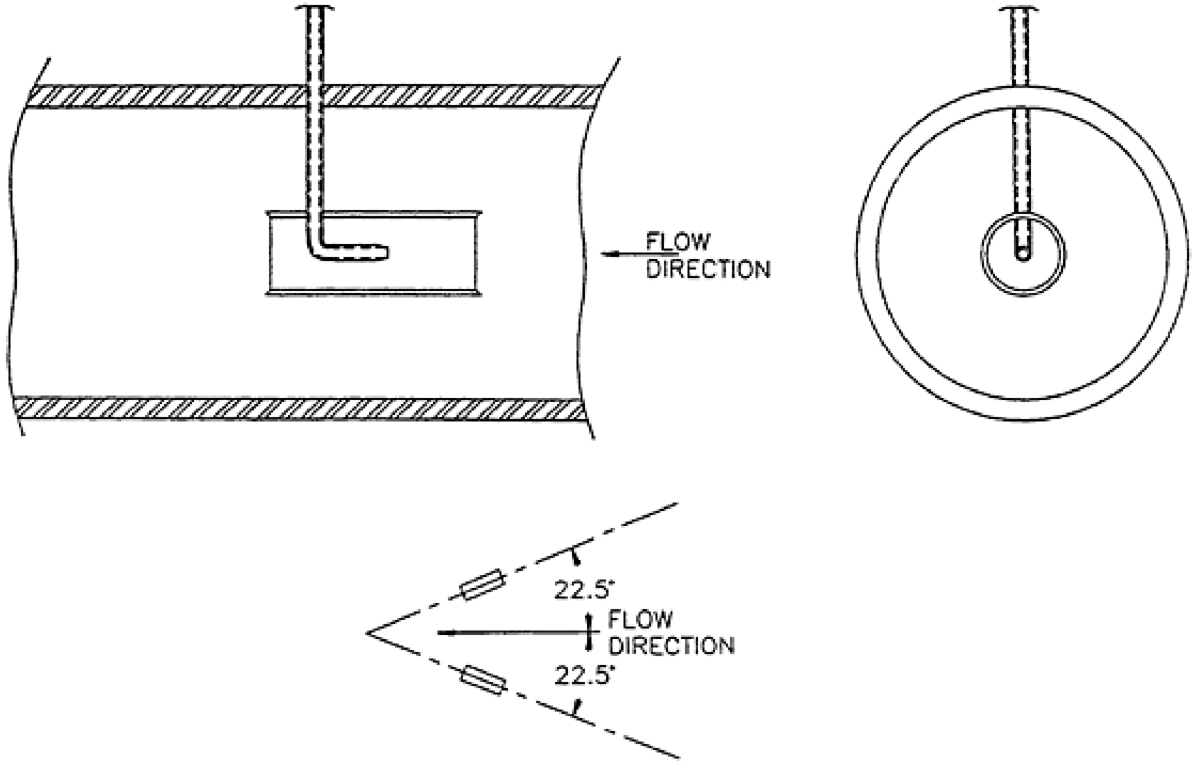

Figure 10-11. Kiel probe. Accurate measurements can be made at angles up to 22.5° with the flow stream.

where k is the ratio of specific heat at constant pressure to that at constant volume. (See ASME Research Committee on Fluid Meters Report, op. cit., p. 105.) Coefficient C is usually close to 1.00 (±0.01) for simple pitot tubes (Fig. 10-10) and generally ranges between 0.98 and 1.00 for pitot-static tubes (Fig. 10-12).

There are certain limitations on the range of usefulness of pitot tubes. With gases, the differential is very small at low velocities; e.g., at 4.6 m/s (15.1 ft/s) the differential is only about 1.30 mm (0.051 in) of water (20°C) for air at 1 atm (20°C), which represents a lower limit for 1 percent error even when one uses a micromanometer with a precision of 0.0254 mm (0.001 in) of water. Equation does not apply for Mach numbers greater than 0.7 because of the interference of shock waves. For supersonic flow, local Mach numbers can be calculated from a knowledge of the dynamic and true static pressures. The free stream Mach number (M_∞) is defined as the ratio of the speed of the stream (V_∞) to the speed of sound in the free stream:

$$A_\infty = \sqrt{\left(\frac{\partial P}{\partial \rho}\right)_{s=c}}$$

(10-15)

$$M_\infty = \frac{V_\infty}{\sqrt{\left(\frac{\partial P}{\partial \rho}\right)_{s=c}}}$$

(10-16)

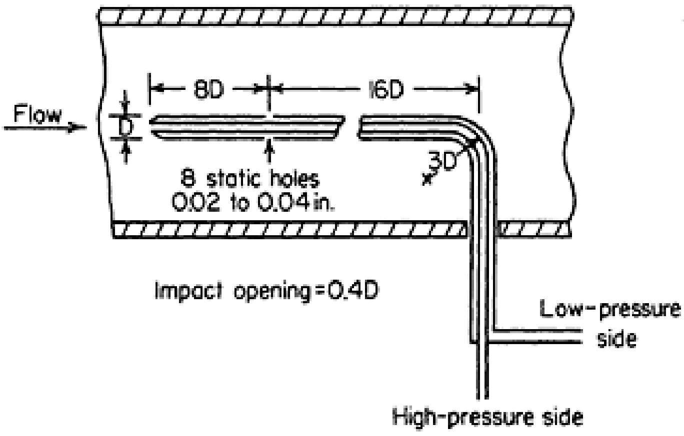

Figure 10-12. Pitot-static tube.

where S is the entropy. For isentropic flow, this relationship and pressure can be written as:

$$M_\infty = \frac{V_\infty}{\sqrt{kRT_s}}$$
(10-17)

The relationships between total and static temperature and pressure are given by the following relationship:

$$\frac{T_T}{T_S} = 1 + \frac{k-1}{2} M^2$$
(10-18)

$$\frac{P_T}{P_S} = \left(1 + \frac{k-1}{2} M^2\right)^{(k-1)/k}$$
(10-19)

With **liquids** at low velocities, the effect of the Reynolds number upon the coefficient is important. The coefficients are appreciably less than unity for Reynolds numbers less than 500 for pitot tubes and for Reynolds numbers less than 2300 for pitot-static tubes [see Folsom, *Trans. Am. Soc. Mech. Eng.,* **78,** 1447–1460 (1956)]. Reynolds numbers here are based on the probe outside diameter. Operation at low Reynolds numbers requires prior calibration of the probe.

The pitot-static tube is also sensitive to **yaw** or **angle of attack** than is the simple pitot tube because of the sensitivity of the static taps to orientation. The error involved is strongly dependent upon the exact probe dimensions. In general, angles greater than 10° should be avoided if the velocity error is to be 1 percent or less.

Disturbances upstream of the probe can cause large errors, in part because of the turbulence generated and its effect on the static-pressure measurement. A calming section of at least 50 pipe diameters is desirable. If this is not possible, the use of straightening vanes or a honeycomb is advisable.

The effect of **pulsating flow** on pitot-tube accuracy is treated by Ower et al., op. cit., pp. 310–312. For sinusoidal velocity fluctuations, the ratio of indicated velocity to actual mean velocity is given by the factor $\sqrt{1 + \lambda^2/2}$, where λ is the velocity excursion as a fraction of the mean velocity. Thus, the indicated velocity would be about 6 percent high for velocity fluctuations of ±50 percent, and pulsations greater than ±20 percent should be damped to avoid errors greater than 1 percent. The error increases as the frequency of flow oscillations approaches the natural frequency of the pitot tube and the density of the measuring fluid approaches the density of the process fluid [see Horlock and Daneshyar, *J. Mech. Eng. Sci.,* **15,** 144–152 (1973)].

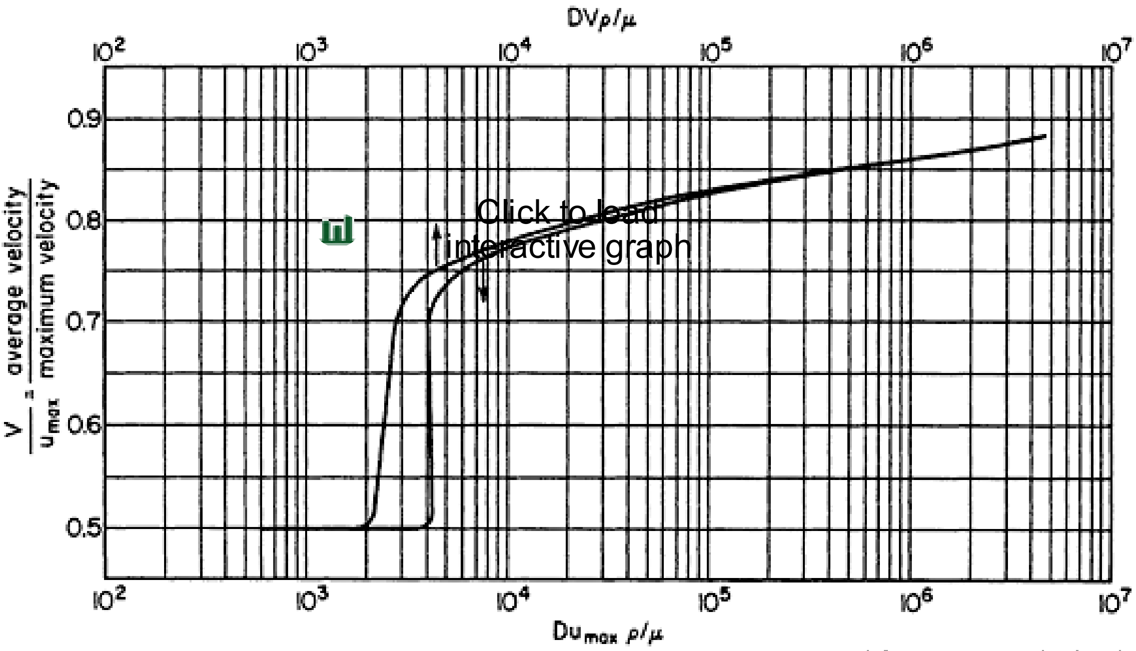

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Figure 10-13. Velocity ration versus Reynolds number for smooth circular pipes. [Based on data from Rothfus, Archer, Klimas, and Sikchi, Am. Inst. Chem. Eng. J., 3, 208 (1957).]

Pressures substantially lower than true impact pressures are obtained with pitot tubes in turbulent flow of dilute polymer solutions [see Halliwell and Lewkowicz, *Phys. Fluids,* **18,** 1617–1625 (1975)].

Special Tubes A variety of special forms of the pitot tube have been evolved. Folsom (loc. cit.) gives a description of many of these special types together with a comprehensive bibliography. Included are the impact tube for **boundary-layer** measurements and **shielded total-pressure tubes.** The latter are insensitive to angle of attack up to 40°.

Chue [*Prog. Aerosp. Sci.,* **16,** 147–223 (1975)] reviews the use of the pitot tube and allied pressure probes for impact pressure, static pressure, dynamic pressure, flow direction and local velocity, skin friction, and flow measurements.

A reversed pitot tube, also known as a **pitometer,** has one pressure opening facing upstream and the other facing downstream. Coefficient C for this type is on the order of 0.85. This gives about a 40 percent increase in pressure differential as compared with standard pitot tubes and is an advantage at low velocities. There are commercially available very compact types of pitometers which require relatively small openings for their insertion into a duct.

The **pitot-venturi** flow element is capable of developing a pressure differential 5 to 10 times that of a standard pitot tube. This is accomplished by employing a pair of concentric venturi elements in place of the pitot probe. The low-pressure tap is connected to the throat of the inner venturi, which in turn discharges into the throat of the outer venturi. For a discussion of performance and application of this flow element, see Stoll, *Trans. Am. Soc. Mech. Eng.,* **73,** 963–969 (1951).

