

FLOW MEASUREMENT HANDBOOK

Flow Measurement Handbook is an information-packed reference for engineers on flow-measuring techniques and instruments. Striking a balance between laboratory ideal and the realities of field experience, it provides a wealth of practical advice on the design, operation, and performance of a broad range of flowmeters.

The book begins with a brief review of essentials of accuracy and flow, how to select a flowmeter, and various calibration methods. Following this, each chapter is devoted to a class of flowmeter and includes detailed information on design, application, installation, calibration, operation, advantages, and disadvantages.

Among the flowmeters discussed are orifice plates, venturi meters, standard nozzles, critical flow venturi nozzles, variable area and other devices depending on momentum of the flow, volumetric flowmeters such as positive displacement, turbine, vortex shedding, swirl, fluidic, electromagnetic and ultrasonic meters, and mass flowmeters including thermal and Coriolis. More than 80 different types and 250 applications are listed in the index. There are also chapters covering probes, a brief introduction to modern control, and manufacturing implications.

For those readers who want more background information, many chapters conclude with an appendix on the mathematical theory behind the techniques discussed. The final chapter takes a look at directions in which the technology is likely to go in the future.

Engineers will use this practical handbook to solve problems in flowmeter design and application and to improve performance.

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Flow Measurement Handbook

**INDUSTRIAL DESIGNS, OPERATING PRINCIPLES,
PERFORMANCE, AND APPLICATIONS**

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To Liz, Sarah and Paul, Mark, John and Rachel

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Preface

This is a book about flow measurement and flowmeters written for all in the industry who specify and apply, design and manufacture, research and develop, maintain and calibrate flowmeters. It provides a source of information on the published research, design, and performance of flowmeters as well as on the claims of flowmeter manufacturers. It will be of use to engineers, particularly mechanical and process engineers, and also to instrument companies' marketing, manufacturing, and management personnel as they seek to identify future products.

I have concentrated on the process, mechanical, and fluid engineering aspects and have given only as much of the electrical engineering details as are necessary for a proper understanding of how and why the meters work. I am not an electrical engineer and so have not attempted detailed explanations of modern electrical signal processing. I am also aware of the speed with which developments in signal processing would render any descriptions which I might give out of date.

In the bibliography, other books dealing with flow measurement are listed, and my intention is not to retread ground covered by them, more than is necessary and unavoidable, but to bring together complementary information. I also make the assumption that the flowmeter engineer will automatically turn to the appropriate standard; therefore, I have tried to avoid reproducing information that should be obtained from those excellent documents. I include a brief list that categorizes a few of the standards according to meter or application. I also recommend that those involved in new developments keep a watchful eye on the regular conferences, which carry much of the latest developments in the business.

I hope, therefore, that this book will provide a signpost to the essential information required by all involved in the development and use of flowmeters, from the field engineer to the chief executive of the entrepreneurial company that is developing its product range in this technology.

In this book, following introductory chapters on accuracy, flow, selection, and calibration, I have attempted to provide a clear explanation of each type of flowmeter so that the reader can easily understand the workings of the various meters. I have then attempted to bring together a significant amount of the published information that explains the performance and applications of flowmeters. The two sources for this are the open literature and the manufacturers' brochures. I have also introduced, to a varying extent, the mathematics behind the meter operations, but to avoid disrupting the text, I have consigned this, in most cases, to the appendices at the end of many chapters. This follows the approach that I have used for technical review papers on turbine meters, Coriolis meters, and to a lesser extent earlier papers on

electromagnetic flowmeters, positive displacement flowmeters, and flowmeters in multiphase flows.

However, when searching the appropriate databases for flowmeter papers, I quickly realized that including references to all published material was unrealistic. I have attempted to select those references that appeared to be most relevant and available to the typical reader of this book. However, the reader is referred to the list of journals and conferences that were especially valuable in writing this book. In particular, the *Journal of Flow Measurement and Instrumentation* has filled a gap in the market, judging by the large number and high quality of the papers published by the journal. It is likely that, owing to the problems of obtaining papers, I have omitted some that should have been included.

Topics that I do not consider to be within the subject of bulk flow measurement of liquids and gases, and that are not covered in this book, are metering pumps, flow switches, flow controllers, flow measurement of solids and granular materials, open channel flow measurement, hot-wire local velocity probes or laser doppler anemometers, and subsidiary instrumentation.

In two areas where I know that I am lacking in first-hand knowledge – modern control methods and manufacturing – I have included a brief review, which should not be taken as expert information. However, I want to provide a source of information for existing and prospective executives in instrumentation companies who might need to identify the type of products for their companies' future developments. This requires a knowledge of the market for each type of flowmeter and also an understanding of who is making each type of instrument. It requires some thought regarding the necessities of manufacturing and production and the implications for this in any particular design.

I have briefly referred to future directions for development in each chapter where appropriate, and in the final chapter I have drawn these ideas together to provide a forward look at flow metering in general.

The techniques for precise measurement of flow are increasingly important today when the fluids being measured, and the energy involved in their movement, may be very expensive. If we are to avoid being prodigal in the use of our natural resources, then the fluids among them should be carefully monitored. Flow measurement contributes to that monitoring and, therefore, demands high standards of precision and integrity.

Acknowledgments

My knowledge of this subject has benefited from many others with whom I have worked and talked over the years. These include colleagues from industry and academia, and students, whether in short courses or longer-term degree courses and research. I hope that the book does justice to all that they have taught me.

In writing this book, I have drawn on the information from many manufacturers, and some have been particularly helpful in agreeing to the use of information and diagrams. I have acknowledged these companies in the captions to the figures. Some went out of their way to provide artwork, and I am particularly grateful to them. Unfortunately, space, in the end, prevented me from using many of the excellent diagrams and photographs with which I was provided.

In the middle of already busy lives, the following people kindly read through sections of the book, of various lengths, and commented on them: Heinz Bernard (Krohne Ltd.), Reg Cooper (Bailey–Fischer & Porter), Terry Cousins (T&SK Flow Consultants), Chris Gimson (Endress & Hauser), Charles Griffiths [Flow Automation (UK) Ltd.], John Hemp (Cranfield University), Yousif Hussain (Krohne Ltd.), Peter Iles-Smith (Yokagawa United Kingdom Limited), Alan Johnson (Fisher-Rosemount), David Lomas (ABB Kent-Taylor), Graham Mason (GEC–Marconi Avionics), John Napper (formerly with FMA Ltd.), Kyung-Am Park (Korea Research Institute of Standards and Science), Bob Peters (Daniel Europe Ltd.), Roger Porkess (University of Plymouth), Phil Prestbury (Fisher-Rosemount), David Probert (Cambridge University), Karl-Heinz Rackebrandt (Bailey–Fischer & Porter), Bill Pursley (NEL), Jane Sattary (NEL), Colin Scott (Krohne Ltd.), John Salusbury (Endress & Hauser), Dave Smith (NEL), Ian Sorbie (Meggitt Controls), Eddie Spearman (Daniel Europe Ltd.), J. D. Summers-Smith (formerly with I.C.I), and Ben Weager (Danfoss Flowmetering Ltd.). I am extremely grateful to them for taking time to do this and for the constructive comments they gave. Of course, I bear full responsibility for the final script, although their help and encouragement was greatly valued.

I am also grateful to Dr. Michael Reader-Harris for his advice on the orifice plate discharge coefficient equation, and to Prof. Stan Hutton for his help and encouragement when Chapter 10 was essentially in the form of technical papers.

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I am also grateful for the help and encouragement given to me by many in the preparation of this book. It would be difficult to name them, but I am grateful for each contribution. The support of my family must be mentioned. In various ways they all contributed – by offering encouragement, by undertaking some literature searches, by doing some early typing work, and by helping with some of the diagrams. I particularly thank my wife whose encouragement and help at every stage, not to mention putting up with a husband glued to the word processor through days, evenings, holidays, etc., ensured that the book was completed.

My editor has been patient, first as I overran the agreed delivery date and then as I overran the agreed length. I am grateful to Florence Padgett for being willing to overlook this lack of precision!

Nomenclature

CHAPTER 1

c_i	Sensitivity coefficient
$f(x)$	Function for Normal distribution
K	K factor in pulses per unit flow quantity
k	Coverage factor
M	Mean of a sample of n readings
m	Index
$N(\mu, \sigma^2)$	Normal curve
n	Number of measurements, Exponent
p	Probability, Index
$\bar{q}$	Mean of n measurements q_j , Exponent
q_j	Test measurement
q_v	Volumetric flow rate
q_{vo}	Volumetric flow rate at calibration point
r	Exponent
s	Exponent
$s(\bar{q})$	Experimental standard deviation of mean of group q_j
$s(q_j)$	Experimental standard deviation of q_j
t	Student's t
U	Expanded uncertainty
$u(x_i)$	Standard uncertainty for the i th quantity
$u_c(y)$	Combined standard uncertainty
x	Coordinate
x_i	Result of a meter measurement, Input quantities
$\bar{x}$	Mean of n meter measurement
y	Output quantity
z	Normalized coordinate $(x - \mu)/\sigma$
μ	Mean value of data for normal curve
ν	Degrees of freedom
σ	Standard deviation (σ^2 variance)
$\Phi(z)$	Area under Normal curve [e.g., $\Phi(0.5)$ is the area from $z = -\infty$ to $z = 0.5$]
$\phi(x)$	Function for normalized Normal distribution

CHAPTER 2

A	Cross-section of pipe
c	Local speed of sound
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
D	Diameter of pipe

d	Diameter of tube bundle straightener tubes
g	Acceleration due to gravity
H	Hodgson's number
K	Pressure loss coefficient
M	Mach number
n	Index as in Equation (2.4)
p	Pressure
p_0	Stagnation pressure
Δp_{loss}	Pressure loss across a pipe fitting
q_v	Volumetric flow rate
q_m	Mass flow rate
R	Radius of pipe
Re	Reynolds number
r	Radial coordinate (distance from pipe axis)
T	Temperature
T_0	Stagnation temperature
V	Velocity in pipe, Volume of pipework and other vessels between the source of the pulsation and the flowmeter position
V_0	Velocity on pipe axis
V_{rms}	Fluctuating component of velocity
$\bar{V}$	Mean velocity in pipe
z	Elevation above datum
γ	Ratio of specific heats
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Density

SUBSCRIPTS

1,2	Pipe sections

CHAPTER 3

c_i	Sensitivity coefficient for the i th quantity
h_{min}	Lubrication film thickness
n	Bearing rotational speed
p	Bearing load
u_i	Standard uncertainty for the i th quantity
η	Friction coefficient
λ	Specific film thickness
μ	Fluid viscosity

σ Combined roughness of the two contacting surfaces of the bearing

CHAPTER 4

C_d Concentration of tracer in the main stream at the downstream sampling point
 $C_{d\text{mean}}$ Mean concentration of tracer measured downstream during time t
 C_i Concentration of tracer in the injected stream
 C_u Concentration of tracer in the main stream upstream of injection point (if the tracer material happens to be present)
 c_i Sensitivity coefficient
 M_n Net mass of liquid collected in calibration
 p Pressure
 q_v Volumetric flow rate in the line
 q_{vi} Volumetric flow rate of injected tracer
 R Gas constant for a particular gas
 T Temperature
 t Collection time during calibration, Integration period for tracer measurement
 V Amount injected in the sudden injection (integration) method
 v Specific volume
 ρ Liquid density

CHAPTER 5

A Function of β and Re
 a_1 Expression in orifice plate bending formula
 a_ϵ Constant
 b Constant
 b_ϵ Constant
 C Discharge coefficient
 C_{Re} Part of discharge coefficient affected by Re
 C_{Taps} Part of discharge coefficient which allows for position of taps
 C_∞ Discharge coefficient for infinite Reynolds number
 c_1 Expression in orifice plate bending formula
 c_ϵ Constant
 D Pipe diameter (ID)
 D' Orifice plate support diameter
 d Orifice diameter
 E Velocity of approach factor $(1 - \beta^4)^{-1/2}$, Thickness of the orifice plate
 E_T Total error in the indicated flow rate of a flowmeter in pulsating flow
 E^* Elastic modulus of plate material
 e Thickness of the orifice
 F Correction factors used to obtain the mass flow of a (nearly) dry steam flow

f Frequency of the pulsation
 H Hodgson number
 h Thickness of orifice plate
 K Loss coefficient, Related to the criterion for Hodgson's number
 $L_1 = l_1/D$
 $L'_2 = l'_2/D$ (The prime signifies that the measurement is from the downstream face of the plate)
 l_1 Distance of the upstream tapping from the upstream face of the plate
 l'_2 Distance of the downstream tapping from the downstream face of the plate (The prime signifies that the measurement is from the downstream face of the plate)
 $M'_2 = 2L'_2/(1 - \beta)$
 n Index
 p_d Downstream pressure
 p_u Upstream pressure
 Δp Differential pressure, pressure drop between pulsation source and meter
 q_m Mass flow rate
 q_v Volumetric flow rate
 Re Reynolds number usually based on the pipe ID
 r Radius of upstream edge of orifice plate
 t Time
 V Volume of pipework and other vessels between the source of the pulsation and the flowmeter position
 $\bar{V}$ Mean velocity in pipe with pulsating flow
 V_{rms} Root-mean-square value of unsteady velocity fluctuation in pipe with pulsating flow
 x Dryness fraction
 α Flow coefficient, CE
 β Diameter ratio, d/D
 γ Ratio of specific heats
 δq_m Small changes or errors in q_m , etc.
 ϵ Expansibility (or expansion) factor
 ϵ_1 Expansibility (or expansion) factor for orifice
 κ Isentropic exponent
 ρ_1 Density at the upstream pressure tapping cross-section
 σ_y Yield stress for plate material
 Φ_{fo}^2 Ratio of two-phase pressure drop to liquid flow pressure drop
 ϕ Maximum allowable percentage error in pulsating flow

CHAPTER 6

C Coefficient of discharge
 C_{Re} Part of coefficient of discharge affected by Reynolds number

C_{tp}	Coefficient for wet gas flow equation
C_{∞}	Discharge coefficient for infinite Reynolds number
D	Pipe ID
d	Throat diameter
E	Velocity of approach factor ($1 - \beta^4$) ^{-1/2}
Fr_g	Superficial gas Froude number
g	Gravitational acceleration
k	Roughness
n	Index
Δp	Differential pressure
q_g	Gas volumetric flow rate
q_l	Liquid volumetric flow rate
q_m	Mass flow rate
q_{tp}	Apparent volumetric flow rate when liquid is present in the gas stream
q_v	Volume flow rate
Re	Reynolds number based on D
Re_d	Reynolds number based on d
V_{sg}	Superficial gas velocity
X	Lockhart-Martinelli parameter
β	diameter ratio d/D
ϵ	Expansibility (or expansion) factor
ρ	Density
ρ_g	Gas density
ρ_l	Liquid density

CHAPTER 7

A_2	Outlet cross-sectional area
A_*	Throat cross-sectional area
a	Constant
a_c	Constant
a_z	Constant
b	Constant
b_c	Constant
b_z	Constant
C	Discharge coefficient
C_R	$= C_* \sqrt{Z}$
C_*	Critical flow function
c	Sound speed
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
d	Throat diameter
d_t	Diameter of tapping
d_2	Outlet diameter
f	Obtained from Equation (7.17)
M	Mach number
M_1	Mach number at inlet when stagnation conditions cannot be assumed
M	Molecular weight
n	Exponent in Equation (7.12)
p_0	Stagnation pressure
p_1	Pressure at inlet when stagnation conditions cannot be assumed
p_{2i}	Ideal outlet pressure
p_{2max}	Actual maximum outlet pressure
p_*	Throat pressure in choked conditions

q_m	Mass flow
R	Universal gas constant
Re_d	Reynolds number based on the throat diameter
T_0	Stagnation temperature
T_*	Throat temperature in choked conditions
X	Mole fraction of each component of a gas mixture
Z	Compressibility factor
Z_0	Compressibility factor at stagnation conditions
β	d/D
γ	Ratio of specific heats
ϵ	Error
κ	Isentropic exponent
ν	Kinematic viscosity
ρ_0	Density at stagnation conditions

CHAPTER 8

A	Cross-sectional area of the pipe, Constant
A'	Constant
A_f	Cross-sectional area of float
A_x	Cross-sectional area of tapering tube at height x
A_2	Annular area around float, Annular area around target
a	Area of target
B	Constant
C	Coefficient
C_c	Contraction coefficient, Constants in curve fit for target meter discharge coefficient
D	Pipe diameter
d	Throat diameter for pipe inlet
E	Full-scale or upper range value of flow rate used in precision calculation
F	Summation error in flow rate
g	Gravity
K	Loss coefficient, Precision class, Bend or elbow meter coefficient
L	Length of laminar flow tube
M	Actual flow rate used in precision calculation
p	Pressure
q_v	Volumetric flow rate
R	Radius of bend or elbow
Re	Reynolds number
V	Velocity, Volume of float
$\bar{V}$	Mean velocity in tube
v	Specific volume of gas
x	Height of float in tube
μ	Viscosity
μ_g	Viscosity of calibration gas at flowing conditions
μ_{std}	Viscosity of reference gas at standard conditions
ρ	Density
ρ_f	Density of float material

CHAPTER 9

E	Young's modulus of elasticity
F	Friction force
g	Acceleration due to gravity
L	Length of clearance gap in direction of flow
ℓ	Axial length of clearance gap
ℓ_{ax}	Axial length of measuring chamber
M	Mass of rotor
N	Rotational speed of rotor
N_i	Rotational speed of cam disk of calibrator (clutch system)
N_{IN}	Rotational speed of shaft into calibrator (epicyclic system)
N_{max}	Maximum rotational speed of rotor
N_o	Rotational speed of outer ring of calibrator (clutch system)
N_{OUT}	Rotational speed of shaft out of calibrator (epicyclic system)
N_s	Slippage in N_{IN}
n	Number of teeth on gear
p_d	Downstream pressure
p_u	Upstream pressure
q_{ideal}	As used in Equation (9.A.6)
$q_{leakage}$	Leakage flow rate
q_{BULK}	Bulk volumetric flow rate
q_{slip}	Volumetric flow rate through meter at no rotation
q_v	Volumetric flow rate
R	Radius of friction wheel
r	Radius of point of friction wheel contact
r_i	Inner radius of measuring chamber
r_o	Outer radius of measuring chamber
r_s	Shaft radius
T	Temperature, Torque
T_{IN}	Torque on input shaft to calibrator
T_{OUT}	Torque out from calibrator
T_0	Constant drag torque
T_1	Speed-dependent drag torque
t	Clearance between stationary and moving members
t_o	Thickness of outer casing
u	Fluid velocity
y	Position coordinate across clearance gap
α_m	Coefficient of linear expansion of metal
α_l	Coefficient of volume expansion of liquid
Δ	Change in quantity, Distance between center of rotation of arm and center of discs in calibrator
δ	Reduction in area of the measuring chamber due to blade sections
θ_c	Angular position of center disk of calibrator
θ_i	Angular position of cam disk of calibrator
θ_o	Angular position of outer ring of calibrator

μ	Viscosity
Φ	Angular position of arm of calibrator

OTHER SUBSCRIPTS

j	Dummy suffix for summation
l	Liquid
m	Metal
$1,2$	Different materials
$1-8$	Epicyclic gears

CHAPTER 10

A	Cross-sectional area of the effective annular flow passage at the rotor blades
a	Turbine meter aerodynamic torque coefficient
a_0, a_1, a_2, a_3	Constant coefficients
B	Axial length of rotor
b	Turbine meter aerodynamic torque coefficient, Bearing length, Frequency response coefficient unless used as an exponent
C	Proportional to $1/K$
C_D	Drag coefficient
C'_D	Drag coefficient adjusted to allow for C_h
C_f	Fluid drag coefficient
C_h	Constant for a particular design (Behaves like an adjustment to the main aerodynamic drag term coefficient)
C_L	Lift coefficient
c	Chord
D	Drag force, Pipe diameter, Impeller diameter
D_1	Rotor response parameter
D_2	Fluid drag parameter
D_3	Nonfluid drag parameter
d_1, d_2	Additional geometric variables
e	Meter error
F	Nonfluid forces
f	Frequency of blade passing
I	Mass moment of inertia of rotor system about rotor axis
i	Incidence angle
$i(r)$	Incidence at radius r
K	Lattice effect coefficient, K factor (pulses/unit volume)
$K(r)$	Lattice coefficient at radius r
K_1, K_2	Turbine meter resisting torque coefficients
K_h	Constant used in equation for helical blade angle
L	Lift force, Helical pitch of blades
N	Number of blades
n	Rotational speed (frequency)
p	Index in error equation
p_{01}	Stagnation pressure at inlet
p_{02}	Stagnation pressure at outlet

[illegible]

CHAPTER 12

A	Used for area of electrode leads forming a loop causing quadrature signals in the electromagnetic flowmeter
a	Pipe radius
B	Magnetic flux density in tesla
$\mathbf{B}$	Magnetic flux density vector
B_0	Maximum value of magnetic flux density
B_θ	θ component of magnetic flux density
b	Inner radius of an annulus of conducting fluid in two phase annular flow
D	Diameter of pipe
$\mathbf{E}$	Electric field vector
f	Frequency of magnetic field excitation
$\mathbf{j}$	Current density vector
l	Length of wire traversing magnetic field
Δp	Pressure of the pipeline fluid above atmospheric
q_v	Volumetric flow rate
r	Radial coordinate
S	Electromagnetic flowmeter sensitivity
t	Pipe wall thickness
U	Electric potential
ΔU_{EE}	Voltage between electrodes
ΔU_P	Voltage across a wire P moving through a magnetic field (similarly for Q and R)
V	Velocity
$\mathbf{V}$	Velocity vector
V_m	Mean velocity in the pipe in meters per second
W	Weighting function
$\mathbf{W}$	Weight function vector
W'	Rectilinear flow weight function
W''	Axisymmetric weight function
W_z	Axial component of W
α	Void fraction
ϵ	Strain
θ	Cylindrical coordinate
μ	Electric permeability
σ	Conductivity
Φ	Scalar potential

CHAPTER 13

A	Weighting factors for Gaussian quadrature
a	Radius of pipe
c	Velocity of ultrasound in the medium, Coefficients of a polynomial
D	Pipe diameter
f	Frequency of stable quartz oscillator for ultrasonic measurement system
f_d	Frequency for the downstream sing-around pulse train
f_t	Transmitted doppler frequency
f_r	Reflected doppler frequency
f_u	Frequency for the upstream sing-around pulse train

Δf	Difference between the sing-around frequencies, Frequency shift in doppler flowmeter
$f(t)$	Function of t
h	Displacement of ultrasonic beam from axis of pipe
I	Driving current in ultrasonic flowmeter
k_s	Adiabatic compressibility
L	Distance along path in transit-time flowmeter, Distance between transducers in ultrasonic correlation flowmeter
N_d, N_n	Counters for ultrasonic measuring system
N_t, N_u	
n	$1/n$ is the index for turbulent profile curve fit, Index
P_R	Ultrasound power reflected
P_T	Ultrasound power transmitted
q_m	Mass flow rate
Re	Reynolds number
$R_{yx}(\tau)$	Correlation coefficient
T	Correlation time for integration
t	Pipe wall thickness
t_d	Downstream wave transit time
t_m	Mean wave transit time
t_u	Upstream wave transit time
Δt	Difference between these two wave transit times
t_1, t_2	Beam positions for ultrasonic flowmeter using Gaussian quadrature
$U_I^{(2)}, U_{II}^{(1)}$	Received voltages for ultrasonic flowmeter
V	Flow velocity in pipe
V_m	Mean velocity in pipe
$V(x)$	Flow velocity profile along pipe chord
$\mathbf{V}_s$	Ultrasonic flowmeter created flow
$\mathbf{v}_o$	Acoustic field for cases (1) and (2)
v	Relative flow velocity in the direction of the acoustic beam
$\mathbf{W}$	Vector weight function
X	Axial length between transducers in ultrasonic flowmeter
Y	Length of chord in the plane of the ultrasonic path
y	Distance along chord used in ultrasonic flowmeter
Z	Acoustic impedance
z	Ultrasonic beam deflection distance on opposite wall
β	Ultrasonic beam deflection angle
γ	Ratio of specific heats for a particular gas
θ	Ultrasound beam angle
λ	Ultrasound wave length
ρ	Density of material
ρ_m	Mean density in ultrasonic meter
τ	Measurement period for ultrasonic system, Time period between ultrasonic wave peaks
τ_m	Mean ultrasonic correlation time

CHAPTER 14

A	Area of duct
c	Speed of sound
K	Constant
k_s	Adiabatic compressibility
p	Pressure
Δp_A	Pressure drop to throat of venturi A
Δp_B	Pressure drop to throat of venturi B
Δp_{AB} etc.	Differential pressure between venturi throats, across limbs of the hydraulic Wheatstone bridge, or across diagonals of the hydraulic Wheatstone bridge
q_m	Total mass flow
q_{vp}	Metering pump volumetric transfer flow
V	Velocity in the meter
γ	Ratio of specific heats
ρ	Density of the fluid

CHAPTER 15

A	Area of the duct, Constant equal to $1/\pi$
B	Constant equal to $(2/\pi)^{0.5}$
C, D	Functions of temperature
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
D	Pipe diameter
d	Heating element diameter
g	Gravitational constant
I	Current through resistance R
K	Constants
k	Thermal conductivity of the fluid
k'	Constant allowing for heat transfer and temperature difference at zero flow
L	Finite difference dimension
Nu	Nusselt number
n	Index
Pe	Peclet number
Pr	Prandtl number
p	Pressure
Q_h	Heat transfer
q_a	Rate of heat addition per unit volume
q_h	Heat flux
$\mathbf{q}_h$	Heat flux vector
q_m	Mass flow rate
q_v	Volumetric flow rate
R	Resistance of heating element
Re	Reynolds number
S	Flow signal
S_0	Flow signal at start
St	Stanton number
T	Absolute temperature of the fluid
T_1, T_2, T_c	Temperatures used in finite difference approximation
ΔT	Measured temperature difference in Kelvins
t	Time from start of flow change
V	Fluid velocity
$\mathbf{V}$	Vector velocity

$\mathbf{W}$	Weight vector
ρ	Fluid density
μ	Dynamic viscosity of the fluid
ν	Kinematic viscosity
τ	Time constant of flowmeter

CHAPTER 16

q_m	Mass flow
R	Radius of the annulus of angular momentum meter
s	Spring constant
X	Angular momentum
θ	Angular deflection
τ	Time difference between markers on the angular momentum meter rotating assembly
ω	Angular velocity of the rotor

CHAPTER 17

A	Cross-sectional area of the pipe
d	Width of U-tube
F	Force due to Coriolis acceleration
I_s	Inertia in plane of twisting oscillation
I_u	Inertia in plane of normal oscillation
K	Constant, Allows for the fact that the twist of the tube will not form a straight integration
K_s	Spring constant of the U-tube in twisting oscillation
K_u	Spring constant of the U-tube in normal oscillation
l	Length of the U-tube
δm	Element of mass equal to $\rho A \delta r'$
q_m	Mass flow
r	Radius
δr	Elementary length of tube
$\delta r'$	Elementary length of fluid
T	Torque
t	Time
V	Flow velocity
$\mathbf{V}$	Vector velocity
$\mathbf{v}^{(1)}$	Oscillatory velocity field set up in the stationary fluid by the driving oscillator
$\mathbf{v}^{(2)}$	Oscillatory velocity field set up in the stationary fluid by the Coriolis forces
$\mathbf{W}$	Vector weight function
θ	Twist angle of U-tube
θ_0	Amplitude of twist angle of U-tube when in sinusoidal motion
ρ	Density
τ	Difference in transit time of two halves of twisted U-tube
$\nabla \phi$	Phase difference between the total velocities at the two sensing points
Ω	Angular velocity of the pipe caused by the vibration

Ω_0	Amplitude of the angular velocity of the pipe caused by the vibration
ω	Driving frequency
ω_s	Natural frequency of U-tube in twisting oscillation
ω_u	Natural frequency of U-tube in normal oscillation

CHAPTER 18

k	Coefficient
M	Mach number
p	Pressure
p_0	Stagnation pressure
Δp	Dynamic pressure
V	Velocity
γ	Ratio of specific heats
ρ	Density

CHAPTER 20

C	Cost
cp	Component precision
D	Pipe diameter
d	Width of Coriolis flowmeter U-tube
eq	Electronics quality
K	Constant
K_s	Spring constant of the U-tube in twisting oscillation
ma	Manufacturing accuracy
mc	Material consistency
$p_1, p_2,$ p_3, p_4	Dimensional and other factors
q_m	Mass flow rate
q_v	Volumetric flow rate
S	Flow signal
Δt	Time difference between upstream and downstream waves or pulses
t_m	Mean time of transit used to obtain sound speed
θ	Ultrasound beam angle
τ	Difference in transit time of two halves of twisted U-tube
ω	Driving frequency
ω_s	Natural frequency of U-tube in twisting oscillation

COMBINED ALL CHAPTERS

A	Cross-section of pipe, Function of β and Re , Cross-sectional area of the pipe, Constant, Cross-sectional area of the effective annular flow passage at the rotor blades, Constant of value about 3.0 used in equation for avoiding cavitation in flows past vortex meters, Used for area of electrode leads forming a loop causing quadrature signals in the electromagnetic flowmeter, Weighting factors for Gaussian quadrature, Area of the duct, Constant equal to $1/\pi$

A'	Constant
A_f	Cross-sectional area of float
A_{min}	Flow area past bluff body
A_x	Cross-sectional area of tapering tube at height x
A_2	Outlet cross-sectional area, Annular area around float, Annular area around target
A_*	Throat cross-sectional area
a	Constant, Area of target, Turbine meter aerodynamic torque coefficient, Area when integrating vorticity, Pipe radius
$a_0, a_1,$ a_2, a_3	Constant coefficients
a_1	Expression in orifice plate bending formula
a_c	Constant
a_z	Constant
a_ϵ	Constant
B	Constant, Axial length of rotor, Constant of value about 1.3 used in equation for avoiding cavitation in flows past vortex meters, Magnetic flux density in tesla, Constant equal to $(2/\pi)^{0.5}$
$\mathbf{B}$	Magnetic flux density vector
B_0	Maximum value of magnetic flux density
B_θ	θ component of magnetic flux density
b	Inner radius of an annulus of conducting fluid in two-phase annular flow
b	Constant, Turbine meter aerodynamic torque coefficient, Bearing length, Frequency response coefficient unless used as an exponent
b_c	Constant
b_z	Constant
b_ϵ	Constant
C	Discharge coefficient, Coefficient, Proportional to $1/K$, Function of temperature, Cost
C_c	Contraction coefficient, Constants in curve fit for target meter discharge coefficient
C_D	Drag coefficient
C'_D	Drag coefficient adjusted to allow for C_h
C_d	Concentration of tracer in the main stream at the downstream sampling point
C_{dmean}	Mean concentration of tracer measured downstream during time t
C_f	Fluid drag coefficient
C_h	Constant for a particular design (Behaves like an adjustment to the main aerodynamic drag term coefficient)
C_i	Concentration of tracer in the injected stream
C_L	Lift coefficient
C_R	$= C_* \sqrt{Z}$

C_{Re}	Part of discharge coefficient affected by Re	f_t	Transmitted doppler frequency
C_{Taps}	Part of discharge coefficient that allows for position of taps	f_r	Reflected doppler frequency
C_{tp}	Coefficient for wet gas flow equation	f_u	Frequency for the upstream sing-around pulse train
C_u	Concentration of tracer in the main stream upstream of injection point (if the tracer material happens to be present)	Δf	Difference between the sing-around frequencies, Frequency shift in doppler flowmeter
C_∞	Discharge coefficient for infinite Reynolds number	$f(t)$	Function of t
C_*	Critical flow function	$f(x)$	Function for Normal distribution
c	Sound/ultrasound speed, Chord, Coefficients of a polynomial	g	Acceleration due to gravity
c_1	Expression in orifice plate bending formula	H	Hodgson number, Streamwise length of bluff body
c_i	Sensitivity coefficient	h	Thickness of orifice plate, Parallel flats on sides of bluff body at leading edge, Displacement of ultrasonic beam from axis of pipe
c_p	Specific heat at constant pressure	h_{min}	Lubrication film thickness
cp	Component precision	I	Mass moment of inertia of rotor system about rotor axis, Driving current in ultrasonic flowmeter, Current through resistance R
c_v	Specific heat at constant volume	I_s	Inertia in plane of twisting oscillation
c_ϵ	Constant	I_u	Inertia in plane of normal oscillation
D	Pipe diameter (ID), Drag force, Impeller diameter, Function of temperature	i	Incidence angle
D'	Orifice plate support diameter	$i(r)$	Incidence at radius r
D_1	Rotor response parameter	$\mathbf{j}$	Current density vector
D_2	Fluid drag parameter	K	Pressure loss coefficient, Related to the criterion for Hodgson's number, Precision class, Bend or elbow meter coefficient, Lattice effect coefficient, K factor (pulses/unit volume), Calculation factor for shedding frequency (Zanker and Cousins 1975), Constant, Allows for the fact that the twist of the tube will not form a straight integration
D_3	Nonfluid drag parameter	$K(r)$	Lattice coefficient at radius r
d	Diameter of tube bundle straightener tubes, Orifice diameter, Throat diameter, Heating element diameter, Width of Coriolis flowmeter U-tube	K_1, K_2	Turbine meter resisting torque coefficients
d_t	Diameter of tapping	K_h	Constant used in equation for helical blade angle
d_1	Additional geometric variable	K_s	Spring constant of the U-tube in twisting oscillation
d_2	Outlet diameter, Additional geometric variable	K_u	Spring constant of the U-tube in normal oscillation
E	velocity of approach factor $(1 - \beta^4)^{-1/2}$, Thickness of the orifice plate, Full-scale or upper range value of flow rate used in precision calculation, Young's modulus of elasticity, Fluid compressibility	k	Coverage factor, Roughness, Thermal conductivity of the fluid, Coefficient
$\mathbf{E}$	Electric field vector	k'	Constant allowing for heat transfer and temperature difference at zero flow
E_T	Total error in the indicated flow rate of a flowmeter in pulsating flow	k_s	Adiabatic compressibility
E^*	Elastic modulus of plate material	L	Length of laminar flow tube, Length of clearance gap in direction of flow, Lift force, Helical pitch of blades, Length of bluff body across pipe between end fittings, Distance along path in transit-time flowmeter, Distance between transducers in ultrasonic correlation flowmeter, Finite difference dimension
e	Thickness of the orifice, Meter error		
eq	Electronics quality		
F	Correction factors used to obtain the mass flow of a (nearly) dry steam flow, Summation error in flow rate, Friction force, Nonfluid forces, Force due to Coriolis acceleration		
Fr_g	Superficial gas Froude number		
f	Frequency of the pulsation, Obtained from Equation (7.17), Frequency of blade passing, Shedding frequency, Frequency of magnetic field excitation, Frequency of stable quartz oscillator for ultrasonic measurement system		
f_d	Frequency for the downstream sing-around pulse train		

L_1	l_1/D	p_0	Stagnation pressure
L'_2	l'_2/D (The prime signifies that the measurement is from the downstream face of the plate)	p_u	Upstream pressure
l	Length of wire traversing magnetic field, Length of the U-tube	p_v	Saturated liquid vapor pressure
l_1	Distance of the upstream tapping from the upstream face of the plate	p_{01}	Stagnation pressure at inlet
l'_2	Distance of the downstream tapping from the downstream face of the plate (The prime signifies that the measurement is from the downstream face of the plate)	p_{02}	Stagnation pressure at outlet
ℓ	Axial length of clearance gap	p_1	Pressure at inlet when stagnation conditions cannot be assumed, Static pressure at inlet, p_1 p_2 dimensional and other factors
ℓ_{ax}	Axial length of measuring chamber	p_2	Static pressure at outlet
M	Mean of a sample of n readings, Mach number, Actual flow rate used in precision calculation, Mass of rotor	p_{2i}	Ideal outlet pressure
M	Molecular weight	p_{2max}	Actual maximum outlet pressure
M_1	Mach number at inlet when stagnation conditions cannot be assumed	p_*	Throat pressure in choked conditions
M'_2	$= 2L'_2/(1 - \beta)$	Δp_A	Pressure drop to throat of venturi A
M_n	Net mass of liquid collected in calibration	Δp_B	Pressure drop to throat of venturi B
m	Index	Δp_{AB} etc.	Differential pressure between venturi throats, across limbs of the hydraulic Wheatstone bridge, or across diagonals of the hydraulic Wheatstone bridge
δm	Element of mass equal to $\rho A \delta r'$	Δp_{loss}	Pressure loss across a pipe fitting
ma	Manufacturing accuracy	Δp	Differential pressure, Pressure drop between pulsation source and meter, Pressure drop across vortex meter (about 1 bar at 10 m/s), Pressure of the pipeline fluid above atmospheric, Dynamic pressure
mc	Material consistency	Q_h	Heat transfer
N	Rotational speed of rotor, Number of blades	q	Actual average flow rate; Exponent in error equation
N_i	Rotational speed of cam disk of calibrator (clutch system)	$\bar{q}$	Mean of n measurements q_i
N_{IN}	Rotational speed of shaft into calibrator (epicyclic system)	q_0	Time average flow rate over period T
N_{max}	Maximum rotational speed of rotor	q_1	Initial flow rate
N_o	Rotational speed of outer ring of calibrator (clutch system)	q_2	Final flow rate
N_{OUT}	Rotational speed of shaft out of calibrator (epicyclic system)	q_a	Rate of heat addition per unit volume
N_s	Slippage in N_{IN}	q_b	Base flow rate
N_d, N_n	Counters for ultrasonic measuring system	q_{BULK}	Bulk volumetric flow rate
N_t, N_u	Normal curve	q_g	Gas volumetric flow rate
$N(\mu, \sigma^2)$	Nusselt number	q_h	Heat flux
Nu	Number of measurements, Exponent in equations, Bearing rotational speed, Number of teeth on gear, Rotational speed (frequency)	q_h	Heat flux vector
n	Probability, Pressure, Bearing load, Exponent in error equation	q_{ideal}	As used in Equation (9.A.6)
p	Ultrasound power reflected	q_i	Indicated average flow rate
P_R	Ultrasound power transmitted	q_j	Test measurement
P_T	Peclet number	q_l	Liquid volumetric flow rate
Pe	Prandtl number	$q_{leakage}$	Leakage flow rate
Pr	Atmospheric pressure	q_m	Mass flow rate
p_{atmos}	Downstream pressure	q_{max}	Maximum flow rate for which the meter is designed
p_d	Minimum back pressure 5D downstream of vortex meter	q_{min}	Minimum flow rate for which the meter is designed
p_{gmin}		q_n	Normal flow rate
		q_t	Flow rate at time t
		q_{tp}	Apparent volumetric flow rate when liquid is present in the gas stream
		q_{trans}	Flow rate at change of precision
		q_{slip}	Volumetric flow rate through meter at no rotation
		q_v	Volumetric flow rate (in the main line)

q_{vo}	Volumetric flow rate at calibration point	T_1, T_2, T_c	Temperatures used in finite difference approximation
q_{vp}	Metering pump volumetric transfer flow	T_*	Throat temperature in choked conditions
q_{vi}	Volumetric flow rate of injected tracer	ΔT	Measured temperature difference in Kelvins
R	Radius of pipe, Gas constant for a particular gas, Radius of bend or elbow, Radius of friction wheel, Resistance of heating element, Radius of the annulus of angular momentum meter	t	Student's t , Collection time during calibration, Integration period for tracer measurement, Time, Clearance between stationary and moving members, Pipe wall thickness
R	Universal gas constant	t_B	Blade thickness
Re	Reynolds number usually based on the pipe ID	t_d	Downstream wave time
Re_d	Reynolds number based on the throat diameter	t_m	Mean time of transit used to obtain sound speed
$R_{yx}(\tau)$	Correlation coefficient	t_o	Thickness of outer casing
r	Radial coordinate (distance from pipe axis), Radius of upstream edge of orifice plate, Radius of point of friction wheel contact, Exponent, Radius	t_R	Relaxation time
r_h	Hub radius	t_u	Upstream wave time
r_i	Inner radius of measuring chamber	Δt	Time difference between upstream and downstream waves or pulses
r_j	Journal bearing radius	t_1, t_2	Beam positions for ultrasonic flowmeter using Gaussian quadrature
r_t	Tip radius	U	Expanded uncertainty, Electric potential
r_o	Outer radius of measuring chamber, Meter bore radius	ΔU_{EE}	Voltage between electrodes
r_s	Shaft radius	ΔU_P	Voltage across a wire P moving through a magnetic field (similarly for Q and R)
δr	Elementary length of tube	$U_I^{(2)} U_{II}^{(1)}$	Received voltages for ultrasonic flowmeter
$\delta r'$	Elementary length of fluid	$u(x_i)$	Standard uncertainty for the i th quantity
S	Slope of no-flow decay curve at standstill, Strouhal number, Electromagnetic flowmeter sensitivity, Flow signal	$u_c(y)$	Combined standard uncertainty
S_0	Flow signal at start	V	Fluid velocity, Amount injected in the sudden injection (integration) method, Volume of pipework and other vessels between the source of the pulsation and the flowmeter position, Volume of float, Nondimensional fluid velocity
St	Stanton number	$\bar{V}$	Mean velocity in pipe with pulsating flow
s	Exponent, Experimental standard deviation, Blade spacing, Length along curve when integrating velocity around a vortex, Spring constant	V_{rms}	Root-mean-square value of unsteady velocity fluctuation in pipe with pulsating flow
$s(\bar{q})$	Experimental standard deviation of mean of group q_i	V	Velocity vector
$s(q_i)$	Experimental standard deviation of q_i	v ⁽¹⁾	Oscillatory velocity field set up in the stationary fluid by the driving oscillator
T	Temperature, Torque, Fundamental period of pulsating flow	v ⁽²⁾	Oscillatory velocity field set up in the stationary fluid by the Coriolis forces
T_B	Bearing drag torque	V_m	Mean velocity in the pipe in meters per second
T_c	Temperature at calibration	V_{max}	Maximum value of V_z , Velocity of flow past bluff body
T_d	Driving torque	V_{min}	Minimum value of V_z
T_{Fo}	Mechanical friction torque on rotor at zero speed	V_o	Velocity on pipe axis
T_h	Hub fluid drag torque	V_s	Ultrasonic undisturbed flow
T_{IN}	Torque on input shaft to calibrator	v_o	Acoustic field for cases (1) and (2)
T_n	Nonfluid drag torque	V_{sg}	Superficial gas velocity
T_{OUT}	Torque out from calibrator	V_z	Axial velocity (instantaneous inlet fluid velocity of pulsating flow)
T_o	Stagnation temperature, Constant drag torque, Temperature at operation		
T_r	Retarding torque		
T_t	Blade tip clearance drag torque		
T_W	Hub disk friction drag torque		
T_1	Speed-dependent drag torque		

V_0	Time average value of V_z over period T	δ	Reduction in area of the measuring chamber due to blade sections, Deflection of flow at blade outlet from blade angle, Shear layer thickness
V_1	Inlet relative flow velocity	δq_m etc.	Small changes or errors in q_m , etc.
V_2	Outlet relative flow velocity	ϵ	Expansibility (or expansion) factor, Error, Strain
$\bar{V}$	Mean velocity in pipe	ϵ_1	Expansibility (or expansion) factor for orifice
v	Specific volume of gas, Relative flow velocity in the direction of the acoustic beam	η	Friction coefficient, Flow deviation factor
W	Rotor blade velocity, Nondimensional instantaneous rotor velocity under pulsating flow, Weighting function	θ	Cylindrical coordinate, Ultrasound beam angle, Angular deflection, Twist angle of U-tube
W	Weight function vector	θ_c	Angular position of center disk of calibrator
W'	Rectilinear flow weight function	θ_l	Angular position of cam disk of calibrator
W''	Axisymmetric weight function	θ_o	Angular position of outer ring of calibrator, Amplitude of twist angle of U-tube when in sinusoidal motion
W_h	Relative velocity at the hub	κ	Isentropic exponent
W_z	Axial component of W	λ	Specific film thickness, Ultrasound wave length
w	Diameter or width of bluff body	μ	Mean value of data for normal curve, Dynamic viscosity, Fluid viscosity, Electric permeability
X	Axial length between transducers in ultrasonic flowmeter, Mole fraction of each component of a gas mixture, Angular momentum, Lockhart-Martinelli parameter	μ_g	Viscosity of calibration gas at flowing conditions
x	Coordinate, Dryness fraction, Height of float in tube	μ_{std}	Viscosity of reference gas at standard conditions
x_i	Result of a meter measurement	ν	Degrees of freedom, Kinematic viscosity
$\bar{x}$	Mean of n meter measurements	ρ	Density, Fluid density
Y	Length of chord in the plane of the ultrasonic path, Tangential force	ρ_f	Density of float material
γ	Output quantity, Wetness fraction, Position coordinate across clearance gap, Distance along chord used in ultrasonic flowmeter	ρ_m	Mean density in ultrasonic meter
Z	Compressibility factor, Axial force, Acoustic impedance	ρ_0	Density at stagnation conditions
Z_o	Compressibility factor at stagnation conditions	σ	Standard deviation (σ^2 variance), Combined roughness of the two contacting surfaces of the bearing, Conductivity
z	Normalized coordinate $(x - \mu)/\sigma$, Elevation above datum, Ultrasonic beam deflection distance on opposite wall	σ_y	Yield stress for plate material
α	Flow coefficient CE, Wave shape coefficient, Thermal coefficient of expansion, Angle between inlet flow direction and far field flow, Void fraction	τ	Nondimensional time (t/T), Period of modulation, Measurement period for ultrasonic system, Time period between ultrasonic wave peaks, Time constant of flowmeter, Time difference between markers on the angular momentum meter rotating assembly, Difference in transit time of two halves of twisted U-tube
α_m	Coefficient of linear expansion of metal	τ_m	Mean ultrasonic correlation time
α_1	Coefficient of volume expansion of liquid	τ_0	Rotor coast time to standstill
β	Diameter ratio d/D , Blade angle at radius r measured from axial direction of meter, Ultrasonic beam deflection angle	Φ	Angular position of arm of calibrator, Scalar magnetic potential
β_1	Relative inlet angle of flow	$\Phi(z)$	Area under Normal curve [e.g., $\Phi(0.5)$ is the area from $z = -\infty$ to $z = 0.5$]
β_2	Relative outlet angle of flow		
β_m	Mean of the inlet and outlet flow field		
γ	Ratio of specific heats		
Γ	Full-flow amplitude relative to average flow		
Δ	Change in quantity, Distance between center of rotation of arm and center of discs in calibrator		

Φ_{fo}^2	Ratio of two-phase pressure drop to liquid flow pressure drop	ω	Instantaneous rotor angular velocity, Vorticity, Angular velocity of the rotor, Driving frequency
ϕ	Maximum allowable percentage error in pulsating flow	ω_s	Natural frequency of U-tube in twisting oscillation
$\phi(x)$	Function for normalized Normal distribution	ω_u	Natural frequency of U-tube in normal oscillation
$\Delta\phi$	Phase difference between the total velocities at the two sensing points	OTHER SUBSCRIPTS	
Ω	Injected swirling flow rate/total flow rate, Angular velocity of the pipe caused by the vibration	j	Dummy suffix for summation
Ω_o	Amplitude of the angular velocity of the pipe caused by the vibration in sinusoidal motion	l	Liquid
		m	Metal
		1,2	Pipe sections, Different materials
		1-8	Epicyclic gears

CHAPTER 1

Introduction

1.1 INITIAL CONSIDERATIONS

Some years ago at Cranfield, where we had set up a flow rig for testing the effect of upstream pipe fittings on certain flowmeters, a group of senior Frenchmen were being shown around and visited this rig. The leader of the French party recalled a similar occasion in France when visiting such a rig. The story goes something like this.

A bucket at the end of a pipe seemed particularly out of keeping with the remaining high tech rig. When someone questioned the bucket's function, it was explained that the bucket was used to measure the flow rate. Not to give the wrong impression in the future, the bucket was exchanged for a shiny new high tech flowmeter. In due course, another party visited the rig and observed the flowmeter with approval. "And how do you calibrate the flowmeter?" one visitor asked. The engineer responsible for the rig then produced the old bucket!

This book sets out to guide those who need to make decisions about whether to use a shiny flowmeter, an old bucket, nothing at all, or a combination of these! It also provides information for those whose business is the design, manufacture, or marketing of flowmeters. I hope it will, therefore, be of value to a wide variety of people, both in industry and in the science base, who range across the whole spectrum from research and development through manufacturing and marketing. In my earlier book on flow measurement (Baker 1988/9), I provided a brief statement on each flowmeter to help the uninitiated. This book attempts to give a much more thorough review of published literature and industrial practice.

This first chapter covers various general points that do not fit comfortably elsewhere. In particular, it reviews recent guidance on the accuracy of flowmeters (or calibration facilities).

The second chapter reviews briefly some essentials of fluid mechanics necessary for reading this book. The reader will find a fuller treatment in Baker (1996), which also has a list of books for further reading.

A discussion of how to select a flowmeter is attempted in Chapter 3, and some indication of the variety of calibration methods is given in Chapter 4, before going in detail in Chapters 5–17 into the various high (and low) tech meters available. Chapter 18 deals with probes, Chapter 19 gives a brief note on modern control systems, and Chapter 20 provides some reflections on manufacturing and markets. Finally, Chapter 21 raises some of the interesting directions in which the technology is likely to go in the future.

In this book, I have tried to give a balance between the laboratory ideal, the manufacturer's claims, the realities of field experience, and the theory behind the practice. I am very conscious that the development and calibration laboratories are sometimes misleading places, which omit the problems encountered in the field (Stobie 1993), and particularly so when that field happens to be the North Sea. In the same North Sea Flow Measurement Workshop, there was an example of the unexpected problems encountered in precise flow measurement (Kleppe and Danielsen 1993), resulting, in this case, from a new well being brought into operation. It had significant amounts of barium and strontium ions, which reacted with sulfate ions from injection water and caused a deposit of sulfates from the barium sulfate and strontium sulfate that were formed.

With that salutary reminder of the real world, we ask an important – and perhaps unexpected – question.

1.2 DO WE NEED A FLOWMETER?

Starting with this question is useful. It may seem obvious that anyone who looks to this book for advice on selection is in need of a flowmeter, but for the process engineer it is an essential question to ask. Many flowmeters and other instruments have been installed without careful consideration being given to this question and without the necessary actions to ensure proper documentation, maintenance, and calibration scheduling being taken. They are now useless to the plant operator and may even be dangerous components in the plant. Thus before a flowmeter is installed, it is important to ask whether the meter is needed, whether there are proper maintenance schedules in place, whether the flowmeter will be regularly calibrated, and whether the company has allocated to such an installation the funds needed to achieve this ongoing care. Such care will need proper documentation.

