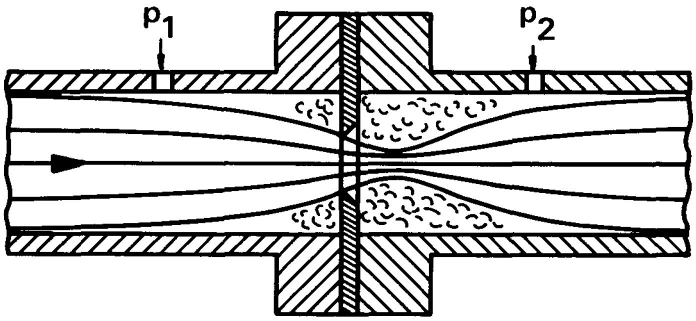
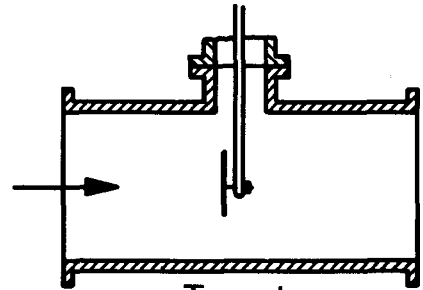
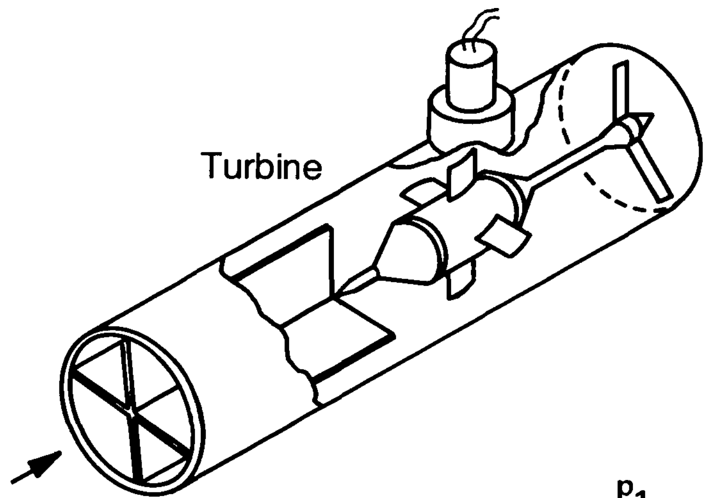
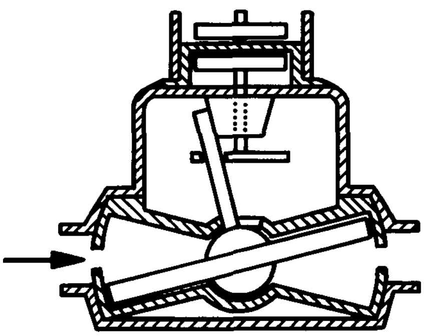


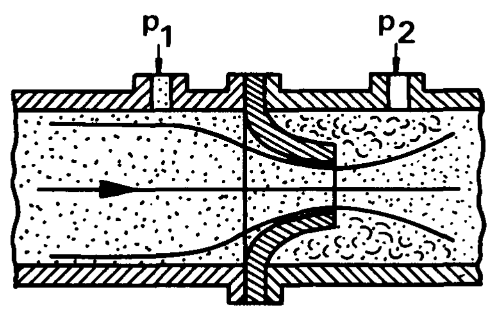
Orifice Plate


Target

Turbine




Nutating Disc



Nozzle

NBS BUILDING SCIENCE SERIES 159

On-Site Calibration of Flow Metering Systems Installed in Buildings

U.S. DEPARTMENT OF COMMERCE • NATIONAL BUREAU OF STANDARDS

NATIONAL BUREAU OF STANDARDS

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Issued January 1984

Library of Congress Catalog Card Number: 83-600626

National Bureau of Standards Building Science Series 159
Natl. Bur. Stand. (U.S.), Bldg. Sci. Ser. 159, 154 pages (Jan. 1984)
CODEN: BSSNBV

U.S. GOVERNMENT PRINTING OFFICE
WASHINGTON: 1984

For sale by the Superintendent of Documents, U.S. Government Printing Office, Washington, DC 20402

ABSTRACT

The measurement of flow of the various fluids (air, water, steam) used in building service systems is usually the most difficult parameter to obtain and maintain. Consequently, in energy management and control systems (EMCS), the flowrate or the total quantity of flow is often the least accurate measurement. However, in most systems the energy consumed depends directly on this parameter.

Since the majority of fluid flow measuring techniques require the sensing element to be located in the stream of the fluid being monitored, flow measuring devices often are the most difficult instruments to calibrate initially and to maintain in calibration within the required accuracy. This report summarizes the various types of flowmetering devices used in EMCS, various methods for their initial calibration and, when practical, techniques for maintaining their calibration while they are in service. Emphasis is placed on the use of transfer reference meter systems, where the working meter is calibrated on site by connecting it in series with a calibrated transfer meter of any variety. Other methods of calibration are also described.

Reference tables and the necessary equations for flow calculations are presented throughout the text and in the appendicies. Illustrative examples are given in detail for the calculation of flow using each type of metering device described. These examples are extremely helpful in field calibration when the metering being calibrated is of a different type than the meter being used as a reference. Because of this, the reader is encouraged to review these examples.

Key words: calibration methods; flowmetering devices; flow nozzle meters; multiple pitot-static tube assemblies; orifice meters; positive displacement meters; reverse-pitot tube assemblies; target meters; turbine meters; ultrasonic flowmeters; venturi meters; vortex shedding meters.

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NOMENCLATURE

		Units
a	Acceleration	ft/s^2
C	Coefficient of discharge for an orifice, flow nozzle or venturi meter	none
CF	Meter calibration factor	*
d	Diameter of an orifice, flow nozzle throat, or venturi throat	in.
D	Pipe inside diameter	in.
F	Force	lb_f
F_a	Area factor for thermal expansion of an orifice, nozzle or venturi	none
F_g	Correction factor for local acceleration due to gravity	none
F_{m1}	Combined conversion factor for mercury to convert inches of mercury at a known temperature to psi	psi/in. Hg
F_{m2}	Correction factor for density of air in the high pressure leg of a mercury manometer	none
F_{m3}	Correction factor for density of water in the high pressure leg of a mercury manometer	none
F_{w1}	Combined conversion factor for water to convert inches of water at a known temperature to psi	psi/in. H_2O
F_{w2}	Correction factor for density of air in the high pressure leg of a water manometer	none
f	Frequency	Hz
g_c	Dimensional and proportional constant relating force, mass and acceleration (= 32.1740)	(lb•ft)/(lb
g_L	Local acceleration due to gravity	ft/s^2

* As designated

NOMENCLATURE (Continued)

		Units
h	Differential pressure at manometer temperature	in. H_2O, or in. Hg
h_w	Differential pressure at reference conditions, 68 °F and standard gravity	in. H_2O
I	Transmitter current	ma DC
K	Meter calibration factor	*
m	Mass	lb
M	Mass rate of flow	lb/hr
p	Pressure relative to existing atmospheric pressure	psig
P	Absolute pressure	psia
P_b	Barometric pressure	psia in. Hg at 32 °F
P_g	Saturation pressure	psia
ΔP	Differential pressure between points in a flow system; flowmeter differential pressure	*
Q	Volumetric rate of flow	ft^3/min
r	Pressure ratio P_2/P_1 where $P_1 - P_2 = \Delta P$	none
R	Gas constant	(psia)ft^3/lb °R
R_D	Pipe Reynolds number $DV\rho/\mu$	none
t	Temperature	°F
T	Absolute temperature	°R
v	Specific volume	ft^3/lb
V	Velocity	ft/s
Y	Expansion factor for air or steam	none

* As designated

NOMENCLATURE (Continued)

		Units
Z	Gas compressibility factor	none
β	Ratio d/D	none
γ	Specific heat ratio c_p/c_v	none
ρ	Density	lb/ft^3 lb/gallon
μ	Dynamic viscosity	lb/(ft•s)
ν	Kinematic viscosity	ft^2/s
ϕ	Denotes "function of"	none

Note: This list comprises principal symbols; additional symbols are defined as needed

SI CONVERSIONS

In view of the presently accepted practice of the building industry in the United States and the status of engineering references and tables used by Engineering Management Control Systems (EMCS) personnel, common U.S. units of measurements have been used in this report. In recognition of the United States as a signatory to the General Conference of Weights and Measures, which gave official status to the SI system of units in 1960, appropriate conversion factors have been provided in the table below. The reader interested in making further use of the coherent system of SI units is referred to: NBS SP330, 1972 Edition, "The International System of Units;" E830-72, ASTM Metric Practice Guide (American National Standards Z210.1); or 1976 Edition of ASHRAE SI Metric Guide for Heating, Refrigerating, Ventilating and Air-Conditioning. Additional SI conversions are given in appendices B and C of this report.

Metric Conversion Factors

To convert from	To	Multiply by*
Acceleration		
ft/s^2	metre per second^2 (m/s^2)	3.048000E-01
Area		
ft^2	metre^2 (m^2)	9.290304E-02
$in.^2$	metre^2 (m^2)	6.451600E-04
Energy		
British thermal unit	joule (J)	1.055056E+03
Force		
pound-force (lbf)	newton (N)	4.448222E+00
Length		
ft	metre (m)	3.048000E-01
in.	metre (m)	2.540000E-02
Mass		
lb	kilogram (kg)	4.535924E-01

*The notation "xE±y," where x and y are numbers, is a standard form for indicating multiplication of the number x by the number 10 raised to the power ± y.

Metric Conversion Factors (cont.)

To convert from	To	Multiply by*
Mass per unit time		
lb/h	kilogram per second (kg/s)	1.259979E-04
lb/min	kilogram per second (kg/s)	7.559873E-03
lb/s	kilogram per second (kg/s)	4.535924E-01
Mass per unit volume		
lb/ft^3	kilogram per $metre^3$ (kg/m^3)	1.601846E+01
$lb/in.^3$	kilogram per $metre^3$ (kg/m^3)	2.767990E+04
lb/gal (U.S. liquid)	kilogram per $metre^3$ (kg/m^3)	1.198264E+02
Pressure or stress		
atmosphere (standard)	pascal (Pa)	1.013250E+05
in. of mercury (32°F)	pascal (Pa)	3.38638E+03
in. of mercury (60°F)	pascal (Pa)	3.37685E+03
in. of water (39.2°F)	pascal (Pa)	2.49082E+02
in. of water (60°F)	pascal (Pa)	2.4884E+02
lbf/ft^2	pascal (Pa)	4.788026E+01
lbf/in^2 (psi)	pascal (Pa)	6.894757E+03
Temperature		
degree Fahrenheit	degree Celsius (°C)	subtract 32 and divide by 1.8
degree Celsius	degree Fahrenheit (°F)	multiply by 1.8 and add 32
degree Fahrenheit	kelvin (K)	add 459.67 and divide by 1.8
degree Celsius	kelvin (K)	add 273.15
kelvin	degree Celsius (°C)	subtract 273.15
degree Rankine	kelvin (K)	divide by 1.8
Velocity (includes speed)		
ft/h	metre per second (m/s)	8.466667E-05
ft/min	metre per second (m/s)	5.080000E-03
ft/sec	metre per second (m/s)	3.048000E-01
in./s	metre per second (m/s)	2.540000E-02

*(see preceding page)

Metric Conversion Factors (cont.)

To convert from	To	Multiply by*
Viscosity (dynamic)		
lb/(ft·s)	pascal-second (Pa·s)	1.488164E+00
Viscosity (kinematic)		
ft^2/s	metre^2 per second (m^2/s)	9.290304E-02
Volume		
ft^3	metre^3 (m^3)	2.831685E-02
gallon (U.S. liquid)	metre^3 (m^3)	3.785412E-03
in.^{3}	metre^3 (m^3)	1.638706E-05
Volume per unit time		
ft^3/min	metre^3 per second (m^3/s)	4.719474E-04
ft^3/s	metre^3 per second (m^3/s)	2.831685E-02
gal (U.S. liquid)/min	metre^3 per second (m^3/s)	6.309020E-05

*(see preceding page)

DISCLAIMER

Certain commercial equipment and instrumentation are identified by name in this report in order to adequately describe the types, capabilities, and technical features of hardware available for flow metering and the on-site calibration of flow metering instrumentation. In no case does such identification imply recommendation or endorsement by the National Bureau of Standards, nor does it imply that the material or equipment identified is necessarily the best available for the purpose.

1. INTRODUCTION

In monitoring the performance of building service systems, one of the most difficult parameters to measure is the flow of the various fluids (air, water, steam). As a consequence, in Energy Management and Control Systems (EMCS), the flowrate or the total quantity of flow is often the least accurate measurement. The energy consumed, however, normally depends directly on this parameter.

This document is intended as an aid or guide to building services personnel (service managers, engineers, technicians) who have a basic understanding of fluid flow and who may be faced with the task of making flow measurements in building systems. The main purpose of this document is to summarize calibration methods for flowmeter systems installed in buildings and to present under one cover essentially all necessary data and equations needed, with the exception of steam tables, for the calculation of meter performance. Flowmeter types and their characteristics as they relate to calibration are also discussed. Fluids covered are air, water, and dry or super-heated steam. Emphasis is placed on the use of transfer reference meter systems where the working meter is calibrated on site by connecting it in series with a transfer meter, which need not be the same type as the working meter. Design characteristics of various types of flowmeters are listed in chapter 13 of reference 1.

While the transfer meter method is usually considered the most desirable choice, a second method used is the direct calibration method. In this case, either the working meter is removed from the building system and calibrated directly at a primary calibration facility employing a volumetric or weighing-type fluid collection and measurement system, or the working meter is calibrated on site against a volumetric or weighing system.

A third method, employed with differential pressure (ΔP) meter systems of orifice, nozzle or venturi types, is to perform a direct calibration of the ΔP transducer. Then, using published data for discharge coefficients and fluid expansion factors for the meter primary element, the system accuracy can be estimated and the flow characteristics for the complete metering system can be determined.

For best accuracy, all flowmeters need to be calibrated using the working or service fluid under typical operating conditions including flowrates, temperatures, and pressures. Accuracies approaching those of primary calibration facilities (tenths of a percent) may be obtained with an on-site calibration using a volumetric or weighing-type collection and measurement system. However, this approach is usually too costly, requiring excessive time, space and/or labor. Also, this degree of accuracy is not often needed in EMCS. The transfer meter method is considered most feasible since the on-site installation and operational efforts are generally less demanding and good accuracy is possible. Reliable, accurate flow measurements depend on certain factors or conditions which often receive too little attention. Two of these factors are: (1) improper meter installation causing a distorted, asymmetrical flow pattern at the meter entrance, and (2) lack of sufficient follow-up or continuing calibration effort to assess the system performance for its entire service period. Admittedly, an adequate installation in existing EMCS lines of the transfer reference meter system and/or the working meter is often difficult, impractical, or virtually impossible. In any new construction, facilities for a transfer reference meter should be considered along with the design of the building mechanical systems.

The types of meters to be discussed include differential pressure types (orifice, flow nozzle, venturi, pitot-static tube, and reversed-type pitot); volumetric types (positive displacement, turbine and vortex); target (impact) and the ultrasonic meter (transit time). Each type will be discussed for use on water, air and steam as appropriate. A brief description of each meter is given, followed by the basic hydraulic or performance equation, installation notes, and sample calculations. Emphasis is placed on the orifice, nozzle, and venturi meters since they are used extensively as both transfer and working meters.

In an effort to make this publication most helpful to the intended user, supplementary data and information are included in the appendix sections as follows: Discharge coefficients and fluid expansion factors for orifice plates, flow nozzles, and venturi meters along with corresponding tolerance (uncertainty, accuracy) data for these quantities are given in appendix A. The sources for these data are references 2, 10, and 11. Properties of air, water, and steam along with mass/volume conversion factors needed for calculation of flow quantities are presented in appendix B. Temperature and pressure relations are given in appendix C. In particular, conversion factors for mechanical pressure gauges, pressure transducers, and liquid manometers are discussed. The relations between mass and volume rate of flow for gases are discussed in appendix D. Appendix E is a compilation of several example calculations given in detail. Those readers already familiar with the flow measurement techniques discussed here may prefer to read these examples first, before reviewing the main body of this report.

Many flow measurements involve the quantities force and mass. In the English system, the same word "pound" refers to both and therefore it becomes necessary to distinguish between the two quantities. In this report, "lb" refers to pound mass, and "lb_f" refers to pound force. These two quantities are related through Newton's second law, force = mass x acceleration. The pound mass is defined in terms of the kilogram and the pound force is defined as the force exerted on a pound mass when it is subjected to the standard acceleration of gravity of 32.1740 ft/s^2. When Newton's second law is used in equation form, a conversion factor designated g_c is necessary to maintain both numerical and dimensional equality, i.e., $F = ma/g_c$. In the English system when the unit of force is designated lb_f and the unit of mass is lb, $g_c = 32.1740\ (lb \cdot ft)/(lb_f \cdot s^2)$.

Throughout this report, the reader will find references to scheduled maintenance calibration of the various types of metering methods described. Scheduled on-site calibration is recommended for all types of flow meters and their transmitting systems. The extent and type of on-site calibration varies from one type of meter to another. However, one major factor applicable to all types of flow metering calibration must be called to the attention of the reader in the introduction of this report to avoid repeating it throughout the text and to emphasize its importance when carrying out the actual on-site calibration and when setting up maintenance schedules for routine calibration: the deterioration of primary elements of the meter (the parts which interact with the flowing fluid) by abrasion and chemical and/or electrolytic reactions has been found to be a major problem in the continuous monitoring of fluids. This is especially true in monitoring the flow of liquids. The designer of a flow system is not always aware of the characteristics of the components in a metering device nor is he always aware of possible changes in the characteristics of the fluid which is to be monitored. In general, the deterioration of a metering system by abrasion and chemical and/or electrolytic reactions is a continuing problem that is hidden in the initial design and is often found to be the basis of on-site meter calibration problems. When continous deterioration of any metering system becomes apparent from previous calibration records or from an erratic response of a metering system which cannot be readily diagnosed, one should always examine the primary parts of the meter for evidence of abrasion and chemical and/or electrolytic reaction with other components in the system. The transducer (i.e., a device which converts one form of energy into another) is also subject to deterioration from environmental conditions. In the case of the metering devices utilizing differential pressure (ΔP) measurements, the calibration of the transducers often is more critical than that of the primary elements.

2. ORIFICE, FLOW NOZZLE AND VENTURI METERS

Since differential pressure meters of the orifice, flow nozzle, and venturi types are used extensively as both transfer and working meters, a brief description of them and the hydraulic equation from which the performance is calculated will be given here. These meters have been used for flow measurements in closed conduits dating back to the past century. Their characteristics have been investigated extensively and are well known. Bernoulli's theorem (1738) is the basis for the hydraulic equation.

2.1 PHYSICAL CHARACTERISTICS

Figures 1, 2, 3, and 4 indicate the shapes and pressure tap locations for orifice, flow nozzle, and venturi meters. When the meter is to receive a direct calibration, the choice of meter type is largely a matter of allowable building system pressure loss versus meter cost. Orifice meters are simple in design. They are manufactured from flat plates and therefore are relatively inexpensive, but they produce the largest pressure losses. The design of nozzle and venturi meters is more complex, with manufacture necessary from machined castings or welded sections, but these meters offer lower pressure losses. Table 1 gives pressure loss data for these types as a function of the beta ratio, $\beta = d/D$, where d is the meter orifice or throat diameter and D is the inside diameter of the pipe.

The primary element is known as that part of the meter system which interacts with the flowing fluid. In these meters, this is the orifice plate, flow nozzle, or venturi. The interaction causes fluid acceleration and pressure change. The secondary element is the instrument system which senses and measures this interaction; in these cases, the secondary element is a ΔP transducer system or a differential manometer including the pressure sensing lines.

2.2 HYDRAULIC EQUATION

Two forms of the hydraulic equation from which the flowrate is calculated are:

$$M = 358.93\ (C\ Y\ d^2\ F_a)[\rho\ h_w/(1 - \beta^4)]^{1/2} \qquad (2\text{-}1)$$

$$Q = 5.982\ (C\ Y\ d^2\ F_a)[h_w/\rho(1 - \beta^4)]^{1/2} \qquad (2\text{-}2)$$

where
- 358.93 $(lb^{1/2}/hr)(ft^{3/2}/in.^{5/2})$
- 5.982 $(ft^{3/2}/min)(lb^{1/2}/in.^{5/2})$
- M = mass flowrate, lb/hr
- Q = volume flowrate, ft^3/min
- C = coefficient of discharge, dimensionless
- Y = fluid expansion factor, dimensionless
- d = diameter of orifice, nozzle throat, or venturi throat, in.
- ρ = density of fluid entering primary element, lb/ft^3
- h_w = differential pressure $P_1 - P_2$, inches of water at 68 °F
- β = ratio d/D, dimensionless, where D is the pipe inside diameter
- F_a = area factor for thermal expansion of a primary element, dimensionless

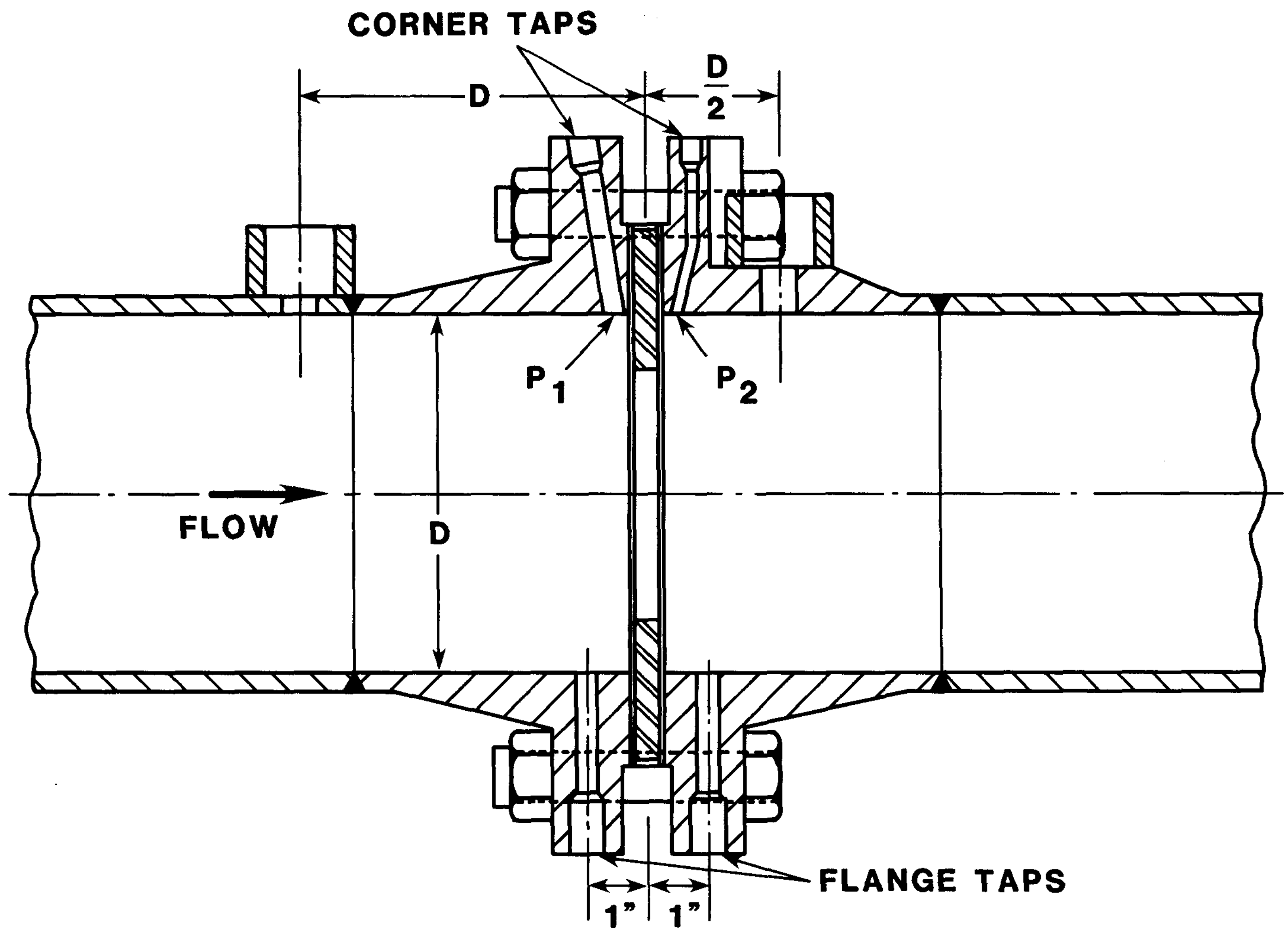


Figure 1. Thin-plate, square-edged orifice mounted between flanges. For illustrative purposes, three pairs of pressure taps are shown: D and D/2 taps, flange taps (1", 1"), and corner taps which sense the pressures in the upstream and downstream "corners" of the orifice plate. The upstream pressure tap is P_1, and downstream, P_2. [2]

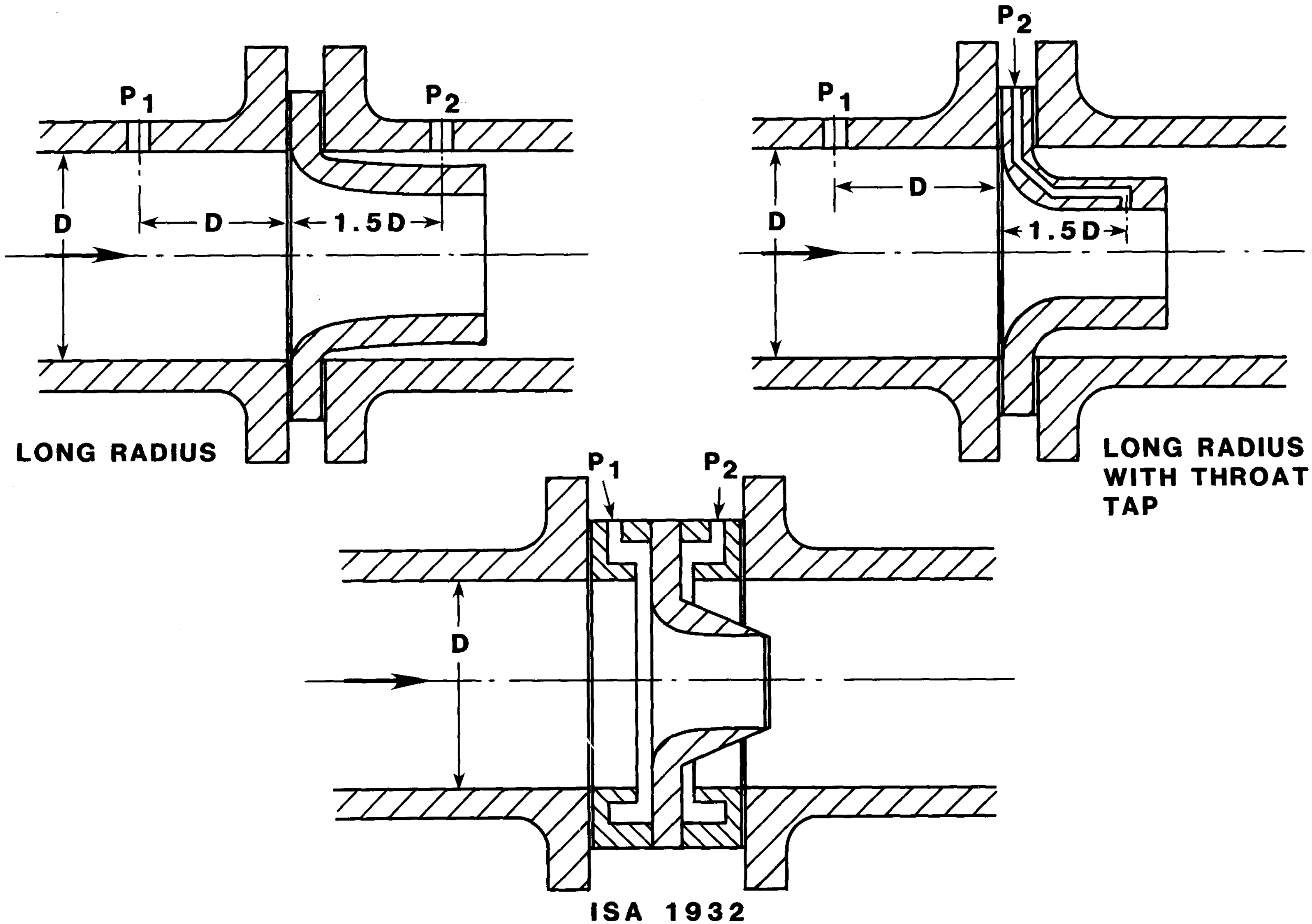


Figure 2. Flow nozzles showing location of pressure taps. [2]

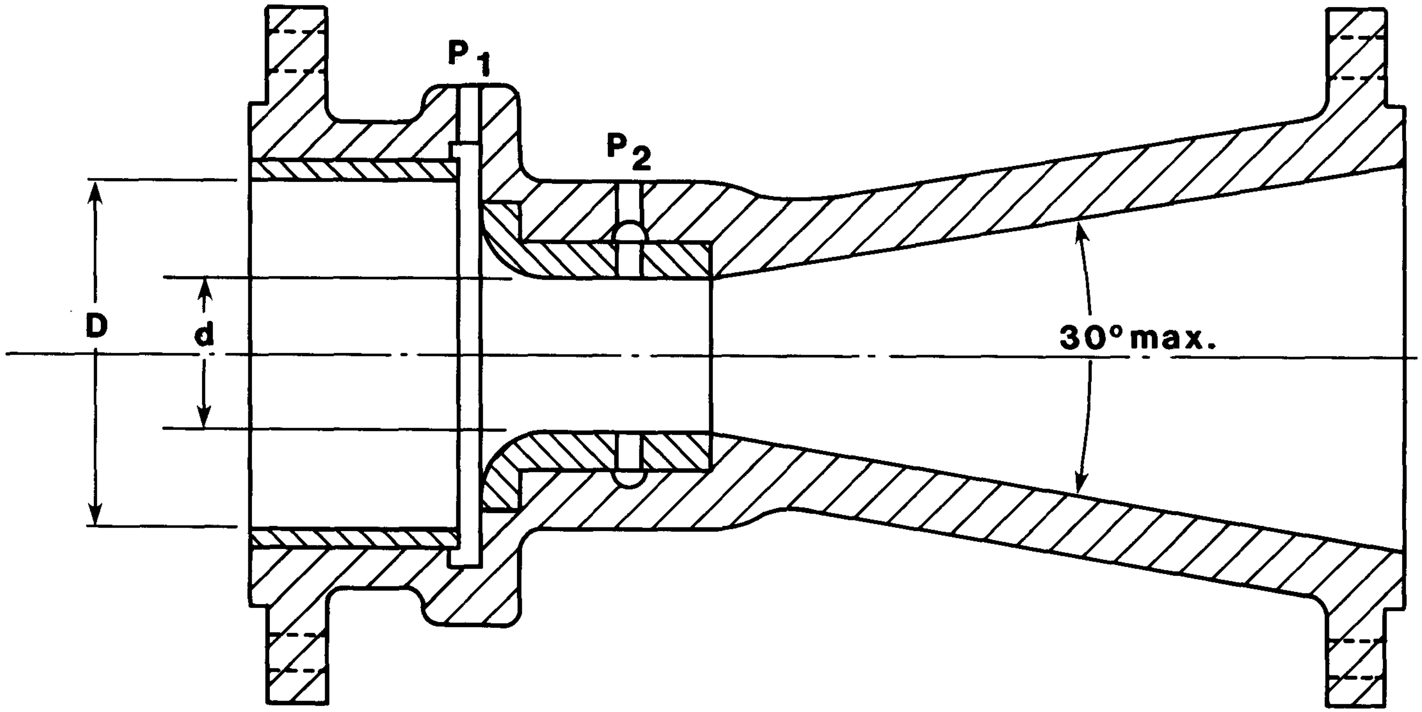


Figure 3. Nozzle venturi meter showing location of pressure taps. [2]

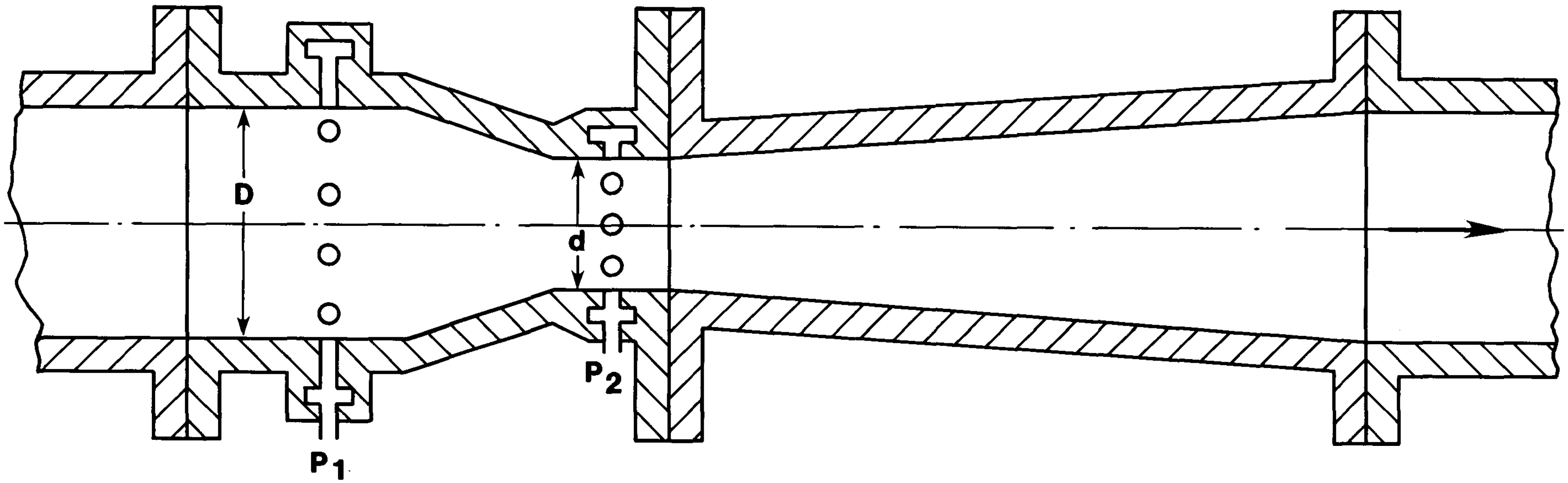


Figure 4. Classical venturi meter showing location of pressure taps. [2]

Table 1. Pressure Loss Through Differential Pressure Primary Elements

	Pressure Loss in Percent of Actual Differential Pressure			
Ratio β	Orifice*	Flow Nozzle*	Nozzle Venturi**	Classical Venturi*
0.3	88	86	26	13
.4	82	75	22	12
.5	73	64	17	11
.6	63	53	15	10
.7	52	41	13	11

Classical venturi meters, 7° outlet cone

* Reference 2

** BARCO (Aeroquip) catalog 866

The theory of flow of fluids in terms of pressure differences and a derivation of these equations may be found in reference 2, pages 47-56. Equations (2-1) and (2-2) apply for subsonic flow.

When calibrating a working meter on site, the flowrate M or Q is measured either by the transfer reference meter system or by a direct calibration system. The quantity h_w is the working meter output. The fluid expansion factor is considered a known quantity; for liquids, Y = 1.000 and for gases, Y is calculated from gas relations. Thus, with d, F_a, ρ, and β known, common practice is to determine the discharge coefficient, C, for the particular working meter.

The coefficient of discharge is defined as

$$C = \frac{\text{Actual rate of flow}}{\text{Theoretical rate of flow}}$$

In turbulent flow, this coefficient must be measured experimentally. It depends on the shape of the primary element (orifice, nozzle, venturi), the pressure tap locations (flange; 1D, 1/2D; corner), the β ratio, and the Reynolds Number, $R_D = D\ V\ \rho/\mu$, where V is the fluid average velocity and μ is the dynamic viscosity. Its dependence on flowrate is through V, and on fluid properties through ρ and μ. It is important to note that once a ΔP meter is calibrated using one fluid, its performance with other fluids can be predicted through this C vs R_D relationship. Data for this relationship are given in appendix A for reference purposes.

The fluid expansion factor, Y, accounts for any change in density as the fluid flows through the primary element. For flow through nozzle and venturi meters, a correction based on ideal flow is used. This correction is dependent on the pressure ratio P_1/P_2, on the β ratio, and on the gas specific heat ratio $\gamma = C_p/C_v$. For flow through orifices, an empirical relation is used. For liquids, Y = 1. Appendix A gives equations and tabular data for Y.

The factor F_a accounts for any expansion or contraction in the meter throat or orifice area due to temperature increase or decrease from the (ambient) temperature which existed when diameter d was determined. For operating temperatures within 50 °F of ambient, this effect can usually be ignored. Figure B-8 in appendix B includes graphical data for F_a as a function of temperature for several materials used in primary elements.

3. CALIBRATION METHODS FOR DIFFERENTIAL PRESSURES AND OTHER TYPES OF METERS

3.1 TRANSFER REFERENCE FLOWMETER SYSTEM

A transfer reference flowmeter system consists of one or more flowmeters with upstream and downstream piping sections along with flow straightening vanes and (usually) all necessary transducer and readout equipment. This system is calibrated at a primary calibration facility such as NBS or a qualified independent laboratory and then installed and used on site as the reference meter to calibrate the working meter(s). For ΔP reference systems, the pressure transducers and their sensing lines must be included in all primary and routine calibrations since they may be influenced by the system performance. For maximum accuracy, the transfer flowmeter should be calibrated using the same fluid at test conditions (pressure, temperature, flowrates) which can be expected to occur during later use on site. In this way, the performance of the working meter can be "traceable" to a primary calibration facility. Periodic recalibration of the transfer reference flowmeter system is imperative. Maintenance of a reference file on each meter will help the user determine the frequency of a recalibration schedule.

Use of transfer reference systems is often the most practical method for calibrating building-systems working flowmeters. Direct calibration may be too expensive, particularly where large numbers of on-site meters are involved. While there is some loss of accuracy over that achievable by direct calibration of the working meter at a primary calibration facility, perhaps of order 2:1, there is a compensating gain in efficiency since one transfer meter may be used to calibrate many working meters. On-site calibration may tend to mask errors in meter performance due to local piping system characteristics such as distorted flow patterns downstream of pipe bends, or temperature gradient effects in ΔP meter sensing lines. Although such effects may be unknown, every effort should be made to eliminate distorted flow patterns through use of good meter installation practices. For example, the configuration shown in figure 5 is preferred over that shown in figure 6.

An ideal fluid flowmeter responds only to flowrate or fluid quantity collected and, for a given flow, it will always produce the same output. In the real world, however, many factors can affect the flowmeter performance. To serve as a suitable transfer (or working) meter, all parameters or factors which affect performance must be known and must remain under adequate control. For example, many types of meters are influenced by fluid viscosity. Furthermore, the ΔP meter calibration factor or discharge coefficient often varies with flowrate. In addition, swirling flow or transverse flow components such as those immediately downstream from elbows and valves influence the accuracy of many meters. Many meters fail to operate satisfactorily in dirty fluids containing foreign particles such as rust or pipe scale which have detrimental effects on meter repeatability. All such factors need adequate attention and control.

Figure 5 shows a flow schematic for on-site calibration of a working meter system with the transfer meter system downstream, and figure 6 gives a similar schematic with the transfer meter system upstream. In each, valve functions are as follows:

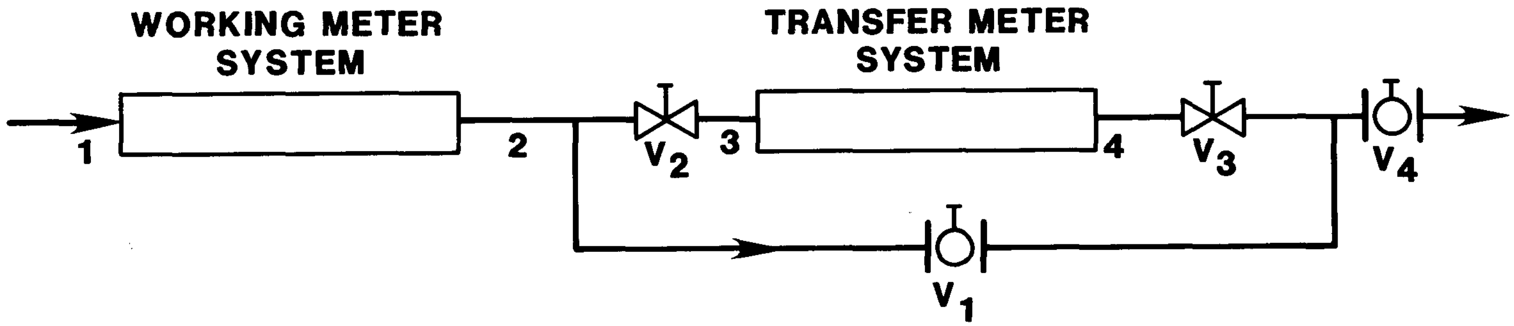


Figure 5. On-site calibration, transfer meter system downstream

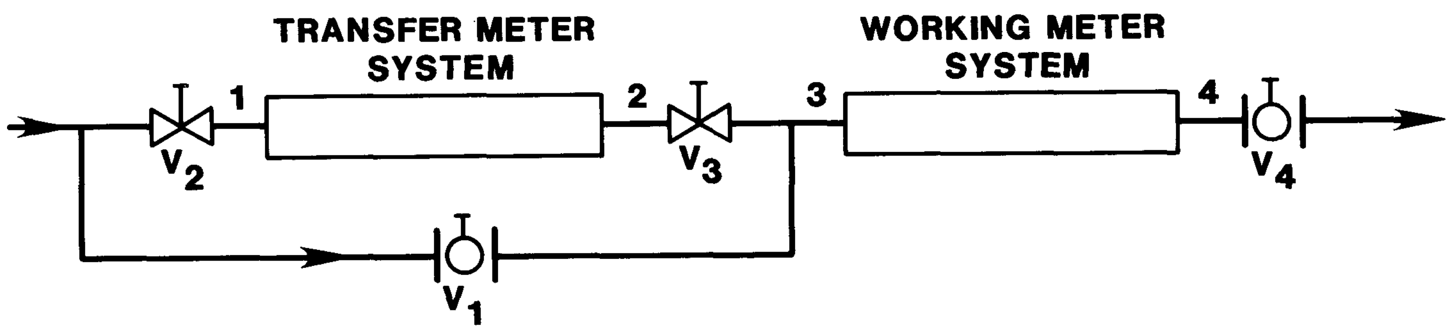


Figure 6. On-site calibration, transfer meter system upstream

Valve	Calibration	Normal	Notes
V1	Closed	Open	Blocking valve, positively no leakage allowed, low ΔP, suggest ball valve
V2, V3	Open	Closed	Isolation valves, full open during calibration, low ΔP, suggest gate valves
V4	Control	Open	Flowrate control valve, must be downstream of both systems, fine control necessary, low ΔP when open, suggest ball valve

Note: Valves V1 - V4 should be permanent in the working system to permit periodic re-calibration.

Since all fluid must pass through both meters, the need for a high quality, non-leaking blocking valve V1, can not be overemphasized. An alternative would be the use of three valves in lieu of V1 in a Tee arrangement with the leg valve vented to the atmosphere during calibration. Any leakage would be apparent through this vent valve. Likewise, there should be no leakage through the branch circuits between the two meter systems. Valves V2 and V3 allow the transfer meter system to be deactivated or used elsewhere during normal operation. With the transfer meter system downstream (figure 5), V4 could be eliminated, with flow control accomplished through V3. However, V2 should never be used for flow control in either case because this could disturb the flow pattern in the transfer meter system. For the same reason (flow pattern control), the transfer meter system downstream configuration is preferred.