Traversing for Mean Velocity Mean velocity in a duct can be obtained by dividing the cross section into a number of equal areas, finding the local velocity at a representative point in each, and averaging the results. In the case of **rectangular passages,** the cross section is usually divided into small squares or rectangles and the velocity is found at the center of each. In circular pipes, the cross section is divided into several equal annular areas as shown in Fig. 10-13. Readings of velocity are made at the intersections of a diameter and the set of circles which bisect the annuli and the central circle.

For an N-point traverse on a circular cross section, make readings on each side of the cross section at

$$100 \times \sqrt{(2n-1)/N} \text{ percent} \quad (n = 1, 2, 3 \text{ to } N/2)$$

of the pipe radius from the center. Traversing several diameters spaced at equal angles about the pipe is required if the velocity distribution is unsymmetrical. With a normal velocity distribution in a circular pipe, a 10-point traverse theoretically gives a mean velocity 0.3 percent high; a 20-point traverse, 0.1 percent high.

For normal velocity distribution in straight circular pipes at locations preceded by runs of at least 50 diameters without pipe fittings or other obstructions, the graph in Fig. 10-13 shows the ratio of mean velocity V to velocity at the center u_{max} plotted against the Reynolds number, where D = inside pipe diameter, ρ = fluid density, and μ = fluid viscosity, all in consistent units. Mean velocity is readily determined from this graph and a pitot reading at the center of the pipe if the quantity $Du_{max}\rho/\mu$ is less than 2000 or greater than 5000. The method is unreliable at intermediate values of the Reynolds number.

Methods for determining mean flow rate from probe measurements under nonideal conditions are described by Mandersloot, Hicks, and Langejan [*Chem. Eng.* (*London*), no. 232, CE370-CE380 (1969)].

The **hot-wire anemometer** consists essentially of an electrically heated fine wire (generally platinum) exposed to the gas stream whose velocity is being measured. An increase in fluid velocity, other things being equal, increases the rate of heat flow from the wire to the gas, thereby tending to cool the wire and alter its electrical resistance. In a constant-current anemometer, gas velocity is determined by measuring the resulting wire resistance; in the constant-resistance type, gas velocity is determined from the current required to maintain the wire temperature, and thus the resistance, constant. The difference in the two types is primarily in the electric circuits and instruments employed.

The hot-wire anemometer can, with suitable calibration, accurately measure velocities from about 0.15 m/s (0.5 ft/s) to supersonic velocities and detect velocity fluctuations with frequencies up to 200,000 Hz. Fairly rugged, inexpensive units can be built for the measurement of mean velocities in the range of 0.15 to 30 m/s (about 0.5 to 100 ft/s). More elaborate, compensated units are commercially available for use in unsteady flow and turbulence measurements. In calibrating a hot-wire anemometer, it is preferable to use the same gas, temperature, and pressure as will be encountered in the intended application. In this case the quantity $I^2R_w/\Delta t$

can be plotted against $\sqrt{V}$, where I = hot-wire current, R_w = hot-wire resistance, Δt = difference between the wire temperature and the gas bulk temperature, and V = mean local velocity. A procedure is given by Wasan and Baid [*Am. Inst. Chem. Eng. J.,* **17,** 729–731 (1971)] for use when it is impractical to calibrate with the same gas composition or conditions of temperature and pressure. Andrews, Bradley, and Hundy [*Int. J. Heat Mass Transfer,* **15,** 1765–1786 (1972)] give a calibration correlation for measurement of small gas velocities. The hot-wire anemometer is treated in considerable detail in Dean, op. cit., chap. VI; in Ladenburg et al., op. cit., art. F-2; by Grant and Kronauer, *Symposium on Measurement in Unsteady Flow,* American Society of Mechanical Engineers, New York, 1962, pp. 44–53; ASME Research Committee on Fluid Meters Report, op. cit., pp. 105–107; and by Compte-Bellot, *Ann. Rev. Fluid Mech.,* **8,** pp. 209–231 (1976).

The hot-wire anemometer can be modified for liquid measurements, although difficulties are encountered because of bubbles and dirt adhering to the wire. See Stevens, Borden, and Strausser, David Taylor Model Basin Rep. 953, December 1956; Middlebrook and Piret, *Ind. Eng. Chem.,* **42,** 1511–1513 (1950); and Piret et al., *Ind. Eng. Chem.,* **39,** 1098–1103 (1947).

The **hot-film anemometer** has been developed for applications in which use of the hot-wire anemometer presents problems. It consists of a platinum-film sensing element deposited on a glass substrate. Various geometries can be used. The most common involves a wedge with a 30° included angle at the end of a tapered rod. The wedge is commonly 1 mm (0.039 in) long and 0.2 mm (0.0079 in) wide on each face. Compared with the hot wire, it is less susceptible to fouling by bubbles or dirt when used in liquids, has greater mechanical strength when used with gases at high velocities and high temperatures, and can give a higher signal-to-noise ratio. For additional information see Ling and Hubbard, *J. Aeronaut. Sci.,* **23,** 890–891 (1956); and Ling, *J. Basic Eng.,* **82,** 629–634 (1960).

The **heated-thermocouple anemometer** measures gas velocity from the cooling effect of the gas stream flowing across the hot junctions of a thermopile supplied with constant electrical power input. Alternate junctions are maintained at ambient temperature, thus compensating for the effect of ambient temperature. For details see Bunker, *Proc. Instrum. Soc. Am.,* **9,** pap. 54-43-2 (1954).

A glass-coated bead **thermistor anemometer** can be used for the measurement of low fluid velocities, down to 0.001 m/s (0.003 ft/s) in air and 0.0002 m/s (0.0007 ft/s) in water [see Murphy and Sparks, *Ind. Eng. Chem. Fundam.,* **7,** 642–645 (1968)].

The **laser-Doppler anemometer** measures local fluid velocity from the change in frequency of radiation, between a stationary source and a receiver, due to scattering by particles along the wave path. A laser is commonly used as the source of incident illumination. The measurements are essentially independent of local temperature and pressure. This technique can be used in many different flow systems with transparent fluids containing particles whose velocity is actually measured. For a brief review of the laser-Doppler technique see Goldstein, *Appl. Mech. Rev.,* **27,** 753–760 (1974). For additional details see Durst, Melling, and Whitelaw, *Principles and Practice of Laser-Doppler Anemometry,* Academic, New York, 1976.

10.1.9. FLOWMETERS

In the process industries, flow measurement devices are the largest market in the process instrumentation field. Two web sites for process equipment and instrumentation, www.globalspec.com, and www.thomasnet.com, both list more than 800 companies that offer flow measurement products. There are more than one hundred types of flowmeters commercially available. The aforementioned web sites not only facilitate selection and specification of commercial flowmeters, but also provide electronic access to manufacturers' technical literature.

Devices that measure flow can be categorized in two areas as follows:

1. All types of measuring devices in which the material passes without being divided into isolated quantities. Movement of the material is usually sensed by a primary measuring element which activates a secondary device. The flow rate is then inferred from the response of the secondary device by means of known physical laws or from empirical relationships.

2. A positive-displacement meter, which applies to a device in which the flow is divided into isolated measured volumes. The number of fillings of these known volumes are measured with respect to time.

The most common application of flow measurement in process plants is flow in pipes, ducts, and tubing. Table 10-4 lists widely used flowmeters for these closed conduits as well as the two major classes of open-channel flowmeters. Table 10-4 also lists many other types of flowmeters that are discussed later in this subsection.

This subsection summarizes selection and installation of flowmeters, including the measurement of pressure and velocities of fluids when the flow measurement technique requires it.

10.1.10. INDUSTRY GUIDELINES AND STANDARDS

Because flow measurement is important, many engineering societies and trade organizations have developed flow-related guidelines, standards, and other publications (Table 10-5). The reader should consult the appropriate standards when specifying, installing, and calibrating flow measurement systems.

There are also numerous articles in scholarly journals, trade magazines, and manufacturers' literature related to flow measurement.

Different types of flowmeters differ markedly in their degrees of sensitivity to flow disturbances. In the most extreme cases, obtaining highly accurate flow measurements with certain types of flowmeters may require $60D$ upstream straight pipe and $20D$ downstream. Valves can be particularly problematic because their effects on a flowmeter vary with valve position. Numerous types of flow straighteners or conditioners, as shown in Fig. 10-9, can significantly reduce the required run of straight pipe upstream of a given flowmeter.

10.1.11. CLASSIFICATION OF FLOWMETERS

Table 10-4 lists the major classes of flowmeters, along with common examples of each. Brief descriptions are provided in this subsection, followed by more details in subsequent subsections.

Differential Pressure Meters Differential pressure meters or head meters measure the change in pressure across a special flow element. The differential pressure increases with increasing flow rate. The pitot tubes described previously work on this principle. Other examples include orifices [see also Eqs. (6-111) and (8-102), and Fig. 10-14], nozzles (Fig. 10-19), targets, venturis (see also Sec. 8 and Fig. 10-17), and elbow meters. Averaging pitot tubes produce a pressure differential that is based on multiple measuring points across the flow path.

Differential pressure meters are widely used. Temperature, pressure, and density affect gas density and readings of differential pressure meters. For that reason, many commercial flowmeters that are based on measurement of differential pressure often have integral temperature and absolute pressure measurements in addition to differential pressure. They also frequently have automatic temperature and pressure compensation.

Velocity Meters Velocity meters measure fluid velocity. Examples include electromagnetic, propeller, turbine, ultrasonic Doppler, ultrasonic transit time, and vortex meters. Section 8 describes the principles of operation of electromagnetic, turbine, ultrasonic, and vortex flowmeters.

Mass Meters Mass flowmeters measure the rate of mass flow through a conduit. Examples include Coriolis flowmeters and thermal mass flowmeters. Coriolis flowmeters can measure fluid density simultaneously with mass flow rate. This permits calculation of volumetric flow rate as well. Section 8 includes brief descriptions of Coriolis and thermal mass flowmeters.

Volumetric Meters Volumetric meters (also called positive-displacement flowmeters) are devices that mechanically divide a fluid stream into discrete, known volumes and count the number of volumes that pass through the device. See Spitzer (2005, op. cit.).

Variable-Area Meters Variable-area meters, which are also called rotameters, offer popular and inexpensive flow measurement devices. These meters employ a float inside a tube that has an internal cross-sectional area that increases with distance upward in the flow path through the tube. As the flow rate increases, the float rises in the tube to provide a larger area for the flowing fluid to pass.

Open-Channel Flow Measurement Open-channel flow measurements are usually based on measurement of liquid level in a flow channel constructed of a specified geometry. The two most common flow channels used are weirs and flumes. See Spitzer (2005, op. cit.).