The water industry in the United Kingdom has provided examples of the problems associated with unmaintained instruments. Most of us who are involved in the metering business will have sad stories of the incorrect installation or misuse of meters. Reliability-centered maintenance recognizes that the inherent reliability depends on the design and manufacture of an item, and if necessary this will need improving (Dixey 1993). It also recognizes that reliability is preferable in critical situations to extremely sophisticated designs, and it uses failure patterns to select preventive maintenance.

In some research into water consumption and loss in urban areas, Hopkins et al. (1995) found that obstacles to accurate measurements were

- ❑ buried control valves,
- ❑ malfunctioning valves,
- ❑ valve gland leakage,
- ❑ hidden meters that could not be read, and
- ❑ locked premises denying access to meters.

They commented that “water supply systems are dynamic functions having to be constantly expanded or amended. Consequently continuous monitoring, revisions and amendments of networks records is imperative. Furthermore, a proper

programme of inspection, maintenance and subsequent recording must be operative in respect of inter alia:

- ❑ networks,
- ❑ meters,
- ❑ control valves,
- ❑ air valves,
- ❑ pressure reducing valves,
- ❑ non-return valves."

They also commented on the poor upstream pipework at the installation of many domestic meters.

So I make no apology for emphasizing the need to assess whether a flowmeter is actually needed in any specific application.

If the answer is yes, then there is a need to consider the type of flowmeter and whether the meter should be measuring volume or mass. In most cases, the most logical measure is mass. However, by tradition and industrial usage, there are places where volume measurement may be the norm, and as a result, the regulations have been written for volume measurement. This results in a Catch-22 situation. The industry and the regulations may, reasonably, resist change to mass flow measurement until there is sufficient industrial experience, but industrial experience is not possible until the industry and the regulations allow. The way forward is for one or more forward-looking companies to try out the new technology and obtain field experience, confidence in the technology, and approval.

In this book, I have made no attempt to alert the reader to the industry-specific regulations and legal requirements, although some are mentioned. Some regulations are touched on by the various authors, and Miller (1996) is a source of information on many documents. The main objective of the Organisation Internationale de Métrologie Légale (OIML) is to prevent any technical barriers to international trade resulting from conflicting regulations for measuring instruments. With regard to flow measurement, it is particularly concerned with the measurement of domestic supplies and industrial supplies of water and gas (Athane 1994). This is because there are two parties involved, the supplier and the consumer, and the consumer is unlikely to be able to ascertain the correct operation of the meter. In addition these measurements are not monitored continually by the supplier, the meters may fail without anyone knowing, the usage is irregular and widely varying in rate, the measurements are not repeatable, and the commodities have increased in value considerably in recent years.

In order to reduce discussions and interpretation problems between manufacturers and authorized certifying institutes, the European Commission is mandating the European standardization body (CEN/CENELEC) to develop harmonized standards that will give the technical details and implementation of the requirements based on OIML recommendations. These are such that a measuring instrument complies with essential requirements, assuming that the manufacturer has complied with them (Nederlof 1994).

The manufacturer will also be fully aware of the electromagnetic compatibility (EMC), which relates to electromagnetic interference. In particular, the EMC

characteristics of a product are that

- ❑ the level of electromagnetic disturbance generated by the instrument will not interfere with other apparatus, and
- ❑ the operation of the instrument will not be adversely affected by electromagnetic interference from its environment.

In order to facilitate free movement within the European area the CE mark identifies products that conform to the European essential requirements, and all products must be so marked within the European Economic Area (DTI 1993, Chambers 1994).

First, we consider the knotty problem of how accurate the meter should be.

1.3 HOW ACCURATE?

There continues to be inconsistency about the use of terms that relate to accuracy and precision. This stems from a slight mismatch between the commonly used terms and those that the purists and the standards use. Thus we commonly refer to an accurate measurement, when strictly we should refer to one with a small value of uncertainty. We should reserve the use of the word *accurate* to refer to the instrument. A high quality flowmeter, carefully produced with a design and construction to tight tolerances and with high quality materials as well as low wear and fatigue characteristics, is a precise meter with a quantifiable value of repeatability. Also, it will, with calibration on an accredited facility, be an accurate meter with a small and quantifiable value of measurement uncertainty. In the context of flowmeters, the word *repeatability* is preferred to *reproducibility*. The meanings are elaborated on later, and I regret the limited meaning now given to *precision*, which I have used more generally in the past and shall slip back into in this book from time to time! In the following chapters, I have attempted to be consistent in the use of these words. However, many claims for accuracy may not have been backed by an accredited facility, but I have tended to use the phrase “measurement uncertainty” for the claims made.

Hayward (1977) used the story of William Tell to illustrate precision. William Tell had to use his cross-bow to fire an arrow into an apple on his little son’s head. This was a punishment for failing to pay symbolic homage to an oppressive Austrian ruler. Tell succeeded because he was an archer of great skill and high accuracy.

An archer’s ability to shoot arrows into a target provides a useful illustration of some of the words related to precision. So Figure 1.1(a) shows a target with all the shots in the bull’s-eye. Let us take the bull’s-eye to represent $\pm 1\%$, within the first ring $\pm 3\%$, and within the second ring $\pm 5\%$. Ten shots out of ten are on target, but how many will the archer fire before one goes outside the bull’s-eye? If the archer, on average, achieves 19 out of 20 shots within the bull’s-eye [Figure 1.1(b)], we say that the archer has an uncertainty of $\pm 1\%$ (the bull’s-eye) with a 95% confidence level (19 out of 20 on the bull’s-eye: $19 \div 20 = 0.95 = 95 \div 100 = 95\%$).

Suppose that another archer clusters all the arrows, but not in the bull’s-eye, Figure 1.1(c). This second archer is very consistent (all the shots are within the same size circle as the bull’s-eye), but this archer needs to adjust his aim to correct the

offset. We could say that the second archer has achieved high repeatability of $\pm 1\%$, but with a bias of 4%. We might even find that 19 out of 20 shots fell within the top left circle so that we could say that this archer achieved a repeatability within that circle of $\pm 1\%$ with a 95% confidence. Suppose this archer had fired one shot a day, and they had all fallen onto a small area [Figure 1.1(c)], despite slight changes in wind, sunshine, and archer's mood, then we term this good day-to-day repeatability. But how well can we depend on the archer's bias? Is there an uncertainty related to it?

Finally, a third archer shoots 20 shots and achieves the distribution in Figure 1.1(d). One has missed entirely, but 19 out of 20 have hit the target somewhere. The archer has poor accuracy, and the uncertainty in this archer's shots is about five times greater than for the first, even though the confidence level at which this archer performs is still about 95%.

If the third archer has some skill, then the bunching of the arrows will be greater in the bull's-eye than in the next circle out, and the distribution by ring will be as shown in Figure 1.1(e).

We shall find that the distribution of readings of a flowmeter results in a curve approximating a Normal distribution with a shape similar to that for the shots. Figure 1.1(f) shows such a distribution where 95% of the results lie within the shaded area and the width of that area can be calculated to give the uncertainty, $\pm 1\%$ say, of the readings with a 95% confidence level. In other words, 19 of every 20 readings fall within the shaded area.

With this simplistic explanation, we turn to the words that relate to precision.

Accuracy

It is generally accepted that *accuracy* refers to the truthfulness of the instrument. An instrument of high accuracy more nearly gives a true reading than an instrument of low accuracy. Accuracy, then, is the quality of the instrument. It is common to refer to a measurement as accurate or not, and we understand what is meant. However, the current position is that accuracy should be used as a qualitative term and that no numerical value should be attached to it. It is, therefore, incorrect to refer to

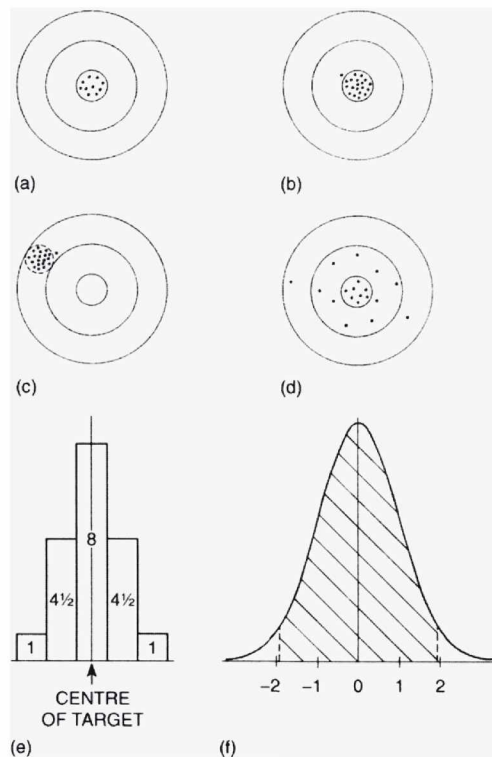


Figure 1.1. Precision related to the case of an archery target. **(a)** Good shooting – 10 out of 10 arrows have hit the bull's-eye. An accurate archer? **(b)** Good shooting? – 19 out of 20 arrows have hit the bull's-eye. An accurate archer and a low value of uncertainty ($\pm 1\%$) with a 95% confidence level. **(c)** Shots all fall in a small region but not the bull's-eye. Good repeatability ($\pm 1\%$) but a persistent bias of 4%. **(d)** Shots, all but one, fall on the target – 19 out of 20 have hit the target. A $\pm 5\%$ uncertainty with 95% confidence level. **(e)** Distribution of shots in (d) on a linear plot, assuming that we can collapse the shots in a ring semicircle onto the axis. **(f)** The Normal distribution, which is a good approximation for the distribution of flowmeter readings.

a measurement's accuracy of, say, 1%, when, presumably, this is the instrument's measurement uncertainty, as is explained later.

Repeatability

In a process plant, or other control loop, we may not need to know the accuracy of a flowmeter as we would if we were buying and selling liquid or gas, but we may require repeatability within bounds defined by the process. *Repeatability* is the value below which the difference between any two test results, taken under constant conditions with the same observer and with a short elapsed time, are expected to lie with 95% confidence.

Precision

Precision is the qualitative expression for repeatability. It should not take a value and should not be used as a synonym for accuracy.

Uncertainty

Properly used, *uncertainty* refers to the quality of the measurement, and we can correctly refer to an instrument reading having an uncertainty of $\pm 1\%$. By this we mean that the readings will lie within an envelope $\pm 1\%$ of the true value. Each reading will, of course, have an individual error that we cannot know in practice, but we are interested in the relationship of the readings to the true value. Because *uncertainty* is referred to the true value, by implication it must be obtained using a national standard document or facility. However, because it is a statistical quantity, we need also to define how frequently the reading does, in fact, lie within the envelope; hence the confidence level.

Confidence level

The *confidence level*, which is a statement of probability, gives this frequency, and it is not satisfactory to state an uncertainty without it. Usually, for flow measurement, this is 95%. We shall assume this level in this book. A confidence level of 95% means that we should expect on average that 19 times out of 20 ($19/20 = 95/100 = 95\%$) the reading of the meter will fall within the bracket specified (e.g., $\pm 1\%$ of actual calibrated value).

Linearity

Linearity may be used for instruments that give a reading approximately proportional to the true flow rate over their specified range. It is a special case of *conformity* to a curve. Note that both terms really imply the opposite. *Linearity* refers to the closeness within which the meter achieves a truly linear or proportional response. It is usually defined by stating the maximum deviation (or nonconformity, e.g., $\pm 1\%$ of flow rate) within which the response lies over a stated range. With modern signal processing, linearity is probably less important than conformity to a general curve. *Linearity* is most commonly used with such meters as the turbine meter.

Range and Rangeability

An instrument should have a specified range over which its performance can be trusted. Therefore, there will be upper and lower range values. This reflects the fact that probably no instrument can be used to measure a variable when there are no limitations on the variable. Without such a statement, the values for uncertainty, linearity, etc., are inadequate. The ratio of upper range value and lower range value may be called the *rangeability*, but it has also been known as the *turndown ratio*. The difference between upper and lower or negative range values is known as *span*. It is important to note whether the values of uncertainty, linearity, etc., are a percentage of the actual flow rate or of the full-scale flow [sometimes referred to as full-scale deflection (FSD), full-scale reading (FSR), maximum scale value, or upper range value (URV)].

1.4 A BRIEF REVIEW OF THE EVALUATION OF STANDARD UNCERTAINTY

Kinghorn (1982) points out the problem with terminology in matters concerning statistics and flow measurement. To the engineer and the statistician, words such as *error* and *tolerance* may have different meanings. The word *tolerance* was used for what is now known as uncertainty.

In providing an introduction to the terminology of uncertainty in measurement, I shall aim to follow the guidance in BIPM et al. (1993), which is usually known as the *Guide*, and also in a document consistent with the *Guide*, which provides the basis for uncertainty estimates in laboratories accredited in the United Kingdom (NAMAS 1997). The reader is strongly advised to consult this document, which is full of clear explanations and useful examples. Those wishing to pursue background arguments are referred to van der Grinten's (1994, 1997) papers and other papers in the proceedings of the 1997 North Sea Flow Measurement Workshop.

Random error, the random part of the experimental error, causes scatter, as the name suggests, and reflects the quality of the instrument design and construction. It is the part that cannot be calibrated out, and the smaller it is, the more precise the instrument is. It may be calculated by taking a series of repeat readings resulting in the value of the standard deviation of a limited sample n , and sometimes called the experimental standard deviation:

$$s(q_j) = \left\{ \frac{1}{n-1} \sum_{j=1}^n (q_j - \bar{q})^2 \right\}^{1/2} \quad (1.1)$$

where $\bar{q}$ is the mean of n measurements q_j . The experimental standard deviation of the mean of this group of readings is given by

$$s(\bar{q}) = \frac{s(q_j)}{\sqrt{n}} \quad (1.2)$$

Where too few readings have been taken to obtain a reliable value of $s(q_j)$, an earlier calculation of $s(q_j)$ from previous data may be substituted in Equation (1.2). In obtaining the overall uncertainty of a flowmeter or a calibration facility, there will be values of group mean experimental standard deviation for various quantities, and

so NAMAS (1997) defines a standard uncertainty for the i th quantity as

$$u(x_i) = s(\bar{q}) \quad (1.3)$$

where x_i is one of the input quantities. For those with access to NAMAS (1997), this is, essentially, dealt with there as a Type A evaluation of standard uncertainty.

Systematic error, according to flowmeter usage, is that which is unchanging within the period of a short test with constant conditions. This is, essentially, dealt with in NAMAS (1997) under the heading Type B evaluation of standard uncertainty. It is also called *bias*. However, in modern flowmeters and in calibration facilities, it is likely that this bias or systematic error will result in a meter adjustment, or a rig correction. The resulting uncertainty in that adjustment or correction will contribute to the overall uncertainty. The systematic uncertainty, therefore, may derive from various factors such as

- a. uncertainty in the reference and any drift,
- b. the equipment used to measure or calibrate,
- c. the equipment being calibrated in terms of resolution and stability,
- d. the operational procedure, and
- e. environmental factors.

From these we deduce further values of $u(x_i)$.

There has been debate about the correct way to combine the random and systematic uncertainties. We can combine random and systematic uncertainties conservatively by arithmetic addition. This results in a conservative estimate. NAMAS (1997) has followed the *Guide* in taking the square root of the sum of the squares of the standard uncertainties in consistent units. Thus the combined standard uncertainty is

$$u_c(y) = \sqrt{\sum [c_i u(x_i)]^2} \quad (1.4)$$

where y is the output quantity. To ensure consistent units, a sensitivity coefficient c_i will be required for each input x_i , although in practice this may be unity in most cases (as in Figure 4.3).

The final step (and we have glossed over many important details in NAMAS 1997) is to deduce from u_c the bracket within which the reading of, say, the meter lies.

In the past, bearing in mind that u_c or its components have been derived from standard deviations, we have used Student's t value which for a number of readings n is given by

n	t
10	2.26
20	2.09
> 30	2.0

for a 95% confidence level. The *Guide* replaces this, in general, with a coverage factor k , to obtain the expanded uncertainty

$$U = k u_c(y) \quad (1.5)$$

The recommended value is $k = 2$, which gives a confidence level of 95.45% taken as 95%, assuming a Normal distribution. If this assumption is not adequate, then we need to revert to Student's t .

The net result is that the assumption of a factor of 2 has now been given a systematic basis. The reader who is interested in more details about the basis of Normal and t distributions is referred to Appendix 1.A.

1.5 SENSITIVITY COEFFICIENTS

Suppose that output quantity, a flow rate, has the relationship

$$y = x_1^p x_2^q x_3^r x_4^s \quad (1.6)$$

then if x_2 , x_3 , and x_4 are held constant, we can differentiate y with respect to x_1 and obtain the partial derivative. This is the slope of the curve of y against x_1 when the other variables are kept constant. It also allows us to find the effect of a small change in x_1 on y . This slope (or partial derivative) is the sensitivity coefficient c_1 for x_1 and may be found by calculation. It will have the value $c_1 = p x_1^{(p-1)} x_2^q x_3^r x_4^s$, where the values of x_1 , x_2 , x_3 , and x_4 will be at the calibration point and may be dimensional. In some cases, it may be a known coefficient (e.g., a temperature coefficient of expansion). For cases where it is difficult to calculate, it may be possible to find the coefficient by changing x_1 by a small amount and observing the change in y . In some cases, the sensitivity coefficient may provide a conversion between different sets of units (e.g., where output quantity or velocity may be obtained from a dimension, a pressure, a movement, or a voltage).

1.6 WHAT IS A FLOWMETER?

We take as a working definition of an ideal flowmeter:

A group of linked components that will deliver a signal uniquely related to the flow rate or quantity of fluid flowing in a conduit, despite the influence of installation and operating environment.

The object of installing a flowmeter is to obtain a measure of the flow rate, usually in the form of an electrical signal, which is unambiguous and with a specified expanded uncertainty. This signal should be negligibly affected by the inlet and outlet pipework and the operating environment. Thus the uncertainty of measurement of a flowmeter should be reported as $y \pm U$, where U , the uncertainty band, might have a value of, say, 0.5%, and it should be made clear whether this is related to rate, full-scale deflection (FSD) or other value that might be a combination of these [e.g., in the form $\pm a$ (rate) $\pm b$ (FSD)]. The range should be given (e.g., 1 m³/h to 20 m³/h).

The statement of performance should include the coverage factor $k = 2$ and the level of confidence of approximately 95%, and, if appropriate, the authority that accredited the calibration facility (national or international).

In addition, the ranges of properties for which it can be used should be specified, such as fluid, flow range (beyond calibration), maximum working pressure, temperature range of fluid, and ambient temperature range.

It is useful to introduce two factors that define the response of flowmeters, although they are most commonly used for linear flowmeters with pulse output. The K factor is the number of pulses per unit quantity. In this book, we shall take it as number of pulses per unit volume when dealing with turbine and vortex meters:

$$K = \frac{\text{Pulses}}{\text{True volume}}$$

whereas the meter factor is usually defined as

$$\text{Meter factor} = \frac{\text{True volume}}{\text{Indicated volume}}$$

The reader should keep a wary eye for other definitions of meter factor such as the reciprocal of the K factor.

Let us take a specific example of a fictitious, but reasonably realistic flowmeter. In Figure 1.2(a), a typical flowmeter envelope is shown. It defines an approximately linear flowmeter with a 10:1 turndown and an uncertainty of $\pm 1\%$ of rate with a confidence level of 95% against a traceable standard calibration. This is a reasonable performance for a flowmeter and probably satisfies most requirements in industry. This, let us assume, is the performance specification the manufacturer carries in its sales literature. Actually the characteristic of the flowmeter may be the curve shown in Figure 1.2(a). If the company works to a high standard of manufacture, then the company may know that this characteristic lies within close tolerances in all cases. It may, therefore, only be necessary for the manufacturer to calibrate each flowmeter at, say, 90% of FSD, or 50% and 90% of FSD, in order to make the claim that the characteristic falls within the envelope specified in the sales literature.

If the meters are actually of this standard, it may well be feasible to calibrate them in much greater detail so that a 5-, 10-, or even 20-point calibration may provide a characteristic that ensures that the reading is known to, say, $\pm 0.2\%$. The values obtained from the calibration will then be programmed into a flow computer, which will interpret each reading of the flowmeter against this look-up table. Since we are comparing the flowmeter's signal to a linear one, if it were without error it would also be linear. Consequently, companies sometimes record linearity within their literature. In this case, it would also be $\pm 1\%$ with a 10:1 turndown.

The envelope that has been discussed gives the uncertainty at each flow rate in terms of the actual flow rate. Because of the physical basis of some flowmeters, this method is not appropriate, and the uncertainty may then be given in terms of the full scale. Figure 1.2(b) shows such an envelope where the performance of the flowmeter would be defined as $\pm 1\%$ FSD. It is apparent that the uncertainty in the flowmeter's reading at, say, half scale is $\pm 2\%$ of rate and at 20% of reading will be as much as $\pm 5\%$ of rate. The problem often arises wherein the user has a particular flow range that does not match that of the actual instrument. The user's full flow may only be at 60% of the instrument's range, and so for the user the instrument has an uncertainty, at best, of 1.7% of rate.

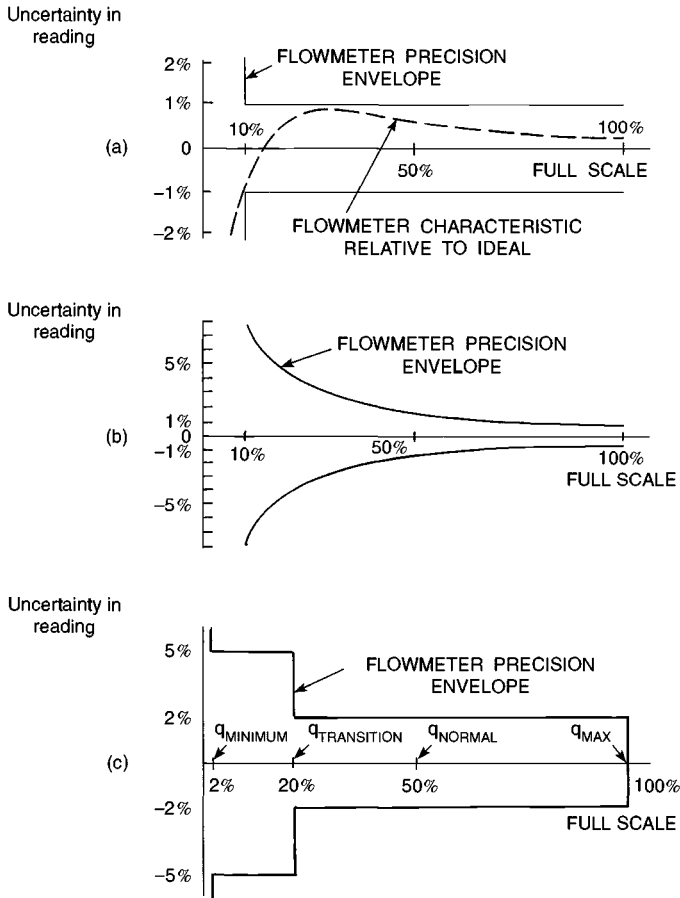


Figure 1.2. Required envelope for a flowmeter. **(a)** Envelope as a percentage of rate. **(b)** Envelope as a percentage of FSD. **(c)** Stepped envelope with increased uncertainty at low range values.

A third type of envelope is shown in Figure 1.2(c). This is particularly common in the specifications for water and gas meters. In the example shown, the meter has an uncertainty of $\pm 2\%$ of rate from full flow down to 20%. Below this value of flow rate, the uncertainty is $\pm 5\%$ of flow rate down to 2% of range. In practice, a meter might have more steps in its envelope.

In many cases, the specification of uncertainty by the manufacturer may be a combination of these. As indicated earlier, it is common to have an uncertainty that combines a value based on rate and another on full-scale deflection. In addition, there may be allowances to be added for zero drift, temperature change, and, possibly, even pressure change. In some flowmeters, viscosity is important but is probably accommodated by charts showing the variation in performance with viscosity.

One note of caution! Clever electronics can take any signal, however non-linear, and straighten the characteristic before the signal is output. Suppose such a procedure were used for the characteristic in Figure 1.2(a), below 10% of range.

The characteristic is probably very sensitive to minor variations in this region, and any attempt to use the characteristic could lead to disguised, but serious, errors.

Variation of temperature and pressure can have effects on the performance of a flowmeter, as can humidity, vibration, and other environmental parameters (Baker 1988/9).

Often the units used in a manufacturer's catalog are not those that you have calculated. For this reason, conversion factors that provide conversions to four significant figures of flow rate, velocity, temperature, pressure, length, etc., have been included in Table 1.1. If not otherwise specified the International System of units (SI) based on meter, kilogram, second is assumed.

Table 1.1. Conversion for some essential measurements from Imperial, U.S., and other units to metric, to four significant figures

Length	1 in ^a = 25.4 mm 1 ft = 0.3048 m
Volume	1 ft ³ = 0.0283 m ³ 1 ft ³ = 28.32 l (liter) 1 bbl (barrel) = 0.1590 m ³
Mass	1 lb = 0.4536 kg 1 long ton (2,240 lb) = 1,016 kg 1 short ton (2,000 lb) = 907.2 kg 1 metric tonne (2,205 lb) = 1,000 kg
Density	1 lb/ft ³ = 16.02 kg/m ³
Temperature	(Temperature in °F – 32)/1.8 = Temperature in °C
Pressure	1 psi = 6,895 N/m ²
Viscosity	Dynamic viscosity: SI (metric) unit is the Pascal second (Pas) to which the more common unit, the centipoise (cP), is related by 1 cP = 10 ⁻³ Pas. Kinematic viscosity: SI (metric) unit is m ² /s to which the more common unit, the centistoke (cSt), is related by 1 cSt = 10 ⁻⁶ m ² /s = 1 mm ² /s.
Velocity	1 ft/s = 0.3048 m/s
Volumetric flow rate	1 ft ³ /s (1 cusec) = 0.02832 m ³ /s 1 Imp gal/s = 0.004546 m ³ /s 1 Imp gal/s = 4.546 l/s 1 U.S. gal/s = 0.003785 m ³ /s 1 U.S. gal/s = 3.785 l/s 1 Imp gal/h = 0.004546 m ³ /h 1 Imp gal/h = 4.546 l/h 1 U.S. gal/h = 0.003785 m ³ /h 1 U.S. gal/h = 3.785 l/h
Mass flow rate	1 lb/h = 0.4536 kg/h 1 lb/s = 0.4536 kg/s

^a An approximate conversion is 4 in. to 100 mm.

1.7 CHAPTER CONCLUSIONS (FOR THOSE WHO PLAN TO SKIP THE MATHEMATICS!)

I have tried to bring together, within the scope of this book, essential information for all who may have dealings with flowmeters and flow measurement (although I have tried to avoid duplicating too much of the excellent and practical information given by Miller 1996). For this reason, the chapters not only address the technical aspects but also the selection, maintenance, calibration, and typical applications of the various meters. There is also a section on manufacture and production. I hope that the book will provoke the prospective entrepreneur, the small and medium-sized enterprises (SME), or the major instrument company to assess the market needs and the relevant development and production needs of their companies for new devices.

One plea regarding manufacturers' brochures is in order. It would be helpful to have a standard format for the core information to which the manufacturer could add special features and presentation to reflect customer demands (e.g., from different countries and industries). I often find myself faced with two catalogs from the same manufacturer. I search them to find how they differ. They clearly are printed differently, and yet it is not clear whether they refer to different meters. Do just the catalogs differ, or is there a difference between the flowmeters described in each catalog? This may seem an unlikely problem, but the preparation of this book confirms that confusing catalogs are still with us.

The corollary of the last plea (for a readable and complete brochure) is that the manufacturer needs to be told everything about the fluid and the installation in order to decide whether the meter can handle the requirements. We shall return to this point in much greater detail in Chapter 3.

I have endeavored to reflect the requirements of standard documents, but the reader is encouraged to obtain relevant standards. A partial list is included together with a bibliography at the end of this book. Standards organizations and professional bodies have produced a range of essential documents that cover many aspects of flow metering. Upp (1993) provides a useful digest of some of the documents such as API's Manual of Petroleum Measurement Standards, AGA documents which for many flow measurement applications (e.g., Section 14.3) are the same as API, and ASME and ISA. Another example from the United Kingdom is IGE (1988), which recommends standards of good practice and provides an extremely thorough statement on gas meter installation, which, although not carrying the wider backing of a standard, is clearly very authoritative and should provide a companion volume. Such topics as commissioning and maintenance are covered, and the manual encourages procedures for routine maintenance and record keeping as do I (cf. Olsen 1974, Kulin 1975).

As the reader moves into the following chapters, two sets of information may be useful. Table 1.1 lists the conversion factors for Imperial, U.S., and metric units, and Table 1.2 relates volumetric and mass flow rate to linear velocity in various sizes of tube (Baker 1988/9). It is common in flow measurement to require the velocity of flow, and Table 1.2 provides an order of magnitude.

Finally, the whole matter of accuracy and the limits of accuracy, when related to all the parameters that influence a flowmeter's operation, remains an area with unanswered questions.

Table 1.2. Velocity in pipes for various flow rates to two significant figures

						Mean Velocity (m/s) in a Circular Pipe of Diameter							
	m ³ /h ^a	l/min	gal/min	gal/min	ft ³ /min	10 mm	25 mm	50 mm	100 mm	200 mm	500 mm	1000 mm	2000 mm
Very low	10 ⁻³	0.017	3.7 × 10 ⁻³	4.4 × 10 ⁻³	5.9 × 10 ⁻⁴	3.5 × 10 ⁻³	5.7 × 10 ⁻⁴	1.4 × 10 ⁻⁴	3.5 × 10 ⁻⁵				
	10 ⁻²	0.17	3.7 × 10 ⁻²	4.4 × 10 ⁻²	5.9 × 10 ⁻³	3.5 × 10 ⁻²	5.7 × 10 ⁻³	1.4 × 10 ⁻³	3.5 × 10 ⁻⁴	8.8 × 10 ⁻⁵	1.4 × 10 ⁻⁵		
	0.1	1.7	0.37	0.44	5.9 × 10 ⁻²	0.35	5.7 × 10 ⁻²	1.4 × 10 ⁻²	3.5 × 10 ⁻³	8.8 × 10 ⁻⁴	1.4 × 10 ⁻⁴	3.5 × 10 ⁻⁵	
	1	17	3.7	4.4	0.59	3.5	0.57	0.14	3.5 × 10 ⁻²	8.8 × 10 ⁻³	1.4 × 10 ⁻³	3.5 × 10 ⁻⁴	8.8 × 10 ⁻⁵
	10	170	37	44	5.9	35	5.7	1.4	0.35	8.8 × 10 ⁻²	1.4 × 10 ⁻²	3.5 × 10 ⁻³	8.8 × 10 ⁻⁴
	100	1700	370	440	59	350	57	14	3.5	0.88	0.14	3.5 × 10 ⁻²	8.8 × 10 ⁻³
	1000	1.7 × 10 ⁴	3700	4400	590	3500	570	140	35	8.8	1.4	0.35	8.8 × 10 ⁻²
	10 ⁴	1.7 × 10 ⁵	3.7 × 10 ⁴	4.4 × 10 ⁴	5900	3.5 × 10 ⁴			350	88	14	3.5	0.88
Very high	10 ⁵	1.7 × 10 ⁶	3.7 × 10 ⁵	4.4 × 10 ⁵	5.9 × 10 ⁴	3.5 × 10 ⁵				880	140	35	8.8
	10 ⁶	1.7 × 10 ⁷	3.7 × 10 ⁶	4.4 × 10 ⁶	5.9 × 10 ⁵	3.5 × 10 ⁶						350	88

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^a Since water has a density of 1,000 kg/m³ (approximately), the mass flow rate in kilograms per hour of water may be obtained by multiplying this column by 1,000.

1.8 MATHEMATICAL POSTSCRIPT

I have left this note to the end so that those who are not concerned with advanced mathematical concepts can ignore it.

I have included essential mathematics in the main text of the book. In certain flowmeters, the mathematical theory is more complex (e.g., the turbine meter), and the theory has, accordingly, been consigned to an appendix after the relevant chapter.

One important and interesting mathematical approach, which starts to develop a unified theory of flow measurement, was first suggested by Shercliff (1962) and significantly extended by Bevir (1970). Both applied it to electromagnetic flowmeters where it has been highly successful. Hemp (1975) has also applied this theory to electromagnetic flowmeters, but he has developed the theory for other types of flowmeter: ultrasonic (1982), thermal mass (1994a), and Coriolis (1994b, and Hemp and Hendry 1995). In Chapter 12, on electromagnetic flowmeters, an appendix describes the essential mathematics. The weight function developed in this theory provides a measure of the importance of flow in each part of the meter with respect to the overall meter signal. The flow at each point of a cross-section is weighted with this function. Ideally the weighting should result in a true summation of the flow in the meter to obtain a volume flow rate. It has been possible to approach this ideal for the electromagnetic flowmeter.

For the other types of meters, the reader will be given only a brief explanation and will be referred to relevant papers. This is partly for want of space in this book and partly because the theory, although interesting, is still being developed and may not yet have reached a sufficiently significant stage for the designer.

A second mathematical physics theory, first (to my knowledge) applied by Hemp (1988) to flow measurement, is reciprocity. This, essentially, states that if you apply a voltage to one end of an electrical network and measure the current at the other end, you find that by reversing the ends and hence the direction you obtain the same relationship. Hemp has proposed this as a means of eliminating some errors in flowmeters to which the theory is applicable.

APPENDIX 1.A

Statistics of Flow Measurement

1.A.1 INTRODUCTION

The main needs of the engineer are to

- ❑ understand and be able to give a value to the uncertainty of a particular measurement;
- ❑ know how to design a test to provide data of a known uncertainty;
- ❑ be able to combine measurements, each with its own uncertainty, into an overall value; and

- determine the absolute accuracy of an instrument at the end of a traceable ladder of measurement.

The international and national documents set the recommended approach for flow metering. Most standard statistics books will provide the essentials (Rice 1988; cf. Campion et al. 1973, which is often quoted but may not be easy to obtain), but good school texts may be more accessible (Crawshaw and Chambers 1984, Eccles et al. 1993a, 1993b). Hayward (1977) is an extremely well written and elegant little book, which deserves to be updated and reprinted; Kinghorn (1982) provides a well-written and useful brief review of the main points; and Mattingly (1982) addresses some of the problems concerned with transfer standards.

1.A.2 THE NORMAL DISTRIBUTION

The Normal distribution, Figure 1.A.1(a), is also known as the Gaussian distribution after Carl Friedrich Gauss, who proposed it as a model for measurement errors (Rice 1988). The notation used for the Normal curve is $N(\mu, \sigma^2)$ which is the distribution under the curve

$$f(x) = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{1}{2}[(x-\mu)/\sigma]^2} \quad (1.A.1)$$

where μ is the mean value of the data, and σ^2 is the variance. Alternatively, σ is the standard deviation for the whole population. We can simplify the curve (normalize it) by putting $z = (x - \mu)/\sigma$ and obtaining [Figure 1.A.1(b)]

$$\phi(z) = \frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}z^2} \quad (1.A.2)$$

With the form of Equation (1.A.2), the curve does not vary with the size of the parameters μ and σ .

What the curve tells us (in relation to instrument measurements) is that the statistical chance of an instrument reading giving a value near to the mean μ is high, but the farther away the reading is from the mean, the less the chance is of its occurring (indicated by the curve decreasing in height the further one moves from the mean), and as values of the reading get farther still from μ , so the chance gets less and less.

The area under the curve of Equation (1.A.2) [Figure 1.A.1(b)], which reaches to infinity each way, is unity, and this is the

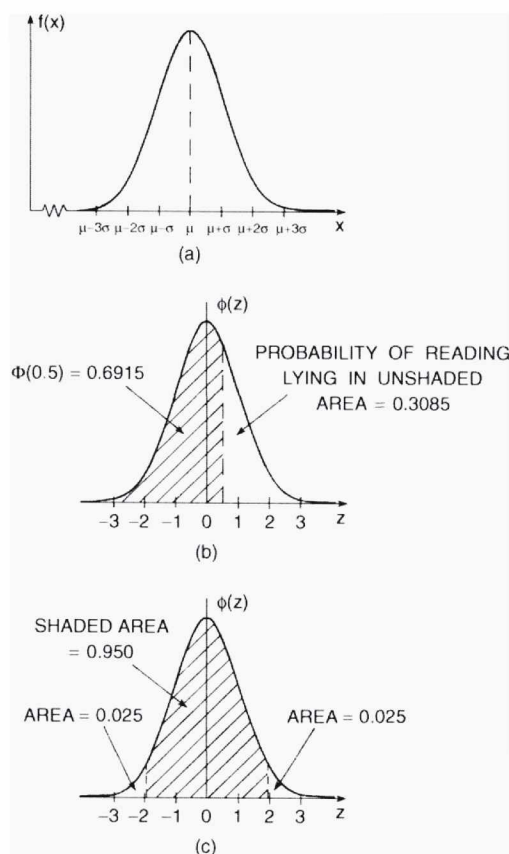


Figure 1.A.1. The Normal distribution.

probability of the reading lying within this curve (obviously). The area under the curve between $z = -\infty$ and some other value of z is given by

$$\Phi(z) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^z e^{-\frac{1}{2}t^2} dt \quad (1.A.3)$$

and is the probability that a reading will lie within that range and is obtained numerically and given in Table 1.A.1 in normalized form. For instance, if $z = 0.5$, $\Phi(z) = 0.6915$, where $\Phi(z)$ is the area under the curve from $z = -\infty$ to $z = 0.5$ in this case. So the chance of a reading lying beyond this point is 0.3085, or about 30%.

We shall be interested in the chance that a flowmeter reading will fall between certain limits each side of $z = 0$, the mean value. A chance of 95% is often used and is called a 95% confidence level. This means that 19 times out of 20 the reading will fall between the limits. This requires that the central area of the curve [Figure 1.A.1(c)] has a value of 0.95, or 0.475 each side of the mean. To obtain z from this value, we need to add $0.475 + 0.5 = 0.975$, and this gives a value (Table 1.A.1) of $z = 1.96$. If we put this in terms of x , we obtain

$$x - \mu = 1.96\sigma \quad (1.A.4)$$

or the band around the mean value of the reading within which 95% of the readings statistically should fall, is approximately $\pm 2\sigma$, or two standard deviations from the mean.

If we are interested, not in the spread of individual readings, but in the spread of the mean of small sets of readings, a statistical theorem called the Central Limit Theorem provides the answer. If a sample of n readings has a mean value of M , then the distribution of means like M is given by $N(\mu, \sigma^2/n)$. This is intuitively reasonable because one would expect that the scatter of means of groups of n readings would have a smaller variance, σ^2/n , than the readings themselves, as well as a smaller standard deviation, $\sigma/\sqrt{n}$. In this discussion, we have skated over the need to know the value of the standard deviation of the whole normal population. If we do not know σ , then we can approximate it with the value of the standard deviation s of the small set of n . So if $n \geq 30$, it is usually sufficiently precise to take $\sigma = s$. If $n < 30$, the standard deviation should be taken as $\sigma = s\sqrt{n}/\sqrt{n-1}$.

Table 1.A.1. A selection of values from the normal distribution function $\Phi(z)$

z	$\Phi(z)$	Symmetrical Central Area Under Curve
0	0.5000	
0.5	0.6915	
1.28	0.8997	
1.282	0.9000	0.80
1.29	0.9015	
1.64	0.9495	
1.645	0.9500	0.90
1.65	0.9505	
1.96	0.9750	0.95
2.57	0.99492	
2.576	0.99500	0.99
2.58	0.99506	
3.29	0.99950	0.999
3.30	0.99952	

After D. V. Lindley and W. F. Scott, *New Cambridge Statistical Tables*, 2nd ed., Cambridge: Cambridge Univ. Press. Table 4, pp. 34, 35.

1.A.3 THE STUDENT *t* DISTRIBUTION

We now need to look at one more subtlety of these estimates. The Normal distribution assumed that we had obtained many readings and could with confidence know

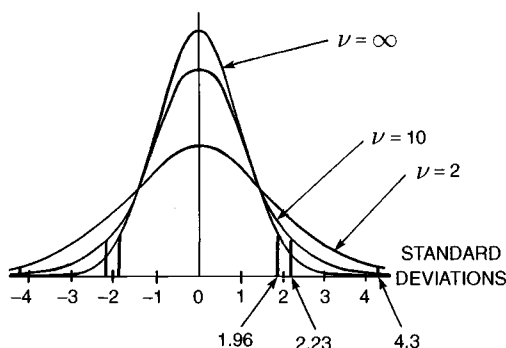


Figure 1.A.2. Student's t distribution curves compared with the Normal curve. Note $p = 5\%$ as related to Table 1.A.2 for both tails.

that they formed a Normal distribution. We can agree that if the error is random, then it is a fair assumption that many readings would form a Normal distribution. However, often we have only a few readings, and these may not be uniformly distributed within the curve of Figure 1.A.1. Too many may lie outside the 1.96σ limit. For this reason, we use the Student t distribution, which allows for small samples on the assumption that the distribution, as a whole, is Normal. Figure 1.A.2 shows the effect of the small number of readings. Since, with a small number of readings, one has to be subtracted from all the others to obtain a

mean, the number of independent values is one less than the number of readings, and so the statisticians say that there is one less degree of freedom than the number of readings. In Figure 1.A.2, ν is the symbol for the degree of freedom, and $\nu = n - 1$, where n is the number of readings. For $\nu \rightarrow \infty$, the t distribution tends to a Normal curve with mean zero and variance unity.

Figure 1.A.2 shows clearly the larger area spreading beyond the Normal curve in which the readings may lie and the reason for the greater uncertainty. The curves are used in a similar way to the Normal curve, but, as an alternative, Table 1.A.2 provides the information we need. If we have 10 readings, say, and so select the value of $\nu = 9$ for the degree of freedom, and if we wish to find the limits for a confidence level of 95%, we shall need to use the 5% column. We obtain $t = 2.262$, which we can apply to obtain the limits for a 95% confidence of $\pm 2.262\sigma/\sqrt{n}$ on the mean values of groups of readings, where n is the number of readings in the group. We should

Table 1.A.2. A selection of values from the student t function

n	ν	$p(\%)$ (Total)				
		20	10	5	1	0.1
		$p/2$ (%) (per tail)				
		10	5	2.5	0.5	0.05
2	1	3.078	6.314	12.71	63.66	636.6
3	2	1.886	2.920	4.303	9.925	31.60
5	4	1.533	2.132	2.776	4.604	8.610
10	9	1.383	1.833	2.262	3.250	4.781
20	19	1.328	1.729	2.093	2.861	3.883
30	29	1.311	1.699	2.045	2.756	3.659
61	60	1.296	1.671	2.000	2.660	3.460
121	120	1.289	1.658	1.980	2.617	3.373
∞	∞	1.282	1.645	1.960	2.576	3.291

After D. V. Lindley and W. F. Scott, *New Cambridge Statistical Tables*, 2nd ed., Cambridge: Cambridge Univ. Press, Table 10, p. 45.

note, however, that the 95% confidence level from Table 1.A.2 gives a t value that varies little from 2.0 if $\nu \geq 20$. The limits for 95% confidence will then be $\pm 2.0 \sigma / \sqrt{n}$ on the mean values of groups of readings.

I have always been puzzled by the name Student, but Eccles et al. (1993b) explain that the originator of this technique was William S. Gosset, born 1876, who used the pseudonym *Student*.

1.A.4 PRACTICAL APPLICATION OF CONFIDENCE LEVEL

The method described in Section 1.4 leads to the following steps (cf. Hayward 1977):

- i. Write down systematic uncertainties and derive the standard uncertainty for each component.
- ii. Write down random uncertainties and derive the standard uncertainty for each component.
- iii. Calculate the combined standard uncertainty for uncorrelated input quantities (and refer to NAMAS 1997 if correlated).
- iv. Obtain the expanded uncertainty using $k = 2$ for 95% confidence.

Taking a simple example, where we need to revert to t , suppose that we obtain a series of volumetric flow readings from a 50 mm ID flowmeter with the flow set at $10 \text{ m}^3/\text{h}$:

10.06, 10.01, 9.95, 9.99, 9.85, 10.02, 10.03, 10.12, 9.90, 9.98.

The results are plotted in Figure 1.A.3. The mean of these readings is 9.991, and the standard deviation is 0.07752. We could thus conclude that the true reading of this meter fell in the bracket $9.991 \pm 2.262 \times 0.07752 / \sqrt{9} = 9.991 \pm 0.05845$, or between 9.93 and 10.05 with a 95% confidence. This fairly brackets the value of $10 \text{ m}^3/\text{h}$.

The actual readings should have all fallen within 9.991 ± 0.07752 , or 9.91 and 10.07. In fact three fell outside this bracket – rather higher than the 1 in 19 implied by the 95% confidence level. We might wish to look more closely at the procedure

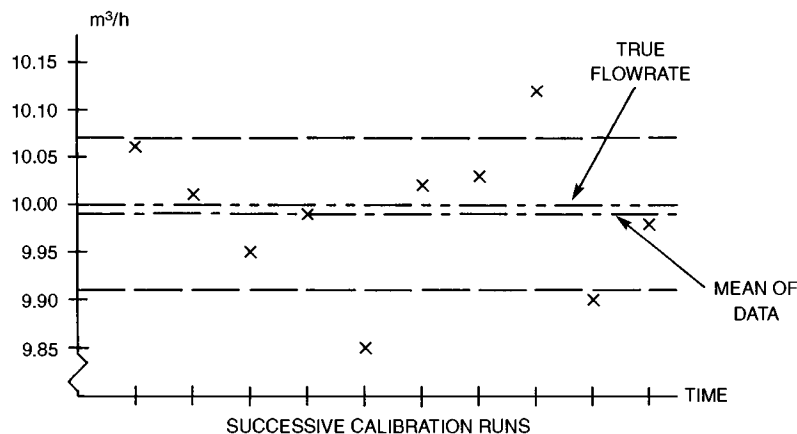


Figure 1.A.3. A set of flowmeter test readings (after Kinghorn 1982) for a fixed flow rate.

for obtaining these results since this suggests a possible problem with the means for obtaining the data.

1.A.5 TYPES OF ERROR

There are essentially four types of error (Kinghorn 1982).

- ❑ *Spurious errors* result from obvious failures, obvious in the sense that they can be identified and documented. Readings with these should be eliminated.
- ❑ *Random errors* cause a variation in the output reading even when the input parameter has not changed.
- ❑ *Constant systematic error*, which is also called bias, may vary over the range but is constant in time, and could, in principle, be corrected out of the reading.
- ❑ *Variable systematic error* (bias) slowly varies with time, usually in a consistent direction, and may be caused by wear in bearings of a rotating meter, fatigue in components of a vibrating meter, erosion of geometry, etc.

Figure 1.A.4 illustrates these errors. Clearly one of the readings is so far out that there must be some explanation other than randomness. It is comforting to know that some of the most eminent experimentalists of the past have had cause to discard readings in critical experiments!

The scatter around the mean line will provide the basis of the calculation which we did in Section 1.A.4. The constant systematic error (bias) can be seen and could be built into a flow computer. The change in the mean value with time shows the changing systematic error, which is, in part, the reason for regular recalibration of meters.

The repeatability is related to the closeness of readings. If we expect a reading to lie within a band given by $\pm 2s$, the worst case difference between successive readings that fall within this band would be $4s$ [$(2 \times 2)s$], but a less extreme working value is obtained from the root of the sum of the squares (or the quadrature; cf. Pythagoras and the length of the hypotenuse):

$$\sqrt{(2s)^2 + (2s)^2} \quad \text{or} \quad 2\sqrt{2}s$$

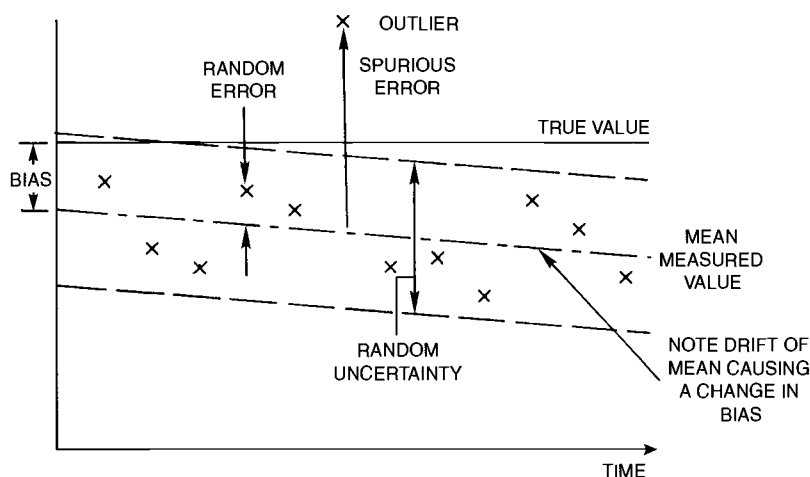


Figure 1.A.4. Diagram to show the various types of error (after Kinghorn 1982).

1.A.6 COMBINATION OF UNCERTAINTIES

If we combine uncertainties due to the nature of a flowmeter's operating equation, then we take the following approach. Suppose that the flowmeter has the equation

$$q_v = x_1 x_2^n x_3 x_4^m \quad (1.A.5)$$

To obtain the uncertainty in q_v , we need the partial derivative of q_v with respect to x_1 , x_2 , etc. The required result can be achieved, either by differentiating the equation as it stands or by first taking logarithms of both sides. We shall skip this and go straight to the result:

$$\frac{u_c(q_v)}{q_v} = \pm \frac{u(x_1)}{x_1} \pm \frac{nu(x_2)}{x_2} \pm \frac{u(x_3)}{x_3} \pm \frac{mu(x_4)}{x_4} \quad (1.A.6)$$

The problem with this equation is that the arithmetic sum of the uncertainties is usually overpessimistic. It is, therefore, recommended that they be combined in quadrature, or by the root-sum-square (rss) method. This leads to the following equation:

$$\frac{u_c(q_v)}{q_v} = \sqrt{\left(\frac{u(x_1)}{x_1}\right)^2 + \left(\frac{nu(x_2)}{x_2}\right)^2 + \left(\frac{u(x_3)}{x_3}\right)^2 + \left(\frac{mu(x_4)}{x_4}\right)^2} \quad (1.A.7)$$

There are complications beyond this equation. x_2 may appear in the equation as $x_5 + x_6$. In this case, $x_5 + x_6$ will need to be dealt with first and will require careful consideration as to whether the actual errors in these quantities are combining, canceling, or random.