Figure 7 gives minimum straight pipe requirements for each part of the transfer and working meter systems. Each metering system requires a minimum of about 20D of straight pipe. Flow straightening vanes or tube bundles are necessary in both cases, unless very long straight pipe lengths (estimated greater than 100D to 150D minimum) exist upstream. It is noted that most standards, including reference 2, recommended shorter lengths when no straightening vanes are used. However, more recent research indicates that these lengths are too short to eliminate the predicted effects on accuracy. See reference 4 for experiments conducted with elbows upstream of orifice plates and reference 5 for experimental results which show that swirling flow decays at quite low rates in pipes at Reynolds numbers comparable to those existing in building systems (turbulent flow). Also, it must be realized that tubular type flow straighteners only eliminate swirling or transverse velocity components; they will not completely reshape the axial velocity profile to a fully developed or even a symmetrical one. Therefore, a meter which requires a fully developed velocity profile such as the transit time ultrasonic type may not perform to best accuracy unless some other flow conditioning device such as a perforated plate type flow straightener or very long straight runs are used to aid in development of a fully developed profile. However, positive displacement (PD) meters are not normally affected by swirling flow and flow straighteners may often be

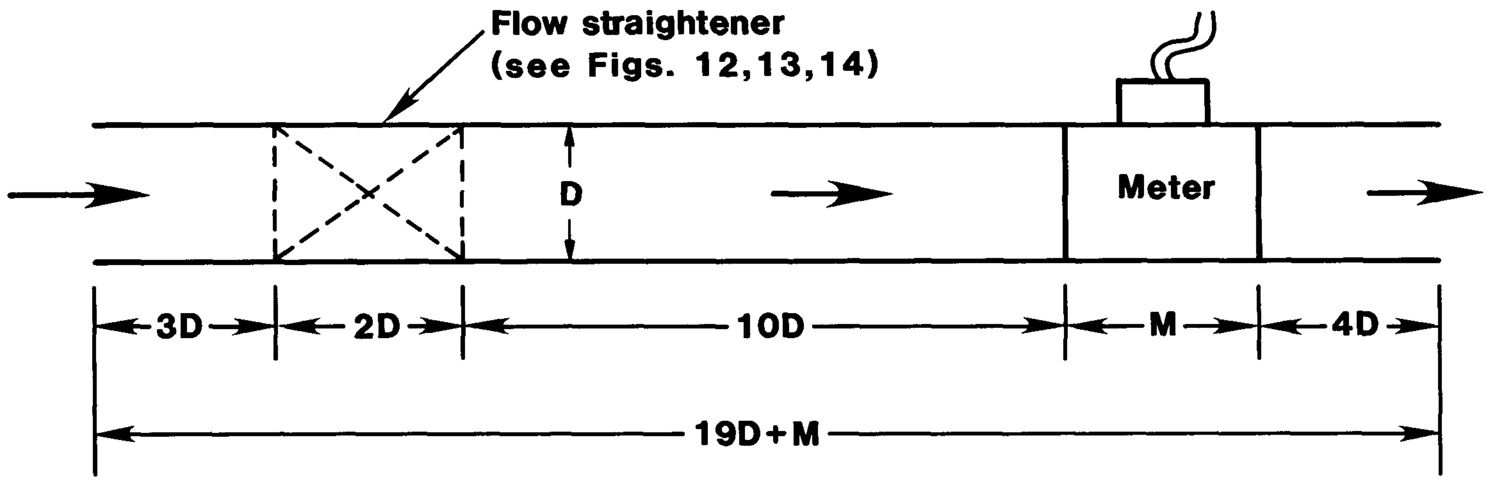


Figure 7. A schematic diagram of promoting straight flow in transfer and working meter systems. Relative pipe dimensions shown are minimum (applicable for positions 1 and 3 in figures 5 and 6)

eliminated. Consequently, PD meters are often recommended for situations where adequate straight piping runs are unavailable.

3.2 DIRECT CALIBRATION METHOD

The calibration of the working meter system against a volumetric or weighing type fluid measurement system is known as a direct calibration. The working meter system is either removed from the building system and calibrated at the primary calibration facility, or the working meter is calibrated using on-site volumetric or weighing facilities. When it is to be calibrated at the primary facility, it is important that the entire working meter system (figure 7) be removed and calibrated at this facility. During on-site calibration, the fluid is directed into the volumetric or weighing system downstream of the working meter with precautions taken to ensure that no leakage or branch circuits cause fluid to be lost before entering the calibration equipment. The following comments pertain to direct calibration on site.

For building systems involving flowing water, a direct on-site calibration may be more satisfactory than the transfer method for differential pressure (ΔP) meters of sizes accommodating less than 2-inch pipe for the following reasons: the uncertainties of the discharge coefficients are large and sometimes unavailable; and, flow measurements in these smaller sizes can usually be made with satisfactory accuracy using commercially available equipment of reasonable size and cost. (For example, commercially available scales offer good accuracy at reasonable cost.)

The calibration of a volumetric metering system by use of smaller reference volumes or by physical measurement is often considered less convenient. Therefore, volumetric-type meters are often calibrated using a volumetric-type meter which has been calibrated using the gravimetric calibration method with the water in the same temperature range.

A gravimetric-type calibrator system for water is shown in figure 8. The basic equipment are the scales and weigh tank, a timer, a thermometer, valves, and a timer switch. Gravimetric systems are classified into two types according to the method used in collecting the water: the static-weigh and the dynamic-weigh methods.

Static-Weigh

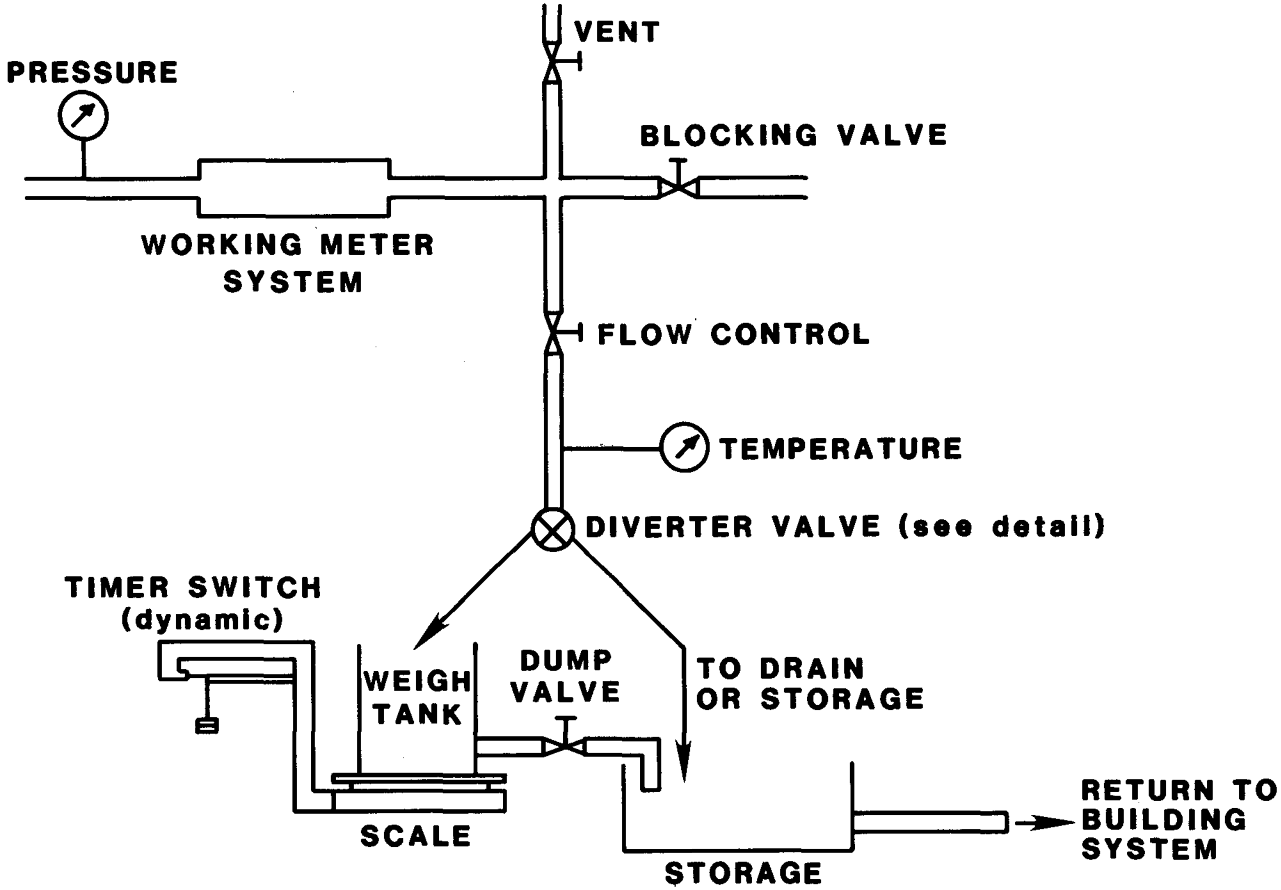
Operation consists of:

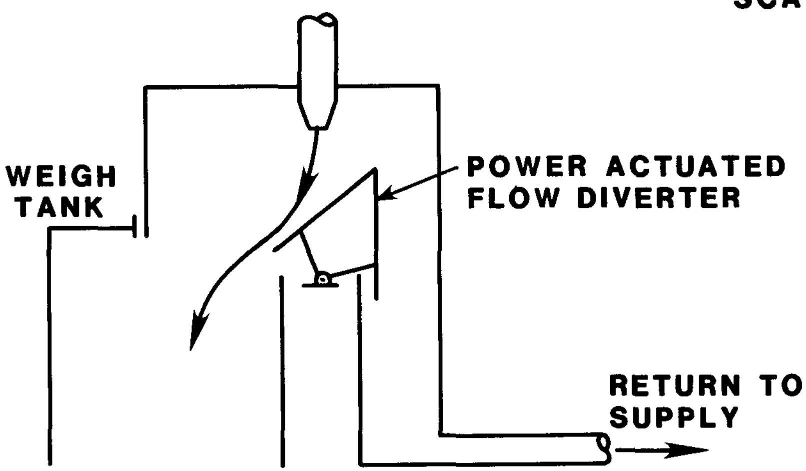
1. Set flowrate.

2. Determine tare weight (mass) while water is diverted to drain or storage.

3. Divert flow to weigh tank, starting timer (or working meter counter for totalizing operation).

4. After collection of desired amount of water, divert flow to drain or storage, stopping timer or counter.





DIVERTER VALVE DETAIL

5. Determine gross weight (mass).

6. Drain weigh tank through dump valve (or pump out).

   Flowrate = Net Weight (mass)/Time Interval
   Meter Output/Pulse = Net Weight (mass)/Total Count (pulse type output).

Weighing is performed with water stationary in the tank and the net weight (mass) can be determined quite accurately. The weighing, however, requires two operations and thus becomes time consuming. A critical component is the diverter valve. Its function is to direct the flow as desired to drain or storage, or to the weigh tank without disturbing the rate of flow through the working meter. The timer switch is connected directly to the diverter valve and should be adjusted to activate at the midpoint of the diverter operation. This switching is often performed using a suitable photocell and light source to develop a sharp on/off relationship at the midpoint. When the desired uncertainty is one percent or more, manual timing will usually be satisfactory if the diverter valve and timer are operated with the required care.

Dynamic-Weigh

Operation consists of:

1. Set flowrate with dump valve open and tare weight on the weigh beam.

2. Close dump valve. Tank begins to fill.

3. When tare weight is balanced, beam rises, starting timer or counter.

4. Place weights on scale beam pan corresponding to the desired amount of water being collected, lowering beam.

5. When total counterpose weights are balanced, beam rises again, stopping timer or counter.

Thus, the time interval or total count for a preselected amount of water to enter the weigh tank is determined, and the weighing operation is completed "on the run" while the water enters the tank. This method has the advantage of consuming less time and offers greater convenience compared to the static method. The timer switch activated by the weigh beam may be a direct contact, magnetic, capacitance, or optical type. When load cells are used for weighing, the timer start/stop circuitry must, of course, sense the load cell system output. Possible errors due to the change in inertia of the water and added weights should be considered. This inertia error is discussed in reference 22. When the desired uncertainty is one percent or more, manual timer actuation may be used.

Pertinent features of both of these systems include:

The weigh tank capacity should be sufficient for a minimum collection time of 30 seconds at the maximum flowrate plus tare weight collection time. For manual timing, a minimum collection time of 1 minute is recommended.

- It is important that no part of the system touches or rubs against the weigh tank. If it is mandatory that the system be connected to the weigh tank, then the construction should be such that no vertical force components are exerted on the scale from the system.

- It is essential that all of the flow and only that flow passing through the working meter be collected in the weigh tank. There must be no leakage through the blocking valve or through any branch circuits. Any air or vapor must be vented at the high points. Keep all meter discharge lines as short as possible.

- Commercially available scales of conventional design with weigh beam and counterpoise weights are recommended for high accuracy at reasonable cost. For a calibration accuracy of 0.5 percent, a desirable scale accuracy would be 0.05 to 0.1 percent of reading. The scale should be calibrated in place with static weights, Commercial Class C, before the weigh tank is installed, and after the installation of the tank connections is complete. This procedure checks on possible interference from any connections. In selecting a scale, the scale sensitivity (weight necessary for shifting the weigh beam from an equilibrium position at the midpoint to an equilibrium position at either extremity) should be sufficient for the desired accuracy. However, more sensitivity than necessary should be avoided because scale "settling time" usually increases with sensitivity which increases the static weighing time unnecessarily. During dynamic weighing, increasing beam oscillations may generate unnecessary problems.

- When weighing to an accuracy of 0.5 percent or better, the air buoyancy effect should be considered. This effect is about 0.12 percent when water is weighed in air.

- No leakage is allowed through the dump valve during the weighing or collection operations.

- The minimum net weight (mass) of water collected should be at least 10 percent of the scale capacity. When totalizing pulse-type meter outputs, the minimum net weight (mass) should correspond to at least 1000 pulses, preferably 5000.

- Water density is determined from the temperature and pressure, table B-1. Linear interpolation may be used if necessary. A mercury-in-glass thermometer is preferred, and it should be graduated in single degrees, accurate to one degree Fahrenheit or better.

- Water temperature should not exceed about 150 °F due to liquid volatility and possible evaporation losses. At this temperature level, the weigh tank should be provided with a cover to reduce this loss. Safety precautions should be taken to avoid personnel injury when handling water at these temperatures in a system open to the atmosphere.

An example calibration of an orifice meter with a gravimetric calibrator system, including sample data and calculations, is included in appendix E, example E.5.

Direct calibration of meters used to measure the flow of a gas such as air requires a closed weigh tank or a volumetric tank. Such approaches are used in primary calibration facilities, but because of their complexity and costs they are considered generally impractical for calibrating working meters on-site.

Flowmeters in building systems monitoring steam flow may receive a direct calibration by measuring condensate flow provided all steam flowing through the working meter is condensed and flows through a single pipe at a convenient location. Great care must be exercised to trace all lines to assure that no loss through branch lines and that no leakage occurs. The major advantage to the steam-condensate method is that liquid flow is easier to measure accurately than is gas flow. Figure 9 shows a flow schematic for a calibration for steam flowmeters. Long pipe lines and service equipment such as heaters will normally be located between the working meter and the calibrator. Thus, calibration should be attempted only during periods of steady steam load. Even then, long periods may be required to obtain a steady flowrate in the meter and at the calibrator. The system should be vented as necessary to expel all air. The calibrator may be a gravimetric type as discussed above, or a positive displacement meter properly calibrated at a primary facility.

Procedures and equipment which can result in accurate measurements of the mass flowrate, M, have been described above. Such measurements require the services of personnel who are knowledgeable and experienced in measurement techniques and have access to the required equipment, space and labor. The level of accuracy thus attained may not be needed nor economical. Alternatives include the use of a transfer reference meter system as discussed previously. For a ΔP meter, a differential-pressure performance evaluation of the flow measurement system based on the direct calibration of the ΔP transducer only is required.

3.3 PERFORMANCE EVALUATION FROM DIRECT CALIBRATION OF ΔP TRANSDUCER SYSTEMS ONLY

In some situations, an on-site calibration of the working meter using a direct calibration or a transfer reference meter may not be feasible. However, with differential pressure (ΔP) transducers used on meters of the orifice, venturi, or flow nozzle types, their performance can be evaluated by performing direct calibration of the ΔP, pressure, P, and temperature, T, monitoring systems only. By using published data to obtain discharge coefficients and fluid expansion factors for the meter primary element, the flow characteristics or performance of the complete working meter system can then be determined. This approach requires relatively little straight pipe run since no transfer meter system is involved. However, the ΔP, P, and T systems need to be calibrated. Appendix A is a compilation of available data and equations for discharge coefficients and fluid expansion factors for orifice, flow nozzle, and venturi type primary elements. Equations which may be used to assign uncertainties to the discharge coefficient and expansion factor are also included.

The disadvantages of this method include loss of accuracy due to the uncertainties assigned to the discharge coefficient and the expansion factor. In addition, when the ΔP transducer is required to perform only within a narrow band corresponding to its full scale output, the loss of accuracy at the low or

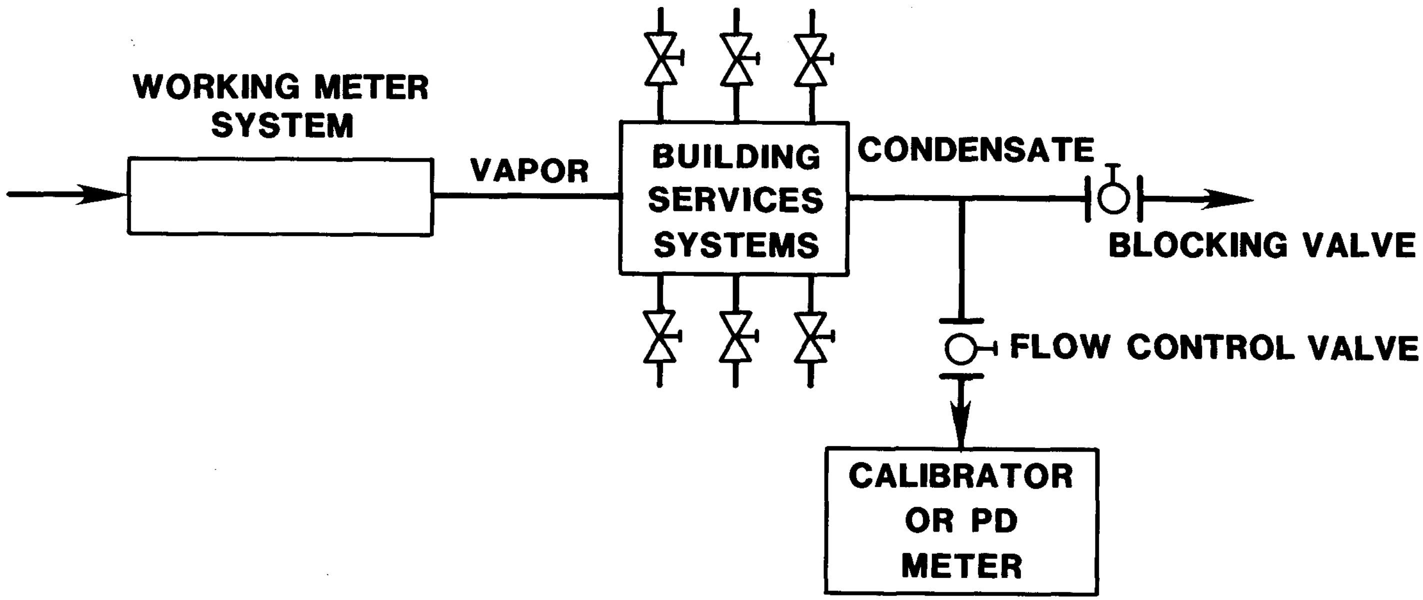


Figure 9. Calibration system for steam flowmeters

part scale ΔP values can become quite large. For example, a transducer rated at 0.5 percent full scale would need only produce a differential pressure measurement accurate to within 5 percent at a ΔP value of 10 percent full scale to still comply with its rated performance. This indicates the importance of using a transducer which has excellent repeatability characteristics and the need for periodic calibration over the operating range. Transducers are available which specify accuracy of the actual reading plus a small portion of the full scale reading (e.g., $\pm$ 0.15 percent of reading + 0.005 percent f.s.). Regardless of the specified accuracy, when a transducer is first put into service, 3 or 4 calibrations might be performed over a period of 6 months to a year and the results used to establish a performance curve from which linearity and hysteresis corrections can be made if necessary. Once the transducer performance has been established, intervals between the periodic calibrations may be increased. It should be noted that the ΔP transducers often require a specified "warm-up" period and respond to changes in the ambient conditions by loss of accuracy.

When a system's flow rate more than doubles (range 4:1 in ΔP), the use of two ΔP transducers may be preferred; one for HIGH ΔP and one for LOW ΔP. Significant improvements in the uncertainty are possible through the use of this technique. For example, figure 10 shows the actual systematic error e_{MS}/M, percent of rate, as a function of flowrate, M, for a HVAC system installed at NBS which uses two such transducers. For a flow range of 500 to 2000 lb/hr, the systematic error ranged from about 2 to 5 percent with two transducers, whereas with one transducer (HIGH), the error would have ranged from about 2 to 15 percent. While e_{MS} depended on the combined effects of several errors in this case, the error in ΔP dominated at low ΔP values. In this example, the ΔP transducer error was expressed on a full scale basis and no linearity corrections were made.

Reference 23 is a detailed analysis of the performance of an energy monitoring system for steam flow at NBS using the above approach of direct calibration of the ΔP, P, and T systems only. An appropriate method and sample calculation is given for estimating the uncertainty in the total energy consumed over a one-year period.

3.4 ADDITIONAL FACTORS TO BE CONSIDERED IN THE ON-SITE CALIBRATION OF DIFFERENTIAL PRESSURE METERS USING TRANSFER REFERENCE METERS

This section deals with on-site calibration of ΔP meters of the orifice, flow nozzle, and venturi types using transfer meter systems as the reference. The transfer reference meter is calibrated off site and its flow characteristics or performance including an estimated uncertainty, are known. The transfer reference meter is not necessarily a ΔP meter. Reasons for calibrating a ΔP meter on site include the situations when a direct flowmeter-system calibration using a gravimetric calibrator approach is not practical; when the assignable accuracy would be too low with calibration of the ΔP transducer only; or, when the performance must be verified.

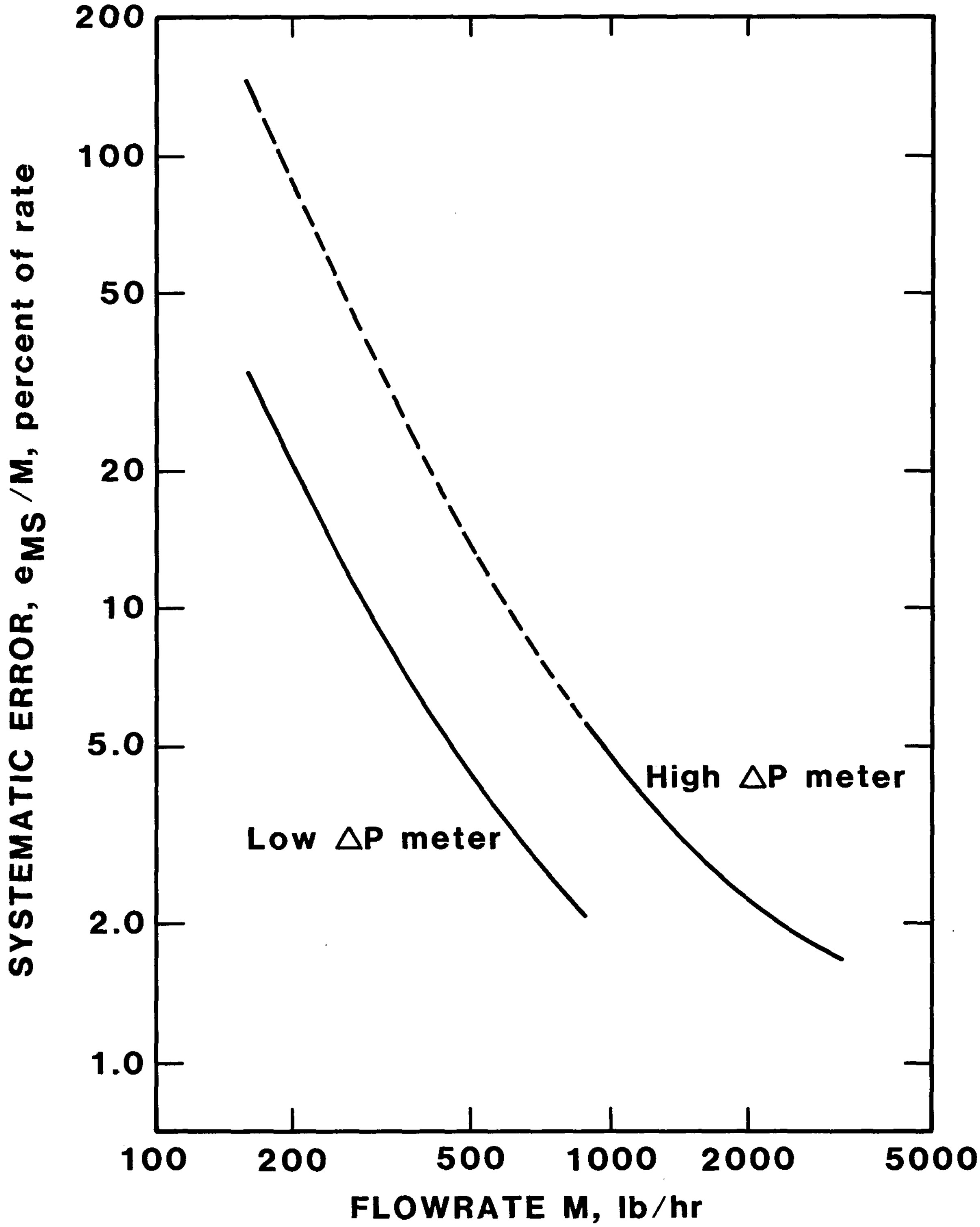


Figure 10. Systematic error in flowrate M

Pertinent features of ΔP meters include:

- Flowrate varies with the square root of the pressure differential $(\Delta P)^{1/2}$, as indicated in equations (2-1) and (2-2).

- Primary elements (the parts which interact with the flowing fluid) may be used in air, water, or dry (superheated) steam. For wet steam, consult reference 1.

- Discharge coefficients and fluid expansion factors for these primary elements are well established and used extensively. However, the on-site calibration determines the discharge coefficient which may vary with conditions at the particular installation.

- Accurate measurement of the output of a ΔP metering system is relatively difficult (particularly at ΔP values of a few inches of water) compared with meters producing a pulse type output (e.g., vortex shedding, positive displacement, turbine).

- Since flowrate varies with $(\Delta P)^{1/2}$, the practical flow range is often limited to about 3:1 for a single ΔP transducer because many transducers are rated on a full scale ΔP basis.

- For best accuracy of an on-site calibration, the transfer reference meter should be calibrated using the working fluid under test conditions which include temperatures and flowrates that duplicate those of the working meter.

- The detailed calculations necessary to obtain the flowrate or total flow through equations (2-1) or (2-2) make a strong case for an automatic data processing approach when measurements are to be made over a long period or when many flows are to be monitored.

Meter system design, construction, and installation should receive careful attention regardless of whether or not the meter is to be directly calibrated. Common trouble areas include the pressure taps and the pressure sensing lines. The pressure differential sensed by the transducer should be exactly the same as that existing at the meter taps. The pressure tap hole at the inner surface of the pipe or meter section should contain no burrs or roughness or other irregularities. Special precautions should be taken when installing the sensing lines, keeping these lines short and close together, and including adequate provisions for flushing, venting, and drainage. Figure 11 shows a recommended sensing line schematic. The bypass valves facilitate transducer zero ΔP checks. To ensure a positive check on bypass valve leakage (liquids) during normal operation, the leg valve in the Tee network remains open to the atmosphere. The likelihood of accumulation of rust, scale, and other solids in the lines and in the ΔP cell make periodic flushing of the lines imperative.

Another common trouble area is the lack of sufficient straight pipe lengths upstream and downstream. As discussed previously, it is important that the flow field posseses no swirling components. According to recent research

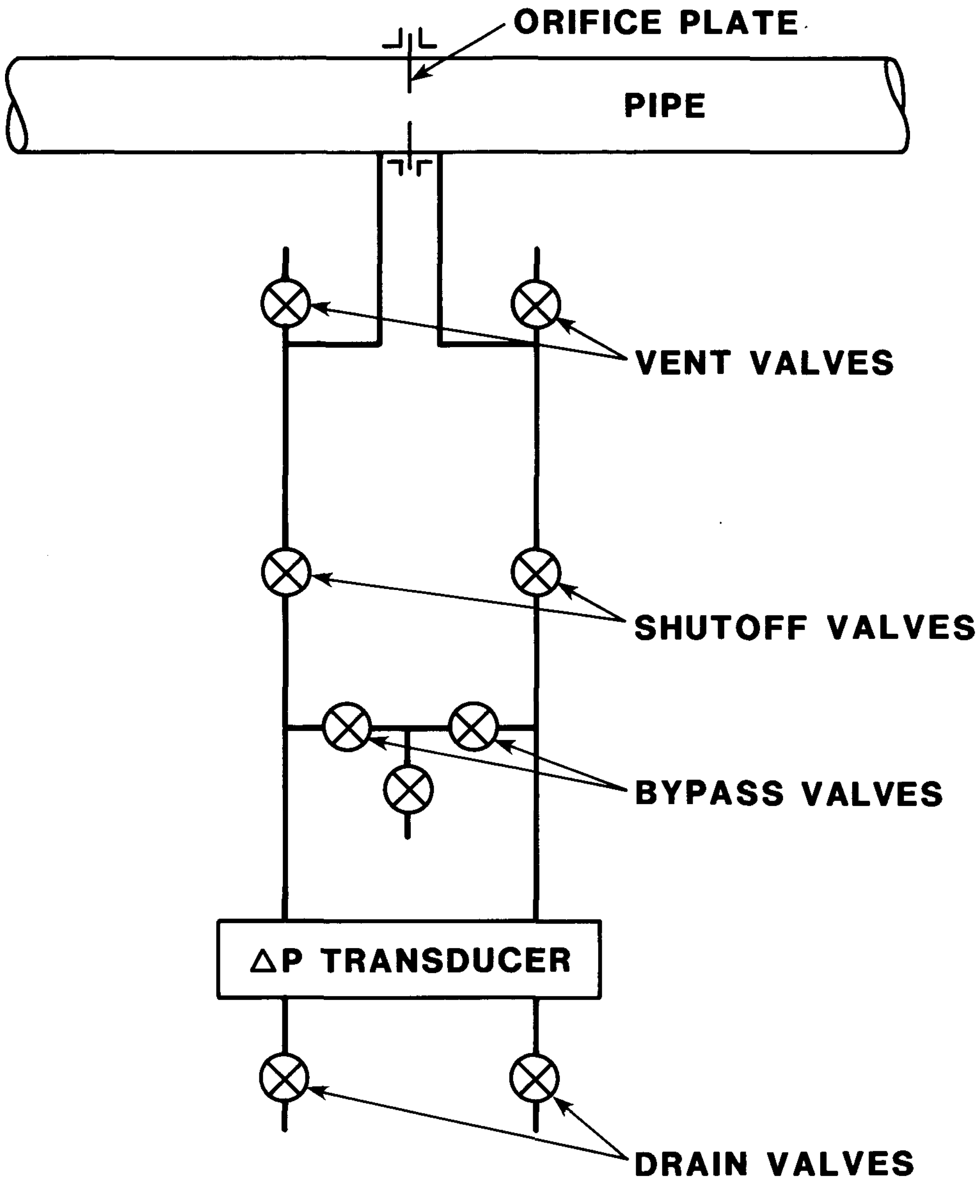


Figure 11. Orifice plate sensing line and valve schematic. Applies to flow nozzles and venturi meters as well.

(references 4 and 5) most standards are simply too optimistic about the lengths of straight pipe needed to dampen the swirling components. This fact alone can often justify calibration on site when specific accuracy requirements are to be met. For best accuracy, the straight length upstream should be 100 to 150 pipe diameters minimum. If this is not feasible, a flow straightener section or vane assembly should be located a minimum of 10D upstream of the meter as indicated in figure 7. Suitable flow straightener designs are shown in figures 12, 13, and 14. The diameter of the straightener tubes in the tubular design should be D/4 maximum. When thin wall tubes are used, the pressure losses will be small. The perforated plate design will cause relatively large pressure loss but it will tend to reshape and improve the symmetry of the axial velocity profile. For more detailed discussions of design and construction, consult references 2 and 3.

An example calculation of the calibration of an orifice meter system using a transfer meter system is given in appendix E, example E.6.

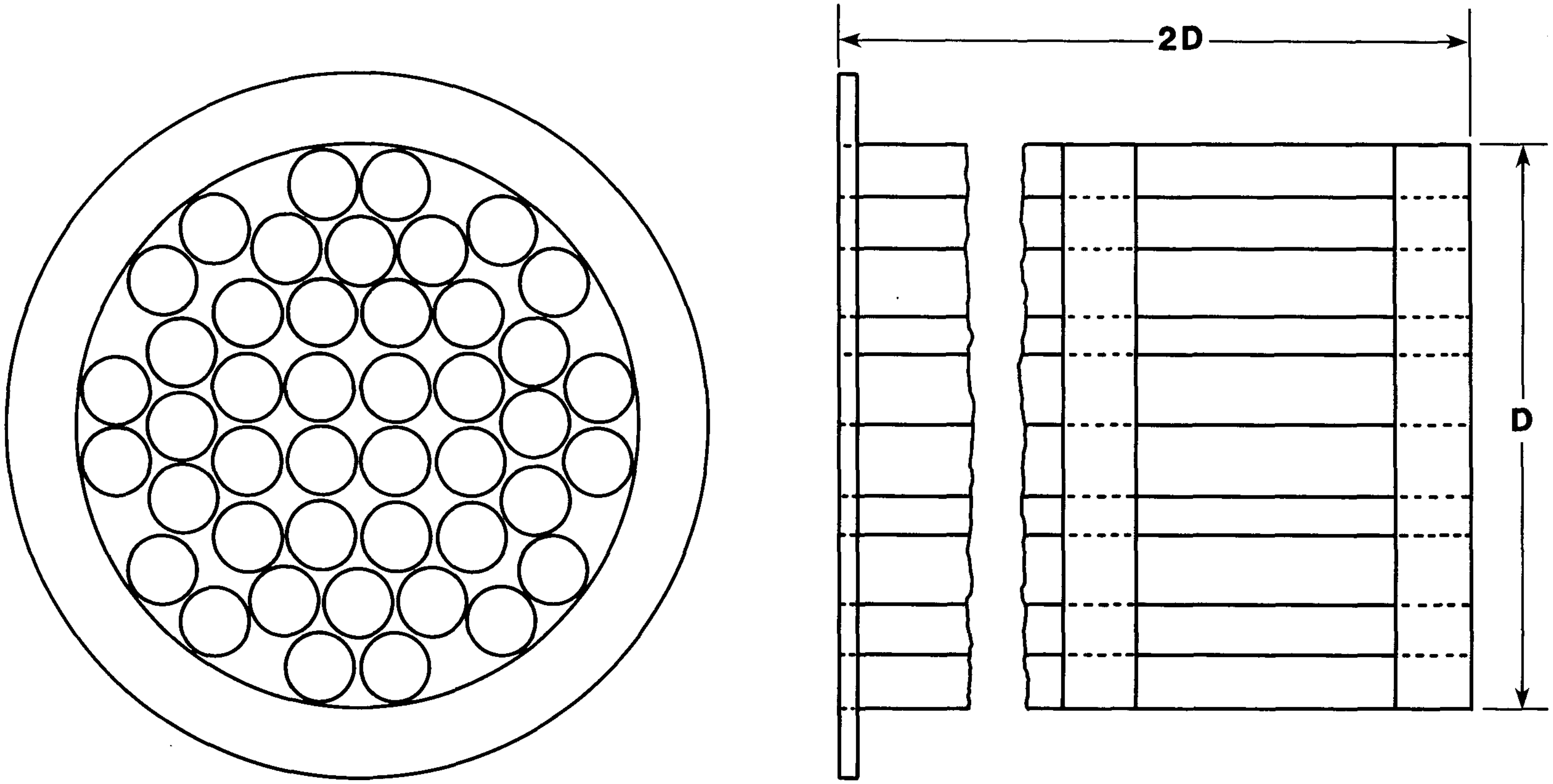


Figure 12. Tubular flow straightener design [2]

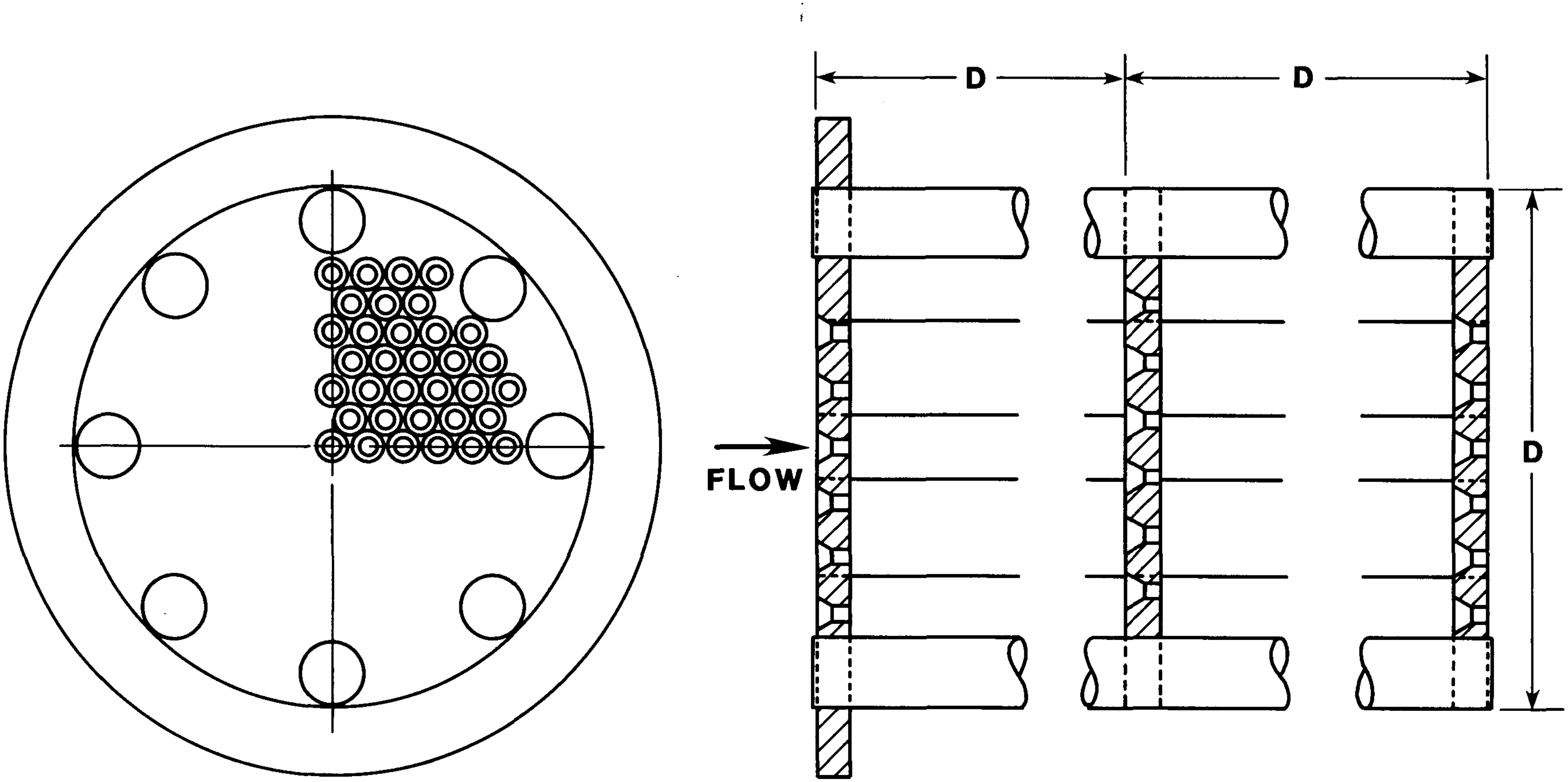


Figure 13. Perforated plate flow straightener [2]

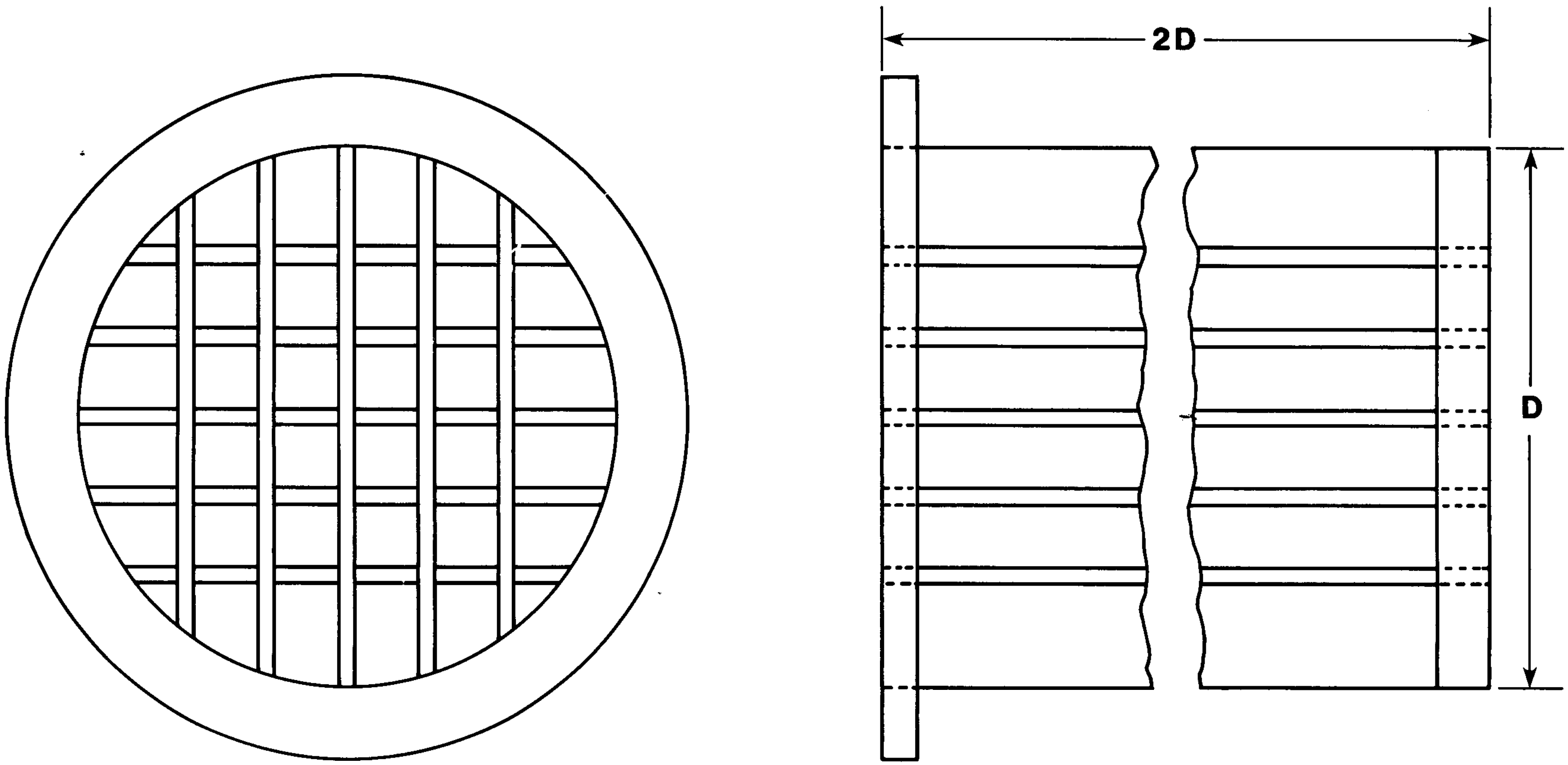


Figure 14. Cross plate flow straightener [2]

4. ON-SITE CALIBRATION OF OTHER FLOW METERING SYSTEMS

The types of flowmeters discussed in this "other" category are indicated in the table below. Depending upon the application and accuracy requirements, they all require calibration. The fluids indicated are those considered most likely to be encountered in building service and do not necessarily encompass all possible fluid applications. Included in this section are meter descriptions and summaries of operating principles, graphs showing typical performance, installation and use notes, basic equations for calculating the flow, and sample calculations for on-site calibration.

Flowmeter Type	Water	Air	Steam
Positive Displacement	x		
Vortex Shedding	x	x	x
Turbine	x		
Pitot-Static Types			
Reverse Pitot	x	x	x
Multiple Pitot Static		x	
Target	x		
Ultrasonic	x		
Insertion Type Turbine	x	x	x

4.1 POSITIVE DISPLACEMENT (PD) FLOWMETER

This is a quantity-type meter in which a chamber is completely filled with fluid and then emptied. Counting each filling indicates the flow. The counter may be a mechanical, totalizing type with disc or wheel type readout; or an electromagnetic pulse or optical type readout may be used. This will allow the meter to be used for either flowrate or totalizing applications. In some cases, the primary element is magnetically coupled to the counter, eliminating shaft sealing. The PD meter may be considered a special type of fluid motor with a high volumetric efficiency and operating under light load.

The PD meter is widely used in commercial, industrial and domestic applications, metering both gases and liquids, including water. For liquids, the types of primary elements used include the nutating disc; reciprocating piston; oscillating or rotary piston; rotating gear, lobed impeller; and sliding and rotating vanes. Figure 15 is a cross-section view of a nutating disc type.