Table 10-4. Comparison of Flowmeter Technologies

Flowmeter technology	Accuracy* (+/–)	Turndown	Fluids†	Pipe sizes,‡ in	Maximum pressure,‡ psig	Temperature range,‡ °F	Pipe run	Relative pressure loss
Differential Pressure Meters								
Pitot		8:1	L, G	1 to 96	2.76E-4 to 200	–200 to 750		L
Averaging pitot	1% R	8:1	L, G, S	0.25 to 72	8800	–20 to 2370		L
Orifice							Long	
Square-edged	0.5 to 1.5% R	4:1	L, G, S	0.5 to 40	8800	–4 to 2300		M
Eccentric	2% R	4:1	L, G, S	0.5 to 40	8800	–4 to 2300		M
Segmental	2% R	4:1	L, G, S, SL	0.5 to 40	8800	–4 to 2300		M
Orifice and multi-variable flow transmitter	0.5 to 1% R	10:1	L, G, S	> 0.5	4000	1000		M
Venturi	0.5 to 1.5% R	10:1	L, G, S, SL	1 to 120	8800	–4 to 2300		L
Flow nozzle	0.5 to 2% R	8:1	L, G, S	2 to 80	>1000	<1000		M
V cone	0.5% F	10:1	L, G, S, SL	0.25 to 120	6000	To 400		M
Wedge	0.5 to 5% R	10:1	L, G, S, SL	0.5 to 24	>600	‡		M
Velocity Meters								
Correlation	0.5% R	10:1	L, G, SL	1 to 60	Piping limits	To 600	Long	L
Electromagnetic	0.2 to 2% R	10:1	L	0.15 to 60	5000	–40 to 350	Short	L
Propeller	2% R	15:1	L	2 to 12	230	0 to 300		M
Turbine	0.15 to 1% R	10:1	L, G	0.5 to 30	6000	–450 to 600	Short	M
Ultrasonic Doppler	1 to 30% R	50:1	L, G, SL	0.5 to 200	6000	–40 to 250	Long	L
Ultrasonic transit time	0.5 to 5% R	Down to zero flow	L, G	1 to 540	6000	–40 to 650	Long	L
Vortex	0.5 to 2% R	20:1	L, G, S	0.5 to 16	1500	–330 to 800	Short	M
Mass Meters								
Coriolis	0.1 to 0.3% R	10:1 to 80:1	L, G	0.06 to 12	5700	–400 to 800	None	L, M
Thermal (for gases)	1% F	50:1	G	0.125 to 8	4500	32 to 572	Short	L
Thermal (for liquids)	0.5% F	50:1	L	0.06 to 0.25	4500	40 to 165	Short	L
Volumetric	0.15 to 2% R	10:1	L	0.25 to 16	2000	–40 to 600	None	M to H
Variable area	1 to 5% F	10:1	L, G	0.125 to 6	6000	<1000	None	M
Open-Channel Flowmeters								
Weirs	2 to 5% R	25:1	L	Wide range	NA	NA		L
Flumes	3 to 10% R	40:1	L	Wide range	NA	NA		L

*F = full scale, R = rate. †L = liquid, G = gas, S = steam, SL = slurry.
‡Dependent on the material selection and application. Readers should consult manufacturers for current capabilities.
Adapted from J. Pomroy, *Chemical Engineering*, pp. 94–102, May 1996; J. W. Dolenc, *Chemical Engineering Progress*, pp. 22–32, Jan. 1996; R. C. Baker, *Introductory Guide to Flow Measurement*, American Society of Mechanical Engineers, New York, 2003; R. W. Miller, *Flow Measurement Engineering Handbook*, 3d ed., McGraw-Hill, New York, 1996; D. W. Spitzer, *Industrial Flow Measurement*, 3d ed., The Instrumentation, Systems, and Automation Society, Research Triangle Park, N.C., 2005; and manufacturers' literature at www.globalspec.com.

10.1.12. DIFFERENTIAL PRESSURE FLOWMETERS

General Principles If a constriction is placed in a closed channel carrying a stream of fluid, there will be an increase in velocity, and hence an increase in kinetic energy, at the point of constriction. From an energy balance, as given by Bernoulli's theorem [see Sec. 6, subsection "Energy Balance," Eq. (6-16)], there must be a corresponding reduction in pressure. Rate of discharge from the constriction can be calculated by knowing this pressure reduction, the area available for flow at the constriction, the density of the fluid, and the coefficient of discharge *C*. The last-named is defined as the ratio of actual flow to the theoretical flow and makes allowance for stream contraction and frictional effects. The metering characteristics of commonly used differential pressure meters are reviewed and grouped by Halmi [*J. Fluids Eng.*, **95,** 127–141 (1973)].

The term **static head** generally denotes the pressure in a fluid due to the head of fluid above the point in question. Its magnitude is given by the application of Newton's law (force = mass × acceleration). In the case of **liquids** (constant density), the static head p_h Pa (lbf/ft^2) is given by

$$p_h = h\rho g/g_c$$

(10-20)

where h = head of liquid above the point, m (ft); ρ = liquid density; g = local acceleration due to gravity; and g_c = dimensional constant.

The head developed in a compressor or pump is the energy force per unit mass. In the measuring systems it is often misnamed as (ft) while the units are really ft-lb/lbm or kilojoules.

For a compressor or turbine, it is represented by the following relationship:

$$E = U_1 V_{\theta 1} - U_2 V_{\theta 2}$$

(10-21)

Table 10-5. Guidelines, Standards, and Other Publications Related to Flow Measurement

Technical society	Number of guidelines and standards*
American Gas Association (AGA)	2
American Petroleum Institute (API)	11
American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE)	5
American Society of Mechanical Engineers (ASME)	18
ASTM International (ASTM)	17
British Standards Institution (BSI)	100
Deutsches Institut für Normung E. V. (DIN)	48
International Electrotechnical Commission (IEC)	6
Instrumentation, Systems, and Automation Society (ISA)	3
International Organization for Standardization (ISO)	212
SAE International (SAE)	6

*Number of documents identified by searching for *flow measurement* on `http://global.ihs.com`, the web site of a clearinghouse of industry guidelines, codes, and standards.

where U is the blade speed and V_θ is the tangential velocity component of absolute velocity. This equation is known as the Euler equation.

Orifice Meters A **square-edged** or **sharp-edged** orifice, as shown in Fig. 10-14, is a clean-cut square-edged hole with straight walls perpendicular to the flat upstream face of a thin plate placed crosswise of the channel. The stream issuing from such an orifice attains its minimum cross section (vena contracta) at a distance downstream of the orifice which varies with the ratio β of orifice to pipe diameter (see Fig. 10-15).

For a centered circular orifice in a pipe, the pressure differential is customarily measured between one of the following pressure-tap pairs. Except in the case of flange taps, all measurements of distance from the orifice are made from the upstream face of the plate.

1. *Corner taps.* Static holes drilled one in the upstream and one in the downstream flange, with the openings as close as possible to the orifice plate.

2. *Radius taps.* Static holes located one pipe diameter upstream and one-half pipe diameter downstream from the plate.

3. *Pipe taps.* Static holes located 2 1/2 pipe diameters upstream and eight pipe diameters downstream from the plate.

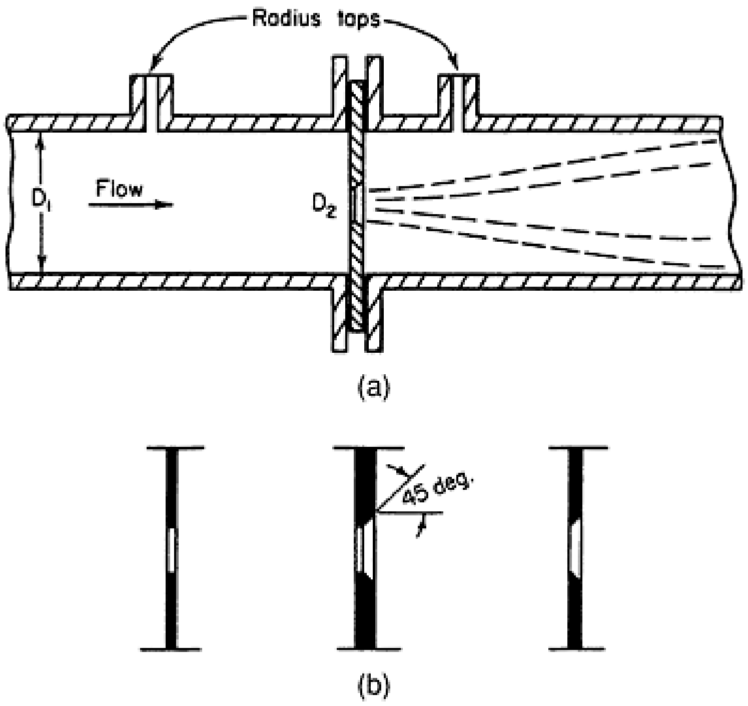

Figure 10-14. Square-edged or sharp-edged orifices. The plate at the orifice opening must not be thicker than one-thirtieth of the pipe diameter, one-eighth of the orifice diameter, or one-

fourth of the distance from the pipe wall to the edge of the opening. (a) Pipe-line orifice. (b) Types of plates.

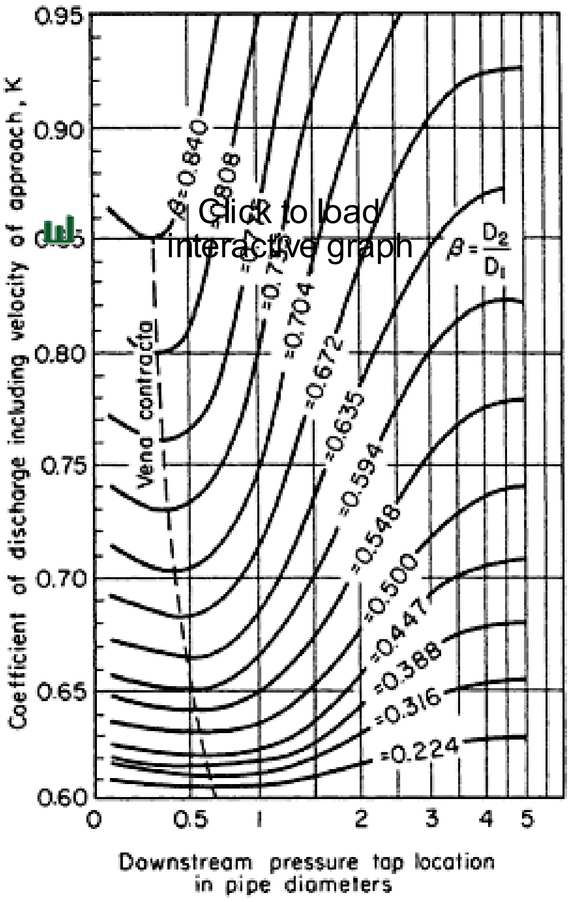

Figure 10-15. Coefficient of discharge for square-edged circular orifices for NRe > 30,000 with the upstream tap located between one and two pipe diameters from the orifice plate. [Spitzglass, Trans. Am. Soc. Mech. Eng., 44, 919 (1922).]

4. *Flange taps.* Static holes located 25.4 mm (1 in) upstream and 25.4 mm (1 in) downstream from the plate.

5. *Vena-contracta taps.* The upstream static hole is one-half to two pipe diameters from the plate. The downstream tap is located at the position of minimum pressure (see Fig. 10-15).

Radius taps are best from a practical standpoint; the downstream pressure tap is located at about the mean position of the vena contracta, and the upstream tap is sufficiently far upstream to be unaffected by distortion of the flow in the immediate vicinity of the orifice (in practice, the upstream tap can be as much as two pipe diameters from the plate without affecting the results). Vena-contracta taps give the largest differential head for a given rate of flow but are inconvenient if the orifice size is changed from time to time. Corner taps offer the sometimes great advantage that the pressure taps can be built into the plate carrying the orifice. Thus the entire apparatus can be quickly inserted in a pipe line at any convenient flanged joint without having to drill holes in the pipe. Flange taps are similarly convenient, since by merely replacing standard flanges with special orifice flanges, suitable pressure taps are made available. Pipe taps give the lowest differential pressure, the value obtained being close to the permanent pressure loss.

The practical working equation for weight rate of discharge, adopted by the ASME Research Committee on Fluid Meters for use with either gases or liquids, is

$$w = q_1\rho_1 = CYA_2\sqrt{\frac{2g_c(p_1 - p_2)\rho_1}{1 - \beta^4}}$$

$$= KYA_2\sqrt{2g_c(p_1 - p_2)\rho_1}$$

(10-22)

where A_2 = cross-sectional area of throat; C = coefficient of discharge, dimensionless; g_c = dimensional constant; $K = C/\sqrt{1 - \beta^4}$, dimensionless; p_1, p_2 = pressure at upstream and downstream static pressure taps respectively; q_1 = volumetric rate of discharge measured at upstream pressure and temperature; w = weight rate of discharge; Y = expansion factor, dimensionless; β = ratio of throat diameter to pipe diameter, dimensionless; and ρ_1 = density at upstream pressure and temperature.