1.A.7 UNCERTAINTY RANGE BARS, TRANSFER STANDARDS, AND YODEN ANALYSIS

It is sometimes useful to indicate the range of uncertainty that has been estimated from the experimental method in each reading. This can be done by using bars that give uncertainty limits on each experimental point. This will then indicate, for a particular flow rate, the likely uncertainty in the reading. This is shown in Figure 1.A.5(a). In some cases where flow rate varies, there may be an uncertainty in both primary flow rate measurement and reading of second meters. In this case, there will need to be uncertainty bars in both

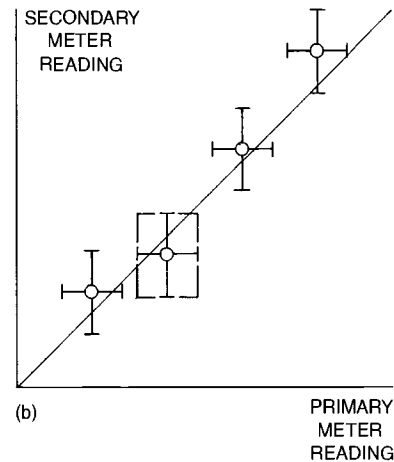
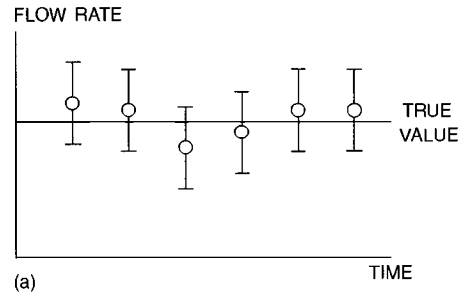
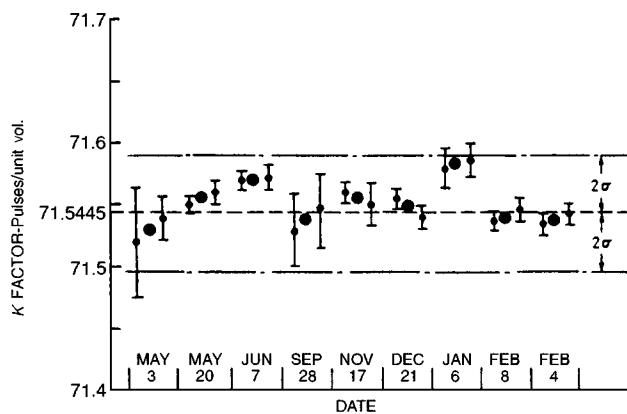
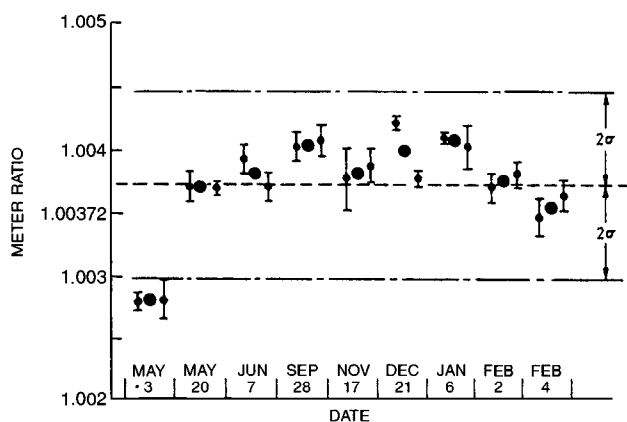


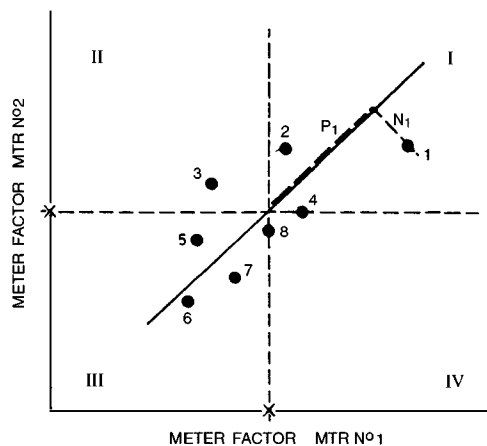
Figure 1.A.5. (a) Uncertainty limit bars on readings. **(b)** Uncertainty limit bars on both flow rate and readings.



(a)



(b)



(c)

Figure 1.A.6. Turbine meter as a transfer standard (from Mattingly 1982; reproduced with the author's permission). **(a)** Typical turbine meter control chart for meter factor. **(b)** Typical turbine meter control chart for ratio. **(c)** Graphical representation of the Youden plot.

directions, and the rectangle [Figure 1.A.5(b)] will define the limits of the possible uncertainty. In other words, the maximum uncertainty will have been obtained by quadrature.

Mattingly (1982) describes the procedure for checking the validity of different flow measurement laboratories using a Measurement Assurance Program (MAP) [or Proficiency Testing as referred to by Mattingly in a recent draft report on the approach of the National Institute of Standards and Technology (NIST)] where a good-quality flowmeter acting as a transfer standard is exchanged between laboratories. Figure 1.A.6(a) shows the results of such a cycle of checks, and the bars on the experimental points presumably indicate the uncertainty of the turbine meter at a particular laboratory. Various people have suggested the use of two flowmeters usually in series, but as a possible alternative in parallel, to enhance the accuracy of a transfer standard. In this case, the ratio of the signals from the two meters will give an indication as to whether there has been a shift in either, and, if there has not been, the reading of the meters will provide, with greater confidence, the accuracy of the facility. In Figure 1.A.6(b), Mattingly (1982) gave a typical control chart for the ratio of a pair of National Bureau of Standards turbine meters. If the ratio falls outside an agreed tolerance band, the cause of the error will need to be found before confidence in the meter readings is restored.

For laboratory comparison, the transfer package with two meters should be run at one or two agreed flow rates. The position of the two flowmeters may be interchanged to obtain a second set of data. Using the Youden analysis in Figure 1.A.6(c), each laboratory is represented by a single point (1, 2, 3, etc.) resulting from plotting the meter factor of the two meters on the two axes. It is apparent that the position of the points relates to the type of error. Essentially, if the points lie in quadrants I or III, then the meters are reading the same, and the error can be attributed to the flow rig. If the readings are in quadrants II or IV, the flowmeters are not agreeing, and the error may be due to a malfunction in one of them. The reader is referred to Mattingly's (1982) article for further details of this procedure (cf. Youden 1959).

CHAPTER 2

Fluid Mechanics Essentials

2.1 INTRODUCTION

In a recent book (Baker 1996), I provided an introduction to fluid mechanics and thermodynamics, particularly aimed at instrumentation. I do not, therefore, propose to repeat what is written there but rather to confine myself to essentials. In addition, Noltingk (1988) has provided a very valuable handbook on general instrumentation. In this book, I shall use the term *fluid* to mean liquid or gas and will refer to either liquid or gas only when the more general term does not apply.

2.2 ESSENTIAL PROPERTY VALUES

Flowmeters generally operate in a range of fluid temperature from -200°C (-330°F) to 500°C (930°F), with line pressures up to flange rating for certain designs. Typical values of density and viscosity are given in Table 2.1.

It should also be noted that liquid viscosity decreases with temperature, whereas gas viscosity increases with temperature at moderate pressures. In common fluids, such as air and water, the value of viscosity is not dependent on the shear taking place in the flow. These fluids are referred to as Newtonian in their behavior as compared with others where the viscosity is a function of the shear taking place. The behavior of such fluids, known as non-Newtonian, is very different from normal fluids like water and air. Newtonian fluid behavior is a good representation for the behavior of the bulk of fluids.

2.3 FLOW IN A CIRCULAR CROSS-SECTION PIPE

An essential dimensionless parameter that defines the flow pattern at a particular value of the parameter is the Reynolds number

$$\text{Re} = \frac{\rho V D}{\mu} \quad (2.1)$$

where ρ is the density of the fluid, μ is the dynamic viscosity, and, when applied to flow measurement, V is the velocity in the pipe, and D is the pipe diameter. Typical values of Reynolds number are: for water with $\mu/\rho = 10^{-6} \text{ m}^2/\text{s}$, $V = 1 \text{ m/s}$ (3.3 ft/s), $D = 0.1 \text{ m}$ (4 in.) $\text{Re} = 10^5$; and for air at ambient conditions with $\mu/\rho = 1.43 \times 10^{-5} \text{ m}^2/\text{s}$, $V = 10 \text{ m/s}$ (33 ft/s), $D = 0.1 \text{ m}$ (4 in.) $\text{Re} = 0.7 \times 10^5$.

Table 2.1. Typical values of viscosity (approximate values at 1 bar)

	Temperature (°C)	Density, ρ (kg/m ³)	Viscosity	
			Dynamic, μ (cP)	Kinematic, ν (cSt)
Water	20	1,000	1.002	1.002
Benzene	20	700	0.647	0.92
Oxygen	0	1.43	0.019	13.3
Nitrogen	0	1.25	0.017	13.6
CO ₂	0	1.98	0.014	7.1
Air	0	1.29	0.017	13.2

Adapted from Kaye and Laby (1966).

If Re is less than about 2,000, and the fluid has had a sufficient length of pipe to reach a steady state, it all moves parallel to the axis of the pipe. At the pipe wall, the fluid “sticks” to the pipe in, what is known as, a nonslip condition. The velocity of the fluid increases, therefore, from zero at the pipe wall to a maximum at the center, and the shape of the profile is parabolic. In this case, the flow is called laminar.

At about $Re = 2,000$, a major change takes place, and the smooth parallel nature of the flow gives way to eddies in the flow. These eddies mix the high velocity at the pipe axis with the lower velocity near the pipe wall. The resulting profile is flatter (although it still goes to zero at the pipe wall). When fully developed, this is known as a turbulent profile. It is of a well-defined shape and with a known range of eddy sizes. It may, therefore, be misleading to use the term *turbulent profile* for any other profiles created by upstream disturbances due to bends, etc. Between the turbulent flow and the laminar flow, there is a transition when the flow alternates randomly, in space and time, between laminar flow and turbulent flow.

Any flowmeter application to non-Newtonian fluids will need advice from the flowmeter manufacturer. It is probable that clear bore flowmeters, if applicable, will be most suitable. A flowmeter with relatively low sensitivity to profile, such as electromagnetic or multibeam ultrasonic designs, will be worth consideration provided the fluid is conducting for the first type or transmits an adequate level of sound for the second.

It is apparent from the calculations of Reynolds number for water and air that, in the majority of industrial applications, the flows will be turbulent, and, therefore, the behavior of the turbulent profile, and of flowmeters subject to it, is of primary importance to us.

For laminar flow, we can use the equation

$$V = V_0 \left[1 - \left(\frac{r}{R} \right)^2 \right] \quad (2.2)$$

where V_0 is the velocity on the axis of the pipe, r is the radial point at which we are measuring velocity, and R is the pipe radius and where V is zero at the wall of the pipe because of the nonslip condition for a fluid at a solid boundary.

Table 2.2. Approximate turbulent velocity profiles from equation (2.4)

Re	4×10^3	2.3×10^4	1.1×10^5	1.1×10^6	2.0×10^6 to 3.2×10^6
n	6.0	6.6	7.0	8.8	10
$V/\bar{V}$ (at $r = 0.75R$)	1.003	1.004	1.004	1.005	1.005
$V/\bar{V}$ (at $r = 0.758R$)	0.998	0.999	1.000	1.002	1.002

The mean velocity in the pipe is then given by

$$\bar{V} = \frac{V_0}{2} \quad (2.3)$$

For turbulent flow an approximate curve-fit is

$$V = V_0(1 - r/R)^{1/n} \quad (2.4)$$

where V_0 is, again, the center line velocity, and we can relate n to Re from experimental data (Table 2.2). Note that pipe roughness, if the pipe is not hydraulically smooth, will also affect the profile.

This expression, therefore, provides not only a convenient, although approximate, representation of the boundary layer shape but also an index adjustment to allow for the change in Reynolds number. The resulting profile shapes for the laminar and the turbulent regimes, based on Equation (2.4), are shown in Figure 2.1.

One interesting feature of these turbulent profiles is that the ratio of the velocity at about the 3/4 point to the mean velocity is approximately unity. The mean velocity is given by

$$\frac{\bar{V}}{V_0} = \frac{2n^2}{(n+1)(2n+1)} \quad (2.5)$$

Hence

$$V/\bar{V} = \frac{(n+1)(2n+1)}{2n^2} (1 - r/R)^{1/n} \quad (2.6)$$

Table 2.2 provides the values of this ratio for various values of Re and two radial positions. Thus if a single measurement of velocity in a pipe is to be used to obtain the mean of the velocity in a turbulent flow, a point at $0.758R$ may be found best. This figure will clearly be of importance when we consider where to position the probes described in Chapter 18.

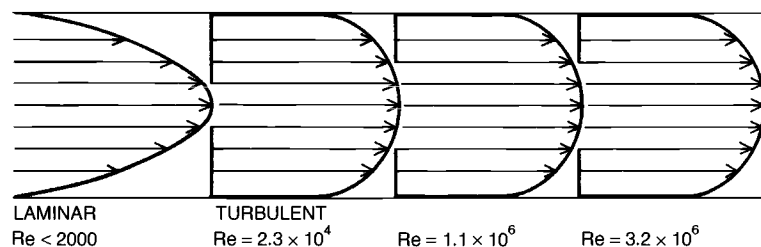


Figure 2.1. Laminar and turbulent pipe profiles.

A final important point about the profiles in Figure 2.1 is that there is much difference between the flows, even though they are drawn as smooth curves. In the laminar profile, the fluid shears smoothly over itself, but it all moves essentially parallel to the axis of the pipe. In the turbulent profiles, the curves represent a mean profile and ignore the turbulent eddies, which can be as much as 10% of the velocity of the mean flow and cause fluctuations in all directions. In the context of flow measurement, this is particularly significant in that the meter must measure accurately against this background fluctuation.

So far we have assumed that the pipe flow profile is fully developed. To achieve such a profile requires an upstream straight length of pipe greater than may be available in many industrial flowmeter installations. 60D is sometimes quoted as a minimum requirement.

We, therefore, have a problem when installing a flowmeter in that the calibration conditions for a fully developed profile are unlikely to be achieved in the installation and that the meter accuracy will be degraded. We have four options.

- i. Find an installation point where fully developed flow occurs.
- ii. Calibrate the flowmeter with the upstream pipework (downstream has little effect beyond a maximum of about 8D).
- iii. Take extensive measurements of the effects of bends, valves, T-pieces, multiple bends, expansions, and contractions on a particular flowmeter design and allow for the change in precision.
- iv. Attempt to reorder the flow to recreate a turbulent profile of the sort used for calibration.

Outside a few laboratory situations, (i) is probably unlikely to be an option, but (ii) may not be realistic or financially viable unless in situ calibration is possible. Option (iii) will be a recurring theme through the discussion of most types of flowmeters. An alternative approach is to compute the profiles resulting from pipe fittings. One example of such an approach is that of Langsholt and Thomassen (1991) who used a commercially available computer program to obtain the flow downstream of various piping fittings. Some indication that computer prediction and experiment gave similar swirl decay downstream of a double bend in perpendicular planes was given, and 41D reduced the angle to 3.3° for measured values and 2.4° for simulated values. Mottram and Rawat (1986) suggested that the swirl from two 90° bends in perpendicular planes is reduced by 50% in smooth pipes after $L/D = 45$ and in rough pipes after an L/D value in the range 10 to 20 (cf. however Mattingly et al. 1987 who noted swirl effects on a turbine meter after 90D).

Langsholt and Thomassen (1991) also provided some flow contours at points in a complex pipe configuration. In the context of this chapter, it is important to give some attention to (iv).

2.4 FLOW STRAIGHTENERS AND CONDITIONERS

Flow straighteners and flow conditioners have been used, for many years, to attempt to reorder a profile that has been disturbed by upstream fittings. The flow straightener was designed to remove swirl from the flow. Swirl, the bodily rotation of the fluid in

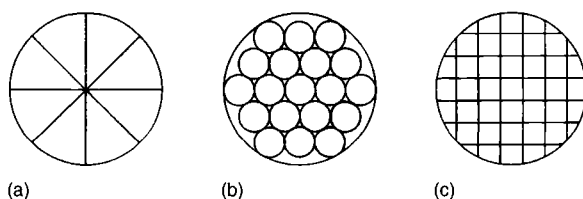


Figure 2.2. Approximate diagrams of flow straighteners: (a) Etoile; (b) Tube bundle; (c) Box (a honeycomb layout may also be used).

a pipe, takes a long length to decay and can introduce severe metering errors. The three types of straightener commonly used are shown in Figure 2.2.

It is essential that they are set straight or they will become the cause of swirling flow rather than the correction for it. Examples of the effectiveness of etoile straighteners of various lengths were given by Kinghorn et al. (1991). Their conclusions follow:

- An eight vaned (on four diameters) straightener of $0.5D$ length virtually removed even the maximum swirl.
- The flat profiles downstream of the straighteners appear to have caused most of the residual negative errors in the orifice coefficient, as opposed to the positive errors due to swirling flow.
- To ensure an error of less than $\pm 0.5\%$, a straightener $1D$ long should be placed upstream of the orifice meter at least
 - $6D$ for plates with a beta ratio of 0.5 and
 - $14D$ for plates with a beta ratio of 0.8 .
- Even with $16D$ separation, virtually all tests suggested some effect from the straightener.
- A $1D$ straightener is recommended for removing swirl and minimizing head loss.

Flow conditioners attempt to reorder the flow profile to recreate a fully developed turbulent profile. However, this reordering must address both mean profile shape and turbulent eddy pattern. Examples of conditioners are given in Figure 2.3.

Recent published work (e.g., that due to Laws) has thrown much helpful light on the effectiveness of these devices. Bates (1991) reported on field use by Total Oil Marine of a K-lab flow conditioner. Strong swirl was found to be present in the gas metering system on a process platform in the Alwyn North field, which had been installed to ISO 5167 in three 14-in. lines with beta ratio of 0.6 . The meter was separated from a pipe reduction of 0.86 by $30D$, but upstream of this there were other fittings. Greasy deposits on the orifice plate gave an early indication of swirl being present. This was rectified by installing K-lab flow conditioners. This conditioner is machined from solid, with holes arranged approximately as in Figure 2.3(b). (See Erdal et al. 1994 on the development of the K-Lab conditioner and Spearman et al. 1991 on LDV measurements downstream of a Mitsubishi flow conditioner and of an orifice plate in a combined package.)

Karnik et al. (1991, 1994) addressed the important question of how closely the flow downstream of a flow conditioner represented fully developed flow as far as a flowmeter was concerned. For undisturbed fully developed flow, they found that,

for Re of about 0.9×10^5 , the profile was approximated by $n = 7.4$ in Equation (2.4), compared with $n = 7$ in Table 2.2. Measurements of velocity and turbulence profiles and of orifice plate performance were made to determine the effect of an elbow upstream of the tube bundle with various spacings between the elbow, the flow straightener, and the metering position. They confirmed the point that, even though the profile may be approximately correct, the turbulence characteristics are unlikely to be correct, and the orifice plate appears to be influenced by both.

Laws and colleagues have done some useful work on straighteners and conditioners (Laws 1991, Laws and Ouazzane 1992, 1995b, 1995c). Her work initially, and rather unexpectedly, suggested that tube bundles of length recommended by ISO 5167 (hexagonal pack of 19 tubes of $L \geq 20d$), AGA(3), and ANSI/API 2530:1985 (circumferential pack with $L > 10d$) are little more, and possibly less, effective than one of only $L = 1.25d$. She found that this was as effective in swirl elimination and attenuation of profile nonuniformities. She also suggested that the AGA design was more effective and had a lower Δp . This led her to suggest that perforated plates may be a better and more easily constructed option.

This observation was reinforced by work on the Zanker flow conditioner with thicker perforated plate, which performed, in some cases, better without downstream honeycomb (Laws and Ouazzane 1992). A criterion of $\pm 6\%$ of the fully developed profile $100D$ downstream was used to assess whether the resultant Zanker profiles at $8.5D$ were satisfactory. The reported experiments suggested that, for a 12-mm plate in a 100-mm line, the Zanker flow conditioner without downstream honeycomb performed better than with the honeycomb. Laws and Ouazzane (1995b) also suggested that a thick Zanker plate (10% of the pipe diameter) with upstream honeycomb flow straightener is preferable to the Zanker straightener with thinner plate and downstream straightener. Not surprisingly, they suggested that the Laws plate is superior. In further tests, Laws and Ouazzane (1995c) found that the etoile flow straightener was not a very good flow

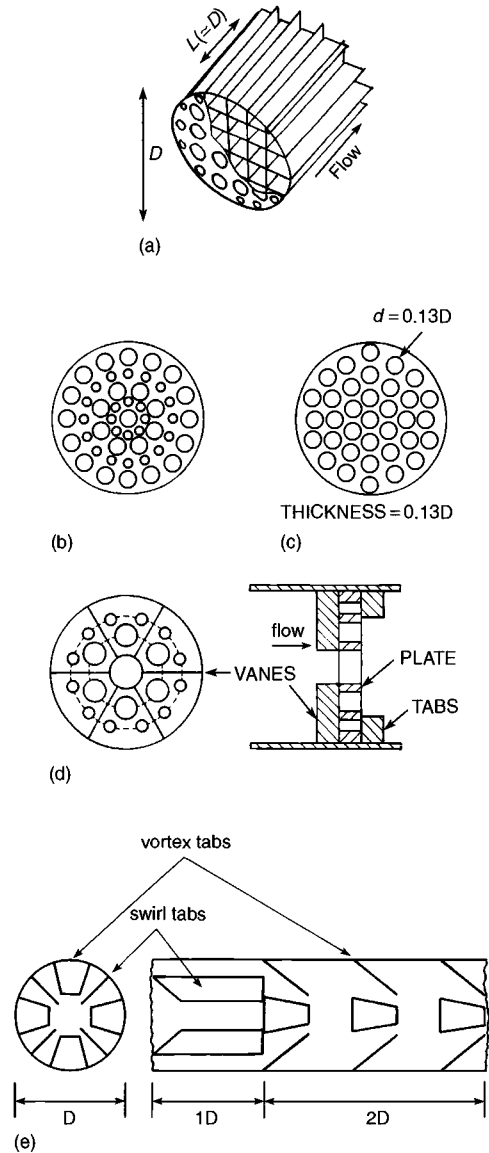


Figure 2.3. Approximate diagrams of flow conditioners: (a) Zanker; (b) K-Lab Mark 2; (c) Mitsubishi; (d) Laws (after Laws 1990, Laws and Ouazzane 1995a); (e) Vortab [reproduced with permission of Laaser (UK) Ltd.].

conditioner, although it was effective as a straightener. They suggested that if the etoile is made with the vanes stopping short of the pipe axis, the wake from the hub of the etoile is removed and the performance is improved.

Laws (1990) described her new flow conditioner, which benefited from these findings. It has a central hole and two rings of holes on $0.4616D$ and $0.8436D$. The plate thickness is given as $0.123D$ and, for best performance, has an arrangement of vanes and tabs as shown in Figure 2.3(d). The number of holes is 1:7:13 (or 1:6:12 in later versions), the ratio of open area outer to inner ring is 1.385, and the ratio of inner ring to central hole is 5.42. The total open area is 51.55%. Laws claims that the unit will be an enhancement to most flow installations, enabling flow conditions to be maintained close to fully developed irrespective of the operating conditions and thus minimizing errors due to installation effects. The loss coefficient is about 1.4, and the claim appears to be made that with a flow distortion device $3D$ from the conditioner and the conditioner $3D$ upstream of the upstream pressure tapping of an orifice plate, the shift in performance is of order 0.2% (Laws and Ouazzane 1995a).

Smith et al. (1989) proposed a different approach to the recreation of turbulent profiles using vortex-generating devices. Their results looked promising for the conditioner $5D$ upstream of the flowmeter when they appear to claim less than 1% shift. The device consists of four axial swirl tabs of about $1D$ length, which project about 40% of the radius into the flow from the wall and are essentially a straightening device to remove swirl. Two sets are recommended; they should be set at 45° relative to each other. These are followed by, preferably, two sections of $1D$ length each, with four trapezoidal tabs (vor-tabs) spaced around the circumference of the pipe and projecting inward with an angle of about 30° to the pipe wall in the general direction of the flow. Laws and Harris (1993) tested this device and considered that it did quite well in recreating a turbulent profile but appeared uncertain of the stability of the resulting profile. The loss is claimed to be only 0.7 dynamic heads. A commercially available device, the Vortab [Figure 2.3(e)], appears to exploit the same idea and consists of a $1D$ straightening section followed by a $2D$ section with three sets of vor-tabs.

Gallagher et al. (1994) discussed the development of the Gallagher conditioner, which also consists of an antiscirl device, settling chamber, and profile device mounted sequentially in the pipe.

2.5 ESSENTIAL EQUATIONS

The *continuity equation* results from the classical physics concept that mass can neither be created nor destroyed and so the same mass must pass through each point in a pipe

$$\text{Mass flow} = q_m = \rho q_v = \rho VA \quad (2.7)$$

and we can relate the velocity at different pipe sections 1 and 2 (Figure 2.4)

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \quad (2.8)$$

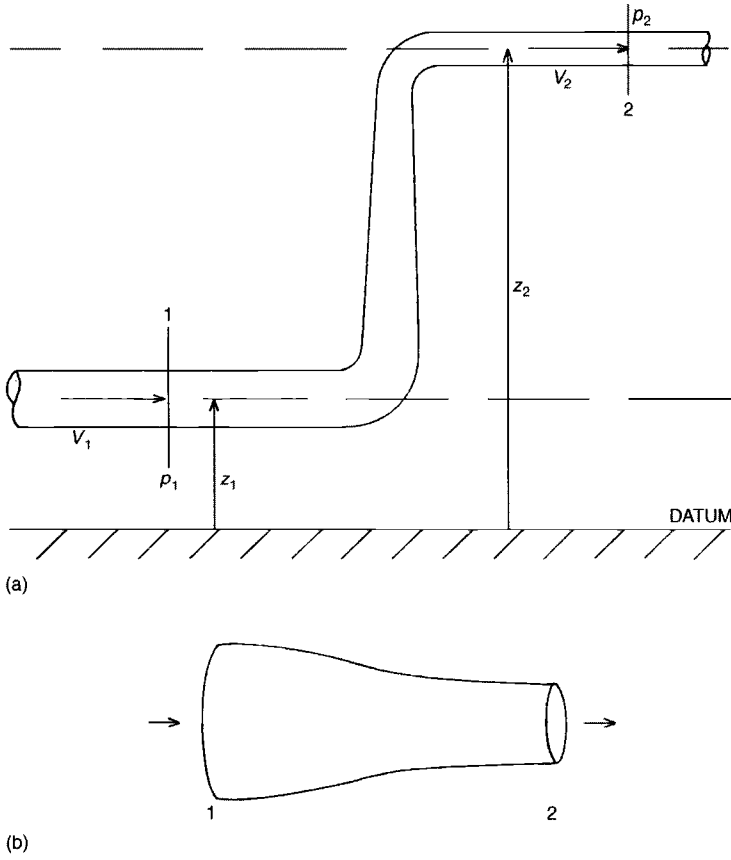


Figure 2.4. Duct sections. **(a)** Varying height and duct section. **(b)** Varying area duct to show positions referred to in equations.

Bernoulli's equation can be derived from energy considerations (Baker 1996) and relates the pressure change to the velocity change. Using Figure 2.4(a), we can relate pressure, velocity, and height above datum in a compressible fluid without flow losses as

$$\int_1^2 \frac{dp}{\rho} + \frac{(V_2^2 - V_1^2)}{2} + g(z_2 - z_1) = 0 \quad (2.9)$$

This equation is known as the compressible fluid form of Bernoulli's equation. If ρ is constant for an incompressible fluid with constant density, then we can rewrite the Bernoulli equation as

$$\frac{p_2 - p_1}{\rho} + \frac{(V_2^2 - V_1^2)}{2} + g(z_2 - z_1) = 0 \quad (2.10)$$

Note from this, assuming that we can neglect changes in z , that, if the flow is brought to rest, the resulting pressure, p_0 is the total or stagnation pressure, the pressure while fluid is flowing is the static pressure, and the difference, $\Delta p = \frac{1}{2}\rho V^2$, is the dynamic pressure.

If we neglect changes in height and combine Equations (2.8) and (2.10), we obtain an equation that gives the relationship between pressure change and flow rate through a duct such as in Figure 2.4(b)

$$p_2 - p_1 + \frac{q_m^2}{2\rho} \left(\frac{1}{A_2^2} - \frac{1}{A_1^2} \right) = 0 \quad (2.11)$$

Loss coefficients result, in real flows, from viscosity in the fluid, and it is essential to know these losses in a piping system to ensure that the pump, or whatever creates the flow, will be sufficient to maintain the required flow. To understand how to calculate losses, the reader is referred to the excellent book by Miller (1990).

Thus all flowmeters create a flow loss, although clear bore meters do not significantly add to the loss due to an equal length of straight pipe. However, if the profile is highly disturbed and flow conditioning is used, these devices cause an additional pressure loss which is given by

$$\Delta p_{\text{loss}} = K \left(\frac{1}{2} \rho V^2 \right) \quad (2.12)$$

K may be up to order 5 for the devices in Figures 2.2 and 2.3, but for some flow conditioners it may be as much as 15 since the flow is effectively going through a contraction with very poor downstream pressure recovery.

For a fuller derivation of these equations, the reader is referred to Baker (1996) or other fluid mechanics books.

2.6 UNSTEADY FLOW AND PULSATION

In this section, we consider the effect of unsteadiness on flowmeters. The use of fluid instability in flowmeters is covered in Chapter 11 where it is an essential part of the meter designs.

Most flowmeters are affected by unsteady or pulsatile flows for the following reasons.

- Pulsatile flow affects the velocity profile in the pipe, and the distorted profile may, in turn, affect the flowmeter response. Hakansson and Delsing (1994) found that flattening of the flow profile occurred due to the pulsating flows. This, in the case of the ultrasonic flowmeter, causes incorrect averaging of the diametral beam.
- The characteristic of the meter may be nonlinear, and the resulting output average will not correspond to that for the flow. The differential pressure flowmeters with their square law are examples.
- The inertia of parts of the flowmeter or of the fluid may not allow the meter to track the pulsating flow correctly. The gas turbine meter is highly affected by this. The importance of this effect in the orifice plate has been the cause of some discussion in the literature for which the reader is referred to Mottram's papers.
- The natural operating frequency of the flowmeter may be near the pulsation frequency, and this may cause aliasing or other errors. Vortex meters are sensitive to such pulsations.

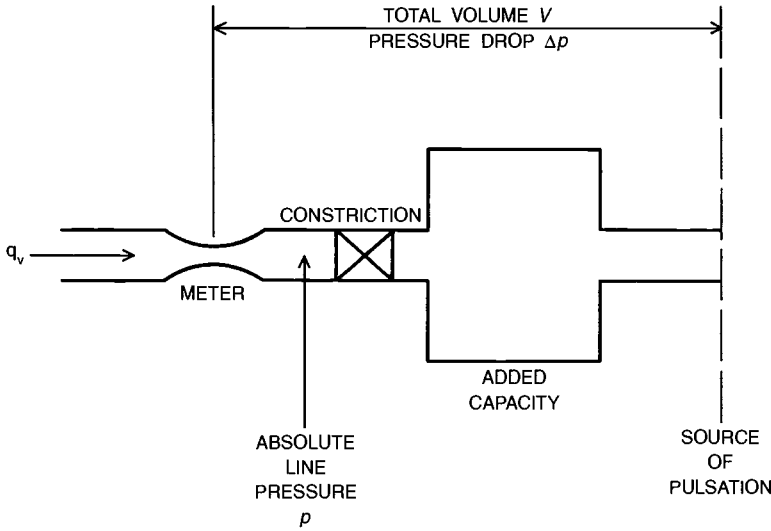


Figure 2.5. Diagram to indicate symbols used in the Hodgson number calculation.

- The secondary instrumentation may not be able to follow the pulsation, and, as a consequence, errors may be introduced. The manometer is the most obvious secondary device to be affected, but pressure transducers, connecting leads, and other devices may also be affected.

One approach associated with Hodgson was developed to show how much damping was needed to reduce pulsation to acceptable levels.

Mottram (1989) reviews some of the work on Hodgson's number (cf. Mottram 1981) but makes the wise comment that "if you can't measure (the size of the pulsation) damp it" (Mottram 1992). The Hodgson number is given by (Figure 2.5)

$$H = \frac{V f \Delta p}{q_v p} \quad (2.13)$$

where V is the total volume of pipework and other vessels between the source of the pulsation and the flowmeter position, Δp is the pressure drop over the same distance, f is the frequency of the pulsation, q_v is the volumetric flow rate, and p is the absolute line pressure. With this number, it is possible to plot curves to show the likely error levels for various values of H (cf. Section 5.5 for orifice plates).

Mottram gives a simplified value for the damping based on the Hodgson number, allowing for the worst case where resonance occurs, of

$$\frac{H}{\gamma} = \frac{1}{4\pi} \frac{(V_{rms}/\bar{V})_{ud}}{(V_{rms}/\bar{V})_d} \quad (2.14)$$

where $(V_{rms}/\bar{V})_{ud}$ is the undamped velocity ratio, $(V_{rms}/\bar{V})_d$ is the damped velocity ratio, and γ is the isentropic exponent.

He cautions that the criterion resulting from using this with the Hodgson number is untested. He also suggests that the Hodgson number should be doubled where the pulsation amplitude is estimated and not measured.

Some meters [e.g., differential pressure (cf. Section 5.5), turbine, or vortex] are seriously affected by pulsating flow, whereas others (e.g., electromagnetic or ultrasonic time-of-flight) are probably little affected.

2.7 COMPRESSIBLE FLOW

When a gas flows at velocities comparable to the speed of sound, its behavior is markedly different from the behavior of incompressible fluids. Such high velocity flows occur in sonic or critical nozzles.

To appreciate the special features of compressible flow of a gas, we shall start by imagining that we have the nozzle in Figure 2.6(a) set up in an experimental rig with a compressor sucking air downstream of the control valve. We consider a convergent-divergent nozzle that reduces in area from the inlet to a minimum at the throat, and then increases in area to the exit, where it opens into the back pressure chamber. As the flow increases, there would be a steadily increasing noise emitted

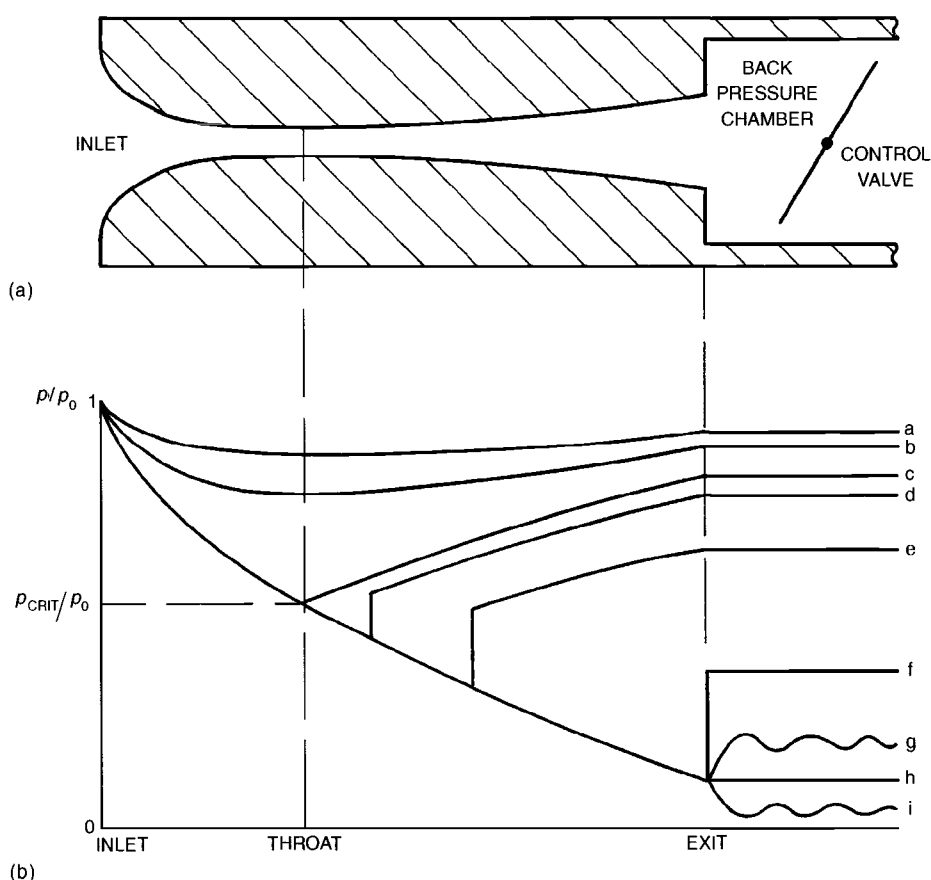


Figure 2.6. Convergent-divergent nozzle: (a) geometry; (b) p/p_0 against distance through convergent-divergent nozzle.

by the nozzle. A point is reached where the mass flow becomes constant, and the noise disappears irrespective of the downstream pressure.

What is happening? The normal information transfer mechanisms between the changes downstream and the upstream flow appear to have ceased. The loss of sound is clearly linked to this loss of information transfer, and it is intuitively reasonable to link the information transfer to the very small pressure waves that constitute sound and to suggest that, when the velocity is as fast or faster than sound speed, it is impossible for the waves to move upstream, and so impossible also to communicate upstream any flow change that happens downstream. In this case, sound speed has been reached at the throat of the nozzle.

It is beyond the scope of this book to derive the full equations for compressible flow, but we state three equations that will be useful in later discussions. These are for the special case of flow when conditions are adiabatic, isentropic, and reversible. Although an idealization, the flow for the inlet to the nozzle is a good approximation to such a flow.

$$\frac{T_0}{T} = 1 + \frac{\gamma - 1}{2} M^2 \quad (2.15)$$

$$\frac{p_0}{p} = \left(1 + \frac{\gamma - 1}{2} M^2 \right)^{\gamma/(\gamma-1)} \quad (2.16)$$

$$\frac{q_m \sqrt{c_p T_0}}{A p_0} = \frac{\gamma}{\sqrt{\gamma - 1}} M \left(1 + \frac{\gamma - 1}{2} M^2 \right)^{-\frac{1}{2}(\gamma+1)/(\gamma-1)} \quad (2.17)$$

They give the value of T , p , and q_m/A for a given γ and M , where

	Units
A = area	m^2
c = local sound speed	m/s
c_v = specific heat at constant volume	J/kgK
c_p = specific heat at constant pressure	J/kgK
M = Mach number V/c	dimensionless
p = pressure	Pa
p_0 = stagnation pressure	Pa
q_m = mass flow rate	kg/s
T = temperature	K
T_0 = stagnation temperature	K
V = gas velocity	m/s
γ = ratio of specific heats c_p/c_v (isentropic exponent)	dimensionless

Note that, in an isentropic flow without heat transfer, the stagnation temperature and pressure do not change and represent the values if the gas were to be brought to rest. However, stagnation pressure will change where irreversibility occurs (e.g., through a shock wave or in a flow with friction), and stagnation temperature will change where heat transfer takes place.

Figure 2.6(b) gives the plot for pressure variation through the convergent-divergent nozzle. As gas flows through the convergent-divergent nozzle in Figure 2.6(a), the velocity of the gas increases toward the narrowest point in the duct, the throat. The flow is created by a low pressure downstream of the nozzle. As the back pressure is reduced, so the pressures through the nozzle fall, and the velocity increases [a and b in Figure 2.6(a)]. In the sonic or critical condition, the pressure downstream is reduced until the velocity of the gas at the throat has increased to the speed of sound when the Mach number M is unity. Figure 2.6(b) shows that this condition can be achieved with the flow downstream of the throat subsonic (c), partly supersonic and partly subsonic with shock waves (d and e), or with the flow all supersonic (f, g, h, and i with various types of flow in the plenum). Had the nozzle stopped at the throat, it would still be possible to obtain sonic conditions there, but there would be no possibility of the pressure recovery that occurs in the diffuser in c, d, and e.

Equations (2.16) and (2.17) give the variation of pressure and area of the duct as we move toward the throat and M increases. At the throat where $M = 1$, we obtain two important relationships: the critical pressure ratio

$$\frac{p_*}{p_0} = \left(\frac{\gamma + 1}{2} \right)^{-\gamma/(\gamma-1)} \quad (2.18)$$

and the mass flow rate at choked or critical conditions

$$q_m = \frac{A_* p_0}{\sqrt{c_p T_0}} \frac{\gamma}{\sqrt{\gamma - 1}} \left(\frac{\gamma + 1}{2} \right)^{-\frac{1}{2}(\gamma+1)/(\gamma-1)} \quad (2.19)$$

where * as a subscript indicates throat conditions. For air, with $\gamma = 1.4$, the critical pressure ratio for sound speed at the throat is $p_*/p_0 = 0.528$. It can be shown that the pressure recovery in the diffuser allows the exit/inlet pressure ratio to be much higher than this while still achieving critical conditions (sonic at throat).

The importance of choked or critical conditions is twofold.

- i. The flow rate is unaffected by downstream variations because the sonic throat condition acts as a block to downstream changes (see Baker 1996 for a fuller description).
- ii. The mass flow of gas may be obtained from a knowledge of γ for the gas, the throat area A_* , and the upstream stagnation values p_0 and T_0 .

We shall consider the practical version of Equation (2.19) in Chapter 7.

2.8 MULTIPHASE FLOW

The term *multiphase flow* is somewhat misleading because it covers both multicomponent and multiphase. Thus dirty gas, air-in-water, cavitational, and steam flows may all be referred to as multiphase flows within this broad use of the term. Because of the dearth of data, the fluid engineer attempts to learn from data from different sources.

We may consider first the flow from an oil well, making the assumption that this is vertical. This is not strictly true, but it will give us some basic concepts. This was described by Baker and Hayes (1985). The crude oil will reach the well-head, having flowed up a pipe of about 100-mm bore for distances of several kilometers or about 30,000 pipe diameters. The flow will therefore, presumably, be fully developed. Initially, the flow will be single-phase, essentially oil only. As oil is removed from the well, the pressure in the reservoir will decrease, and the gas fraction in the well flow line will increase, and appear as gas bubbles. This is known as bubbly flow (Figure 2.7).

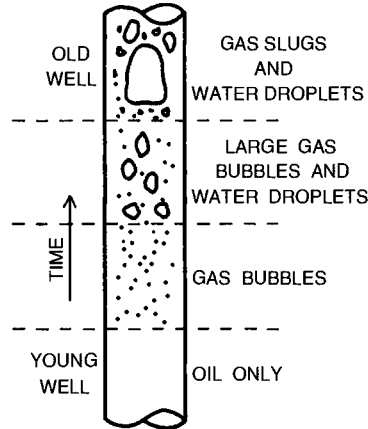


Figure 2.7. An example of three-phase vertical flow from an oil well.

With further aging, the gas bubbles will become larger. An equilibrium size distribution will result from breakup due to turbulence and from coalescence due to the breakdown of the liquid film on close approach of two bubbles. In addition, the water content is likely to increase. Water will be present as droplets and will form a third phase. Yet further aging will result in slugs of gas that travel up the center of the pipe leaving a slower layer of moving liquid on the wall (which, at times, may even reverse in direction during the passage of the slugs). These slugs tend to overtake each other forming larger slugs many meters long. These slugs may be up to 20 m long and may form an equilibrium size distribution giving a balance between coalescence due to bubbles overtaking each other and breakup due to instabilities when they become too large. As a further complication, the oil flow may contain waxy deposits and sand. Even though separation of components is a standard process, the development of subsea systems may require multicomponent handling.

We next consider horizontal two-phase flows (Figure 2.8). The most obvious effect is the loss of axisymmetry. Gravity now causes the less-dense phase to migrate to the top of the pipe. Thus, in a gas-liquid flow, the gas will move to the top of the pipe as bubbles. If these are allowed to become large, plugs of gas result, and as these coalesce, slugs of gas take up regions against the top of the pipe. Eventually, a sufficient number of these will lead to stratified flow.

Alternatively, the mixtures may be of two liquids (e.g., water in oil). The droplets of water will sink toward the bottom of the pipe, mirroring the behavior of air

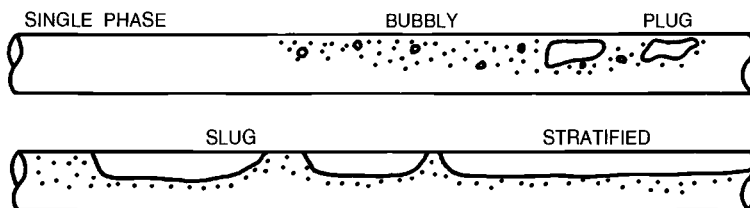


Figure 2.8. Horizontal two-phase flow.

bubbles, and will eventually drop out onto the bottom of the pipe causing a continuous layer of water.

With a sufficient length of straight pipe (100D or more), a fully developed flow may be achieved. However, in most applications we shall not be able to predict the resulting flow or how it will affect equipment in the line. This presents a considerable challenge to the development of instrumentation for handling oil well flows in sub-sea installations. It may be possible to mix the fluid to create a more homogeneous fluid for flow measurement, but this will cause severe turbulence and a changing profile, conditions generally considered unsuitable for such a measurement, not to mention the pressure loss which may result.

Liquids may contain gases in solution. For water, the maximum amount is about 2% by volume. The gas in solution does not increase the volume of the liquid by an amount equal to its volume since the gas molecules “fit” in the “gaps” in the liquid molecular structure. For hydrocarbons, the amount of gas that can be held in solution is very large, and the GOR (Gas-to-oil ratio), which is the volume of gas at standard conditions to the volume of liquid, can range up to 100 or more. In either case, but particularly the latter, changes in flow conditions (e.g., a pressure drop) can cause the gas to come out of solution causing a two-phase flow with, possibly, severe effects on instrument precision.

In low head flows (e.g., in a sewage works), the flowing stream may entrain air, and a dispersion of air in water may result.

2.9 CAVITATION, HUMIDITY, DROPLETS, AND PARTICLES

Cavitation may occur in certain liquid flows at pressures around ambient. Cavitation is the creation of vapor cavities within the liquid caused by localized boiling at low pressure. It can cause damage (e.g., in pumps, propellers, valves, and other flow components) because the cavities can collapse very quickly with large, although localized, impacts and can erode solid surfaces. It can also cause errors in flowmeter readings because it results in a larger volume than for the liquid alone.

High humidity may also create problems if it results in a consequent large amount of water vapor changing to liquid droplets in the gas (e.g., in critical flow venturi nozzles).

Particulate matter can, in addition, cause wear and may need to be removed with a fine filter.

Much can be learned by computing the trajectories of droplets and particles (cf. Ahmad et al. 1986, Hayes 1988), and such an approach may be usefully applied to flowmeters. Droplets and particles will affect their performance in various ways.

- a. They will cause flowmeters to read incorrectly, and most often this will be due to the flowmeter responding to the volume flow of the gas phase and not fully accounting for the greater mass carried in solid and liquid.
- b. They may cause erosion of the flowmeter body and sensing element.
- c. They may give false pulse readings in vortex flowmeters by hitting the pressure sensor.
- d. They may become trapped in vortex structures around both stationary and moving parts of the meter.

The basic flow around most meter internals is now predictable from standard computer programs, and a bubble/droplet/particle trajectory model should be combined with such a prediction. From such studies, it may be possible to design internals less susceptible to bubble/droplet/particle effects or to understand better the effects on, and to optimize the design of, meters such as the turbines described by Mark et al. (1990a, 1990b).

For electromagnetic and ultrasonic meters, further work on the interaction of the field and bubble/droplet/particle may lead to more sophisticated sensors. Further study should also address the effect of large bubbles on the performance of such meters and on ways of flow-pattern mapping in general.

2.10 GAS ENTRAPMENT

Gas entrapment appears to occur in some flow geometries with important effects. Thomas et al. (1983) observed that "Transient large eddies (vortices) in turbulent free shear flows entrain and transport large quantities of bubbles, and may also force the coalescence of bubbles." At the conference where this paper was presented, the appearance of this phenomenon was discussed in relation to two papers in particular, which suggested the presence of the phenomenon. In one of these, Baker and Deacon (1983) had tested a turbine meter that appeared to exhibit hysteresis in its response to increasing and decreasing fractions of air in water. This may have been due to the particular meter or the flow circuit, but it raised speculation as to whether vortex structures in the vicinity of the ball bearing fluid access path could be entrapping air and holding it in the bearing after the external air content in the flow had dropped. Presumably such entrapment could occur in the vortex downstream of an orifice plate [Figure 2.9(a)].

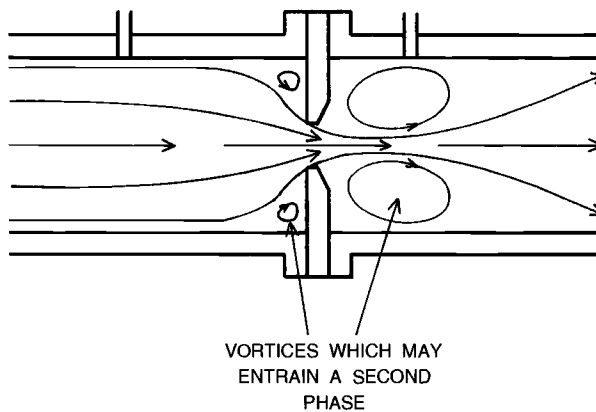
Hulin and Foussat (1983) observed the entrapment of a second phase in the shed vortices behind a meter bluff body. Figure 2.9(b) shows a diagram of this mechanism.

I have also observed, in some field data from an ultrasonic flowmeter, a behavior that could result from entrapment of air. A meter in a low head flow, where air was entrained with the water, periodically failed. The possibility that, in such a flow, the transducer cavities could cause small local vortices which could entrain the air and block the ultrasonic beam offered a possible explanation [Figure 2.9(c)].

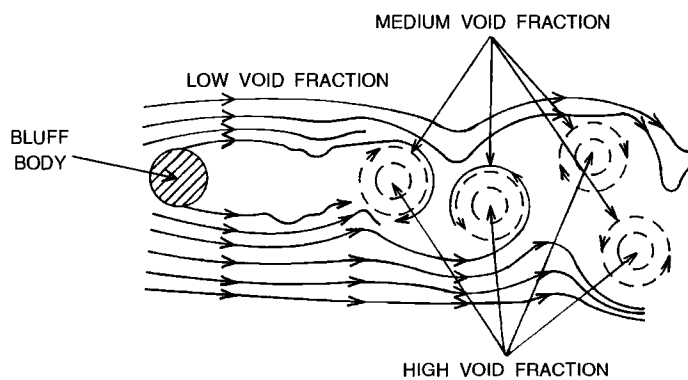
There is considerable scope for investigating in other applications the appearance of such entrapment and the likely effect. In flow measurement, entrapment may also occur around the throat of a Dall tube and behind an averaging pitot tube or other probes inserted in the flow. In all these cases, it is likely to affect the reading of the flowmeter. Entrapment may also occur in pumps, where it may reduce performance, and in pipework where abrupt area changes exist.