The advantages of the PD meter include the ability to measure flow despite a wide range of fluid viscosities, a digital type output, and accuracy which is

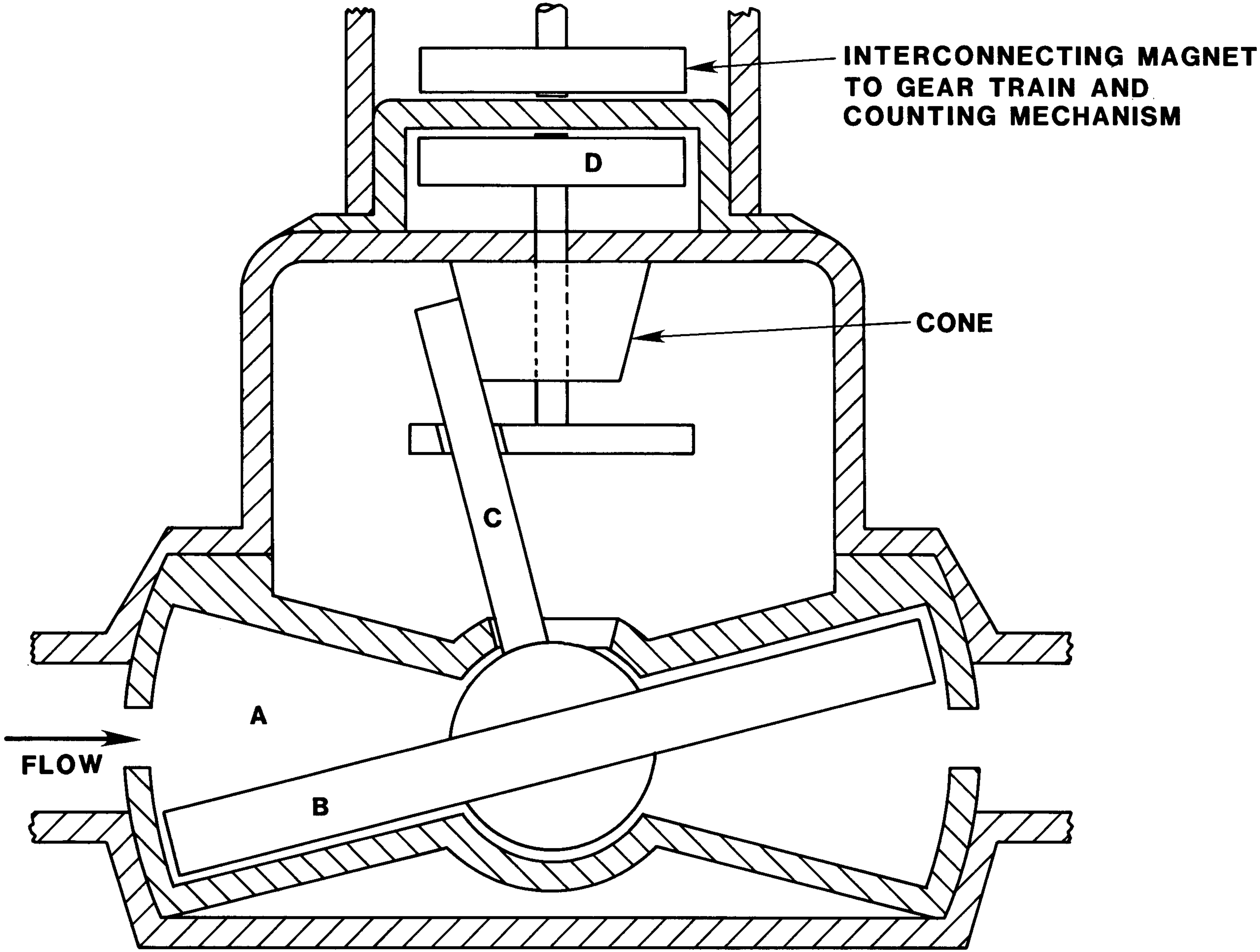


Figure 15. Cross-section of a positive displacement meter. Liquid flowing through chamber (A) causes disc (B) to nutate. This motion results in the rotation of spindle (C) and drive magnet (D).

relatively insensitive to upstream flow patterns. Thus, long straight pipes or flow straighteners upstream may not be necessary. Disadvantages include the presence of significant loading due to the mechanical and fluid friction of the metering element as well as that arising from the torque required to drive the readout or registering mechanism. Because of this loading, a pressure differential is required to drive the meter; as a result there is a small amount of fluid leakage past the "sealing" surfaces. This leakage is known as "slip" or "slippage". The accuracy of the meter is largely determined by this leakage or "slip". No two meters, even of the same design, will have identical proportions of leakage. Therefore it is important, for applications requiring accuracies better than about one half percent, that each meter be calibrated individually, and that all parameters or factors which affect the performance be known and adequately controlled.

Factors associated with leakage which are known to affect the meter performance include: liquid properties, viscosity and lubricity, meter design, meter temperature, and readout loading. Other factors influencing meter performance include flowrate; pressure level in the meter; contamination in the fluid; and, possibly, meter operating position.

Thus, in the selection of a PD meter either as a working meter or as a transfer reference meter, the flowing fluid properties (viscosity, lubricity, and temperature) should be considered. The pressure level and allowable pressure loss also need consideration. For water, typical pressure losses at the rated flow may range from 2-5 psi for capillary or film seal meters (example: nutating disc, oscillating piston) and from 10-15 psi for packed seal (reciprocating piston) meters. For most applications, an absolute pressure level (psia) equivalent to the vapor pressure plus 3 or 4 times the meter pressure loss at the rated flow is probably sufficient for proper operation. The user should refer to the manufacturer's specifications for details on each meter. In calibration, the pressure drop across the meter should be noted.

The meter should be treated as a precision instrument. Filtering of the fluid so that solid particle size is much smaller than the meter clearances is necessary. Although optimum conditions are not known, filtration in the range of 25 to 50 microns appears adequate to ensure normal meter operation without excessive wear. No meter, including the nutating-disc type widely used in domestic and industrial water applications, should operate continuously in fluids containing solid particles.

Before use as a transfer reference meter, a PD meter should first be calibrated using the same fluid at or near the temperatures and pressures which will exist during field usage. Thus fluid viscosity and lubricity, meter dimensions, and seal leakage are duplicated during calibration and use as a transfer reference device. In this manner, meter performance is duplicated. Meter performance is affected by meter orientation (vertical vs. horizontal, if permitted by the manufacturers). Positional changes vary the mechanical friction in the metering element and in the readout mechanism, and modify the lubricant level in the gear reduction mechanism of the readout. A safe approach is to calibrate and use the meter in the same position. Magnetic and optical readouts may not impose the same positional operating restriction. The user should always refer to the manufacturer's specifications regarding the positioning of the meter.

Several points should be noted with regard to the meter readout, when the PD meter is used as a transfer reference. Although the mechanical readout has long been the "work horse" for domestic, commercial and industrial applications, other types such as the magnetic or optical types are available. A small change in the loading of the metering element by a mechanical readout may have a significant effect on the slip characteristics of the meter, especially those of the film seal type. Piston-type meters with packed seals are said to be less affected by small changes in readout loading. Because wear or the presence of dirt or corrosion can change readout loading, recalibration of the meter on a regular schedule, with more frequent intervals at the beginning of the program, should be considered for long term measurement programs.

Also, with regard to readouts, the meter should not be calibrated with one readout and used with another because of the influence of readout loading on the meter accuracy. The readout and metering element should be connected by a positive drive, without cams, clutches or the like which may cause relative motion between the readout and the metering element; the sole exception is the case where the meter and readout have been designed to be coupled magnetically. In this type, the rotary motion of the metering element is magnetically coupled to a mechanical readout or else it drives a pulse generator, producing an AC voltage which varies with a frequency exactly proportional to the speed of the metering element. Such a system, by proper design, should result in decreased loading on the metering element and increased resolution in the readout. The meter is also more readily adaptable to both rate and totalization applications. Suitable external instrumentation such as an electronic counter is used to count the pulses received from the meter.

Air and vapor in the PD meter should be avoided because of effects on meter operation and accuracy. Thus, during installation, meter layout should be planned to avoid locations where air could accumulate and valves should be provided for venting the system during use, usually at points of high elevation. Throttling upstream of the meter should be avoided whenever possible.

Pulsed-readout meter performance can be expressed conveniently in terms of a "calibration factor" plotted as a function of meter output frequency when the fluid, viscosity, and meter temperature are known, i.e., when a given meter is operated on a single fluid at a known temperature. Figure 16 shows such a plot of meter performance. While the sample calibration factor, pulses/gallon, is shown to vary with frequency (flowrate), the calibration factor may be essentially constant over considerable flow ranges such as 10:1 or larger for a typical meter operating on a single fluid. Also, large changes in kinematic viscosity such as 1×10^{-5} to 20×10^{-5} ft^2/s may have a rather small influence on the meter factor, tyically $\pm$ 0.5 percent for a five or ten-fold increase in flow.

Denoting the calibration factor by the symbol K, the volume flowrate Q, is

$$Q = 60(f/K) \text{ GPM} \tag{4-1}$$

and the mass flowrate M, is

$$M \doteq 3600\ (\rho)(f/K)\ \text{lb/hr} \tag{4-2}$$

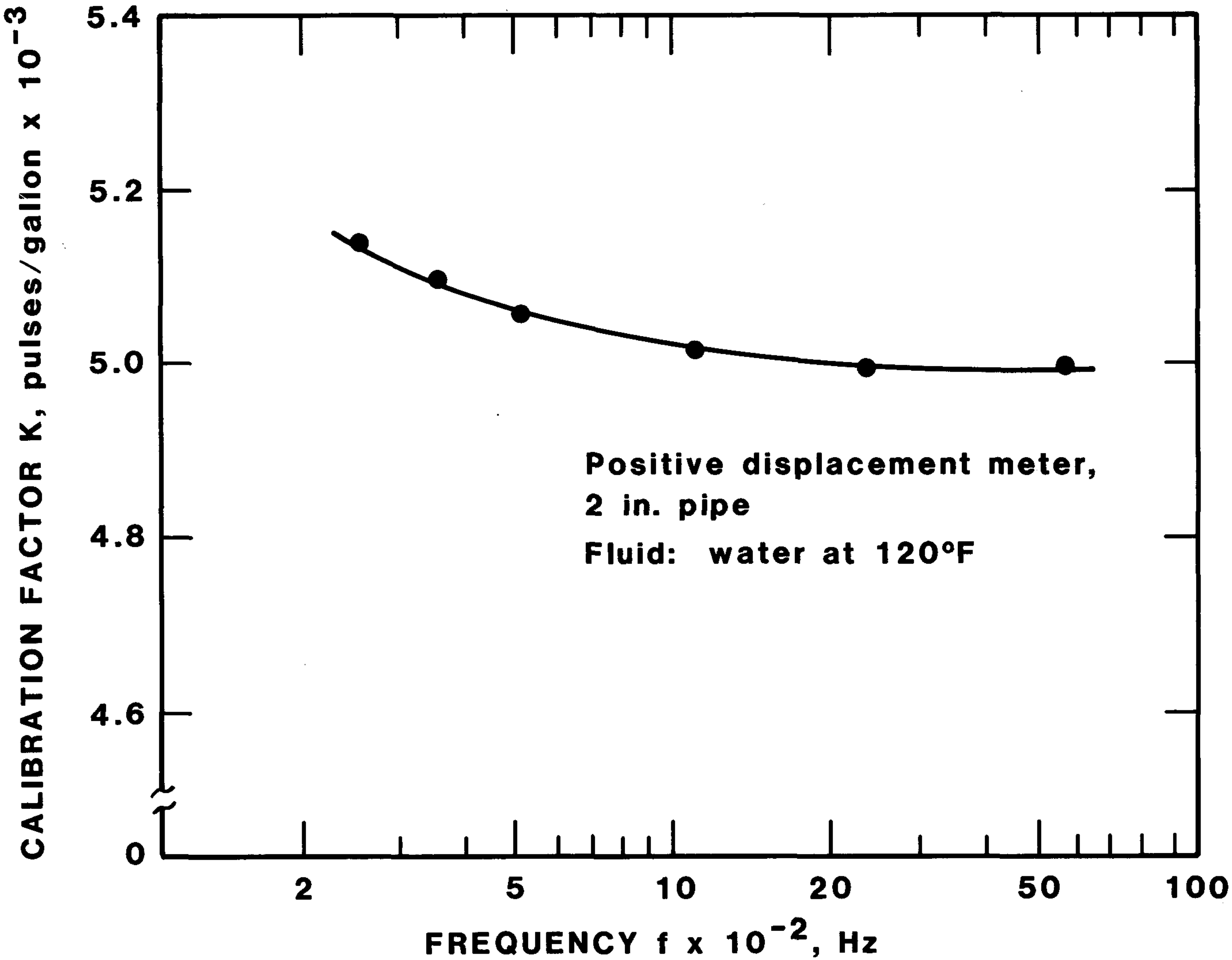


Figure 16. Positive displacement meter performance

where, t, is the pulse frequency in Hz, K has units pulses/gallon, and ρ is the density, lb/gallon.

The actual totalized flow (ATF) where the meter register reads directly in volume units, is

$$ATF = [(K_o/K)\ (\text{net meter registration})], \tag{4-3}$$

where K_o is the calibration factor corresponding to the current setting of the meter register. The quantity K_o/K is defined as the meter factor (MF), a dimensionless number, ideally equal to 1.000.

An example calculation for on-site calibration of a positive displacement meter is given in appendix E, example E.7.

In summary, the PD meter is considered quite suitable for building system applications. Advantages include: (1) a digital output, (2) frequent calibrations are usually not needed once a meter element is calibrated and used with clean fluids, (3) elimination of periodic and long term calibration programs for output signal transducer systems, and (4) a calibration factor essentially independent of flowrate.

4.2 VORTEX SHEDDING FLOWMETER

This meter, part of the industrial scene for about 10 years, operates on the principle that the frequency of vortex shedding for fluid flow around a submerged object is proportional to the fluid stream velocity. Flowrate is measured by detecting the frequency. Figure 17 shows design details of one meter. The vortices are shed behind the bluff body.

Advantages include lack of moving parts in the primary element* and a digital output. Accurate measurement of the probe output is a much simpler measurement task than accurate measurement of, for example, the ΔP from a differential pressure meter. Meter configurations are available for both gases and liquids in pipes one inch or more in diameter, and at temperatures up to about 400 °F or 500 °F. The meter output is usually expressed in dimensionless terms according to the following equation:

$$f\ D/V = \phi(D\ V\ \rho/\mu) = \phi(D\ V/\nu) \tag{4-4}$$

where
$f\ D/V$ = Strouhal number, dimensionless
$D\ V\ \rho/\mu$ = Reynolds number, dimensionless
V = fluid velocity
D = characteristic linear dimension of the meter
f = frequency of vortex shedding
ρ = fluid density

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* Some models do employ moving parts in the secondary element that detects the vortex frequency.

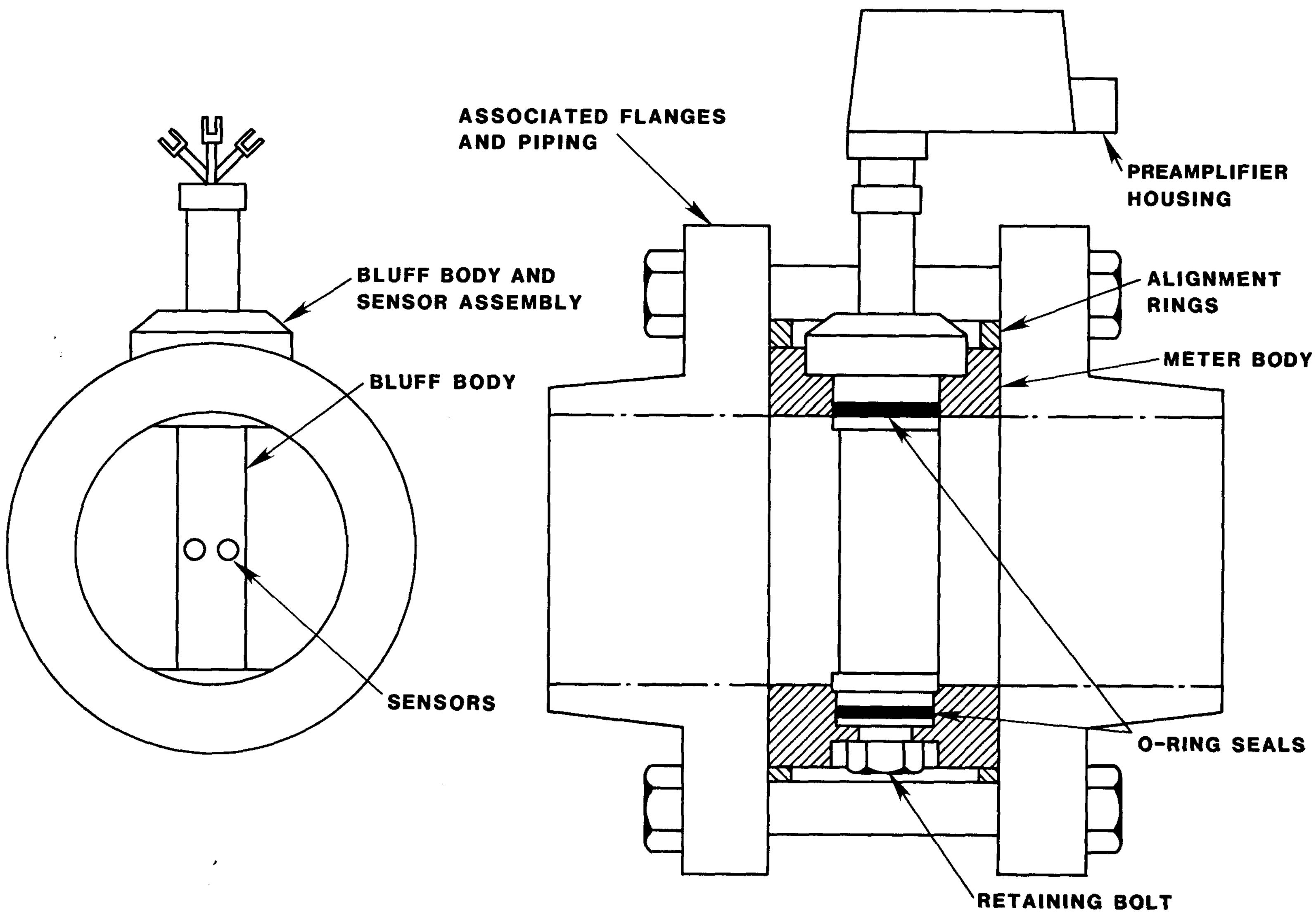


Figure 17. Mechanical design of a vortex shedding flowmeter [13]

μ = fluid dynamic viscosity
ν = fluid kinematic viscosity ($\nu = \mu/\rho$).

The function ϕ is determined by calibration for each geometrical shape of meter in the same sense that the coefficient of discharge C for an orifice is determined experimentally for each geometry (such as concentric orifice, beta ratio, pressure tap configuration). In principle, the performance of a meter calibrated on one fluid is predictable when used with another fluid provided that each fluid is incompressible and that its properties ρ and μ are known. The meter is a volumetric type as indicated by use of the Strouhal number f D/V, where V is proportional to Q, the volumetric flowrate.

When a vortex shedding meter is calibrated using a single fluid at a specific temperature, its performance can be conveniently expressed in terms of a calibration factor K, such as pulses/ft^3 plotted as a function of frequency f, rather than the dimensionless parameters of equation (4-4). When a meter is to be used for several fluids, a function f/ν may be used instead of DV/ν since D is constant for a given meter and f varies nearly directly with V. Verification of the meter's accuracy requires calibrating the meter with several fluids to encompass the range of interest for ν. Note that the function ϕ accounts for changes in meter performance with fluid kinematic viscosity, ν, only. Any changes in performance due to changes in dimensions with temperature are uncorrected. In both of these cases, the volume and mass flowrates are computed from:

$$Q = (60)(f/K) \text{ GPM} \tag{4-1}$$

$$M = (3600)(\rho)(f/K) \text{ lb/hr} \tag{4-2}$$

where f is the frequency in Hz, K has units ft^3/gallon, and ρ is the liquid density, lb/ft^3. Figure 18 is a sample plot of meter performance on different fluids.

At Reynolds numbers greater than about 15,000, the vortex shedding flowmeter calibration factor K, in units such as gallon/pulse, is essentially constant within 0.5 to 2 percent for flowrates varying from about 10:1 to 100:1, according to the manufacturers' specifications. Pressure losses at the rated flow can vary from a few inches of water to a few psi depending upon the fluid. Pulse output frequencies are relatively low. For example, a particular 2-inch diameter meter in water flowing at a rate of 100 GPM produced an output pulse frequency of about 50 Hz. Thus, for adequate resolution, sampling times of several seconds or more may be needed for frequency measurements, or interval measurement techniques can be used (time to count, say, 10 pulses).

The vortex shedding meter should <u>not</u> be considered a "cure-all" for meter installation troubles encountered in building systems. The same requirements exist for vortex meters as for orifice meters in terms of straightening vanes and straight runs of pipe upstream and downstream. Such conditions are needed to minimize the effects of transverse velocity components and (abnormal) upstream turbulence on the steady formation of vortices behind the meter obstruction body. The meter system design of figure 7 is considered adequate,

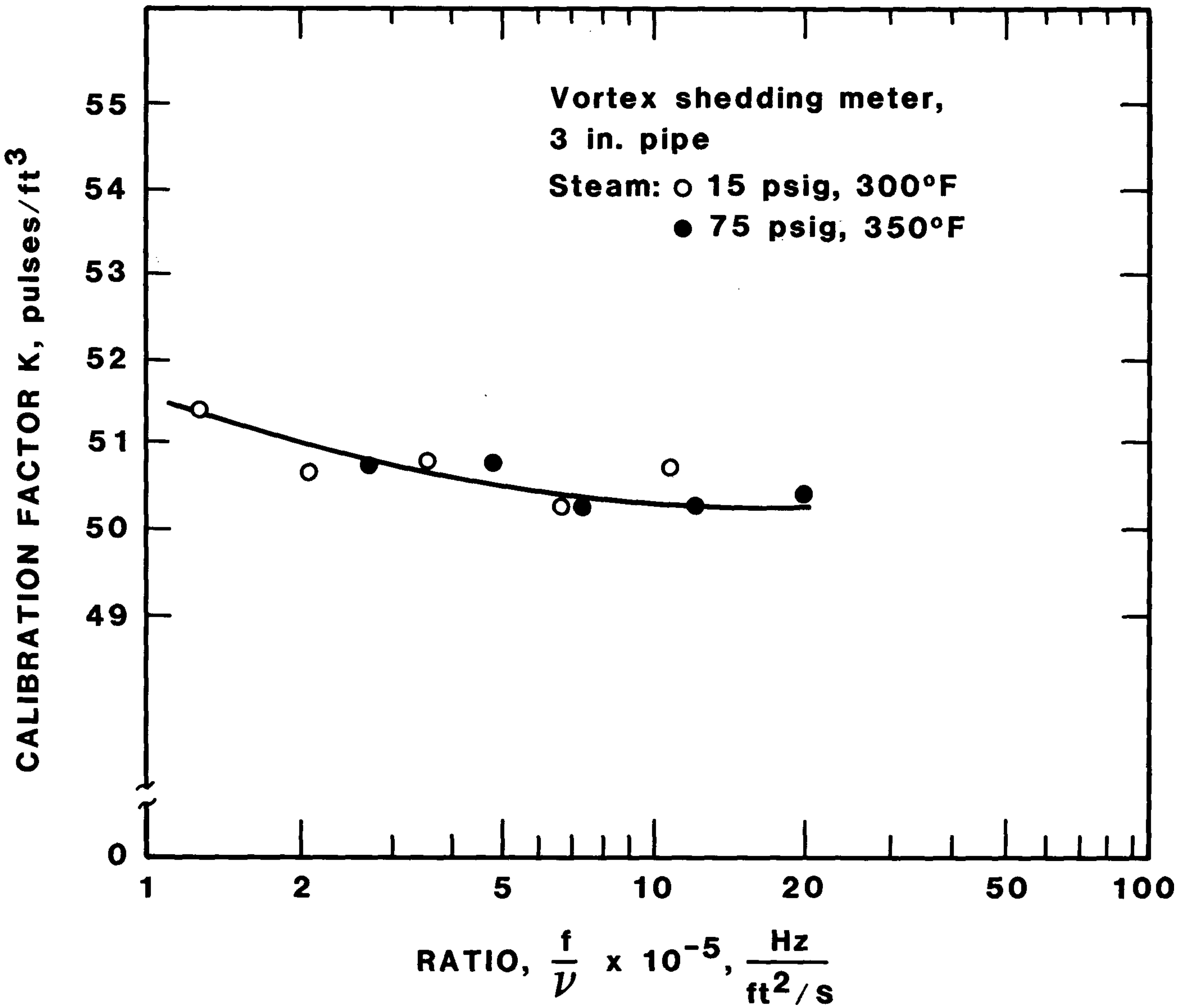


Figure 18. Vortex shedding flowmeter performance

although some manufacturers may recommend longer straight pipe lengths between the meter and the flow straightener.

No effect on the performance of the meter due to the position or orientation of the section of pipe enclosing the meter should be expected. The pressure level should be high enough to ensure that no vapor formation (in the case of liquid flow) occurs in the meter. An absolute pressure, psia, equal to the fluid vapor pressure plus 3 on 4 times the pressure loss at the rated flow should be sufficient. Since this meter element has no moving parts, fluid filtration or straining is usually not required. However, changes in internal dimensions caused by excessive scale build up or abrasion could influence the calibration factor. Internal passages to the vortex sensing transducers must remain clear and open. With each meter design, there will be a minimum fluid flow necessary to create a steady vortex flow.

An example calculation for on-site calibration of a vortex shedding meter is given in appendix E, example E.8.

4.3 TURBINE METER

Figure 19 shows a typical turbine meter. This meter contains a bladed rotor which rotates at a velocity proportional to the volume rate of flow. Most models employ a magnetic pickup, as shown, in which passing rotor blades vary the reluctance of a magnetic circuit, and generate an AC voltage in the pickup coil. The pulse frequency is directly proportional to rotor speed. This signal is sensed as an indication of flow. It can be counted by an electronic counter, or converted to an analog signal using circuits converting frequency to voltage.

The calibration factor is expressed in electrical pulses generated per unit volume of throughput, e.g., pulses/gallon. This factor is sensitive to flowrate, fluid density, and viscosity, the fluid flow pattern at the meter entrance, and sometimes the meter orientation. For a meter of a specified shape, meter performance can be expressed in terms of dimensionless parameters as

$$Q/(n\ D^3) = \phi\ (n\ D^2\ \rho/\mu) = \phi\ (n\ D^2/\nu) \tag{4-5}$$

where Q = volume flowrate
n = speed of rotor
D = a characteristic linear dimension of the meter
ρ = fluid density
μ = dynamic viscosity
ν = kinematic viscosity ($\nu = \mu/\rho$).

The function ϕ, determined by calibration, describes the performance provided retarding forces acting on the rotor (bearing friction and electromagnetic forces) are insignificant and the fluid is incompressible. When considering one particular meter of fixed size and shape, the quantity D is constant and the above dimensionless quantities may be reduced to the form

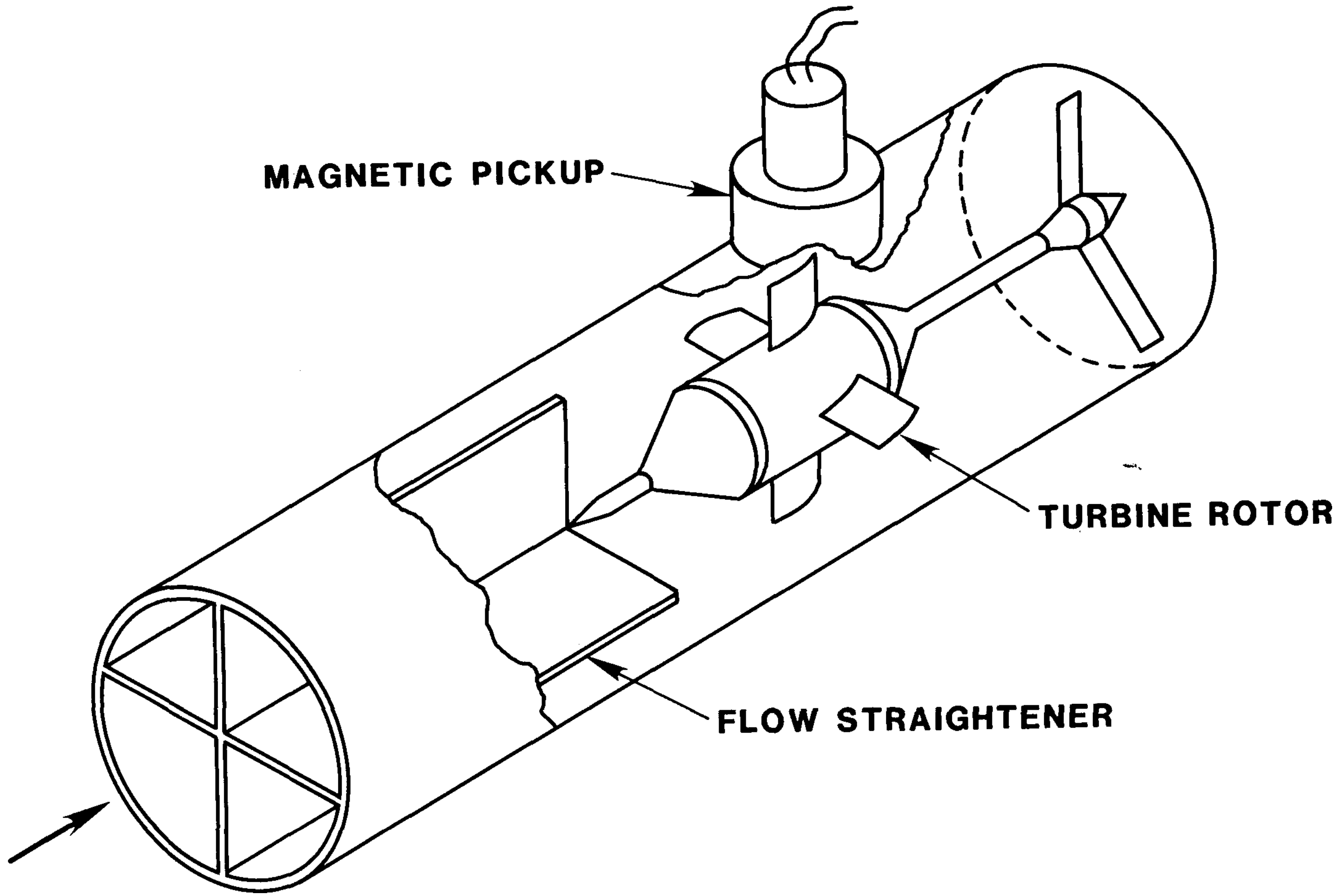


Figure 19. An axial flow turbine meter for liquids, with an electromagnetic pickup coil [2]

$$f/Q = \phi(f/\nu) \tag{4-6}$$

where f is the pulse frequency and f/Q has dimensions pulses/unit volume.

Unfortunately no single performance curve (function ϕ) exists for all turbine meters such as the corresponding coefficient of discharge data which exists, for example, for orifice meters. Therefore, each meter must be calibrated to obtain its performance or function ϕ given above. When used as a transfer reference meter and for best accuracy, each meter should be calibrated using the same liquid to be used during the application. This applies in particular to a liquid such as water which has a different lubricity characteristic as compared to, say, liquid hydrocarbons. Meter performance on water may be different from that obtained using liquid hydrocarbons of similar viscosity as noted in reference 14. The difference is apparently due to changes in retarding forces and bearing friction. Also, the temperature during calibration and application should not differ appreciably since the function ϕ does not account for change in meter dimensions with temperature.

The frequency output of a typical turbine meter can be linear, (i.e., f/Q = constant) within $\pm$ 0.5 percent at its higher rates of flow when operating with low viscosity fluids, such as water. This range of linear operation may extend over a flow range of 10:1 or higher, depending on meter design and size, and on liquid viscosity. For example, with water in the range 40 - 450° F, the kinematic viscosity ν, varies from 1.66×10^{-5} to 0.16×10^{-5} ft^2/s. Thus, the temperature range needs consideration for each application, and f/ν remains the controlling variable when temperature varies significantly. At the lower rates of flow for each meter, frequency output becomes very nonlinear presumably resulting from the combined effects of retarding forces due to electromagnetic and mechanical loading on the turbine. There is always some minimum flowrate below which retarding forces are greater than the fluid forces causing rotation and the rotor ceases to turn. Figure 20 shows performance characteristics of a turbine meter in which the calibration factor K = f/Q is plotted as a function of f/ν. When the meter is calibrated and the operator is using a single fluid at a known temperature, the usual case in building system applications, a convenient graph, as with the vortex shedding meter, is a plot of f/Q vs f. With both plots, the volume flowrate is once again:

$$Q = 60(f/K) \text{ GPM} \tag{4-1}$$

where f is the pulse frequency in Hz and K has units pulses/gallon. Similarly, the mass flowrate is

$$M = 3600(\rho)(f/K) \text{ lb/hr} \tag{4-2}$$

where ρ is the density, lb/gallon.

The pressure drop through the meter depends on its design and varies with ρQ^2. Typically, this may be 5 to 10 psi at the rated flow. The pressure level in the meter should be high enough so that cavitation or formation of vapor does not occur. An entrance pressure of 50 psia plus the vapor pressure of the water is

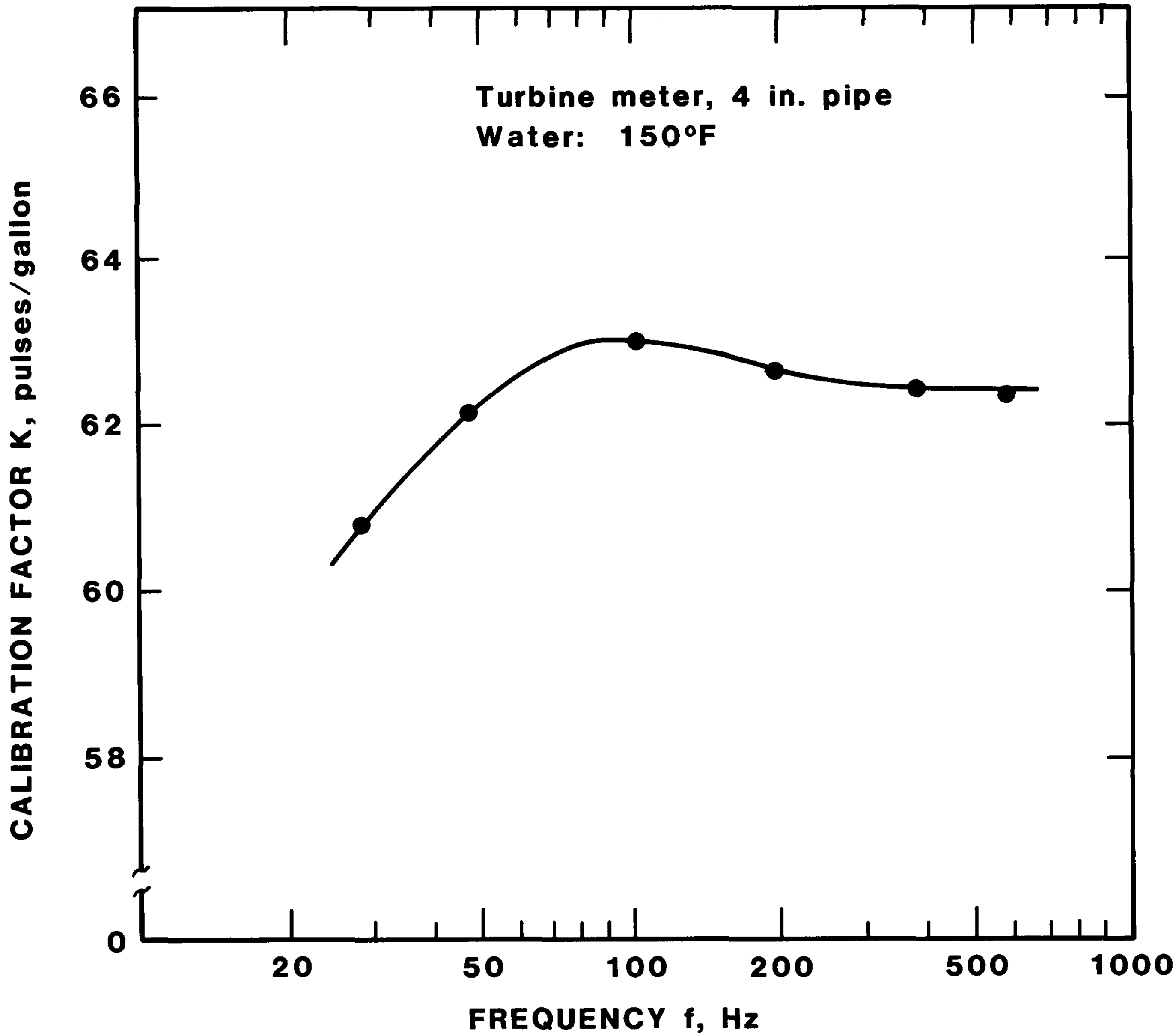


Figure 20. Performance of a turbine meter

usually adequate to overcome pressure losses in the meter and connecting piping, and to prevent cavitation in the meter.

Long term, satisfactory turbine meter operation requires a clean fluid. When dirt or particulate matter are entrained in the fluid, meter performance may be adversely affected due to increased bearing friction. Therefore, a suitable filter upstream is needed. The dirt problem is one of the common difficulties in using turbine meters in the water lines of building systems because these systems easily may become contaminated. The level of contamination tolerable for heating, ventilating and air-conditioning (HVAC) equipment may very likely be intolerable for turbine meters. The required level of filtration depends on both the meter size and the bearing design. For smaller meters such as the one-inch size, filtration to 50 microns is typical. Meter bearing material and design should be compatible with operation in water.

Calibration should be performed periodically and may be performed either on a flowmeter calibration facility or on site using suitable transfer reference flowmeters as shown in figure 5 or 6. In either case, for best accuracy, it is important that the flow straightener, the meter, and the necessary lengths of upstream and downstream piping, as shown in figure 7 for example, be calibrated as a unit. For best accuracy, both the meter and the pickup unit should be installed in the same positions as that used during calibration.

Maintenance procedures should include servicing the filter equipment, including changing filters as necessary to avoid excessive flow blockage and/or pressure loss. Also, the meter should be removed from the line periodically and examined for possible deterioration of its components or for solids deposited on them. Recalibration should be conducted during these maintenance activities.

Severe overspeeding of the meter should be avoided. Such a problem may occur when venting air from a piping system or by subjecting the meter to very high flowrates above the rated value. High rotor speeds when the meter is dry, in particular, can hasten bearing wear and failure. Subjecting the meter to intense pressure or flow pulsations or to mechanical vibration may hasten bearing failure. Exposure to excessive dirt and solid particles may cause bearing seizure.

In summary, the turbine meter has advantages of small size, excellent repeatability, and a digital output. Also, the calibration factor is often essentially constant when the meter is operated within specified ranges of flowrate and viscosity; this fact simplifies flow calculations. Installation as part of a meter system using a flow straightener and calibration in water at the required temperature are essential for best accuracy. However, the amount of dirt often encountered in building systems raises a question as to its general suitability for this application. Clean liquids are required for satisfactory long term operation.

An example calculation for calibration in water is given in appendix E, example E.9.

4.4 TARGET METER

A sketch of a target meter for closed channel flow is shown in figure 21. Other names used for this type meter include vane, force, and drag force. The change in momentum of the fluid as it accelerates past the target causes a drag force which deflects the target. This deflection is measured by a force transducer such as a strain gauge system or linear differential transformer. With incompressible flow past the target, this drag force depends on the size and shape of the target and the proportion of the pipe cross section it occupies, the fluid velocity, and the fluid properties of density and viscosity. The drag coefficient is dimensionless and is defined as:

$$C_d = \frac{2\, g_c\, F}{\rho\, a\, V^2} \tag{4-7}$$

where
F = drag force, lb_f
ρ = fluid density, lb/ft^3
g_c = 32.1740 $(ft \cdot lb)/(lb_f \cdot s^2)$
a = target area, ft^2
V = fluid velocity upstream, ft/s.

While the drag coefficient has been studied for such shapes as the flat plate, the sphere, and the circular cylinder in a free stream, expressing C_d as a function of target Reynolds number, drag coefficients for the various shapes of target meters used in closed channel flow are not well known. Each meter, therefore, needs an individual calibration covering the Reynolds number ranges of interest. Figure 22 is a plot of C_d versus Reynolds number for two target meters, where $R_D = D\, V\, \rho/\mu$.

When C_d is known, the volumetric flowrate Q and the mass flowrate M are calculated from the following equations:

$$Q = C_d \left(\frac{D^2 - d^2}{d}\right)\sqrt{\frac{\pi}{4}\,\frac{2g_c\, F}{\rho}} = C_d \left(\frac{D^2 - d^2}{d}\right)\left(\frac{\pi\, g_c\, F}{2\rho}\right)^{1/2} \tag{4-8}$$

and

$$M = C_d \left(\frac{D^2 - d^2}{d}\right)\sqrt{\frac{\pi}{4}\, 2g_c\, F\, \rho} = C_d \left(\frac{D^2 - d^2}{d}\right)\left(\frac{\pi\, g_c\, F\rho}{2}\right)^{1/2} \tag{4-9}$$

where d = target diameter.

Although the theory for the target meter is "old", these meters have been on the industrial scene only about 25 years. Their principal advantages include their ability to handle fluids with suspended particles, their lack of moving parts, and their absence of pressure sensing lines.

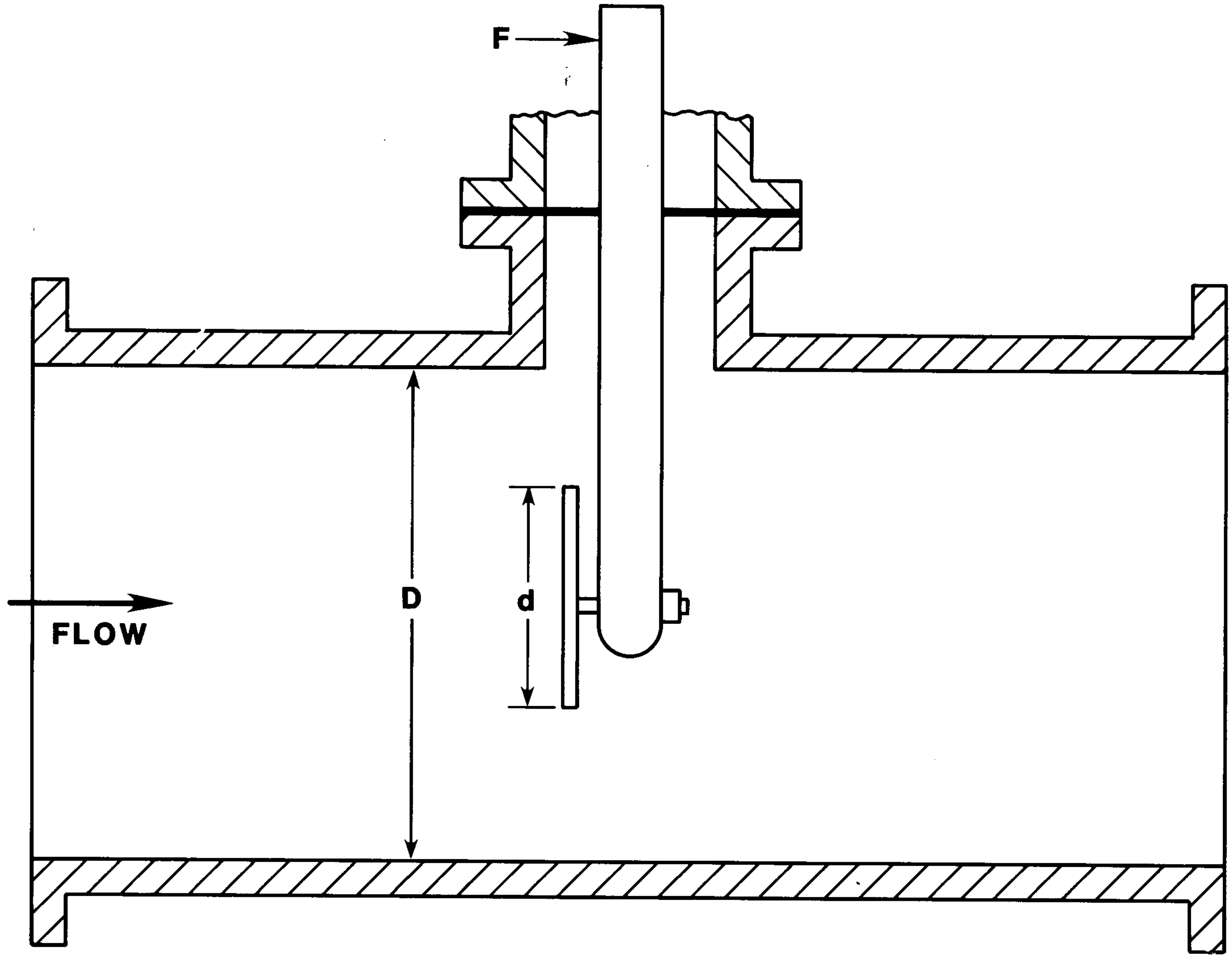
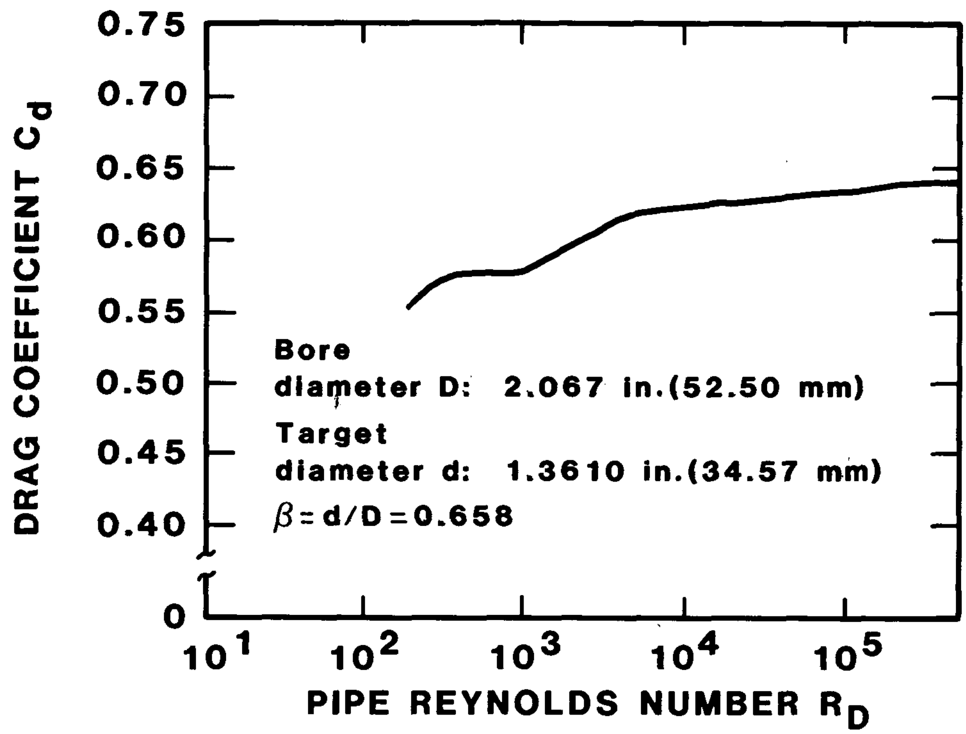


Figure 21. Sketch of a target meter [2]



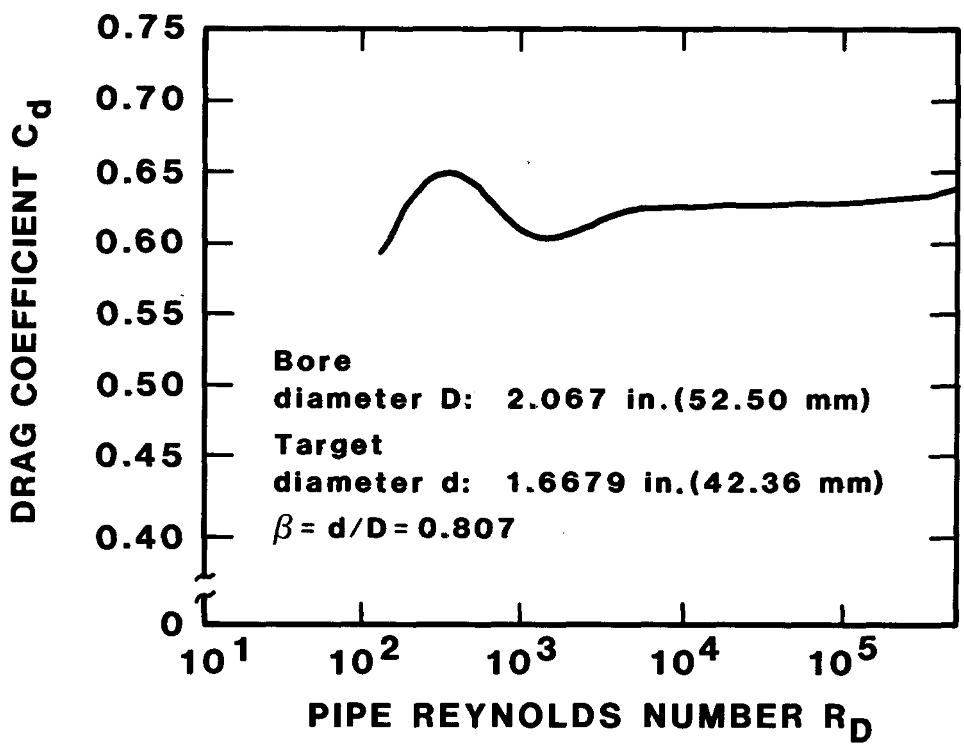


Figure 22. Drag coefficient C_d for two 2-inch target meters, d/D = 0.658 and 0.807 [15]

The output is usually an analog signal, nonlinear with flowrate. Additional electronics are needed to convert it to a digital signal, linearize it with flowrate and/or totalize the flow. Accurate measurement of the meter output is deemed comparable to accurate measurement of pressure or differential pressure, except that a problem in limiting the noise in the electrical lines to acceptable limits may exist.