For the case of subsonic flow of a gas ($r_c < r < 1.0$), the expansion factor Y for orifices is approximated by

$$Y = 1 - [(1 - r)/k](0.41 + 0.35\beta^4)$$

(10-23)

where r = ratio of downstream to upstream static pressure (p_2/p_1), k = ratio of specific heats (c_p/c_v), and β=diameter ratio. (See also Fig. 10-18.) Values of Y for supercritical flow of a gas ($r < r_c$) through orifices are given by Benedict [*J. Basic Eng.,* **93,** 121–137 (1971)]. For the case of **liquids,** expansion factor Y is unity, and Eq. (10-27) should be used, since it allows for any difference in elevation between the upstream and downstream taps.

Coefficient of discharge C for a given orifice type is a function of the Reynolds number N_{Re} (based on orifice diameter and velocity) and diameter ratio β. At Reynolds numbers greater than about 30,000, the coefficients are substantially constant. For square-edged or sharp-edged concentric circular orifices, the value will fall between 0.595 and 0.620 for vena-contracta or radius taps for β up to 0.8 and for flange taps for β up to 0.5. Figure 10-15 gives the coefficient of discharge K, including the velocity-of-approach factor ($1/\sqrt{1 - \beta^4}$), as a function of β and the location of the downstream tap. Precise values of K are given in *ASME PTC,* op. cit., pp. 20–39, for flange taps, radius taps, vena-contracta taps, and corner taps. Precise values of C are given in the ASME Research Committee on Fluid Meters Report, op. cit., pp. 202–207, for the first three types of taps.

The discharge coefficient of sharp-edged orifices was shown by Benedict, Wyler, and Brandt [*J. Eng. Power,* **97,** 576–582 (1975)] to increase with edge roundness. Typical as-purchased orifice plates may exhibit deviations on the order of 1 to 2 percent from ASME values of the discharge coefficient.

In the transition region (N_{Re} between 50 and 30,000), the coefficients are generally higher than the above values. Although calibration is generally advisable in this region, the curves given in Fig. 10-16 for corner and vena-contracta taps can be used as a guide. In the laminar-flow region ($N_{Re} < 50$), the coefficient C is proportional to $\sqrt{N_{Re}}$. For $1 < N_{Re} < 100$, Johansen [*Proc. R. Soc.* (*London*), **A121,** 231–245 (1930)] presents discharge-coefficient data for sharp-edged orifices with corner taps. For $N_{Re} < 1$, Miller and Nemecek [ASME Paper 58-A-106 (1958)] present correlations giving coefficients for sharp-edged orifices and short-pipe orifices (L/D from 2 to 10). For short-pipe orifices (L/D from 1 to 4), Dickerson and Rice [*J. Basic Eng.,* **91,** 546–548 (1969)] give coefficients for the intermediate range ($27 < N_{Re} < 7000$). See also subsection "Contraction and Entrance Losses."

Permanent pressure loss across a concentric circular orifice with radius or vena-contracta taps can be approximated for turbulent flow by

$$(p_1 - p_4)/(p_1 - p_2) = 1 - \beta^2$$

(10-24)

where p_1, p_2 = upstream and downstream pressure-tap readings respectively, p_4 = fully recovered pressure (four to eight pipe diameters downstream of the orifice), and β=diameter ratio. See *ASME PTC,* op. cit., Fig. 5.

See Benedict, *J. Fluids Eng.,* **99,** 245–248 (1977), for a general equation for pressure loss for orifices installed in pipes or with plenum inlets. Orifices show higher loss than nozzles or venturis. Permanent pressure loss for laminar flow depends on the Reynolds number in addition to β. See Alvi, Sridharan, and Lakshmana Rao, loc. cit., for details.

For the case of **critical flow** through a square- or sharp-edged concentric circular orifice (where $r \leq r_c$, as discussed earlier in this subsection), use Eqs. (10-31), (10-32), and (10-33) as given for critical-flow nozzles. However, unlike nozzles, the flow through a sharp-edged orifice continues to increase as the downstream pressure drops below that corresponding to the critical pressure ratio r_c. This is due to an increase in the cross section of the vena contracta as the downstream pressure is reduced, giving a corresponding increase in the coefficient of discharge. At $r = r_c$, C is about 0.75, while at $r \cong 0$, C has increased to about 0.84. See Grace and Lapple, loc. cit.; and Benedict, *J. Basic Eng.,* **93,** 99–120 (1971).

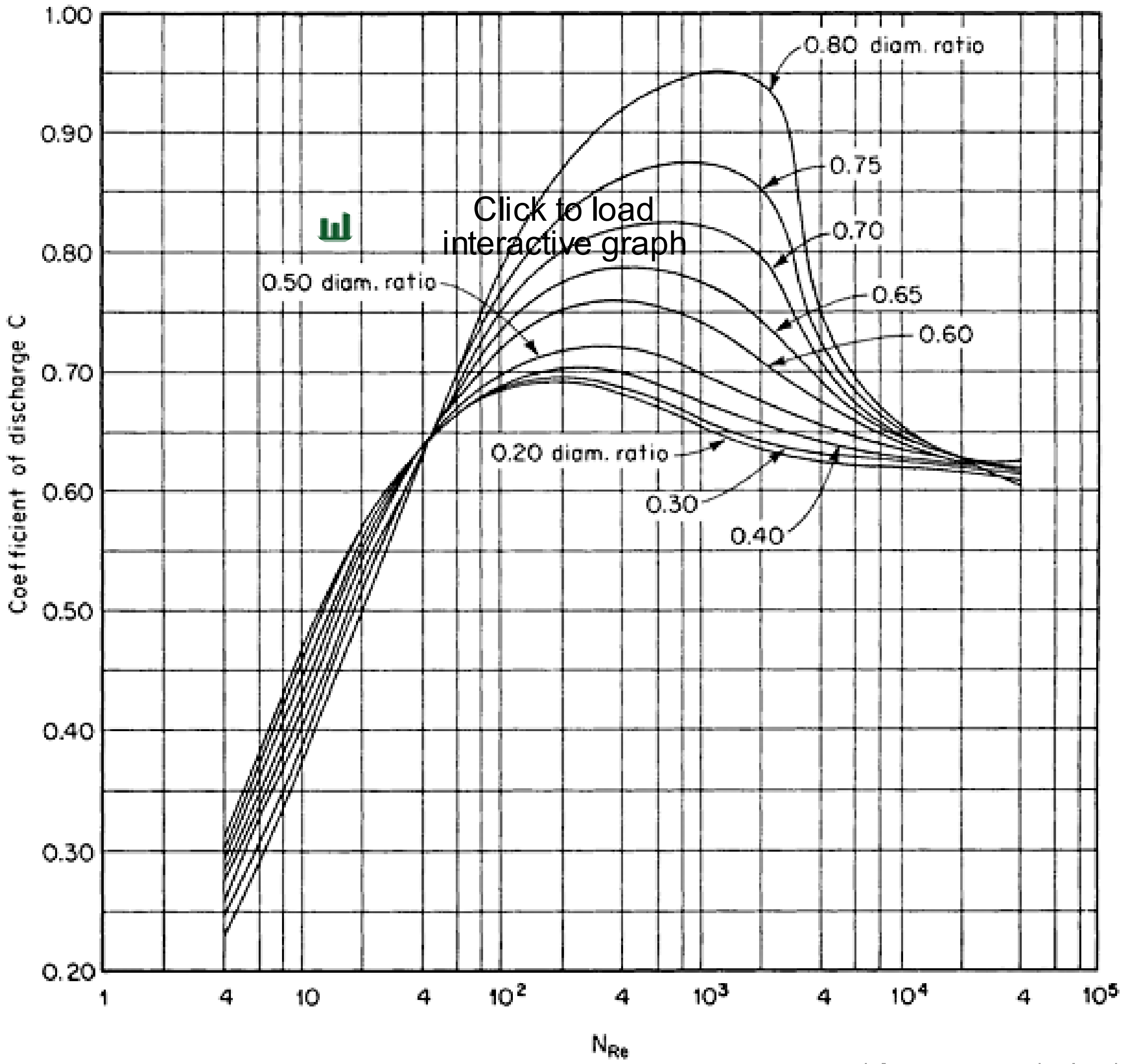

Figure 10-16. Coefficient of discharge for square-edged circular orifices with corner taps. [Tuve and Sprenkle, Instruments, 6, 201 (1933).]

Measurements by Harris and Magnall [*Trans. Inst. Chem. Eng.* (*London*), **50,** 61–68 (1972)] with a venturi ($\beta = 0.62$) and orifices with radius taps ($\beta = 0.60 - 0.75$) indicate that the discharge coefficient for **nonnewtonian fluids,** in the range N'_{Re} (generalized Reynolds number) 3500 to 100,000, is approximately the same as for newtonian fluids at the same Reynolds number.

Quadrant-edge orifices have holes with rounded edges on the upstream side of the plate. The quadrant-edge radius is equal to the thickness of the plate at the orifice location. The advantages claimed for this type versus the square- or sharp-edged orifice are constant-discharge coefficients extending to lower Reynolds numbers and less possibility of significant changes in coefficient because of erosion or other damage to the inlet shape.

Values of discharge coefficient *C* and Reynolds numbers limit for constant *C* are presented in Table 10-6, based on Ramamoorthy and Seetharamiah [*J. Basic Eng.,* **88,** 9–13 (1966)] and Bogema and Monkmeyer (*J. Basic Eng.,* **82,** 729–734 (1960)]. At Reynolds numbers above those listed for the upper limits, the coefficients rise abruptly. As Reynolds numbers decrease below those listed for the lower limits, the coefficients pass through a hump and then drop off. According to Bogema, Spring, and Ramamoorthy [*J. Basic Eng.,* **84,** 415–418 (1962)], the hump can be eliminated by placing a fine-mesh screen about three pipe diameters upstream of the orifice. This reduces the lower N_{Re} limit to about 500.

Permanent pressure loss across quadrant-edge orifices for turbulent flow is somewhat lower than given by Eq. (10-24). See Alvi, Sridharan, and Lakshmana Rao, loc. cit., for values of discharge coefficient and permanent pressure loss in laminar flow.

Slotted orifices offer significant advantages over a standard square-edged orifice with an identical open area for homogeneous gases or liquids [G. L. Morrison and K. R. Hall, *Hydrocarbon Processing* **79,** 12, 65–72 (2000)]. The slotted orifice flowmeter only requires compact header configurations with very short upstream pipe lengths and maintains accuracy in the range of 0.25 percent with no flow conditioner. Permanent head loss is less than or equal to that of a standard orifice that has the same β ratio. Discharge coefficients for the slotted orifice are much less sensitive to swirl or to axial velocity profiles. A slotted orifice plate can be a "drop in" replacement for a standard orifice plate.

Segmental and **eccentric orifices** are frequently used for gas metering when there is a possibility that entrained liquids or solids would otherwise accumulate in front of a concentric circular orifice. This can be avoided if the opening is placed on the lower side of the pipe. For liquid flow with entrained gas, the opening is placed on the upper side. The pressure taps should be located on the opposite side of the pipe from the opening.