2.11 STEAM

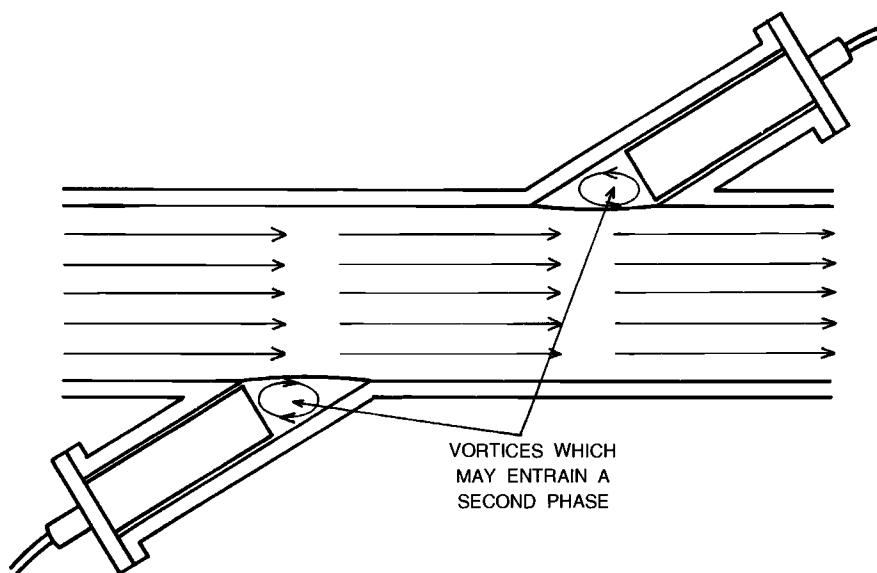
One truly two-phase fluid is steam. Superheated steam may be treated as a gas, and its properties are well tabulated. However, it is increasingly important to measure the flow of wet steam made up, say, of about 95% (by mass) vapor and about 5%



(a)



(b)



(c)

Figure 2.9. Possible mechanisms for gas entrapment: **(a)** Upstream and downstream of an orifice plate; **(b)** in vortices; **(c)** in an ultrasonic transducer cavity.

liquid. The droplets of the liquid are carried by the vapor but will not follow the vapor stream precisely. As with water droplets in oil, the liquid will drop through the vapor to land on the pipe wall, and we may obtain an annular flow regime until sufficient turbulence is created to reentrain this liquid. The measurement of such a flow causes major problems because the pressure and temperature remain constant while the dryness fraction changes. It is therefore not possible to deduce the dryness fraction or density from the pressure and temperature, and, in addition, the droplets may cause an error when we attempt to measure the flow.

Some pointers from the literature will be referred to in relation to some of the types of flowmeter. However, the best and safest advice is to use a water separator upstream of the flowmeter and to measure the resulting flow of dry saturated steam.

2.12 CHAPTER CONCLUSIONS

I have been highly selective in choosing topics for inclusion in this chapter. Once again the reader is referred to fuller texts on fluid mechanics. Baker (1996) was written, particularly, with the industrial flowmeter user in mind and enlarges on some of the points made in this chapter.

The development of computational fluid dynamics provides a tool for analyzing flowmeter behavior, and references will be mentioned in the other chapters. I consider that perturbation methods should be explored. These methods would allow a greater precision in defining performance change between the ideal flow through a meter and the disturbed flow due to turbulence change, profile distortion, or swirl.

The effects of small amounts of a second phase offer possibilities for investigating the behavior of particle/bubble/droplet in flows through meters by developing, for instance, methods such as those of Ahmad et al. (1986) or Hayes (1988).

Work on flow conditioning is at present subject to thorough investigation by Laws and others as indicated, and Mottram and others continue to provide important insights into the effect of pulsating flows.

The critical flow venturi nozzle, as we shall see, appears to be capable of higher precision than has been accepted. There may be value in fluid behavior investigations in this and other areas to define ultimate limits of accuracy.

Specification, Selection, and Audit

3.1 INTRODUCTION

The object of this chapter is to assist the reader in specifying a proposed flowmeter's operational requirements as fully as possible and hence to enable the reader to select the right flowmeter. The reason for wishing to select a flowmeter may be obvious: a flow rate to be measured. However, the reason in some cases may be less obvious. If you are a flowmeter manufacturer, or prospective manufacturer, you may wish to identify a type and design that would meet a particular market niche. If you are a member of an R&D team, you may be exploring measurement areas that are inadequately covered at present.

In preparing this chapter, I have developed earlier ideas (Baker 1988/9, Baker and Smith 1990), but I have also benefited from the ideas of others such as Endress et al. (1989). Baker (1988/9) was the best distillation I could offer in the late 1980s. There is much there that I would still endorse. In Baker and Smith (1990), we took a different line. We provided the means to specify the user's needs in as much detail as possible. We then provided a form for communication with the manufacturer.

Many people have attempted to provide a means of selecting a flowmeter. A glance at only a few of the many books on flow measurement in the bibliography will indicate this. I used to feel that the way forward was to write an expert system to do the job. I am less convinced about that now. One reason for this is that manufacturers appear to be extending the flowmeter types they offer and their applications, so that keeping the expert system up-to-date is a major task, and the decision on the best meter for a particular job may rest on more subtle considerations than an expert system is likely to include. This suggests that the expert system should narrow the field pending the final choice in consultation with manufacturers or consultants.

I have picked out a few examples from the literature of application to specialist areas. This selection is not intended to be exhaustive but rather to complement the applications identified for each flowmeter type in other chapters.

3.2 SPECIFYING THE APPLICATION

The change that I perceive over the last 10 years is that the main manufacturers have recognized the need to have within their armory a meter for every occasion. In some cases, this has led to existing meters being applied to new and more demanding applications. When I wrote Baker (1988/9), I was fairly skeptical as to the extent that

flowmeter types could be eliminated for particular applications. I am now more so. On a regular course at Cranfield University, we divide up into working groups in order to select flowmeters for particular applications. Almost invariably there are three or four meters suitable to a lesser or greater extent for each application.

I have, therefore, developed the theme of Baker and Smith (1990) in this chapter. We start with a form and discuss the implications of each item for user and manufacturer. Before the reader turns to the form and concludes that it is unrealistic for a user or manufacturer to complete or respond to such a form, I would hasten to defend its inclusion because

- i. it provides a check list of features that might otherwise be forgotten in specifying the meter, and
- ii. it is there to be modified, simplified, and reduced in length and complexity, until it suits individual needs.

The form is set out in Appendix 3.A.1 under eight headings, which are labeled alphabetically. It can be photocopied and/or adapted. It is aimed at the user and current manufacturer, but it should provide a starting point for the prospective manufacturer, who may wish to identify a type and design for a particular market niche, or for a member of an R&D team, which is exploring new development areas. In summary the eight headings are:

- A. General information about the meter manufacturer
- B. General description of the meter user's business and specific application
- C. Details of fluid to be measured
- D. Site details
- E. Electrical and control details
- F. Other user information such as preferred accuracy, materials, maintenance, and cleaning
- G. Manufacturer's additional information and requirements such as warranty, filtration requirements, and price
- H. Other considerations and final conclusion

The audit of a flowmeter should link closely with the specification and selection. To ensure consistency, the final section on audit reverts to the specification table and reviews the correctness of the original assessment (if there was one) or the appropriateness of the selection.

3.3 NOTES ON THE SPECIFICATION FORM

A INFORMATION ABOUT THE MANUFACTURER

In approaching a manufacturer, it is important to know whether it will want to sell you its one and only flowmeter type – however appropriate or inappropriate for your application – or whether it has a range of meters, which will ensure that one is appropriate to your needs. It is also useful to know how many the manufacturer sells and the satisfaction of other customers. A manufacturer may be wise to include

illustrations of a high quality production line in its literature as well as indications of its international sales and servicing structure.

In making this point, I have no intention of excluding the smaller high quality manufacturer whose products will be sold on their special features and adaptability. Such a manufacturer will have its own strengths. Indeed, the manufacturer should be clear as to what they are and base its selling strategy on them. In fact, many, if not most, manufacturers' outlets in a particular country will fall in the category of small and medium-sized enterprises (SME) and are likely to give a personalized service.

B USER'S BUSINESS AND SPECIFIC APPLICATION

The object of this section is to provide the manufacturer with a clear overview of the application, site, and essential background to maintenance and safety on site.

The relevant safety documents and those dealing with other relevant standards and legislation on electromagnetic compatibility (EMC) should be referred to (note the CE marking requirement in Europe). Endress et al. (1989) provides a useful digest of the safety requirements for Europe and the United States.

C FLUID DETAILS

The user should endeavor to provide the manufacturer with the fullest information possible to avoid any incompatibility between fluid and meter. Of those properties requested, conductivity only applies to liquids where an electromagnetic flowmeter is to be considered. Opacity would be relevant only for meters that use optical sensing through the fluid. Compressibility may relate to the behavior of differential pressure and ultrasonic flowmeters, and it may be useful for the manufacturer in advising on such applications.

Lubricity has not been included in the list. On the basis of definitions in dictionaries and encyclopedias, *lubricity* may be taken to mean boundary lubricating properties.* It is a term frequently used in the flowmeter business, where the measured fluid is often expected to act as lubricant for the flowmeter rotor. *Lubricity* generally is used to indicate the ability of the metered fluid to lubricate the bearings of a rotating meter insert.

Lubricity does not appear to be a term used by lubrication experts. Its meaning would seem to be related to the high friction coefficient in a lubricated bearing at low viscosity. This is best illustrated by the Stribeck diagram, Figure 3.1(a) (from Summers-Smith 1994). This shows that as the viscosity μ or rotational speed n of the bearing decreases or the load p increases, the friction coefficient η falls to a certain point. In this region, hydrodynamic lubrication occurs, and the curve has a positive and fairly constant slope. Following this for smaller values of viscosity, the curve has a sharp negative slope. Where the curve meets the vertical axis, boundary lubrication takes place as the surfaces come into contact. This is matched by a curve with a similar shape [Figure 3.1(b)] which shows the friction coefficient η as a function of the specific film thickness λ , the ratio of the lubrication film thickness $h_{\min}$, and the combined roughness of the two contacting surfaces σ . The minimum occurs (region

* Private communication from J. D. Summers-Smith.

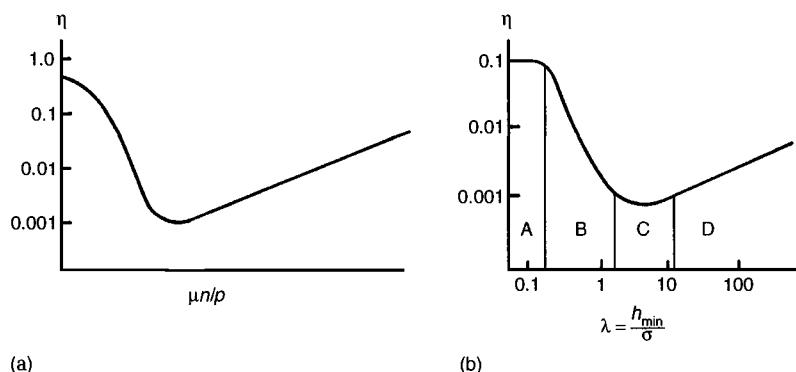


Figure 3.1. Friction variation (from Summers-Smith 1994; reproduced with permission of the author and Professional Engineering Publishing): **(a)** Stribeck diagram for a journal bearing; **(b)** Coefficient of friction as a function of specific film thickness.

C) where the clearance is slightly larger than the surface roughness dimension, and the rapid increase in friction coefficient with decreasing viscosity occurs (region B) due to the smallness of the clearance allowing interference between the roughness on the opposing faces.

It, therefore, seems likely that low lubricity has been used to refer to situations where $\mu n/p$ is very small, where μ is fluid viscosity, n is rotational speed, and p is load. Many gases will require external lubrication in turbine meters, whereas a bearingless flowmeter may be suitable for hydrofluoric acid and very dry liquids such as carbon disulfide.

Some gases (e.g., oxygen) require special lubrication. Because it oxidizes steel pipes with the danger of chunks of rust detaching from the pipe wall, a maximum velocity is sometimes specified for oxygen flows.

Abrasiveness caused by solids in the flow will clearly be a problem for bearings and will lead to wear unless the particulate matter is filtered out.

D SITE DETAILS

Much of the data requested would be clarified and reinforced by a site plan. Details of access and environmental hazards are clearly essential. The access may be such that the meter has to be man-handled into position, placing severe limitations on weight.

E ELECTRICAL AND CONTROL DETAILS

Whether the proposed instrument becomes part of a sophisticated modern telemetry system or is a stand-alone with local power – not to mention all the intermediate variants – will make a lot of difference to meter specification. In addition, all safety details for the site must be set out either here or under environmental hazards in D. Appropriate documents on safety should be consulted.

F OTHER USER INFORMATION

In specifying accuracy, it is essential that the user does not overstate requirements or that the manufacturer does not oversell the product. The accuracy should be realistic when the total installation constraints are taken into account. Response time should

be appropriate to the requirements of the plant. For a plant with, say, a dosing period of 10 s, the response will need to be rapid enough to achieve the required dosing precision. The instrument identified by the manufacturer should fit in with existing plant schedules as far as possible and should be compatible with extreme conditions (e.g., steam cleaning of hygienic process plant).

G MANUFACTURER'S PROPOSED METER AND REQUIREMENTS

This section allows the manufacturer to specify the meter that most nearly meets the user's requirements. The manufacturer should make clear any ways in which the meter will not meet the specification. The details of the calibration procedure for the flowmeter should be spelled out.

The form may well have been supplied to several manufacturers, and this section will provide the user with a comparison of the possibilities.

H ANY OTHER CONSIDERATIONS AND CONCLUSIONS

This is an opportunity for the manufacturer to add further information and for the user to summarize the decision on selection.

3.4 FLOWMETER SELECTION SUMMARY TABLES

Table 3.1 is an update of the table in Baker (1988/9) resulting from the fuller information included in this book. It lists the main types of flowmeter as set out in the chapters of this book and attempts to encapsulate the key features.

For convenience, they are subdivided into momentum-, volume-, and mass-sensing meters. This subdivision also emphasizes that, for instance, a meter responding to momentum will be sensitive to density change. Where a less common meter of the same family has particular features that make it unique or particularly useful in certain applications, these may be indicated in notes in the table and in the discussion in subsequent chapters, where other points arising from this chapter will be elaborated. This table, therefore, provides a first cut at the selection for the task which has been identified by the specification questionnaire. The values given are my best estimates at the time of writing, but they probably fail to reflect the inherent variation in designs.

Nature of the Fluid to Be Metered

The first choice, liquid or gas, will cause a few types of meters to be eliminated. The selection of suitable meters for slurries and two (or more)-phase flows, is severely restricted at present. In Table 3.1, X indicates unsuitable, and * indicates suitable except where some are more suitable (**) than others. A ? indicates that it is not common but might be appropriate in this application. The electromagnetic flowmeter has found a valuable role in slurry flow measurement. However, the Coriolis mass flowmeter is sometimes claimed to be suitable for slurry, and an ultrasonic flowmeter has been specially designed for this purpose. The electromagnetic flowmeter is also useful in two-phase flows when the continuous phase is conducting. However, venturis are usable, but with caution. The ultrasonic correlation flowmeter and the doppler flow monitor may also be worth considering in such flows.

Flowmeters with rotating parts will be less suitable for dirty, abrasive fluids or for fluids that do not lubricate adequately. Abrasion may also detract from the performance of meters, such as the vortex meter. Cavitation should be avoided. The liquid must be conducting for current commercial electromagnetic flowmeters.

Flowmeter Range and Specification

Accuracy is perhaps the most difficult to determine because both user and manufacturer are prone to overstate their requirements and capabilities. In Table 3.1,

***is for random error < 0.1% rate,

**is for random error < 0.5% rate, and

*is for random error < 2% rate.

The estimates reflect my view of the accuracy of flowmeters. These values should be achievable with a particular type of flowmeter. For a standard orifice the uncertainty before calibration due to the discharge coefficient is 0.5–0.75%, and overall uncertainty will be greater than this when errors in the measurement of dimensions, pressure, and density are included. After calibration, the total uncertainty will be less than this, and it is reasonable to assume that the random error will be less than 0.5%. This appears to be confirmed by data. Some manufacturers may feel able to claim higher values for the various meters than those in the table, but the reader should ask for justification for such claims. Commercial flowmeter claims of better than 0.5% indicate a level of accuracy that is probably only achieved in a few designs. Some manufacturers may not always achieve the values indicated. The repeatability of a variable area meter is often quoted as a percentage of full scale, and so may be just outside the * rating for flow rates over, say, the top half of the range.

Precision relates to random error. It does not allow for changes in bias error caused by changes in method of measurement, location, or long elapsed times. Thus, if the reader is concerned about long-term repeatability or total uncertainty, these estimates may not provide the most appropriate guide.

Diameter, temperature, and flow are my best estimates based on manufacturers' literature. In all cases, the diligent reader can probably find examples of greater ranges than those given, and manufacturers are always seeking to extend the capability of their instruments. Note that, in the column for flow range, the values in parentheses indicate kilograms per hour rather than cubic meters per hour.

The maximum pressure has not been included because, apart from exceptionally high pressure applications, when one might turn to an orifice plate or a turbine meter, there appear to be versions of most designs that are offered for normal industrial ranges. In some cases, minimum pressures may be important because cavitation may be a danger, and the manufacturer should be consulted.

The pressure loss is given as H (high), M (medium), or L (low). Only meters with a clear bore have been categorized as L because there is virtually no more pressure loss than in a similar length of pipe. The orifice plate meter is H/M, and positive displacement meter and vortex are H/M, although their relative losses will depend on such things as the instrumentation being driven by the positive displacement meter.

Table 3.1. Flowmeter selection summary tables

1 Flowmeter type	2 Liquid or Gas	3 Slurry	4 Other Two-Phase	5 Accuracy	6 Typical Turndown ^d	7 Dia. ^e Range (mm)	8 Temp. ^f Range (°C)	9 Flow Range ^f (m ³ /h (kg/h))	10 Pressure Loss	11 Sensitivity to Installation ^g	12 Initial Cost	Notes
Momentum^a												
Orifice	L	X	?	**	5:1	50–1,000		1–3 × 10 ⁵	H	H/M	L/M	Concentric ISO orifice with differential pressure cell assumed
	G			**	5:1	50–1,000		10–4 × 10 ⁶				
Venturi	L	*	*	**	5:1	50–1,200		30–7,000	M	M	M/H	
	G			**	5:1	50–1,200		400–10 ⁵				
Nozzles	L	?	*	**	5:1	50–630		2–1.7 × 10 ⁴		H/M	M	Does not include critical nozzle.
	G			**	5:1	50–630		20–2.5 × 10 ⁵				
Variable-area	L	X	X		10:1	15–150	–200 to 350	10 ^{–4} –100	M	L/M	L	Glass/plastic assumed; higher ratings for steel. Consult manufacturers.
	G				10:1	15–150		10 ^{–2} –2,000				
Other Venturi-like	L	?	?	?	3:1	13–1,200			M	H/M	M/H	
	G				3:1	13–1,200						
Averaging pitots	L	X	X	?	10:1	25–12,000	up to 450	10–3 × 10 ⁴	L	H	L	
	G				10:1	25–12,000		200–6 × 10 ⁵				
Laminar	G	X	X	?	20:1	?		up to 120	H	L	M	Particularly appropriate for pulsating flows.
Volume												
Positive displacement	L	X ^b	X ^b	***	10:1	4–200	–50 to 290	0.01–2,000	H/M	L	H/M	Turndown given may be achieved for rotary positive displacement gas meter.
	G			***	80:1		–40 to 65	0.01–1,200				
Turbine	L	X	X	***	10:1	5–600	–265 to 310	0.03–7000	M	H	L/M	High precision instruments assumed rather than water meters, etc.
	G			***	30:1	25–600	–10 to 50	0.01–25,000				
Vortex	L	X	X	**	20:1	12–200	–200 to 400	3–2,000	H/M	H/M	L/M	Fluidic flowmeter suitable for wide range of utility flows.
	G			**	20:1	12–200		50–10 ⁴				
Electromagnetic	L	**	**	**	100:1	2–3,000	–50 to 190	10 ^{–2} –10 ⁵	L	M	M	Only available commercially for conducting liquids.
Transit-time U.S.	L	? ^b	? ^b	**	20:1	10–2,000	–200 to 260	3–10 ⁵	L	H/M	M/H	Single beam are more sensitive to installation.
	G			**	30:1	20–1,000	–50 to 260	0.04–10 ⁵		H/M	M/H	Correlation and doppler for two-phase flows.
Mass												
Multisensor	L	** ^c	** ^c	?	?	?	?		H		H	Specialist instrument especially for engine testing.
Wheatstone bridge	L	X	X	**	50:1	6–60	–18 to 150	(0.05–2.3 × 10 ⁴)	?	L	H	
Thermal	L	X	X	*	15:1	2–6	0 to 65	(0.002–100)	M	L(CTMF)	M	Higher flows require CTFM with bypass or ITMF.
	G			*	50:1	6–200	–50 to 300	(2 × 10 ^{–4} –8000)	L/M	M/H(ITMF)		
Angular mom.	L	X	X	*	7:1	20–50	–40 to 150	(100–4.5 × 10 ³)	M	M	M	Particularly suitable for aircraft fuel flow.
Coriolis	L	*	X	**	100:1	6–200	–240 to 200	(1–7 × 10 ⁵)	M/L	L	H	Straight single tube has essentially zero pressure drop. Gas ranges are probably more limited.
	G			**								

Key: ***, Very high; **, More suitable/high; *, Suitable/medium; ?/blank, Uncertain/lower; X, Unsuitable. L = Low; M = Medium; H = High.

^a Some proprietary devices offer special features: higher differential pressure, linear characteristics.

^b Some designs have been produced for slurries.

^c Multisensor systems have been developed specially for multiphase.

^d Typical estimates that may be exceeded by using smart transducers and may be greater or less than the value attained by some designs.

^e Larger or smaller sizes may be available.

^f Other ranges may be available.

^g Flow conditioning may be used in some applications to reduce this effect.

Table 3.2. Rough selection criteria for liquid and gas flowmeters

[illegible]

Reproduced from Endress et al. (1989) with permission.

Key: ○ = only under special conditions (e.g., special version, only certain diameters); ● = Property of material; ✓ = suitable.

^a Only if liquid is electrically conductive.^b Suitable for steam measurement.

Installation effects caused by upstream pipework will be a recurring theme in most chapters. The positive displacement meter may be assumed to be insensitive to upstream installation effects and is, therefore, rated L. A few others, also, have low sensitivity to installation. However, most flowmeters are affected, and these have been categorized for a fitting, such as a bend, at 5D upstream.

- L, negligible effect;
- M, <2% increase in uncertainty;
- H, >2% increase in uncertainty.

These categorizations are my best estimates, and manufacturers may be prepared to uprate them.

Most flowmeters are sensitive to pulsation over part of their range, with the possible exception of the ultrasonic flowmeter and the laminar flowmeter, which was specially designed to cope with pulsating gas flows.

The flowmeters with essentially a clear straight bore (nonintrusive) are the electromagnetic, ultrasonic, and some Coriolis designs. The only flowmeter that may fail and block the line is the positive displacement flowmeter. All the others have partial line blockage, although the venturi has such a smooth change in section that its effect is probably of small concern.

Only the ultrasonic family offers clamp-on options (noninvasive) at present.

Response time will depend on fluid inertia, component inertia, or electrical damping. The first two are inherent to the meter type and are not reducible. The electrical response may be speeded up in some meters provided stability and noise permit.

The installation of the flowmeter will also be affected by environmental considerations, such as ambient temperature, humidity, exposure to weather, electromagnetic radiation, and vibration. These will need to be assessed in consultation with particular manufacturers. The area safety and explosion classification should also be checked.

Price and Cost

This is possibly the most important consideration and is difficult to tabulate because the range of price for any design is wide and the data are always changing! Therefore, I have attempted to rate the initial cost of the flowmeters as H (high), M (medium), or L (low). However, the medium bracket tends to become a "catch-all" and covers, for instance, electromagnetic and turbine flowmeters, which can actually have an initial cost range of 4:1 for a 100-mm diameter design. For this reason, L/M or M/H have been used where low or high cost versions are available.

Initial cost will, of course, include both purchase price and cost of installation. There will also be ongoing costs associated with maintenance, energy loss due to the presence of the flowmeter, and savings based on the information provided by the flowmeter. If the flowmeter is clearly required, then the savings should outweigh the costs. This does not reduce the responsibility of the user to install the best and most economical instrument for the job!

Table 3.2 is reproduced with permission from Endress et al. (1989) and gives a rough guide to the flowmeters most likely to be useful, but starting with the nature of the fluid and the pipe size.

3.5 OTHER GUIDES TO SELECTION AND SPECIFIC APPLICATIONS

General Selection Advice

Apart from my introductory guide (Baker 1988/9), the reader is referred to the British Standard Guide (BS 7405:1991) to the Selection and Application of Flowmeters for a review of each type of flowmeter and a guide to selection (cf. Furness's 1991 review of BS 7405). Miller (1996) offers some guidance on selection, as does Hayward (1979), which is still good value, even if a bit dated. The Institute of Petroleum guide (IoP 1982) gives a comprehensive list of information required.

Furness and Heritage (1989) gave a breakdown of considerations when selecting a flowmeter, similar to those in the British Standard. In particular, they have used the 11 categories of meter type set out in the Standard, although slightly modified to include open channel flow measurement. This is an interesting subdivision, but it is not one I have chosen to use in this book. A useful aspect of the directory is its ability to provide the names of all flowmeter manufacturers. However, the names change with great rapidity in this field, as company buy-outs take place.

A review of flowmeters by Ginesi (1991) covers well-trodden ground, but it also is a reminder that some less common meters (e.g., the segmental wedge flowmeter or the V-cone) may offer solutions.

Refinery Flow Measurement

Furness and Jelffs (1991) sought to make a case for using meters to measure flows in and out of refineries rather than depending on tank transfer. They also mention briefly some problems in flow metering.

- ❑ For gas metering, the gas often contains liquid.
- ❑ Provided there are no liquid particles, the orifice plate should retain its sharpness, although they suggest that these meters seldom achieve an accuracy of better than 4%.
- ❑ Flare gas metering poses problems of large turndown.
- ❑ Rotating meters are usually unsuitable for naphtha because external lubrication is necessary.
- ❑ Black oils require positive displacement meters with heavy-duty rotors and large clearances, possibly with heat tracing.
- ❑ On-line density measurement (they claimed) had a minimum uncertainty of ± 0.3 – 0.5% .
- ❑ Most Coriolis meters (they claimed) are influenced by density in a way that can be checked only by proving gravimetrically, and the measurement within the meter of density is not sufficiently accurate to use a compact prover to obtain a $\pm 0.2\%$ requirement.

They also refer to Shewhart Control Charts, which plot meter factor against time, with limits to indicate change that requires action by the plant staff; Moving Average Charts, which detect long-term meter factor drift; and Cumulative Sum (CUSUM) Charts, which indicate the occurrence of an event due to some assignable cause.

Water Industry

Fowles (1994) commented that the selection of an instrument for a required duty is a very complex decision-making process. If the wrong selection is made, it can result in abortive cost, erroneous information, and incoherent operational decisions. In addressing these problems, the user is often directed to American, British, European, and international standards. However, such specifications are often not suited to the specific and often unique demands of the industry.

Reynolds (1994) discussed water industry specifications that have led to data sheets on electromagnetic, ultrasonic, differential pressure, insertion flowmeters, and ultrasonic level instruments. The data sheets consist, first, of technical specification and option selection to be completed by the purchaser and, second, supplier-provided information that will influence cost of ownership. A European set of instrument specifications is also being established.

Nuclear Industry

Finlayson (1992) gave information on the use of flowmeters at one plant, which had 1,500 flow measurements:

Type	For Radioactive		For Nonradioactive	
	Liquid	Gas	Liquid	Gas
Electromagnetic	✓		✓	
Ultrasonic	✓		✓	
Differential pressure	✓	✓	✓	✓
Variable area			✓	✓
Turbine	✓	✓		✓
Positive displacement			✓	

For differential pressure, the paper gave diagrams of traps, vents, etc., and listed a number of considerations.

- ☐ Stainless steel should be used for corrosive fluids and when behind shielding.
- ☐ All welded construction should be used to avoid leakage.
- ☐ All vents should be routed to a suitable drain for safety.
- ☐ Drain legs should take the shortest route and be valved before penetrating plant shielding.
- ☐ Sensors should be housed in a shield bulge or a glove box.
- ☐ Drain cocks within a shield bulge should have extended spindles to the outside of the bulge.
- ☐ Transmitters should be located outside the bulge or glove box in a lockable enclosure or equipment room.

The paper described some of the flowmeters and modifications needed for the application, including a time-of-flight ultrasonic one.

Cryogenic Applications

Rivetti and Martini (1990) reviewed various types of flowmeters for space applications and concluded that only in a few cases were they suitable for cryogenic service

and that, even then, care needed to be taken over the use of materials and the sensors. The use of superfluid helium for space applications required new approaches. They discussed specifically designed transducers of small size and weight with good accuracy and reliability. Thermal transducers and turbine meters that used superconducting magnet suspension for the bearings appeared to be the most promising, although much work still remains to be done.

3.6 DRAFT QUESTIONNAIRE FOR FLOWMETER AUDIT

In this chapter we have taken a detailed look at the questions that a user and a manufacturer need to answer before selecting a flowmeter for a particular application. We have then looked at two selection tables, which do little more than indicate the range of possible alternatives for most applications. We have also reviewed some published guidance.

This final section sets out the essential information needed if the flowmeter installed is to be audited in the future. The objectives of such an audit should probably establish one or more of the following:

- ☐ The suitability of the flowmeter for the task,
- ☐ The current state of the meter,
- ☐ The optimum specification for a new flowmeter, and
- ☐ The suitability of the flowmeter's position for precise measurement and maintenance.

If in situ calibration is to be used, then the options need to be considered. The list in Appendix 3.A.2, based on Baker and Smith (1990), offers a starting point for the flowmeter owner. The list should be used either with the specification questionnaire if such exists or to create such a questionnaire for future meter actions.

3.7 FINAL COMMENTS

To the company that uses flowmeters, note that the results of auditing flowmeters may vary widely. The replacement of meters because of age, wear, or incorrect sizing may result in more precise measurements provided installation constraints allow it. However, this may not mean that the user reduces costs because correct measurement may mean higher costs. One is tempted to suggest that, for statistical reasons, the overall gain will be hardly worth the cost of the audit and replacement exercise. Of course, the need for greater accuracy will exist if an absolute flow rate is needed for control or fiscal reasons.

To the person responsible for purchase, make sure that you have specified the requirements carefully, and do not be content until you are sure that you have found a commercial device that matches your requirements in all points.

To the manufacturer, make sure that you have a clear understanding of the following:

- a. what you are offering and why it is superior to your competitors' designs;

- b. whether you expect to sell your products because you can offer a complete range for 95% of applications, or because you have the only device for particular applications;
- c. the weaknesses in your products that need to be addressed and the gaps in your range that need to be filled.

To the research center, identify the applications that have not been met and the technology most likely to meet them.

To the inventor, start from the application problem, look for features that may be inherently flow-dependent, and see if they can be used as the basis for a flowmeter. Remember that the power of modern signal processing should allow the mechanical design to be very simple.

The development of an expert system to satisfy all parties would clearly be valuable, but its production and updating may cost too much to make it worthwhile. I should like to prove myself wrong about this one day.

APPENDIX 3.A

Specification and Audit Questionnaires

3.A.1 SPECIFICATION QUESTIONNAIRE

	Response
A	Information about the manufacturer
A1	Manufacturer and address
A2	Contact person and position in company
A3	Company size and international standing
A4	Flow product range and turnover
A5	Other products
A6	Accreditation (BS 5750, ISO 9000, UK NAMAS, etc.)
B	User's business and specific application
B1	Company's site address
B2	Contact person and position in company
B3	General description of application and telemetry links
B4	Site safety requirements
B5	Current general policy on instrument maintenance
C	Fluid details
C1	Chemical composition, state (gas or liquid), and materials that are incompatible with the chemical
C2	Purity of chemical and if contaminated or multicomponent, the nature of other components, likelihood of deposition
C3	Commonly known properties
	C3.1 Density
	C3.2 Viscosity (if non-Newtonian, state type)
	C3.3 Conductivity

Response

- C4 Other features if known
 - C4.1 Abrasiveness
 - C4.2 Opacity
 - C4.3 Compressibility
 - C4.4 Flammability and other safety considerations
- D Site details (a site plan should be attached if possible)
- D1 Flow range: maximum
minimum
normal flow
likelihood and size of over-range flows
- D2 Pipework (a diagram should be attached if possible
extending to at least 50D upstream, particularly in cases
where there are fittings that create swirl)
 - D2.1 Outside diameter
 - D2.2 Inside diameter
 - D2.3 Material of pipe and pipe lining if present
 - D2.4 Flange size and details or other connection method
 - D2.5 Nearest upstream pipe fitting
 - D2.6 Distance to upstream fitting
 - D2.7 Distance to nearest downstream fitting
 - D2.8 Orientation of pipe run
- D3 Environment, etc.
 - D3.1 Hazards including required safety and explosion-proof
classification
 - D3.2 Access, size, and weight limitations including
vehicle access, machinery transport, and shelter
 - D3.3 Flow source (gravity, pumped, etc.)
 - D3.4 Flow stability
 - D3.5 Mounting requirements (including maximum weight)
 - D3.6 Ambient working pressure: maximum
minimum
normal
 - D3.7 Fluid working pressure: maximum
minimum
normal
 - D3.8 Ambient working temperature: maximum
minimum
normal
 - D3.9 Fluid working temperature: maximum
minimum
normal
 - D3.10 Maximum acceptable pressure drop
 - D3.11 Electromagnetic and radio frequency interference at site
- E Electrical and control details
- E1 Overall control system in use with communication protocol
details
- E2 Power supply availability and variability

Response

- E3 Preferred signal transmission if not defined previously:
- 0-20 mA
 - 4-20 mA
 - 0-1 kHz
 - 0-10 kHz
 - Pulse output: frequency range
 - pulse size
 - Digital
 - Other
- E4 Failure system:
- fail soft one operating and one stand-by channel
 - fail passive two operating channels
 - fail operational three operating channels and one decision element
 - fail safe output goes to safe condition
- E5 Interelement protection (IP) given by IP XY
- X has range 0-6 for protection against access of foreign bodies (Endress et al. 1989)
 - Y has range 0-8 for protection against ingress of water (Endress et al. 1989)
- F Other user information
- F1 Accuracy: uncertainty
- repeatability
- F2 Response time
- F3 Preferred construction materials
- F4 Existing instrument maintenance schedules
- F5 Line cleaning arrangements
- G Manufacturer's proposed meter and requirements
- G1 Manufacturer's proposed meter specification
- G2 Special requirements resulting from this proposal (e.g., filtration, power supply, environment)
- G3 Calibration details, rig conditions, and results
- G4 Warranty period and conditions
- G5 Delivery
- G6 Price
- H Any other considerations and conclusions
- H1 Considerations and constraints by the manufacturer not covered elsewhere
- H2 Agreement of the user to the proposed meter
- H3 Conclusion and decision

3.A.2 SUPPLEMENTARY AUDIT QUESTIONNAIRE

This questionnaire should be read in conjunction with the specification questionnaire in Appendix 3.A.1.

Answer

- A Information about the manufacturer
Not relevant
- B User's business and specific application
If the auditor is not part of the user's company, then
this section may usefully be completed.
- C Fluid details
Is the meter of suitable design, construction, and
materials for this fluid?
- D Site details (Is the site plan correct?)
On the plan, mark power and signal cables and the position of
the main meter electronics.
What is the meter power consumption?
What is the flow range?
Are these within the meter specification?

Pipework

Is the site of the meter satisfactory? If not, should
meter be moved?

Environment, etc.

Is the meter suitable for this environment?
Is meter design suitable for these hazards?
Is the meter capable of operating within these
parameters?
Is access sufficient for proper maintenance?

- E Electrical and control details
Do power, control, and signal systems match requirements
for operating and protection?
Is there power and system access for audit and
maintenance work?
What is the signal level?
Is the meter permanently energized and if not what is
the warm-up procedure?
What is the control system for the meter?
Are there special: alarms?
low flow cutoffs?
- F Other user information
Has the meter accuracy been checked? If so, what
is the uncertainty? repeatability?
What are the materials of the meter? Do they match preferred
construction materials?
Have instrument maintenance schedules been adhered to?
If so, are there any apparent operational changes or
peculiarities that may have caused problems etc.?
Has the meter reading changed unexpectedly?
What are the line-cleaning arrangements?

Answer**G Manufacturer's information and requirements**

Meter specification:

Manufacturer?

Type?

Model?

Serial number?

Specification?

Where specification is unavailable, obtain key details
as far as possible.

Expected accuracy?

Construction date?

Installation date?

Subsequent checking/calibration/maintenance?

Is documentation available? If so, list.

Are all special requirements met (e.g., filtration, power
supply, environment)?

Has the meter met the expected accuracy? If not, was it
unrealistic/unnecessarily stringent?

H Any other considerations and conclusions

Actions resulting from this audit?

Calibration

4.1 INTRODUCTION

The object of calibration is to benchmark a flowmeter to an absolute datum. Just as a benchmark tells us how a particular geographical point compares in height with sea level datum, so a calibration of a flowmeter tells us how the signal from the flowmeter compares with the absolute standard of a national laboratory. The analogy is not perfect at this stage. The national laboratory standard must also be compared with other more fundamental measures of time and mass, and it will be essential to check and compare different national standards in different countries from time to time.

There is a desire to reference back to fundamental measurements such as mass, length, and time. Thus if we can measure mass flow on a calibration facility by using fundamental measurements of mass and time, this will bring us nearer to the absolute values than, say, obtaining the volume of a calibration vessel by using weighed volumes of water and deducing the volume from the density. The first is more correctly termed primary calibration, whereas the second fails strictly to achieve this. It is likely that the final accuracy will reflect this. Liquid calibration facilities can achieve a rather higher accuracy than is possible for most of the gas calibration facilities. In part, this difference will result from the lower density and the increased difficulties of handling a gas.

The result of a calibration will be both a comparison with the national standard and also a range within which the reading is likely to lie.

4.1.1 CALIBRATION CONSIDERATIONS

Figure 4.1 shows three different approaches to obtaining the data. Although the selection of calibration points and their optimum distribution over the range is not obvious, most of the reference books seem to omit any comment on this. Figure 4.1(a) records data at one flow rate. This might be appropriate if the flowmeter were expected to operate at one rate. It shows the fact that there is variation with time to which the repeatability refers. Such a plot might be used to compare the calibration facility itself with other facilities or to test the consistency of diverter operation. Figure 4.1(b) is probably the most usual plot. Here a number of points have been taken over the full range of the flowmeter. Figure 4.1(c) shows a characteristic that has been investigated in four positions, and three readings have been obtained in each.

The number of points will depend on the confidence of the user in the calibration facility, the quality and linearity of the flowmeter, and the statistical requirements for

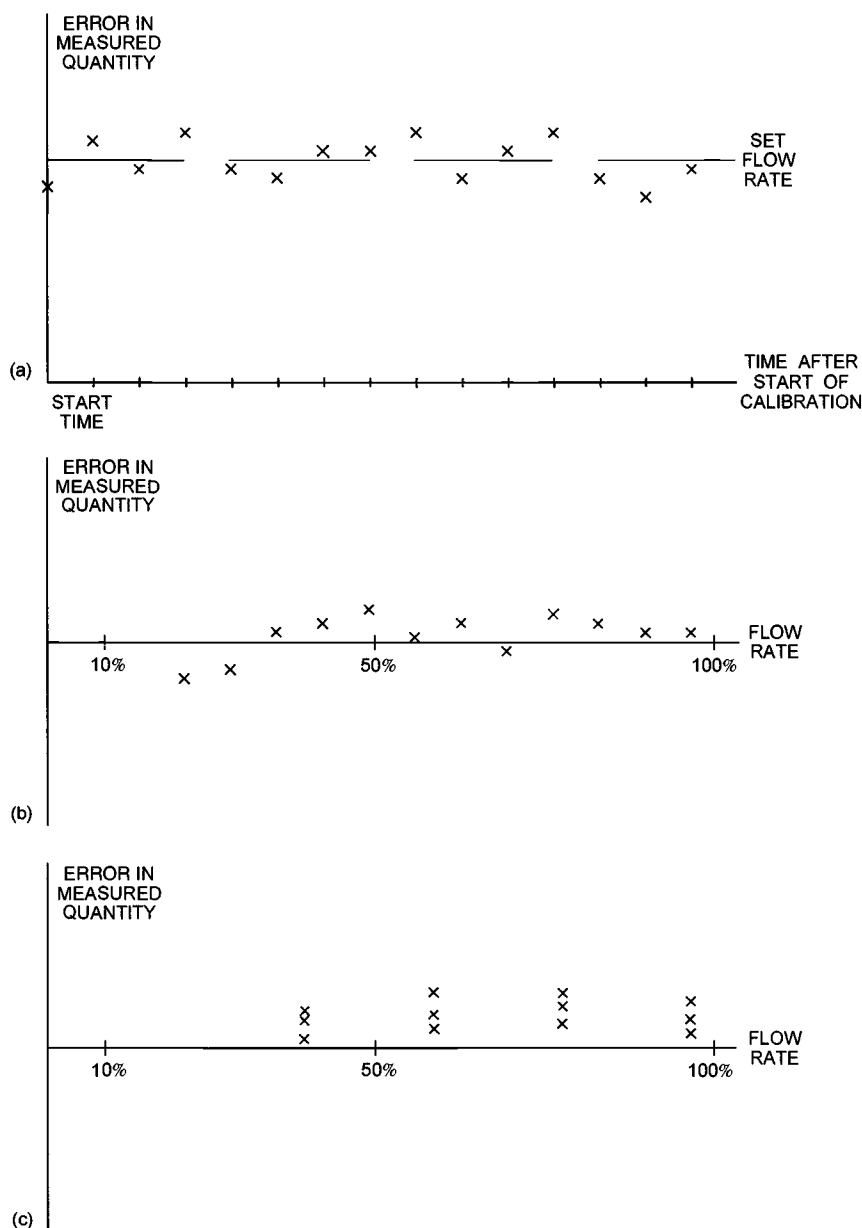


Figure 4.1. Diagram to show different calibration plots. **(a)** Measurements at a constant flow rate taken over a short period of time. **(b)** Measurements of the full characteristic of the flowmeter. **(c)** Measurements repeated at certain points in the flowmeter characteristic.

the required confidence level. Thus a manufacturer may have sufficient confidence in the quality of manufacture of its products to use, say, just two calibration points, and three readings might be taken for each to guard against one spurious reading. It might then be justified in claiming a $\pm 1\%$ envelope for its product. This is unlikely to be a nationally accredited calibration. On the other hand, 20 points or more along a complete characteristic of adequate linearity may be needed to ensure, for a high

quality meter, an acceptable value of uncertainty. At any point on the characteristic, there will need to be sufficient neighboring points to achieve the required confidence level. (See Section 1.A.4 where there is such a calculation.)

Ideally the flowmeter would be calibrated under conditions identical with service ones. In practice, in situ calibration is not very accurate, and a calibration facility will provide the reference conditions for an instrument given that installation and other influence conditions will increase the uncertainty. We shall therefore wish to consider the following factors.

Fluid: If the service fluid is water, then we shall have few problems, although it will still be necessary at least to check for gas in solution, for purity, and for temperature. However, if the fluid is anything else, there will be a range of decisions. If we cannot match the fluid perfectly, we shall add to the final uncertainty of the calibration because the effect of, say, calibrating a meter for oxygen service on air will require a calibration adjustment. This adjustment will, itself, have an uncertainty in it.

Flow Range: It is obviously preferable if this is identical to that for the service. However, if the fluid conditions are not identical, then there may be a compromise between velocity through meter, mass/volumetric flow through meter, and Reynolds number. If the fluid for calibration differs from the fluid for service, the differing viscosity may cause the Reynolds numbers not to match. The importance of this is that the Reynolds number defines the shape of the flow profile and the turbulence spectrum (if the pipe is smooth). The profile shape (and the turbulence), as we noted in Chapter 2 and as we shall see for many of the flowmeters in this book, can affect performance. The viscosity may also change the nature of the flow over the internals of flowmeters, which in turn may alter their performance. Even if the fluid is the same, the temperature may cause changes in viscosity, which may be significant. There are also, of course, flow rate limitations due to the calibration facility.

Pipe Size and Configuration: The service pipe may not be of the same size as that on the calibration facility. Many flowmeters are sensitive to step changes in pipe diameter upstream of the inlet. It may therefore be necessary to calibrate the flowmeter with the upstream service pipework in place. It may be wise to spigot the flowmeter to the pipe, so that if they are separated, they can be reunited in precisely the same way as when they were calibrated as a unit. In cases where there are major changes in the diameter, or fittings causing disturbance, and it is impractical to install enough upstream pipe in the actual facility, it may be necessary to use a flow conditioner package upstream, and the calibration, if possible, should be done with the complete conditioner package and flowmeter bolted together in the calibration facility. These requirements will be discussed in relation to the various flowmeter designs.

In addition, the upstream straight length of pipe should be sufficient to create a fully developed flow profile. This length is usually taken as of order $60D$, although some suggestions have been made that, even after $70D$, the flow has not settled, and it is known that swirl is very persistent.

Pipe Material and Finish: One aspect of this, which may be important (although possibly of secondary effect in many applications), is the roughness of the inside surface – the pipe should be smooth to ensure that the flow profile is predictable.

Roughness may be defined as the arithmetic mean deviation of surface contour from a mean line based on the minimum sum of the square of the deviation. The maximum permitted roughness will probably be in the range 10^{-3} – 10^{-4} D. In some cases, there may be questions of compatibility with the service conditions. For instance, the electromagnetic flowmeter can be sensitive to surrounding pipework if it has special electrical or magnetic properties. Also if the flowmeter is transferred with a length of pipe, this may result in incompatibility with the flow calibration facility.

Steady Flow: Most flowmeters are affected by unsteady conditions. There are several reasons for this; they are set out in Section 2.6. However, the calibration facility should be designed to ensure that flow is steady.

Environmental Conditions: Even though ideally the flowmeter should be unaffected by its environment, in practice this is unlikely to be the case. It is important that any effects are controlled and recorded. Where the service is very likely to be subject to external effects, temperature and pressure variation, humidity, immersion in water, vibration, or hazardous environment, the selection of the meter prior to calibration will need to allow for these. If the meter is in a vulnerable position, there is always the possibility of accidental damage. And, of course, Joe Bloggs, who always taps his spanner on any convenient piece of pipe as he walks through the plant, may find a tender part of the flowmeter!

Experience of instrument performance evaluation (Cornish 1994–5) (reference condition tests and environmental tests) yielded a table of instruments failing to meet performance specification. Out of 162 evaluated, 51% failed manufacturers' specification in 1989–93. Manufacturers' instructions may, in some cases, also be wanting. On the other hand, I have heard it suggested that a high proportion of flowmeters (possibly up to 70% in the water industry) may fail to meet manufacturers' recommended installation requirements.

It should be remembered that, having taken great trouble to calibrate the flowmeter in the correct flow conditions, the actual service installation will probably have many of the shortcomings that we have been at pains to eliminate from the calibration facility and will, therefore, raise questions about the value of the calibration. In subsequent chapters, we shall review much of the available advice in these circumstances.

4.1.2 TYPICAL CALIBRATION LABORATORY FACILITIES

I have found that, among the many papers to which reference is made in this chapter, Mattingly (1982) and Pursley (1986), although a bit dated, still provide excellent reviews.

At the top of the range of facilities are national rigs such as those at the National Engineering Laboratory in Scotland or those listed by Mattingly (1982) at the National Bureau of Standards (now NIST, the U.S. National Institute of Standards and Technology):

High water flow	625 kg/s max	±0.13%	Static-weighing
Low water flow	2.5 kg/s max	±0.13%	Dynamic-weighing
High liquid hydrocarbon flow	100 kg/s max	±0.13%	Static-weighing

Low liquid hydrocarbon flow	10 kg/s max	$\pm 0.13\%$	Dynamic-weighing
High air flow	4980 m ³ /h	$\pm 0.25\%$	Constant volume tank and critical nozzles
Low air flow	90 m ³ /h	$\pm 0.25\%$	Bell provers, mercury-sealed piston devices and critical nozzles

In a recent draft report by Mattingly, the NIST appears now to have uncertainties of $\pm 0.12\%$ for the liquid flow facilities and $\pm 0.22\%$ for the dry air flow facility.

The reader is also referred to King (1992 cf. King 1991), who described calibration facilities at the National Engineering Laboratory, Scotland, in addition to the main NEL gas and water facilities. These further facilities were for variable viscosity oils, user-defined oils, and two-component and multiphase fluids. These are in addition to the standard-type facilities for air, water, and oil. The laboratory includes:

- ❑ three gravimetric primary standards for 1–100 l/s for kerosene, gas oil, and 15-cSt oil to 0.05% of volume at 95% confidence level;
- ❑ one high flow secondary for 1–200 l/s for any of the three oils;
- ❑ two small gravimetric primary standards for user-defined oils 0.1–10 l/s and 0.01–2.5 l/s to 0.05%;
- ❑ a water-in-oil facility of 1–50 l/s for flow studies; and
- ❑ a multiphase secondary standard for oil and water with air or nitrogen in horizontal or vertical lines.

In connection with the measurement of multiphase flow, King suggested that uncertainties of 5% of volumetric flow are likely.

From these examples, the reader can get a good idea of the scope of facilities available in leading laboratories that specialize in flow calibration. It is also likely that the uncertainty claims will have improved with time and experience for these facilities.

4.1.3 CALIBRATION FROM THE MANUFACTURER'S VIEWPOINT

Weager (1993/4), who approached the subject from the standpoint of a manufacturer, described aspects of development of a NAMAS (National Measurement Accreditation Service)-approved flow facility. He commented that “the most used and abused term in any brochure for a measuring instrument is ‘accuracy’, and even the phrase ‘typical accuracy’ has been employed.” Standards committees continue to address the specification of flowmeters (cf. list of standards). Western European Calibration Cooperation (WECC) document 19 also deals with uncertainty. Figure 4.2 shows the typical quality of a NAMAS-approved laboratory, with uncertainty in the range ± 0.1 – 0.2% . The quantity passed was derived as in Figure 4.3. The national authority should ensure that round-robin tests of a transfer standard package are regularly organized between commercial laboratories and national and international standard laboratories to ensure that no undiagnosed changes have occurred. It is essential that the national laboratories are as aware of their own unexpected changes as are other laboratories and are open about the way they address these problems.