Since no study has been published to date on the effects of installation conditions, it is recommended that each meter be mounted in a metering section with straight lengths of pipe upstream and downstream together with a flow straightener, as shown in figure 7. Some manufacturers may recommend more than 10D of straight pipe upstream between the meter and the flow straightener.

Target meter systems are available for water in pipe sizes 1/2 inch to 8 inches and larger, flowrates to 2000 GPM and larger, with full scale pressure drops as low as a few tenths of a psi for the larger-size meters. Nominal flow range ratio is 10:1 (transmitter output 100:1). Available transmitter output signals include 0-10 volts DC and 4-20 mA DC. These outputs vary directly with force input as in equations (4-8) and (4-9). Outputs which vary linearly with flowrate, i.e., with $(F)^{1/2}$ are also available. These two outputs are illustrated in figure 23. While the linear Q output is generally more useful, accuracy suffers because of the added electronics.

For best accuracy, the target meter should be calibrated with the fluid to be used on site using the Q^2 transmitter. Thus, for on-site calibration using flowing water at a known temperature, the meter performance can be given through a plot of a calibration factor CF as a function of transmitter output, volts or mA, where

$$CF = GPM/(volts)^{1/2} \text{ or } GPM/(mA)^{1/2} \tag{4-10}$$

Then flowrates can be calculated directly from an expression, for example:

$$Q = (CF)(I - I_o)^{1/2} = [GPM/(mA)^{1/2}](mA)^{1/2} = GPM \tag{4-11}$$

where I_o = current at zero flow
I = current at flowrate Q

The calibration factor CF will then account for variation in C_d with flowrate and for the nonlinear characteristics of the transmitter electronics. Corresponding expressions can be written for mass flowrate M. Example E.10 in appendix E includes sample calculations and a plot of meter performance for the calibration of a target meter in water.

The target meter is attractive for building services because of its ability to handle particulate matter and the absence of pressure sensing lines. For best accuracy, the meter should be calibrated with the fluid to be used. As with the ΔP meter, the output signal varies with Q^2 which may limit its useful operating flow rate as compared to a PD meter. For calculating flowrates, the nonlinearity characteristic can be accomodated by using a calibration factor having the dimensions of flowrate/(output)$^{1/2}$.

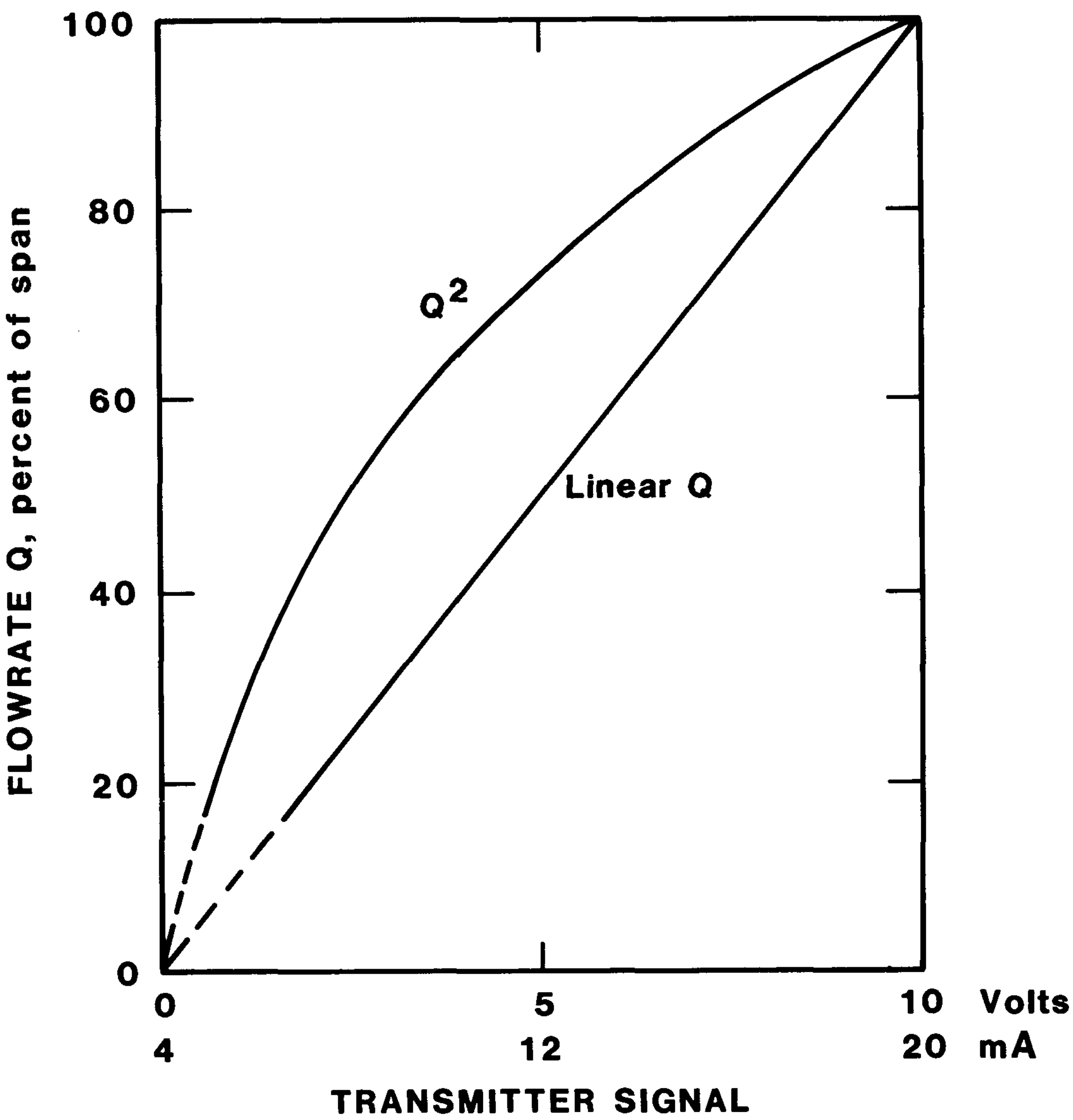


Figure 23. Target meter flowrate outputs for linear Q and nonlinear Q^2 transmitter output systems

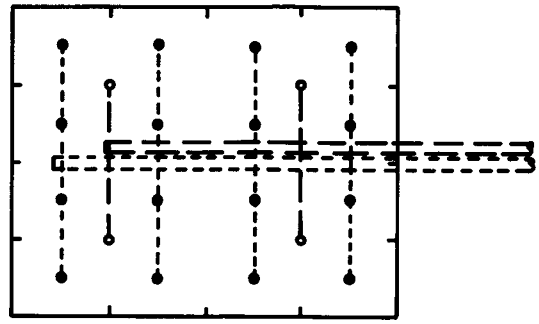
4.5 MULTIPLE PITOT-STATIC AND REVERSE-PITOT TUBE ASSEMBLIES

Several designs of flowmeters of these types are available. Figure 24 shows the schematic diagrams of an assembly designed for mounting in an air duct system. It contains 16 pitot-tubes spaced on an equal area traverse basis and connected to a common impact pressure line, and four static legs connected to a common static pressure line. Since the velocity profile varies in the duct cross section, an average value of impact pressure and a average value of static pressure are obtained. The ΔP is measured by means of a manometer or pressure transducer system. An advantage of this assembly is that a single ΔP measurement is used instead of 16 measurements, or instead of the traversing approach with a single tube. Because of the variation in impact pressure (velocity) from tube to tube and to account for the blockage effect of the assembly, reliable measurement of flowrates require that the assembly be calibrated on site over the range of velocities to be encountered. A flow straightener (honeycomb) is installed upstream of the assembly to eliminate transverse or swirling velocity components.

Figure 25 shows a commercially available reversed type of pitot tube assembly. This consists of a probe which extends completely across the diameter of a pipe and which senses impact pressure at four specific openings along the upstream side of the probe. The resulting impact pressure is some average value which connects to a manometer or ΔP transducer system. A single reversed pressure tube senses the pressure on the downstream side of the probe. Advantages include increased differential pressure for a given fluid velocity compared to a conventional pitot-static tube, and ease of probe installation.

In fully developed turbulent flow, the velocity profile (velocity V as a function of radius r) changes with Reynolds number R_D becoming more uniform as R_D increases; or with a single fluid, as flowrate increases. For a detailed analytical analysis of the influence of the velocity profile on the coefficient of discharge, see reference 16. A significant influence on the coefficient of discharge by an increase in velocity can be expected. Also, when the flow is asymmetrical, such as the case downstream of elbows and valves, significant errors on the impact pressure will result. The static hole is located in the probe wake region; a region of higher turbulence and fluctuating pressures. For all these reasons, it is strongly recommended that the probe be mounted in a metering section such as shown on figure 7 and calibrated on site with the working fluid over the range of flowrates to be encountered. If necessary, to aid in producing a symmetrical velocity profile, the flow straightener should be of the perforated plate design rather than the tubular design (at the expense of a larger pressure loss). (See figures 12 and 13.)

The reversed pitot-tube sensing probes are suitable for flowing air, water or steam. However, flowing hot water at pressure levels approaching the vapor pressure should be avoided to eliminate possible cavitation at the static hole. Building system pressures of 50 to 100 psig should present no problem. To illustrate this point, consider the static pressure behind a circular cylinder where the static pressure has an approximate value corresponding to $C_p = -1.3$. The dimensionless pressure coefficient C_p is defined by the following relationship:



- Velocity taps at centers of 16 equal areas
- Static pressure taps at centers of 4 equal areas

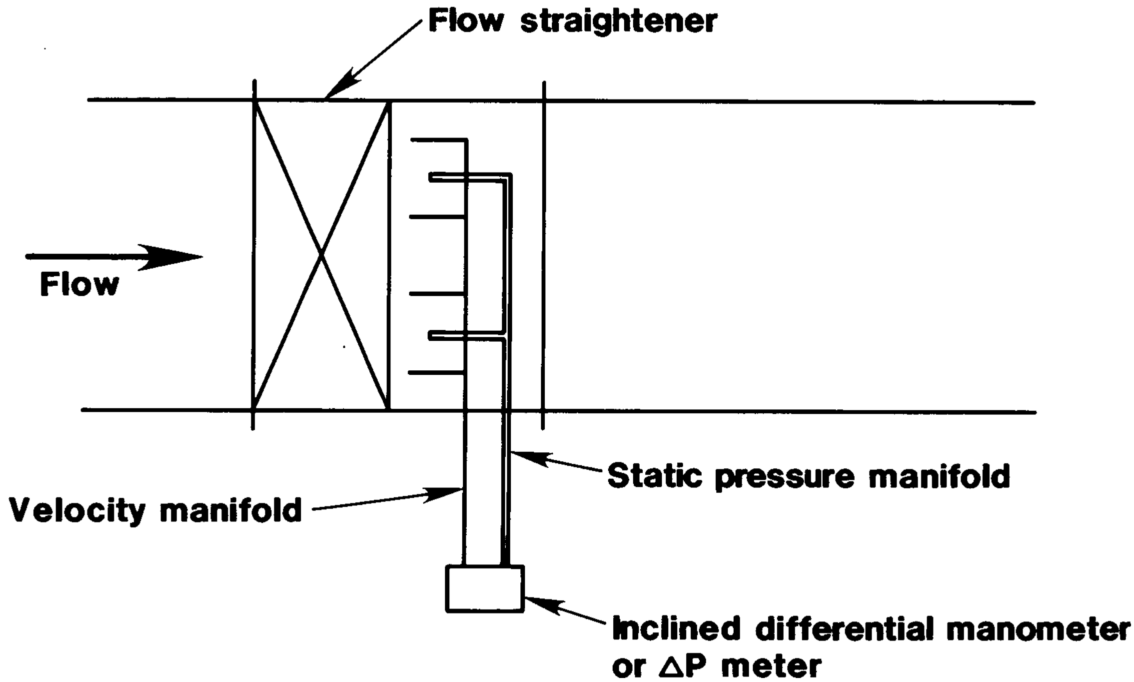


Figure 24. Pitot-static rake assembly for measuring air flow in a duct. The ends of the velocity taps are open to receive the impact pressure. The ends of the static pressure taps are closed and a ring of radial perforations receive the static pressure

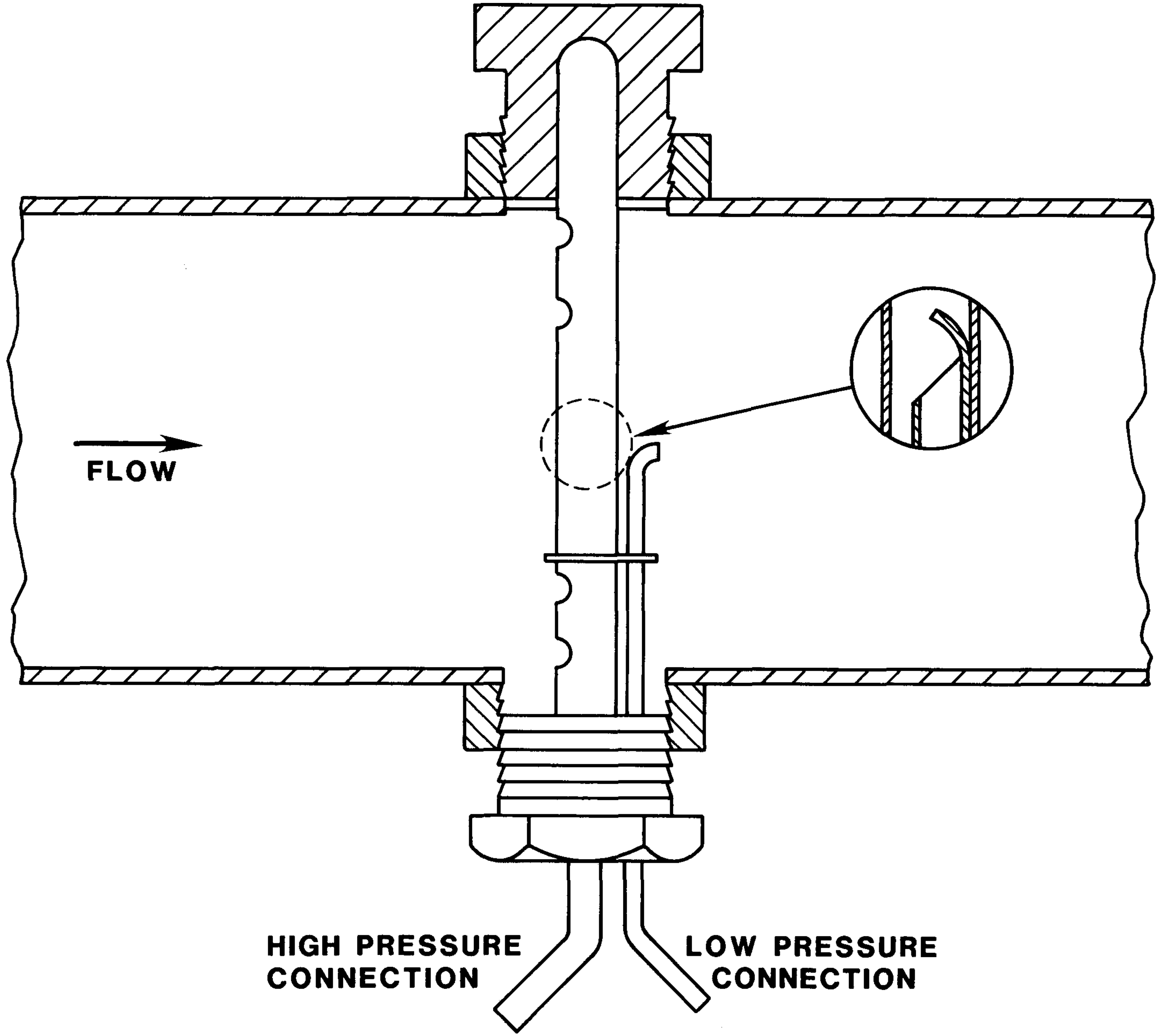


Figure 25. Reversed pitot-tube assembly [2]

$$C_p = \frac{2g_c\ (P - P_o)}{\rho\ V^2}\ . \qquad (4\text{-}12)$$

For example, with water flowing at V = 20 ft/s

$$P - P_o = (C_p\ \rho\ V^2)/(2\ g_c)$$

$$= \frac{(-1.3)(62.4\ \text{lb/ft}^3)(20\ \text{ft/s})^2}{(2)(32.174)[(\text{lb}\cdot\text{ft})/(\text{s}^2\cdot\text{lb}_f)](144\ \text{in.}^2/\text{ft}^2)}\text{, or}$$

$$P - P_o = -3.5\ \text{psi}$$

This means the pressure P at the static hole is 3.5 psi less than system pressure P_o. Thus, if the water temperature is 212°F, system pressure P_o must be at least 14.7 + 3.5 = 18.2 psia (or 3.5 psig) to avoid flashing of the water at the static hole. A pressure P_o of 7 to 10 psig would probably provide an adequate degree of safety from flashing.

The pressure differential, ΔP, for both of these probe systems is nonlinear with flowrate, where $\Delta P \propto Q^2$ or $\Delta P \propto M^2$. When used with a ΔP transmitter where the current or voltage output varies directly with ΔP, or when used with a manometer system, a calibration factor of the following form must be used:

$$CF = \text{flowrate}/(\text{output})^{1/2}\ .$$

This will allow direct calculation of flowrate Q (or M) from an expression such as

$$Q = (CF)(I - I_o)^{1/2}\ \text{or} \qquad (4\text{-}13)$$

$$Q = (CF)(\text{inches of manometer fluid})^{1/2} \qquad (4\text{-}14)$$

Refer to figure 23 and the discussion in section 4.4 which discusses a similar situation for the target meter. In these mutliple assemblies, with an on-line calibration, the calibration factor CF will then account for any effects of changing velocity profiles on the probe performance.

Multiple pressure probe assemblies offer the advantage of a single ΔP output resulting from sensing impact pressure (velocity) and static pressure at multiple points. For reliable flow measurement, calibration is required. For a large flow range such as 10:1, HIGH and LOW ΔP transmitter systems such as discussed in section 3.3 may be necessary. The nonlinearity in the ΔP can be accommodated by using a calibration factor of the form: flowrate/(output)$^{1/2}$. Electronic noise from a ΔP transducer generated by small changes in the velocity profiles can also be a problem which is usually solved with little effort.

4.6 ULTRASONIC FLOWMETER

Ultrasonic flowmeters can be readily attached to the outside of existing liquid-filled pipes without shutdown, diversion, special sections, or isolation valves. The economic advantage of ultrasonic flowmeters over conventional flowmeters increases with pipe size.

The primary elements are nonintrusive and are sensitive over a wide range of velocities of the liquid flowing within the pipe without inducing a pressure drop or other disturbance to the medium being monitored.

Several ultrasonic techniques are used in the measurement of flow. The majority of instruments of this type use various configurations of acoustic sources (transmitters) and detectors (receivers). The velocity of the fluid flowing in the pipe is detected from the effect of the moving liquid on the sound waves. In general, this effect is determined by comparing the signals transmitted with those received. The two techniques most commonly used to detect the effect of the flowing liquid on the sound waves are known as the transit-time meters and the Doppler meters.

The transit-time flowmeters utilize the fact that fluid flow along the acoustic path affects the time it takes for the acoustical signal to travel from the transmitter to the receiver. Transit-time meters transmit bursts of ultrasonic energy across the pipe as shown in figure 26. The transit times in the downstream (t+) and upstream (t-) directions are given by:

$$t+ = \frac{\text{distance}}{\text{conduction velocity}} = \frac{L}{c + \hat{u}\cos\theta}\text{, and, } t- = \frac{L}{c - \hat{u}\cos\theta}$$

where t = time in seconds
L = distance between the two transceivers
c = velocity of sound in the liquid being monitored
$\hat{u}$ = velocity of the liquid averaged along the path of the ultrasound. $\hat{u} \cong 1.33\,\bar{u}$ for laminar flow, and, for turbulent flow, $\hat{u} \cong 1.07\,\bar{u}$ where $\bar{u}$ = the average velocity of the flow over the cross-sectional area.
θ = angle between the direction of travel of the acoustical signals and direction of fluid flow

Note that $\hat{u}$ differs from $\bar{u}$ because the ultrasonic path is along a single line rather than averaged over the cross-sectional area.

The downstream transit time is shorter than the upstream transit time by

$$\Delta t \cong \frac{2L\,\hat{u}\cos\theta}{c^2}$$

and thus the average fluid velocity along the acoustical path, $\hat{u}$, is proportional to Δt. A short acoustic pulse is transmitted alternately in the upstream and downstream directions. Unfortunately, the resulting Δt is in the nanosecond range and higher quality electronics are required to achieve adequate stability. It should be noted that this type of meter requires relatively clean liquids since minute particles can absorb or disperse sound energy.

Ultrasonic flowmeter utilizing the Doppler effect function on the shift in frequency produced when the sound waves are reflected from moving bodies. In this type of ultrasonic flowmeter, a single transmitter projects a continuous ultrasonic beam at about 0.5 MHz through the pipe wall to the liquid flow being

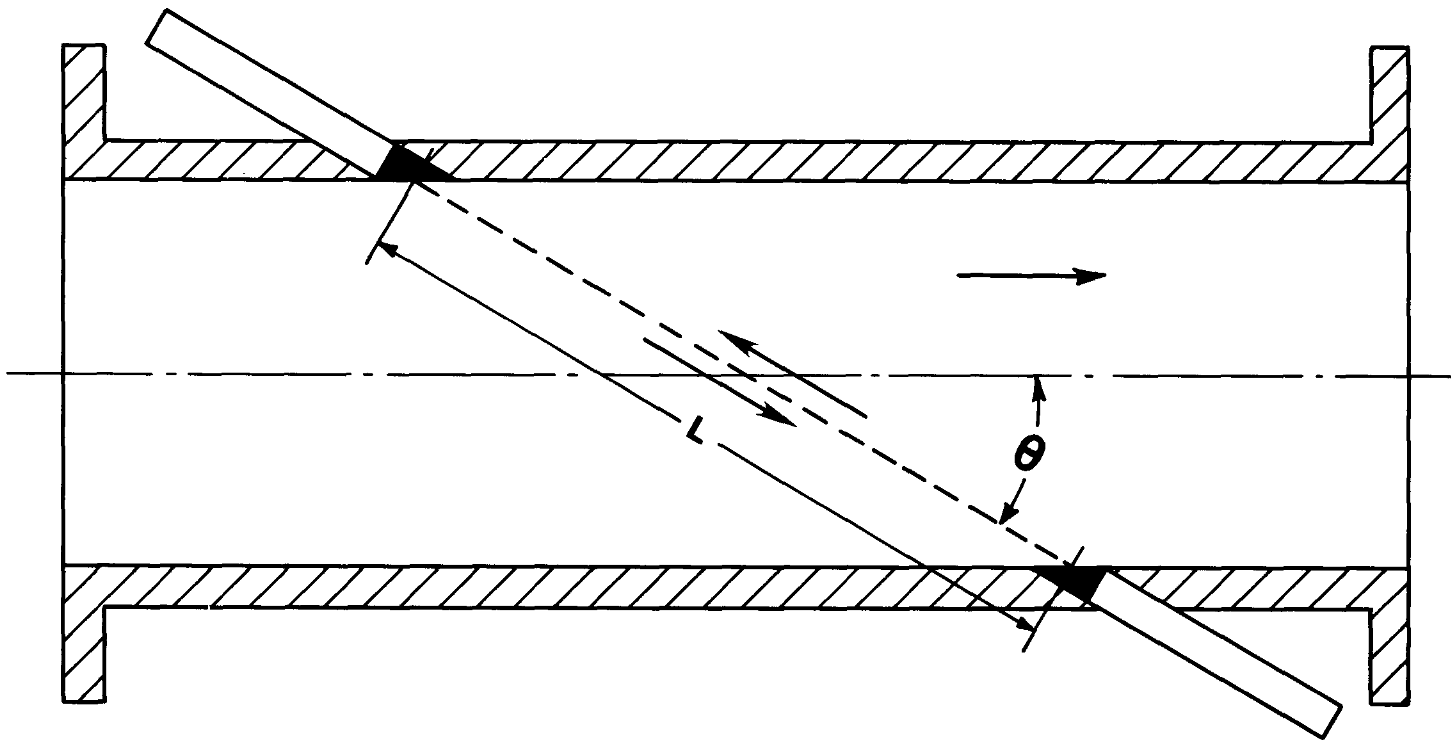


Figure 26. Single path, transit-time ultrasonic flowmeter

monitored. The particles carried by the liquid have velocity components in the direction of the ultrasonic beam since the beam makes an angle which is less than 90 ° to the direction of flow. The reflected ultrasonic waves detected at the receiver (which can be located adjacent to the transmitter or on the opposite side of the pipe) are shifted in frequency in proportion to the velocity of the liquid. The shift in frequency is determined and used to measure the flow rate. Doppler meters are suitable for liquids containing a consistent density of acoustically reflective particles.

Since the transmit-time meters require relatively clean liquids and the Doppler type meters require relatively small acoustically reflective particles, and these conditions are rarely met in typical EMCS applications, frequent periodic calibrations of these meters are required.

The vast differences in the designs of currently available ultrasonic flowmeters requires that the reader consult a particular manufacturer's literature for further information on calibration procedures [17,18,19].

4.7 INSERTION TYPE TURBINE METER

This is another type of turbine meter in which a meter assembly is mounted on the end of a stem or strut and positioned as desired in a large pipe, duct, or open channel. Figure 27 shows a typical installation sketch. Advantages include low cost compared to the "full bore" turbine meter for large pipe installations; easy installation including no disruption to the fluid flow when an isolation valve is used, insuring essentially no pressure loss; and a digital output. Other characteristics, including the influence of fluid properties, density and viscosity, fluid lubricity, the effects of retarding forces due to bearing friction and electromagnetic loading, and the effects of flow patterns have all been discussed above for full bore turbine meters (section 4.3). These characteristics apply generally to the insertion meter. Again, a clean fluid is imperative for satisfactory, long-term operation.

Insertion turbine meters are available for use in water, air, and steam. Typical velocities range from 50 or 60 feet/second maximum for liquids to 300 feet/second maximum for gases. Manufacturers typically claim a linearity of $\pm$ 1 percent full scale for 10:1 flow range.

The insertion turbine meter, like the target meter, senses local fluid velocity only and thus, for measuring flow in closed conduits (pipes and ducts), the performance is dependent on the velocity profile. Figure 28 shows velocity profiles for fully developed turbulent flow as a function of Reynolds number R_D. Such profiles normally occur only in very long, straight pipes. Furthermore, it should be noted that the conversion factor between measured velocity and flowrate will be flow dependent. An iteration is recommended for each measurement. Note also that velocity gradients are lowest near the pipe center, indicating a preferred probe position.

The insertion type turbine meter is attractive because of its relatively low cost for large pipe sizes and its ease of installation. For best accuracy, the meter should be calibrated on site with the operating fluid over the operating flow and temperature ranges.

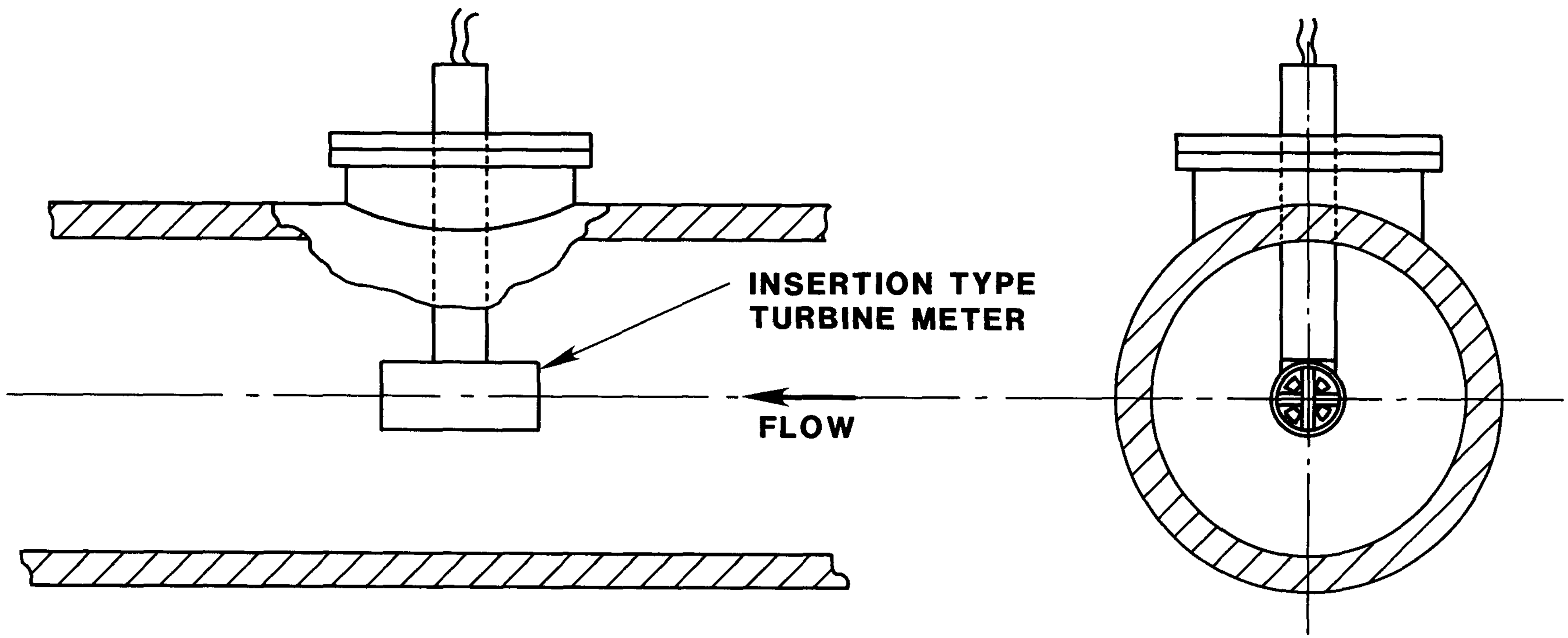


Figure 27. Insertion type turbine meter. Typical installation.

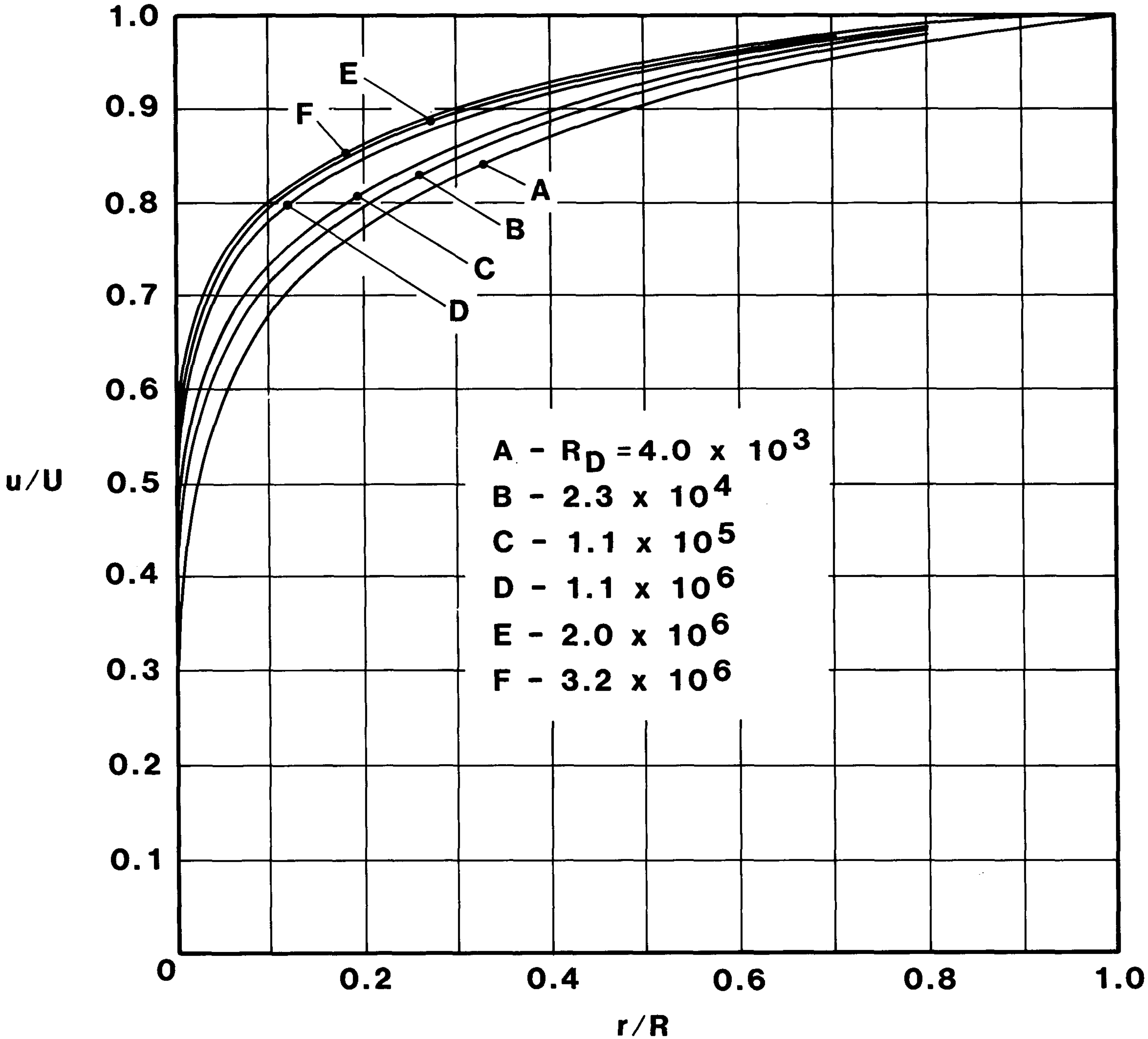


Figure 28. Velocity profiles for fully developed turbulent flow, where R is pipe radius, r is radial position, U is the center line velocity, and u is the axial velocity at radius r. The distributions are normalized [21]

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ACKNOWLEDGMENTS

Discussions with Mr. Kenneth Benson of the Fluid Engineering Division at NBS on flowmeter characteristics and calibration techniques have been especially helpful. The authors also wish to acknowledge the excellent work of Dr. James F. Schooley of the NBS Center for Absolute Physical Quantities in editing this report and contributing many technical improvements. Also, long past discussions on similar subjects with Mr. Montgomery R. Shafer Jr. (retired, deceased) of the former Fluid Meters Section at NBS are recalled. Mr. Shafer's very significant contributions through his work on flowmeter characteristics and calibration techniques are remembered by many, and his paper (reference 14) on turbine meter characteristics is still a classic.

APPENDIX A

Coefficient of Discharge C and Fluid Expansion Factor Y for Orifices, Flow Nozzles and Venturi Meters

(The data in this appendix were taken from references 2, 10, or 11)

Table A.1. Coefficient of Discharge C

Orifice Plate with Corner Taps

R_D / β	5×10^3	10^4	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.23	0.6012	0.5997	0.5987	0.5983	0.5980	0.5979	0.5977	0.5975	0.5974	0.5973
0.24	0.6018	0.6000	0.5990	0.5986	0.5982	0.5981	0.9579	0.5977	0.5975	0.9575
0.26	0.6031	0.6009	0.5996	0.5991	0.5987	0.5985	0.5983	0.5980	0.5978	0.5978
0.28	0.6044	0.6019	0.6003	0.5997	0.5992	0.5989	0.5987	0.5983	0.5982	0.5981
0.30	0.6060	0.6029	0.6011	0.6004	0.5997	0.5994	0.5992	0.5987	0.5983	0.5984
0.32	0.6077	0.6040	0.6019	0.6011	0.6003	0.6000	0.5997	0.5991	0.5989	0.5988
0.34	0.6095	0.6053	0.6028	0.6018	0.6010	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	0.6115	0.6066	0.6037	0.6026	0.6016	0.6012	0.6008	0.6001	0.5997	0.5995
0.38	0.6136	0.6081	0.6048	0.6035	0.6024	0.6018	0.6014	0.6005	0.6002	0.6000
0.40	0.6159	0.6096	0.6059	0.6044	0.6031	0.6025	0.6020	0.6011	0.6006	0.6004
0.42	0.6184	0.6113	0.6070	0.6054	0.6039	0.6032	0.6026	0.6016	0.6011	0.6008
0.44	0.6210	0.6130	0.6082	0.6064	0.6047	0.6039	0.6033	0.6021	0.6016	0.6013
0.46	0.6238	0.6148	0.6095	0.6074	0.6056	0.6047	0.6040	0.6027	0.6021	0.6017
0.48	--	0.6167	0.6108	0.6085	0.6064	0.6055	0.6047	0.6032	0.6025	0.6021
0.50	--	0.6187	0.6121	0.6096	0.6073	0.6062	0.6053	0.6037	0.6030	0.6026
0.52	--	0.6027	0.6135	0.6107	0.6082	0.6070	0.6060	0.6042	0.6034	0.6029
0.54	--	0.6228	0.6148	0.6117	0.6090	0.6077	0.6066	0.6047	0.6037	0.6032
0.56	--	0.6249	0.6162	0.6128	0.6098	0.6084	0.6072	0.6050	0.6040	0.6035
0.58	--	0.6270	0.6175	0.6138	0.6105	0.6089	0.6077	0.6053	0.6042	0.6036
0.60	--	0.6291	0.6187	0.6147	0.6111	0.6094	0.6080	0.6055	0.6043	0.6036
0.62	--	0.6311	0.6198	0.6155	0.6116	0.6098	0.6083	0.6055	0.6042	0.6035
0.64	--	0.6330	0.6208	0.6161	0.6119	0.6099	0.6083	0.6053	0.6039	0.6031
0.65	--	0.6339	0.6212	0.6164	0.6120	0.6099	0.6082	0.6051	0.6037	0.6028
0.66	--	0.6348	0.6216	0.6165	0.6120	0.6099	0.6081	0.6048	0.6033	0.6025
0.67	--	0.6356	0.6219	0.6167	0.6120	0.6097	0.6079	0.6045	0.6029	0.6021
0.68	--	0.6363	0.6222	0.6167	0.6118	0.6095	0.6076	0.6041	0.6025	0.6016
0.69	--	0.6370	0.6223	0.6167	0.6116	0.6092	0.6072	0.6036	0.6019	0.6010
0.70	--	0.6376	0.6224	0.6165	0.6113	0.6088	0.6067	0.6030	0.6012	0.6003
0.71	--	0.6382	0.6224	0.6163	0.6109	0.6083	0.6061	0.6023	0.6004	0.5994
0.72	--	0.6386	0.6222	0.6160	0.6103	0.6076	0.6054	0.6014	0.5995	0.5985
0.73	--	0.6389	0.6220	0.6155	0.6097	0.6069	0.6046	0.6004	0.5985	0.5974
0.74	--	0.6391	0.6216	0.6149	0.6089	0.6060	0.6036	0.5993	0.5973	0.5962
0.75	--	0.6392	0.6211	0.6141	0.6079	0.6049	0.6025	0.5980	0.5959	0.5948
0.76	--	0.6391	0.6204	0.6132	0.6068	0.6037	0.6012	0.5966	0.5944	0.5932
0.77	--	0.6385	0.6196	0.6121	0.6055	0.6023	0.5997	0.5949	0.5927	0.5915
0.78	--	--	0.6185	0.6108	0.6039	0.6007	0.5980	0.5931	0.5908	0.5895
0.79	--	--	0.6173	0.6093	0.6022	0.5988	0.5960	0.5910	0.5886	0.5873
0.80	--	--	0.6158	0.6076	0.6003	0.5968	0.5939	0.5887	0.5862	0.5849

Table A.2. Coefficient of Discharge C

Orifice Plate with Taps at 1D and 1/2D

β \ R_D	5×10^3	10^4	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5997	0.5985	0.5979	0.5976	0.5974	0.5973	0.5972	0.5970	0.5969	0.5969
0.22	0.6006	0.5992	0.5984	0.5980	0.5977	0.5976	0.5975	0.5973	0.5972	0.5971
0.24	0.6017	0.6000	0.5989	0.5985	0.5981	0.5980	0.5978	0.5976	0.5974	0.5974
0.26	0.6030	0.6008	0.5995	0.5990	0.5986	0.5984	0.5982	0.5979	0.5977	0.5977
0.28	0.6043	0.6017	0.6002	0.5996	0.5991	0.5988	0.5986	0.5932	0.5981	0.5980
0.30	--	0.6028	0.6010	0.6003	0.5996	0.5993	0.5991	0.5986	0.5984	0.5983
0.32	--	0.6039	0.6018	0.6010	0.6002	0.5999	0.5996	0.5990	0.5988	0.5987
0.34	--	0.6052	0.6027	0.6017	0.6009	0.6004	0.6001	0.5995	0.5992	0.5990
0.36	--	0.6066	0.6037	0.6026	0.6016	0.6011	0.6007	0.6000	0.5997	0.5995
0.38	--	0.6080	0.6047	0.6035	0.6023	0.6018	0.6013	0.6005	0.6001	0.5999
0.40	--	0.6096	0.6059	0.6044	0.6031	0.6025	0.6020	0.6011	0.6006	0.6004
0.42	--	--	0.6071	0.6054	0.6040	0.6033	0.6027	0.6017	0.6012	0.6009
0.44	--	--	0.6084	0.6065	0.6049	0.6041	0.6035	0.6023	0.6017	0.6014
0.46	--	--	0.6098	0.6077	0.6059	0.6050	0.6043	0.6030	0.6023	0.6020
0.48	--	--	0.6112	0.6089	0.6069	0.6059	0.6051	0.6036	0.6030	0.6026
0.50	--	--	0.6127	0.6102	0.6079	0.6068	0.6060	0.6043	0.6036	0.6032
0.52	--	--	0.6143	0.6115	0.6090	0.6078	0.6068	0.6051	0.6042	0.6038
0.54	--	--	0.6159	0.6129	0.6101	0.6088	0.6077	0.6058	0.6049	0.6044
0.56	--	--	0.6176	0.6143	0.6113	0.6098	0.6087	0.6065	0.6055	0.6049
0.58	--	--	--	0.6157	0.6124	0.6108	0.6095	0.6072	0.6061	0.6055
0.60	--	--	--	0.6171	0.6135	0.6118	0.6104	0.6079	0.6067	0.6060
0.62	--	--	--	0.6185	0.6146	0.6128	0.6112	0.6085	0.6072	0.6065
0.64	--	--	--	0.6198	0.6156	0.6136	0.6120	0.6090	0.6076	0.6068
0.66	--	--	--	0.6211	0.6166	0.6144	0.6127	0.6094	0.6079	0.6071
0.68	--	--	--	0.6223	0.6175	0.6151	0.6132	0.6097	0.6081	0.6072
0.70	--	--	--	--	0.6182	0.6157	0.6136	0.6099	0.6081	0.6071
0.71	--	--	--	--	0.6185	0.6159	0.6138	0.6099	0.6081	0.6071
0.72	--	--	--	--	0.6187	0.6161	0.6139	0.6098	0.6080	0.6069
0.73	--	--	--	--	0.6190	0.6162	0.6139	0.6097	0.6078	0.6067
0.74	--	--	--	--	0.6191	0.6163	0.6139	0.6096	0.6076	0.6065
0.75	--	--	--	--	0.6193	0.6163	0.6138	0.6094	0.6073	0.6062

Table A.3. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 2 in.