Coefficient *C* for a square-edged eccentric circular orifice (with opening tangent to pipe wall) varies from about 0.61 to 0.63 for β's from 0.3 to 0.5, respectively, and pipe Reynolds numbers > 10,000 for either vena-

Table 10-6. Dischar ge Coefficients for Quadrant-Edge Orifices

			Limiting N^*_{Re} for constant coefficient†	
β	***C*‡**	***K*‡**	**Lower**	**Upper**
0.225	0.770	0.771	5,000	60,000
0.400	0.780	0.790	5,000	150,000
0.500	0.824	0.851	4,000	200,000
0.600	0.856	0.918	3,000	120,000
0.630	0.885	0.964	3,000	105,000

*Based on pipe diameter and velocity.

†For a precision of about ±0.5 percent.

‡Can be used with corner taps, flange taps, or radius taps.

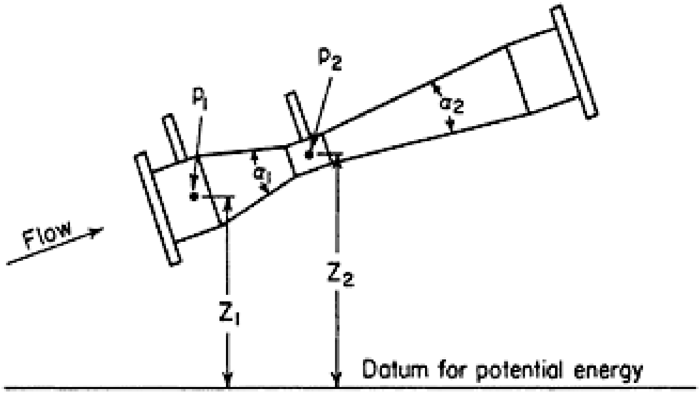

Figure 10-17. Herschel-type venturi tube.

contracta or flange taps (where β = diameter ratio). For square-edged segmental orifices, the coefficient C falls generally between 0.63 and 0.64 for $0.3 \le \beta \le 0.5$ and pipe Reynolds numbers > 10,000, for vena-contracta or flange taps, where β=diameter ratio for an equivalent circular orifice = $\sqrt{\alpha}$ (α = ratio of orifice to pipe cross-sectional areas). Values of expansion factor Y are slightly higher than for concentric circular orifices, and the location of the vena contracta is moved farther downstream as compared with concentric circular orifices. For further details, see ASME Research Committee on Fluid Meters Report, op. cit., pp. 210–213.

For permanent pressure loss with segmental and eccentric orifices with laminar pipe flow see Lakshmana Rao and Sridharan, *Proc. Am. Soc. Civ. Eng., J. Hydraul. Div.,* **98** (HY 11), 2015–2034 (1972).

Annular orifices can also be used to advantage for gas metering when there is a possibility of entrained liquids or solids and for liquid metering with entrained gas present in small concentrations. Coefficient K was found by Bell and Bergelin [*Trans. Am. Soc. Mech. Eng.,* **79,** 593–601 (1957)] to range from about 0.63 to 0.67 for annulus Reynolds numbers in the range of 100 to 20,000 respectively for values of $2L/(D - d)$ less than 1 where L = thickness of orifice at outer edge, D = inside pipe diameter, and d = diameter of orifice disk. The annulus Reynolds number is defined as

$$N_{Re} = (D - d)(G/\mu)$$

(10-25)

where G = mass velocity ρV through orifice opening and μ=fluid viscosity. The above coefficients were determined for β's (= d/D) in the range of 0.95 to 0.996 and with pressure taps located 19 mm (3/4 in) upstream of the disk and 230 mm (9 in) downstream in a 5.25-in-diameter pipe.

Venturi Meters The standard Herschel-type venturi meter consists of a short length of straight tubing connected at either end to the pipe line by conical sections (see Fig. 10-17). Recommended proportions (*ASME PTC,* op. cit., p. 17) are entrance cone angle $\alpha_1 = 21 \pm 2°$, exit cone angle α_2 = 5 to 15°, throat length = one throat diameter, and upstream tap located 0.25 to 0.5 pipe diameter upstream of the entrance cone. The straight and conical sections should be joined by smooth curved surfaces for best results. **Rate of discharge** of either gases or liquids through a venturi meter is given by Eq. (10-22).

For the flow of **gases,** expansion factor *Y,* which allows for the change in gas density as it expands adiabatically from p_1 to p_2, is given by

$$Y = \sqrt{r^{2/k}\left(\frac{k}{k-1}\right)\left(\frac{1-r^{(k-1)/k}}{1-r}\right)\left(\frac{1-\beta^4}{1-\beta^4 r^{2/k}}\right)}$$

(10-26)

for venturi meters and flow nozzles, where $r = p_2/p_1$ and k = specific heat ratio c_p/c_v. Values of Y computed from Eq. (10-26) are given in Fig. 10-18 as a function of r, k, and β.

For the flow of **liquids,** expansion factor Y is unity. The change in potential energy in the case of an inclined or vertical venturi meter must be allowed for. Equation (10-22) is accordingly modified to give

$$w = q_1\rho = CA_2\sqrt{\frac{[2g_c(p_1-p_2)+2g\rho(Z_1-Z_2)]\rho}{1-\beta^4}}$$

(10-27)

where g = local acceleration due to gravity and Z_1, Z_2 = vertical heights above an arbitrary datum plane corresponding to the centerline pressure-reading locations for p_1 and p_2 respectively.

Value of the **discharge coefficient** C for a **Herschel-type venturi meter** depends upon the Reynolds number and to a minor extent upon the size of the venturi, increasing with diameter. A plot of C versus pipe Reynolds number is given in *ASME PTC,* op. cit., p. 19. A value of 0.984 can be used for pipe Reynolds numbers larger than 200,000.

Permanent pressure loss for a Herschel-type venturi tube depends upon diameter ratio β and discharge cone angle α_2. It ranges from 10 to 15 percent of the pressure differential ($p_1 - p_2$) for small angles (5 to 7°) and from 10 to 30 percent for large angles (15°), with the larger losses occurring at low values of β (see *ASME PTC,* op. cit., p. 12). See Benedict, *J. Fluids Eng.,* **99,** 245–248 (1977), for a general equation for pressure loss for venturis installed in pipes or with plenum inlets.

For flow measurement of **steam and water mixtures** with a Herschel-type venturi in 2 1/2-in- and 3-in-diameter pipes, see Collins and Gacesa, *J. Basic Eng.,* **93,** 11–21 (1971).

A variety of **short-tube** venturi meters are available commercially. They require less space for installation and are generally (although not always) characterized by a greater pressure loss than the corresponding Herschel-type venturi meter. Discharge coefficients vary widely for different types, and individual calibration is recommended if the manufacturer's calibration is not available. Results of tests on the Dall flow tube are given by Miner [*Trans. Am. Soc. Mech. Eng.,* **78,** 475–479 (1956)] and Dowdell [*Instrum. Control Syst.,* **33,** 1006–1009 (1960)]; and on the Gentile flow tube (also called Beth flow tube or Foster flow tube) by Hooper [*Trans. Am. Soc. Mech. Eng.,* **72,** 1099–1110 (1950)].

The use of a **multiventuri system** (in which an inner venturi discharges into the throat of an outer venturi) to increase both the differential pressure for a given flow rate and the signal-to-loss ratio is described by Klomp and Sovran [*J. Basic Eng.,* **94,** 39–45 (1972)].

Flow Nozzles A simple form of flow nozzle is shown in Fig. 10-19. It consists essentially of a short cylinder with a flared approach section. The approach cross section is preferably elliptical in shape but may be conical. Recommended contours for long-radius flow nozzles are given in *ASME PTC,* op. cit., p. 13. In general, the length of the straight portion of the throat is about one-half throat diameter, the upstream pressure tap is located about one pipe diameter from the nozzle inlet face, and the downstream pressure tap about one-half pipe diameter from the inlet face. For subsonic flow, the pressures at points 2 and 3 will be practically identical. If a conical inlet is preferred, the inlet and throat geometry specified for a Herschel-type venturi meter can be used, omitting the expansion section.

Rate of discharge through a flow nozzle for subcritical flow can be determined by the equations given for venturi meters, Eq. (10-22) for gases and Eq. (10-27) for liquids. The expansion factor *Y* for nozzles is the same as that for venturi meters [Eq. (10-26), Fig. 10-18]. The value of the discharge coefficient *C* depends primarily upon the pipe Reynolds number and to a lesser extent upon the diameter ratio β. Curves of recommended coefficients for long-radius flow nozzles with pressure taps located one pipe diameter upstream and one-half pipe diameter downstream of the inlet face of the nozzle are given in *ASME PTC,* op. cit., p. 15. In general, coefficients range from 0.95 at a pipe Reynolds number of 10,000 to 0.99 at 1,000,000.

The performance characteristics of pipe-wall-tap nozzles (Fig. 10-19) and throat-tap nozzles are reviewed by Wyler and Benedict [*J. Eng. Power,* **97,** 569–575 (1975)].

Permanent pressure loss across a subsonic flow nozzle is approximated by

$$p_1 - p_4 = \frac{1-\beta^2}{1+\beta^2}(p_1 - p_2)$$

(10-28)

where p_1, p_2, p_4 = static pressures measured at the locations shown in Fig. 10-19; and β=ratio of nozzle throat diameter to pipe diameter, dimensionless. Equation (10-28) is based on a momentum balance assuming constant fluid density (see Lapple et al., *Fluid and Particle Mechanics,* University of Delaware, Newark, 1951, p. 13).

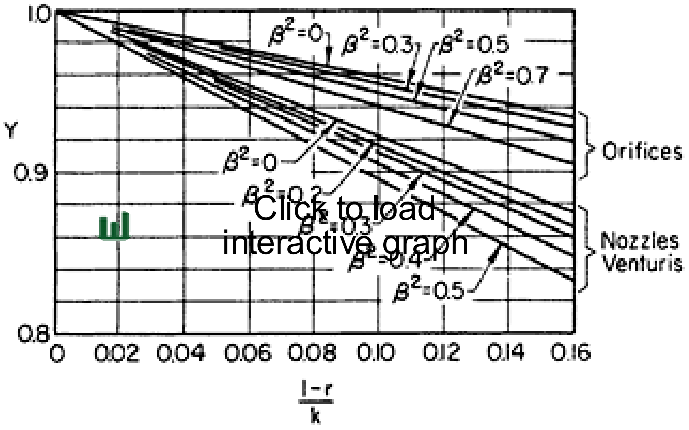

Figure 10-18. Values of expansion factor Y for orifices, nozzles, and venturis.

See Benedict, loc. cit., for a general equation for pressure loss for nozzles installed in pipes or with plenum inlets. Nozzles show higher loss than venturis. Permanent pressure loss for laminar flow depends on the Reynolds number in addition to β. For details, see Alvi, Sridharan, and Lakshamana Rao, *J. Fluids Eng.,* **100,** 299–307 (1978).

Critical Flow Nozzle For a given set of upstream conditions, the rate of discharge of a gas from a nozzle will increase for a decrease in the absolute pressure ratio p_2/p_1 until the linear velocity in the throat reaches that of sound in the gas at that location. The value of p_2/p_1 for which the acoustic velocity is just attained is called the critical pressure ratio r_c. The actual pressure in the throat will not fall below $p_1 r_c$ even if a much lower pressure exists downstream.

The **critical pressure ratio** r_c can be obtained from the following theoretical equation, which assumes a perfect gas and a frictionless nozzle:

$$r_c^{(1-k)/k} + \left(\frac{k-1}{2}\right)\beta^4 r_c^{2/k} = \frac{k+1}{2}$$

(10-29)

This reduces, for $\beta \le 0.2$, to

$$r_c = \left(\frac{2}{k+1}\right)^{k/(k-1)}$$

(10-30)

where k = ratio of specific heats c_p/c_v and β= diameter ratio. A table of values of r_c as a function of k and β is given in the ASME Research Committee on Fluid Meters Report, op. cit., p. 68. For small values of β, r_c = 0.487 for k = 1.667, 0.528 for k = 1.40, 0.546 for k = 1.30, and 0.574 for k = 1.15.