SUMMARY OF NAMAS ACCREDITATION			
Test Rig	Accredited flow range l/s	Measurement Uncertainty for Flowrate	Measurement Uncertainty for Quantity passed
100kg	0.5 - 2.5	+/- 0.1%	+/- 0.1%
1200kg	5 - 25	+/- 0.2%	+/- 0.1%
8tonnes	6 - 200	+/- 0.1%	+/- 0.1%
46tonnes	200 - 1200	+/- 0.1%	+/- 0.1%

Note:- Measurement Uncertainty for Flowrate above is derived from interlaboratory comparisons over the whole flowrange, with the National Engineering Laboratory and is not the value as calculated in the table headed "Calculation of Uncertainty Budgets"

Figure 4.2. Summary of NAMAS accreditation (reproduced with permission of Danfoss Flowmetering Ltd.).

Clay et al. (1981) described just such intercomparison tests between three U.K. laboratories using orifice and turbine meters and disclosed a small, 0.1%, bias in one of the laboratories.

Figure 4.3 is an example of the derivation of uncertainty in an actual commercial laboratory. The traceability for a particular laboratory is shown in Figure 4.4 for a water facility. In addition, flow rate uncertainty is assessed by laboratory intercomparison (with the National Engineering Laboratory) using a transfer standard and hence providing a further traceability.

The reality is that most flowmeters are sold with a production calibration and without a fully traceable calibration certificate. The precision claimed for some flowmeters is in conflict with the reality of the precision of the calibration facility. A flow rig with $\pm 0.2\%$ uncertainty cannot be used to credit a flowmeter with a lower uncertainty of, say, $\pm 0.15\%$. It would be useful if manufacturers justified, on the basis of a calibration facility accuracy, their uncertainty values.

4.2 APPROACHES TO CALIBRATION

There are several approaches to calibration, and in the remainder of this chapter they are reviewed in turn.

Dedicated Flow Calibration Facility: This is at present the surest way to achieve a high accuracy. The facility will usually be traceable back to fundamental measurements of time and mass.

CALCULATION OF UNCERTAINTY BUDGETS

Test Rig:- 8 Tonnes [Low Flowrate]
Date:- 21.05.97

Flowrate:-	6	l/s
Target Weight:-	4000	kg
Typical Density:-	998.0000	kg/m ³
Diversion Time:-	667	secs

Measurement quantity:-

Mass

Symbol	Source of uncertainty	Value ± g	Probability Distribution	Divisor	c_i	$u_i(W_x)$ ± g
W_S	Calibration of standard weights	50	normal	2.0	1.0	25
S_D	Discrimination of weighscale	1000	rectangular	1.73	1.0	578
S_L	Linearity of weighscale	100	normal	2.0	1.0	50
S_H	Hysteresis of weighscale	0				
S_T	Temperature stability of weighscale	200	rectangular	1.73	1.0	116
S_R	Repeatability of weighscale	1000	normal	2.0	1.0	500
$u_c(W_x)$	Combined uncertainty		normal			775
$U(W_x)$	Expanded uncertainty		normal (k=2)			1550

Density (densitometer method)

Symbol	Source of uncertainty	Value ± kg/m ³	Probability Distribution	Divisor	c_i	$u_i(D_x)$ ± kg/m ³
D_{TX}	Calibration of Density Transducer	0.200	normal	2.0	1.0	0.100
D_T	Temperature compensation	0.025	rectangular	1.73	1.0	0.014
D_P	Pressure compensation	0.005	rectangular	1.73	1.0	0.003
D_S	Long term stability	0.200	rectangular	1.73	1.0	0.116
$u_c(D_x)$	Combined uncertainty		normal			0.154
$U(D_x)$	Expanded uncertainty		normal (k=2)			0.307

Time

Symbol	Source of uncertainty	Value ±	Probability Distribution	Divisor	c_i	$u_i(T_x)$ secs
T_C	Calibration of Timer	0.001 %	normal	2.0	1.0	0.003
T_D	Discrimination of Timer display	0.1 ms	rectangular	1.73	1.0	0.000
T_T	Temperature stability of Timer	0.001 %	rectangular	1.73	1.0	0.004
T_{DIV}	Diverter operation	126 ms	rectangular	1.73	1.0	0.073
$u_c(T_x)$	Combined uncertainty		normal			0.073
$U(T_x)$	Expanded uncertainty		normal (k=2)			0.146

Summary - Flowrate measurement

Symbol	Source of uncertainty	Probability Distribution	$u_i(\text{FLOW})$ ± l/s	$u_i(\text{FLOW})$ ± %
$u_c(\text{FLOW})$	Combined uncertainty (FLOW)	normal	0.0016	0.027
$U(\text{FLOW})$	Expanded uncertainty (FLOW)	normal (k=2)	0.0032	0.054

Summary - Quantity passed measurement

Symbol	Source of uncertainty	Probability Distribution	$u_i(\text{FLOW})$ ± l/s	$u_i(\text{FLOW})$ ± %
$u_c(\text{FLOW})$	Combined uncertainty (QUANTITY)	normal	0.0015	0.025
$U(\text{FLOW})$	Expanded uncertainty (QUANTITY)	normal (k=2)	0.0030	0.049

Figure 4.3. Example of calculation of uncertainty budgets in flow rate measurement. Note that the symbols, not included in the nomenclature of this book, are defined in the table. Summation of uncertainties is by means of root-sum-squares. (Reproduced with permission of Danfoss Flowmetering Ltd.).

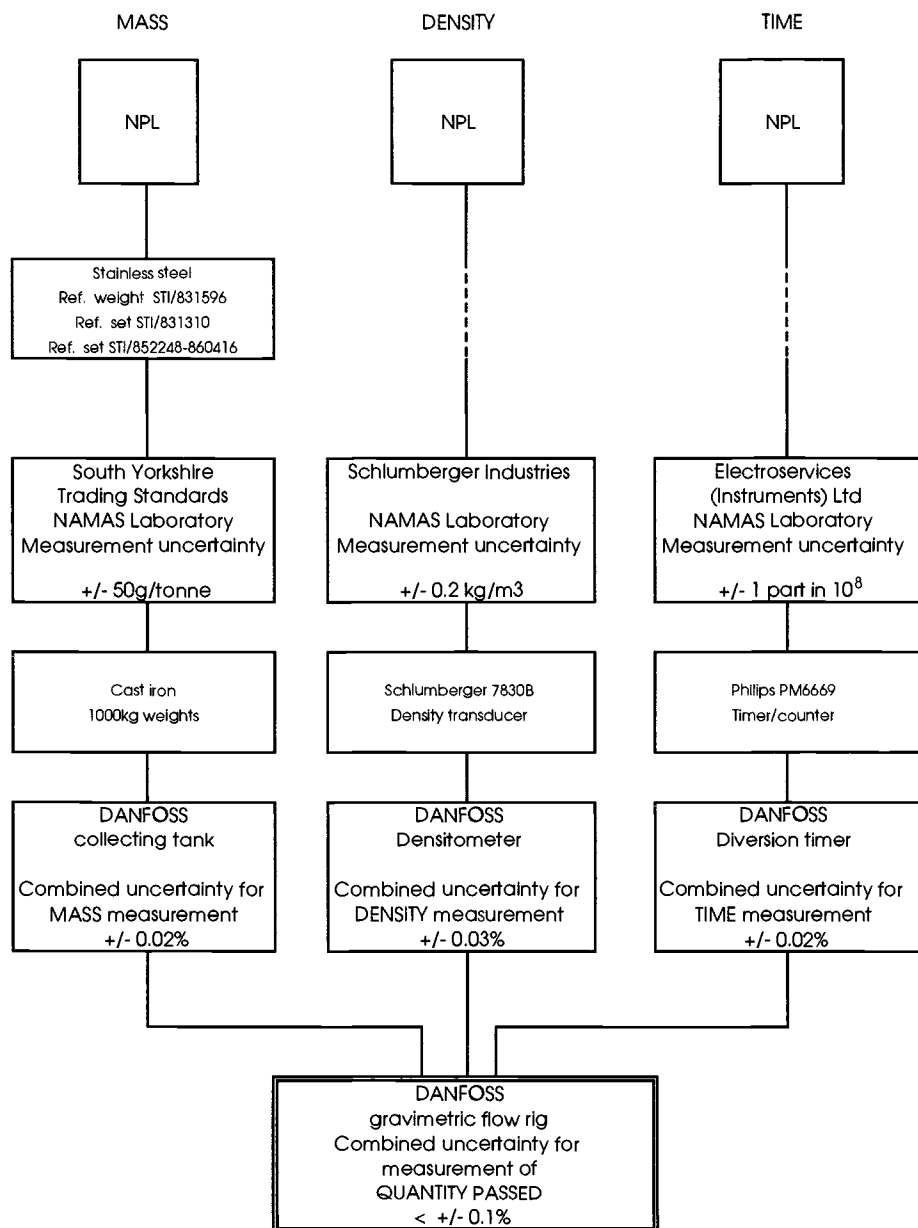


Figure 4.4. Traceability of measurements made on a commercial 8t test facility. (Reproduced with permission of Danfoss Flowmetering Ltd.).

Master Meters: For a simple facility to compare two meters, the master meter offers a suitable transfer standard.

In Situ Calibration: Although there are various ways of calibrating on site, the ultimate accuracy is usually far less than on a calibration facility, and the skill and experience needed is greater. It may, therefore, be preferable to use the term *in situ verification*.

Dry Calibration: Dry calibration is not strictly calibration. An orifice meter, if correctly constructed according to the standards, can be measured, and from the measurements it should be possible to deduce the flow for a particular pressure drop. The same is true of other standard meters such as the venturi, nozzle, and critical flow venturi nozzle. Other meters are less predictable. No flow test is used, and so *dry calibration* is something of a misnomer, since we are really talking about the ability of standards or theory to predict performance. The reader is referred to the appropriate chapter for the likely predictability of particular flowmeters such as the electromagnetic and ultrasonic.

4.3 LIQUID CALIBRATION FACILITIES

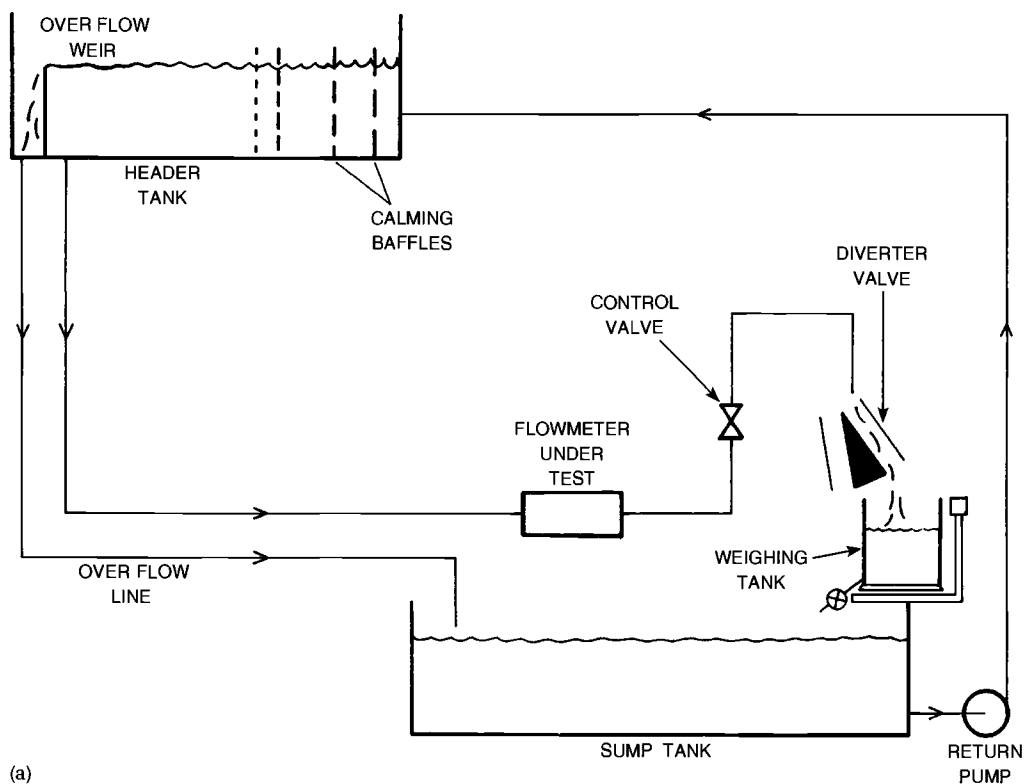
4.3.1 FLYING START AND STOP

Figure 4.5(a) shows the outline of such a facility, which is most commonly used for water. This is known as the static weighing method because the water is diverted into a tank for a fixed period, and the tank can then be measured with the water static in the tank. In its ideal form, there would be a header tank of large enough size so that change in level was negligible. A header tank, with a weir to retain constant head, is shown in Figure 4.5(b). The flow from this, by gravity, would pass through the flowmeter under test and thence to a control valve before discharging into a diverter valve. This valve would either allow the flow to drop into a weighing tank or into the sump tank. Water in the weighing tank would be drained into the sump tank once the weight had been taken. The beauty of this design is that the flow would be extremely steady due to the header tank and errors would be restricted to the measurement of the time during which the flow was diverted into the weighing tank and to the weight itself. Figure 4.6 shows a diagram of the diversion process and the source of timing error. Mattingly (1982) elaborates on the error, and the reader is referred to the original article for full details.

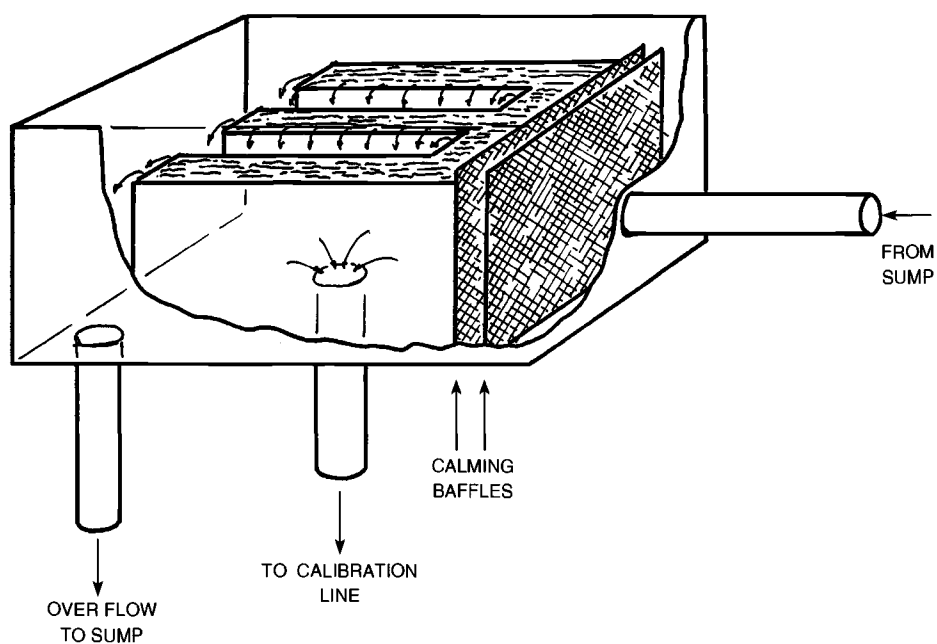
The essential problem is that it is physically impossible to divert the flow to and from the weighing tank instantaneously. The time at which the diversion takes place is, therefore, not at a precise moment, and the changing flow is illustrated in Figure 4.6. The timer is set to start when volume a equals volume b. This is when the lost water shown by volume b is just balanced by the gained water in volume a, which passed before timing began. The same compromise is needed at the end of the timing period with c equal to d. The setting is often, but not necessarily, the same at the beginning and end of the diversion (cf. Buttle and Kimpton 1989 for the design of a diverter).

In order to check the validity of this compromise method, one test is to divert many times after short periods, comparing the result with one diversion over a period equal to the total of the short runs. This will tend to emphasize the errors in the diversion, and an adjustment can be made to minimize them. However, there may be variation in the diversion flow depending on flow rate, and this will require a more sophisticated compensation method.

It is reasonable to assume that the time measurement can now be several orders of accuracy higher than the other measured quantities. The errors will therefore



(a)



(b)

Figure 4.5. Flying start and stop liquid flow calibration facility with static weighing: **(a)** Outline diagram; **(b)** Typical constant head tank design (after Harrison 1978a).

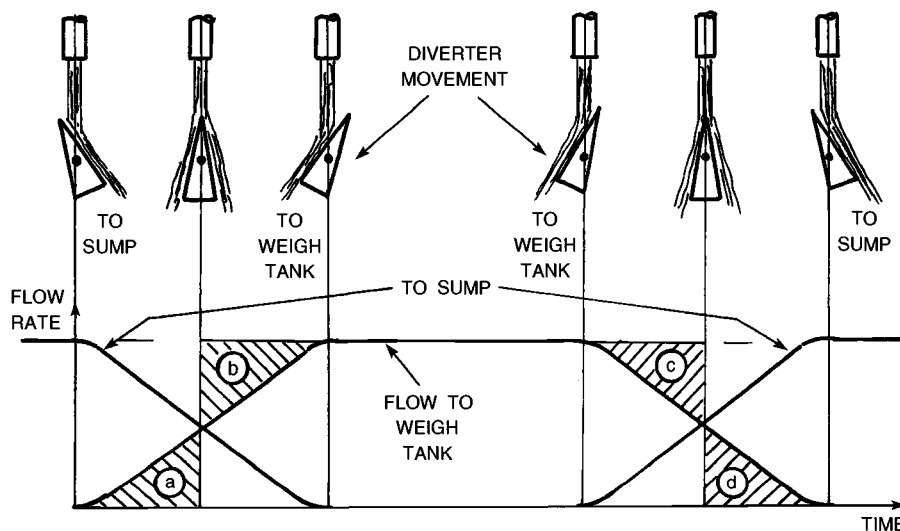


Figure 4.6. Diagram of the flow division for a diverter valve in a flying start and stop facility (after Harrison 1978a).

come from the weight measurements and the setting of the diverter. Scott (1975a) suggests that the true weight of water is about $1/1,000$ greater than the apparent weight. For very accurate weighing, therefore, the upthrust due to immersion of steel test weights and the water in the weigh tanks in air will need to be allowed for. The typical overall accuracy of such a facility is currently of order 0.1% (cf. Kinghorn et al. 1981 in which the U.K. primary standard is described and an uncertainty of 0.1% or better is claimed).

One variant on the weighing method is to use a dynamic weighing system where the flow enters the one tank, and a fast drain is available in this tank. The flow entering the empty tank rises in the tank, and the weight is measured (usually electronically) at the start of flowmeter output measurement. At the point at which sufficient water has been collected, the second weight is measured, and the drain valve is opened. Pursley (1986) advises that the tank should have vertical sides and that the liquid jet should enter vertically. There are problems with the dynamic nature of this at high flow rates: Does the inertia of the weighing tank affect the measurement? Does the jet momentum have any effect?

Another variant is known as the substitution weighing method. At the start, the weighing machine has the nearly empty tank and a calibrated weight of the same value as the liquid that will enter the weigh tank. The weight is removed, and the water is allowed to enter until the balance is restored. If standing start and stop is used, then the final balance can be perfected with additional small weights (Pursley 1986).

Gyroscopic weighing cells have been applied in recent years to increase the performance, but load cells may offer an adequate alternative.

In practice, it is often inconvenient to use a header tank and necessary, therefore, to use some means (a pump) for recycling the liquid. This in turn is likely to make it more difficult to ensure that the flow is absolutely steady.

Scott (1975a) encouraged protection of the calibration weights and their integrity by keeping them in soft material.

4.3.2 STANDING START AND STOP

The standing start and stop amounts to allowing fluid to pass through the meter for a fixed time and then comparing the total registered by a reference meter with the amount collected. The amount collected could be measured either as a volume or a weight. The problem with this method is that the meter must follow the changing flow rate accurately, and many meters are unable to do so. Figure 4.7 is a diagram to indicate the variation between the meter's response and the actual changing flow rate. Each meter will differ according to how it measures the varying flow. There will, therefore, be an error in the measurement in most cases, particularly if the meter is of poor accuracy at low flows.

For standing start and stop, it is clearly essential to be sure that the same amount of fluid has passed through the meter as arrives in the mass or volume measuring tank. This usually requires particular attention to the design of weirs and sight glasses, which confirm that the level in the tank is the same both before and after the run and that the flow stops quickly. Examples of weirs are shown in Figures 4.8(a, b). It also requires care to ensure that valves do not leak. The block and bleed is one system to achieve this [Figure 4.8(c)].

A typical measuring vessel for volumetric measurement is shown in Figure 4.9. The vessel is calibrated using weighed quantities of distilled water at a known temperature. The top and bottom of the vessel are of small diameter so that small volume changes cause measurable level changes compared with the large volume contained in the center portion of the vessel. The initial fluid level is usually fixed by a weir and sight glass. The fluid is then allowed to enter the vessel and is stopped when it rises into the narrow top section. Against this, there is either a calibrated sight glass or a means to measure the surface level to obtain the final volume. It has been found that the vessel will need to be filled and emptied three or four times before it has wetted sufficiently on the inner surface to ensure that a consistent and correct reading is obtained. When a consistent reading is obtained, the value can be used to calibrate the flowmeter.

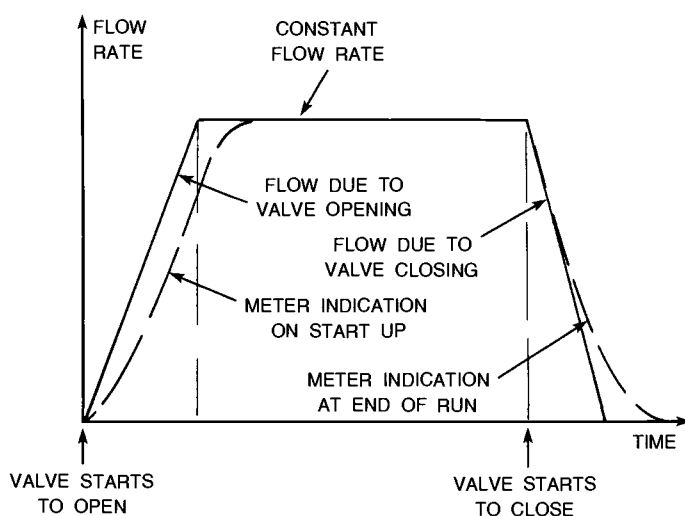


Figure 4.7. Discrepancy between changing flow rate and the meter's response.

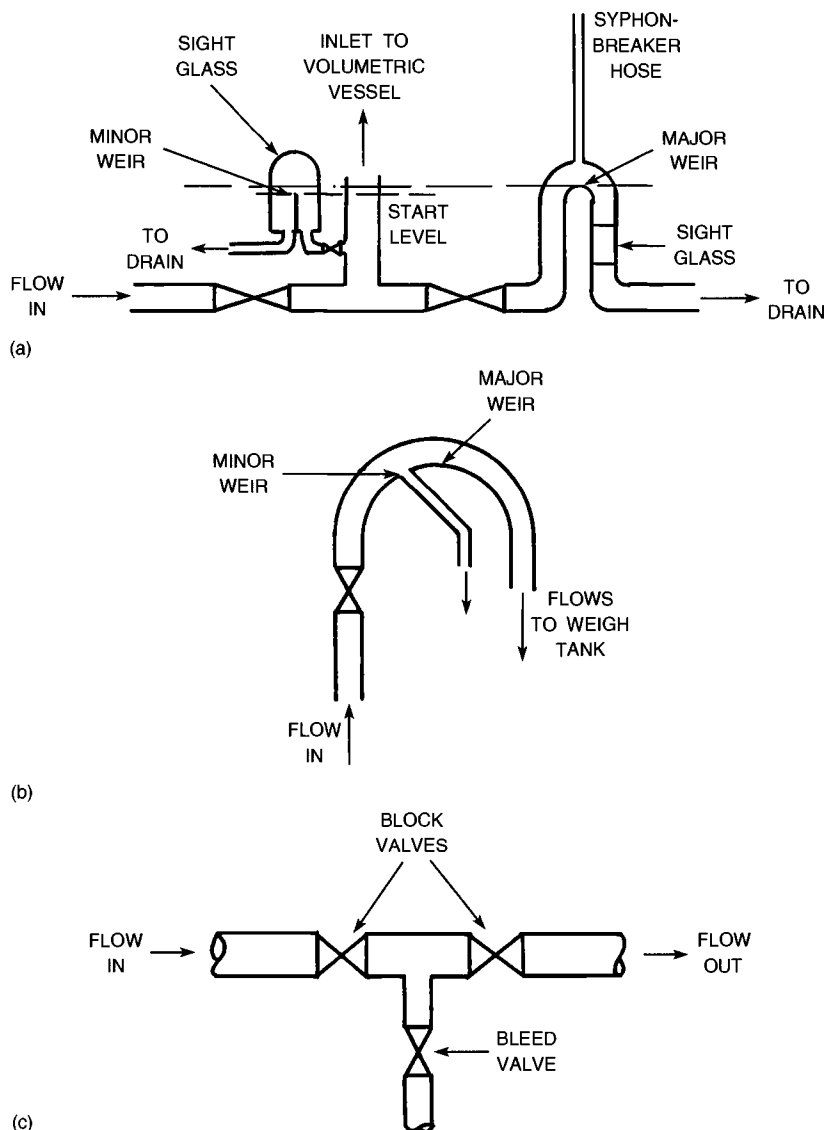


Figure 4.8. Examples of weirs, swan necks, and block-and-bleed valves in calibration (after NEL). **(a)** Arrangement for precise starting level for volumetric tank. **(b)** Arrangement for standing start and stop gravimetric system. **(c)** Block-and-bleed valving system.

The viscosity of the liquid will also affect the emptying of the vessel, and as a result Pursley (1986) recommends a maximum viscosity of 5 cSt for this calibration method.

Proving vessels of this type are used to calibrate meters to uncertainties of $\pm 0.1\%$ for custody transfer of hydrocarbon liquids. A typical vessel could have a capacity of 500 liters. Such vessels, of various sizes, have been used for a variety of liquids including foods.

Another type of volumetric facility uses the falling head method. A tall tank is filled with liquid. The liquid is allowed to flow through an outlet pipe to a flowmeter.

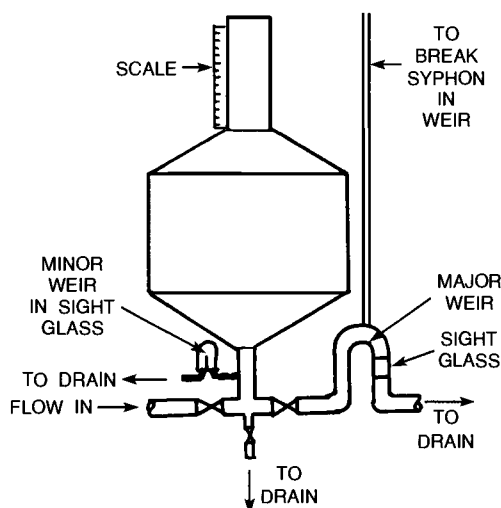


Figure 4.9. Volumetric proving vessel for standing start and stop calibrations (after NEL).

The volume between two level switches is known, and the meter is calibrated against this volume. The disadvantage of this method is that the flow rate may change slightly as the head falls.

4.3.3 LARGE PIPE PROVERS

Figure 4.10(a) shows a typical design of prover, and Figure 4.10(b) is a photograph of a commercial system. The prover consists of a carefully constructed length of pipe. In a bidirectional prover, the sphere passes both ways. A sphere is projected into the pipe, and its passage is recorded past two set points where there are detectors. When it passes the first, the meter-pulsed output is counted until the sphere passes the second, and the total number of pulses is obtained.

The volume swept is known, and the time taken is recorded. With this data, the flow per pulse can be checked with the prover volume, and the actual flow rate can be obtained. Unidirectional provers are also available. Instruments of this type are commonly included in hydrocarbon metering stations, where oil and hydrocarbon products are transshipped, and fiscal and custody transfer requirements come into play. Figure 4.10(b), a photograph of a prover, shows the valving arrangements to bring the prover into play. Figure 4.10(a) also shows the separate flanged connections for use when the prover is being calibrated. This may be done by connecting up a mobile proving vessel or transfer standard flowmeter. The calculation of the internal volume of the prover by measurement, even if the accuracy is achievable, would seem unlikely to be an economical method (cf. Paulsen 1991 on prover ball material problems and materials).

Claims are made that these devices can achieve uncertainties of the order 0.1% (or less) with calibration.

4.3.4 COMPACT PROVERS

The large bulk of the pipe prover, which leads to weight (e.g., in oil platform installations) and space considerations, and also the amount of liquid needed to complete the calibration have led to the design of very compact piston provers [Figure 4.11(a)]. These devices still depend on the swept volume of a tube but use a piston moving over a much shorter distance. Figures 4.11(b)–(d) show the stages in the measurement cycle. Thus, with great manufacturing precision, it appears that the performance of these compact provers can rival that of the large pipe prover. Compact provers will, certainly, target the same level of accuracy as pipe provers in order to provide a realistic alternative.

One problem that arises in the use of these compact prover results is the small amount of fluid that passes. Thus, if a flowmeter produces a pulse for a certain volume of fluid passed, the number of pulses resulting from the fluid passed by a compact prover may be very low, causing a discrimination error. This has led to

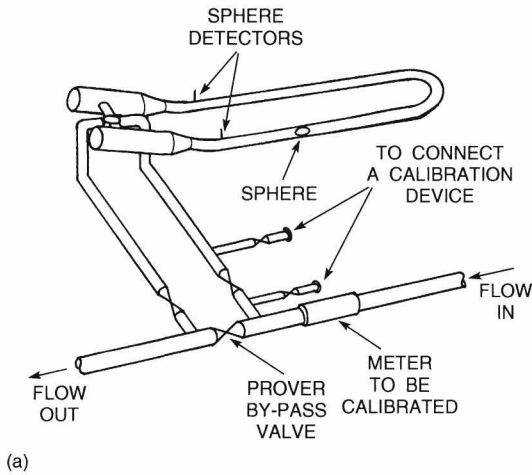
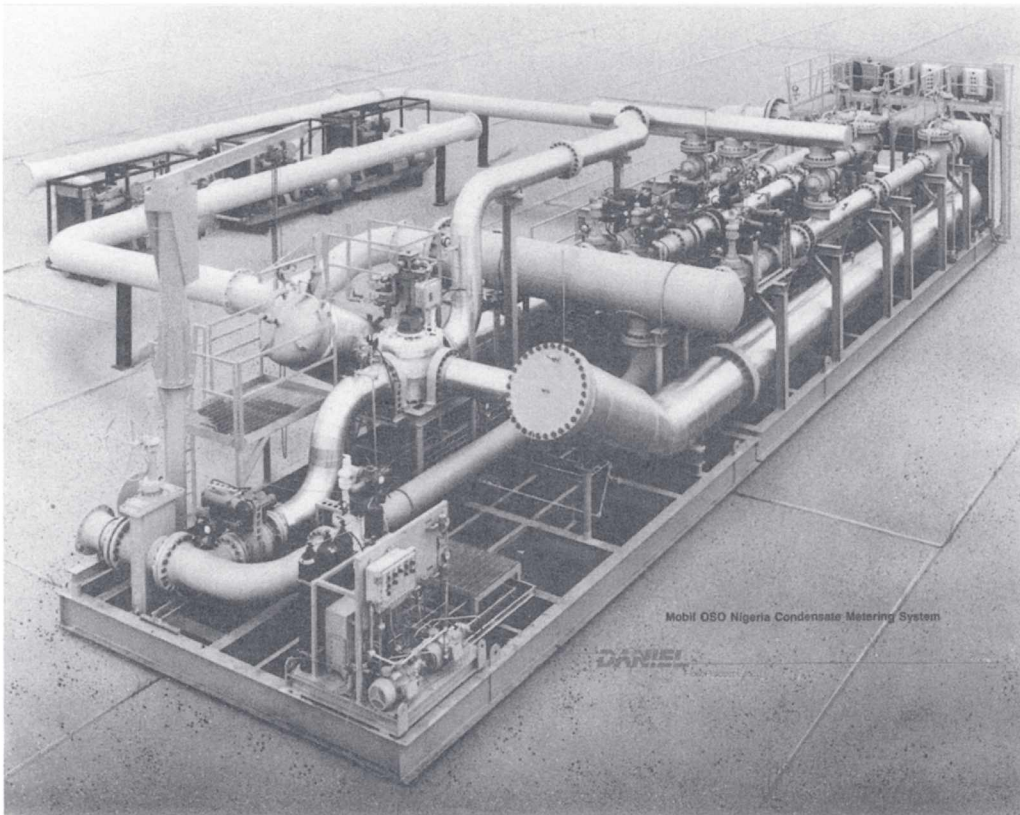


Figure 4.10. Large pipe provers: **(a)** Schematic of bidirectional prover (after NEL); **(b)** Photograph of a bidirectional prover (reproduced with permission of Daniel Europe Ltd.).



special techniques to overcome the problem, such as pulse interpolation (Furness and Jelffs 1991) in order to achieve the required 0.01% resolution (based on 10,000 pulses for the pipe prover). Reid and Pursley (1986) used a double chronometry method for pulse interpolation [Figure 4.12(a)]. As the first prover volume switch is activated, two timers are started. The first timer (A) runs for the full period of

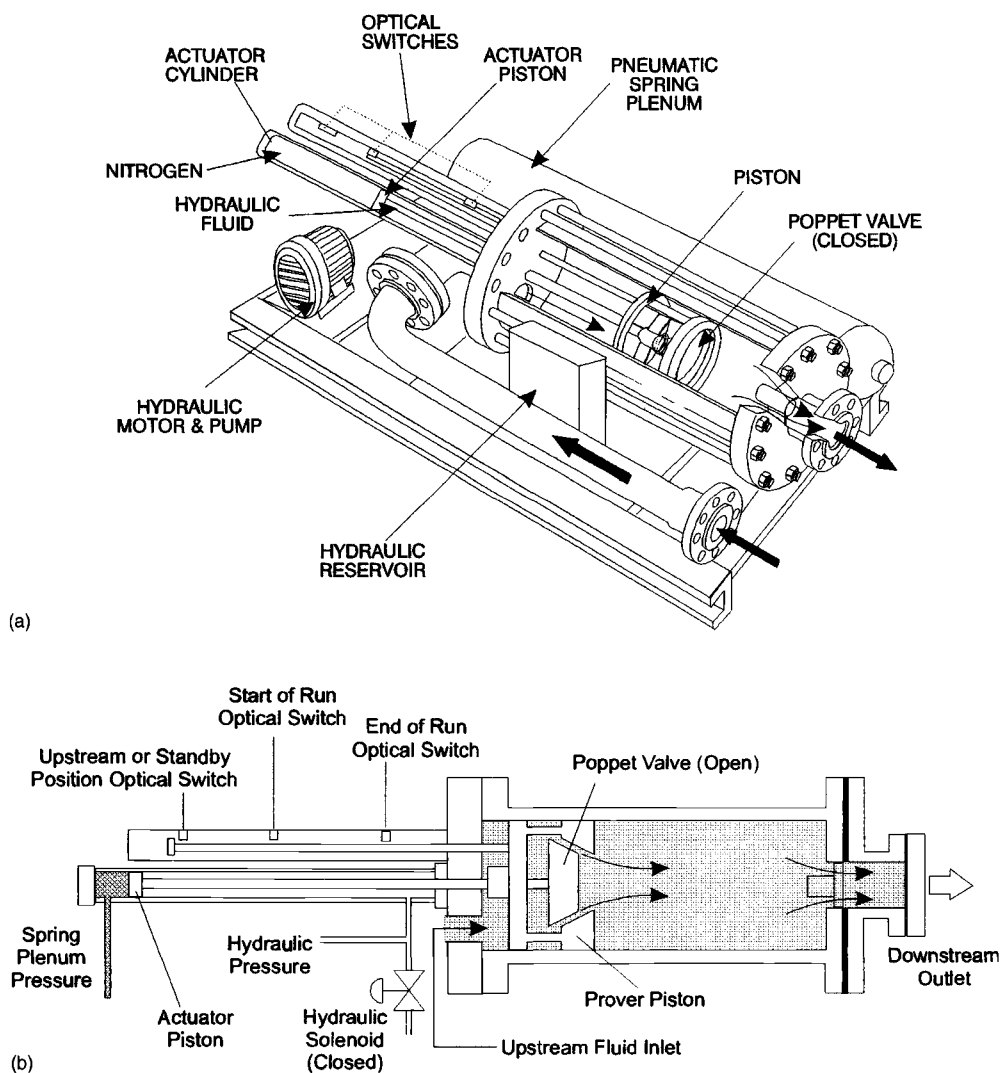
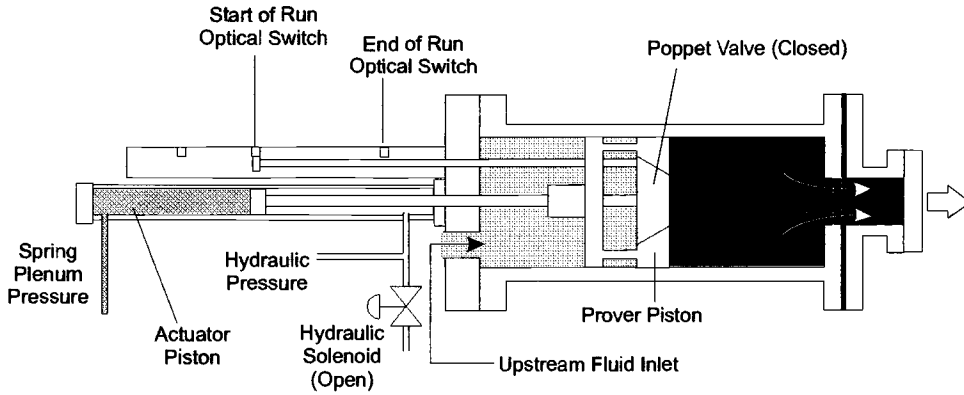


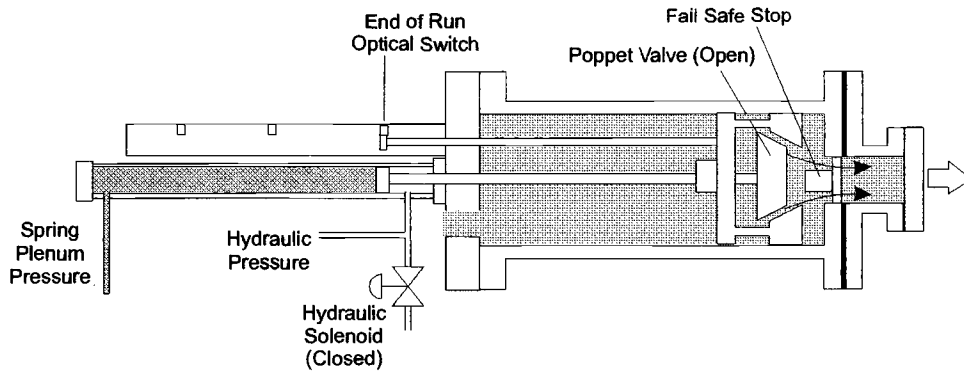
Figure 4.11. Compact prover (reproduced with permission of Fisher-Rosemount Ltd.): **(a)** Principal components; **(b)** Standby mode; **(c)** Start of run; **(d)** End of run.

the piston movement, from activation of the first prover volume switch until the second prover volume switch is activated (volume D). The second timer runs until the first turbine pulse and starts again at the second prover volume switch, running until the subsequent turbine pulse. Alternatively, the second timer (B) runs from the first turbine pulse after the first volume switch until the first pulse after the second volume switch (C pulses in total). Thus the full time, the number of pulses, and the fractional period between pulses are all obtained. The turbine pulses per unit volume K can be found by multiplying (Time A/Time B) by the total pulses C and dividing by the displacer volume D [Figure 4.12(b)].

An advantage of the compact prover (Furness and Jelffs 1991), especially for meters delivering products to ships, is that it can be easily cleaned between grades



(c)



(d)

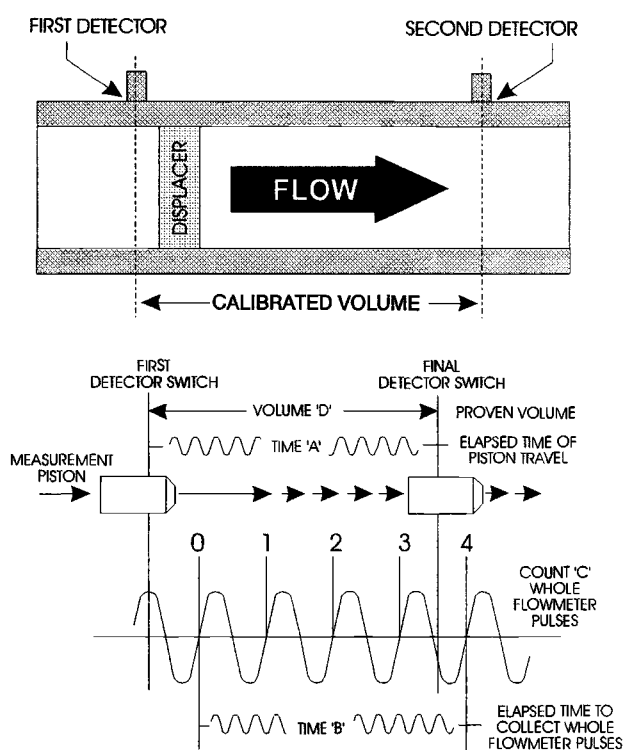
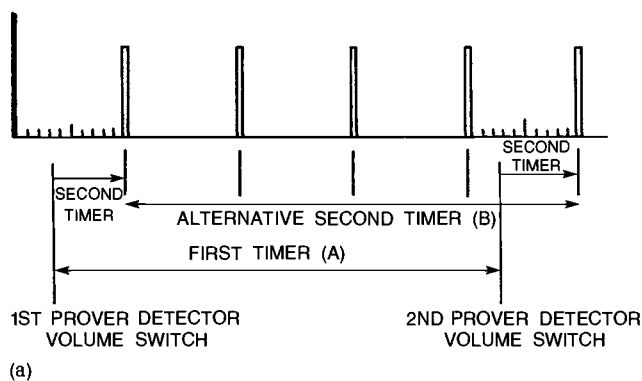
Figure 4.11. (continued).

and so eliminate contamination. It can also be skid mounted. Wherever possible, comparisons between ship and meter quantities should be monitored continuously to provide warning of meter factor drift (cf. Hannisdal 1991 on alternative metering concepts to reduce space and weight).

4.4 GAS CALIBRATION FACILITIES

4.4.1 VOLUMETRIC MEASUREMENT

The bell prover uses a standing start and stop. Figure 4.13 shows the main features. An inverted vessel – the bell – is held by a cable and counterweight so that it dips into a sealing fluid. This is usually water or a light oil. The flowmeter to be calibrated is connected into a pipe that allows the gas in the gas cavity under the bell to flow out. The bell is allowed to sink into the liquid so that the gas is expelled from the cavity through the outlet pipe and so through the flowmeter under test. It is therefore possible to take the height of the vessel at start and end and to deduce the volume of gas that has passed. For high accuracy, various other factors need to be taken into account. The volume of gas in the vessel will be affected by the pressure and temperature within the vessel. To a lesser extent, it will be affected



$$K = \frac{\text{TIME A}}{D} \times \frac{C}{\text{TIME B}}$$

(b)

Figure 4.12. Interpolation of flowmeter pulses for compact prover. **(a)** General concept and double chronometry method for pulse interpolation (after Reid and Pursley 1986). **(b)** Double chronometry method as used by Fisher-Rosemount (reproduced with permission).

by change in the volume of the vessel. The humidity of the gas will be important if water is used instead of oil. The movement of the vessel, if continuous monitoring of the flow is taking place, may need to be allowed for. This method is particularly suitable for small domestic gas meters and is capable of achieving an uncertainty of as little as $\pm 0.2\%$ if particular care is taken. There are a number of elderly bell provers still in use, but there are also some new ones providing high accuracy in national laboratories.

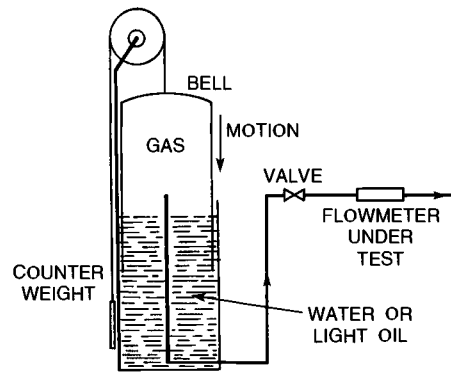


Figure 4.13. Bell prover.

A similar approach, using a large bag of known volume, accommodates larger volumes.

Bellinga and Delhez (1993, cf. Bellinga et al. 1981) of Nederlandse Gasunie, Groningen, described the conversion of a portable piston prover for liquids to use with gases; the prover is capable of flows from 20 to 2000 m³/h. The cylinder length was 12 m with a measuring section of 5.22 m, diameter of 0.584 m, and volume of 1.3988 m³. They also described the traceability chain before the prover came into operation. During the initial calibration of the piston prover, the volume was determined by weighing the quantity of water displaced by the piston when moving through the measuring section. A dynamic calibration was achieved by using a master meter. As a check, the volume displaced was obtained from measurement of the dimensions of the prover. They claimed that these different methods should agree to within 0.03%. The pressure difference needed to drive the piston at 150 m³/h was 20 mbar, but at lower flow rates the motion became unsteady and was corrected by reducing the mass of the piston and adding a dynamic damping system. Bellinga and Delhez also mentioned other small changes to valves, valve operating speed, and electronics that include pulse interpolation. The aim was for the prover to be accepted by the Netherlands Measurement Institute as the primary standard for high pressure gas meter calibration. The meters calibrated from this prover should have an uncertainty in meter factor of as good as 0.1–0.15%. Reid and Pursley (1986) considered the use of nitrogen as a pressure-balancing gas to reduce the differential pressure across the piston and hence to reduce the leakage and to provide stability of motion at low flow rates.

Reid and Pursley claimed 500:1 operation in air for pressures greater than 30 bar with agreement to 1% and repeatability of order 0.25% up to 8 bar.

Jongerius et al. (1993) described a new low flow test facility consisting of four mercury seal piston provers at the Nederlands Meetinstituut with an operating range of 2×10^{-5} –3.5 m³/h at approximately ambient conditions and operable, in principle, on any gas. The low flow facility is traceable to the primary length standard, and uncertainty is about 0.2% for the smallest prover and 0.15% for the others.

4.4.2 MASS MEASUREMENT

Figure 4.14 shows the scheme for a gravimetric calibration facility. The system shown depends on the use of critical nozzles (which will be explained in Chapter 7)

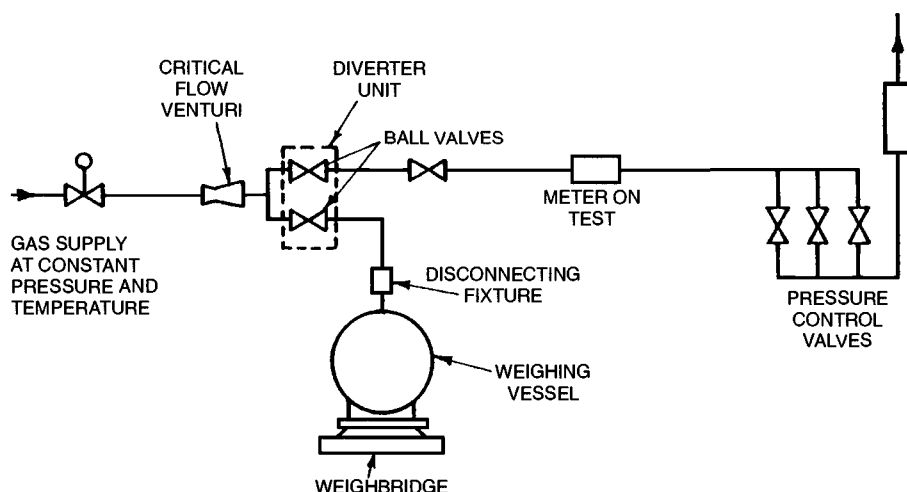


Figure 4.14. Scheme for a gravimetric gas calibration facility (Pursley 1986 reproduced with permission of NEL).

as transfer standards. The facility is used to calibrate the sonic nozzles, then to calibrate the flowmeter under test, and, if appropriate, to recalibrate the nozzles. The problem with gravimetric measurement in gases is the low density of the gas and the consequent problems of mass measurement. In addition, the flow control in the calibration facility must be such that the flow rate through the meter under test is the same as that into the weighing vessel. The facility at the National Engineering Laboratory, Scotland, is used for flows of up to 5 kg/s at pressures of up to 50 bar to within an estimated uncertainty of $\pm 0.3\%$ (Brain 1978).

4.4.3 GAS/LIQUID DISPLACEMENT

The problems of calibration of gas flow has led to the use of a device that exchanges liquid flow by volume for gas flow by volume, say, by means of a piston. The liquid is metered to a high accuracy, and the gas at controlled temperature and pressure is used to calibrate the flowmeter.

Lapszewicz (1991) described a flowmeter for gas mixtures in which liquid is displaced, and the time and volume between two points on a tube is measured. A valving system allows the gas to enter and be exhausted.

4.4.4 p_vT METHOD

Provided that the gas under calibration obeys a well-defined relationship between the pressure p , the specific volume v , and the temperature T (e.g., for an ideal gas),

$$pv = RT$$

where R is the gas constant, the change in quantity in a pressure vessel can be deduced from the change in pressure and temperature, and the volume or mass

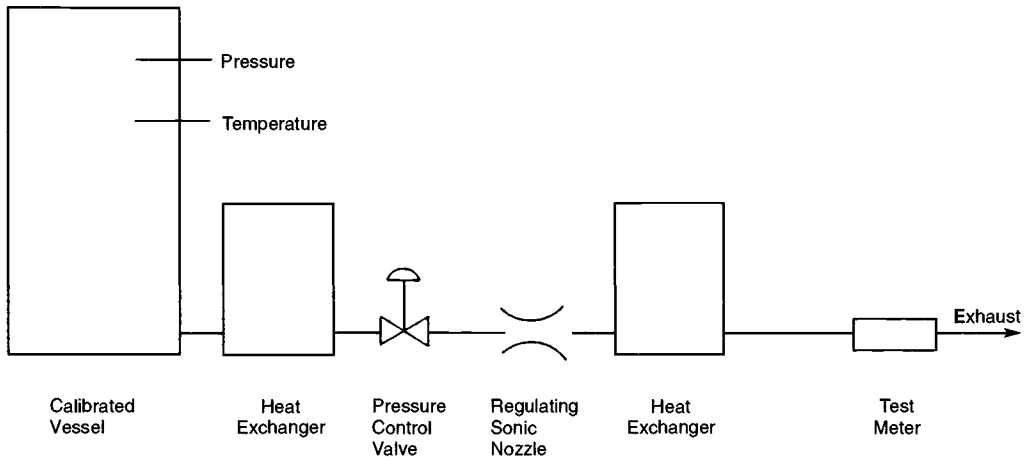


Figure 4.15. Diagram of a typical facility for the pVT calibration method (Pursley 1986 reproduced with permission of NEL).

passing through the meter under test can be deduced. A simple facility is shown in Figure 4.15. The flow and temperature of the gas are controlled by heat exchanger and pressure regulation, and the flow rate may be stabilized by a critical nozzle followed by a further heat exchanger.