β \ R_D	5×10^3	10^4	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.25	0.6023	0.6003	0.5992	0.5987	0.5983	0.5981	0.5980	0.5977	0.5976	0.5975
0.26	0.6029	0.6008	0.5995	0.5990	0.5986	0.5984	0.5982	0.5979	0.5977	0.5976
0.28	0.6043	0.6017	0.6002	0.5996	0.5990	0.5988	0.5986	0.5982	0.5980	0.5979
0.30	--	0.6028	0.6009	0.6002	0.5996	0.5993	0.5990	0.5986	0.5984	0.5983
0.32	--	0.6039	0.6017	0.6009	0.6002	0.5998	0.5995	0.5990	0.5988	0.5986
0.34	--	0.6051	0.6026	0.6017	0.6008	0.6004	0.6001	0.5994	0.5992	0.5990
0.36	--	0.6065	0.6038	0.6025	0.6015	0.6010	0.6006	0.5999	0.5996	0.5994
0.38	--	0.6080	0.6047	0.6034	0.6022	0.6017	0.6013	0.6004	0.6001	0.5998
0.40	--	--	0.6058	0.6043	0.6030	0.6024	0.6019	0.6010	0.6006	0.6003
0.42	--	--	0.6070	0.6054	0.6039	0.6032	0.6026	0.6016	0.6011	0.6008
0.44	--	--	0.6083	0.6064	0.6048	0.6040	0.6034	0.6022	0.6016	0.6013
0.46	--	--	0.6096	0.6076	0.6057	0.6049	0.6041	0.6028	0.6022	0.6019
0.48	--	--	0.6111	0.6088	0.6067	0.6058	0.6050	0.6035	0.6028	0.6024
0.50	--	--	0.6126	0.6100	0.6078	0.6067	0.6058	0.6042	0.6034	0.6030
0.52	--	--	0.6141	0.6113	0.6088	0.6076	0.6067	0.5049	0.6041	0.6036
0.54	--	--	0.6157	0.6127	0.6099	0.6086	0.6075	0.6056	0.6047	0.6042
0.56	--	--	0.6174	0.6140	0.6110	0.6096	0.6084	0.6063	0.6053	0.6047
0.58	--	--	--	0.6154	0.6121	0.6106	0.6093	0.6070	0.6059	0.6053
0.60	--	--	--	0.6168	0.6132	0.6115	0.6101	0.6076	0.6064	0.6057
0.62	--	--	--	0.6182	0.6143	0.6124	0.6109	0.6082	0.6069	0.6062
0.64	--	--	--	0.6195	0.6153	0.6133	0.6117	0.6087	0.6073	0.6065
0.65	--	--	--	0.6201	0.6158	0.6137	0.6120	0.6089	0.6074	0.6066
0.66	--	--	--	0.6208	0.6162	0.6141	0.6123	0.6091	0.6076	0.6067
0.67	--	--	--	0.6214	0.6167	0.6144	0.6126	0.6092	0.6076	0.6068
0.68	--	--	--	0.6219	0.6171	0.6147	0.6128	0.6093	0.6077	0.6068
0.69	--	--	--	0.6225	0.6174	0.6150	0.6130	0.6094	0.6077	0.6068
0.70	--	--	--	--	0.6177	0.6152	0.6132	0.6094	0.6077	0.6067
0.71	--	--	--	--	0.6180	0.6154	0.6133	0.6094	0.6076	0.6066
0.72	--	--	--	--	0.6183	0.6156	0.6134	0.6094	0.6075	0.6064
0.73	--	--	--	--	0.6185	0.6157	0.6134	0.6092	0.6073	0.6062
0.74	--	--	--	--	0.6186	0.6157	0.6134	0.6091	0.6071	0.6059
0.75	--	--	--	--	0.6187	0.6157	0.6133	0.6088	0.6068	0.6056

Table A.4. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 3 in.

β \ R_D	5×10^3	10^4	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5997	0.5986	0.5979	0.5976	0.5974	0.5973	0.5972	0.5970	0.5970	0.5969
0.22	0.6006	0.5992	0.5984	0.5981	0.5978	0.5976	0.5975	0.5973	0.5972	0.5972
0.24	--	0.6000	0.5989	0.5985	0.5982	0.5980	0.5979	0.5976	0.5975	0.5974
0.26	--	0.6008	0.5996	0.5991	0.5986	0.5984	0.5982	0.5979	0.5978	0.5977
0.28	--	0.6018	0.6002	0.5997	0.5991	0.5989	0.5987	0.5983	0.5981	0.5980
0.30	--	0.6028	0.6010	0.6003	0.5997	0.5994	0.5991	0.5987	0.5985	0.5983
0.32	--	0.6040	0.6018	0.6010	0.6003	0.5999	0.5996	0.5991	0.5988	0.5987
0.34	--	--	0.6027	0.6018	0.6009	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	--	--	0.6037	0.6026	0.6016	0.6011	0.6008	0.6000	0.5997	0.5995
0.38	--	--	0.6048	0.6035	0.6024	0.6018	0.6014	0.6006	0.6002	0.6000
0.40	--	--	0.6059	0.6045	0.6032	0.6026	0.6021	0.6011	0.6007	0.6005
0.42	--	--	0.6071	0.6055	0.6040	0.6033	0.6028	0.6017	0.6012	0.6010
0.44	--	--	0.6084	0.6066	0.6049	0.6042	0.6035	0.6023	0.6018	0.6015
0.46	--	--	0.6098	0.6077	0.6059	0.6050	0.6043	0.6030	0.6024	0.6020
0.48	--	--	--	0.6089	0.6069	0.6059	0.6051	0.6036	0.6030	0.6026
0.50	--	--	--	0.6102	0.6079	0.6068	0.6059	0.6043	0.6036	0.6032
0.52	--	--	--	0.6115	0.6090	0.6078	0.6068	0.6050	0.6042	0.6037
0.54	--	--	--	0.6128	0.6100	0.6087	0.6077	0.6057	0.6048	0.6043
0.56	--	--	--	0.6141	0.6111	0.6097	0.6085	0.6064	0.6054	0.6048
0.58	--	--	--	--	0.6122	0.6106	0.6093	0.6070	0.6059	0.6053
0.60	--	--	--	--	0.6132	0.6115	0.6101	0.6076	0.6064	0.6057
0.62	--	--	--	--	0.6142	0.6123	0.6108	0.6080	0.6068	0.6060
0.64	--	--	--	--	0.6151	0.6131	0.6114	0.6084	0.6070	0.6063
0.65	--	--	--	--	0.6155	0.6134	0.6117	0.6086	0.6071	0.6063
0.66	--	--	--	--	0.6159	0.6137	0.6119	0.6087	0.6072	0.6064
0.67	--	--	--	--	0.6162	0.6140	0.6121	0.6088	0.6072	0.6063
0.68	--	--	--	--	0.6165	0.6142	0.6123	0.6088	0.6072	0.6063
0.69	--	--	--	--	0.6168	0.6144	0.6124	0.6088	0.6071	0.6061
0.70	--	--	--	--	0.6170	0.6145	0.6124	0.6087	0.6069	0.6060
0.71	--	--	--	--	0.6172	0.6146	0.6124	0.6086	0.6067	0.6057
0.72	--	--	--	--	0.6173	0.6146	0.6124	0.6084	0.6065	0.6054
0.73	--	--	--	--	--	0.6145	0.6122	0.6081	0.6061	0.6051
0.74	--	--	--	--	--	0.6144	0.6120	0.6077	0.6057	0.6046
0.75	--	--	--	--	--	0.6142	0.6118	0.6073	0.6052	0.6041

Table A.5. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 4 in.

β \ R_D	10^4	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5986	0.5979	0.5976	0.5974	0.5973	0.5972	0.5971	0.5970	0.5969
0.22	0.5992	0.5984	0.5981	0.5978	0.5976	0.5975	0.5973	0.5972	0.5972
0.24	0.6000	0.5990	0.5985	0.5982	0.5980	0.5979	0.5976	0.5975	0.5974
0.26	0.6009	0.5996	0.5991	0.5986	0.5984	0.5983	0.5979	0.5978	0.5977
0.28	0.6018	0.6003	0.5997	0.5991	0.5989	0.5987	0.5983	0.5981	0.5980
0.30	--	0.6010	0.6003	0.5997	0.5994	0.5991	0.5987	0.5985	0.5984
0.32	--	0.6019	0.6010	0.6003	0.5999	0.5996	0.5991	0.5989	0.5987
0.34	--	0.6028	0.6018	0.6009	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	--	0.6037	0.6026	0.6016	0.6011	0.6008	0.6000	0.5997	0.5995
0.38	--	0.6048	0.6035	0.6024	0.6018	0.6014	0.6006	0.6002	0.6000
0.40	--	--	0.6045	0.6032	0.6025	0.6020	0.6011	0.6007	0.6004
0.42	--	--	0.6055	0.6040	0.6033	0.6027	0.6017	0.6012	0.6009
0.44	--	--	0.6065	0.6049	0.6041	0.6035	0.6023	0.6017	0.6014
0.46	--	--	0.6077	0.6058	0.6049	0.6042	0.6029	0.6023	0.6020
0.48	--	--	0.6088	0.6068	0.6058	0.6050	0.6035	0.6029	0.6025
0.50	--	--	--	0.6078	0.6067	0.6058	0.6042	0.6034	0.6030
0.52	--	--	--	0.6088	0.6076	0.6066	0.6048	0.6040	0.6035
0.54	--	--	--	0.6098	0.6085	0.6074	0.6054	0.6045	0.6040
0.56	--	--	--	0.6108	0.6093	0.6082	0.6060	0.6050	0.6045
0.58	--	--	--	0.6118	0.6102	0.6089	0.6066	0.6055	0.6049
0.60	--	--	--	0.6127	0.6110	0.6096	0.6070	0.6058	0.6052
0.62	--	--	--	0.6135	0.6117	0.6102	0.6074	0.6061	0.6054
0.64	--	--	--	--	0.6123	0.6107	0.6077	0.6063	0.6055
0.65	--	--	--	--	0.6125	0.6108	0.6077	0.6063	0.6055
0.66	--	--	--	--	0.6127	0.6110	0.6077	0.6062	0.6054
0.67	--	--	--	--	0.6129	0.6111	0.6077	0.6061	0.6053
0.68	--	--	--	--	0.6130	0.6111	0.6076	0.6060	0.6051
0.69	--	--	--	--	0.6131	0.6111	0.6075	0.6058	0.6049
0.70	--	--	--	--	0.6131	0.6110	0.6073	0.6055	0.6045
0.71	--	--	--	--	0.6130	0.6109	0.6070	0.6052	0.6042
0.72	--	--	--	--	0.6128	0.6106	0.6066	0.6047	0.6037
0.73	--	--	--	--	0.6126	0.6103	0.6062	0.6042	0.6031
0.74	--	--	--	--	0.6123	0.6099	0.6056	0.6036	0.6025
0.75	--	--	--	--	--	0.6094	0.6050	0.6029	0.6018

Table A.6. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 6 in.

β \ R_D	10^4	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5986	0.5979	0.5977	0.5974	0.5973	0.5972	0.5971	0.5970	0.5969
0.22	0.5993	0.5984	0.5981	0.5978	0.5977	0.5975	0.5973	0.5972	0.5972
0.24	--	0.5990	0.5986	0.5982	0.5980	0.5979	0.5976	0.5975	0.5974
0.26	--	0.5996	0.5991	0.5987	0.5984	0.5983	0.5980	0.5978	0.5977
0.28	--	0.6003	0.5997	0.5992	0.5989	0.5987	0.5983	0.5981	0.5980
0.30	--	0.6010	0.6003	0.5997	0.5994	0.5992	0.5987	0.5985	0.5984
0.32	--	0.6019	0.6010	0.6003	0.5999	0.5996	0.5991	0.5989	0.5987
0.34	--	--	0.6018	0.6009	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	--	--	0.6026	0.6016	0.6012	0.6008	0.6000	0.5997	0.5995
0.38	--	--	0.6035	0.6024	0.6018	0.6014	0.6006	0.6002	0.6000
0.40	--	--	--	0.6031	0.6025	0.6020	0.6011	0.6007	0.6004
0.42	--	--	--	0.6040	0.6033	0.6027	0.6017	0.6012	0.6009
0.44	--	--	--	0.6048	0.6040	0.6034	0.6022	0.6017	0.6014
0.46	--	--	--	0.6057	0.6049	0.6041	0.6028	0.6022	0.6019
0.48	--	--	--	0.6067	0.6057	0.6049	0.6034	0.6027	0.6024
0.50	--	--	--	0.6076	0.6065	0.6056	0.6040	0.6033	0.6029
0.52	--	--	--	--	0.6074	0.6064	0.6046	0.6038	0.6033
0.54	--	--	--	--	0.6082	0.6071	0.6052	0.6043	0.6038
0.56	--	--	--	--	0.6090	0.6078	0.6057	0.6047	0.6041
0.58	--	--	--	--	0.6098	0.6085	0.6061	0.6051	0.6044
0.60	--	--	--	--	0.6105	0.6091	0.6065	0.6053	0.6047
0.62	--	--	--	--	--	0.6095	0.6068	0.6055	0.6048
0.64	--	--	--	--	--	0.6099	0.6069	0.6055	0.6047
0.65	--	--	--	--	--	0.6100	0.6068	0.6054	0.6046
0.66	--	--	--	--	--	0.6100	0.6068	0.6053	0.6044
0.67	--	--	--	--	--	0.6100	0.6066	0.6051	0.6042
0.68	--	--	--	--	--	0.6099	0.6064	0.6048	0.6039
0.69	--	--	--	--	--	0.6098	0.6062	0.6045	0.6036
0.70	--	--	--	--	--	0.6096	0.6058	0.6041	0.6031
0.71	--	--	--	--	--	0.6093	0.6054	0.6036	0.6026
0.72	--	--	--	--	--	0.6089	0.6049	0.6030	0.6020
0.73	--	--	--	--	--	--	0.6043	0.6023	0.6012
0.74	--	--	--	--	--	--	0.6035	0.6015	0.6004
0.75	--	--	--	--	--	--	0.6027	0.6006	0.5994

Table A.7. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 8 in.

β \ R_D	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5979	0.5977	0.5974	0.5973	0.5972	0.5971	0.5970	0.5970
0.22	0.5984	0.5981	0.5978	0.5977	0.5975	0.9573	0.5972	0.5972
0.24	0.5990	0.5986	0.5982	0.5980	0.5979	0.5976	0.5975	0.5974
0.26	0.5996	0.5991	0.5987	0.5985	0.5983	0.5980	0.5978	0.5977
0.28	0.6003	0.5997	0.5992	0.5989	0.5987	0.5983	0.5981	0.5980
0.30	--	0.6003	0.5997	0.5994	0.5992	0.5987	0.5985	0.5984
0.32	--	0.6010	0.6003	0.5999	0.5997	0.5991	0.5989	0.5987
0.34	--	0.6018	0.6009	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	--	--	0.6016	0.6012	0.6008	0.6001	0.5597	0.5995
0.38	--	--	0.6024	0.6018	0.6014	0.6006	0.6002	0.6000
0.40	--	--	0.6031	0.6025	0.6020	0.6011	0.6007	0.6004
0.42	--	--	0.6040	0.6033	0.6027	0.6016	0.6011	0.6009
0.44	--	--	0.6048	0.6040	0.6034	0.6022	0.6017	0.6014
0.46	--	--	--	0.6048	0.6041	0.6028	0.6022	0.6018
0.48	--	--	--	0.6056	0.6048	0.6034	0.6027	0.6023
0.50	--	--	--	0.6065	0.6056	0.6040	0.6032	0.6028
0.52	--	--	--	0.6073	0.6063	0.6045	0.6037	0.6032
0.54	--	--	--	--	0.6070	0.6050	0.6041	0.6036
0.56	--	--	--	--	0.6077	0.6055	0.6045	0.6040
0.58	--	--	--	--	0.6083	0.6059	0.6048	0.6042
0.60	--	--	--	--	0.6088	0.6063	0.6051	0.6044
0.62	--	--	--	--	0.6092	0.6064	0.6052	0.6044
0.64	--	--	--	--	--	0.6065	0.6051	0.6043
0.65	--	--	--	--	--	0.6064	0.6050	0.6041
0.66	--	--	--	--	--	0.6063	0.6048	0.6039
0.67	--	--	--	--	--	0.6061	0.6045	0.6037
0.68	--	--	--	--	--	0.6059	0.6042	0.6033
0.69	--	--	--	--	--	0.6055	0.6039	0.6029
0.70	--	--	--	--	--	0.6051	0.6034	0.6024
0.71	--	--	--	--	--	0.6046	0.6028	0.6018
0.72	--	--	--	--	--	0.6040	0.6021	0.6011
0.73	--	--	--	--	--	0.6033	0.6014	0.6003
0.74	--	--	--	--	--	0.6025	0.6005	0.5993
0.75	--	--	--	--	--	0.6015	0.5994	0.5983

Table A.8. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 10 in.

β \ R_D	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5979	0.5977	0.5974	0.5973	0.5972	0.5971	0.5970	0.5970
0.22	0.5984	0.5981	0.5978	0.5977	0.5976	0.5973	0.5972	0.5972
0.24	0.5990	0.5986	0.5982	0.5980	0.5979	0.5976	0.5975	0.5975
0.26	--	0.5991	0.5987	0.5985	0.5983	0.5980	0.5978	0.5977
0.28	--	0.5997	0.5992	0.5989	0.5987	0.5983	0.5981	0.5980
0.30	--	0.6003	0.5997	0.5994	0.5992	0.5987	0.5985	0.5984
0.32	--	--	0.6003	0.5999	0.5997	0.5991	0.5989	0.5987
0.34	--	--	0.6009	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	--	--	0.6016	0.6012	0.6008	0.6001	0.5997	0.5995
0.38	--	--	0.6024	0.6018	0.6014	0.6006	0.6002	0.6000
0.40	--	--	--	0.6025	0.6020	0.6011	0.6006	0.6004
0.42	--	--	--	0.6032	0.6027	0.6016	0.6011	0.6009
0.44	--	--	--	0.6040	0.6034	0.6022	0.6016	0.6013
0.46	--	--	--	0.6048	0.6041	0.6028	0.6022	0.6018
0.48	--	--	--	--	0.6048	0.6033	0.6027	0.6023
0.50	--	--	--	--	0.6055	0.6039	0.6032	0.6027
0.52	--	--	--	--	0.6062	0.6045	0.6036	0.6032
0.54	--	--	--	--	0.6069	0.6050	0.6041	0.6035
0.56	--	--	--	--	0.6076	0.6054	0.6044	0.6039
0.58	--	--	--	--	--	0.6058	0.6047	0.6041
0.60	--	--	--	--	--	0.6061	0.6049	0.6042
0.62	--	--	--	--	--	0.6063	0.6050	0.6042
0.64	--	--	--	--	--	0.6062	0.6048	0.6041
0.65	--	--	--	--	--	0.6061	0.6047	0.6039
0.66	--	--	--	--	--	0.6060	0.6045	0.6037
0.67	--	--	--	--	--	0.6058	0.6042	0.6034
0.68	--	--	--	--	--	0.6055	0.6039	0.6030
0.69	-	--	--	--	--	0.6051	0.6035	0.6025
0.70	--	--	--	--	--	0.6047	0.6029	0.6020
0.71	--	--	--	--	--	0.6041	0.6023	0.6013
0.72	--	--	--	--	--	0.6035	0.6016	0.6006
0.73	--	--	--	--	--	0.6027	0.6008	0.5997
0.74	--	--	--	--	--	0.6018	0.5998	0.5987
0.75	--	--	--	--	--	0.6008	0.5987	0.5976

Table A.9. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 15 in.

β \ R_D	2×10^4	3×10^4	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5979	0.5977	0.5974	0.5973	0.5972	0.5971	0.5970	0.5970
0.22	--	0.5981	0.5978	0.5977	0.5976	0.5973	0.5973	0.5972
0.24	--	0.5986	0.5982	0.5980	0.5979	0.5976	0.5975	0.5975
0.26	--	--	0.5987	0.5985	0.5983	0.5980	0.5978	0.5977
0.28	--	--	0.5992	0.5989	0.5987	0.5983	0.5982	0.5981
0.30	--	--	0.5997	0.5994	0.5992	0.5987	0.5985	0.5984
0.32	--	--	0.6003	0.6000	0.5997	0.5991	0.5989	0.5988
0.34	--	--	--	0.6005	0.6002	0.5996	0.5993	0.5991
0.36	--	--	--	0.6012	0.6008	0.6001	0.5997	0.5995
0.38	--	--	--	0.6018	0.6014	0.6006	0.6002	0.6000
0.40	--	--	--	--	0.6020	0.6011	0.6006	0.6004
0.42	--	--	--	--	0.6027	0.6016	0.6011	0.6009
0.44	--	--	--	--	0.6033	0.6022	0.6016	0.6013
0.46	--	--	--	--	0.6040	0.6027	0.6021	0.6018
0.48	--	--	--	--	--	0.6033	0.6026	0.6022
0.50	--	--	--	--	--	0.6038	0.6031	0.6027
0.52	--	--	--	--	--	0.6044	0.6035	0.6031
0.54	--	--	--	--	--	0.6049	0.6040	0.6034
0.56	--	--	--	--	--	0.6053	0.6043	0.6037
0.58	--	--	--	--	--	0.6056	0.6046	0.6039
0.60	--	--	--	--	--	0.6059	0.6047	0.6040
0.62	--	--	--	--	--	0.6060	0.6047	0.6040
0.64	--	--	--	--	--	0.6059	0.6045	0.6037
0.65	--	--	--	--	--	0.6058	0.6043	0.6035
0.66	--	--	--	--	--	0.6056	0.6041	0.6033
0.67	--	--	--	--	--	0.6054	0.6038	0.6029
0.68	--	--	--	--	--	0.6050	0.6034	0.6025
0.69	--	--	--	--	--	0.6046	0.6029	0.6020
0.70	--	--	--	--	--	0.6041	0.6024	0.6014
0.71	--	--	--	--	--	0.6035	0.6017	0.6007
0.72	--	--	--	--	--	0.6028	0.6009	0.5999
0.73	--	--	--	--	--	0.6020	0.6000	0.5989
0.74	--	--	--	--	--	0.6010	0.5990	0.5979
0.75	--	--	--	--	--	0.5999	0.5978	0.5966

Table A.10. Coefficient of Discharge C

Orifice Plate with Flange Taps

D = 30 in.

β \ R_D	5×10^4	7×10^4	10^5	3×10^5	10^6	10^7
0.20	0.5974	0.5973	0.5972	0.5971	0.5970	0.5970
0.22	0.5978	0.5977	0.5976	0.5974	0.5973	0.5972
0.24	--	0.5981	0.5979	0.5977	0.5975	0.5975
0.26	--	0.5985	0.5983	0.5980	0.5978	0.5978
0.28	--	--	0.5987	0.5983	0.5982	0.5981
0.30	--	--	0.5992	0.5987	0.5985	0.5984
0.32	--	--	0.5997	0.5991	0.5989	0.5988
0.34	--	--	--	0.5996	0.5993	0.5991
0.36	--	--	--	0.6001	0.5997	0.5995
0.38	--	--	--	0.6005	0.6002	0.6000
0.40	--	--	--	0.6011	0.6006	0.6004
0.42	--	--	--	0.6016	0.6011	0.6008
0.44	--	--	--	0.6021	0.6016	0.6013
0.46	--	--	--	0.6027	0.6021	0.6017
0.48	--	--	--	0.6032	0.6026	0.6022
0.50	--	--	--	0.6038	0.6030	0.6026
0.52	--	--	--	0.6043	0.6035	0.6030
0.54	--	--	--	0.6048	0.6038	0.6033
0.56	--	--	--	--	0.6042	0.6036
0.58	--	--	--	--	0.6044	0.6038
0.60	--	--	--	--	0.6045	0.6038
0.62	--	--	--	--	0.6044	0.6037
0.64	--	--	--	--	0.6042	0.6034
0.65	--	--	--	--	0.6040	0.6032
0.66	--	--	--	--	0.6037	0.6029
0.67	--	--	--	--	0.6034	0.6025
0.68	--	--	--	--	0.6029	0.6020
0.69	--	--	--	--	0.6024	0.6015
0.70	--	--	--	--	0.6018	0.6008
0.71	--	--	--	--	0.6011	0.6001
0.72	--	--	--	--	0.6002	0.5992
0.73	--	--	--	--	0.5992	0.5982
0.74	--	--	--	--	0.5981	0.5970
0.75	--	--	--	--	0.5969	0.5957

Table A.11. Coefficient of Discharge C

ISA 1932 Flow Nozzle

2" $\leq$ D $\leq$ 20"

β \ R_D	2 x 10^4	3 x 10^4	5 x 10^4	7 x 10^4	10^5	3 x 10^5	10^6	2 x 10^6
0.30	--	--	--	0.9860	0.9868	0.9879	0.9883	0.9883
0.32	--	--	--	0.9850	0.9860	0.9874	0.9878	0.9878
0.34	--	--	--	0.9841	0.9851	0.9867	0.9871	0.9872
0.36	--	--	--	0.9830	0.9842	0.9859	0.9864	0.9865
0.38	--	--	--	0.9818	0.9831	0.9850	0.9855	0.9856
0.40	--	--	--	0.9804	0.9819	0.9839	0.9845	0.9846
0.42	--	--	--	0.9790	0.9805	0.9827	0.9833	0.9834
0.44	0.9617	0.9694	0.9751	0.9773	0.9790	0.9813	0.9820	0.9821
0.46	0.9593	0.9672	0.9732	0.9756	0.9773	0.9797	0.9804	0.9805
0.48	0.9567	0.9650	0.9711	0.9736	0.9754	0.9779	0.9786	0.9787
0.50	0.9542	0.9626	0.9689	0.9715	0.9733	0.9758	0.9766	0.9767
0.52	0.9516	0.9601	0.9665	0.9691	0.9709	0.9735	0.9743	0.9744
0.54	0.9490	0.9575	0.9639	0.9665	0.9683	0.9709	0.9717	0.9718
0.56	0.9464	0.9548	0.9611	0.9636	0.9655	0.9680	0.9688	0.9689
0.58	0.9437	0.9519	0.9581	0.9605	0.9623	0.9648	0.9655	0.9656
0.60	0.9411	0.9489	0.9548	0.9572	0.9588	0.9612	0.9619	0.9620
0.62	0.9384	0.9458	0.9513	0.9535	0.9550	0.9573	0.9579	0.9580
0.64	0.9358	0.9425	0.9475	0.9495	0.9509	0.9529	0.9535	0.9536
0.66	0.9332	0.9390	0.9434	0.9451	0.9464	0.9481	0.9487	0.9487
0.68	0.9306	0.9354	0.9390	0.9404	0.9414	0.9429	0.9433	0.9434
0.70	0.9280	0.9316	0.9342	0.9353	0.9361	0.9372	0.9375	0.9375
0.72	0.9255	0.9276	0.9292	0.9298	0.9303	0.9309	0.9311	0.9311
0.74	0.9230	0.9235	0.9238	0.9239	0.9240	0.9241	0.9242	0.9242
0.76	0.9207	0.9191	0.9180	0.9175	0.9172	0.9168	0.9166	0.9166
0.78	0.9184	0.9146	0.9118	0.9107	0.9099	0.9088	0.9084	0.9084
0.80	0.9162	0.9100	0.9053	0.9034	0.9020	0.9001	0.8996	0.8995

Table A.12. Coefficient of Discharge C

Long Radius Flow Nozzle

2" $\leq$ D $\leq$ 24"

β \ R_D	10^4	2×10^4	5×10^4	10^5	2×10^5	5×10^5	10^6	5×10^6	10^7
0.20	0.9673	0.9759	0.9834	0.9873	0.9900	0.9924	0.9936	0.9952	0.9556
0.22	0.9659	0.9748	0.9828	0.9868	0.9897	0.9922	0.9934	0.9951	0.9955
0.24	0.9645	0.9739	0.9822	0.9864	0.9893	0.9920	0.9933	0.9951	0.9955
0.26	0.9632	0.9730	0.9816	0.9860	0.9891	0.9918	0.9932	0.9950	0.9954
0.28	0.9619	0.9721	0.9810	0.9856	0.9888	0.9916	0.9930	0.9950	0.9954
0.30	0.9607	0.9712	0.9805	0.9852	0.9885	0.9914	0.9929	0.9949	0.9954
0.32	0.9596	0.9704	0.9800	0.9848	0.9882	0.9913	0.9928	0.9948	0.9953
0.34	0.9584	0.9696	0.9795	0.9845	0.9880	0.9911	0.9927	0.9948	0.9953
0.36	0.9573	0.9688	0.9790	0.9841	0.9877	0.9910	0.9926	0.9947	0.9953
0.38	0.9562	0.9680	0.9785	0.9838	0.9875	0.9908	0.9925	0.9947	0.9952
0.40	0.9552	0.9673	0.9780	0.9834	0.9873	0.9907	0.9924	0.9947	0.9952
0.42	0.9542	0.9666	0.9776	0.9831	0.9870	0.9905	0.9923	0.9946	0.9952
0.44	0.9532	0.9659	0.9771	0.9828	0.9868	0.9904	0.9922	0.9946	0.9951
0.46	0.9522	0.9652	0.9767	0.9825	0.9866	0.9902	0.9921	0.9945	0.9951
0.48	0.9513	0.9645	0.9763	0.9822	0.9864	0.9901	0.9920	0.9945	0.9951
0.50	0.9503	0.9639	0.9759	0.9819	0.9862	0.9900	0.9919	0.9944	0.9950
0.52	0.9494	0.9632	0.9754	0.9816	0.9860	0.9898	0.9918	0.9944	0.9950
0.54	0.9485	0.9626	0.9750	0.9813	0.9858	0.9897	0.9917	0.9944	0.9950
0.56	0.9476	0.9619	0.9746	0.9810	0.9856	0.9896	0.9916	0.9943	0.9950
0.58	0.9468	0.9613	0.9743	0.9808	0.9854	0.9895	0.9915	0.9943	0.9949
0.60	0.9459	0.9607	0.9739	0.9805	0.9852	0.9893	0.9914	0.9942	0.9949
0.62	0.9451	0.9601	0.9735	0.9802	0.9850	0.9892	0.9914	0.9942	0.9949
0.64	0.9443	0.9596	0.9731	0.9800	0.9848	0.9891	0.9913	0.9942	0.9948
0.66	0.9435	0.9590	0.9728	0.9797	0.9846	0.9890	0.9912	0.9941	0.9948
0.68	0.9427	0.9584	0.9724	0.9795	0.9845	0.9889	0.9911	0.9941	0.9948
0.70	0.9419	0.9579	0.9721	0.9792	0.9843	0.9888	0.9910	0.9941	0.9948
0.72	0.9411	0.9573	0.9717	0.9790	0.9841	0.9887	0.9910	0.9940	0.9947
0.74	0.9403	0.9568	0.9714	0.9787	0.9839	0.9886	0.9909	0.9940	0.9947
0.76	0.9396	0.9562	0.9710	0.9785	0.9838	0.9884	0.9908	0.9940	0.9947
0.78	0.9388	0.9557	0.9707	0.9783	0.9836	0.9883	0.9907	0.9939	0.9947
0.80	0.9381	0.9552	0.9704	0.9780	0.9834	0.9882	0.9907	0.9939	0.9947

Table A.13. Coefficient of Discharge C

Nozzle Venturi Meter

$2.5'' \leq D \leq 20''$

$d \geq 2''$

$1.5 \times 10^5 \leq R_D \leq 2 \times 10^6$

β	C
0.316	0.9847
0.320	0.9846
0.340	0.9843
0.360	0.9838
0.380	0.9833
0.400	0.9826
0.420	0.9818
0.440	0.9809
0.460	0.9798
0.480	0.9786
0.500	0.9771
0.520	0.9755
0.540	0.9736
0.560	0.9714
0.580	0.9689
0.600	0.9661
0.620	0.9630
0.640	0.9595
0.660	0.9556
0.680	0.9512
0.700	0.9464
0.720	0.9411
0.740	0.9352
0.760	0.9288
0.775	0.9236

Table A.14. Coefficient of Discharge C

Classical Venturi Meter

	Rough-cast Entrance Cone	Machined Entrance Cone	Rough Welded Sheet-Metal Entrance Cone
Coefficient C	0.984	0.995	0.985
Tolerance on C, %	0.70	1.00	1.50
β	0.30 to 0.75	0.40 to 0.75	0.40 to 0.70
D, in.	4 to 32	2 to 10	8 to 48
R_D	2×10^5 to 2×10^6	2×10^5 to 1×10^6	2×10^5 to 2×10^6

Table A.15. References for Equations for Coefficient of Discharge C

Primary Element	Source
Thin-plate, square-edge orifice for corner, flange, and D, D/2 taps	Reference 11 or Reference 10
ISA 1932 flow nozzle	Reference 10
Long radius flow nozzle	Reference 10
Nozzle venturi meter	Reference 10
Classical venturi meter	No known references

Table A.16. Fluid Expansion Factor Y for Flow Nozzles and Venturi Meters

γ = 1.3 (steam)

β	β^4 \ r	0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55
0.20	0.0016	0.9707	0.9407	0.9099	0.8781	0.8454	0.8117	0.7768	0.7406	0.7030
.30	.0081	.9705	.9402	.9092	.8773	.8445	.8106	.7756	.7393	.7016
.40	.0256	.9698	.9390	.9074	.8750	.8417	.8075	.7722	.7357	.6978
0.50	.0625	.9683	.9362	.9034	.8700	.8358	.8008	.7648	.7278	.6896
.55	.0915	.9671	.9338	.9001	.8658	.8309	.7952	.7588	.7214	.6829
.60	.1296	.9654	.9305	.8954	.8599	.8240	.7876	.7505	.7126	.6738
0.65	.1785	.9629	.9259	.8889	.8519	.8146	.7771	.7392	.7007	.6614
.70	.2401	.9594	.9193	.8798	.8406	.8016	.7627	.7237	.6844	.6447
.725	.2763	.9570	.9150	.8739	.8333	.7933	.7535	.7139	.6742	.6343
0.75	.3164	.9542	.9098	.8667	.8246	.7833	.7426	.7023	.6622	.6221
.775	.3608	.9507	.9034	.8580	.8141	.7714	.7297	.6886	.6481	.6077
.80	.4096	.9462	.8955	.8473	.8013	.7570	.7141	.6723	.6313	.5908
0.82	.4521	.9418	.8876	.8368	.7888	.7431	.6992	.6568	.6155	.5750
.84	.4979	.9362	.8779	.8241	.7739	.7266	.6817	.6387	.5971	.5567
.86	.5470	.9292	.8658	.8084	.7557	.7067	.6608	.6172	.5756	.5353

γ = 1.4 (air)

β	β^4 \ r	0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55
0.20	0.0016	0.9728	0.9448	0.9160	0.8863	0.8556	0.8238	0.7908	0.7565	0.7207
.30	.0081	.9726	.9444	.9154	.8855	.8546	.8227	.7896	.7552	.7193
.40	.0256	.9719	.9432	.9137	.8833	.8520	.8198	.7864	.7517	.7156
0.50	.0625	.9706	.9405	.9099	.8785	.8464	.8133	.7793	.7441	.7076
.55	.0915	.9694	.9383	.9067	.8745	.8416	.8080	.7734	.7378	.7010
.60	.1296	.9678	.9352	.9023	.8690	.8351	.8006	.7653	.7292	.6920
0.65	.1785	.9655	.9309	.8962	.8613	.8261	.7905	.7543	.7175	.6798
.70	.2401	.9622	.9247	.8876	.8506	.8136	.7765	.7392	.7016	.6633
.725	.2763	.9600	.9207	.8819	.8436	.8056	.7676	.7297	.6915	.6530
0.75	.3164	.9573	.9158	.8751	.8353	.7960	.7571	.7184	.6797	.6409
.775	.3608	.9540	.9097	.8669	.8252	.7845	.7445	.7050	.6657	.6266
.80	.4096	.9498	.9022	.8566	.8128	.7705	.7292	.6889	.6491	.6097
0.82	.4521	.9457	.8947	.8466	.8009	.7570	.7147	.6736	.6334	.5939
.84	.4979	.9405	.8856	.8344	.7864	.7409	.6975	.6557	.6152	.5755
.86	.5470	.9338	.8740	.8194	.7688	.7215	.6769	.6344	.5936	.5541

$$Y = \left(r^{2/\gamma} \frac{\gamma}{\gamma - 1} \frac{1 - r^{(\gamma-1)/\gamma}}{1 - r} \frac{1 - \beta^4}{1 - [(\beta^4)(r^{2/\gamma})]} \right)^{1/2}$$

where r is the pressure ratio P_2/P_1 and γ is the specific heat ratio c_p/c_v.

Limits of use are given in reference 10 and are the same as those for discharge coefficient C. See tables A-11 through A.14. $P_2/P_1 \leq 0.75$.

Table A.17. Uncertainty of Discharge Coefficients

Orifice Meters

	Corner Taps	Flange Taps	D and D/2 taps
$\beta \leq 0.6$	0.6%	0.6%	0.6%
$0.6 \leq \beta \leq 0.8$	β%		
$0.6 < \beta \leq 0.75$		β%	β%

Flow Nozzles

	ISA 1983	Long Radius
$\beta \leq 0.6$	0.8%	
$\beta > 0.6$	$(2\beta - 0.4)$%	
$0.2 \leq \beta \leq 0.8$		2%

Classical Venturi

Rough cast entrance	0.7%
Machined entrance	1.0%
Rough-welded sheet iron entrance	1.5%

Nozzle Venturi Meter

The uncertainty in α is $(1.2 + 1.5\ \beta^4)$% where $\alpha = C/(1-\beta^4)^{1/2}$

Source: Reference 10

Notes: Quantities β, R_D and D are assumed known withour error.

The above uncertainties are given for reference purposes. They would be useful in estimating the uncertainty in flowrate when the primary element is used uncalibrated and the secondary element (ΔP transducer) only receives calibration.

Table A.18. Uncertainty of Expansion Factors [10]

Orifice Meters

$\beta \leq 0.75$	$4(\Delta P/P_1)\%$	
$0.75 < \beta \leq 0.8$	$8(\Delta P/P_1)\%$	(corner taps only)

Flow Nozzles

ISA 1932 and Long Radius	$2(\Delta P/P_1)\%$

Venturi Meters

Nozzle venturi and Classical venturi meters	$(4+100\beta^8)(\Delta P/P_1)\%$

Notes: Quantities β, ΔP and P_1 are assumed known without error.

The above uncertainties are given for reference purposes. They would be useful in estimating the uncertainty in flowrate when the primary element is used uncalibrated and the secondary element (ΔP transducer) only receives calibration, or when the primary element has been calibrated with a liquid and is to be used to monitor gas (air or steam).