Under **critical flow conditions,** only the upstream conditions p_1, v_1, and T_1 need be known to determine flow rate, which, for $\beta \le 0.2$, is given by

$$w_{max} = CA_2\sqrt{gk\left(\frac{p_1}{v_1}\right)\left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}}$$

(10-31)

For a **perfect gas,** this corresponds to

$$w_{max} = CA_2p_1\sqrt{g_ck\left(\frac{M}{RT_1}\right)\left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}}$$

(10-32)

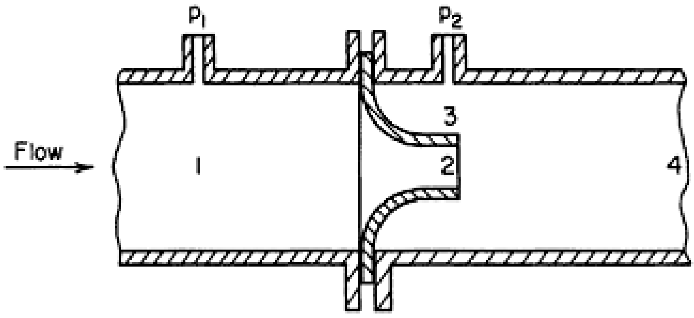

Figure 10-19. Flow-nozzle assembly.

For air, Eq. (10-31) reduces to

$$w_{max} = C_1CA_2p_1/\sqrt{T_1}$$

(10-33)

where A_2 = cross-sectional area of throat; C = coefficient of discharge, dimensionless; g_c = dimensional constant; k = ratio of specific heats, c_p/c_v; M = molecular weight; p_1 = pressure on upstream side of nozzle; R = gas constant; T_1 = absolute temperature on upstream side of nozzle; v_1 = specific volume on upstream side of nozzle; C_1 = dimensional constant, 0.0405 SI units (0.533 U.S. customary units); and w_{max} = maximum-weight flow rate.

Discharge coefficients for critical flow nozzles are, in general, the same as those for subsonic nozzles. See Grace and Lapple, *Trans. Am. Soc. Mech. Eng.,* **73,** 639–647 (1951); and Szaniszlo, *J. Eng. Power,* **97,** 521–526 (1975). Arnberg, Britton, and Seidl [*J. Fluids Eng.,* **96,** 111–123 (1974)] present discharge-coefficient correlations for circular-arc venturi meters at critical flow. For the calculation of the flow of natural gas through nozzles under critical-flow conditions, see Johnson, *J. Basic Eng.,* **92,** 580–589 (1970).

Elbow Meters A pipe elbow can be used as a flowmeter for liquids if the differential centrifugal head generated between the inner and outer radii of the bend is measured by means of pressure taps located midway around the bend. Equation (10-27) can be used, except that the pressure-difference term ($p_1 - p_2$) is now taken to be the differential centrifugal pressure and β is taken as zero if one assumes no change in cross section between the pipe and the bend. The discharge coefficient should preferably be determined by calibration, but as a guide it can be estimated within ±6 percent for circular pipe for Reynolds numbers greater than 10^5 from

$C = 0.98\sqrt{R_c/2D}$, where R_c = radius of curvature of the centerline and D = inside pipe diameter in consistent units. See Murdock, Foltz, and Gregory, *J. Basic Eng.,* **86,** 498–506 (1964); or the ASME Research Committee on Fluid Meters Report, op. cit., pp. 75–77.

Accuracy Square-edged orifices and venturi tubes have been so extensively studied and standardized that reproducibilities within 1 to 2 percent can be expected between standard meters when new and clean. This is therefore the order of reliability to be had, if one assumes (1) accurate measurement of meter differential, (2) selection of the coefficient of discharge from recommended published literature, (3) accurate knowledge of fluid density, (4) accurate measurement of critical meter dimensions, (5) smooth upstream face of orifice, and (6) proper location of the meter with respect to other flow-disturbing elements in the system. Care must also be taken to avoid even slight corrosion or fouling during use.

Presence of **swirling flow** or an **abnormal velocity distribution** upstream of the metering element can cause serious metering error unless calibration in place is employed or sufficient straight pipe is inserted between the meter and the source of disturbance. Table 10-7 gives the minimum lengths of straight pipe required to avoid appreciable error due to the presence of certain fittings and valves either upstream or downstream of an orifice or nozzle. These values were extracted from plots presented by Sprenkle [*Trans. Am. Soc. Mech. Eng.,* **67,** 345–360 (1945)]. Table 10-7 also shows the reduction in spacing made possible by the use of straightening vanes between the fittings and the meter. Entirely adequate straightening vanes can be provided by fitting a bundle of thin-wall tubes within the pipe. The center-to-center distance between tubes should not exceed one-fourth of the pipe diameter, and the bundle length should be at least 8 times this distance.

The distances specified in Table 10-7 will be conservative if applied to venturi meters. For specific information on requirements for venturi meters, see a discussion by Pardoe appended to Sprenkle (op. cit.). Extensive data on the effect of installation on the coefficients of venturi meters are given elsewhere by Pardoe [*Trans. Am. Soc. Mech. Eng.,* **65,** 337–349 (1943)].

In the presence of **flow pulsations,** the indications of head meters such as orifices, nozzles, and venturis will often be undependable for several reasons. First, the measured pressure differential will tend to be high, since the pressure differential is proportional to the square of flow rate for a head meter, and the square root of the mean differential pressure is always greater than the mean of the square roots of the differential pressures. Second, there is a phase shift as the wave passes through the metering restriction which can affect the differential. Third, pulsations can be set up in the manometer leads themselves. Frequency of the pulsation also plays a part. At low frequencies, the meter reading can generally faithfully follow the flow pulsations, but at high frequencies it cannot. This is due to inertia of the fluid in the manometer leads or of the manometric fluid, whereupon the meter would give a reading intermediate between the maximum and minimum flows but having no readily predictable relation to the mean flow. Pressure transducers with flush-mounted diaphragms can be used together with high-speed recording equipment to provide accurate records of the pressure profiles at the upstream and downstream pressure taps, which can then be analyzed and translated into a mean flow rate.

Table 10-7. Locations of Orifices and Nozzles Relative to Pipe Fittings

Type of fitting upstream	$\frac{D_2}{D_1}$	Distances in pipe diameters, D_1			
		Distance, upstream fitting to orifice		Distance, vanes to orifice	Distance, nearest downstream fitting from orifice
		Without straight-ening vanes	With straight-ening vanes		
Single 90° ell, tee, or cross used as ell	0.2	6			2
	0.4	6			
	0.6	9	9		
	0.8	20	12	8	4
2 short-radius 90° ells in form of S	0.2	7			2
	0.4	8	8		
	0.6	13	10	6	
	0.8	25	15	11	4
2 long- or short-radius 90° ells in perpendicular planes	0.2	15	9	5	2
	0.4	18	10	6	
	0.6	25	11	7	
	0.8	40	13	9	4
Contraction or enlargement	0.2	8	Vanes have no advantage		2
	0.4	9			
	0.6	10			
	0.8	15			4

Globe valve or stop check	0.2	9	9	5	2
	0.4	10	10	6	
	0.6	13	10	6	
	0.8	21	13	9	4
Gate valve, wide open, or plug cocks	0.2	6	Same as globe valve		2
	0.4	6			
	0.6	8			
	0.8	14			4

The rather general practice of producing a steady differential reading by placing restrictions in the manometer leads can result in a reading which, under a fixed set of conditions, may be useful in control of an operation but which has no readily predictable relation to the actual average flow. If calibration is employed to compensate for the presence of pulsations, complete reproduction of operating conditions, including source of pulsations and waveform, is necessary to ensure reasonable accuracy.

According to Head [*Trans. Am. Soc. Mech. Eng.,* **78,** 1471–1479 (1956)], a pulsation-intensity limit of $\Gamma = 0.1$ is recommended as a practical pulsation threshold below which the performance of all types of flowmeters will differ negligibly from steady-flow performance (an error of less than 1 percent in flow due to pulsation). Γ is the peak-to-trough flow variation expressed as a fraction of the average flow rate. According to the ASME Research Committee on Fluid Meters Report (op. cit., pp. 34–35), the fractional metering error E for **liquid flow** through a head meter is given by

$$(1 + E)^2 = 1 + \Gamma^2/8$$

(10-34)

When the pulsation amplitude is such as to result in a greater-than-permissible metering error, consideration should be given to installation of a pulsation damper between the source of pulsations and the flowmeter. References to methods of pulsation-damper design are given in the subsection “Unsteady-State Behavior.”

Pulsations are most likely to be encountered in discharge lines from reciprocating pumps or compressors and in lines supplying steam to reciprocating machinery. For **gas flow,** a combination involving a surge chamber and a constriction in the line can be used to damp out the pulsations to an acceptable level. The surge chamber is generally located as close to the pulsation source as possible, with the constriction between the surge chamber and the metering element. This arrangement can be used for either a suction or a discharge line. For such an arrangement, the metering error has been found to be a function of the Hodgson number N_H, which is defined as

$$N_H = Qn\, \Delta p_s / qp_s$$

(10-35)

where Q = volume of surge chamber and pipe between metering element and pulsation source; n = pulsation frequency; Δp_s = permanent pressure drop between metering element and surge chamber; q = average volume flow rate, based on gas density in the surge chamber; and p_s = pressure in surge chamber.

Herning and Schmid [*Z. Ver. Dtsch. Ing.,* **82,** 1107–1114 (1938)] presented charts for a simplex double-acting compressor for the prediction of metering error as a function of the Hodgson number and *s,* the ratio of piston discharge time to total time per stroke. Table 10-8*a* gives the minimum Hodgson numbers required to reduce the metering error to 1 percent as given by the charts (for specific heat ratios between 1.28 and 1.37). Schmid [*Z. Ver. Dtsch. Ing.,* **84,** 596–598 (1940)] presented similar charts for a duplex double-acting compressor and a triplex double-acting compressor for a specific heat ratio of 1.37. Table 10-8*b* gives the minimum Hodgson numbers corresponding to a 1 percent metering error for these cases. The value of $Q\, \Delta p_s$ can be calculated from the appropriate Hodgson number, and appropriate values of Q and Δp_s selected so as to satisfy this minimum requirement.

10.1.13. VELOCITY METERS

Anemometers An anemometer may be any instrument for measurement of gas velocity, e.g., a pitot tube, but usually the term refers to one of the following types.

The vane **anemometer** is a delicate revolution counter with jeweled bearings, actuated by a small windmill, usually 75 to 100 mm (about 3 to 4 in) in diameter, constructed of flat or slightly curved radially disposed vanes. Gas velocity is determined by using a stopwatch to find the time interval required to pass a given number of meters (feet) of gas as indicated by the counter. The velocity so obtained is inversely proportional to gas density. If the original calibration was carried out in a gas of density ρ_0 and the density of the gas stream being metered is ρ_1, the true gas velocity can be found as follows: From the calibration curve for the instrument, find $V_{t,0}$ corresponding to the quantity $V_m\sqrt{\rho_1/\rho_0}$, where V_m = measured velocity. Then the actual velocity $V_{t,1}$ is equal to $V_{t,0}\sqrt{\rho_0/\rho_1}$. In general, when working with air, the effects of atmospheric-density changes can be neglected for all velocities above 1.5 m/s (about 5 ft/s). In all cases, care must be taken to hold the anemometer well away from one's body or from any object not normally present in the stream.