4.4.5 CRITICAL NOZZLES

The design and operation of these devices will be described in Chapter 7. Here we note only that they are rather inflexible in that the flow through them is, essentially, defined by the inlet pressure and the area of the throat of the nozzle. Thus to accommodate a range of flows, they need to be set up as a bank of flowmeters. A typical bank of such flowmeters is shown in Figure 7.2. The approach will be to set these in series with high quality master flowmeters. In order that the flow range is as complete as possible, the mass flows through the bank need to be in a progression such as 1, 2, 4, 8, 16. This will allow flow rates of 1, 2, 3, 4, 5, ..., 31, by suitable combination of the nozzles (cf. Aschenbrenner 1989 who described the calibration of a bank of 16 nozzles). In such a system, it will, clearly, be essential to have a means of checking the integrity of the valving system so that accidental leaks do not affect the accuracy. With this range, it is then possible to calibrate the master meters over the same range and to use these either on the facility, or as transfer standards (c.f. Bignell 1996a on comparison techniques for small sonic nozzles).

4.4.6 SOAP FILM BURETTE METHOD

Figure 4.16 shows a simple arrangement of the soap film burette calibration system (Pursley 1986). This is suitable for very low flow rates. The gas enters the burette after leaving the flowmeter, and a soap film, created within the gas flow, is driven up the burette by the flow of gas. The movement can be observed and timed by eye, or sensing methods, probably optical, can be used to measure its transit time. Clearly the burette will need to be calibrated for volume. Flow range is 10^{-7} – 10^{-4} m^3/s at conditions near to ambient (Brain 1978).

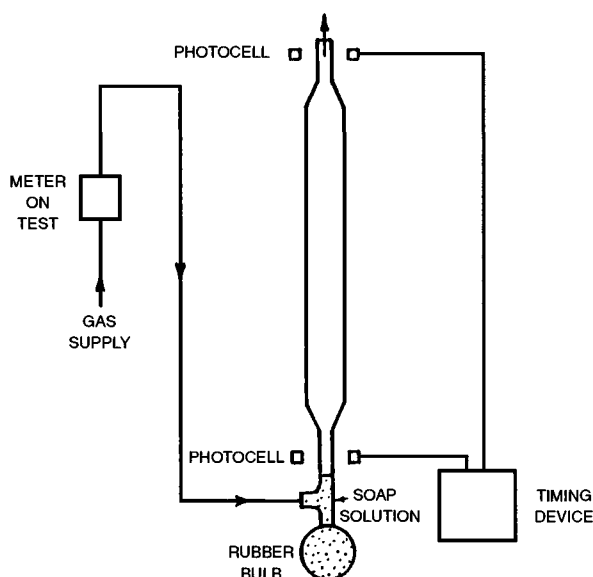


Figure 4.16. Soap film burette (Pursley 1986 reproduced with permission of NEL).

4.5 TRANSFER STANDARDS AND MASTER METERS

Calibrated master meters may also be used to measure the flow in a specially constructed flow facility in which the masters are in series with the meter to be calibrated. This may be the most economical approach for many manufacturers. The cost will, of course, include the regular recalibration of the master meters. Such a facility will need careful design to achieve:

- ❑ steady flow,
- ❑ fully developed flow profile upstream of both the reference (master) meter and the meter under calibration,
- ❑ adequate range for all meters to be tested,
- ❑ flexible means to accommodate flowmeters of different diameters and axial lengths and to mount them in the line, and
- ❑ high quality sensing and recording systems to provide the comparison between the meter under calibration and the master meter.

A good transfer standard should have the following characteristics (Harrison 1978b):

- ❑ highly repeatable,
- ❑ stable with time,
- ❑ wide flow range,
- ❑ insensitive to installation,
- ❑ simple and robust,
- ❑ compact,

- ❑ low head loss,
- ❑ suitable for a variety of fluids, and
- ❑ available in a wide range of sizes.

Meters suitable for use as masters for transfer standard work are, therefore, the high accuracy meters, which will be described in later chapters. In particular, these include positive displacement meters and turbine meters. For conducting liquids, electromagnetic meters may, now, have reached a sufficient standard to be considered in some cases. Ultrasonic multibeam meters may, now, also be found to be adequate, as may Coriolis meters. In gas flows, the ultrasonic multibeam may be suitable, and the critical nozzle meters, as indicated earlier, are very appropriate despite their inflexible flow range. Wet gas meters, although a fairly elderly design, are still used by some standards laboratories as transfer standards.

The problem with using one meter on its own is that the stability of its signal may not be observable, and errors may creep in without any check. For this reason, master meters are often used in pairs, either in series, so that the consistency of their readings is continually checked, or in parallel when one is used most of the time and the second is kept as a particularly high accuracy meter for occasional checks.

A development to achieve very high performance and to reduce sensitivity to installation is the calibration of the meter with the upstream pipework and a flow straightener permanently in position, and the use of two of these packages in series (Figure 4.17; this arrangement was suggested by Mattingly et al. 1978, with or without the initial flow straightener). In this way, any deviation caused by installation or by drift of one meter will appear as a relative shift in calibration between the two meters. Such meters or meter pairs are referred to as transfer standards in that they allow a calibration standard to be transferred (with only a small increase in uncertainty) from a nationally certified facility to a manufacturer's facility or a research laboratory facility.

The meters selected as transfer standards must be capable of retaining a high performance with removal, transport, calibration, subsequent removal, transport, reinstallation, removal, storage, etc! Several of the main flowmeter designs will meet this requirement, and the actual selection will depend on other factors. The line liquid should be without gas bubbles or cavitation. The meters should be selected for optimum range; in some cases, overlapping ranges may be an advantage.

Apart from flow profile, the pipework for transfer standard installation would have the following features:

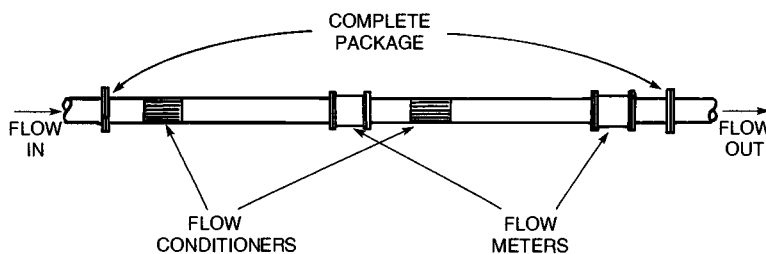


Figure 4.17. Transfer package allowing two meters, conditioners, and pipework to be calibrated together.

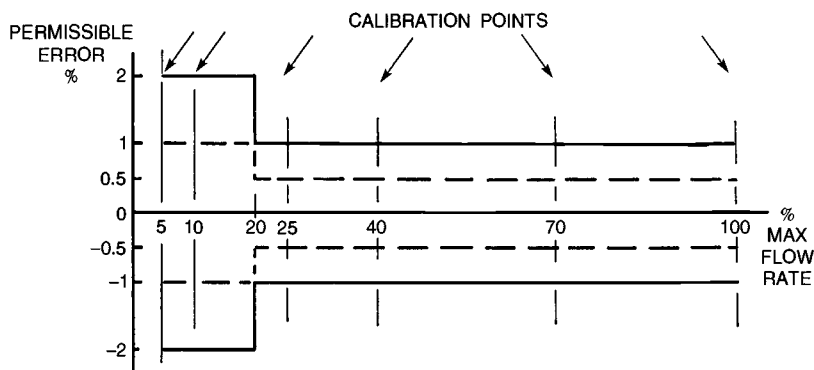


Figure 4.18. Maximum permissible errors: solid line, ISO 9951; broken line, Gasunie specification (after van der Kam and Dam 1993 with permission of Elsevier Science).

- uniquely linking meters with convenient valves to control flow path and for ease of access;
- suitable environment, meeting safety requirements;
- freedom from vibration and cathodic protection; and
- surrounding pipework clean, in good condition, and of the same diameter as transfer package.

Van der Kam and Dam (1993) described the replacement of orifice meters by turbines, in export stations of Nederlandse Gasunie. Even though ISO 9951 specifies a Maximum Permissible Error (MPE) as $\pm 1\%$ in the upper range, van der Kam and Dam reported that, in their experience, turbines can operate within an envelope of $\pm 0.5\%$ down to about 25% maximum flow and of $\pm 1\%$ in the lower range (Figure 4.18). They suggested that the four-path ultrasonic meter would probably be best of all.

Pereira and Nunes (1993) described a facility with an array of ISA 1932 nozzles as the calibration source. The sizes were such as to allow a 50:1 turndown in air, and where nozzles were not needed, rubber balls were inserted in the inlets to seal them off and prevent flow. Uncertainty was claimed to be in the range 0.5–0.8%.

4.6 IN SITU CALIBRATION

A new flowmeter should have some performance specification. The need to keep documentary evidence on flowmeters should be reinforced. It may start well with a manufacturer's minimal or full calibration. The flowmeter should, therefore, have a known performance under reference conditions. Let us assume that this is $\pm 1\%$ uncertainty with 95% confidence, a reasonable and not uncommon value. What happens when we install this flowmeter on site?

In many cases, there may not be an adequate upstream straight length to retain the calibration, and this may be further degraded by misalignment, etc. Let us assume that this will cause a total uncertainty of $\pm 2\%$. The meter then starts its working life, and we cease to have any satisfactory means of predicting the likely drift that results. Recent experience from the water industry suggests that this may range from 5% to as much as 50%. Because of this uncertainty, it is necessary to determine

a documentation, maintenance, and recalibration schedule for each meter. One way to decide the period between recalibration is by experience. If, after the first recalibration, the shift is unacceptably large, then the next period to recalibration should be half the length. If the calibration is essentially unchanged, then the period may be increased. Descriptions such as “initial period,” “essentially unchanged,” and “unacceptably large” are rather subjective, but they may be given a value based on lost revenue against costs of maintenance and calibration. The outcome of all this is the need to recalibrate, and one choice is in situ calibration with a likely uncertainty of, say, $\pm 3\%$, considerably poorer than originally obtained on a test stand.

In deciding between the merits of test stand and in situ calibration, for a specific case, the reader should develop a table with pros and cons in terms of ultimate uncertainty and total costs (including removal, transport, calibration, lost revenue, and reinstallation). Other factors such as damage to the meter should also be allowed for in the decision.

Methods of in situ calibration include

- ❑ provers (sphere, piston, and tank);
- ❑ other meters (transfer standards);
- ❑ pipework features (inlets and bends);
- ❑ drop test;
- ❑ in situ meter measurement and inspection (dry calibration);
- ❑ clamp-on ultrasonic flowmeters (see Chapter 13); and
- ❑ probes (see Chapter 18) and tracers.

Traceability for the provers and transfer standards will be through a calibration facility. For other methods, traceability will be more tenuous, and in situ verification will be a more appropriate term.

Provers

Provers have already been covered but are mentioned here for completeness. The sphere prover (Figure 4.10) is widely used in permanent systems with banks of turbine meters in the oil industry. The piston prover can be trailer mounted. Claims of order 0.1% or better are sometimes made for these devices (cf. Eide 1991 who claimed $\pm 0.03\%$). The volumetric tank prover (Figure 4.9) discussed earlier is reckoned capable of the same sort of accuracy level.

Other Meters

An example of a transfer standard used in a survey of domestic water consumption in Scotland (Harrison and Williamson 1985) is shown in Figure 4.19 and was capable of calibrating meters from 0.004 to 35.7 m³/h. This was achieved by using three meters within the transfer standard:

- ❑ an Avery-Hardoll PD meter in series with a Kent 50-mm Helix 3000 flowmeter – range 35.7–3.7 m³/h,
- ❑ a Kent 50-mm Master 2000 flowmeter in series with a Kent 50-mm Helix 3000 flowmeter – range 10.8–0.7 m³/h, and
- ❑ two Kent 12-mm PSM meters in series – range 2.7–0.004 m³/h.

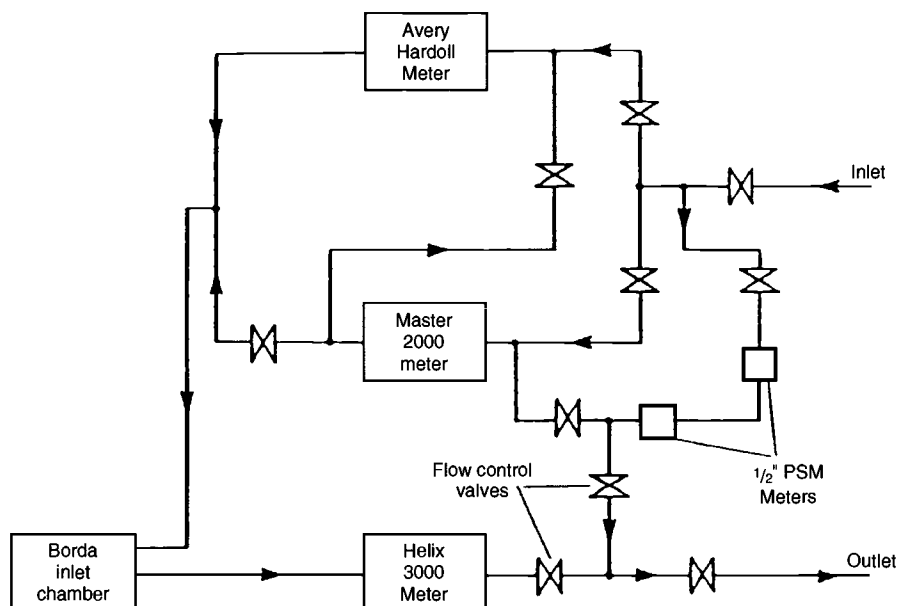


Figure 4.19. Wide flow range reference meter transfer standard (Harrison and Williamson 1985 reproduced with permission of NEL).

The Borda inlet chamber allowed the flow to settle prior to the Helix meter, which is sensitive to installation.

It is obviously necessary for provers, such as this one, to be coupled into the site pipework in series with the meter to be tested.

Pipework Features – Inlets, Bends/Elbows

The inlet from a large tank to a pipe may be used to obtain a measure of the flow rate. The pressure difference between the inside and the outside as flow goes round a bend or elbow may be used to obtain flow rate. The accuracy of these methods will depend on how well-formed the pipework features are. The equations and methods are described in Chapter 8.

Drop Test

In the drop test method, the volumetric flow through the meter is compared with the volume change due to a level change in a reservoir (Figure 4.20). To achieve this, we require the following:

- A suitable system with the pipework leading from the reservoir so as to allow closure of all flows except that which leads from the reservoir to the meter under test. There must be confidence that the water leaving the reservoir all passes through the test meter. There must also be a valve suitable for controlling the flow through the meter.
- Strapping tables, which give the relationship between water level and the volume of water in the reservoir.
- A means to obtain level of water in the reservoir.

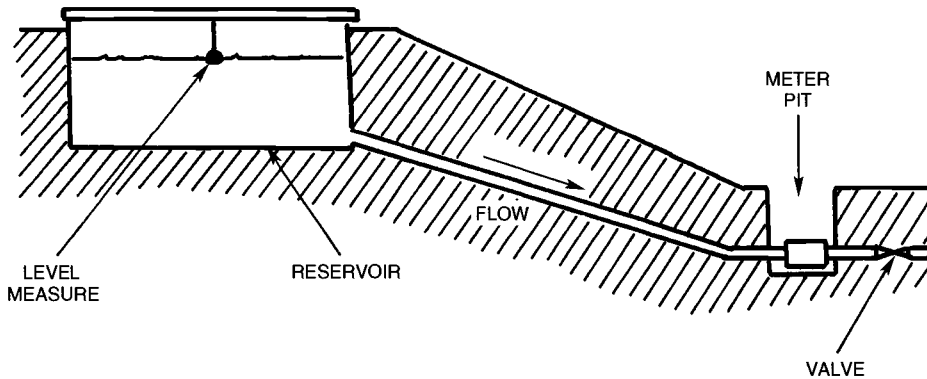


Figure 4.20. Diagram of a suitable system for a drop test.

- d. Satisfactory means of communication between the person who is taking the level measurements and the person who is taking the volumetric readings on the meter. Of course, this process may be automated.

Typical uncertainty for such a test may be

Strapping tables (say 1:10,000 dimensional precision)	0.02%
Level measurement (say 2 mm in 10 cm)	2.0%
Timing errors for rate (say 0.2 s in 100 s)	0.2%

Within this group of errors, although the volume of the reservoir may be affected by tank supports, etc., the level is likely to be predominant. The measurement of level, often in places with poor accessibility, small changes, and surface waves, may result in such an error, and the preceding example may be overly optimistic for some sites. Taking the root of the sum of the squares of these errors gives 2.01%, which may be optimistic.

In Situ Measurement of Meter Dimensions (Dry Calibration)

By measuring the dimensions of a differential pressure flowmeter, it may be possible to deduce the performance. If the measurements confirm that the design meets the ISO requirements, then the flow rate should be predictable with an uncertainty of about 1.0–2.5%.

The term *dry calibration* was particularly coined for the electromagnetic flowmeter where it is possible to measure the magnetic field strength and the meter bore and so deduce the meter performance (Al-Rabeh and Baker 1986). Although the methods have increased in sophistication, it has been found to be not a very satisfactory approach except under very carefully controlled conditions. However, an uncertainty of order 5% may be achievable. This will, of course, require checking all secondary instrumentation as well as measurements of the meter geometry.

The use of retrofitted ultrasonic transducers also depends on the measurement of pipe dimensions and the positions of the transducers. There is not enough experience on whether this may offer genuine calibration in an in situ mode.

Clamp-on Ultrasonic Flowmeters

The possibility of using clamp-on transit-time ultrasonic meters is being explored. Apart from the inherent uncertainties of using such meters on pipework of unknown condition, it is also essential to avoid installations that are seriously affected by upstream pipe fittings. These meters are discussed in more detail in Chapter 13.

Probes

Probes will be covered in Chapter 18. However, the use of probes for in situ calibration is common, and a few points need to be made here.

- If a single-point measurement is required, the reader is referred to Table 2.2 where it is shown that, for a well-developed turbulent profile, if the probe is placed near the three-quarter radius position ($0.76 \times$ radius from the pipe axis), the velocity at that point is approximately equal to the mean velocity in the pipe. However, this is a region of strong shear and turbulence, and incorrect positioning of the probe may result in measurement errors. If the probe is positioned on the axis of the pipe, the actual positioning is less critical, but the velocity at that point is not proportional to the mean velocity in the pipe as the Reynolds number changes. Allowing for a probe calibration of $\pm 1\%$ uncertainty, the best that is likely to be achieved for the total flow is of order 3% and may be much poorer than this.
- If the velocity profile across the pipe is required, then multiple measurements will be necessary. These can be equally spaced, but to obtain mean velocity in the pipe it may be quicker and preferable to use an integration method that selects fewer positions, which may or may not be equally weighted (cf. ISO 1977b, 1988 and also list of standards). The log linear positioning of measurements (with each measurement equally weighted) follows:

Number of points radially	Position, r/R		
3	0.359	0.730	0.936
4	0.310	0.632	0.766
	0.958		
5	0.278	0.566	0.695
	0.847	0.962	
6	0.253	0.517	0.635
	0.773	0.850	0.973

Salami (1971) recommended that at least six traverse points on each of six radii were needed to keep errors down to 0.5% for nonaxisymmetric profiles. If we assume that the profile is axisymmetric, then we may take one diameter and select the number of points according to the time available. The specification of the positions, the recording of the data from the probe, and the calculation of the mean flow in the pipe are often done today by a portable flow computer designed for the purpose. We shall also need a value of the pipe internal diameter (ID). For further information, the reader is referred to the standards. Realistically we are unlikely to achieve an uncertainty of better than 3%.

- The act of inserting a probe will alter the flow in the pipe, and the probe's response will vary with insertion position. This is mentioned in Chapter 18, but the manufacturer of the probe will need to supply detailed information.

Tracers

Tracer methods for obtaining flow in an unknown system are an elegant and attractive approach, but they have a disadvantage. Those undertaking them really need extensive experience; otherwise, the resulting accuracy may not be very high. The method can conveniently be categorized as constant rate injection, sudden injection, or time of transit.

In the constant rate method, a solution of the tracer material in the same fluid as that in the main stream is injected from a branch pipe, which has a valve, at a known rate q_{vi} to the line (Figure 4.21). The concentration in the injected stream C_i is known; the concentration in the main stream, if the tracer material happens to be present, is measured as C_u ; and, far enough downstream, the concentration resulting from the injection and thorough mixing C_d is also measured. The equation that gives conservation of tracer material and hence the volumetric flow rate in the line q_v is

$$C_u q_v + C_i q_{vi} = C_d (q_v + q_{vi}) \quad (4.1)$$

and rewriting this to obtain the unknown flow rate in the line we have

$$q_v = q_{vi} \frac{C_i - C_d}{C_d - C_u} \quad (4.2)$$

For this method to be successful, we do not need to know the size of pipe, but we do need to ensure that all the tracer material goes downstream, and that there is a genuine steady state, without accumulation of tracer in trap areas in the pipework. It is probably necessary for the upstream concentration C_u to be less than 15% of the downstream concentration after injection C_d . Clearly the tracer must be such as can be injected into the line without any health or environmental problems, and it must be cheap enough to use in continuous injection. If we make the assumption, to obtain an idea of the accuracy, that $C_i \gg C_d \gg C_u$, then with a high quality flowmeter for the injection and a precise concentration in the injected flow, the downstream value C_d is likely to be the least reliable of the dominant measurements and will affect the accuracy. Hermant (1962) in the discussion of his paper appears to claim $\pm 1\%$ as the uncertainty compared with use of current meters. Clayton et al. (1962a, b), using a radioactive tracer, were able to claim better than 1% for the method including using a portable system. They suggested that 110–220D should be allowed for mixing.

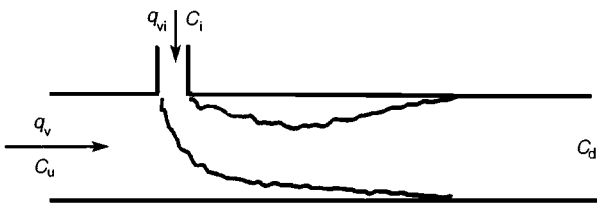


Figure 4.21. Constant rate injection method (dilution method).

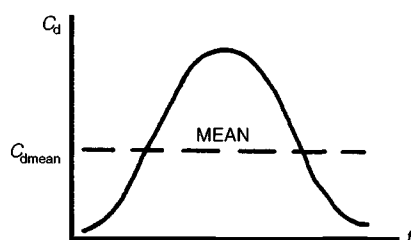


Figure 4.22. Concentration variation downstream with sudden injection method (integration).

In the sudden injection method (integration), one known amount of V is injected with a known concentration C_i , and the variation in the downstream concentration is measured with time as in Figure 4.22. If this curve is integrated, and we assume that the flow rate is constant during the measurement period, we obtain the total of tracer passing, or the mean over the measuring period. Thus

$$q_v = \frac{VC_i}{\int_0^t (C_d - C_u) dt} \quad (4.3)$$

$$= \frac{VC_i}{t(C_{d\text{mean}} - C_u)} \quad (4.4)$$

where $C_{d\text{mean}}$ is the mean concentration measured downstream. In addition to the constraints for the constant rate method, the accuracy of sampling the concentration is likely to be lower because instantaneous measurements will be needed and the accuracy of the concentration sensor will introduce uncertainty. However, the amount of tracer used will be less, which may, therefore, allow a wider choice of tracer. (cf. Spragg and Seatonberry 1975).

In addition to these two methods, it may be possible to sense the passing of tracer at two points in the line, or, indeed, the variation in some naturally occurring component that might allow a correlation method to be used. It should be remembered that, in this case, the measure will be of velocity and not necessarily of volumetric flow (cf. Aston and Evans 1975 who estimated likely uncertainties within $\pm 1\%$ for steady flows and $\pm 2\%$ for normally fluctuating flows).

Scott (1982) also described the Allen salt-velocity method, which measures the time of transit between two stations between which the volume of the pipe is known. Again he mentions the need to inject the tracer explosively, in this case a minimal amount of brine, and the requirement for a sensitive detection system. The marker point as the brine passed each station was taken as the midpoint of the half height trace [Figure 4.23 (cf. Hooper 1962 and Clayton et al. 1962b who suggested uncertainties of order 1%)].

Various tracers have been used. For the constant rate method, sample analysis is likely to be the most accurate method of obtaining C_u and C_d , the values of concentration. In the case of sudden injection, a continuous detector will be needed. Examples follow:

Main fluid	Tracer	Detector
Water	Brine	Electrical conductivity
Other liquids	Dyes or chemicals	Color or light sensing
Liquids or gases	Temperature	Temperature sensor
	Radioactivity	Scintillation or geiger counter
Gases	CO ₂ , N ₂ O, He, methane	Infrared spectrometer

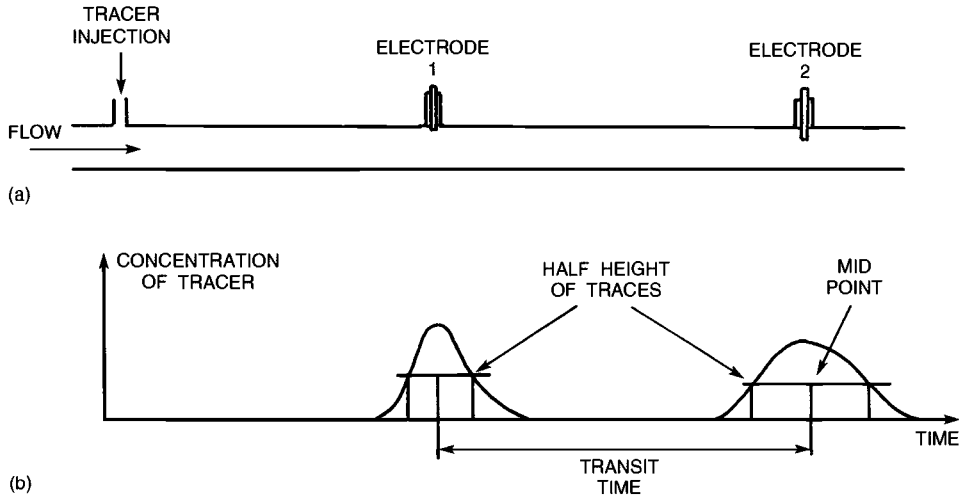


Figure 4.23. The Allen salt-velocity method for time of transit between two stations: **(a)** Pipe for which volume is known; **(b)** Marker points as the brine passes each station (after Scott 1982).

4.7 CALIBRATION UNCERTAINTY

Mattingly (1990/1) gives a very useful run down of calibration uncertainty. Before the facility is built, the uncertainty can be estimated. For a static gravimetric liquid facility,

$$q_v = \frac{M_n}{\rho t} \quad (4.5)$$

where q_v is the volumetric flow rate, M_n is the net mass of liquid collected, ρ is the liquid density, and t is the collection time. The uncertainties (cf. Equation 1.A.7 for alternative symbols) can be combined as

$$\frac{\Delta q_v}{q_v} \leq \left\{ \left(\frac{\Delta M_n}{M_n} \right)^2 + \left(\frac{\Delta \rho}{\rho} \right)^2 + \left(\frac{\Delta t}{t} \right)^2 \right\}^{1/2} \quad (4.6)$$

or

$$\frac{\Delta q_v}{q_v} \leq \left| \frac{\Delta M_n}{M_n} \right| + \left| \frac{\Delta \rho}{\rho} \right| + \left| \frac{\Delta t}{t} \right| \quad (4.7)$$

By inserting percentages of reading, an initial estimate can be obtained. This presupposes that Equation (4.5) is correct. However, other factors are likely to be involved. At the National Institute of Standards and Technology (NIST), Gaithersburg, Maryland, for liquid flow, the three values to 3 standard deviations were

$$\frac{\Delta M_n}{M_n} = 0.02\% \quad \frac{\Delta \rho}{\rho} = 0.02\% \quad \frac{\Delta t}{t} = 0.01\%$$

and combine to $\pm 0.03\%$ for Equation (4.6) and $\pm 0.05\%$ for Equation (4.7). For gas, the volume was 0.04% , density pressure effects were 0.13% , temperature effects were 0.05% , and collection time for the device was 0.01% and for switching was 0.02% , giving totals of $\pm 0.15\%$ and $\pm 0.25\%$, respectively. The next stage is to obtain data

from the facility, and the third stage is to obtain systematic errors from round-robin tests. These round-robin tests are known as Measurement Assurance Programs (MAPs). The systematic error is either root-sum-squared with the random error or, preferably, obtained by straight addition.

A satisfactory transfer standard is formed by using two meters, often turbines, with a flow conditioner placed between them in a tandem configuration (Figure 4.17), as already mentioned for laboratory flow facilities. Adequacy of the data is established by specifying the number of repeat calibrations done for each flow rate and meter configuration. These will ensure statistical significance. The Youden (1959) procedure is recommended (Chapter 1).

4.8 TRACEABILITY AND ACCURACY OF CALIBRATION FACILITIES

For a calibration to be acceptable, the ultimate source of the measurement must be known, and the calibration must be traceable to that standard, as in Figure 4.4. Laboratories holding national standards exist in several countries (Benard 1988), and the address of the nearest should be available from national government information or from laboratories such as:

Country	Laboratory
United Kingdom	National Engineering Laboratory, East Kilbride, Scotland
United States of America	National Institute of Standards and Technology, Gaithersburg, MD

National standards should themselves be traceable back to more fundamental measures of mass, time, and length. As a result of this, a traceability chain is formed. Each link is formed from a facility, or a flowmeter calibrated on a facility or against a flowmeter of lower uncertainty. The chain needs to be rechecked with sufficient frequency to ensure continuing confidence.

It should also be clear that, if a transfer standard with an uncertainty of, say, $\pm 0.25\%$ is used to calibrate a flowmeter, the uncertainty in the flow rate measurement made by that flowmeter will be greater than $\pm 0.25\%$ due to additional random errors in the flowmeter.

Harrison (1978b) clarified the meaning of traceability with the following three points:

- a. Each standard used for calibration purposes has itself been calibrated against a standard of higher quality up to the level at which the higher-quality instrument is the accepted national standard. This is usually a unique item held in a national standards laboratory, but it could, in some cases, be a local standard of equivalent quality built and operated to a national specification and confirmed as operating to that specification.
- b. The frequency of such calibration, which is dependent on the type, quality, stability, use, and environment of the lower-quality standard, is sufficient to

establish reasonable confidence that its value will not move outside the limits of its specification between successive calibrations.

- c. The calibration of any instrument against a standard is valid in exact terms only at the time of calibration and its performance thereafter must be inferred from a knowledge of the factors mentioned in b.

The standards for mass and time are national, derived from equivalent international standards. The international one for mass is in Paris and for time is based on a fundamental frequency, which is in turn broadcast nationally.

Thus calibration uncertainties achievable at present appear to be: for liquids about 0.1% and for gases 0.2–0.3% or possibly better. When total mass or volume, rather than flow rate, is required, these values may be improved. On the other hand, in situ calibration is likely to be at best 2% and often 5% or lower and should be seen as a last resort in most cases. The exception is where a meter installation is equipped with off-takes so that a high quality transfer standard prover or meter can be coupled in series and will result in an accuracy approaching that of a dedicated facility (cf. Johnson et al. 1989 for further useful discussion of component uncertainties for a gas facility).

4.9 CHAPTER CONCLUSIONS

For most readers, the essential value of this chapter will be knowledge of calibration methods and accuracy levels, as well as signposts pointing to where to obtain further detailed information.

For those in the business of designing, installing, and commissioning flow calibration facilities, there are some fundamental questions to be addressed, not all of which may, yet, be answerable. This chapter has attempted to flag some of these questions and to point to relevant literature. Some of these questions concern:

- ❑ limits of accuracy for a meter in a turbulent flow;
- ❑ limits of accuracy for a test stand;
- ❑ the nature of flow diversion and its repeatability;
- ❑ the design of provers and the interpolation of pulse trains;
- ❑ ultimate in situ accuracy including
 - more user-friendly tracer methods,
 - more experience with retrofitted ultrasonic meters,
 - improved confidence in clamp-on ultrasonic meters,
 - improved probe measurements;
- ❑ theoretical optimum accuracy as traced from national standards and round-robin meter exchange.

Because, therefore, existing methods may not meet all needs (Paton 1988), new developments continue to appear. One relatively recent development, which is now a standard instrument, is the piston prover. Another development that appeared a few years ago is the gyroscopic load cell, which has been applied to the weighing of

collection tanks in some calibration laboratories (cf. Bignell's 1996b proposal for a positive displacement gas flow standard).

For low gas flows (0.05–5 ml/s), Robinson et al. (1986) suggested

- ❑ laminar flowmeters,
- ❑ bubble flow calibration (see Section 4.4.6), and
- ❑ critical flow venturi nozzle

and achieved agreement of within 1% between laminar and bubble methods. However, the bubble method, too, has its potential errors due to detergent build-up.

Finally, we may question whether there are other factors resulting from environment, impurities, or other components in the fluid that may affect the calibration of certain flowmeter types or the behavior of the test stand.

Orifice Plate Meters

5.1 INTRODUCTION

The orifice plate flowmeter, the most common of the differential pressure (DP) flowmeter family, is also the most common industrial flowmeter. It is apparently simple to construct, being made of a metal plate with an orifice that is inserted between flanges with pressure tapings formed in the wall of the pipe. It has a great weight of experience to confirm its operation. However, it is far more difficult to construct than appears at first sight, and the flow through the instrument is complex.

Some key features of the geometry of the orifice and of the flow through it are shown in Figure 5.1. The behavior of the orifice plate may be predicted, but the predictions derive from experimental observation and data. The inlet flow will usually be turbulent and will approach the orifice plate where an upstream pressure tapping (one diameter before the orifice plate) will measure the pressure at the wall, that is in this case the static pressure. (Flange and corner tapings will be discussed later). The pressure across the pipe will have a constant time-mean value because the only velocity across the pipe is due to turbulent eddies. The flow close to the orifice plate will converge toward the orifice hole, possibly causing a recirculation vortex around the outside corner of the wall and orifice plate. The inward momentum of the flow at the orifice hole will continue downstream of the hole, so that the submerged jet coming out of the hole will reduce to a smaller cross-section than the hole, which is known as the vena contracta, the narrowest point of the submerged jet. Outside this submerged jet is another larger recirculation zone, and the downstream pressure tapping (the one after the orifice) set in the wall or in the corner by the orifice plate senses the pressure in the vena contracta across this recirculation zone. Downstream of this point, diffusion takes place with considerable total pressure loss.

The data on which the orifice predictions are based may be presented in three ways (cf. Miller 1996):

1. The most accurate method is to use a discharge coefficient-Reynolds number curve for the required geometry, which includes all dimensional effects and other influences.
2. To reduce the number of curves, a datum curve is used in conjunction with correction factor curves. This was essentially the procedure adopted for the British Standard 1042: Part 1: 1964.
3. For convenience with the advent of modern flow computers, the data are reduced to a best fit equation. This is essentially the procedure with the most recent

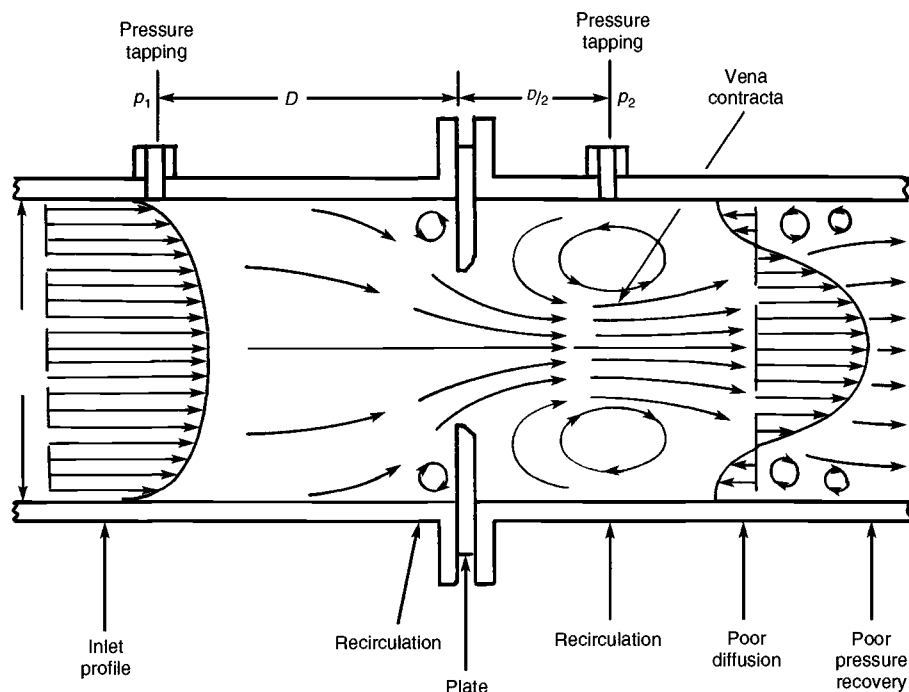


Figure 5.1. Diagram to show geometry and flow patterns in the orifice plate flowmeter.

versions of the International Standard, and its form will be dealt with in the following pages.

The most common orifice plate is a metal disk spanning the pipe with a precisely machined hole in the center of the plate; it is usually mounted between flanges on the abutting pipes, with pressure tapings fitted in precisely defined positions and to precise finishes. The differential pressure is measured by manometer, Bourdon tube, or a pressure transducer, and the flow is deduced from the equations and probably computed using a flow computer.

The importance of the orifice is its simplicity and predictability, but to achieve high accuracy it is essential that the detailed design of the meter is the same as that from which the original data were obtained, and that the flow profile entering the meter is also the same. To ensure that the details are correct, the national and international standards lay down the precise requirements for constructing, installing, and operating the orifice meter. It must be stressed that departing from the standard requirements removes the predictability and prevents the standard from being used to obtain the flow prediction.

In this chapter, I have sought to avoid duplicating the standards or the very thorough presentation of Miller (1996) and instead have attempted to present recent published information on the performance of the meter. It is, therefore, essential that the reader have a copy of the relevant standards from which to work and, in particular, ISO 5167-1 (BS EN ISO 5167-1). The reader is also strongly advised to seek out a copy of Miller (1996). Access to Bean (1971), Spink (1978), Danen (1985), and Spitzer (1991) may also be useful because all have extremely valuable information

for those concerned with the design, installation, and maintenance of orifice plate meters.

5.2 ESSENTIAL BACKGROUND EQUATIONS

Mass Flow Rate Equation

The mass flow rate is related to the differential pressure by the equation (c.f. Equation 2.11 for flow without loss)

$$q_m = \frac{CE\epsilon\pi d^2 \sqrt{2\rho_1 \Delta p}}{4} \quad (5.1)$$

where C is the coefficient of discharge, E is the velocity of approach factor $(1 - \beta^4)^{-1/2}$ where β is the diameter ratio d/D of orifice diameter to pipe internal diameter, ϵ is the expansibility (or expansion) factor, Δp is the differential pressure, and ρ_1 is the density at the upstream pressure tapping cross-section. The value of the volumetric flow can be obtained from this using the relationship:

$$q_v = q_m / \rho \quad (5.2)$$

where ρ is the density of the fluid at the appropriate conditions of pressure and temperature.

Coefficient of Discharge

A simple expression for the discharge coefficient is

$$C = C_\infty + \frac{C_{Re}}{Re^n} \quad (5.3)$$

where C_∞ is the coefficient for infinite Reynolds number, C_{Re} is a constant for a particular installation, Re is the Reynolds number based on the pipe internal diameter, and n is the index to which this is raised. The relative simplicity of Equation 5.3 is not matched by the most commonly used expressions for C , C_∞ or C_{Re} either in their complexity or variety.

The development of the equation commonly referred to as the API or AGA equation can be found by referring to Bean (1971) and Spitzer (1991). Reader-Harris and Sattary (1990, cf. Reader-Harris 1989) reported that a joint meeting of European Community and American Petroleum Institute (API) flow measurement experts in New Orleans in November 1988 unanimously accepted a new equation. API 2530 was revised to include the Reader-Harris/Gallagher equation. This allowed for an additional term for small pipes. Additional data have now allowed an adjustment to the equation for small orifice diameters. This was reported by Reader-Harris et al. (1995).

The previous coefficient of discharge adopted by the ISO and British Standards Institution was given by an equation due to Stolz (1978, 1988). This equation was retained in the standard until 1998. For details of the thorough work that has taken place in the last 10 years or so, the reader is referred to Reader-Harris et al. (1995) and Spencer (1993) (cf. Fling and Whetstone 1985).

The result is that current American practice appears to be to use API Chapter 14 Section 3 (14.3) and a new expression for the discharge coefficient C has been adopted by ISO, the Reader-Harris/Gallagher equation. However, the reader is advised to refer to the document most appropriate to the application.

The form of these new equations is

$$C = C_{\infty} + \frac{C_{Re}}{Re^n} + C_{Taps} + C_{Small\ orifice} \quad (5.4)$$

where all coefficients are functions of β . C_{Taps} includes values of $L_1 = l_1/D$ and l_1 is the distance of the upstream tapping from the upstream face of the plate, and $L'_2 = l'_2/D$ and l'_2 is the distance of the downstream tapping from the downstream face of the plate. The prime signifies that the measurement is from the downstream and not the upstream face of the plate. C_{Re} reflects the slope of the characteristic due to change of Re .

I have decided to include the provisional form of the equation in Appendix 5.A. However, it is essential that the reader refers to the standard document to check the latest version adopted in the United States, United Kingdom, or elsewhere and to obtain all the conditions necessary for its valid use. These include ranges for parameters, detailed design, pipe smoothness, etc.

The data for the new equation appear to have been obtained for orifice plates with diameter ratios 0.1–0.75, throat Reynolds numbers from 1700 to 5×10^7 , and pipe diameters from 50 to 250 mm. Data points for orifice diameters less than 12.5 mm were very scattered and were not included.

The uncertainty is in the range 0.5–0.75% for $D \geq 71.12$ mm, and the precise value depends on β . For small orifices, an additional uncertainty is combined arithmetically and can be of order 0.4% for $D = 25.4$ mm (1 in.).

Miller (1996) gave the proposed Stolz II and NEL/TC 28 equations and tabulated the values for these compared with ISO and ANSI/API. Perusal of the variation across all equations is quite instructive. A difference between the new equations of at least 0.15% is quite common, with differences occasionally as high as 1% for $Re = 5,000$. Comparison of both old and both new gives a spread of discharge coefficients of less than 0.1% to nearly 2%. The best agreement appears to be for $\beta = 0.5$ and $Re > 10,000$, where variation across all four equations is less than 0.1% to about 0.5%. One benefit, therefore, of the extensive work on refining equations has been to increase our confidence in existing values of the discharge coefficient and in indicating where the main divergences from these occur. Spitzer, also, makes the point that the actual maximum variation in discharge coefficient obtained from the main equations for a range of Reynolds numbers from 20,000 to 2,000,000 is only of order 0.12%. For a $\beta = 0.5$, $Re = 10^6$, and corner tapings, I estimate that the value of C for the latest version of ISO is about 0.13% greater than for Stolz, which generally confirms Spitzer's comments.

Expansibility Factor

The expansibility (expansion) factor provides an adjustment factor to allow differential pressure devices to be calibrated on water, an essentially incompressible fluid for these purposes, for use on compressible gas. It, essentially, provides an adjustment factor to the coefficient of discharge that allows for the compressibility of the gas.

Reader-Harris (1998) provides a review of the past development of this factor and the current recommended equation for it. It is found to be virtually independent of Reynolds number. For many years, it was given by an equation of the form:

$$\epsilon_1 = 1 - (a_\epsilon + b_\epsilon \beta^4) \frac{\Delta p}{\kappa p_1} \quad (5.5)$$

where a_ϵ and b_ϵ are constant coefficients, κ is the isentropic exponent, which for an ideal gas is equal to γ , the ratio of specific heats, and if κ is not known γ should be used. p_1 is the pressure at the upstream tapping. However, according to Kinghorn (1986) the coefficients in common use in Equation (5.5) were probably in error.

In the most recent version of the standard, a new equation for ϵ_1 is given but may be due for a further update. It is of the form

$$\epsilon_1 = 1 - (a_\epsilon + b_\epsilon \beta^4 + c_\epsilon \beta^8) \left\{ 1 - \left(\frac{p_2}{p_1} \right)^{1/\kappa} \right\} \quad (5.6)$$

provided that $p_2/p_1 \geq 0.75$. c_ϵ is another constant coefficient. The final bracket may be approximated by $\Delta p/\kappa p_1$ if p_2/p_1 is very close to unity, but the work of Reader-Harris (1998) suggests that the expression in the final bracket is preferable. He gave the following equation, following careful analysis of the data, which may now be in the latest ISO standard:

$$\epsilon_1 = 1 - (0.351 + 0.256\beta^4 + 0.93\beta^8) \left\{ 1 - \left(\frac{p_2}{p_1} \right)^{1/\kappa} \right\} \quad (5.7)$$

He gave a relative uncertainty for ϵ_1 of $3.5\Delta p/\kappa p_1$ %, which differs slightly in detail from that in ISO. He also gave a very useful theoretical derivation and obtained an equation that differs from Equation (5.7) by about the uncertainty in Equation (5.7).

Pressure Loss

The expression in the ISO standard for the pressure loss across the orifice plate now appears to be

$$\text{Pressure loss} \approx \frac{\sqrt{1 - \beta^4(1 - C^2)} - C\beta^2}{\sqrt{1 - \beta^4(1 - C^2)} + C\beta^2} \Delta p \quad (5.8)$$

Taking a typical value of C of about 0.6, this results in a pressure loss for a plate with $\beta = 0.5$ (and $E = 1.033$) of about $0.73\Delta p$ [cf. Uner 1997 who suggested that the previous ISO equation (for nozzles and orifice plates) gives anomalous results if used outside the limits of applicability of the ISO standard].

An alternative for orifice plates allowed by a recent version of the standard is

$$\text{Pressure loss} \approx (1 - \beta^{1.9}) \Delta p \quad (5.9)$$

which, for the preceding case, also gives $0.73\Delta p$.

The most recent version appears to have provided a loss coefficient K given by

$$K = \frac{\text{Pressure loss}}{\frac{1}{2}\rho_1 V^2} \quad (5.10)$$

where

$$K = \left(\frac{\sqrt{1 - \beta^4(1 - C^2)}}{C\beta^2} - 1 \right)^2 \quad (5.11)$$

One of the most welcome changes in the latest version of the standard is the elimination of the symbol α , the flow coefficient, which was of little value. It has been replaced by CE , to which it was equal, or other equivalent expression.

5.3 DESIGN DETAILS

Design details are set out fully in the ISO 5167 standard. It is important that the detailed requirements are followed so that performance can be predicted, but also so that new data can be added to existing data, to increase our overall knowledge. In this book, some of the requirements will be touched on, but the standard should be referred to for full details.

The design is, of necessity, an iterative procedure starting from the known requirements of, typically, design flow rate, pipe size, and differential pressure. Danen (1985) gives some helpful calculation flowcharts for each type of differential meter. The first educated guess will relate to the value of β . With this and the selected values, a new value of β is found via the flow rate equation. Iteration continues until a satisfactory value of β is achieved. However, it is likely that most users will have access to a computer program for this procedure (either from a manufacturer or in their own software library) that will enable the optimum size of plate to be obtained. These will ensure a balance between too great a pressure drop through the plate and too small a differential pressure for measurement. It is often suggested (Miller 1996) that the pressure drop across the orifice plate should be of the order of 100 in. of water (2.5 m or 25 kPa) and that the beta ratio should be close to 0.5. In some cases, this may not be achievable. The pressure loss can never exceed the difference between supply and demand pressures, and the minimum pressure should be kept above the liquid's vapor pressure to avoid cavitation. Figure 5.2, which has been suggested in various forms by various people, illustrates the compromise decision that needs to be made between these factors.

For wide-range metering, Miller suggests the use of two transmitters ranged for 180 in. of water (4.5 m and 45 kPa) and 20 in. of water (0.5 m and 5 kPa).

- a. *The plate.* Some details of the plate geometry are shown in Figure 5.3 for a pipe of diameter D and with an orifice diameter of d . The plate must be flat with parallel faces, and concentric. Flatness is defined in the standards. The upstream face is the most critical because this has a strong effect on the flow entering the orifice. In addition to the flatness, the plate must have a roughness of less than $10^{-4}d$. It is useful to indicate the correct flow direction on the plate in a position that can be seen when the plate is installed. The plate may have to be rather thick to withstand the forces due to the flow, and in this case the downstream edge of the orifice is beveled at an angle of $45^\circ \pm 15^\circ$, so that the final thickness of the

actual orifice opening is between $0.005D$ and $0.02D$, and constant to within $0.001D$. All edges of the orifice must be clear of burrs, etc., and the upstream edge must be sharp, defined as of radius less than $0.0004d$. If d is less than 25 mm (cf. 125 mm in the previous version of the standard), it is necessary to measure the radius to ensure that this condition is met. Sun et al. (1996) gave an equation for change in discharge coefficient with edge radius

$$\frac{\Delta C}{C} \% = 0.85 \ln(10^3 r/d) + 1.74 \tag{5.12}$$

for $0.0002 < r/d < 0.0035$. This appears to give a rather larger change than most of the data in BS 1042: Section 1.5: 1997, although some data for $D=150$ mm lines indicate changes greater than Equation (5.12). If these are valid, then even for $r/d < 0.0004$, $\Delta C/C$ could be nearly 1% high. (Sun et al. also suggested an orifice chamfered at entry and referred to the Russian National Code RD 50-411-83.)

Two methods have been used to measure the edge radius (Hobbs and Humphreys 1990): a stylus that followed the contour and allowed magnifications

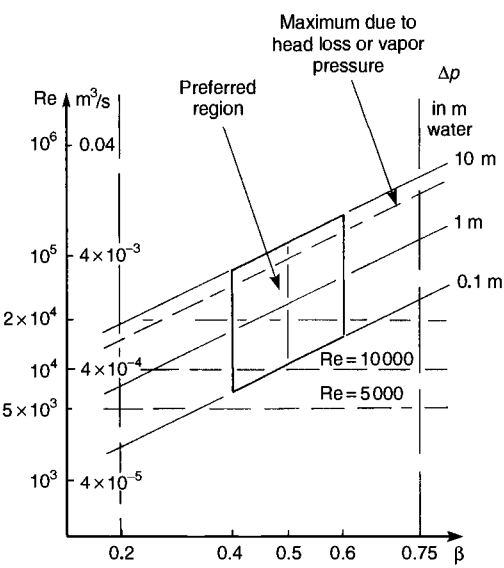


Figure 5.2. Compromise decision for sizing an orifice plate for water with D and $D/2$ tappings with a line size of $D = 50$ mm.