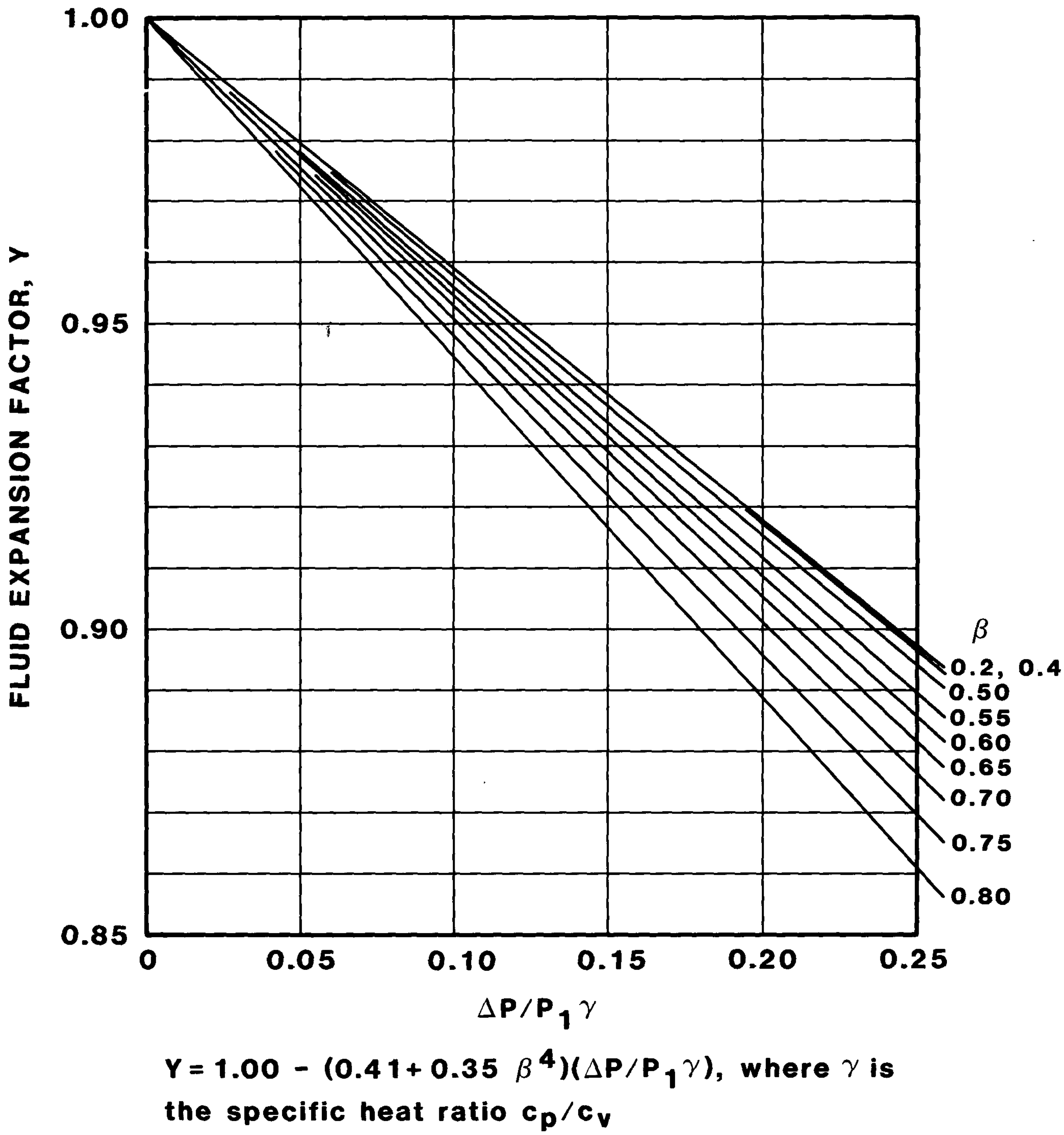

Figure A.1. Fluid expansion factor Y for thin-plate, square-edged orifice plates with corner taps, flange taps, and 1D and 1/2D taps. Pressure ratio $P_1/P_2 \leq 0.75$. Limits of use for β and R_D are the same as those for C. See tables A-1 through A-10. [2,10]

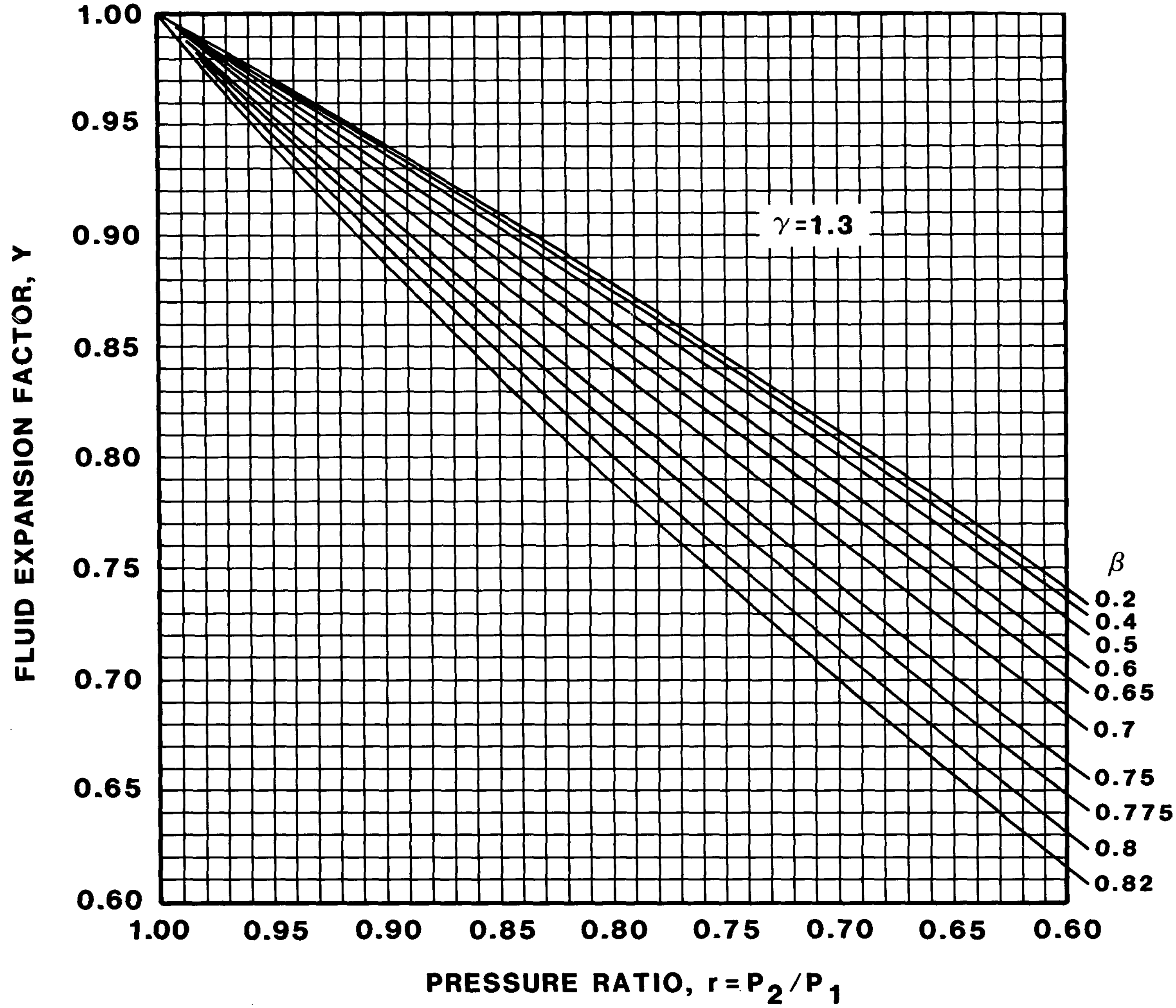

Figure A.2. Fluid expansion factor Y for flow nozzles and venturi meters, γ = 1.3 (steam). For limits of use, see table A.16. [2]

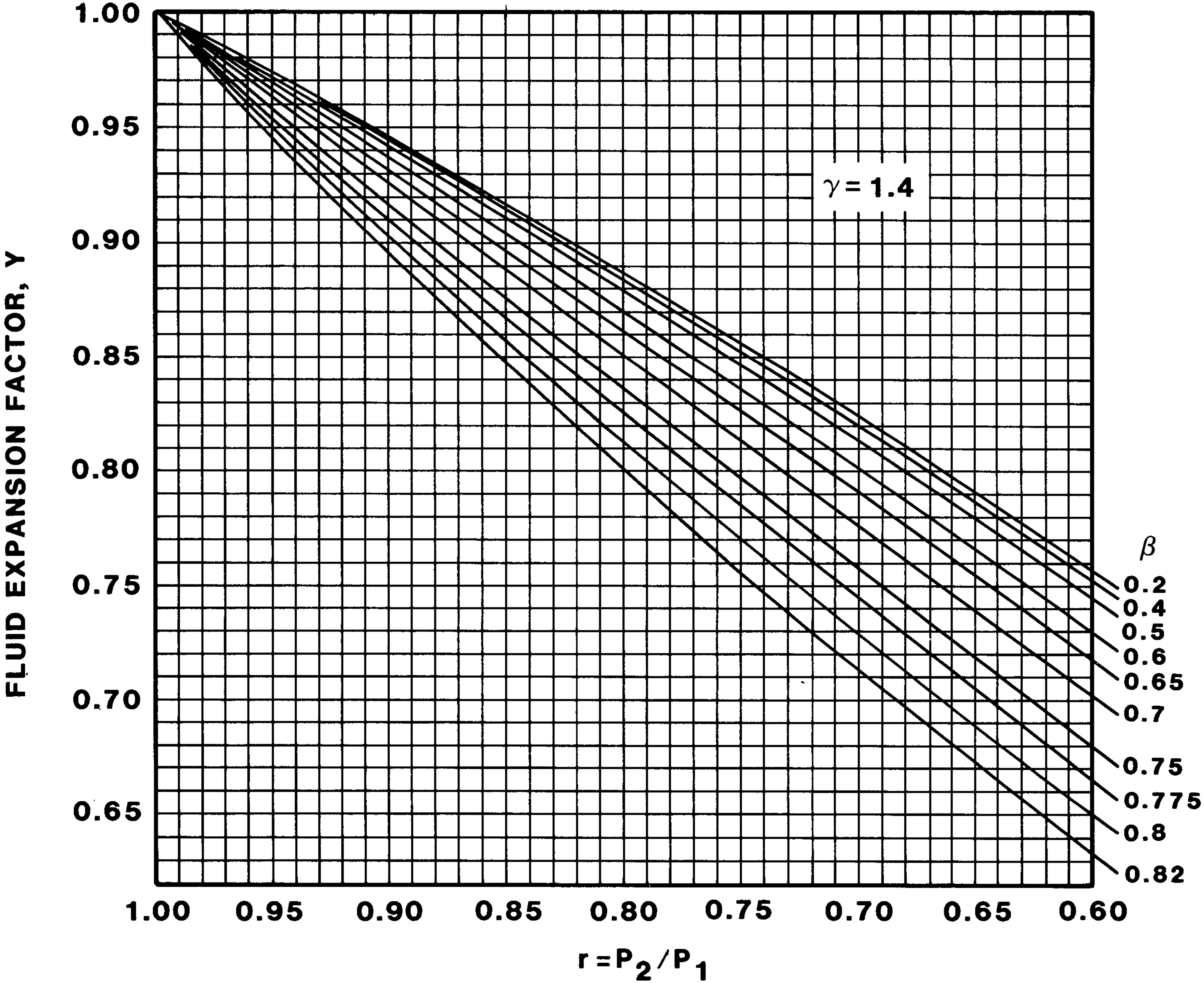

Figure A.3. Fluid expansion factor Y for the flow nozzles and venturi meters, $\gamma = 1.4$ (air). For limits see table A.16. [2]

APPENDIX B

FLUID PROPERTIES AND FLOW QUANTITY CONVERSION FACTORS

Table B.1. Density of Saturated and Compressed Liquid Water (lb/ft^3) [6]

Temperature (°F)	Pressure, psia Saturated	500	1000	Temperature (°F)	Pressure, Psia Saturated	500	1000
32	62.4140	62.5217	62.6288	75	62.2654	62.3618	62.4575
33	.4167	.5240	.6308	76	.2568	.3530	.4486
34	.4191	.5260	.6324	77	.2479	.3440	.4395
				78	.2389	.3349	.4302
35	62.4212	62.5277	62.6336	79	.2297	.3255	.4207
36	.4229	.5289	.6345				
37	.4242	.5299	.6351	80	62.2203	62.3160	62.4111
38	.4252	.5305	.6353	81	.2107	.3063	.4013
39	.4258	.5308	.6352	82	.2009	.2964	.3913
				83	.1910	.2864	.3811
40	62.4261	62.5307	62.6348	84	.1809	.2762	.3708
41	.4261	.5304	.6341				
42	.4257	.5297	.6330	85	62.1706	62.2658	62.3603
43	.4251	.5287	.6317	90	.1166	.2113	.3055
44	.4241	.5274	.6301	95	.0585	.1529	.2467
45	62.4229	62.5258	62.6282	100	61.9964	.0906	.1841
46	.4213	.5239	.6260	105	.9307	.0246	.1180
47	.4194	.5218	.6235				
48	.4173	.5193	.6208	110	61.8612	61.9551	62.0483
49	.4149	.5166	.6178	115	.7884	.8821	61.9754
				120	.7121	.8059	.8992
50	62.4122	62.5136	62.6145	125	.6326	.7265	.8198
51	.4092	.5104	.6110	130	.5500	.6440	.7375
52	.4059	.5068	.6072				
53	.4024	.5031	.6031	135	61.4643	61.5584	61.6521
54	.3986	.4990	.5988	140	.3757	.4700	.5640
				145	.2842	.3787	.4730
55	62.3946	62.4947	62.5943	150	.1899	.2847	.3793
56	.3903	.4902	.5895	155	.0928	.1880	.2830
57	.3858	.4854	.5845				
58	.3810	.4804	.5793	160	60.9932	61.0887	61.1841
59	.3760	.4752	.5738	165	.8909	60.9868	.0827
				170	.7862	.8824	60.9789
60	62.3707	62.4697	62.5681	175	.6789	.7756	.8726
61	.3652	.4640	.5622	180	.5693	.6665	.7640
62	.3595	.4581	.5560				
63	.3535	.4519	.5497	185	60.4573	60.5549	60.6531
64	.3474	.4455	.5431	190	.3430	.4411	.5400
				195	.2265	.3250	.4246
65	62.3410	62.4390	62.5363	200	.1076	.2068	.3070
66	.3344	.4322	.5293	205	59.9866	.0863	.1873
67	.3275	.4251	.5221				
68	.3205	.4179	.5147	210	59.8635	59.9636	60.0655
69	.3132	.4105	.5071	215	.7382	.8389	59.9416
				220	.6108	.7120	.8156
70	62.3058	62.4029	62.4993	225	.4813	.5830	.6875
71	.2981	.3950	.4914	230	.3497	.4520	.5574
72	.2902	.3870	.4832				
73	.2822	.3788	.4748	235	59.2161	59.3189	59.4253
74	.2739	.3704	.4663	240	.0804	.1838	.2912

Table B.1. Density of Saturated and Compressed Liquid Water (lb/ft^3) [6]

Temperature (°F)	Pressure, psia Saturated	500	1000	Temperature (°F)	Pressure, Psia Saturated	500	1000
245	58.9428	0.0467	.1551	385	54.2597	54.3546	54.5140
250	.8031	58.9075	.0171	390	.0606	.1531	.3155
255	.6614	.7663	58.8770	395	53.8590	53.9489	.1144
				400	.6548	.7418	53.9105
260	58.5177	58.6231	58.7350	405	.4481	.5318	.7039
265	.3720	.4779	.5910				
270	.2244	.3306	.4450	410	53.2387	53.3187	53.4944
275	.0747	.1814	.2970	415	.0267	.1026	.2819
280	57.9231	.0301	.1471	420	52.8119	52.8833	.0665
				425	.5942	.6607	52.8480
285	57.7695	57.8768	57.9952	430	.3737	.4348	.6262
290	.6139	.7215	.8413				
295	.4563	.5641	.6854	435	52.1503	52.2053	52.4012
300	.2966	.4046	.5275	440	51.9238	51.9723	.1728
305	.1350	.2431	.3675	445	.6942	.7354	51.9409
				450	.4615	.4948	.7054
310	56.9713	57.0795	57.2056	455	.2255	.2501	.4661
315	.8056	56.9137	.0415				
320	.6378	.7459	56.8754	460	50.9862	51.0012	51.2229
325	.4680	.5758	.7072	465	.7434	50.7479	50.9757
330	.2960	.4036	.5369	470	.4971	.4971	.7243
				475	.2472	.2472	.4686
335	56.1220	56.2291	56.3644	480	49.9935	49.9935	.2082
340	55.9458	.0524	.1897				
345	.7674	55.8735	.0128	485	49.7359	49.7359	49.9431
350	.5869	.6922	55.8337	490	.4744	.4744	.6731
355	.4042	.5085	.6523	495	.2087	.2087	.3978
				500	48.9387	48.9387	.1170
360	55.2192	55.3225	55.4687				
365	.0320	.1340	.2826				
370	54.8424	54.9430	.0942				
375	.6506	.7495	54.9033				
380	.4563	.5534	.7099				

Table B.2. Density of Mercury (lb/ft^3) [20]

Temperature (°F)	ρ	Temperature (°F)	ρ	Temperature (°F)	ρ	Temperature (°F)	ρ
- 5	851.88814	35	848.45654	75	845.04735	115	841.65689
- 4	.80205	36	.37101	76	844.96245	116	.57198
- 3	.71598	37	.28611	77	.87692	117	.48771
- 2	.62992	38	.20059	78	.79202	118	.40343
- 1	.54388	39	.11506	79	.70712	119	.31853
0	.45785	40	.02953	80	.62221	120	.23425
1	.37184	41	847.94401	81	.53731	121	.14997
2	.28584	42	.85848	82	.45241	122	.06507
3	.19963	43	.77295	83	.36751	123	840.98079
4	.11390	44	.68805	84	.28261	124	.89651
5	.02795	45	.60253	85	.19770	125	.81161
6	850.94201	46	.51700	86	.11280	126	.72733
7	.85609	47	.43210	87	.02790	127	.64306
8	.77019	48	.34657	88	843.94300	128	.55815
9	.68430	49	.26104	89	.85810	129	.47388
10	.59843	50	.17552	90	.77319	130	.38960
11	.51257	51	.08999	91	.68829	131	.30532
12	.42673	52	.00509	92	.60401	132	.22104
13	.34090	53	846.91956	93	.51911	133	.13677
14	.25509	54	.83466	94	.43421	134	.05249
15	.16956	55	.74913	95	.34931	135	839.96821
16	.08404	56	.66423	96	.26441	136	.88393
17	849.99851	57	.57871	97	.17950	137	.79965
18	.91236	58	.49380	98	.09523	138	.71538
19	.82683	59	.40828	99	.01032	139	.63110
20	.74068	60	.32338	100	842.92542	140	.54682
21	.65516	61	.23847	101	.84114	145	.12543
22	.56963	62	.15295	102	.75624	150	838.70467
23	.48348	63	.06805	103	.67196	155	.28390
24	.39795	64	845.98314	104	.58706	160	837.86376
25	.31243	65	.89762	105	.50216	165	.44362
26	.22690	66	.81272	106	.41788	170	.02348
27	.14137	67	.72719	107	.33298	175	836.60397
28	.05522	68	.64229	108	.24870	180	.18445
29	848.96970	69	.55739	109	.16380	185	835.76556
30	.88417	70	.47248	110	.07952	190	.34667
31	.79864	71	.38758	111	841.99462	195	834.92778
32	.71312	72	.30268	112	.91034	200	.50888
33	.62759	73	.21715	113	.82544	205	.09062
34	.54207	74	.13225	114	.74116	210	883.67297
						212	833.50567

Table B.3. Conversion Factors for Mass, Volume, and Mass and Volume Rate of Flow

VOLUME

Multiply Number of → By ↘ To Obtain ↓	Cubic centimeters	Liters	Cubic inches	Cubic feet	US gallons
Cubic centimeters	1	1.0000 x 10^{3}	16.387	2.8317 x 10^{4}	3785.4
Liters	1.0000 x 10^{-3}	1	1.6387 x 10^{-2}	28.316	3.7853
Cubic inches	6.1024 x 10^{-2}	61.025	1	1728	231
Cubic feet	3.5315 x 10^{-5}	3.5316 x 10^{-2}	5.7870 x 10^{-4}	1	0.13368
US gallons	2.6417 x 10^{-4}	0.26418	4.3290 x 10^{-3}	7.4805	1

All underlined figures are exact

VOLUME RATE OF FLOW

Multiply Number of → By ↘ To Obtain ↓	Cubic centimeters per minute	Liters per minute	Cubic inches per minute	Cubic feet per minute	US gallons per minute
Cubic centimeters per minute	1	1.0000 x 10^{3}	16.387	2.8317 x 10^{4}	3785.4
Liters per minute	1.0000 x 10^{-3}	1	1.6387 x 10^{-2}	28.316	3.7853
Cubic inches per minute	6.1024 x 10^{-2}	61.025	1	1728	231
Cubic feet per minute	3.5315 x 10^{-5}	3.5316 x 10^{-2}	5.7870 x 10^{-4}	1	0.13368
US gallons per minute	0.26417 x 10^{-3}	0.26418	4.3290 x 10^{-3}	7.4805	1

Table B.3. Conversion Factors for Mass, Volume, and Mass and Volume Rate of Flow [8] (Continued)

MASS

Multiply Number of → By → To Obtain ↓	Grams	Kilograms	Ounces*	Pounds*	Tons (short)
Grams	1	<u>1000</u>	28.350	453.59	0.90718 x 10^6
Kilograms	1×10^{-3}	1	28.350 x 10^{-3}	453.59 x 10^{-3}	0.90718 x 10^3
Ounces*	35.274 x 10^{-3}	35.274	1	16	32 x 10^3
Pounds*	2.2046 x 10^{-3}	2.2046	6.25 x 10^{-2}	1	2000
Tons (short)	1.1023 x 10^{-6}	1.1023 x 10^{-3}	3.125 x 10^{-5}	0.5 x 10^{-3}	1

All underlined figures are exact
* Avoirudupois

MASS RATE OF FLOW

Multiply Number of → By → To Obtain ↓	Grams per second	Kilograms per second	Kilograms per minute	Pounds* per second	Pounds* per minute
Grams per second	1	1000	16.667	453.59	7.5599
Kilograms per second	0.001	1	1.6667 x 10^{-2}	0.45359	7.5599 x 10^{-3}
Kilograms per minute	60 x 10^{-3}	60	1	27.216	0.45359
Pounds* per second	2.2046 x 10^{-3}	2.2046	3.6744 x 10^{-2}	1	1.6667 x 10^{-2}
Pounds* per minute	0.13228	132.28	2.2046	60	1

* Avoirudupois

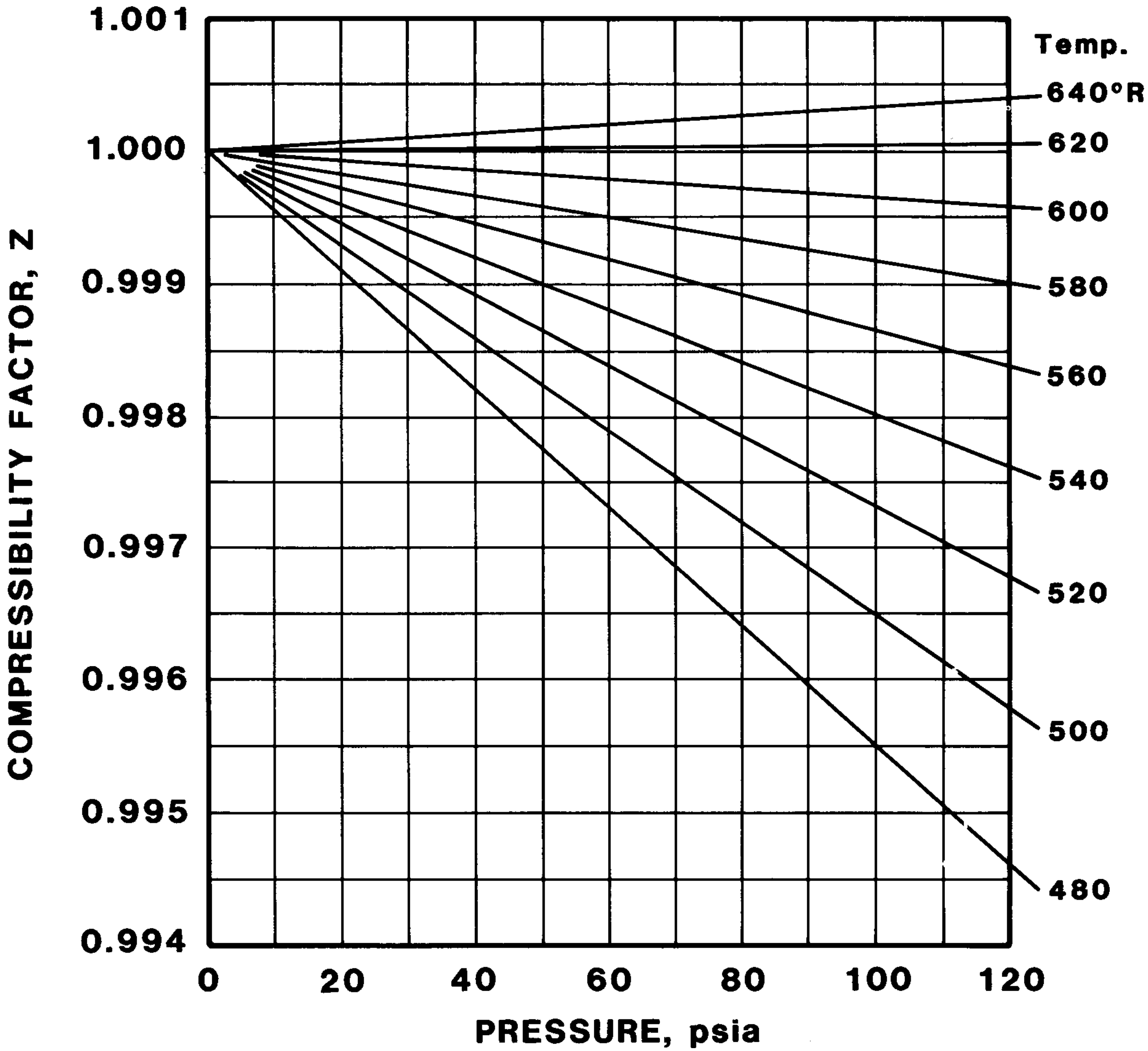

Figure B.1. Compressibility factor for air. [9,12]

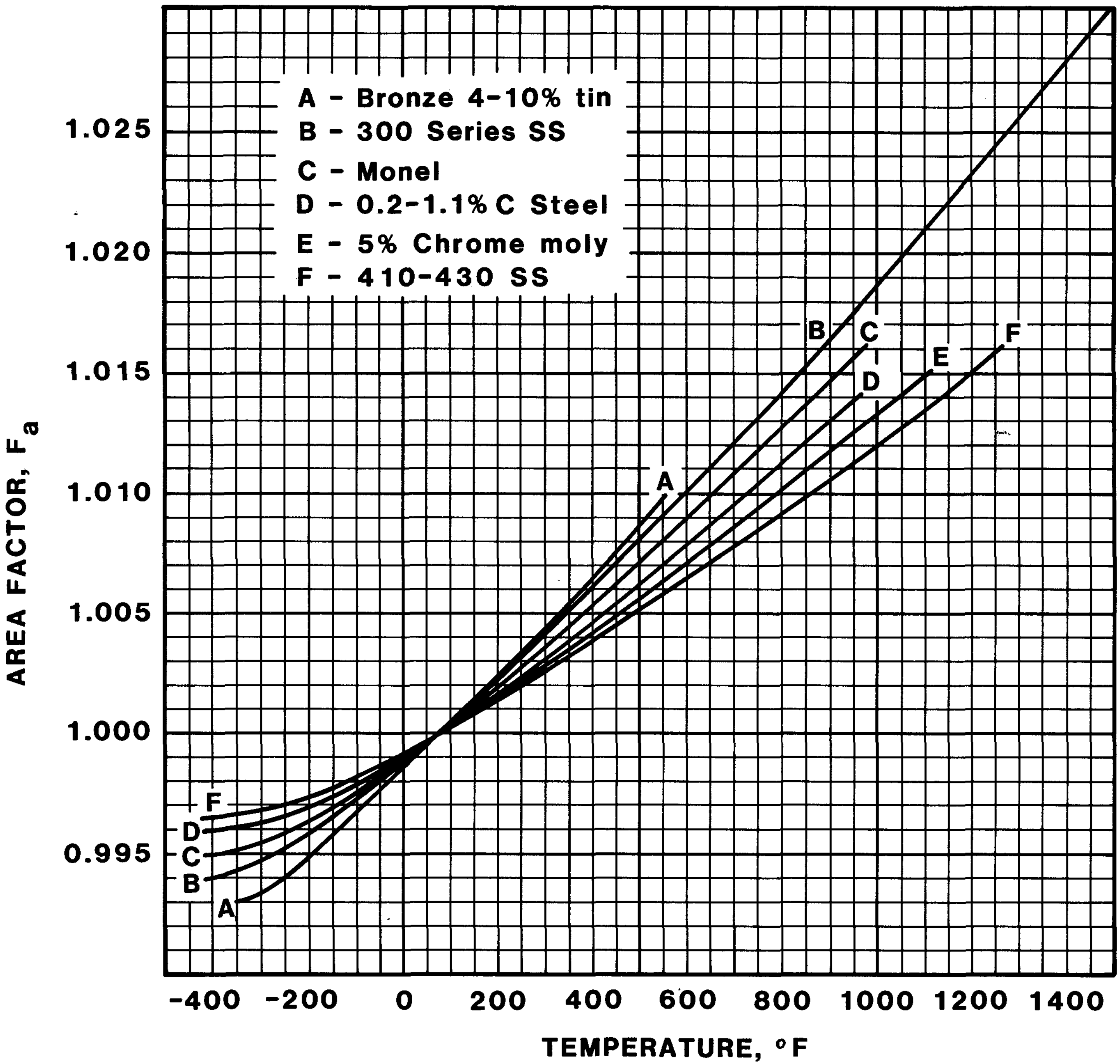

Figure B.2. Area factor F_a, for the thermal expansion of primary elements [2]

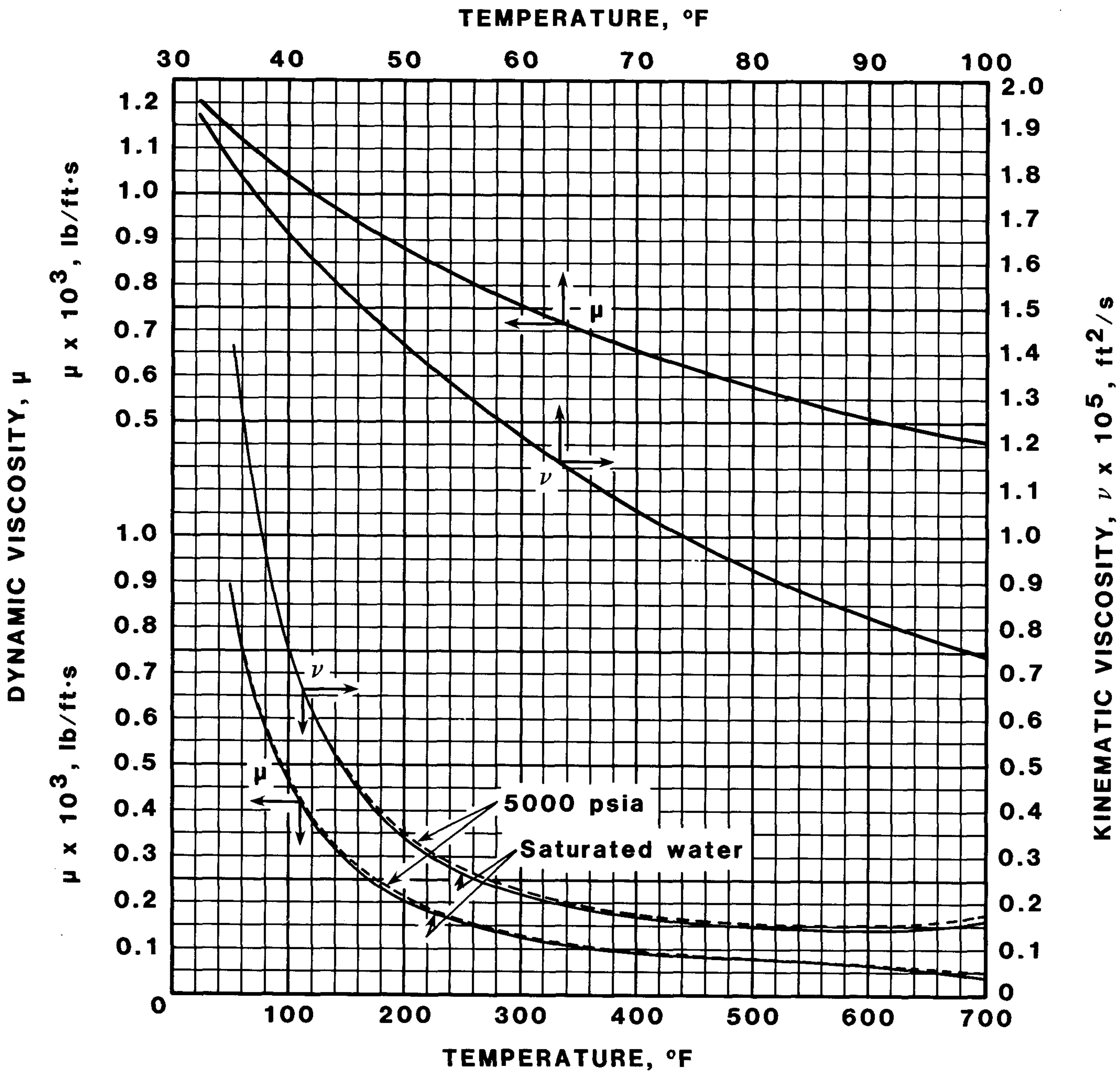

Figure B.3. Dynamic viscosity of water μ, and kinematic viscosity ν. Top curves expand the temperature scale from 30-100°F. Note the negligible effect of pressure. [2]

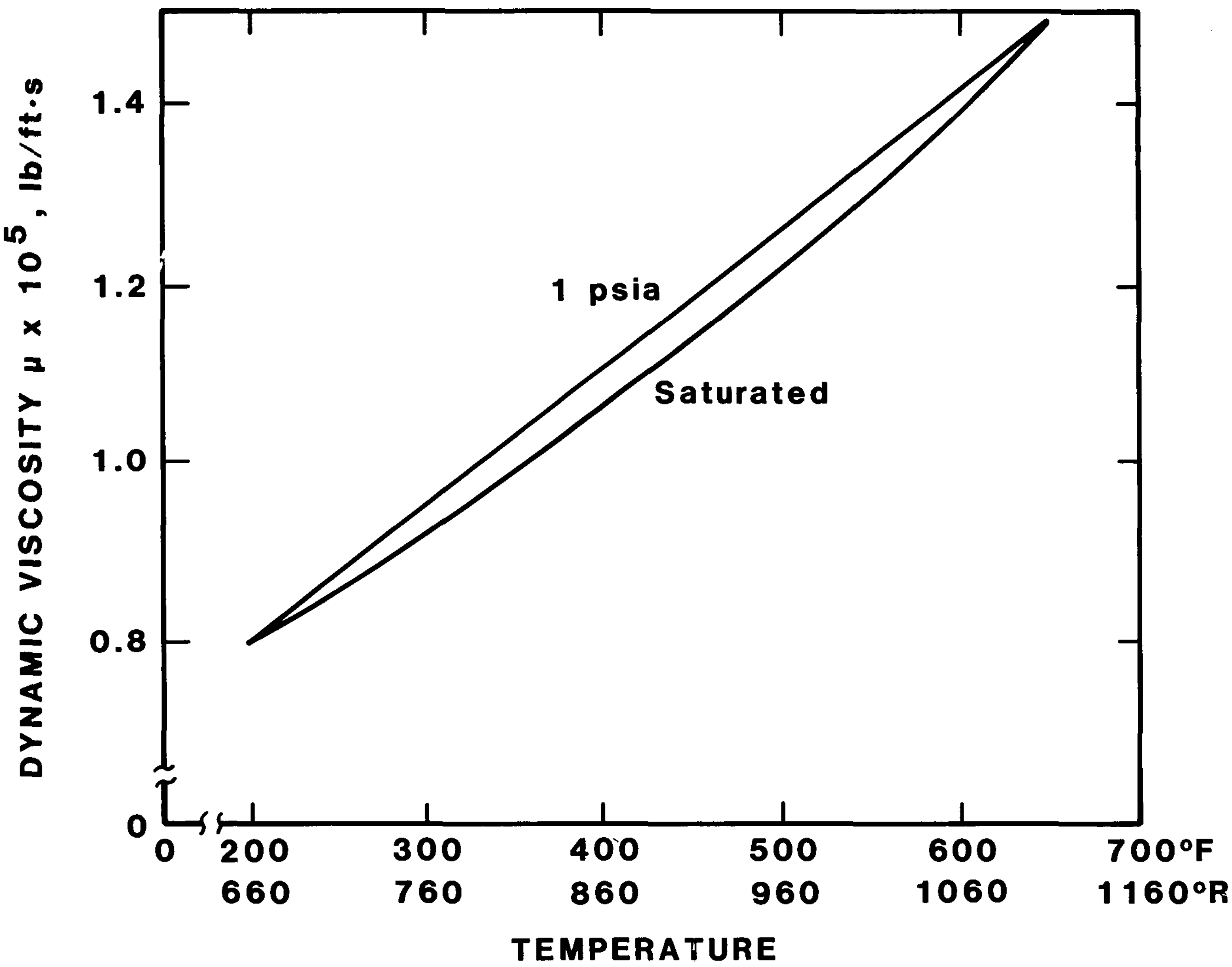

Figure B.4. Dynamic viscosity of saturated and superheated steam [6]

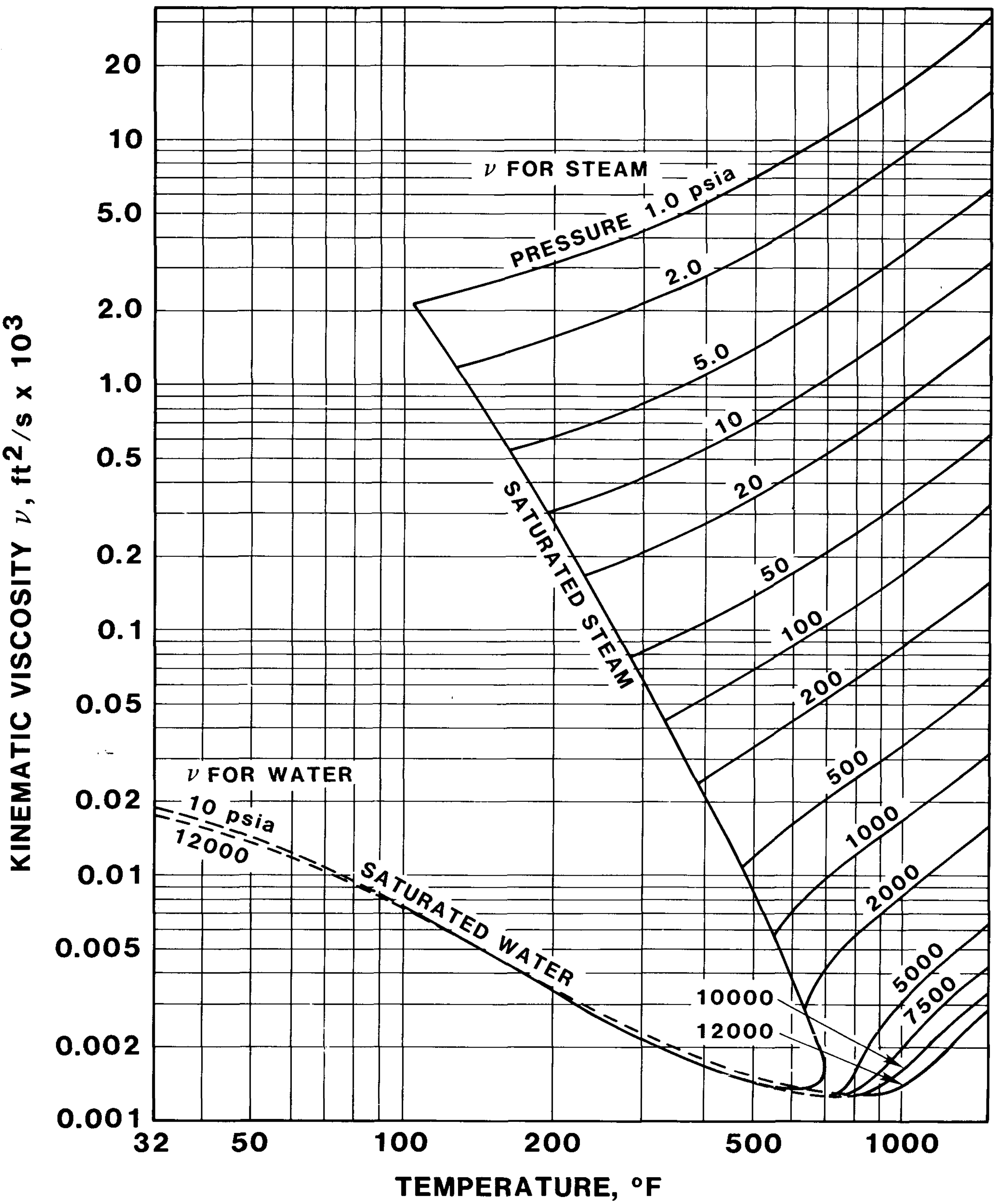

Figure B.5. Kinematic viscosities of steam and water [6]

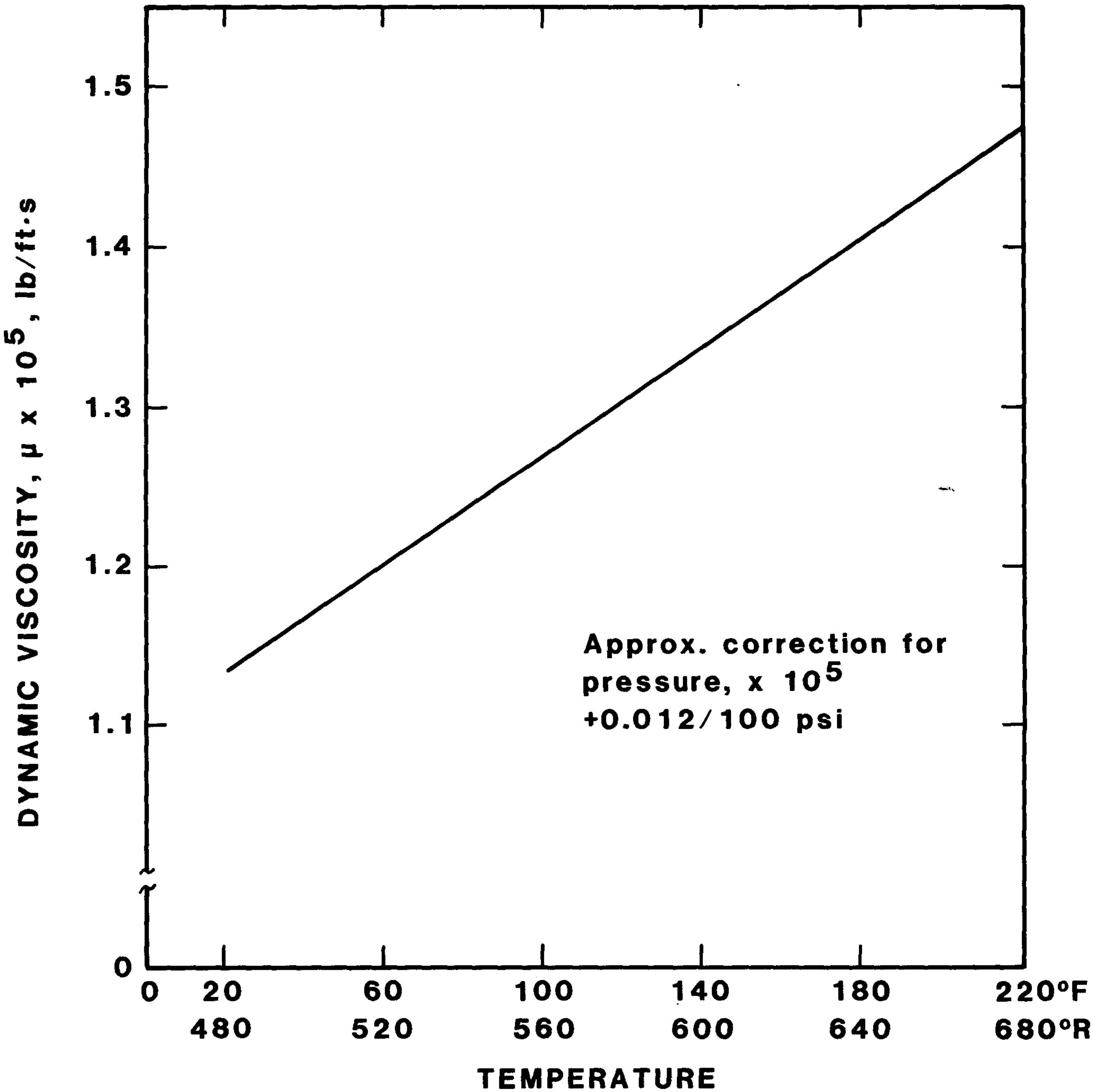

Figure B.6. Dynamic viscosity of air at 14.7 psia [12]

APPENDIX C. TEMPERATURE AND PRESSURE RELATIONS

C.1 TEMPERATURE

The units of temperature commonly used are the degree Fahrenheit, °F, and degree Celsius, °C; and the corresponding units of absolute temperatures, degree Rankine, °R, and the kelvin, K. Absolute temperature in degrees Rankine, designated by the symbol °R, will be used frequently in our compressible fluid calculations. Temperature in degrees Fahrenheit will be designated by the symbol t.

The relations existing among these different units of temperature are:

$$°F = \frac{9}{5}(°C) + 32 \qquad \text{(C-1)}$$

$$°C = \frac{5}{9}(°F - 32) \qquad \text{(C-2)}$$

$$°R = °F + 459.67 \qquad \text{(C-3)}$$

$$K = °C + 273.15 \qquad \text{(C-4)}$$

$$°R = \frac{9}{5}(K). \qquad \text{(C-5)}$$

C.2 PRESSURE

The unit of pressure to be used in our calculations is pounds force per square inch. We will also express differential pressure in terms of the pressure exerted by a column of water at 68 °F and subjected to standard acceleration of gravity, 32.1740 ft/s^2. It will be necessary to designate whether a value of pressure is:

a) Pressure difference between two points in a flow system

   ΔP in psid (pounds per square inch <u>differential</u>), or

   ΔP in inches of water at 68 °F;

b) A pressure relative to the existing atmospheric pressure, using negative values to indicate pressures below atmosphere

   p in psig (pounds per square inch <u>gauge</u>); or

c) Absolute pressure

   P in psia (pounds per square inch <u>absolute</u>).

In the direct measurement of pressure difference ΔP, it is necessary that both sides of the pressure sensing element be connected to pressure taps in the flow system. The pressure difference between these two taps will be derived from the instrument indication. Instruments used for the measurement of pressure difference include liquid manometers, differential pressure gauges of the elastic element type, and ΔP transducers using electrical sensing elements.

Instruments used for the sensing of pressure p relative to atmospheric pressure have one side of the pressure sensing element vented to the atmosphere. Instruments used here include manometers, Bourdon tube and diaphragm capsule gauges, and electrical transducers.

For the direct measurement of absolute pressure P, it is necessary that one side of the pressure sensing element be exposed to zero absolute pressure. Barometers are one form of such instruments. However, the absolute pressure of a point in a flow system usually is not measured directly. Rather, gauge pressure p in psig and barometric pressure P_b in psia are measured. The absolute pressure is then computed by the relation

$$P = p + P_b = \text{absolute pressure in psia.} \qquad \text{(C-6)}$$

Barometric pressure, absolute pressure, gauge pressure, and differential pressure are sensed by many different types of instruments which read out in many different units of pressure or head. These instruments will be classified into three types: namely, Mechanical Pressure Gauges, Manometers, and Electrical Pressure Transducer Systems, defined as follows:

Mechanical Pressure Gauges contain an elastic element such as a Bourdon tube, diaphragm capsule, or spring bellows having a free end whose position changes with variation in pressure and a mechanical linkage system to transmit these variations in position to a revolving pointer located adjacent to a scale calibrated in units of pressure. Such a device senses and indicates directly, and will measure pressure difference ΔP, gauge pressure p, or absolute pressure P depending upon its design.

Manometers contain a liquid such as water or mercury in which the difference in liquid levels increases with increased pressure difference. In elementary designs the positions of the liquid levels are indicated by a scale, calibrated in units of length, placed beside the manometer column(s). Thus, this device senses pressure directly, but its indication, i.e., the difference in liquid levels, is only proportional to pressure. Like the pressure gauge, a manometer will measure pressure difference, gauge pressure, or absolute pressure depending upon its design.

Electrical Pressure Transducer Systems use electrical sensing elements such as metallic or semiconductor strain gauges, variable capacitance or variable inductance devices, differential transformers, potentiometers, or piezoelectric elements to sense deformation of the elastic element. The electrical output of these elements is usually an analog signal which is processed using electronic circuitry producing an analog output such as 4-20 mA DC or 0-10 volts DC, or a variable frequency digital signal.

C.2.1 Corrections and Use of Conversion Factors for Mechanical Pressure Gauges

Calibration of a pressure gauge may show a measurable scale error which varies with gauge reading. When significant, scale error corrections should be applied to change the indicated readings to actual pressures. Aside from applying the scale error correction, it is only necessary to convert from units of pressure in which the gauge is calibrated to pounds per square inch (psi) or to inches of water at 68 °F (h_w). When gauges are calibrated in units of pressure other than psi, the Conversion Factors for Pressure, table C-1, will apply.

C.2.2 Corrections and Use of Conversion Factors Manometers

In converting an indicated level difference or height, h, of a manometer column to pressure in psi, it is necessary to consider the following:

a) The units of length in which the height is expressed (inches, centimeters, etc.),

b) The manometer liquid (water, mercury, etc.),

c) The temperature of the manometer liquid,

d) The density of the fluid in the high pressure leg of the manometer. (This correction is usually applied only when the manometer is used to measure pressure differential between two pressures taps in a flow system.)

e) The local acceleration due to gravity, g_L.

At times, corrections d) and e) will be too small to affect significantly the accuracy of the pressure determination. They may be omitted when such is the case. Correction procedures for each of these five items will now be discussed.

Corrections for Units of Length of the Manometer Scale.

When the scale of the manometer is calibrated in units of length other than inches, it is convenient to convert the length to inches using the conversion factors of figures C.1 and C.2.

Combined Conversion Factor for the Manometer Liquid and its Temperature

A conversion or multiplication factor F_{wl} will be used to convert inches of water at a known temperature to pounds per square inch. Similarly, a multiplication factor F_{ml} will be used to convert inches of mercury at a known temperature to pounds per square inch. Values of these two factors may be obtained from figures C.1 and C.2 for water and mercury manometers, respectively, by entering these figures with the temperature of the liquid in the manometer column. This temperature is usually equivalent to room temperature. Figures C.1 and C.2 cover just the manometer temperature range 50 °F and 110 °F, but a larger range 32 °F to 200 °F is covered in tables 12 and 16 of reference 7, for water and mercury, respectively.