Table 10-8a. Minimum Hodgson Numbers

Simplex double-acting compressor

s	N_H	s	N_H
0.167	1.31	0.667	0.60
0.333	1.00	0.833	0.43
0.50	0.80	1.00	0.34

Table 10-8b. Minimum Hodgson Numbers

Duplex double-acting compressor		**Triplex double-acting compressor**	
s	N_H	s	N_H
0.167	1.00	0.167	0.85
0.333	0.70	0.333	0.30
0.50	0.30	0.50	0.15
0.667	0.10	0.667	0.06
0.833	0.05	0.833	0.00
1.00	0.00	1.00	0.00

Vane anemometers can be used for gas-velocity measurements in the range of 0.3 to 45 m/s (about 1 to 150 ft/s), although a given instrument generally has about a twentyfold velocity range. Bearing friction has to be minimized in instruments designed for accuracy at the low end of the range, while ample rotor and vane rigidity must be provided for measurements at the higher velocities. Vane anemometers are sensitive to shock and cannot be used in corrosive atmospheres. Therefore, accuracy is questionable unless a recent calibration has been made and the history of the instrument subsequent to calibration is known. For additional information, see Ower et al., op. cit., chap. VIII.

Turbine Flowmeters They consist of a straight flow tube containing a turbine which is free to rotate on a shaft supported by one or more bearings and located on the centerline of the tube. Means are provided for magnetic detection of the rotational speed, which is proportional to the volumetric flow rate. Its use is generally restricted to clean, noncorrosive fluids. Additional information on construction, operation, range, and accuracy can be obtained from Baker, pp. 215–252, 2000; Miller, op. cit.; and Spitzer, pp. 303–317, 2005.

The **current meter** is generally used for measuring velocities in open channels such as rivers and irrigation channels. There are two types, the cup meter and the propeller meter. The former is more widely used. It consists of six conical cups mounted on a vertical axis pivoted at the ends and free to rotate between the rigid arms of a U-shaped clevis to which a vaned tailpiece is attached. The wheel rotates because of the difference in drag for the two sides of the cup, and a signal proportional to the revolutions of the wheel is generated. The velocity is determined from the count over a period of time. The current meter is generally useful in the range of 0.15 to 4.5 m/s (about 0.5 to 15 ft/s) with an accuracy of ±2 percent. For additional information see Creager and Justin, *Hydroelectric Handbook,* 2d ed., Wiley, New York, 1950, pp. 42–46.

Other important classes of velocity meters include electromagnetic flowmeters and ultrasonic flowmeters. Both are described in Sec. 8.

10.1.14. MASS FLOWMETERS

General Principles There are two main types of mass flowmeters: (1) the so-called true mass flowmeter, which responds directly to mass flow rate, and (2) the inferential mass flowmeter, which commonly measures volume flow rate and fluid density separately. A variety of types of true mass flowmeters have been developed, including the following: (*a*) the Magnus-effect mass flowmeter, (*b*) the axial-flow, transverse-momentum mass flowmeter, (*c*) the radial-flow, transverse-momentum mass flowmeter, (*d*) the gyroscopic transverse-momentum mass flowmeter, and (*e*) the thermal mass flowmeter. Type *b* is the basis for several commercial mass flowmeters, one version of which is briefly described here.

Axial-Flow Transverse-Momentum Mass Flowmeter This type is also referred to as an angular-momentum mass flowmeter. One embodiment of its principle involves the use of axial flow through a driven impeller and a turbine in series. The impeller imparts angular momentum to the fluid, which in turn causes a torque to be imparted to the turbine, which is restrained from rotating by a spring. The torque, which can be measured, is proportional to the rotational speed of the impeller and the mass flow rate.

Inferential Mass Flowmeter There are several types in this category, including the following:

1. *Head meters with density compensation.* Head meters such as orifices, venturis, or nozzles can be used with one of a variety of densitometers [e.g., based on (*a*) buoyant force on a float, (*b*) hydraulic coupling, (*c*) voltage output from a piezoelectric crystal, or (*d*) radiation absorption]. The signal from the head meter, which is proportional to ρV^2 (where ρ=fluid density and V = fluid velocity), is multiplied by ρ given by the densitometer. The square root of the product is proportional to the mass flow rate.

2. *Head meters with velocity compensation.* The signal from the head meter, which is proportional to ρV^2, is divided by the signal from a velocity meter to give a signal proportional to the mass flow rate.

3. *Velocity meters with density compensation.* The signal from the velocity meter (e.g., turbine meter, electromagnetic meter, or sonic velocity meter) is multiplied by the signal from a densitometer to give a signal proportional to the mass flow rate.

Coriolis Mass Flowmeter This type, described in Sec. 8, offers simultaneous direct measurement of both mass flow rate and fluid density. The coriolis flowmeter is insensitive to upstream and downstream flow disturbances, but its performance is adversely affected by the presence of even a few percent of a gas when measuring a liquid flow.

10.1.15. VARIABLE-AREA METERS

General Principles The underlying principle of an ideal area meter is the same as that of a head meter of the orifice type (see subsection "Orifice Meters"). The stream to be measured is throttled by a constriction, but instead of observing the variation with flow of the differential head across an orifice of fixed size, the constriction of an area meter is so arranged that its size is varied to accommodate the flow while the differential head is held constant.

A simple example of an area meter is a gate valve of the rising-stem type provided with static-pressure taps before and after the gate and a means for measuring the stem position. In most common types of area meters, the variation of the opening is automatically brought about by the motion of a weighted piston or float supported by the fluid. Two different cylinder- and piston-type area meters are described in the ASME Research Committee on Fluid Meters Report, op. cit., pp. 82–83.

Rotameters The rotameter, an example of which is shown in Fig. 10-20, has become one of the most popular flowmeters in the chemical-process industries. It consists essentially of a plummet, or "float," which is free to move up or down in a vertical, slightly tapered tube having its small end down. The fluid enters the lower end of the tube and causes the float to rise until the annular area between the float and the wall of the tube is such that the pressure drop across this constriction is just sufficient to support the float. Typically, the tapered tube is of glass and carries etched upon it a nearly linear scale on which the position of the float may be visually noted as an indication of the flow.

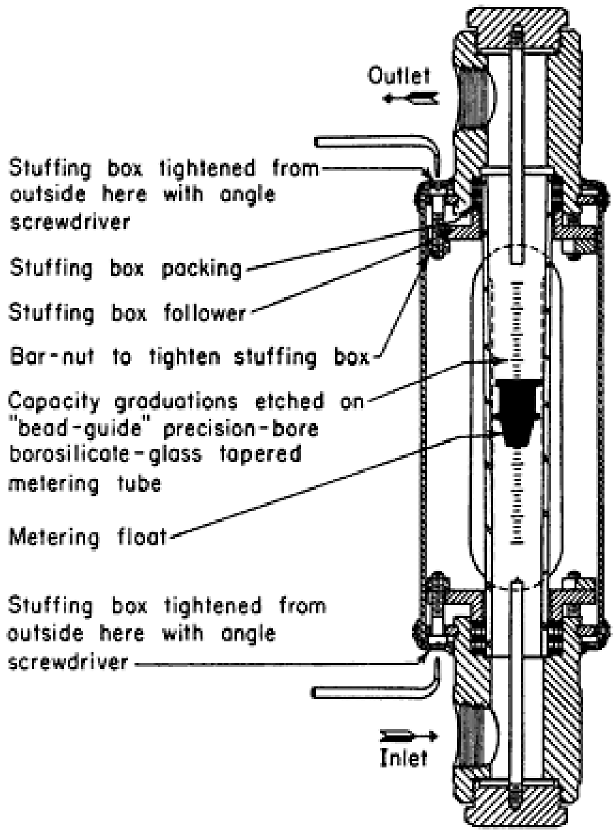

Figure 10-20. Rotameter.

Interchangeable precision-bore glass tubes and metal metering tubes are available. Rotameters have proved satisfactory both for gases and for liquids at high and at low pressures. A single instrument can readily cover a tenfold range of flow, and by providing floats of different densities a two-hundredfold range is practicable. Rotameters are available with pneumatic, electric, and electronic transmitters for actuating remote recorders, integrators, and automatic flow controllers (see Considine, op. cit., pp. 4-35–4-36, and Sec. 8 of this *Handbook*).

Rotameters require no straight runs of pipe before or after the point of installation. Pressure losses are substantially constant over the whole flow range. In experimental work, for greatest precision, a rotameter should be calibrated with the fluid which is to be metered. However, most modern rotameters are precision-made so that their performance closely corresponds to a master calibration plot for the type in question. Such a plot is supplied with the meter upon purchase.

According to Head [*Trans. Am. Soc. Mech. Eng.,* **76,** 851–862 (1954)], flow rate through a rotameter can be obtained from

$$w = q\rho = KD_f\sqrt{\frac{W_f(\rho_f-\rho)\rho}{\rho_f}}$$

(10-36)

and

$$K = \phi\left[\frac{D_t}{D_f}, \frac{\mu}{\sqrt{\frac{W_f(\rho_f-\rho)\rho}{\rho_f}}}\right]$$

(10-37)

where w = weight flow rate; q = volume flow rate; ρ=fluid density; K = flow parameter, $m^{1/2}/s$ ($ft^{1/2}/s$); D_f= float diameter at constriction; W_f= float weight; ρ_f= float density; D_t = tube diameter at point of constriction; and μ = fluid viscosity. The appropriate value of K is obtained from a composite correlation of K versus the parameters shown in Eq. (10-37) corresponding to the float shape being used. The relation of D_t to the rotameter reading is also required for the tube taper and size being used.

The ratio of flow rates for two different fluids A and B at the same rotameter reading is given by

$$\frac{w_A}{w_B} = \frac{K_A}{K_B}\sqrt{\frac{(\rho_f - \rho_A)\rho_A}{(\rho_f - \rho_B)\rho_B}}$$

(10-38)

A measure of self-compensation, with respect to weight rate of flow, for fluid-density changes can be introduced through the use of a float with a density twice that of the fluid being metered, in which case an increase of 10 percent in ρ will produce a decrease of only 0.5 percent in w for the same reading. The extent of immunity to changes in fluid viscosity depends upon the shape of the float.

According to Baird and Cheema [*Can. J. Chem. Eng.,* **47,** 226–232 (1969)], the presence of square-wave pulsations can cause a rotameter to overread by as much as 100 percent. The higher the pulsation frequency, the less the float oscillation, although the error can still be appreciable even when the frequency is high enough so that the float is virtually stationary. Use of a damping chamber between the pulsation source and the rotameter will reduce the error.

Additional information on rotameter theory is presented by Fischer [*Chem. Eng.,* **59**(6), 180–184 (1952)], Coleman [*Trans. Inst. Chem. Eng.,* **34,** 339–350 (1956)], and McCabe, Smith, and Harriott (*Unit Operations of Chemical Engineering,* 4th ed., McGraw-Hill, New York, 1985, pp. 202–205).

10.1.16. TWO-PHASE SYSTEMS

It is generally preferable to meter each of the individual components of a two-phase mixture separately prior to mixing, since it is difficult to meter such mixtures accurately. Problems arise because of fluctuations in composition with time and variations in composition over the cross section of the channel. Information on metering of such mixtures can be obtained from the following sources.

Gas-Solid Mixtures Carlson, Frazier, and Engdahl [*Trans. Am. Soc. Mech. Eng.,* **70,** 65–79 (1948)] describe the use of a **flow nozzle** and a **square-edged orifice** in series for the measurement of both the gas rate and the solids rate in the flow of a finely divided solid-in-gas mixture. The nozzle differential is sensitive to the flow of both phases, whereas the orifice differential is not influenced by the solids flow.

Farbar [*Trans. Am. Soc. Mech. Eng.,* **75,** 943–951 (1953)] describes how a **venturi meter** can be used to measure solids flow rate in a gas-solids mixture when the gas rate is held constant. Separate calibration curves (solids flow versus differential) are required for each gas rate of interest.