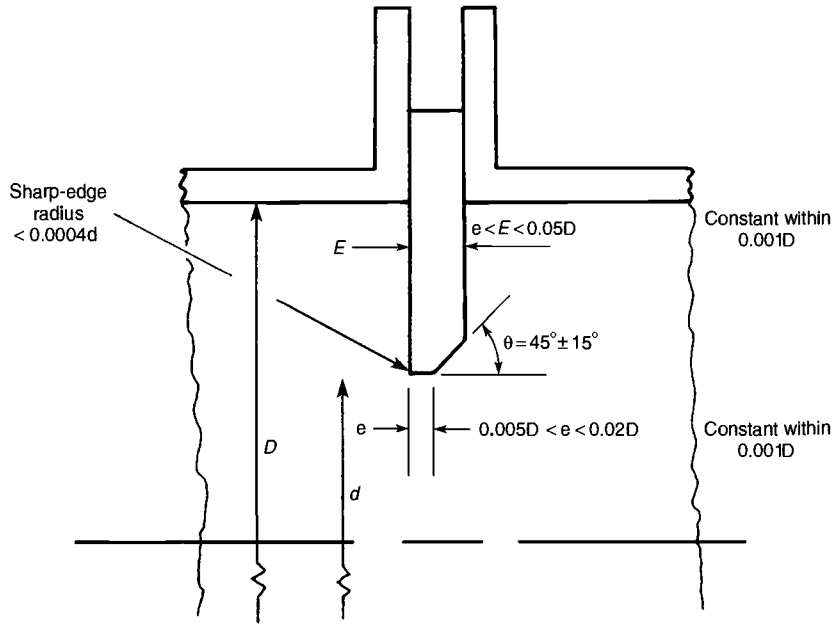


Figure 5.3. Plate geometry: E is thickness of the orifice plate, and e is thickness of the orifice.

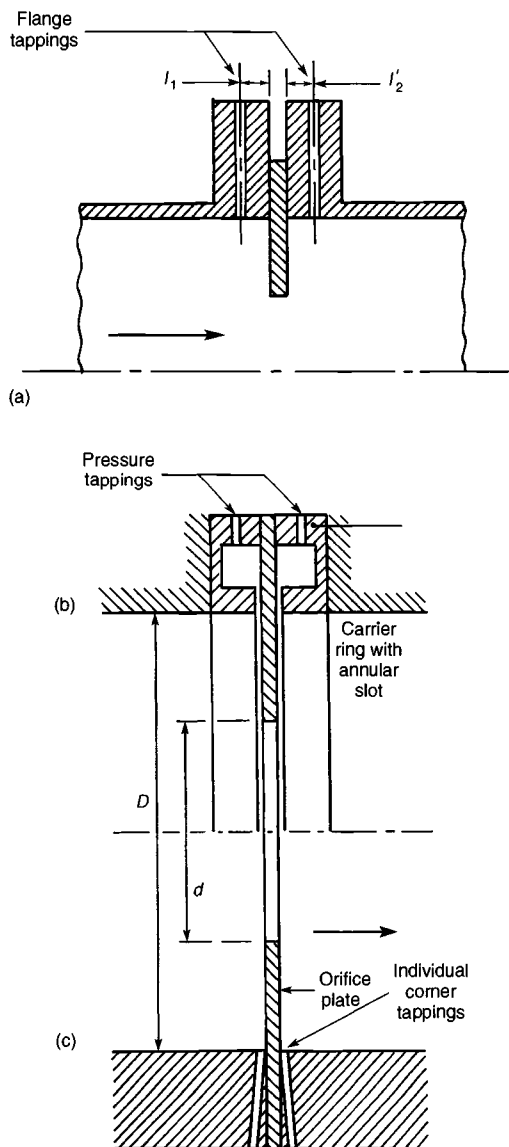


Figure 5.4. Layout of pressure tapings (after ISO 5167-1): **(a)** Flange; **(b)** Corner – ring with annular slot; **(c)** Corner – individual tapings.

of corner tapings reduced the effect of pipe roughness further upstream. Sindt et al. (1989) found up to 1% change in coefficient for $\beta = 0.74$ and $Re = 7 \times 10^6$ between smooth ($2.8 \mu m$) and rough ($8.9 \mu m$), but for $\beta = 0.5$, the change was not measurable.

Constraints are also placed on thermometer pockets near to the flowmeter.

The validity of these values is based on many tests to obtain data. Some recent measurements are compared later.

of up to 500, and a cast around the edge which could be sliced and polished after removal from the edge. Results to 0.005 mm are claimed (cf. Jepson and Chamberlain 1977 who described British Gas's orifice radius inspection system).

- b. *Pressure tapings.* There are three standard methods of sensing the pressure drop across the orifice plate (Figures 5.1 and 5.4): D and $D/2$ tapings (Figure 5.1), flange tapings (Figure 5.4), and corner tapings (Figure 5.4).

Again the standards set out the precision of positioning of the tapings and the details of their manufacture (cf. Zedan and Teyssandier 1990, who concluded that the tolerances that ISO and ANSI/API 2530 allowed for pressure tapping locations were conservative).

5.4 INSTALLATION CONSTRAINTS

The upstream and downstream pipe lengths are set out in some detail in ISO 5167-1. A sample of the information is given in Table 5.1. The values given are claimed to retain the predicted performance of the orifice plate, whereas the values in parentheses increase the measurement uncertainty by 0.5%. For instance, a meter with $\beta = 0.5$ mounted within $14D$ of a 90° bend but with more than $7D$ clear pipe upstream and $3D$ downstream will have an uncertainty in the coefficient of $0.6 + 0.5 = 1.1\%$.

Reader-Harris and Keegans (1986) suggested that $2.5D$ of smooth pipe upstream

Table 5.1. Examples of installation requirements for zero additional uncertainty from the standard. (Bracketed values are $\pm 0.5\%$ additional uncertainty.)

β	Upstream Straight Lengths				
	Single 90° Bend	Two 90° Bends in Perpendicular Planes	Reducer 2D to D Over 1.5D to 3D	Gate Valve Fully Open	Downstream
0.2	10(6)	34(17)	5	12(6)	4(2)
0.3	10(6)	34(17)	5	12(6)	5(2.5)
0.4	14(7)	36(18)	5	12(6)	6(3)
0.5	14(7)	40(20)	6(5)	12(6)	6(3)
0.6	18(9)	48(24)	9(5)	14(7)	7(3.5)
0.7	28(14)	62(31)	14(7)	20(10)	7(3.5)
0.75	36(18)	70(35)	22(11)	24(12)	8(4)

Bends

Some tests have raised questions as to whether the required upstream straight length of pipe is adequate (Branch 1995). Decrease in bend angles does not necessarily reduce the disturbing influence on the meter (Himpe et al. 1994). There does not appear to be an optimum angle between the tapping position and the plane of the bend (Branch 1995), although a considerable influence has been noted by Himpe et al. (1994). Mattingly and Yeh (1991) gave the effect of distorted profile caused by elbows upstream of orifice meters and for the same but with a tube bundle straightener between 3.8D and 5.7D from the bend outlet. A sample of the results is shown in Figure 5.5. They concluded that this flow conditioner may introduce errors. Conditioner/flowmeter combinations should preferably be calibrated as one unit.

Fittings that Induce Swirl

Mattingly and Yeh (1991) also showed that, for Reynolds numbers between 10^4 and 10^5 in a 50-mm water flow facility, the angle of swirl caused by the double elbow out of plane configuration decreased from about 18° to about 6° after 40D and to about 3° after 80D. They questioned the 2° swirl criterion for safe operation of orifice installations (cf. Brennan et al. 1989 and Morrison et al. 1995 whose work may require a knowledge of the level of swirl in a particular pipe installation).

Morrison et al. (1990a) gave measurements that confirmed that larger beta ratio orifice plates are more sensitive to swirl. They noted that expansion of the orifice jet (vena contracta) leaving the plate, resulting from conservation of angular momentum, would be expected to have a greater effect on small beta ratios, contrary to experience. On the other hand, the results of Brennan et al. (1991) appear to be contrary to expectation with increased beta ratios in most cases improving the performance downstream of tees and two elbows in perpendicular planes. In addition, the results show a much better performance by the straighteners than found by Sattary (1991), which is referred to later.

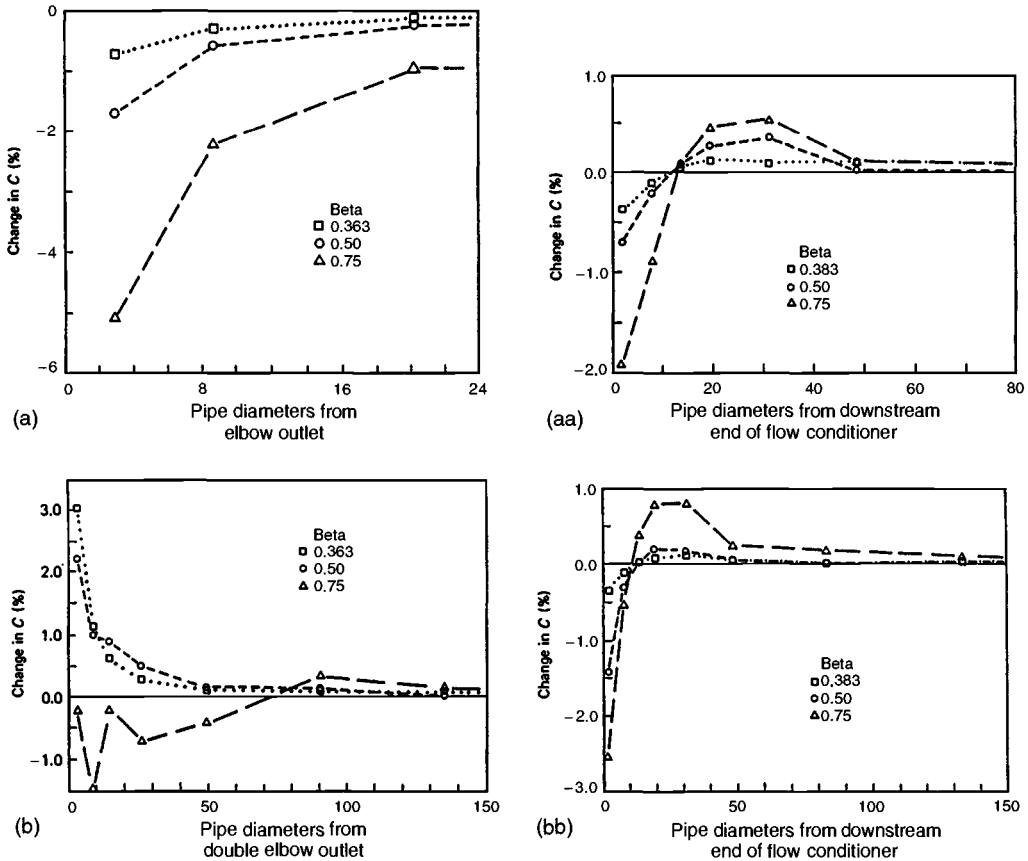


Figure 5.5. Change in orifice coefficient when downstream of (from Mattingly and Yeh 1991 reproduced with permission of Elsevier Science): **(a)** Single elbow; **(b)** Closely coupled double elbows out of plane; **(aa)** Single elbow with a tube bundle straightener; **(bb)** Closely coupled double elbows out of plane with a tube bundle straightener.

Other Profile Distortions

Yeh and Mattingly (1994) measured the profile downstream of a reducer (approximately 3:2), and the effect of the reducer on the coefficient of an orifice plate flow-meter. Their results appeared to confirm the ISO values for $\beta = 0.5$ and less, but not for $\beta = 0.75$ [cf. Morrison et al. 1992 who distorted the inlet profile to an orifice plate by varying the ratio of flow in the outer (wall) area of the pipe to that flowing through the center of the pipe].

Flow Conditioners

McFaddin et al. (1989) found that, for a 100-mm (4-in.) line, a 2.5D long tube bundle flow conditioner (19 × 12 mm tubes) 7D upstream of the orifice with $\beta = 0.75$, the minimum distance specified in the ANSI/API 2530 standard, the error can be as much as 1%.

Kinghorn et al. (1991) found that the flat profiles downstream of etoile straighteners appeared to cause most of the residual negative errors in the orifice coefficient,

as opposed to the positive errors due to swirling flow. To ensure an error of less than $\pm 0.5\%$, a straightener 1D long should be placed upstream at least

- 6D for plates with a beta ratio of 0.5 and
- 14D for plates with a beta ratio of 0.8.

Even with 16D separation, virtually all tests suggested some effect from the straightener.

Karnik et al. (1994) tested the effect of a 19-tube bundle in a 101.6-mm line for good flow conditions and downstream of an elbow and looked at the effect on an orifice plate. Their results are very interesting in showing that even though the profile from the tube bundle at about 20D was very close to the fully developed turbulent profile, approximating to Equation (2.4) with $n = 7.4$, the turbulence intensity pattern is markedly different, suggesting that this is a factor in the response of orifice plates. They showed that the orifice coefficient downstream of the elbow and the tube bundle, compared with the correct value, was high by about 0.2% between 10D and 15D and fell to about 0.1% high by about 20D. There was a point closer to the orifice plate where the coefficients were approximately the same, but it would be unwise to rely on this for precise measurements (cf. Morrow et al. 1991 who used an arrangement that allowed them to slide a tube bundle conditioner along the pipe between a 90° long-radius bend and an orifice plate to obtain the variation of discharge coefficient with position for a gas flow. Overall the results appeared to suggest that, if possible, the conditioner should be next to the bend.)

Sattary (1991) concluded from the EEC orifice plate program on installation effects that the ISO 5167 and AGA Report No 3 minimum straight length specifications needed to be revised. The results for conditioners 19D upstream of an orifice in fully developed flows showed shifts of up to 0.33–0.5% for $\beta = 0.57$, rising to about 0.57–0.75% for $\beta = 0.75$. For a $\beta = 0.2$, there was negligible effect from a tube-bundle conditioner at 5D. The reader concerned with this topic is encouraged to refer to the original paper, which has extensive experimental data and indications that there is variation between the performances of the various straighteners. There is some discrepancy between the various standards. For two bends in perpendicular planes, by far the most serious disturbance, the ISO requirement for no loss in accuracy is that the orifice should be at least 44D downstream for a $\beta = 0.55$ and 70D for a $\beta = 0.75$. On the other hand, the blanket requirement for a flow straightener is that there should be at least 20D from the fitting to the straightener and at least 22D from the straightener to the meter. This is a total of at least 42D plus the length of the straightener, and greater than or equal to all other requirements for β up to 0.75, apart from those for two or more bends in different planes. The equivalent values for AGA for two bends or tees not in the same plane are (Sattary 1991) for $\beta = 0.57$, 23.5D, and for $\beta = 0.75$, 35.5D, and with a short tube bundle straightener for the general case is 17D (straightener distance 5.5D + installation distance 11.5D). Sattary's results appeared to suggest that the values without straightener were not conclusive for ISO, but, apart from the two bends in perpendicular planes, the AGA lengths may be too short.

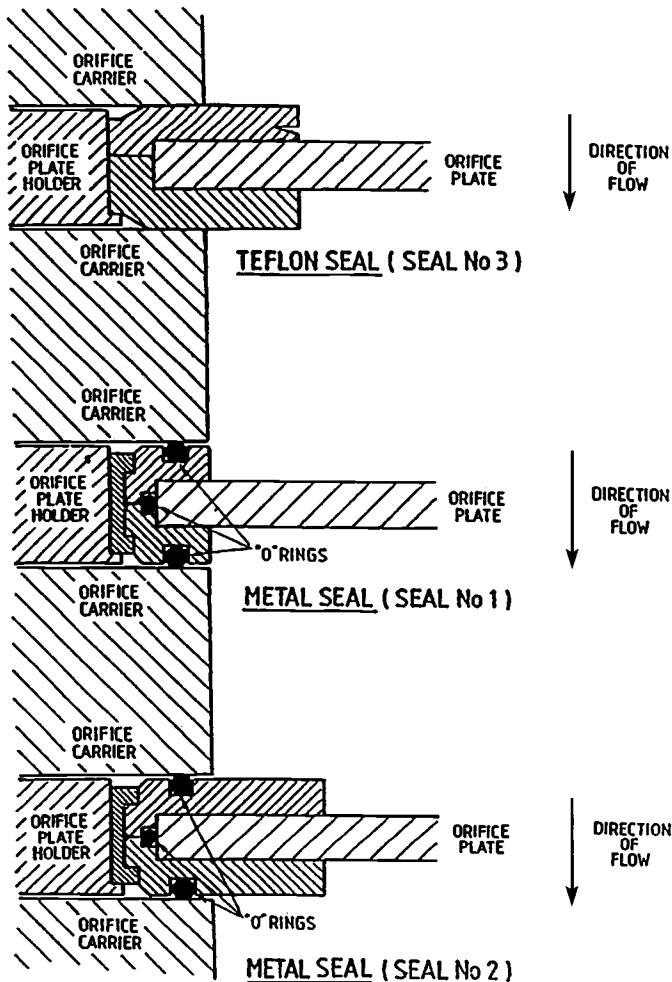


Figure 5.7. Differential seal ring arrangements (from Fulton et al. 1987 reproduced with permission of the authors, the Norwegian Society of Chartered Engineers, and Christian Michelsen Research).

The original research was carried out at the British Gas Research Station in Killingworth, England. It used plates rigidly mounted between flanges and tested at differential pressures between 0 and 120 mbar.

Fulton et al. (1987) undertook further tests to determine the effect of seal ring mountings as shown in Figure 5.7 and differential pressures up to 1000 mbar. Their results appear to suggest that the deflection with the Teflon seal ring was generally about 15% greater than the values from Jepson and Chipchase's (1975) formula, whereas the results for the No. 1 metal seal were close or about 20% less and for the No. 2 metal seal were close or about 30–40% less.

Simpson (1984) provided a useful digest of the papers by Norman et al. (1983, 1984) in which this problem has been developed and experimental work undertaken to validate the results. Simpson suggested that Norman et al.'s equation (essentially

that in BS 1042: Sect. 1.5: 1997)

$$\frac{\Delta q_m}{q_m} = - \left(\frac{\Delta p}{E^*} \right) \left(\frac{D'}{h} \right)^2 \left(\frac{a_1 D'}{h} - c_1 \right) \quad (5.13)$$

where

q_m = mass flow rate

Δp = differential pressure across the plate

E^* = elastic modulus of plate material

D' = orifice plate support diameter

h = thickness of orifice plate (= E in Figure 5.3, following ISO 5167)

could have simpler expressions for a_1 and c_1 than given by Norman et al. because the plate thickness will usually be fixed by the next most appropriate thickness of material. Simpson, therefore, gave (as in the standard)

$$a_1 = \beta(0.135 - 0.155\beta)$$

$$c_1 = 1.17 - 1.06\beta^{1.3}$$

To avoid plastic deformation, the differential pressure must be below

$$\Delta p_y = \sigma_y \left(\frac{h}{D'} \right)^2 \frac{4}{6(0.454 - 0.434\beta)} \quad (5.14)$$

where

σ_y = yield stress for plate material

β = orifice plate diameter ratio d/D

With Equations (5.13) and (5.14), it is possible to design for an error of, say, not more than 0.1% for a certain working pressure and for safe operation without plastic deformation with fault conditions of flow and differential pressure drop.

For a working differential pressure of up to 500 mbar (and up to 1000 mbar under fault conditions without yielding) for $\beta = 0.4$ with 304S.S plate material having $E^* = 195.4 \times 10^9$ Pa and $\sigma_y = 215 \times 10^6$ Pa, Simpson obtained values of

$$h/D' \geq 0.016 \text{ for required maximum error}$$

$$h/D' \geq 0.014 \text{ to avoid plastic deformation}$$

h must, therefore, be equal to or greater than $0.016D'$. It should be noted that, in extreme cases of high differential pressure, the allowed thickness according to the

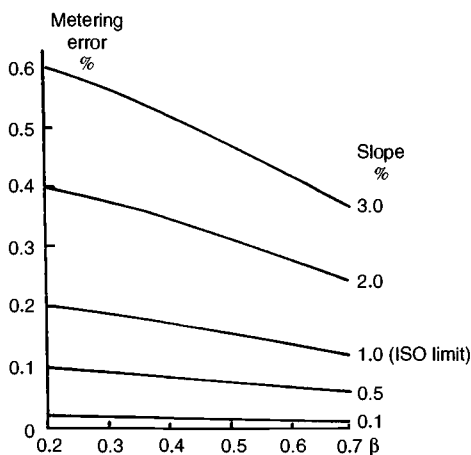


Figure 5.8. Metering error due to initial lack of plate flatness (from Simpson 1984; reproduced with permission of the Institute of Measurement and Control).

standards may be exceeded in meeting these plate bending requirements. The standard also implies that a safety factor on σ_y (of three) is a wise precaution.

Figure 5.8 gives curves for errors due to flatness slope against β ratio.

5.7 EFFECT OF PULSATION

In considering the installation of a flowmeter, the assumption is usually made that the flow is steady. In many cases, this is probably a fair assumption. However, there are situations when it is not the case (e.g., when a reciprocating compressor, an internal combustion engine, or some form of rotary valve is in the line). It is often very difficult to decide whether the flow is indeed steady or pulsating in some way.

We might idealize the pulsating flow as consisting of a sinusoidal ripple superimposed on a steady flow. This idealization is probably seldom valid, and little will actually be known about the amplitude, frequency, or flow profile. Mottram (1992, cf. BS 1042 Section 1.6, which uses some of his work), after the wise comment, "If you can't measure it, damp it!" made some additional useful points.

- ❑ If the frequency is above about 2 Hz, differential pressure transducers will, generally, be too heavily damped to pick it up. (Transducers with a response up to 2 kHz or more are likely to be specified and to be more expensive.)
- ❑ The standard describes some detection techniques including the use of a thermal probe to sense the presence of pulsation.
- ❑ Some flowmeter signals show indication of pulsation.
- ❑ In some cases, the error may be deduced from the raw flowmeter signal.
- ❑ Connecting leads in differential pressure meters can cause resonance and confusing effects.

The orifice plate meter is affected by pulsation, and so it is necessary to reduce the pulsation and to have an idea of the error likely from any residual pulsation.

The effects of pulsation on an orifice meter are:

- a. *Nonlinear (square root) error.* If a signal is averaged to obtain a mean flow, it is necessary that the instantaneous flow be proportional to the signal. If this is not so, an error will be introduced.
- b. *Inertia error.* If the fluid does not follow the changes in the flow rate instantaneously, an error due to the inertia of the fluid will result.
- c. *Velocity profile effect.* This effect is a result of the change in profile from fully developed to the unsteady flow.
- d. *Resonance.* Resonance occurs because some component of the system is resonating at the pulsation frequency.
- e. *Limitations in the pressure measurement device.* If the pressure measurement device does not respond correctly to pulsating flow, the pressure measurement may be incorrect.

Of these, the most important are (a), (d), and (e), the last two of which are related. The latter two are particularly possible in the connecting tubes between the

flow tube and the transducer, and because of the rate of response of the transducer. A manometer will possibly be of little value in an application with pulsation.

Thus for (a), Gajan et al. (1992) reviewed work that showed that, for quite high values of the ratio of rms fluctuating pressure difference to steady pressure difference, up to 0.5 at least, the main error is due to the square root effect and can be eliminated with a high speed response transducer. Even with a slow response transducer, the error for the ratio, if within 0.1, may be within about 1%.

Botros et al. (1992) looked at (d) and (e) and found that the combination of lines and transmitter form a resonator that can amplify or attenuate. They recommended that very short lines of essentially constant diameter and a transmitter with a small volume and a high frequency response should be used.

Commercially available differential pressure transmitters are generally unsuitable for dynamic measurement (Clark 1992). The dynamic response of differential pressure transducer systems is significantly affected by length of transmission line and whether the measurement medium is liquid or gas. Gas and liquid lines appear to be susceptible to oscillations, and in liquid systems there is a need for proper gas venting arrangements. Considerable care is required in order to make reliable dynamic measurements particularly when using a liquid medium (Clark 1992).

The square root problem is shown by

$$\frac{1}{t} \int_0^t \sqrt{\Delta p} dt \leq \sqrt{\frac{1}{t} \int_0^t \Delta p dt} \quad (5.15)$$

A transducer with pressure connections that is capable of responding within a time that is short compared with the period of the fluctuation, and the signal of which is square-rooted immediately and then averaged, will give the correct average value. This is essentially the left-hand side of the inequality (5.15). On the other hand, if the pressure is averaged and then the square root is taken, this is essentially the right-hand side of the inequality and the apparent flow rate will be high compared to the actual flow rate. Mottram's (1981) results for pulsation error are shown in Figure 5.9(a), and they suggest that the experimental results confirm the dominance of the square root error when compared with errors (b) and (c). However, Mainardi et al. (1977) appeared to suggest that (b) and (c) cause a change in the coefficient so that the effect of pulsation for a fluctuating ripple of 6% of the mean flow will cause an error if pulsation is measured with a high frequency response transducer of 1.5%, whereas the error is 4% for a 15% ripple. (Frequencies appeared to be up to about 50 Hz.)

Williams (1970) identified the possible sources of error when manometric devices are used to indicate mean differential pressure in a pulsating flow as

- wave action and resonance effects in the connecting leads,
- volumes of ducts of varying section within the manometer and leads, and
- restrictions in the leads, which cause a nonlinear relationship between flow and pressure drop and result in nonlinear damping.

The effect of pulsation on a manometer is shown in Figure 5.9(b).

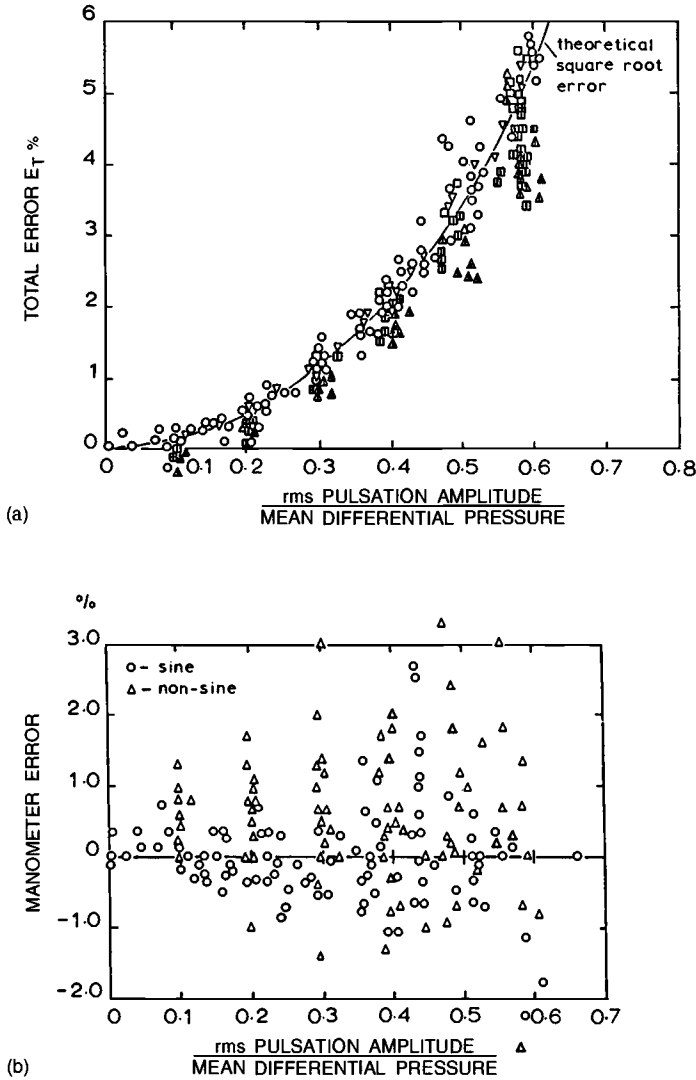


Figure 5.9. Effect of pulsation on pressure measurement (from Mottram 1981; reproduced with permission of the author and of the Instrument Society of America): **(a)** Total error at low pulsation amplitude; **(b)** U-tube manometer error.

Before leaving the subject of pulsation effects, it is important to include a further note (cf. Section 2.6) on the Hodgson number:

$$H = \frac{Vf \Delta p}{q_v p} \quad (5.16)$$

where V is the volume of pipework and other vessels between the source of the pulsation and the flowmeter position (Figure 2.5), Δp is the pressure drop over the same distance, f is the frequency of the pulsation, q_v is the volumetric flow rate, and p is the absolute line pressure. With this number, it is possible to plot curves to show the likely error levels for various values of H . Oppenheim and Chilton (1955)

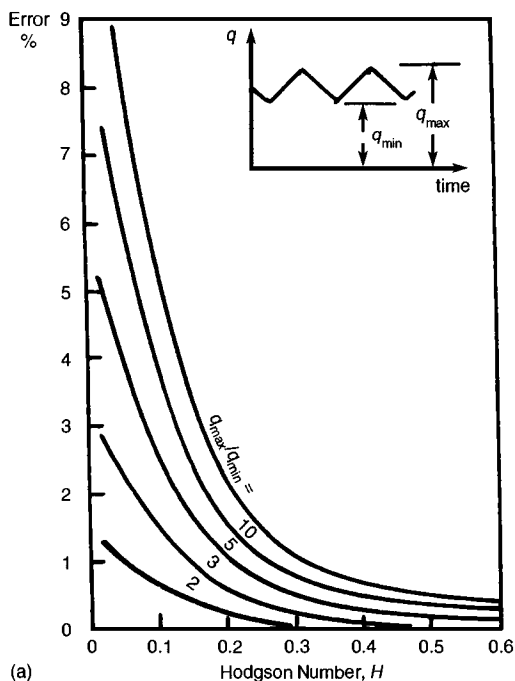
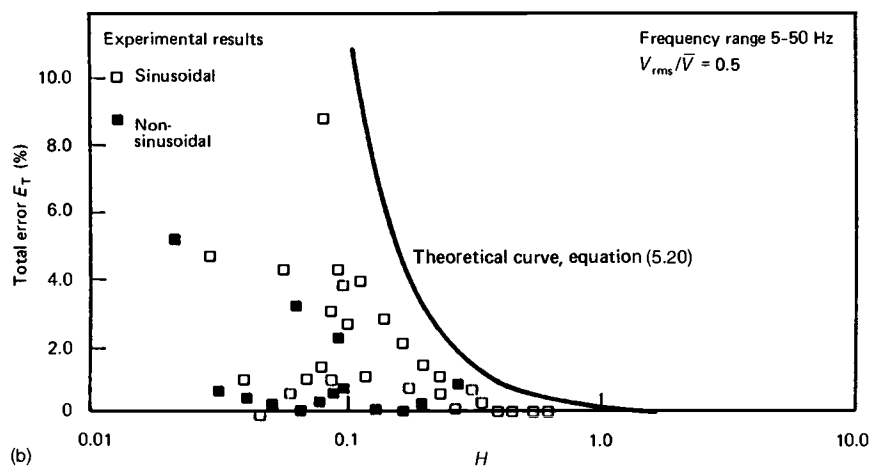


Figure 5.10. Hodgson's number. **(a)** Pulsation error curves for a triangular pulsation of varying amplitude (from Oppenheim and Chilton 1955; reproduced with permission of ASME). **(b)** Comparison between Hodgson's number and experimental results (from Mottram 1989; reproduced with permission of Elsevier Science).



have given plots of error levels for various types of flow. In Figure 5.10(a), one such set of curves is shown.

Mottram (1989) suggested that an expression for the total error in the indicated flow rate of a differential pressure flowmeter is

$$E_T = (1 + (V_{\text{rms}}/\bar{V})^2)^{1/2} - 1 \quad (5.17)$$

which for small amplitude ratios can be written as

$$V_{\text{rms}}/\bar{V} = (2E_T)^{1/2} \quad (5.18)$$

where $\bar{V}$ is the mean velocity and V_{rms} is the rms value of the unsteady velocity fluctuation in the pipe. He then suggested that if a maximum allowable percentage error is ϕ where $\phi = 100E_T$, and using the criterion for damping in Equation (2.14), then

$$\frac{H}{\gamma} \geq \frac{1}{4\pi} \frac{(V_{\text{rms}}/\bar{V})_{\text{ud}}}{\sqrt{2\phi}/10} \quad (5.19)$$

where subscript ud implies undamped and γ is the isentropic index for the gas or

$$\frac{H}{\gamma} \geq K \frac{(V_{\text{rms}}/\bar{V})_{\text{ud}}}{\sqrt{\phi}} \quad (5.20)$$

where $K = 10/(4\sqrt{2}\pi) = 0.563$. He claimed that experimental evidence is available to validate the safety of this criterion provided the flow pulsation amplitude can be measured. Some of Mottram's (1989) data compared with Equation (5.20) with $K = 0.563$ is shown in Figure 5.10(b). Where estimation is necessary, he suggests a safety margin by using $K = 1$. This appears to be confirmed by the data that Mottram (1989) included in his paper (cf. Sparks et al. 1989 who emphasized the problems and the only safe solution – the removal of pulsation).

5.8 EFFECTS OF MORE THAN ONE FLOW COMPONENT

Attempts have been made to relate the differential flowmeter equation to the data for flow of a fluid mixture, most commonly steam, through the orifice. The attempted adjustments have followed three approaches:

1. Adjusting the value of density to reflect the presence of a second component (cf. James 1965–6);
2. Adjusting the discharge coefficient to introduce a blockage factor for the other components, expressed as a function of the dryness (cf. Smith and Leang 1975);
3. Relating the two-phase pressure drop to that which would have occurred if all the flow were passing either as a gas or as a liquid. The ratio of two-phase pressure drop to liquid flow pressure drop Φ_{fo}^2 is equated to a function of x , the dryness fraction.

Miller (1996) favored the third approach as being the easiest to use. It has also been investigated by several workers, and various expressions for Φ_{fo}^2 have been tabulated by Grattan et al. (1981) from the work of Chisholm (1977), Collins and Gacesa (1970), James (1965–1966), and Watson et al. (1967). For other references, the reader is referred to Baker (1991a).

The main reservation concerning some of these correlations relates to whether or not they can be applied to different pipe geometries, fluids, Reynolds numbers, etc.

Steam

Grattan et al. (1981) gave experimental pressure loss data for orifice tests with an empirical curve:

$$\Phi_{\text{fo}}^2 = 1.051 + 291x - 3,796x^2 + 74,993x^3 - 432,834x^4$$

for $0.00005 \leq x \leq 0.1$ (5.21)

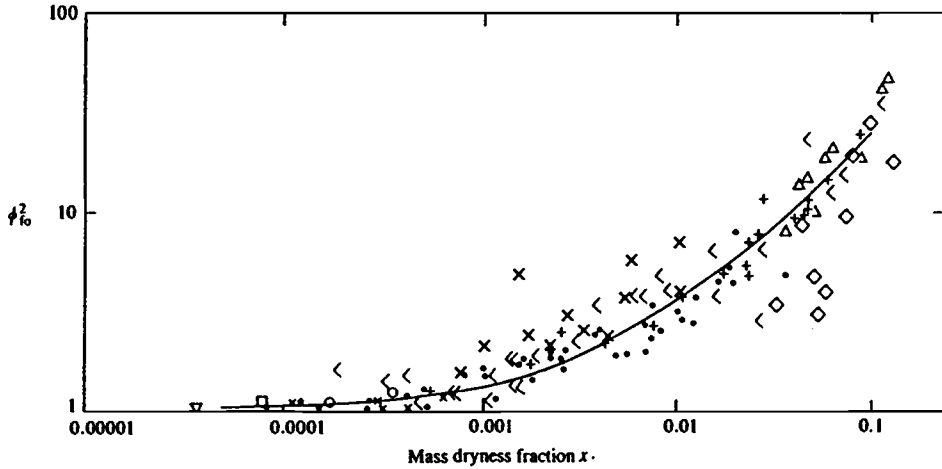


Figure 5.11. Experimental pressure loss data for orifice tests (from Grattan et al. 1981; reproduced by permission of NEL).

and this is shown in Figure 5.11. The scatter is an indication of the limited value of such an empirical curve and of attempts to predict flowmeter performance in two-phase flows. Rooney (1973) also obtained values for the parameter Φ_{fo}^2 (cf. Chisholm, 1967; Chisholm and Leishman, 1969; Chisholm and Watson, 1966).

Owen and Hussein (1991) pointed out that if the water content was very small, the droplets would probably have a negligible effect on the flowmeter response, and so the mass flow should be corrected to allow for the mixture density rather than the vapor density. For wet steam of dryness x greater than about 0.9, this correction is approximately $F = 1/x$. Owen and Hussein compare this value with other corrections in Figure 5.12(a). Note that y is the wetness fraction. In particular, BS 3812 suggested

$$F = \sqrt{2 - x} \quad (5.22)$$

and the *Shell Flowmeter Engineering Handbook* proposed

$$F = 1.74 - 0.74x \quad \text{for } x > 0.95 \quad (5.23)$$

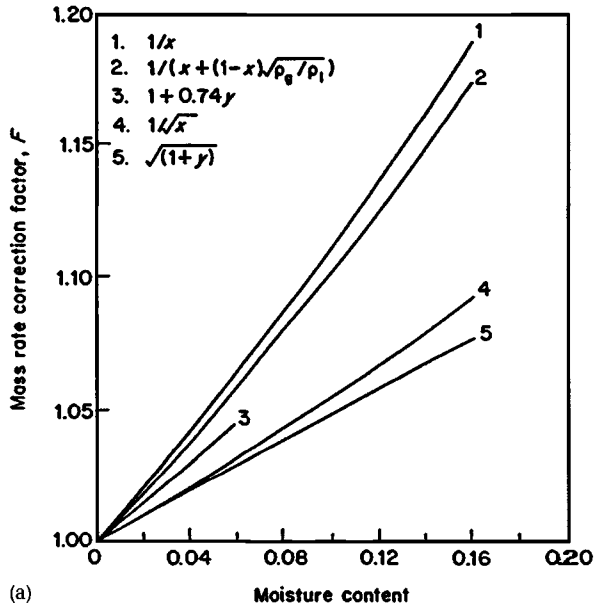
Hussein and Owen (1991) referred to two correlations: James's giving a correction factor of

$$F = \left\{ \frac{\rho_1}{x^{1.5}\rho_1 + (1 - x^{1.5})\rho_g} \right\}^{1/2} \quad (5.24)$$

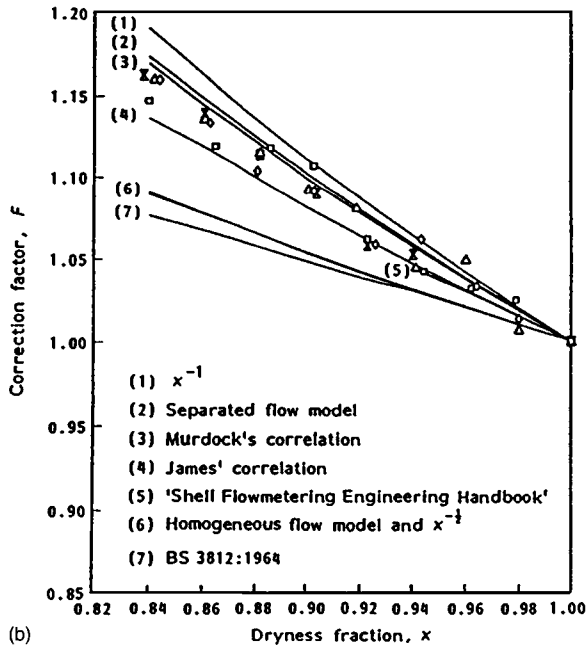
and Murdock's

$$F = \frac{1}{x + 1.26(1 - x)(\rho_g/\rho_1)^{1/2}} \quad (5.25)$$

Figure 5.12(b) for seven alternative correlations suggests that the data scatter around those of James and Murdock. Hussein and Owen (1991) showed that pressure had little effect on correction factors, and that the *Shell Flowmeter Engineering Handbook's* correlation is the best compromise for $x > 95\%$ for the orifice. They wisely advocated



(a)



(b)

Figure 5.12. Wet steam flowmeter correction factor. **(a)** Proposed expressions plotted against moisture content γ , where x is the dryness fraction (from Owen and Hussein 1991; reproduced with permission of Elsevier Science). **(b)** Comparison between data for wet steam correction factor and proposed factors from the literature (from Hussein and Owen 1991; reproduced with permission of Elsevier Science).

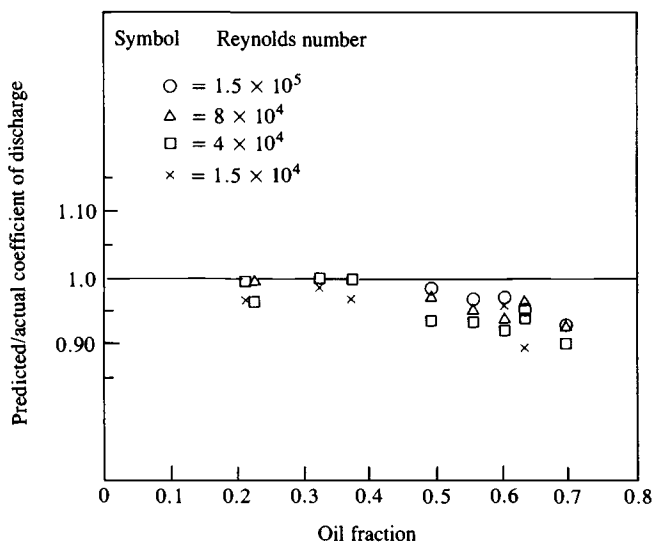


Figure 5.13. Performance of orifice meter in an oil-in-water emulsion (from Pal and Rhodes 1985; reproduced with permission of BHR Group).

an upstream separator and indicated that several types of upstream separators give a value of $x > 95\%$ with reasonably low pressure loss. Owen et al. (1991) provided further convincing experimental evidence for the benefits of fitting a separator in a steam line before an orifice plate. The separator, of course, should make more accurate flow measurement possible. They showed that slugs of water in a gas flow, trapped by valves and then released, can travel at 50 m/s or more and result in impact pressures at 5 bar tank pressure of, in some cases, well over 50 bar. One of the orifices they used was made of 3-mm-thick mild steel and had deformed very substantially after 20 impacts.

Wenran and Yunxian (1995) suggested a simple model for deducing flow rate and phase fraction for steam flows in orifice plates, and they claim that errors were, respectively, 9 and 6.5% for their model. The reader is referred to the original article for more details. Pressure noise has also been suggested as a means of obtaining more information in steam-water flows (Shuoping et al. 1996). (cf. Fischer's 1995 calculation for gas-liquid annular mist).

Oil-in-Water Flows

Pal and Rhodes (1985) reported results of tests of orifice meters in horizontal oil-water mixtures. Figure 5.13 shows the effect of increasing oil content on each meter.

Pal (1993) tested orifices with a range of oil-in-water emulsions ranging from 30 to 84% (inversion took place at 78% oil) and found that stable emulsions introduced coefficient changes within about 3% using a generalized Reynolds number, which is defined in the paper, but for unstabilized emulsions the changes were greater.

Other Applications

Murdock (1961) reported correlations to within $\pm 1.5\%$ for two-phase flow. Others are reported by Lin (1982) and Mattar et al. (1979). Washington (1989) claimed

good predictions using wet gas. Majeed and Aswad (1989) appear to have achieved agreement between measured and predicted flows based on a correlation by Ashford-Pierce when using oil field data. They included a computer listing.

5.9 ACCURACY UNDER NORMAL OPERATION

To achieve the accuracy set out in the standards, it is necessary to construct and install the meter to the detailed specification of the standards.

From the equation

$$q_m = \frac{CE\epsilon\pi d^2 \sqrt{2\rho_1 \Delta p}}{4} \quad (5.1)$$

the standard provides a formula for the total uncertainty (cf. Equations 1.A.7 and 7.17 for alternative symbols) in the mass flow rate:

$$\frac{\delta q_m}{q_m} = \left\{ \left(\frac{\delta C}{C} \right)^2 + \left(\frac{\delta \epsilon}{\epsilon} \right)^2 + \left(\frac{2\beta^4}{1-\beta^4} \right)^2 \left(\frac{\delta D}{D} \right)^2 + \left(\frac{2}{1-\beta^4} \right)^2 \left(\frac{\delta d}{d} \right)^2 + \frac{1}{4} \left(\frac{\delta \Delta p}{\Delta p} \right)^2 + \frac{1}{4} \left(\frac{\delta \rho_1}{\rho_1} \right)^2 \right\}^{1/2} \quad (5.26)$$

where α , ($=CE$) the flow coefficient, has now been eliminated. To obtain the overall uncertainty, the uncertainty in each term is obtained and combined according to Equation (5.26).

It is instructive to put some values into this equation. Let us set out some values and the likely errors in them:

Parameter	Value	Uncertainty %	Comment
D	100 mm	0.2	Will depend on pipe specification.
d	50 mm	0.1	Machine to 0.002 in. or 50 μm .
β	0.5	—	Equal to d/D .
C	0.6051	0.5	From ISO formula or table (value of Re is assumed, but iteration is normally required).
E	1.0328	negligible	
ϵ	1	—	Incompressible liquid.
ρ	1,000 kg/m ³	0.1	Possible variation due to temperature or purity.
Δp	31.54 kPa	1	

From these values, we obtain

$$q_m = 9.75 \text{ kg/s} \quad (\text{Re} = 1.24 \times 10^5)$$

$$\frac{\delta q_m}{q_m} = \left\{ (0.5)^2 + (0)^2 + (0.133)^2 (0.2)^2 + (2.133)^2 (0.1)^2 + \frac{1}{4} (1)^2 + \frac{1}{4} (0.1)^2 \right\}^{1/2}$$

$$= 0.89\%$$

The estimate of uncertainty for gas metering is likely to be considerably larger than this figure. In a paper describing the specification, installation, commissioning, and maintenance of typical orifice metering stations for the British Gas national high pressure transmission system, with flows up to 1.4×10^6 std m³/h at pressures up to 69 bar, Jepson and Chamberlain (1977) concluded that, to achieve uncertainty of less than $\pm 2\%$, a very rigorous checking procedure was needed to ensure that the complete measuring system remained in specification all the time.

Miller (1996) provided a number of special corrections: for steam quality with gas-liquid flows, for saturated liquids with up to 10% saturated vapor, for drain and vent holes, for water vapor, and for indicated differential when the pressure-measuring device is at a different elevation to the differential pressure device and the pressure lines are filled with fluid of a different density.

Ting and Shen (1989) commented that the majority of natural gas flow measurements in the United States were determined by orifice meters. They led a systematic study of measurement by 152.4-mm (6-in.), 101.6-mm (4-in.), and 50.8-mm (2-in.) meters in the Reynolds number range 1 million to 9 million.

Ting and Shen used Honeywell smart static and differential pressure transducers, which had microprocessor-based built-in pressure and temperature compensation, higher span-turndown ratio, improved accuracy, and easy rangeability. Bias was quoted as better than $\pm 0.15\%$ for differential pressure and precision as $\pm 0.015\%$. A precision aneroid barometer of 0.1% full-scale uncertainty was used for atmospheric pressure. Careful calibration of these instruments took place, and the paper implied careful experimentation, the only weakness being that the definition of the inlet pipework was not entirely clear. Their tests showed that the 101.6- and 50.8-mm orifice discharge coefficients agreed with the ANSI and ISO standards within the estimated uncertainty levels of 0.35% with 95% confidence level. However, the 152.4-mm meter agreed for low and medium beta ratio, but for a beta ratio of 0.74, ANSI and ISO gave a discharge coefficient about 2% low. Their plots also, helpfully, show the divergence between the two previous standards, which is not more than about 0.2% over the range.

5.10 INDUSTRIALLY CONSTRUCTED DESIGNS

Orifice Plates

Orifice plates can be obtained from various manufacturers machined to the requirements of ISO 5167 (BS 1042) in stainless steel 316, 321, 304, Hastelloy, tantalum, Inconel, and plastics such as polytetrafluoroethylene (PTFE), polyvinylchloride (PVC), and polyvinylidene fluoride (PVDF). Also available are orifices that are removable from the holding ring, with integral gaskets that may be spiral wound in stainless steel or of PTFE or graphite filled. Size availability extends beyond the common standard's range going down to a 25-mm bore or less. The orifices will then be stamped permanently with detailed information (e.g., tag number, pipe size, rating, orifice

diameter, material), usually on the upstream side. (Figure 5.14). The information will also state which is the upstream side of the plate.

Orifice Plate Carrier Assemblies

Some examples of orifice carrier assemblies are shown in Figure 5.15(a). The materials of construction are generally carbon steels, but stainless steels and duplex steels can be provided. Carriers can be split ring or one piece integral design.

Orifice Metering Run Assemblies

Orifice metering run assemblies are constructed to the appropriate standards and may incorporate flange tappings [Figure 5.15(b)]. Although plates are relatively easy to install and remove, there are occasions when a plate needs to be inspected without disrupting the flow or depressurizing the line. For this reason, at least one manufacturer (Figure 5.16) has developed a system for removing and replacing plates without shutting the line down. This consists of a means for sliding the orifice plate in and out, possibly with a rack and pinion mechanism, and a dual chamber pressure sealing system while the plate is removed.

Sizing

The plate manufacturer may offer a sizing service based on a computer program that undertakes the iteration and optimization, an otherwise laborious procedure.

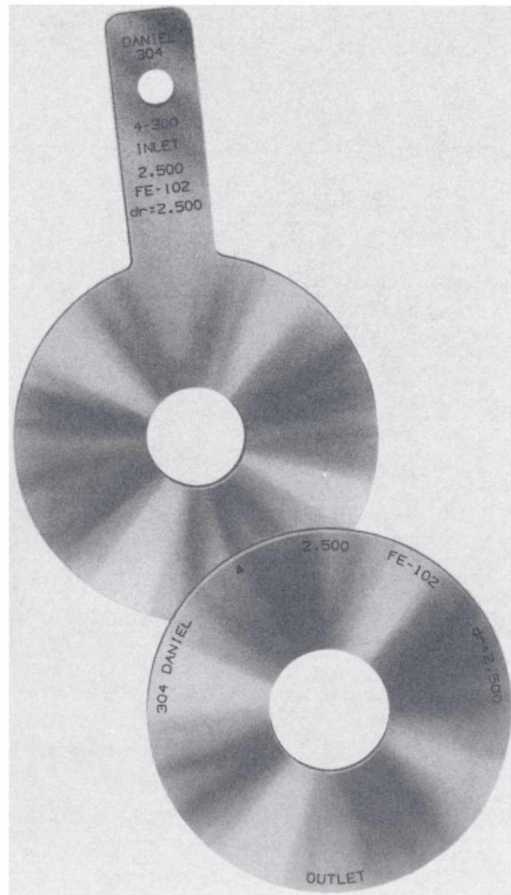


Figure 5.14. To indicate the main features of industrial orifice plates (reproduced with permission of Daniel Europe Ltd.).