Correction for the Density of the Fluid in the High Pressure Leg of the Manometer

Usually a differential pressure is measured simply by the height of the liquid column. Actually, when air is in the high pressure leg, the differential pressure is proportional to the difference of the densities of the liquid column and of the column of equal height of the air on the high pressure side.

Multiplication factors, F_{w2} for a water manometer and F_{m2} for a mercury manometer, will be used to correct for the density of the air in the high pressure leg or well of a manometer used to measure differential pressure between two pressure taps in a flow system. Values of these factors are given in figures C.3 and C.4 for the water and mercury columns, respectively. These figures apply for manometer temperatures in the approximate range of 60 °F to 100 °F. For practial purposes, the factor F_{m2} may have an assumed value of 1.000 except when the manometer is used to measure differential presure between two pressures taps in a flow system at an elevated pressure.

When liquid water is in the high pressure leg of a mercury manometer, the multiplication factor F_{m3} will be used to correct for the density of the water. A value of $F_{m3} = 0.9263$ may be used for the range of 60 °F to 100 °F manometer temperature and pressures to 500 psia.

Correction for the Local Value of the Acceleration Due to Gravity

A multiplication factor F_g will be used to correct the manometer reading for the local acceleration due to gravity. The value of this correction factor may be determined by the relation

$$F_g = g_L/32.1740 \qquad \text{(C-7)}$$

where g_L is the value of the local acceleration due to gravity existing in the laboratory and expressed in ft/s^2.

C.2.3 Equations for Converting Manometer Readings to Pressure in Pounds Per Square Inch

A water manometer reading is converted to psi by the relation:

$$\text{psi} = (\text{inches of water})\ (F_{w1}\ F_{w2}\ F_g) \qquad \text{(C-8)}$$

A mercury manometer reading is converted to psi by the relation:

$$\text{psi} = (\text{inches of mercury})\ (F_{m1}\ F_{m2}\ F_g) \qquad \text{(C-9)}$$

When liquid water is in the high pressure leg, a mercury manometer reading is converted to psi by the relation:

$$\text{psi} = (\text{inches of mercury})\ (F_{m1}\ F_{m3}\ F_g) \qquad \text{(C-10)}$$

C.2.4 Equations for Converting Manometer Readings to Inches of Water at 68 °F, h_w

In all cases:

$$h_w = 27.729 \text{ (psi)} \qquad \text{(Table C.1)}$$

The following may also be used. A water manometer reading is converted to h_w by the relation:

$$h_w = (\text{inches of water}) (\rho/\rho_{68}) (F_{w2}\ F_g) \qquad \text{(C-11)}$$

where ρ is the density of water at the manometer temperature, and ρ_{68} is the density of water at 68 °F.

A mercury manometer reading is converted to h_w by the relation:

$$h_w = (\text{inches of mercury}) (\rho_{Hg}/\rho_{68}) (F_{m2}\ F_g) \qquad \text{(C-12)}$$

where ρ_{Hg} is the density of mercury at the manometer temperature.

When liquid water is in the high pressure leg, a mercury manometer reading is converted to h_w by the relation:

$$h_w = (\text{inches of mercury}) (\rho_{Hg}/\rho_{68}) (F_{m3}\ F_g) \qquad \text{(C-13)}$$

C.2.5 Barometers and Barometric Pressure

It is essential that the barometric or ambient air pressure existing within the laboratory be determined with good accuracy during tests involving the flow of compressible fluids. For computations here, the pressure P_b of the existing atmosphere above zero is expressed in pounds force per square inch absolute, psia. Barometric pressure varies continually throughout the day and from day-to-day depending upon outside atmospheric conditions and the ventilation equipment used to supply air to the laboratory. Normally, P_b at sea level will be within the range 14.2 to 14.9 psia. Exceptions will be noted when severe atmospheric disturbances occur or in laboratories having elevations appreciably different from sea level, e.g., several hundred feet. Two different types of barometers, aneroid and mercury column, are commonly used. Their designs and the conversion of their readings to psia are discussed below.

Aneroid Barometers are essentially a refined version of a mechanical pressure gauge designed to measure absolute pressure. They contain an evacuated diaphragm capsule whose configuration varies with changes in ambient air pressure together with a mechanical linkage system which moves an indicating pointer positioned adjacent to a calibrated scale. The scale units are usually inches of mercury at 32 °F or millimeters of mercury at 0 °C.

Readings from the aneroid barometer do not require corrections for the local acceleration due to gravity. Also, temperature correction factors applicable to mercury columns must not be applied to readings of this instrument. If the aneroid barometer is not self-compensating for changes in ambient temperature, a temperature correction chart or table will be required.

The special temperature correction, if any, as well as any correction for scale error should be applied to the reading of the aneroid barometer to obtain a corrected barometer reading. Conversion factors for pressure from table C.1 are then applied to the corrected reading to convert from the instruments units to psia.

Mercury Column Barometers contain a reservior, vented to the atmosphere, and a vertical column of mercury. A brass scale, calibrated in inches or millimeters, is placed beside the mercury column; and arrangements are provided for the adjustment of this scale prior to each reading.

To obtain an accurate measure of the barometric pressure, the reading from a mercury barometer must be corrected for the thermal expansion of both the mercury and the brass scale, and for the local acceleration due to gravity. It is also necessary to convert the corrected reading of length into pressure units, pounds per square inch absolute, psia.

Tables 17 and 18 of reference 7 give Temperature Corrections for Mercury Barometers with Brass Scales. Correction factors obtained from these tables compensate for the combined thermal expansion of both the mercury and the brass scale. The temperature to be used in entering these tables is the temperature of mercury in the barometer. This is usually indicated by a thermometer permanently attached to the instrument.

C.2.6 Corrections and Use of Conversion Factors for Electrical Pressure Transducers

The pressure transducer system may include readout instrumentation indicating pressure directly, or the output may be an analog signal such as 4-20 mA DC or 0-10 volts DC, or it may be a variable frequency signal. In any case, calibration may show a measurable error which varies with pressure or pressure differential. When significant, error corrections should be applied to change the indicated reading to actual pressures, or corrections should be applied to the electrical output.

The transducer should be calibrated periodically, preferably on site, first to establish and then to monitor any significant errors. The calibration equipment may range in complexity from liquid manometers (vertical, inclined, or micro type using water, mercury, or special manometer fluids) to laboratory reference type pressure transducer equipment incorporating a digital readout and automated computational features.

When the pressure transducer is part of a flowmeter system, its output signal may be expressed in the meter calibration factor CF, for example:

$$CF = GPM/(mA)^{1/2} \tag{C-14}$$

The mA value may refer to transducer (or transmitter) output current, uncorrected. Although corrections would not be made directly, it is still very important that a calibration program be established to determine the transducer error and that the transducer performance be monitored by periodic calibration checks. Otherwise, the meter calibration factor will no longer represent the actual system performance.

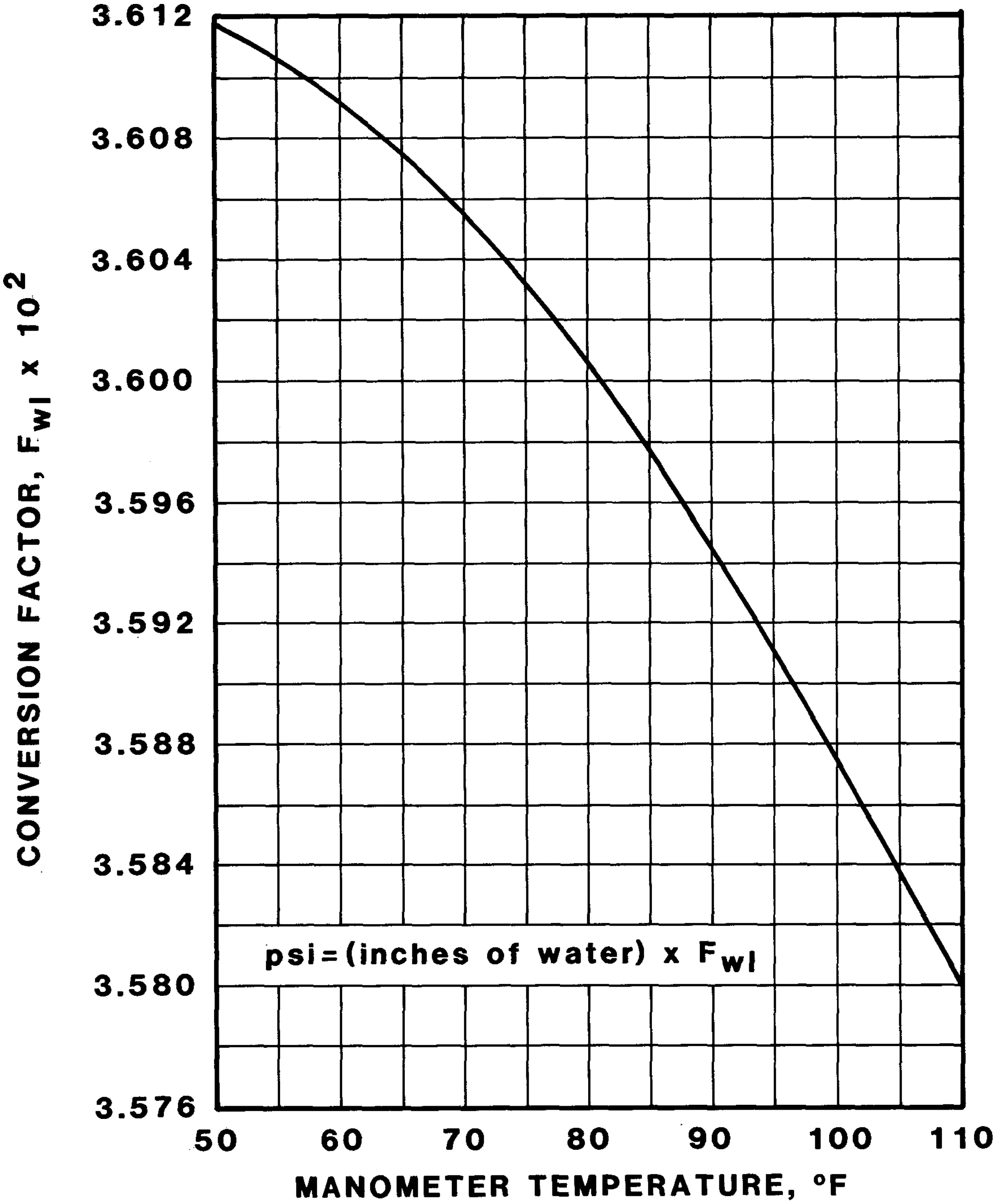

Figure C.1. Conversion factor F_{wl} for water columns [7,9]

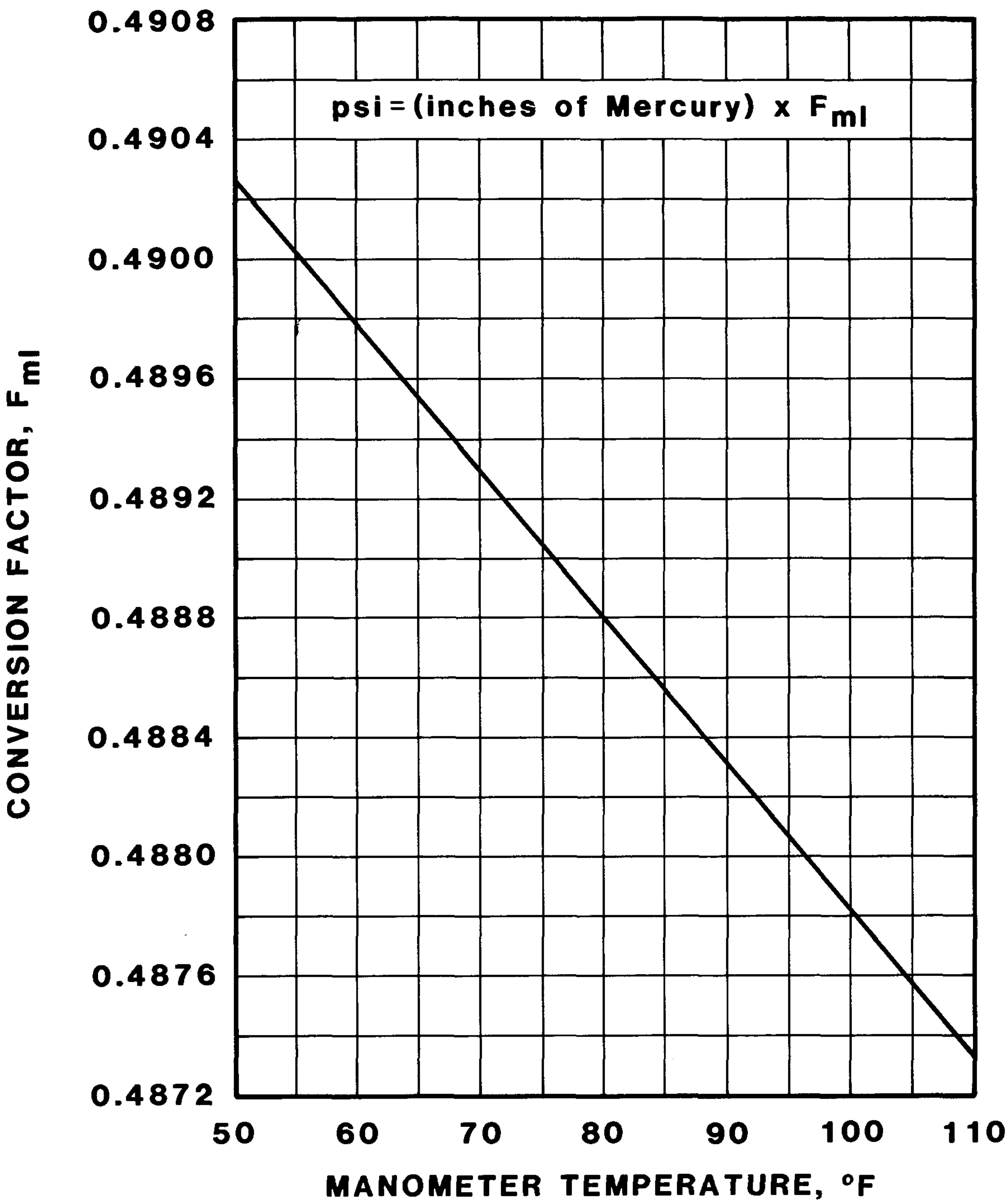

Figure C.2. Conversion factor F_{ml} for mercury columns [7,9]

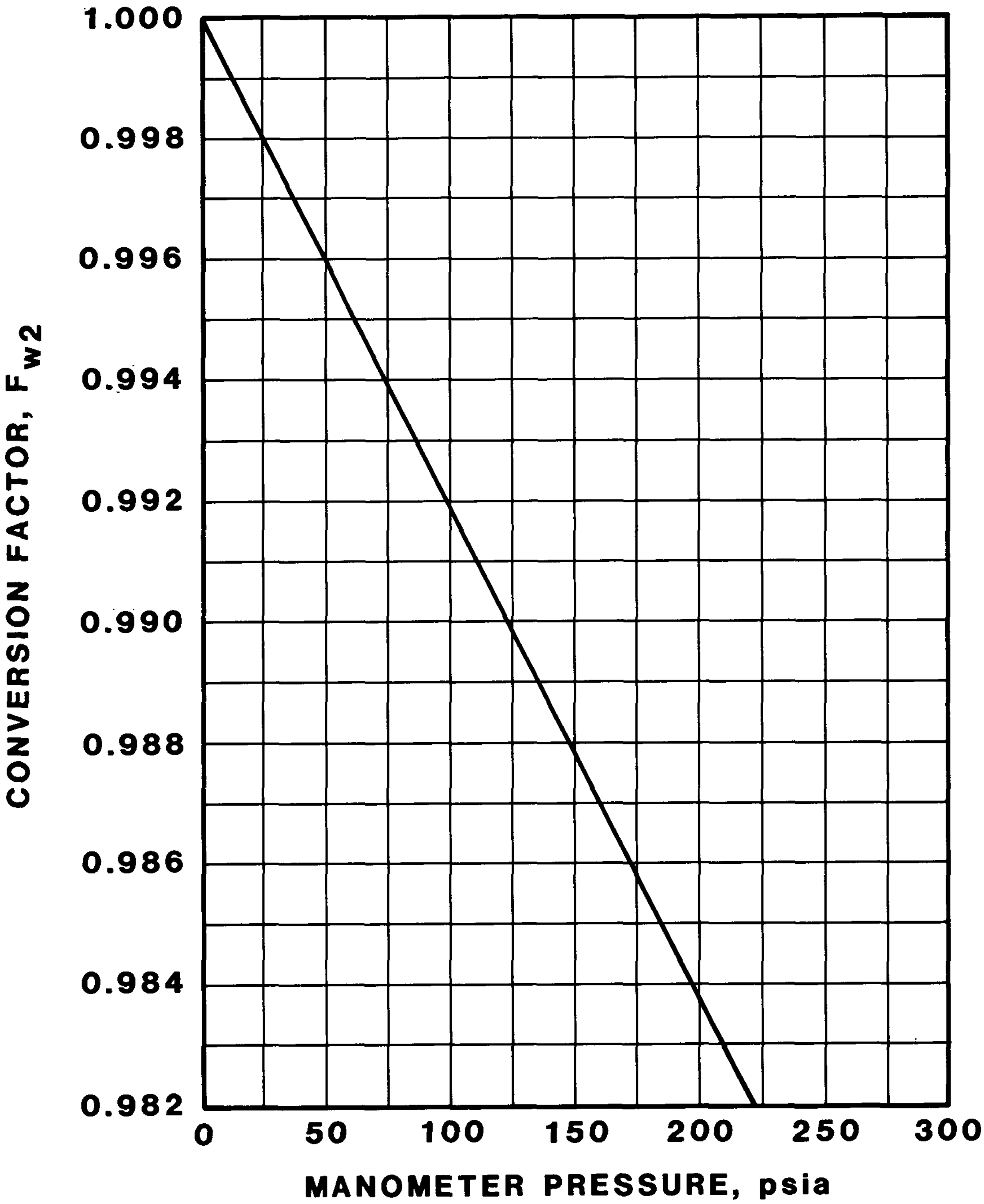

Figure C.3. Water manometer correction factor F_{w2} for air column density [7,9]

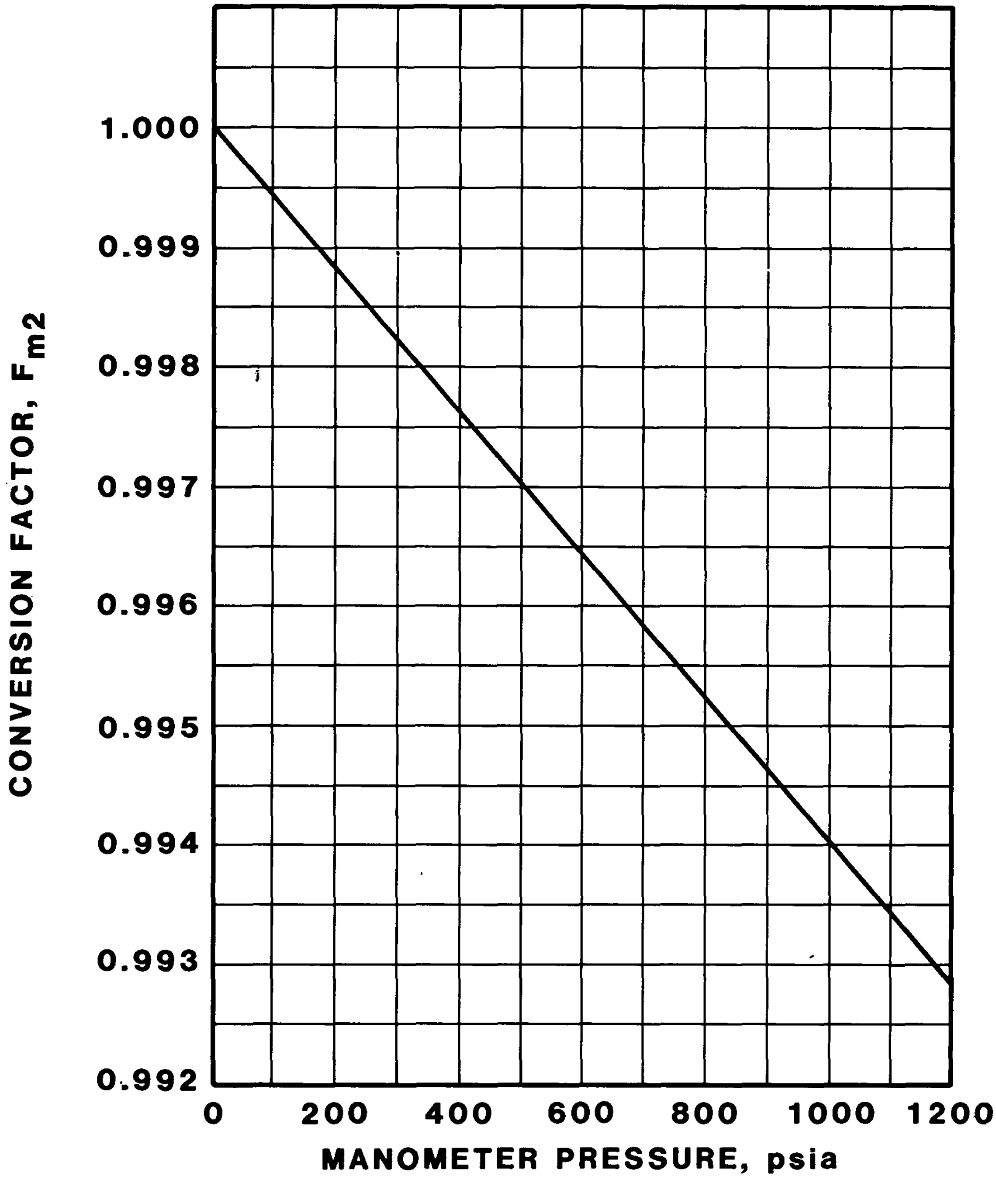

Figure C.4. Mercury manometer correction factor F_{m2} for air column density [7,9]

Table C.1. Conversion Factors for Pressure [7]

Multiply Number of → By To Obtain ↓	Millimeters of mercury at 0 °C	Inches of water at 68 °F	Inches of mercury at 32 °F	Pounds per square inch	Atmospheres*
Millimeters of mercury at 0 °C	1	1.8650	25.4	51.715	760
Inches of water at 68 °F	0.53620	1	13.620	27.729	407.5
Inches of mercury at 32 °F	3.9370×10^{-2}	7.3424×10^{-2}	1	2.0360	29.921
Pounds per square inch	1.9337×10^{-2}	3.6063×10^{-2}	0.49116	1	14.696
Atmospheres*	1.3158×10^{-3}	2.4539×10^{-3}	3.3421×10^{-2}	6.8046×10^{-2}	1

* One atmosphere = 760 mm of mercury at 0 °C = 14.696 pisa.

All liquid columns are assumed subjected to standard gravity, 32.1740 ft/s^2.

APPENDIX D. RELATION BETWEEN MASS AND VOLUME RATE OF FLOW

When calibrating a working meter on site, the occasion may arise when the flow at the transfer conditions of temperature and pressure need to be expressed in terms of the flow at a different temperature and/or pressure existing at the working meter; or at a single station, a mass flowrate may need conversion to a volume flowrate. Thus, the relation between mass and volume rate of flow needs consideration.

Under steady-flow conditions, the mass flowrate M is constant throughout a system provided no leakage occurs. For example in figure 5, using subscripts to denote positions with the system, the conservation of mass flowrate M may be stated by:

$$\text{Mass flowrate} = M = M_1 = M_2 = M_3 = M_4 = \ldots\ldots = M_x \qquad \text{(D-1)}$$

The volume rate of flow Q_x at any point x in the system is the mass flowrate divided by the fluid density ρ_x at that point, or

$$Q_x = M/\rho_x \text{ from which} \qquad \text{(D-2)}$$

$$Q_1 = M/\rho_1;\ Q_2 = M/\rho_2;\ \ldots\ldots\ Q_x = M/\rho_x$$

When water (incompressible fluid) is flowing, density ρ will be essentially constant at building system pressure levels and

$$Q_1 = Q_2 = Q_3 = Q_4 = \ldots\ldots = Q_x \qquad \text{(D-3)}$$

When a gas (air, steam) is flowing, the density ρ_x varies point to point as a function of the temperature and pressure. Since

$$M = M_1 = M_2 = \ldots\ldots = M_x, \text{ then}$$

$$(\rho_1\ Q_1) = (\rho_2\ Q_2) = \ldots\ldots = (\rho_x\ Q_x) \qquad \text{(D-4)}$$

D.1 DENSITY OF DRY AIR, STEAM, AND MOIST AIR

With dry air, the density ρ can be calculated from an equation of state:

$$\rho = P/(Z\ R\ T)$$

where P is the absolute pressure and T, the absolute temperature.

The quantity R is known as the "gas constant" with its value different for each gas and also dependent on the units used in expressing ρ, P and T. The quantity Z is known as the "compressibility factor." It corrects for small variations in the behavior of real gases compared to ideal gases, and its value depends on the gas, and its temperature and pressure. The value of Z is dimensionless, and it is not dependent upon the units of pressure and temperature. A graph giving the Compressibility Factor Z for dry air as a function of P and T is given in appendix B, figure B-1.

For dry air, with P in units pounds force per square inch absolute, psia, and T, the absolute temperature in degrees Rankine °R, the relation for density is

$$\rho = 2.6990\ P/(Z\ T)\ \mathrm{lb/ft^3}, \tag{D-5}$$

where 2.6990 equals the molecular weight of dry air (28.964 lb/lb-mole) divided by the Universal Gas Constant $\bar{R}$,(10.7315) (psia•ft^3)/(lb-mole °R),

and lb has units pounds mass.

With steam, the properties including specific volume, ft^3/lb, are given in tabular form as a function of temperature and pressure in reference 6. Thus:

$$\rho_v = 1/(\text{specific volume})\ \mathrm{lb/ft^3}, \tag{D-6}$$

where ρ_v is the density of steam or water vapor.

For moist air, a relation for the density ρ_m of the mixture is:

$$\rho_m = \frac{2.6990\ (P - P_v)}{(Z\ T)} + \rho_v\ \mathrm{lb/ft^3} \tag{D-7}$$

where $P - P_v$ = partial pressure of the air, and
P_v = partial pressure of the water vapor

The relation for ρ_m is valid at low values of P_v where the behavior of steam approaches ideal gas behavior. The value of P_v depends on the relative humidity, RH, expressed in percent and P_g, the saturation pressure for steam at temperature T, as follows:

$$P_v = (RH/100)\ P_g \tag{D-8}$$

Example calculations E.1, E.2, and E.3 in appendix E deal with relations between mass and volume rate of flow. Example E.4 gives results of calculations for ρ_m for moist air at several pressures, temperatures and moisture contents.

It is noted for both air and steam flow calculations, when the actual volume flowrate at a definite temperature and pressure is known, the volume flowrate may be determined at some other location having a definite temperature and pressure with computation of the mass flowrate unnecessary, i.e.,

$$Q_x = (\rho_1\ Q_1)/\rho_x. \tag{D-9}$$

With air flow, substituting equation (D-5) gives:

$$Q_x = \frac{(P_1\ Z_x\ T_x)}{(P_x\ Z_1\ T_1)}\ Q_1\ , \tag{D-10}$$

since R for any gas is constant. Thus, this relation is convenient to use when mass flowrate is neither known nor required, and volume flowrate at a definite temperature and pressure is known.

D.2 "STANDARD CONDITIONS," ITS MEANING AND USE

As discussed above and shown in example E.3, the actual volumetric flowrate ACFM is not a convenient nor direct measure of the quantity of matter because the numerical value varies with both temperature and pressure. The absolute pressure and temperature of the gas at the particular location must also be stated if the actual volume at that location is to be meaningful in terms of the quantity of matter.

It has also been noted that mass flow is a direct measure of the quantity of matter and its value is independent of the pressure and temperature at the location under consideration.

One other quantity is used, especially in commerical transactions, as a direct indication of the quantity of matter involved in gaseous flow. This is volume flow converted to selected standard conditions of absolute pressure P_s and absolute temperature T_s. Values of volume flow converted to these standard conditions will be referred to as standard cubic feet per minute, SCFM.

The selected standard conditions of pressure and temperature are not necessarily the same for all industries or for all laboratories. Any organization may select any value of P_s and T_s for its standard conditions; however, to prevent confusion, once values for P_s and T_s are selected, they should be used without exception. When two organizations select different standard conditions of pressure and temperature, their values of SCFM will not concur because their standard cubic feet will refer to different quantities of matter. Therefore, the value of SCFM has no definite, accurate meaning unless the values of P_s and T_s associated with the value of SCFM are definitely known.

From the preceding, it should be understood that the "Standard" in SCFM is not standard at all but varies among different industries and laboratories. Hence, this term SCFM has a tendency to confuse as well as help.

APPENDIX E. ILLUSTRATIVE EXAMPLES

Example E.1 Mass and Volume Rates of Flow, Superheated Steam. Steam flows at a mass rate of flow of 2000 lb/hr in a pipe. It is desired to calculate the actual volume rate of flow, cubic feet per minute (ACFM) at two stations, 1 and 2. Pressures are measured with bourdon tube gauges, with cases vented to the atmosphere. Following are the measurements made:

Measurement	Instrument	Reading
Barometric pressure	Aneroid barometer, inches of mercury at 32 °F	29.83 in. Hg at 32 °F
Pressure p_1	Bourdon tube gauge	35.0 psig
Temperature t_1	Resistance thermometer, output converted to °F	300 °F
Pressure p_2	Bourdon tube gauge	15.0 psig
Temperature t_2	Resistance thermometer	260 °F

Compute barometric pressure P_b in psia. No corrections for temperature or local acceleration due to gravity are applied to the aneroid barometer reading. Refer to table C-1 for the conversion factor for P_b.

$$P_b = (0.49116)\ (\text{inches of mercury at } 32\ ^\circ\text{F})$$

$$P_b = (0.49116)\ (29.83) = 14.65 \text{ psia}$$

Computation for station 1:

$$P_1 = (p_1 + p_b) = (35.0 + 14.65) = 49.65 \text{ psia} \qquad \text{(C-6)}$$

To determine ρ_1, use data from steam tables, reference 6, table 3. Interpolation of data for 300 °F gives:

P, psia	v, ft^3/lb
48	9.147
49.65	8.835
50	8.769

The specific volume at station 1 is:

$$v_1 = 8.835 \text{ ft}^3/\text{lb} = 1/\rho_1$$

$$\rho_1 = 1/8.835 = 0.11319 \text{ lb/ft}^3$$

The volume flowrate is:

$$Q_1 = M/\rho_1 = (2000)/(60)(0.11319) = 294.5 \text{ ACFM, or} \tag{D-2}$$

$$Q_1 = (M\ v_1) = (2000)(8.835)/(60) = 294.5 \text{ ACFM}$$

Computation for station 2:

$$P_2 = (p_2 + P_b) = (15.0 + 14.65) = 29.65 \text{ psia}$$

Interpolation at $t_2 = 260\ ^\circ F$:

P, psia	v, ft^3/lb
29.00	14.447
29.65	14.127
30.00	13.954

$$v_2 = 14.127\ ft^3/lb$$

$$Q_2 = (M\ v_2) = (2000)(14.127)/(60) = 470.9 \text{ ACFM}$$

The actual volume rate of flow at station 1 is 295.4 ft^3/minute and at station 2, it is 470.9 ft^3/minute. It should be noted that while the mass flowrate is constant, the volume flowrate is not constant, but varies with the density or specific volume.

Example E.2 <u>Mass and Volume Rates of Flow, Wet Steam</u>. This example deals with wet steam, that is, steam in the saturation state which contains both liquid and vapor. While it is outside the scope of this document, a basic computation for mass and volume flowrates follows. Consider steam flowing at a flowrate of 100 ACFM, at a temperature of 288 °F, and a pressure of 55.8 psig. It is desired to compute the mass flowrate in lb/hr.

Referring to reference 6, it is noted that the temperature and pressure correspond to saturation conditions. Therefore, no determination of density is possible with the information given. In addition, the steam quality "x" (ratio of the mass of vapor to the mass of mixture) is needed.

Assume the quality x is 97.0 percent.

For the saturated state, the specific volume is

$$v = (1 - x)(v_f) + (x)(v_g) \tag{E2-1}$$

where x is expressed as a decimal, v_f is the specific volume of saturated liquid, and v_g is the specific volume of the saturated vapor.

Using equation E2-1 and data in reference 6, table 1, at temperature 288 °F, the specific volume is

$$v = (0.03)(0.0174) + (0.97)(7.6807), \text{ or}$$

$$v = 7.451 \text{ ft}^3/\text{lb}$$

The mean flow rate, M, is:

$$M = (\rho\ Q) = Q/v$$

$$= (100)(60)/(7.451), \text{ or}$$

$$M = 805.3 \text{ lb/hr}$$

Example E.3 Mass and Volume Rates of Flow, Dry Air. Dry air flows through a system as shown in figure 5. Valve V1 is closed. The mass flowrate is known to be 18.00 pounds per minute. It is desired to determine the flowrate entering Q_1, and leaving, Q_4 in actual cubic feet per minute, ACFM. Local acceleration due to gravity is 32.150 ft/s^2. Pressures and temperatures have been measured as follows:

Measurement	Instrument	Reading
Barometric pressure	Aneroid barometer, in. Hg at 32°F	29.836 in. Hg at 32 °F
Pressure p_1	Bourdon tube gauge, case vented to atmosphere, psig	105.0 psig
Temperature t_1	Mercury thermometer, °F	73.7 °F
Pressure h_4	Mercury manometer, low pressure leg vented to atmosphere, in. Hg	55.47 in. Hg at 81.0 °F
Temperature t_4	Mercury thermometer, °C	23.5 °C

Compute the barometric pressure P_b in psia. Note that corrections for instrument temperature and local acceleration due to gravity are not applied to the aneroid barometer reading.

$$P_b = 0.49116 \text{ (inches of mercury at 32 °F)} \qquad \text{(Table C.1)}$$

$$= (0.49116)(29.836), \text{ or}$$

$$P_b = 14.65 \text{ psia}$$

Computation for location 1:

$P_1 = (p_1 + P_b) = (105.0 + 14.65) = 119.65$ psia (C-6)

$T_1 = (t_1 + 459.67) = (73.7 + 459.67) = 533.37$ °R (C-3)

$Z_1 = 0.9973$ (for air at 120 psia and 533 °R) (Figure B.1)

$\rho_1 = 2.6990\ P_1/(Z_1T_1) = \frac{(2.6990)(119.65)}{(0.9973)(533.37)}$, or (D-5)

$\rho_1 = 0.6071\ \text{lb/ft}^3$

$Q_1 = M/\rho_1 = (\text{lb/min})/(\text{lb/ft}^3)$ (D-2)

$Q_1 = (18.00)/(0.6071) = 29.65\ \text{ft}^3/\text{min}$ (at 119.65 psia and 533.4 °R)

Computation for location 4:

Determine the mercury manometer correction factors F_g, F_{m1} and F_{m2}

$F_g = g_L/32.1740 = 32.150/32.1740 = 0.9993$ (C-7)

$F_{m1} = 0.48873$ psi/in. Hg (at 81.0 °F) (Figure C.2)

$F_{m2} = 0.9998$ (at $P_4 = 42$ psia) (Figure C.4)

$p_4 =$ (inches of mercury at 81 °F) $(F_{m1}\ F_{m2}\ F_g)$ (C-9)

$= (55.47)(0.48873)(.9998)(0.9993)$ or

$p_4 = 27.09$ psig

$P_4 = p_4 + P_b = 27.09 + 14.65 = 41.74$ psia (C-6)

$t_4 = (°C)(9/5) + 32 = 23.5(9/5) + 32 = 74.3$ °F (C-1)

$T_4 = t_4 + 459.67 = 74.3 + 459.67 = 533.97$ °R (C-3)

$Z_4 = 0.9991$ (air at 42 psia and 534 °R) (Figure B.1)

$\rho_4 = 2.6990\ P_4/(Z_4T_4) = 2.6990(41.74)/[(0.9991)(533.97)]$ (D-5)

$\rho_4 = 0.2112\ \text{lb/ft}^3$

$Q_4 = M/\rho_4$ or

$Q_4 = 18.000/0.2112 = 85.23\ \text{ft}^3/\text{min}$ (at 41.74 psia and 534.0 °R)

Thus, Q_1 is 29.65 ft^3/min at 119.65 psia and 533.4 °R and Q_4 is 85.23 ft^3/min at 41.75 psia and 534.0 °R. It should be noted that the quantity of air flow is constant at 18.00 lb/min. However, the volume rate of flow is not constant throughout the system, but rather varies with temperature and pressure. Thus whenever the volumetric rate of flow Q is used as a measure of quantity of matter, is is necessary that the absolute pressure and temperature associated with this measurement be stated along with the value of Q.

Example E.4 Density of Moist Air. Calculate the density, ρ_m for moist air at pressure P = 14.7, 50 and 100 psia, temperature t = 80 °F and 170 °F, and relative humidity/100 = 0.50 and 1.0 (saturated air). Compare with dry air (RH = 0).

Calculation for P = 14.7 psia, t = 170 °F and RH/100 = 0.5:

Density of the moist air:

$$\rho_m = \frac{2.6990\ (P - P_v)}{(Z\ T)} + \rho_v \qquad \text{(D-7)}$$

$$T = 170.0 + 459.67 = 629.7\ °R \qquad \text{(C-3)}$$

At t = 170 °F, the saturation pressure of steam is:

$$P_g = 5.9926 \text{ psia} \qquad \text{(Reference 6)}$$

The partial pressure of the water vapor is

$$P_v = (RH/100)(P_g) = (0.5)(5.9926) = 2.9963 \text{ psia} \qquad \text{(D-8)}$$

The partial pressure of the air is

$$P - P_v = 14.7 - 2.9963 = 11.7 \text{ psia}$$

Compressibility Z (air) at T = 629.7 °R and pressure 11.7 psia

$$Z = 1.000 \qquad \text{(Figure B.1)}$$

When RH/100 < 1.000, the water vapor is superheated, and its properties are found in the tables for superheated steam. At t = 170 °F and pressure P_v = 2.9963 psia, the specific volume v is found through interpolation:

P, psia	v, ft^3/lb	
2.9	128.81	(Reference 6, table 3)
2.9963	124.66	
3.0	124.50	

The density ρ_v is:

$$\rho_v = 1/v = 1/124.66 = 0.00802 \text{ lb/ft}^3$$

Substituting the data above into equation (D-7):

$$\rho_m = \frac{(2.6990)(11.70)}{(1.000)(629.7)} + 0.00802, \text{ or}$$

$$\rho_m = 0.05817 \text{ lb/ft}^3$$

For dry air:

$$\rho = \frac{(2.6990)(P)}{(Z\ T)} = \frac{(2.6990)(14.7)}{(1.000)(629.7)} = 0.06301 \text{ lb/ft}^3 \qquad \text{(D-5)}$$

$$\rho_m/\rho = 0.05817/0.06301 = 0.923$$

The following table summarizes the remaining calculations:

Table E.1. Moist Air Data

P psia	RH/100 ---	p_v psia	Z ---	ρ_v lb/ft^3	ρ_m lb/ft^3	ρ lb/ft^3	ρ_m/ρ ---
			t = 80 °F				
100	0.0	0.0	0.998	0.0	0.5011	0.5011	1.000
	0.5	.2534	.998	.00079	.5006	--	0.999
	1.0	.5068	.998	.00158	.5001	--	0.998
50	0.0	0.0	.999	0.0	.2503	.2503	1.000
	0.5	.2534	.999	.00079	.2498	--	0.998
	1.0	.5068	.999	.00158	.2493	--	0.996
14.7	0.0	0.0	.9997	0.0	.07354	.07354	1.000
	0.5	.2534	.9997	.00079	.07306	--	0.993
	1.0	.5068	.9997	.00158	.07256	--	0.987
			t = 170 °F				
100	0.0	0.0	1.0002	0.0	0.4286	0.4286	1.000
	0.5	2.9963	1.0002	.00802	.4237	--	0.989
	1.0	5.9926	1.0002	.01611	.4190	--	0.978
50	0.0	0.0	1.0001	0.0	.2143	.2143	1.000
	0.5	2.9963	1.0001	.00802	.2095	--	0.978
	1.0	5.9926	1.0001	.01611	.2047	--	0.955
14.7	0.0	0.0	1.0000	0.0	0.06301	0.06301	1.000
	0.5	2.9963	1.0000	.00802	.05818	--	0.923
	1.0	5.9926	1.0000	0.1611	.05343	--	0.848

The previous table may be helpful in deciding when a correction for the density of moist air is needed. The data show, for example, with ambient temperature ($t = 80$ °F) and ambient pressure and higher ($P = 14.7$ to 100 psia), that the correction would never exceed about 1 percent, and may often be ignored. On the other hand, when temperatures are higher, the partial pressure of the water vapor becomes more significant and the density ρ_m may be quite different from that of dry air. For example, at 50 psia with saturated air, this difference amounts to about 4.5 percent ($\rho_m/\rho = 0.955$).

Example E.5 Direct Calibration of an Orifice Meter on Site with a Gravimetric Calibrator. An orifice meter with flange taps and diameter $d = 1.200$ in. is mounted in a 2-inch pipe and monitors the flow of water. It is desired to calibrate the meter over a flow range of 10,000 to 20,000 lb/hr using a direct calibration method with static weigh operation as shown in figure 8. Five repeat runs will be taken at four flowrates in the above range. The meter ΔP will be measured by means of a vertical water manometer. Local acceleration due to gravity is 32.145 ft/s^2. Following are the measurements made:

Measurement	Instrument	Reading
Barometric pressure	Aneroid barometer, in. Hg at 32 °F	30.41 in. Hg at 32 °F
Inlet water pressure	Bourdon tube gauge, case vented to atmosphere, psig	59.0 psig
Inlet water temperature	Mercury thermometer, °F	See data below
Manometer temperature	Mercury thermometer, °F	See data
Meter pressure differential, h	Water manometer, inverted, air in the low pressure leg at 59.0 psig; inches of water at manometer temperature	See data
Weigh time	Electronic timer, 5 digits, start/stop actuated thru switch connected to diverter valve	See data
Mass of water	Beam type scale, 1000 lb capacity	See data

The following data were collected during five repeat runs at four flowrates. Flowrate was adjusted as necessary through the flow control valve to maintain h within $\pm$ 0.10 inch of water. The manometer temperature was measured with a thermometer attached, sensing ambient air. The indicated flowrate is net mass/time, designated M'.

Table E.2. Sample Data for Calibration of an Orifice Meter

Temperature Man. °F	Temperature Water °F	h in.H_2O	Time s	Mass Tare lb	Mass Gross lb	Mass Net lb	Flow M' lb/s	
80.0	82.0	64.00	77.799	162.3	615.4	453.1	5.824	
		± .10	80.843	139.3	609.0	469.7	5.810	
			72.008	140.9	562.0	421.1	5.848	
80.3	82.3		74.238	191.1	624.5	433.4	5.838	
			70.805	175.6	590.8	415.2	5.864	
							5.837	Avg
80.3	82.7	46.00	77.225	179.3	561.1	381.8	4.944	
		± .10	67.376	155.6	490.8	335.2	4.975	
			70.264	166.8	513.2	346.4	4.930	
			71.919	156.5	512.5	356.0	4.950	
79.8	83.0		72.546	138.2	498.1	359.9	4.961	
							4.952	Avg
		30.00	84.837	173.0	514.3	341.3	4.023	
79.7	83.3	± .10	97.446	185.1	574.2	389.1	3.993	
			95.675	188.5	571.2	382.7	4.000	
			75.927	155.2	460.2	305.0	4.017	
80.0	83.5		83.579	206.8	541.7	334.9	4.007	
							4.008	Avg
		16.00	103.55	188.6	491.7	303.1	2.927	
80.0	83.6	± .10	119.06	183.6	534.6	351.0	2.948	
			107.88	162.4	478.7	316.3	2.932	
			129.62	194.2	573.6	379.4	2.927	
80.2	83.6		104.29	199.0	505.2	306.2	2.936	
80.0	83.0 Avg						2.934	Avg

Calibration results are to be expressed in terms of the discharge coefficient C, as a function of the pipe Reynolds Number R_D, where C will be determined for each flowrate from equation (2-1):

$$M = 358.93\ (C\ Y\ d^2\ F_a)[(\rho\ h_w)/(1 - \beta^4)]^{1/2}\ \text{lb/hr} \qquad (2\text{-}1)$$

Fluid expansion factor Y = 1.000 since a liquid is flowing.

Orifice thermal expansion F_a = 1.000 since the average water temperature is 83 °F. (Figure B.2)

For 2-inch pipe, D = 2.067 inches (schedule 40). Thus $\beta = d/D = 1.200/2.067 = 0.5806$.

At $t = 83.0\ ^\circ F$ and $p = 59.0$ psig, the water is only slightly compressed and density ρ may be assumed equal to that at saturation temperature with negligible error (0.02 percent). Therefore, $\rho = 62.191\ lb/ft^3$. (Table B.1)

Since the water was weighed in air, the indicated mass flowrate M' needs a correction for the air buoyancy. A nominal value for air density will suffice for this correction. Let $\rho_{air} = 0.075\ lb/ft^3$.

$$\text{Buoyancy Correction} = 1 + (\rho_{air}/\rho_w) = 1 + (0.075/62.191) = 1.0012, \text{ and}$$

$$M = 1.0012(3600)(M') \tag{E5-1}$$

where M' has units lb/s.