Cheng, Tung, and Soo [*J. Eng. Power,* **92,** 135–149 (1970)] describe the use of an **electrostatic probe** for measurement of solids flow in a gas-solids mixture.

Goldberg and Boothroyd [*Br. Chem. Eng.,* **14,** 1705–1708 (1969)] describe several types of solids-in-gas flowmeters and give an extensive bibliography.

Gas-Liquid Mixtures An empirical equation was developed by Murdock [*J. Basic Eng.,* **84,** 419–433 (1962)] for the measurement of gas-liquid mixtures using **sharp-edged orifice** plates with either radius, flange, or pipe taps.

An equation for use with **venturi meters** was given by Chisholm [*Br. Chem. Eng.,* **12,** 454–457 (1967)]. A procedure for determining steam quality via pressure-drop measurement with upflow through either venturi meters or sharp-edged orifice plates was given by Collins and Gacesa [*J. Basic Eng.,* **93,** 11–21 (1971)].

Liquid-Solid Mixtures Liptak [*Chem. Eng.,* **74**(4), 151–158 (1967)] discusses a variety of techniques that can be used for the measurement of solids-in-liquid suspensions or slurries. These include metering pumps, weigh tanks, magnetic flowmeter, ultrasonic flowmeter, gyroscope flowmeter, etc.

Shirato, Gotoh, Osasa, and Usami [*J. Chem. Eng. Japan,* **1,** 164–167 (January 1968)] present a method for determining the mass flow rate of suspended solids in a liquid stream wherein the liquid velocity is measured by an electromagnetic flowmeter and the flow of solids is calculated from the pressure drops across each of two vertical sections of pipe of different diameter through which the suspension flows in series.

10.1.17. FLOWMETER SELECTION

Web sites for process equipment and instrumentation, such as www.globalspec.com and www.thomasnet.com, are valuable tools when selecting a flowmeter. These search engines can scan the flowmeters manufactured by more than 800 companies for specific products that meet the user's specifications. Table 10-4 was based in part on information from these web sites. Note that the accuracies claimed are achieved only under ideal conditions when the flowmeters are properly installed and calibrated for the application.

The purpose of this subsection is to summarize the preferred applications as well as the advantages and disadvantages of some of the common flowmeter technologies.

Table 10-9 divides flowmeters into four classes. Flowmeters in class I depend on wetted moving parts that can wear, plug, or break. The potential for catastrophic failure is a disadvantage. However, in clean fluids, class I flowmeters have often proved reliable and stable when properly installed, calibrated, and maintained.

Table 10-9. Flowmeter Classes

Class I: Flowmeters with wetted moving parts	Class II: Flowmeters with no wetted moving parts
Positive displacement	Differential pressure
Turbine	Vortex
Variable-area	Target
	Thermal
Class III: Obstructionless flowmeters	Class IV: Flowmeters with sensors mou nted external to the pipe
Coriolis mass	Clamp-on ultrasonic
Electromagnetic	Correlation
Ultrasonic	
Adapted from Spitzer, op. cit., 2005.	

Class II flowmeters have no wetted moving parts to break and are thus not subject to catastrophic failure. However, the flow surfaces such as orifice plates may wear, eventually biasing flow measurements. Other disadvantages of some flowmeters in this class include high pressure drop and susceptibility to plugging. Very dirty and abrasive fluids should be avoided.

Because class III flowmeters have neither moving parts nor obstructions to flow, they are suitable for dirty and abrasive fluids provided that appropriate materials of construction are available.

Class IV flowmeters have sensors mounted external to the pipe, and would thus seem to be ideal, but problems of accuracy and sensitivity have been encountered in early devices. These comparatively new technologies are under development, and these problems may be overcome in the future.

Section 8 outlines the following criteria for selection of measurement devices: measurement span, performance, reliability, materials of construction, prior use, potential for releasing process materials to the environment, electrical classification, physical access, invasive or noninvasive, and life-cycle cost.

Spitzer, op. cit., 2005, cites four intended end uses of the flowmeter: rate indication, control, totalization, and alarm. Thus high accuracy may be important for rate indication, while control may just need good repeatability. Volumetric flow or mass flow indication is another choice.

Baker, op. cit., 2003, identifies the type of fluid (liquid or gas, slurry, multiphase), special fluid constraints (clean or dirty, hygienic, corrosive, abrasive, high flammability, low lubricity, fluids causing scaling). He lists the following flowmeter constraints: accuracy or measurement uncertainty, diameter range, temperature range, pressure range, viscosity range, flow range, pressure loss caused by the flowmeter, sensitivity to installation, sensitivity to pipework supports, sensitivity to pulsation, whether the flowmeter has a clear bore, whether a clamp-on version is available, response time, and ambient conditions. Finally, Baker identifies these environmental considerations: ambient temperature, humidity, exposure to weather, level of electromagnetic radiation, vibration, tamperproof for domestic use, and classification of area requiring explosionproof, intrinsic safety, etc.

Note that the accuracies cited in Table 10-4 can be achieved by those flowmeters only under ideal conditions of application, installation, and calibration. This subsection has only given an introduction to issues to consider in the choice of a flowmeter for a given application. See Baker, op. cit., 2003; Miller, op. cit., 1996; and Spitzer, op. cit., 2005, for further guidance and to obtain application-specific data from flowmeter vendors.

10.1.18. WEIRS

Liquid flow in an open channel may be metered by means of a weir, which consists of a dam over which, or through a notch in which, the liquid flows. The terms "rectangular weir," "triangular weir," etc., generally refer to the shape of the notch in a notched weir. All weirs considered here have flat upstream faces that are perpendicular to the bed and walls of the channel.

Sharp-edged weirs have edges like those of square or sharp-edged orifices (see subsection "Orifice Meters"). Notched weirs are ordinarily sharp-edged. Weirs not in the sharp-edged class are, for the most part, those described as **broad-crested weirs.**

The head h_0 on a weir is the liquid-level height above the crest or base of the notch. The head must be measured sufficiently far upstream to avoid the drop in level occasioned by the overfall which begins at a distance about $2h_0$ upstream from the weir. Surface-level measurements should be made a distance of $3h_0$ or more upstream, preferably by using a stilling box equipped with a high-precision level gauge, e.g., a hook gauge or float gauge.

With sharp-edged weirs, the sheet of discharging liquid, called the "nappe," contracts as it leaves the opening and free discharge occurs. Rounding the upstream edge will reduce the contraction and increase the flow rate for a given head. A clinging nappe may result if the head is very small, if the edge is well rounded, or if air cannot flow in beneath the nappe. This, in turn, results in an increase in the discharge rate for a given head as compared with that for a free nappe. For further information on the effect of the nappe, see Gibson, *Hydraulics and Its Applications,* 5th ed., Constable, London, 1952; and Chow, *Open-Channel Hydraulics,* McGraw-Hill, New York, 1959.

Figure 10-21. Rectangular weir.

Flow through a **rectangular weir** (Fig. 10-21) is given by

$$q = 0.415(L - 0.2h_0)h_0^{1.5}\sqrt{2g}$$
(10-39)

where q = volume flow rate, L = crest length, h_0 = weir head, and g = local acceleration due to gravity. This is known as the modified Francis formula for a rectangular sharp-edged weir with two end corrections; it applies when the velocity-of-approach correction is small. The Francis formula agrees with experiments within 3 percent if (1) L is greater than $2h_0$, (2) velocity of approach is 0.6 m/s (2 ft/s) or less, (3) height of crest above bottom of channel is at least $3h_0$, and (4) h_0 is not less than 0.09 m (0.3 ft).

Narrow rectangular notches ($h_0 > L$) have been found to give about 93 percent of the discharge predicted by the Francis formula. Thus

$$q = 0.386Lh_0^{1.5}\sqrt{2g}$$
(10-40)

In this case, no end corrections are applied even though the formula applies only for sharp-edged weirs. See Schoder and Dawson, *Hydraulics,* McGraw-Hill, New York, 1934, p. 175, for further details.

The **triangular-notch weir** has the advantage that a single notch can accommodate a wide range of flow rates, although this in turn reduces its accuracy. The discharge for sharp- or square-edged weirs is given by

$$q = (0.31h_0^{2.5}\sqrt{2g})/\tan\phi$$
(10-41)

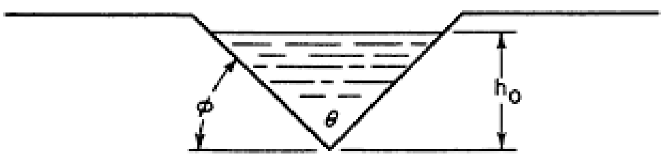

Figure 10-22. Triangular weir.

See Eq. (10-39) for nomenclature. Angle ϕ is illustrated in Fig. 10-22. Equations (10-39), (10-40), and (10-41) are applicable only to the flow of water. However, for the case of triangular-notch weirs Lenz [*Trans. Am. Soc. Civ. Eng.,* **108,** 759–802 (1943)] has presented correlations predicting the effect of viscosity over the range of 0.001 to 0.15 Pa·s (1 to 150 cP) and surface tension over the range of 0.03 to 0.07 N/m (30 to 70 dyn/cm). His equation predicts about an 8 percent increase in flow for a liquid of 0.1-Pa·s (100-cP) viscosity compared with water at 0.001 Pa· s (1 cP) and about a 1 percent increase for a liquid with one-half of the surface tension of water. For fluids of moderate viscosity, Ranga Raju and Asawa [*Proc. Am. Soc. Civ. Eng., J. Hydraul. Div.,* **103** (HY 10), 1227–1231 (1977)] find that the effect of viscosity and surface tension on the discharge flow rate for rectangular and triangular-notch ($\phi = 45°$) weirs can be neglected when

$$(N_{Re})^{0.2}(N_{We})^{0.6} > 900$$
(10-42)

where N_{Re} (Reynolds number) = $\sqrt{gh_0^3}/v$, g = local acceleration due to gravity, h_0 = weir head, ν = kinematic viscosity; N_{We} (Weber number) $= \rho g h_0^2/g_c\sigma$, ρ = density, g_c = dimensional constant, and σ = surface tension.

For the flow of high-viscosity liquids over rectangular weirs, see Slocum, *Can. J. Chem. Eng.,* **42,** 196–200 (1964). His correlation is based on data for liquids with viscosities in the range of 2.5 to 500 Pa·s (25 to 5000 cP), in which range the discharge decreases markedly for a given head as viscosity is increased.

Information on other types of weirs can be obtained from Addison, op. cit.; Gibson, *Hydraulics and Its Applications,* 5th ed., Constable, London, 1952; Henderson, *Open Channel Flow,* Macmillan, New York, 1966; Linford, *Flow Measurement and Meters,* Spon, London, 1949; Lakshmana Rao, "Theory of Weirs," in *Advances in Hydroscience,* vol. 10, Academic, New York, 1975; and Merritt, *Standard Handbook for Civil Engineers,* 2d ed., McGraw-Hill, New York, 1976.

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[1]The line leading from the pressure tap to the gauge is assumed to be filled with fluid of the same density as that in the apparatus at the location of the pressure tap; if this is not the case, ρ_A is the density of the fluid actually filling the gauge line, and the value given for h_A must be multiplied by ρ_A/ρ where ρ is the density of the fluid whose head is being measured.

[2]The line leading from the pressure tap to the gauge is assumed to be filled with fluid of the same density as that in the apparatus at the location of the pressure tap; if this is not the case, ρA is the density of the fluid actually filling the gauge...

[3]The line leading from the pressure tap to the gauge is assumed to be filled with fluid of the same density as that in the apparatus at the location of the pressure tap; if this is not the case, ρA is the density of the fluid actually filling the gauge...