Before calculating C, the meter pressure differential h (h is in in. of H_2O at the manometer temperature) needs correction to account for the local acceleration due to gravity (F_g), the air in the low pressure leg (F_{w2}), and the water manometer temperature of 80 °F (F_{w1}). These corrections are explained in appendix C. Using table C.1 and equation (C-8):

$$h_w = (27.729)\ (h)\ [(F_{w1})(F_{w2})(F_g)] \qquad \text{(Table C.1), (C-8)}$$

At $t = 80\ ^\circ F$

$$F_{w1} = 0.036005 \text{ psi/inch of water.} \qquad \text{(Figure C.1)}$$

At $p = 59.0$ psig

$$P = 59.0 + (0.49116)(30.41) = 73.94 \text{ psia, and} \qquad \text{(Table C.1), (C-6)}$$

$$F_{w2} = 0.9940 \qquad \text{(Figure C.3)}$$

with $g_L = 32.145\ ft/s^2$

$$F_g = g_L/32.1740 = 0.9991. \tag{C-7}$$

Thus, at a water temperature of 68 °F:

$$h_w = 27.729\ (0.036005)(0.994)(0.9991)(h), \text{ or} \tag{C-8}$$

$$h_w = 0.9915\ (h) \tag{E5-2}$$

Substituting equations (E5-1) and (E5-2) into equation (2-1), and solving for C:

$$C = \left(\frac{(1.0012)(M')(3600)}{(358.93)(d^2)}\right)\left(\frac{(1 - \beta^4)}{(\rho)(0.9915)(h)}\right)^{1/2}$$

Substituting above data for ρ, β, and d, this becomes:

$$C = 0.83608(M')/(h)^{1/2} \tag{E5-3}$$

where M' has units lb/s and h has units inches of water. Next, the quantity pipe Reynolds Number R_D needs calculation.

$$R_D = \frac{(D\ V\ \rho)}{\mu}$$

where D = pipe inside diameter
V = average velocity
ρ = density of the water
μ = dynamic viscosity

Since M = (ρ A V), where cross sectional area A = $\pi\ D^2/4$, R_D can be written, after substituting for V:

$$R_D = \frac{D(M/\rho\ A)\rho}{\mu} = \frac{(D\ M)}{(\mu\ \pi\ D^2)/4}, \text{ or}$$

$$R_D = 4(M)/(\mu\ \pi\ D) \qquad \text{(E5-4)}$$

R_D is dimensionless, thus (E5-4) becomes

$$R_D = \frac{4\ M}{(3600\ \mu)\ \pi\ (D/12)}, \text{ or}$$

$$R_D = 0.004244\ (M)/(\mu\ D) \qquad \text{(E5-5)}$$

where M has units of lb/hr, μ has units of lb/(ft•s), and D has units of inches.

At a water temperature of 83 °F, $\mu = 0.56 \times 10^{-3}$ lb/(ft·s). (Figure B-3)

From the data collected and equations (E5-3) and (E5-5), the following final results are tabulated:

h in. H_2O at 80 °F	M' lb/s	M lb/hr	R_D ---	C ----	$M/(h)^{1/2}$ lb/hr(in.)$^{1/2}$
64.00	5.837	21038	77130	0.6100	2630
46.00	4.952	17849	65450	.6104	2632
30.00	4.008	14446	52960	.6118	2637
16.00	2.934	10575	38770	.6132	2644

Thus, this orifice meter has been "calibrated". Its performance has been demonstrated and is known through the discharge coefficient C as determined from direct physical measurements. With such an on-site calibration using a static weigh procedure, quite accurate results for C are possible, comparable to those produced routinely by primary calibration laboratories, about 0.2 percent uncertainty.

When this orifice meter is in service, the above data may be used in several ways. With the discharge coefficient being known as a function of Reynolds number, the orifice could be used in different fluids within the above range of R_D. Since in this case, the discharge coefficient C varied only through a small range of about 0.5 percent (from 0.6100 to 0.6132), an average value of C = 0.6114 could be utilized and the flowrate M determined directly from equation (2-1). If higher accuracy is desired, an iterative procedure is needed since R_D is never known initially: Using C = 0.61, one would calculate M from equation (2-1); then calculate R_D from equation (E5-4), then determine C exactly from the data above (by plotting C vs. R_D and interpolating at the measured value of R_D); and finally calculate M from equation (2-1). When the meter is calibrated with water at or near 83 °F, one could plot the quantity $M/(h)^{1/2}$ as a function of h and calculate M directly. In this latter case, some variation in water temperature in the pipe and in the manometer can be tolerated with a small sacrifice in accuracy. For example, when temperature t varies within a range of ± 10 °F and pressure p varies within ± 10 psi, the "worst case" effect on M would be about ± 0.2 percent, explained as follows: from figure C.1, F_{w1} varies about -0.17 percent/10 °F at 80 °F as does the density ρ of water from table B.1. From figure C.3, F_{w2} varies -0.08 percent/10 psi. The worse case occurs when t and p both increase (or both decrease) together. In equation (2-1), accounting for effects of t on ρ, and t and p on h_w, the absolute value of the error becomes (1/2)(0.17 + 0.17 + 0.08) = 0.2 percent for a change of +10 °F and +10 psi (or -10 °F and -10 psi). The "1/2" factor comes from the exponential 1/2 in equation (2-1) and differential calculus. The change in C (through temperature effects on ρ and μ in R_D) is negligible. Thus, under limited conditions, accurate measurements of M can be made directly from h.

Example E.6. Calibration of an Orifice Meter on Site Using The Transfer Meter Method. A stainless steel orifice meter with corner pressure taps and d = 3.750 inches is mounted in a 6-inch pipe flowing chilled water. A differential pressure transducer system calibrated to read out in inches of water at 68 °F measures the orifice meter ΔP. It is desired to calibrate the orifice meter using a transfer reference meter system located downstream, as shown in figure 5. The transfer meter is a stainless steel nozzle venturi with a throat diameter of d = 4.030 inches and a coefficient of discharge C = 0.9512 as determined by calibration on water at 65 °F at a primary calibration facility. This calibration was on water at 65 °F covering a range $1.5 \times 10^5 \leq R_D \leq 1 \times 10^6$. A mercury manometer is used to measure the nozzle venturi ΔP. Local acceleration due to gravity is 32.172 ft/s^2.

The following represent average values of data taken from 5 repeat observations at a single flowrate:

Measurement	Instrument	Reading
Chilled water temperature	Dial type thermometer, °F	42.0 °F
Water pressure	Bourdon tube gauge, psig	85.3 psig
Working meter (orifice) pressure differential h_w	Pressure transducer system inches of water at 68°F	320.32 in. H_2O at 68 °F
Transfer meter (nozzle venturi), pressure differential, h	Mercury manometer, inches of mercury, water in high pressure leg	7.26 in. Hg at 78.0 °F
Manometer temperature	Mercury thermometer, °F	78.0 °F

Calculations for the nozzle venturi meter:

$C = 0.9512$

$d = 4.030$ inches

$D = 6.065$ inches (6-inch pipe, schedule 40)

$\beta = d/D = 0.6645$

$\rho = 62.443\ lb/ft^3$ (at 42.0 °F and 100 psia) (Table B.1)

$Y = 1.000$ (liquid)

$F_a = 0.9995$ at 42 °F (300 Series Stainless Steel) (Figure B.2)

Obtain h_w, inches of water at 68°F, for the nozzle meter as follows:

$h_w = 27.729$ (psi) (Table C.1)

$h_w = 27.729$ (inches of Hg at 78 °F) $(F_{m1}\ F_{m3}\ F_g)$ (C-10)

$F_{m1} = 0.48892$ psi/in. Hg at 78 °F (Figure C.2)

$F_{m3} = 0.9263$ (Appendix C)

$F_g = g_L/32.1740 = 32.172/32.1740 = 0.99994$ (neglect) (C-7)

$h_w = (27.729)(7.26)(0.48892)(0.9263)$, or

$h_w = 91.17$ inches of water at 68 °F

Next, calculate the flowrate M;

$$M = 358.93\ (C\ Y\ d^2\ F_a)\ [\rho\ h_w/(1 - \beta^4)]^{1/2} \qquad (2\text{-}1)$$

$$= 358.93(0.9512)(1.000)(4.030)^2(0.9995)\left(\frac{(62.443)(91.17)}{1 - (0.6645)^4}\right)^{1/2}, \text{ or}$$

$M = 466,100$ lb/hr

The pipe Reynolds Number R_D is also needed.

$$R_D = 0.004244\ (M)/(\mu\ D) \qquad (E5\text{-}5)$$

$\mu = 1.00 \times 10^{-3}$ lb/(ft•s) at 42 °F (Figure B.3)

$R_D = 0.004244(466,100)/(1.00 \times 10^{-3})(6.065)$, or

$R_D = 326,000$.

The calculation for the transfer meter is complete. The reference flowrate M is 466,100 lb/hr. The pipe Reynolds Number is 326,000 which lies within the range of calibration for the nozzle venturi meter.

Calculation for the orifice meter:

$Y = 1.000$

$d = 3.750$ inches

$\beta = d/D = 3.750/6.065 = 0.6183$

$F_a = 0.9995$ at 42 °F (300 Series SS) (Figure B.2)

$\rho = 62.443$ lb/ft^3

$M = 466,100$ lb/hr

$h_w = 320.32$ in. of water at 68 °F

Solving for the discharge coefficient C in equation (2-1)

$$C = \left[\frac{M}{(358.93)(Y)(d^2)(F_a)}\right]\left[\frac{(1 - \beta^4)}{(\rho\ h_w)}\right]^{1/2}$$

$$= \left[\frac{466,100}{(358.93)(1.000)(3.750)^2(0.9995)}\right]\left[\frac{1 - (0.6183)^4}{(62.443)(320.32)}\right]^{1/2}, \text{ or}$$

$C = 0.6036$

Therefore, the coefficient of discharge C for the orifice (working) meter is 0.6036 at R_D = 326,000.

In particular, it may be noted:

1. The calibration is usually conducted at several flowrates which correspond to building service conditions. Thus, a set of values of C vs. R_D, would be obtained. With an installation conforming to good metering practice, the data for C should be in good agreement with the discharge coefficient data given for corner taps in appendix A.

2. Once a particular meter has been calibrated, i.e., discharge coefficient C established as a function of R_D, it may be used in different fluids within its calibrated range of R_D. Since C changes very slowly with R_D when $R_D > 10^5$, a single value can be used for C with small error, usually not exceeding 0.2 to 0.3 percent, when flowing a given fluid over a flow range not exceeding, say, 2:1. When the flow range is larger, or when maximum accuracy is needed, C should be plotted as function of R_D and an iteration approach used to determine C. (See discussion at the end of example E.5.)

3. The coefficient of discharge C for the transfer meter system applies only to a designated range of R_D, as determined from calibration at an independent facility. Thus, use of the transfer meter outside its calibrated range as expressed in terms of R_D should be avoided.

Example E.7 <u>On-Site Calibration of a Positive Displacement Meter Using The Transfer Meter Method</u>. A positive displacement meter monitors the flow of water at 150 °F and 100 psig. It is to be calibrated on site at nominal 20 GPM using an orifice meter as the transfer reference installed downstream of the PD meter as shown in figure 5. The orifice meter has been calibrated with water with results given in previous example E.5 and plotted here in figure E.1. Salient features of the PD meter and associated electronics equipment are:

Size:	2-inch
Type:	rotary piston
Output:	1 pulse per 0.15 gallon, nominal
Rating:	200 psig and 300 °F
Installation:	The PD meter is mounted in a horizontal 2-inch pipe, insulated, with 75 diameters of straight pipe upstream, 5D downstream, no flow straightener

The orifice meter has the following features:

Size:	Diameter d = 1.200 inches mounted in a 2-inch pipe (D = 2.067 inches), β = 0.5806

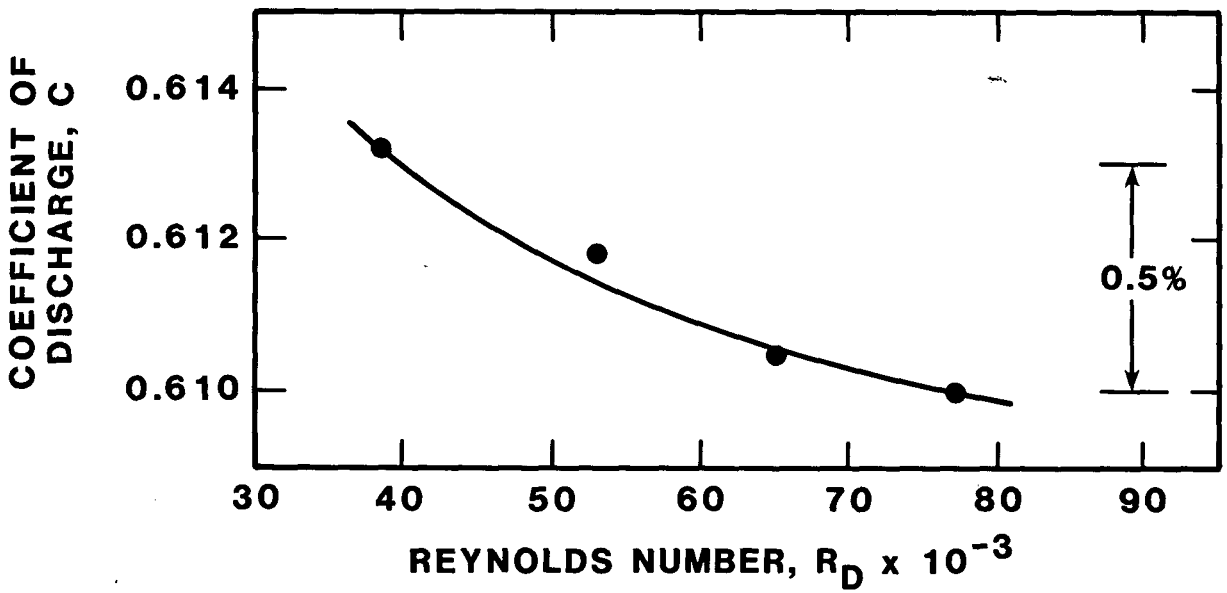

Figure E.1. Coefficient of discharge for an orifice meter, from the results of example E.5

Calibration:	On water at 83 °F, using a gravimetric calibrator, with ΔP measured by a vertical water manometer. (See example E.5).
Installation:	The orifice metering section is installed in a horizontal position downstream of the PD meter as shown in figure 5, with a flow straightener and with the metering section insulated. Temperature and pressure taps are installed at the exit. The differential pressure taps, including the reservoirs are insulated. The sensing lines to the manometer are bare, and installed adjacent to each other. The ΔP is measured by the vertical water manometer.

The time for a preselected number of PD pulses is measured. From the flowrate as measured by the orifice meter, the calibration constant K for the PD meter, gallons per pulse, can be determined for the flowrate of interest.

The following are the measurements made where "Reading" data are the average of 5 repeat runs. The local acceleration due to gravity is 32.10 ft/s^2.

Measurement	Instrument	Reading
Exit water pressure	Bourdon tube gauge, case vented to atmosphere, psig	102.0 psig
Exit water temperature	Dial type thermometer with bulb and capillary, °F	152.0 °F
Manometer temperature	Mercury thermometer, °F	72.0 °F
Orifice meter ΔP	Water manometer, inverted, air in the low pressure leg at 102 psig, inches of water	17.20 in. at 72 °F
PD meter, test time	Electronic preset counter/timer, timer start/stop activated to time period for 150 meter pulses, preset	63.150 seconds

The flowrate for the orifice is calculated from:

$$Q = 5.982\ (C\ Y\ d^2\ F_a)[h_w/\rho(1 - \beta^4)]^{1/2} \qquad (2\text{-}2)$$

Quantities first needing evaluation are C, F_a, h_w, and ρ.

The (flowing) fluid properties ρ and ν are as follows: At 152 °F and 102 psig, the water is a compressed liquid. While the density is nearly that of saturated liquid, interpolation of table B.1 data gives $\rho = 61.172\ lb/ft^3$. At 152 °F, the kinematic viscosity of water ν is $0.47 \times 10^{-5}\ ft^2/s$, from figure B.3. Thus, at 152 °F:

$\rho = 61.172 \text{ lb/ft}^3$

$\nu = 0.47 \times 10^{-5} \text{ ft}^2/\text{s}$

The coefficient of discharge C is needed at pipe Reynolds Number R_D for water at 152 °F. Estimate R_D from the PD meter using the nominal output of 0.15 gallon/pulse as follows:

$$Q = \frac{(150)(0.15)}{(63.150)(7.4805)}, \quad \left[\frac{\text{pulses}}{\text{s}}\right] \left[\frac{\text{gallon}}{\text{pulse}}\right] \left[\frac{\text{ft}^3}{\text{gallon}}\right]$$

$$Q = 0.047 \text{ ft}^3/\text{s}$$

$$R_D = (D\ V\ \rho)/\mu = (D\ V)/\nu = 4\ (Q)/(\pi\ D\ \nu) \qquad \text{(E7-1)}$$

since $\nu = \mu/\rho$, and $V = Q/A = 4\ (Q)/(\pi\ D^2)$. Thus

$$R_D = \frac{(4)(0.047)}{\pi\ (2/12)(0.47 \times 10^{-5})}$$

$R_D = 76{,}000$ estimated

From figure E.1, $C = 0.6100$ at $R_D = 76{,}000$

Since the fluid is incompressible, $Y = 1.000$.

With the orifice plate at the fluid temperature, 152 °F, a small correction for the expansion of the plate can be made through F_a. The orifice plate material is 300 Series SS (stainless steel). From figure B.2:

$F_a = 1.0015$

The orifice ΔP manometer reading in inches of water needs correction to 68 °F. The manometer temperature is 72 °F.

$$h_w = (\text{inches of water})(\rho/\rho_{68})\ (F_{w2}\ F_g) \qquad \text{(C-11)}$$

$$= (17.20)(62.2902/62.3205)(0.9903)(0.9977), \text{ or}$$

$h_w = 16.986$ inches of water, where

$\rho = 62.2902 \text{ lb/ft}^3$ (at 72 °F) (Table B.1)

$\rho_{68} = 62.3205 \text{ lb/ft}^3$ (at 68 °F) (Table B.1)

$F_{w2} = 0.9903$ (at $102 + 14.7 = 117$ psia) (Figure C.3)

and with $g_L = 32.10 \text{ ft/s}^2$

$$F_g = g_L/32.1740 = 0.9977 \qquad \text{(C-7)}$$

Inserting the above data into equation (2-2), the actual volumetric flowrate through the orifice is:

$$Q = 5.982(0.6100)(1.000)(1.200)^2(16.986)^{1/2}/[61.172(1 - (0.5806)^4)]^{1/2}$$

$$= 2.941 \text{ ft}^3/\text{min, or}$$

$$Q = 2.941(7.4805) = 22.00 \text{ GPM}$$

Next, recalculate R_D and check C through R_D:

$$R_D = \frac{4\ (Q)}{(\pi\ D\ \nu)} = \frac{4(2.941)/60}{\pi\ (2/12)(0.47 \times 10^{-5})} = 80{,}000 \qquad \text{(E7-1)}$$

From figure E.1, the corresponding value of C for the orifice is 0.609 and, through equation (2-2), the flowrate Q can be adjusted:

$$Q = 22.00(0.609/0.6100) = 21.96 \text{ GPM}$$

This completes the flow calculation for the orifice. Now, compute the calibration factor for the PD meter. Express K in units gallon/pulse.

$$K = \frac{63.15(21.96)}{150(60)}\ ,\ \left[\frac{\text{seconds}}{\text{pulse}}\right] \left[\frac{\text{gallons}}{\text{minute}}\right] \left[\frac{\text{minutes}}{\text{seconds}}\right]$$

$$K = 0.1540 \text{ gallon/pulse}$$

Thus, the calibration factor for this meter is K = 0.1540 gallon/pulse when water is flowing at or near 22 GPM, 152 °F and 100 psig. At other flowrates, temperatures, and for large changes in pressure, K could vary. With these conditions of 150 °F and 100 psig, and with this orifice meter and manometer, thus PD meter could be calibrated at flowrates down to about 10 GPM, as limited by the orifice calibration for C in the range $40{,}000 < R_D < 80{,}000$. However, in this case $h_w \approx 4.00$ inches of water; accurate measurement becomes increasingly difficult at such a low ΔP. To calibrate the PD meter at flowrates greater than 20 GPM, the orifice would need additional calibration.

Example E.8 <u>On-Site Calibration of a Vortex Shedding Meter Using the Transfer Meter Method</u>. A vortex shedding meter monitors the flow of steam at 360 °F and 45 psig. It is to be calibrated on site using a nozzle venturi meter as the transfer reference meter. The venturi meter is installed downstream of the vortex meter as shown in figure 5. The nozzle venturi meter has been previously calibrated using water as noted below. Salient features of the vortex meter and associated electronics are:

Size: 6-inch

Output: 8 pulses/ft^3, nominal

Rating: 250 psig and 600 °F

Installation: The vortex meter is installed in a vertical 6-inch insulated pipe within a metering section as shown in figure 7 including a perforated plate flow straightener. (See figure 13.)

The nozzle venturi meter has the following features:

Size: Throat diameter d = 2.426 inches. Meter mounted in a 6-inch pipe, D = 6.065 inches, $\beta = 0.400$.

Calibration: On water at 60 °F, C = 0.9675 for R_D range 3×10^5 to 1×10^6

Installation: Mounted in a metering section with a flow straightener. Installed in a vertical pipe, downstream of the vortex meter as shown in figure 5, with the metering section insulated. A temperature tap is installed at the metering section inlet and the pressure taps are located at the venturi inlet (P_1) and throat (P_2). The sensing lines with drain and vent lines connect to a ΔP transducer system calibrated in inches of water at 68 °F.

Material: Bronze, 6 percent tin.

Pulses are counted from the vortex meter with an electronic counter for a preset gate time to accumulate at least, say, 5000 pulses. When the meter frequency is low resulting in excessive run time, a preset counter measuring a time period for a preset number of pulses may be used instead.

From the flowrate measured by the nozzle venturi meter, the calibration constant K for the vortex meter, ft^3/pulses, is calculated for the flowrate of interest.

Following are the measurements made where "reading" data are the averages of 5 repeat runs.

Measurement	Instrument	Reading
Barometric pressure P_b	Aneroid barometer in. Hg at 32 °F	30.41 in. Hg at 32 °F
Inlet nozzle venturi pressure, p_1	Bourdon tube gauge, psig	44.6 psig
Inlet nozzle venturi temperature, t_1	Thermocouple, °F	360 °F
Nozzle venturi ΔP, h_w	Differential pressure transducer system, in. water at 68 °F	123.83 in. water at 68 °F
Vortex meter pulse count	Electronic preset counter/timer, pulses for a 45-second time interval, average of five runs	5478

The flowrate for the nozzle venturi meter is calculated from:

$$Q = 5.982\ (C\ Y\ d^2\ F_a)\ [h_w/\rho(1 - \beta^4)]^{1/2}\ \text{ft}^3/\text{min} \qquad (2\text{-}2)$$

Quantities needing evaluation are: Y, F_a and ρ. First, evaluate density ρ as follows. The inlet pressure and temperature at the nozzle venturi inlet are:

$$P_1 = (p_1 + P_b) = 44.6 + (30.41)(0.49116) = 59.54 \text{ psia} \qquad \text{(Table C.1), (C-6)}$$

$$t_1 = 360\ °F$$

Using steam tables, reference 6, table 3, and from interpolation, the specific volume v_1 at 360 °F and 59.54 psia is 7.989 ft^3/lb.

$$\rho_1 = 1/v_1 = 1/7.989 = 0.1252 \text{ lb/ft}^3$$

To evaluate the expansion factor Y, first find P_2/P_1.

$$\Delta P = (P_1 - P_2) = (3.6063 \times 10^{-2})(\text{in. water at 68 °F}) \text{ or} \qquad \text{(Table C.1)}$$

$$\Delta P = (3.6063 \times 10^{-2})(123.83) = 4.466 \text{ psia}$$

$$P_2 = (P_1 - \Delta P) = 59.54 - 4.47 = 55.07 \text{ psia, and}$$

$$P_2/P_1 = 55.07/59.54 = 0.925 = r$$

The expansion factor Y is found from table A.16 or figure A.2. With r = 0.925, the specific heat ratio γ = 1.3 for steam, and β = 0.400, then with interpolation:

$$Y = 0.9544 \qquad \text{(Table A.16)}$$

With the nozzle venturi at a fluid temperature of 360 °F and constructed of bronze, 6 percent tin:

$F_a = 1.0056$ (Figure B.2)

The flowrate Q can now be calculated. Reynolds number R_D will be calculated later and compared to the R_D range for C. In equation (2-2):

$$Q = \frac{5.982(0.9675)(0.9544)(2.426)^2(1.0056)(123.83)^{1/2}}{[0.1252(1 - 0.400^4)]^{1/2}}$$

$Q = 1042\ ft^3/min.$

Now check that R_D falls within range of calibration for discharge coefficient C. For steam at 360 °F and 59.54 psia:

$\mu = 1.02 \times 10^{-5}$ lb/s ft (Figure B.4)

$\rho = 0.1252\ lb/ft^3$, and

$\nu = \mu/\rho = 8.147 \times 10^{-5}\ ft^2/s$

$$R_D = 4\ (Q)/(\pi\ D\ \nu) = \frac{(4)(1042/60)}{(\pi)(6/12)(8.147 \times 10^{-5})} = 5.4 \times 10^5 \qquad \text{(E7-1)}$$

which conforms to the calibration range for C ($3 \times 10^5 \leq R_D \leq 1 \times 10^6$).

Finally, calculate the calibration constant K for the vortex meter.

$$K = pulses/ft^3 = \frac{(pulses/min)}{(ft^3/min)}$$

From the vortex meter measurements:

f = 5478 pulses/45 seconds = 121.73 Hz, or

f = 121.73(60) = 7304 pulses/min

From the nozzle venturi measurements:

$Q = 1042\ ft^3/min$

Thus:

$K = 7304/1042 = 7.010\ pulses/ft^3$

This value of K applies to one set of conditions: namely, steam temperature 360 °F and steam pressure 45 psig at the vortex meter exit, and to a meter frequency of 122 Hz. Normally, the meter would be calibrated over a range of flowrates, and with constant steam temperature and pressure, the calibration constant K would be plotted as a function of frequency, f. When the steam temperature and/or pressure vary, K would be plotted as a function of f/ν to account for variation of viscosity ν with temperature and pressure. See figure 18, for example. Since the vortex meter has no moving parts, its performance can be predicted through the f/ν plot from results with a single fluid (steam at constant T and P). However, by actual calibration for a few test points with "different" fluids (sets of T and P covering the operating range), the meter performance over a range of conditions can be validated.

Note that the volumetric flowrate measured by the vortex shedding meter is the volume flowrate at the meter exit temperature and pressure. If the volumetric flowrate is needed at a different location in the system or at different values of temperature and/or pressure, or if the mass flowrate is needed, these quantities can be calculated by procedures discussed in appendix D and illustrated in examples E.1 and E.2.

Note also that while the nozzle venturi was calibrated using water, it was used as a reference meter on a compressible fluid (steam in the case mentioned) through use of the fluid expansion factor Y and the meter expansion factor F_a. This is done, in particular, when primary facilities flowing steam are not available but at the expense of an increase in the calibration uncertainty through tolerance values assigned to Y. (See table A.18.)

Example E.9 <u>On-Site Calibration of a Turbine Meter</u>. Water at 160 °F and 50 psig is being monitored by a turbine meter. It is to be calibrated on site using a positive displacement meter as the transfer reference meter. The PD meter has been calibrated previously using water at 160 °F, and it is installed upstream of the turbine meter as shown in figure 6. Salient features of the turbine meter are:

Size:	3-inch
Output:	45 pulses/gallon, nominal
Rating:	600 GPM rated flow, 300 psig and 400 °F
Pressure Loss:	5 psi at rated flow, 600 GPM
Installation:	The turbine meter is installed in a horizontal 3-inch insulated pipe within a metering section as shown in figure 7. The flow straightener is a tubular type as shown in figure 12.

The transfer reference (PD) meter has the following features;

Size:	3-inch
Output:	pulse type, 10 pulses/gallon nominal
Rating:	700 GPM rated flow, 150 psig and 300 °F
Pressure Loss:	2 psi at rated flow

Installation: The PD meter is installed in the insulated horizontal pipe immediately upstream of the turbine metering section in the same (horizontal) position existing during calibration. No flow straightener is used since previous tests showed no significant influence of swirling flow on meter performance.

Calibration: Water in the range 145-180 °F at a primary facility yielded an average value of the calibration factor K = 10.145 pulses/gallon, and K data scatter $\pm$ 0.2 percent for the flow range 30 to 700 GPM.

Pulses are counted from each meter with an electronic counter during concurrent time intervals, counting at least 500 pulses from the PD meter and at least 1000 pulses from the turbine meter. The flow range to be covered by the calibration is 50 to 500 GPM.

The test should be conducted at a pressure level sufficient to avoid cavitation in each meter. At 160 °F, the vapor pressure of water is about 5 psia. From guidelines mentioned in section 5.3, a (minimum) inlet pressure to the turbine meter of 55 psia should be adequate to prevent cavitation. For the PD meter, an absolute pressure level equivalent to the vapor pressure plus 3 or 4 times the meter pressure loss at the rated flow is probably sufficient, as discussed in section 5.1. For this PD meter, this amounts to 5 + 4(2) = 13 psia. Since the PD meter is upstream of the turbine meter, cavitation in the turbine meter would likely occur first. It is noted, however, cavitation usually occurs initially at higher rates of flow, resulting in (sometimes) an abrupt increase in pulse rate/throughput.

Table E.3 summarizes the measurements made and results of the calibration, where the pulse data are averages of 5 repeat runs. The results show no abrupt changes of the calibration factor K for the turbine meter, and it is assumed no cavitation occurred.

Calculations for the PD meter are:

$$f = \text{pulses/time}$$

$$Q = 60(f/K) \tag{4-1}$$

For Run No. 1:

$$f = 2561/30.000 = 85.37 \text{ Hz}$$

$$Q = (60)(85.37)/(10.145) = 504.9 \text{ GPM}$$

Calculations for the turbine meter are:

$$f = \text{pulses/time}$$

$$K = 60(f/Q) \text{ pulses/gallon}$$

Table E.3. Turbine Meter Calibration Data
Water: 162 °F ± 2 °F, 50 ± psig

	Positive Displacement Meter (Reference) K = 10.145 pulses/gallon				Turbine Meter			
Run	Pulses	Time Seconds	Frequency Hz	Flowrate GPM	Pulses	Time Seconds	Frequency Hz	K Pulses/gallon
1	2561	30.000	85.37	504.9	11180	30.000	372.67	44.29
2	1943	30.000	64.77	383.0	8484	30.000	282.8	44.30
3	1281	30.000	42.70	252.5	5602	30.000	186.7	44.38
4	717	30.000	23.9	141.0	3154	30.000	105.1	44.61
5	820	60.000	13.7	80.8	3580	60.000	59.67	44.3
6	512	60.000	8.53	50.5	2204	60.000	36.73	43.6

For Run No. 1:

$f = 11180/30.000 = 372.67$ Hz

$K = (60)(372.67/504.9) = 44.29$ pulses/gallon

A plot of the calibration results are shown in figure E.2 where K is plotted as a function of frequency f. These results show that the calibration factor K varies $\pm$ 0.5 percent for a flow range of about 70 to 500 GPM ($50 \leq f \leq 375$ Hz). Also, it should be noted, these data apply only when the water is near 160 °F.

Example E.10 <u>On-Site Calibration of a Target Meter</u>. A target meter monitors water flowing at 45 °F and 100 psig. It is to be calibrated on site with a transfer meter system consisting of two turbine meters mounted in series. The turbine meters have been calibrated previously on water at 42 °F. Two turbine meters are used instead of one as a check on the transfer-meter operation. Should differences in turbine meter performance be (or appear) larger than an allowed maximum, meter trouble would be suspected and this performance would be investigated. This approach is favored since this type meter tends to be incompatible with foreign material entrained in water and since water is considered to have poor lubricating properties. However, these meters have definite advantages for transfer meter applications, including small size, digital output, and a calibration factor that is essentially constant over large flow ranges when used with a low-viscosity liquid such as water. The turbine meters are installed downstream of the target meter in a metering section as shown in figure 5. At the entrance to the metering section a 50-micron filter is installed. The target meter is installed in a metering section including a flow straightener. Salient features of the target meter are:

Size:	2-inch
Output:	4-20 mA DC, slope and span adjusted for 0-270 GPM nominal, transmitter output varies with Q^2

The turbine meters have the following features:

Size:	2-inch
Output:	pulse type, 3300 pulses/ft^3 nominal
Rating:	300 GPM rated flow, 300 psig and 250 °F
Calibration:	On water in the range 40 °F to 42 °F at a primary facility. Meter No. 1 has a calibration factor K = 3303 pulses/ft^3 with a variation of $\pm$ 0.2 percent in the upper 3:1 flow range. Meter No. 2, K = 3286 pulses/ft^3, with a variation of $\pm$ 0.2 percent in the upper 3:1 flow range.

The target meter will be calibrated over a range of 100 to 270 GPM, expressing the calibration factor CF in units GPM/(mA)$^{1/2}$ as discussed in section 4.4.

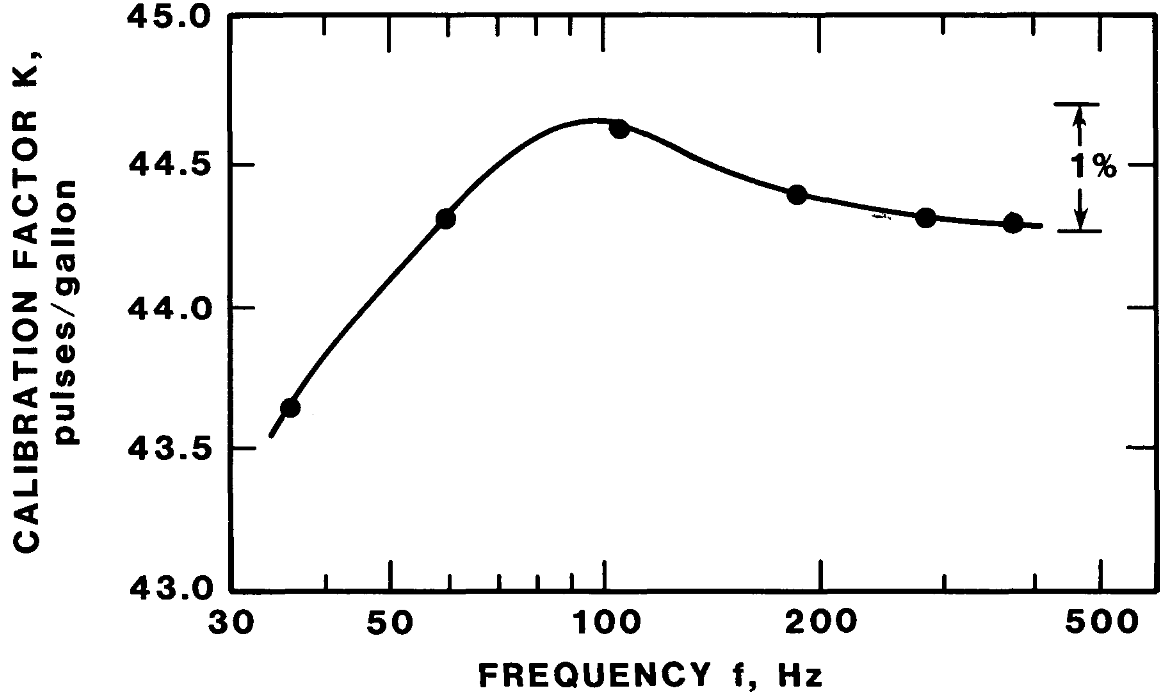

Figure E.2. Calibration factor K for a turbine meter monitoring water flow [water at 162 °F (72.2 °C)].

The reference flowrate will be based on the average readings of the two turbine meters. The target meter transmitter output current will be measured by a digital voltmeter (DVM) (4 1/2 digits) and the two turbine meter ouputs by variable-time-base electronic counters. The time interval should be set large enough to assure a steady count; i.e., no influence of random flow fluctuations, or set to accumulate at least 5000 pulses, whichever is larger.

To check on the turbine meter operation, the frequencies f_1 and f_2 will be compared. To account for the difference and the variation ($\pm$ 0.2 percent) in calibration factors for the two turbine meters, and to allow for meter repeatability, a criterion of $1.000 \leq f_1/f_2 \leq 1.010$ will be used to signify satisfactory transfer meter operation. (Note 3303 + 0.2 percent = 3309; 3286 - 0.2 percent = 3280; 3309/3280 = 1.009). In particular, values of f_1/f_2 which show drift or large scatter outside this range should be investigated.

Table E.4 summarizes the measurements and results of the calibration of the target meter, where data for each test are the average of 5 repeat runs. Note that the transmitter current, I, was checked at zero flow at the beginning and end of each run, test points 1 and 6. Also note that frequency ratio f_1/f_2 for the turbine meters stayed in the range 1.000 to 1.010, indicating satisfactory meter operation according to the criterion stated previously.

Calculations for the turbine meters are:

$$f = \text{pulses/time}$$

$$Q = 60(f/K) \tag{4-1}$$

$$Q_{avg} = (Q_1 + Q_2)/2$$

For test point No. 2:

$$f_1 = 7224/10 = 722.4 \text{ Hz}$$

$$Q_1 = (60)(722.4)(7.4805)/3303 = 98.16 \text{ GPM}$$

$$f_2 = 7202/10 = 720.2 \text{ Hz}$$

$$Q_2 = (60)(720.2)(7.4805)/3286 = 98.37 \text{ GPM}$$

$$Q_{avg} = (Q_1 + Q_2)/2 = 98.27 \text{ GPM}$$

Calculations for the target meter are:

$$I_o = (I_1 + I_6)/2 = (3.998 + 4.002)/2 = 4.000 \text{ mA}$$

The calibration factor is:

$$CF = \text{GPM}/(\text{mA})^{1/2}, \text{ and} \tag{4-10}$$

$$CF = 98.27/(6.074 - 4.000)^{1/2} = 68.24 \text{ GPM}/(\text{mA})^{1/2}$$

The calibration results are plotted in figure E.3 showing the calibration factor CF, GPM/$(I - I_o)^{1/2}$ plotted as a function of transmitter current I. These data apply only for water at or near 45 °F. To obtain the flowrate at any current I, equation (4-11) is applied. For example, when I = 16.00 mA:

$$CF = 69.41 \ GPM/(mA)^{1/2}, \text{ and}$$

$$Q = (CF)(I - I_o)^{1/2} = (69.41)(16 - 4)^{1/2} \text{ or} \qquad (4\text{-}11)$$

$$Q = 240.4 \ GPM$$

Table E.4. Target Meter Calibration Data

Water at 45 °F

Turbine Meters (Reference)

Test Point	No. 1, $K_1 = 3303$ pulses/ft^3: Pulses	Time s	f_1 Hz	Q_1 GPM	No. 2, $K_2 = 3286$ pulses/ft^3: Pulses	Time s	f_2 Hz	Q_2 GPM
1	--	--	--	0	--	--	--	0
2	7224	10	722.4	98.16	7202	10	720.2	98.37
3	10281	10	1028.1	139.70	10196	10	1019.6	139.27
4	7372	5	1474.4	200.3	7364	5	1472.8	201.2
5	10176	5	2035.2	276.55	10095	5	2019.0	275.77
6	--	--	--	0	--	--	--	0

	Reference Meters		Target Meters		
	f_1/f_2	Q_{avg} GPM	I mA	CF $Q_{avg}/(I-I_o)^{1/2}$	Q GPM
1	--	0	3.998	--	0
2	1.003	98.27	6.074	68.24	98.27
3	1.008	139.49	8.139	69.56	141.52
4	1.001	200.8	12.447	69.08	200.8
5	1.008	276.16	19.736	69.62	276.17
6		0	4.002	--	0

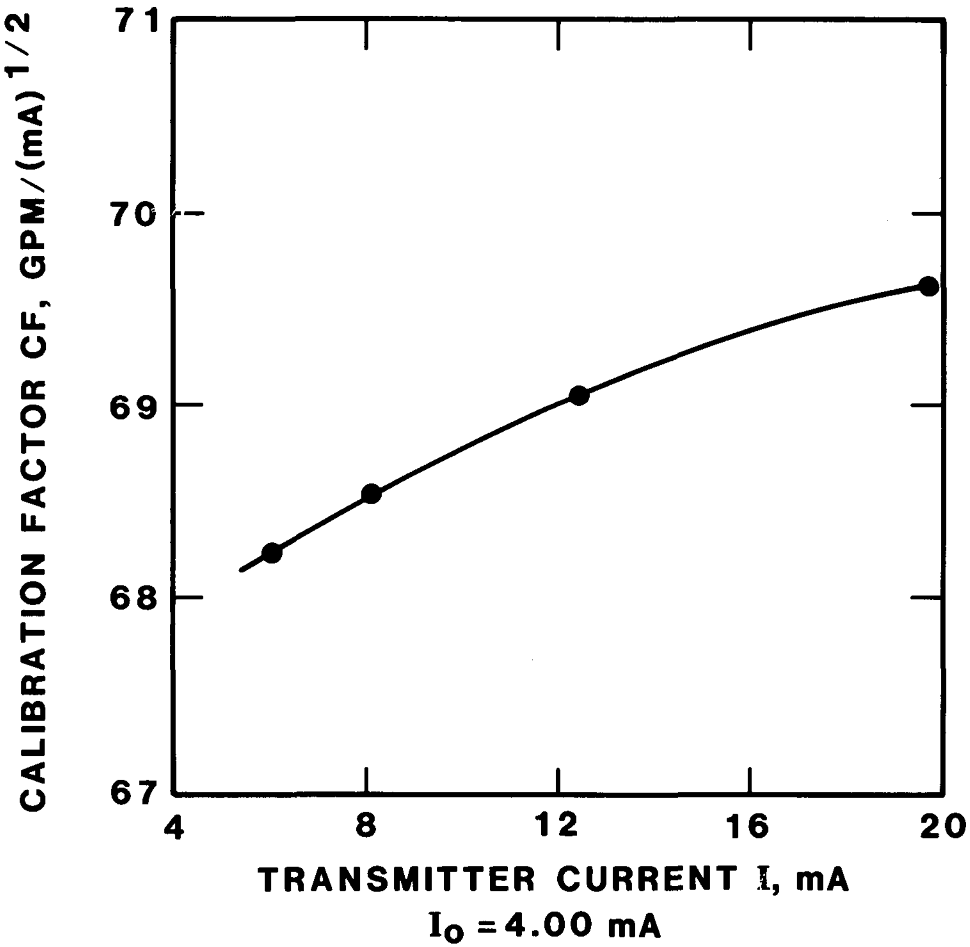

Figure E.3. Performance of a target meter monitoring water flow at 45 °F (7.2 °C)

NBS-114A (REV. 2-80)

U.S. DEPT. OF COMM. BIBLIOGRAPHIC DATA SHEET (See instructions)	1. PUBLICATION OR REPORT NO. NBS BSS 159	2. Performing Organ. Report No.	3. Publication Date January 1984

4. TITLE AND SUBTITLE

On-Site Calibration of Flow Metering Systems Installed in Buildings

5. AUTHOR(S)

David W. Baker and C. Warren Hurley

6. PERFORMING ORGANIZATION *(If joint or other than NBS, see instructions)*

NATIONAL BUREAU OF STANDARDS
DEPARTMENT OF COMMERCE
WASHINGTON, D.C. 20234

7. Contract/Grant No.

8. Type of Report & Period Covered

Final

9. SPONSORING ORGANIZATION NAME AND COMPLETE ADDRESS *(Street, City, State, ZIP)*

Naval Civil Engineering Laboratory
Port Hueneme, CA 93043

10. SUPPLEMENTARY NOTES

Library of Congress Catalog Card Number: 83-600626

[ ] Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT

The measurement of flow of the various fluids (air, water, steam) used in building service systems is usually the most difficult parameter to obtain and maintain. Consequently, in energy management and control systems (EMCS), the flowrate or the total quantity of flow is often the least accurate measurement. However, in most systems the energy consumed depends directly on this parameter.

Since the majority of fluid flow measuring techniques require the sensing element to be located in the stream of the fluid being monitored, flow measuring devices often are the most difficult instruments to calibrate initially and to maintain in calibration within the required accuracy. This report summarizes the various types of flowmetering devices used in EMCS, various methods for their initial calibration and, when practical, techniques for maintaining their calibration while they are in service. Emphasis is placed on the use of transfer reference meter systems, where the working meter is calibrated on site by connecting it in series with a calibrated transfer meter of any variety. Other methods of calibration are also described.

Reference tables and the necessary equations for flow calculations are presented throughout the text and in the appendicies. Illustrative examples are given in detail for the calculation of flow using each type of metering device described. These examples are extremely helpful in field calibration when the metering being calibrated is of a different type than the meter being used as a reference. Because of this, the reader is encouraged to review these examples.

12. KEY WORDS calibration methods; flowmetering devices; flow nozzle meters; multiple pitot-static tube assemblies; orifice meters; positive displacement meters; reverse-pitot tube assemblies; target meters; turbine meters; ultrasonic flowmeters; venturi shedding meters.

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14. NO. OF PRINTED PAGES

154

15. Price

For sale by the Superintendent of Documents, U.S. Government Printing Office, Washington, DC 20402

USCOMM-DC 6043-P80