5.11 PRESSURE CONNECTIONS

Tapping Orientation

The tapping positions should be such that any unwanted component of the line fluid or any second phase in the line fluid will not enter the pressure tapping of the impulse line or become trapped in the impulse line connections. For vertical lines, any position is satisfactory (Miller 1996). Where the line is not vertical, the following recommendations should be noted to reduce the chances of accidental blockage (BS 1042: Part 1: 1964 and Miller 1996):

- For liquids, within an angle of 45° above or below the horizontal;

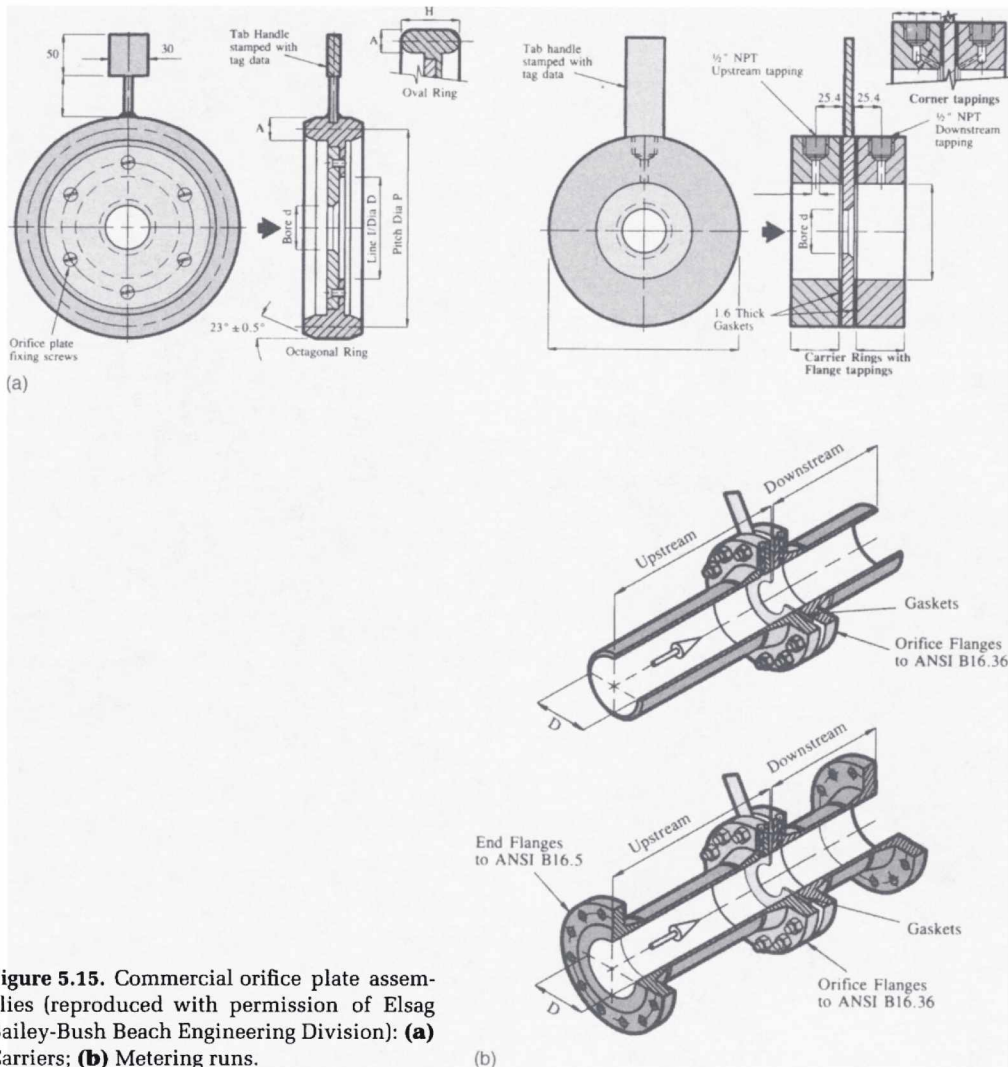


Figure 5.15. Commercial orifice plate assemblies (reproduced with permission of Elsas Bailey-Bush Beach Engineering Division): **(a)** Carriers; **(b)** Metering runs.

- ❑ For dry gases, between the horizontal and vertical upward with a suggestion that for clean noncondensable gases the tapping should be in a vertical position;
- ❑ For moist gases, between an angle of 30° above the horizontal and vertically upward; and
- ❑ For steam and other vapors, horizontal only.

These requirements are illustrated in Figure 5.17. Where a drain hole is provided through the orifice plate, the single tapping should be orientated so that it is between 90° and 180° to the position of the drain hole.

The standard also recommends that, where there is a danger of blockage, provision should be made for rodding out the tapping from outside the line.

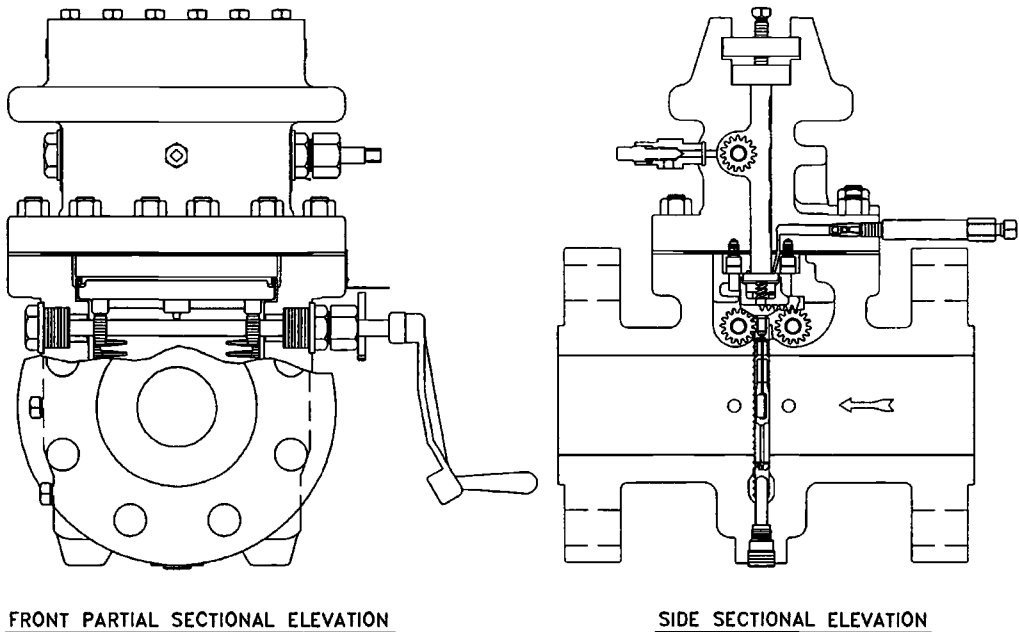


Figure 5.16. Commercial system for removing and replacing plates while on line (reproduced with permission of Daniel Europe Ltd.).

Piezometer Ring Arrangements

In pipe lines of 150 mm (6 in.) or greater, or in lines where one hole may become blocked, it may be useful to use a piezometer ring, as shown in Figure 5.18. In the presence of steam or other vapor flows where condensation may occur, a piezometer ring may form an unwanted trap for condensate. The manifold cross-section should always be equal to or greater than the total area of the individual tappings which it serves. The triple T arrangement may be preferred, and, in any case, traps for condensate and adequate bleed points should be provided. In laboratory applications, the use of transparent tubing is recommended.

Layout of Impulse Lines

It should be remembered that, even though the simple arrangement of manometer leads, with the same fluid in each, results in a deflection, the same as that which would occur for direct connection to the meter

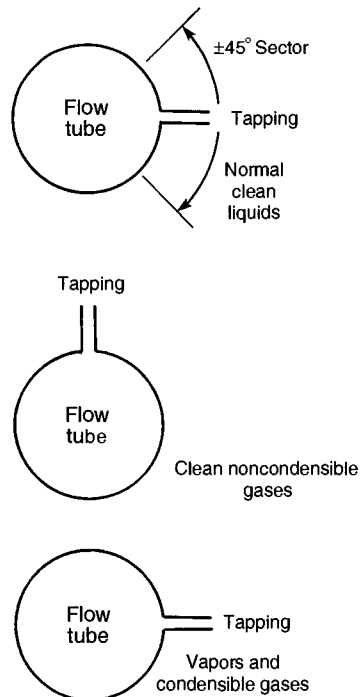


Figure 5.17. Diagram to show positioning of the pressure tapping for nonvertical lines.

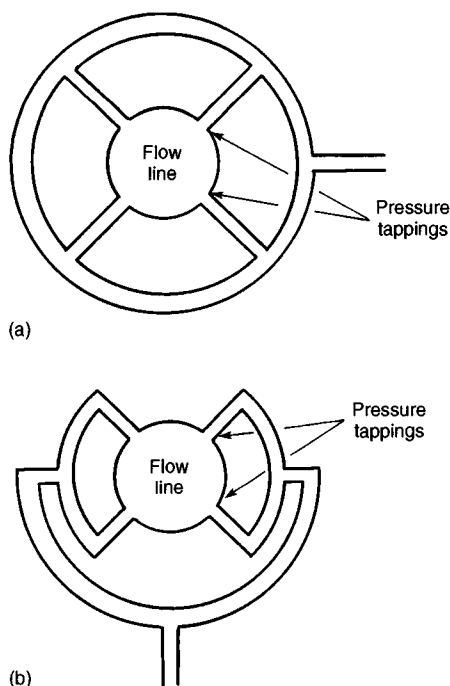


Figure 5.18. Diagram to show piezometer ring connections: **(a)** Conventional; **(b)** Triple T.

line, any variation in the connecting fluid due to air, etc., can cause variation in density and a difference in the pressure across each line. The effects to be guarded against are (BS 1042: Part 1: 1964):

- ❑ Temperature difference between the lines leading to a density difference;
- ❑ Discontinuity in the impulse line fluid due to a second component, be that solid, liquid, or gas; and
- ❑ Blockage of any sort including solidification of the impulse fluid.

The impulse lines should be as short as possible, at the same temperature (which may require lagging together), and at a sufficient gradient to allow drainage, etc., as required.

Miller (1996) should be consulted for detailed guidance on impulse lines, sealant liquids, and transmitter positions. A typical arrangement for steam is shown in Figure 5.19.

5.12 PRESSURE MEASUREMENT

The main methods used for pressure measurement are described in Noltingk (1988) and are manometers, mechanical devices that deflect under pressure, and electromechanical pressure transducers.

A MANOMETER

In a manometer, the height of a column of liquid provides a measure of the pressure in a liquid or gas.

B MECHANICAL DEVICES THAT DEFLECT UNDER PRESSURE

One such device is the Bourdon tube, which has a flattened tube wound into an arc. The tube unwinds under pressure. Another design uses diaphragm elements made up from pairs of corrugated disks with spacing rings welded at the central hole. Pressure will cause the assembly to elongate, and this will in turn be used as a method of registering the pressure change. Other devices based on expanding bellows or diaphragm elements are also used.

C ELECTROMECHANICAL PRESSURE TRANSDUCERS

This is the most important and accurate device for the future and is in a state of continual development. Various methods have been used to obtain an electrical signal from a mechanical deflection. Some of these are capacitance transducers,

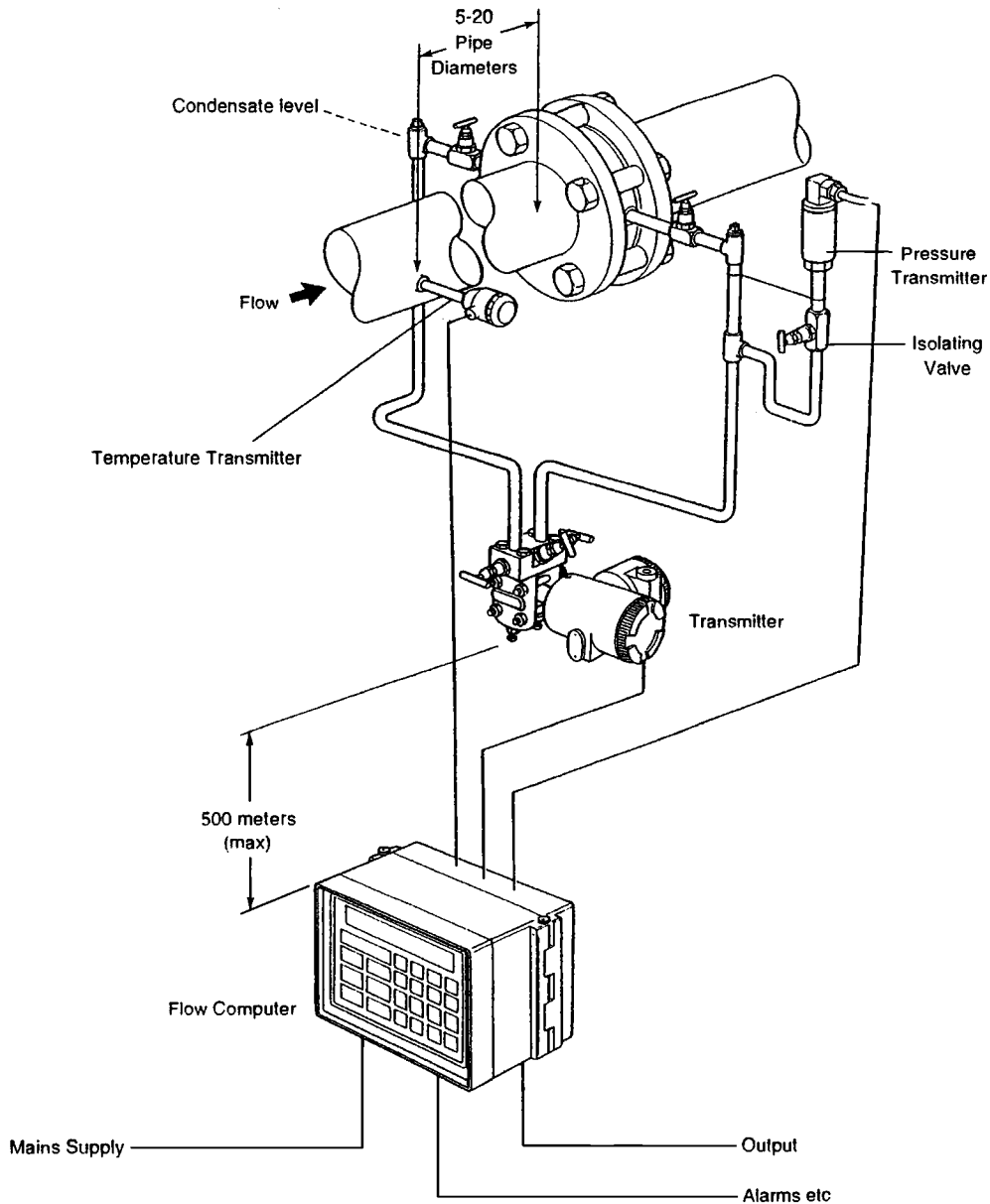


Figure 5.19. Installation for steam (reproduced with permission of Spirax-Sarco Ltd.). [Note: Tappings are often on the same side of the pipe.]

piezoresistive strain gauges, and piezoelectric and resonant devices. In one design, pressure applied to a diaphragm causes it to deflect and to exert a force on a cantilever beam that has a strain gauge bonded to it and that provides an electrical signal. Smart devices can provide condition monitoring, self-diagnosis of temperature, memory, and reference voltages and will provide ranging and temperature compensation. Measurement uncertainty can be as low as, or lower than, 0.1% of span. Regular recalibration at 1- to 6-month intervals is recommended.

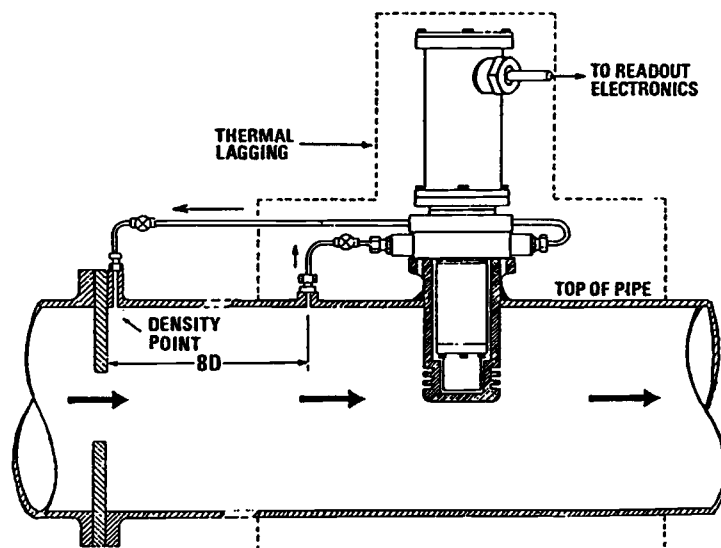


Figure 5.20. Typical example of the installation of a density measuring cell in an orifice metering line (courtesy of the Roxboro Group PLC).

5.13 TEMPERATURE AND DENSITY MEASUREMENT

Methods of temperature and density measurement are described by Noltingk (1988). A typical installation arrangement for a density cell in a metering line is shown in Figure 5.20. The cell is arranged to project into the line so that it is at the gas temperature, and the gas is extracted from the line, passed through the cell, and returned to the line.

5.14 FLOW COMPUTERS

Flow computers are microcomputers programmed to obtain flow from sensors measuring differential pressure, pressure, temperature, density, or other parameters, and are of particular importance in the calculation of the orifice plate flows. When used with smart transmitters, they may offer on-line optimization of differential pressure measurement. They contain stored information on the conversion of frequency and analogue signals to actual units, and they contain algorithms for interpreting density, etc.

Taha (1994) described electronic circuitry to obtain mass flow from an orifice plate with differential pressure, pressure, and temperature sensing.

5.15 DETAILED STUDIES OF FLOW THROUGH THE ORIFICE PLATE, BOTH EXPERIMENTAL AND COMPUTATIONAL

An early study of flow through an orifice plate using laser doppler techniques was reported by Bates (1981) for a BS 1042 orifice plate with $\beta = 0.5$ in the range of

Reynolds number 67,600–183,000. The shape of the profiles upstream at $0.535D$ and $0.32D$ show no recirculation zone in the plate corner, but this may be because the region of any such recirculation is closer to the plate than was investigated. Downstream at $0.5D$ the recirculation region has a zero velocity point at about 25% radius in from the pipe wall, and this moves to about the 20% radius position at $0.9D$ downstream.

Morrison et al. (1990b) reviewed past work on flow patterns in orifice meters, both theoretical and experimental. They used a laser doppler anemometer (LDA) to obtain detailed flows in an orifice. They also obtained (Morrison et al. 1990a) wall pressure distributions, and some of these are shown in Figure 5.21(a) for a $\beta = 0.5$ at $Re = 122,800$ both for conditioned flow and for three angles of swirl. Figure 5.21(a) shows the pipe wall pressure distributions with a minimum pressure at about $D/2$ downstream of the orifice plate. Note that data were not obtained between $0.125D$ upstream and $0.25D$ downstream of the plate. Figure 5.21(b) shows the variation of the velocity on the centerline of the orifice meter, the high peak value at the vena contracta, and possibly a suggestion of poor diffusion.

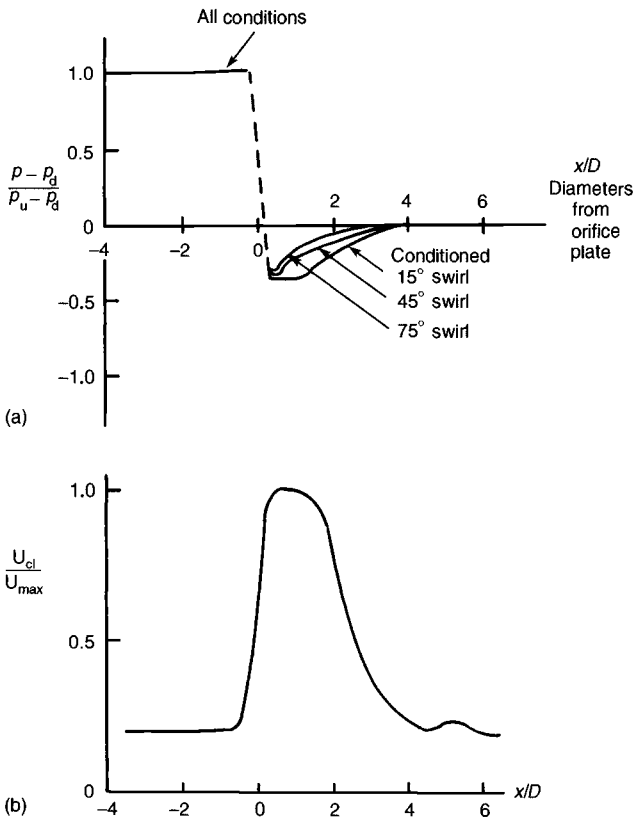


Figure 5.21. Experimental curves (after Morrison et al. 1990a, 1990b). **(a)** Pressure distributions for conditioned flow and for three angles of swirl from pipe wall tapings for an orifice with $\beta = 0.5$ and $Re = 122,800$. **(b)** Center-line velocity distribution using a laser doppler anemometer for an orifice with $\beta = 0.5$ and $Re = 18,400$.

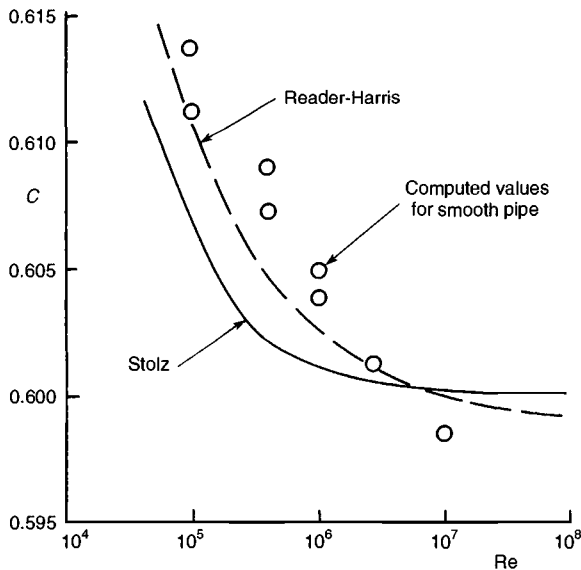


Figure 5.22. Computed values of discharge coefficient C , for an orifice with $\beta = 0.7$, corner tappings, and smooth pipe (after Reader-Harris 1989).

Erdal and Andersson (1997) provided a brief review of some computational fluid dynamics (CFD) work. They emphasized the sensitivity to the use of a particular code (cf. Spencer et al. 1995). Davis and Mattingly (1997) claimed agreement between computed and experimental discharge coefficients to within about 4% for β in the range 0.4–0.7 and Re between 10^4 and 10^6 . Sheikholeslami et al. (1988) and Barry et al. (1992) claimed an agreement of about 2%. Patel and Sheikholeslami (1986) computed the discharge coefficient for $Re = 2,500,000$ with $\beta = 0.4$ to within 1.5%.

Reader-Harris and Keegans (1986) and Reader-Harris (1989) claimed that discharge coefficients could be calculated to within 0.64%. Figure 5.22 shows some of Reader-Harris's computations using the $\kappa - \epsilon$ turbulence model with a computer package (QUICK). In Figure 5.22, computed results for a smooth pipe and a $\beta = 0.7$ are compared with the equation in the standard at that date (Stolz) and with an improved equation (cf. Reader-Harris and Sattary 1990) from which the current ISO equation has developed. Where two symbols occur at one value of Reynolds number, the higher is for the more refined grid. Reader-Harris only considered that he had obtained independence from grid size for $Re \geq 3 \times 10^6$. His results appear to give encouraging agreement with the modified equation.

Spearman et al. (1991) described laser doppler velocimeter (LDV) measurements downstream of a Mitsubishi flow conditioner and of an orifice plate in a combined package.

Erdal and Andersson (1997) indicated the problems which still exist in obtaining adequate modeling of the flow through an orifice plate. They commented that the predictions revealed the need for considerable expertise and care before results, especially for pressure drop, could be predicted with confidence.

5.16 APPLICATION, ADVANTAGES AND DISADVANTAGES

The standard orifice can be used with almost any single-phase Newtonian flow, whereas for high viscosity fluids, a quadrant or conical orifice may be used. Abrasive fluids are likely to change the shape of the leading edge of the plate and therefore the calibration of the orifice meter and are not to be advised. The application of the orifice plate to two-phase flows should be attempted with great caution.

An advantage of the orifice plate flowmeter is that the experience over many years has been well documented and is distilled in the standards, allowing a device to be designed and constructed with confidence. The uncertainty is, also, calculable from the standards. Thus without calibration of the instrument, it is possible to set a value against the uncertainty of the measurements. Few other meters offer this possibility.

Its ease of installation in the line has to be balanced by the need for very careful construction in order to obtain the calculable uncertainty. Some features of the design (e.g., the edge radius of the plate inlet for small plate diameters) are very difficult to achieve.

Another disadvantage is the nonlinear characteristic of the meter due to Equation (5.1). This, in turn, leads to a restricted range since the differential pressure measurement device will need to respond to, say, 100:1 turndown for a 10:1 flow turndown. With smart transducers, this is more possible than in the past. Another problem with the nonlinear characteristic is the effect of pulsation on the reading.

The high pressure loss due to the poor pressure recovery after the plate may be a disadvantage in many applications and will certainly lead to energy losses.

The cost of installation of the orifice will not be markedly less than other full-bore flowmeters, when allowance has been made for pressure transducers, flow computer, etc. Maintenance will be aided if one of the commercial designs which allows the orifice plate to be withdrawn from the line, even under operating conditions, is used. Pressure transducers should be regularly checked. This does not ensure that the total system has retained operational integrity or that the calibration is unchanged by debris in the impact pipes or by fouling or rusting of the flow pipe causing changes in roughness and hence of the profile entering the meter. However, it allows a fuller record to be kept without removal for calibration and, with other operating data, may give greater confidence in the integrity of the instrument.

5.17 CHAPTER CONCLUSIONS

The orifice continues to have an important and valued place in flow measurement because of its longevity and its documentation in the standards and because custom and practice require its use in some industries. The recent update of the standard is evidence for this.

It will, no doubt, continue to have an important niche, and manufacturers of orifice meter components will continue to provide a useful service particularly for fiscal metering of natural gas and oil products (Reader-Harris 1989).

The measurement of C , of C with upstream fittings causing distortion to the inlet profile, of C with upstream conditioners, etc., has been, and will continue to be, the subject of experimental programs. The value of such data may possibly decrease with time, as those seeking high accuracy look to other flow metering devices. However, the orifice will continue to be specified by industry for many applications and legal reasons. The benefit of all the work that has been done, a selection of which has been referred to in this chapter, is that the ISO installation requirements have been generally confirmed. In addition, data on the use of a flow conditioner do not yet instill complete confidence in the resulting orifice performance (but cf. the work by Laws and others reported in Chapter 2).

Users should also be conscious of the degrading of precision that results from corrosion, wear, and deposits on the plate, in the pipe, and in the pressure tapings. The effect of wear on the inlet edge of the plate may be substantial.

The computation of the flow, particularly if coupled with upstream distortion, offers a considerable challenge, and the results can be sensitive to the detailed application of computer programs. It would seem to be a prime candidate for a perturbation approach, where the computer program could be used to calculate changes from the norm rather than attempt to obtain the absolute discharge coefficient. Such an approach might shed further light on the shape of the empirical equations and the trends in discharge coefficient for variation in D , β , and Re (cf. Reader-Harris 1989).

APPENDIX 5.A

Orifice Discharge Coefficient

I am indebted to Dr. Michael Reader-Harris for clarifying the position with regard to the adoption of a new form of the discharge coefficient equation by ISO. At the time that this book is going to press, the ISO has not been finalized, and the United States has not changed its version to reflect the full database used in the modification. Therefore, rather than omit the equation from the book, I have decided to include the provisional form as set out in ISO 5167-1: Amd 1: 1998 and by Reader-Harris and Sattary (1996).

The equation for C , which has come to be known as the Reader-Harris/Gallagher equation, is:

for $Re \geq 4,000$ and $D \geq 71.12$ mm (2.8 in.),

$$C = 0.5961 + 0.0261\beta^2 - 0.216\beta^8 \quad C_\infty \text{ term}$$

$$\left. \begin{aligned} &+ 0.000521 \left(\frac{10^6 \beta}{Re} \right)^{0.7} \\ &+ (0.0188 + 0.0063A)\beta^{3.5} \left(\frac{10^6}{Re} \right)^{0.3} \end{aligned} \right\} \text{ Slope term}$$

$$\begin{aligned}
 & \left. \begin{aligned} & + (0.043 + 0.080e^{-10L_1} - 0.123e^{-7L_1}) \\ & \times (1 - 0.11A)\beta^4(1 - \beta^4)^{-1} \end{aligned} \right\} \text{Upstream tap term} \\
 & - 0.031(M'_2 - 0.8M_2^{1.1})\beta^{1.3} \quad \text{Downstream tap term}
 \end{aligned}$$

Where $D < 71.12$ mm (2.8 in.), the following term should be added:

$$+ 0.011(0.75 - \beta)(2.8 - D/25.4) \quad (D: \text{mm}) \quad (5.A.1)$$

where Re is based on the pipe diameter D ,

$$\begin{aligned}
 M'_2 &= \frac{2L'_2}{1 - \beta} \\
 A &= \left(\frac{19,000\beta}{Re} \right)^{0.8}
 \end{aligned}$$

where $L_1 = l_1/D$ and l_1 is the distance of the upstream tapping from the upstream face of the plate, and $L'_2 = l'_2/D$ and l'_2 is the distance of the downstream tapping from the downstream face of the plate. The ' signifies that the measurement is from the downstream and not the upstream face of the plate.

The terms in A are only significant for small throat Reynolds number. M'_2 is, in fact, the distance between the downstream tapping and the downstream face of the plate divided by the dam height.

For the purposes of Equation (5.A.1) the values of the upstream and downstream lengths are:

	L_1	L'_2
D and $D/2$	1	0.47
Flange	$25.4/D$	$25.4/D$ (D : mm)
Corner	0	0

The uncertainty associated with Equation (5.A.1) is

$$\begin{array}{lll}
 (0.7 - \beta)\% & \text{for} & 0.1 \leq \beta < 0.2 \\
 0.5\% & \text{for} & 0.2 \leq \beta \leq 0.6 \\
 (1.667\beta - 0.5)\% & \text{for} & 0.6 < \beta \leq 0.75
 \end{array}$$

and for $D < 71.12$ mm (2.8 in.), the following should be added arithmetically:

$$+ 0.9(0.75 - \beta)(2.8 - D/25.4) \quad (D: \text{mm})$$

Venturi Meter and Standard Nozzles

6.1 INTRODUCTION

In this chapter, we are concerned with devices defined in international standard documents. Among these, the venturi meter is one of the oldest industrial methods of measuring flow, although other methods may be more common. Chapter 8 deals with other devices such as variable area meters, target meters, averaging pitot tubes, Dall meters, and other venturi, nozzle, and orifice-like devices.

As in the case of the orifice plate, the reader should have access to a copy of the latest version of the ISO standard (5167-1). I have refrained from repeating information that should be obtained from the standard, and include only the minimum information to provide an understanding of the discharge coefficients, the likely uncertainty of the various devices, and the type of material from which they may be constructed. However, although the amount of material published recently is not great, there is a growing interest in the use of some of these devices for difficult metering tasks and a reevaluation of their behavior, which may well lead to several reports in the next few years.

Spink (1978) suggested that a standard orifice should be used unless:

- ❑ The velocity is such as to require the value of β to exceed 0.75: if higher differential pressure or larger pipe is not possible, he suggested considering a flow nozzle;
- ❑ The fluid contains a second phase of some sort – particulate, bubble, or droplet – in suspension: either a venturi or a flow nozzle with vertically downward flow, an eccentric or segmental orifice, or a different type of device such as a target meter or electromagnetic flowmeter should be considered;
- ❑ The fluid is very viscous or results in low Reynolds number flows: the meters already mentioned may be suitable, or a quadrant or semicircular orifice may be worth considering;
- ❑ The flow is very low and the fluid very clean: an integral orifice or a small calibrated meter run may be appropriate;
- ❑ Pumping costs are a major consideration: in which case go to a lower loss device.

If a liquid or a gas flows down a pipe of decreasing cross-section, the velocity (provided, in the case of a gas, that it is below the speed of sound) increases as the area decreases, and the pressure also decreases with the increase in velocity. This behavior allows the measurement of the change in pressure as a means of obtaining

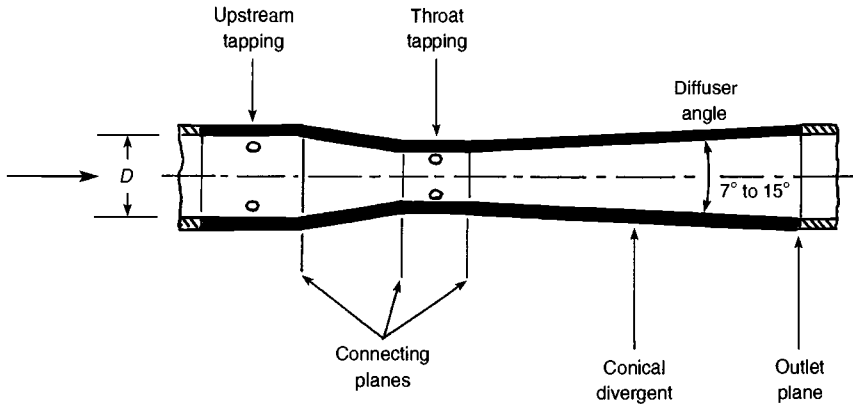


Figure 6.1. Classical venturi meter (after ISO 5167-1 1997).

the flow rate in the duct. If the duct has a smoothly varying cross-section and slowly changing area, the pressure change for a particular flow rate can be predicted from Equation (2.11). This is particularly so for the case of the venturi and the inlet to the standard nozzles, and in contrast to the standard orifice plate, the flow rate can be predicted to within about 1–1.5% as shown from experimental data.

The inlet flow will usually be turbulent and will approach the venturi (Figure 6.1) or nozzle (Figures 6.2–6.4) where an upstream pressure tapping (one at entry to the flowmeter) will measure the static pressure. The convergent flow at the inlet of the venturi or nozzles passes smoothly to the throat, except that it is possible that a small recirculation zone may exist upstream at the corner of nozzle entry. The downstream pressure tapping for the venturi is in the parallel throat section and is set in the wall. In the case of the nozzles, the downstream tapping is either in the throat or in the pipe wall. In the latter case, it senses the pressure of the jet as it leaves the outlet of the nozzle. Downstream of the throat in the venturi, there is a controlled diffusion, the small angle ensuring that good pressure recovery is achieved with a low total pressure loss. The losses at the outlet of the nozzles vary according to the particular design.

The comparison between the flow in the venturi (Figure 6.1), a duct of (relatively) smoothly changing area, and the flow through an orifice plate (Figure 5.1) is marked.

The essential equations governing the behavior of the venturi and the nozzles have been given in Chapter 2 (cf. Baker 1996), and many of the ideas are touched on in Chapter 5. In this chapter, we shall, as far as possible, avoid repetition.

6.2 ESSENTIAL BACKGROUND EQUATIONS

The equation relating mass flow to the differential pressure for venturi and nozzles, is

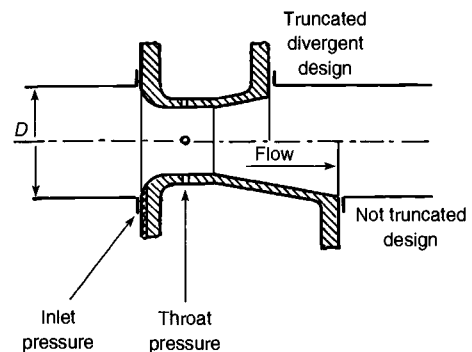


Figure 6.2. Venturi nozzle (after ISO 5167-1 1997).

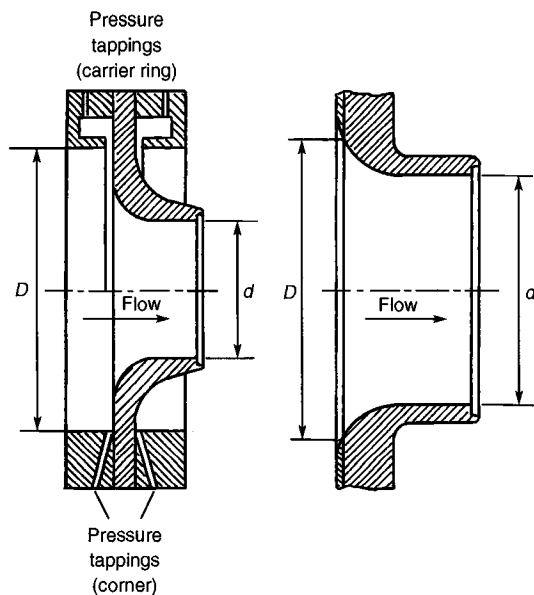


Figure 6.3. ISA 1932 nozzle (after ISO 5167-1 1997).

reproduced from Chapter 5:

$$q_m = \frac{CE\epsilon\pi d^2 \sqrt{2\rho_1 \Delta p}}{4} \quad (5.1)$$

where C is the coefficient of discharge, E is the velocity of approach factor $(1 - \beta^4)^{-1/2}$ where β is the diameter ratio d/D of orifice diameter to pipe internal diameter, ϵ is the expansibility (or expansion) factor, Δp is the differential pressure, ρ_1 is the density at the upstream pressure tapping cross-section, and q_v is related to q_m by Equation (5.2).

The expansibility (or expansion) factor ϵ is given in the standard for each device, venturi or nozzle, which should be consulted for details. The expressions differ from those in Chapter 5.

The discharge coefficients for these instruments are of the form

$$C = C_\infty + C_{Re} f(\beta, Re) \quad (6.1)$$

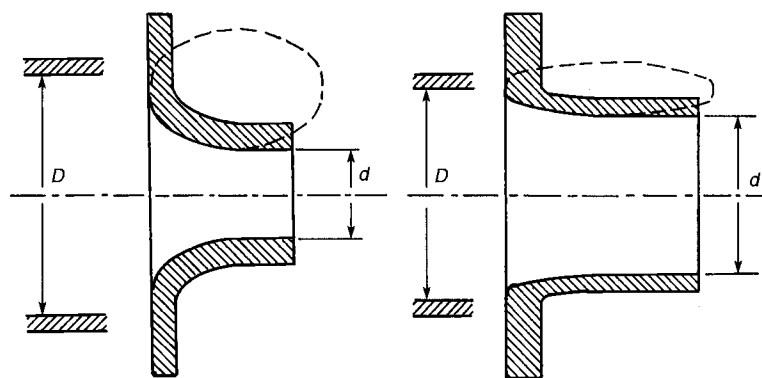


Figure 6.4. Long radius nozzle (after ISO 5167-1 1997).

Table 6.1. Coefficient of discharge for the classical venturi				
Type of Convergent	Constraints	C	Uncertainty in C (%)	
Rough-cast (or as-cast)	$100\text{ mm} \leq D \leq 800\text{ mm}$	0.984	0.7	
	$0.3 \leq \beta \leq 0.75$			
	$2 \times 10^5 \leq Re \leq 2 \times 10^6$			
Machined	$50\text{ mm} \leq D \leq 250\text{ mm}$	0.995	1.0	
	$0.4 \leq \beta \leq 0.75$			
	$2 \times 10^5 \leq Re \leq 1 \times 10^6$			
Rough-welded sheet iron	$200\text{ mm} \leq D \leq 1200\text{ mm}$	0.985	1.5	
	$0.4 \leq \beta \leq 0.7$			
	$2 \times 10^5 \leq Re \leq 1 \times 10^6$			

For the classical venturi meter, the coefficients are given in Table 6.1. It is important to note that, although the flow in the meter is a flow that approaches the ideal and results in a coefficient within about 1% of unity, and therefore very close to the result from Bernoulli’s equation, the uncertainty is greater than for the orifice. This reflects not on the stability of the flow but on the consistency with which the devices can be machined and the pressure tapings formed. The experience that has been amassed of the behavior of these instruments will also relate to this.

Shufang et al. (1996) reported tests outside the ISO limits with a 1,400-mm design using a maximum flow rate of 18,000 m³/h with a system uncertainty of 0.2% and for $1.85 \times 10^6 < Re < 5.78 \times 10^6$ obtained discharge coefficients:

for machined convergent 0.983 ± 0.003
for rough-welded 0.987 ± 0.006

The results for the rough-welded are essentially in line with the annex in the standard, which suggests no change for the rough-welded but an uncertainty of ±2%. However, for the machined convergent, the standard suggests a slight increase in C and uncertainty compared with the decrease reported by Shufang et al. (1996). For the rough-cast or as-cast version, the standard suggests that there appears to be no change for higher Reynolds numbers. There are likely to be changes, mainly decreases in C, for lower Reynolds numbers.

For the nozzles, the standard sets out the discharge coefficient formulas. The reader is referred to ISO 5167–1 for the complete formulas, the conditional requirements and the limits of applicability. For $\beta = 0.5$ and $Re = 10^6$, the terms in Equation (6.1) have values:

	C_∞	Second Term
for the venturi nozzle	0.9771	0
for the ISA 1932	0.9768	−0.00025
for the long radius	0.9965	−0.00462

The approximate uncertainty for each nozzle for a $\beta = 0.5$ is

for the venturi nozzle	1.3%
for the ISA 1932	0.8%
for the long radius	2.0%

Note that in all the equations, Re refers to the Reynolds number based on the upstream pipe internal diameter. However, Re_d based on the throat diameter is sometimes used in the standards, and the reader should be aware of this.

6.3 DESIGN DETAILS

As for the orifice plate, these are set out fully in ISO 5167, which provides the detailed requirements. The reader must refer to the standard to confirm that these are the latest values and to obtain all other details for each of these devices.

For the Classical Venturi Meter

The main features of the venturi design are shown in Figure 6.1. In addition the standard deals in matters such as the smoothness of the finish of the throat and the adjacent curvature. These are required to have $k/d < 10^{-5}$, where k is the roughness and d is the throat diameter. For the machined venturi, the entrance cylinder of length not less than D and the convergent section are required to have a finish to the same quality. The radius of curvature between the sections must be less than the following values:

- 0.25D between entrance cylinder and convergent section,
- 0.25d between the convergent section and the throat,
- 0.25d between the throat and the divergent.

In each case, the standard would prefer that the radius of curvature be zero. The requirements for the other two methods of fabrication of the venturi are also given in the standard.

Pressure tappings are shown in Figure 6.1, and the standard requires that four (at least) shall be provided at both the throat and the upstream positions and specifies their size, finish, and precise positions.

The value of the pressure loss across the venturi meter compared with the differential pressure between inlet and throat can be taken to be in the range 5–20%.

For the Venturi Nozzles

The nozzle is shown in Figure 6.2 and must have a roughness less than $10^{-4}d$. It can be made of any material provided it meets the requirements set out in the standard. Obviously the material must be stable enough to retain its shape and to resist erosion and corrosion by the fluid.

For the ISA 1932 Nozzles

The nozzle is shown in Figure 6.3 and has a roughness as for the venturi nozzle. Again, any unchanging material is acceptable.

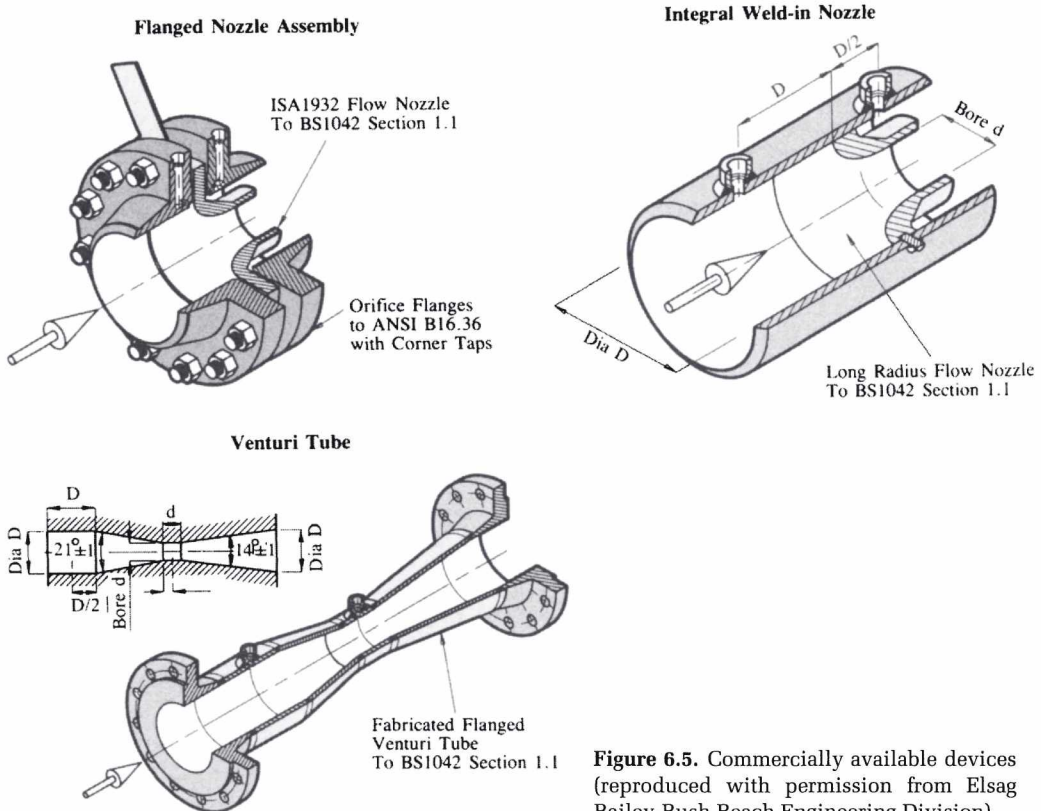


Figure 6.5. Commercially available devices (reproduced with permission from Elsas Bailey-Bush Beach Engineering Division).

For the Long Radius Nozzle

The nozzle is shown in Figure 6.4 and has the same roughness as the venturi nozzle. Again any unchanging material is acceptable.

6.4 COMMERCIALLY AVAILABLE DEVICES

The standards allow prospective users to design and make their own differential pressure device. However, various manufacturers also produce standard equipment. Figure 6.5 shows examples of such devices.

6.5 INSTALLATION EFFECTS

Upstream Fittings

Even though the values recommended by ISO 5167-1 : 1997 for nozzles and venturi nozzles are the same as those for the orifice plate, the values for the classical venturi are generally less. For instance, for a venturi with a diameter ratio of 0.5, the ISO allowance for an upstream bend is 1.5D, and for two or more in the same plane, it is 2.5D. These spacings have raised questions, and, as a result, NEL (1997) provided guidance on installation, which suggested that the ISO may need to be revised as

Table 6.2. Upstream straight lengths required for zero additional uncertainty (in diameters from the outlet of the downstream bend to the upstream pressure tapping plane) as for ISO (old) and as proposed by NEL (1997) (new)

β	Single 90° Bend ($R = 1.5D$)		Two 90° Bends Same Plane		Two 90° Bends Perpendicular Planes		Reducer	
							3:1 Over 3.5D	4:3 Over 2.3D
	Old	New	Old	New	Old ^a	New	Old	New
0.4	0.5	8	1.5	3	*	3	2.5	4
0.5	1.5	9	2.5	10	*	3	5.5	4
0.6	3.0	10	3.5	10	*	3	8.5	4
0.75	4.5	16	4.5	22	*	17	11.5	4

Note: At least four throat diameters should separate the throat tapping from downstream fittings.

^a Indicates that no values are given without 0.5% additional uncertainty.

indicated in Table 6.2. The guidance notes should be consulted for spacings where 0.5% additional uncertainty is relevant.

For the venturi, Ginesi (1991) suggested that the upstream lengths, to avoid calibration change, may be about half those for the orifice, rather than a ninth as suggested by the standards. This agrees with Table 6.2 for a single bend and two bends in the same plane, but there is less agreement for the other two columns. The most surprising values in Table 6.2 are those for two bends in different planes, to which most instruments are very sensitive.

Himpe et al. (1994) described new results for the influence of upstream bends on the discharge coefficient of classical venturi tubes, which suggest that the influence of disturbances does not necessarily decrease with decreasing bend angles, and the influence of the angle between the plane of the pipe bend and the position of the tapping is considerable.

Al-Khazraji et al. (1978) used an eccentric orifice upstream of a venturi to assess the error that results from the flow disturbance. The orientation relative to the single pressure tapping had a negligible effect, but a spacing of 5D reduced the error to about 1%. At 10D spacing, there was, again, negligible effect, showing the same trend as for a single bend with $\beta \leq 0.6$ in Table 6.2.

Two-Phase Flows

Baker (1991a) provides several references relating to the behavior of venturi meters in multiphase flows. Additionally, de Leeuw (1997, cf. 1994) proposed a new correlation for overreading in wet gas horizontal flow which appears to give results within 2% of the data. He claims that the improvement is, in part, because it allows for the gas velocity. Taking q_{tp} as the volumetric flow rate deduced from the differential pressure when liquid is present in the gas stream, q_l as the liquid volumetric flow rate, and q_g as the gas volumetric flow rate, the overreading ratio is given by

$$\frac{q_{tp}}{q_g} = \sqrt{1 + C_{tp}X + X^2} \quad (6.2)$$

where X is the Lockhart-Martinelli parameter and

$$C_{tp} = \left(\frac{\rho_l}{\rho_g} \right)^n + \left(\frac{\rho_g}{\rho_l} \right)^n \quad (6.3)$$

The liquid and gas densities are given by ρ_l and ρ_g and

$$n = 0.606(1 - e^{-0.746Fr_g}) \quad \text{for } Fr_g \geq 1.5 \quad (6.4)$$

and

$$n = 0.41 \quad \text{for } 0.5 \leq Fr_g < 1.5 \quad (6.5)$$

where the superficial gas Froude number is given by

$$Fr_g = \frac{V_{sg}}{\sqrt{gD}} \sqrt{\frac{\rho_g}{\rho_l - \rho_g}} \quad (6.6)$$

where V_{sg} is the superficial gas velocity, and the Lockhart-Martinelli parameter is given by

$$X = \frac{q_l}{q_g} \left(\frac{\rho_l}{\rho_g} \right)^{\frac{1}{2}} \quad (6.7)$$

The relationship is valid for gas densities above 17 kg/m³ and up to the liquid density, for gas Froude numbers above 0.5, and for X up to 0.3 (cf. Chisholm 1977).

6.6 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

Early work on the classical venturi by Herschel was aimed at use in the water industry, and it has continued to be used in measurement of water flows. Where energy conservation is an important consideration in large pumped flows, the low head loss of the venturi becomes attractive.

Rivetti et al. (1994) have described a quite different application of a small size venturi for liquid helium service, which was essentially a scaled down ISO 5167 design. Differential pressure must not be so high as to cause cavitation, and pressure ducts should have a diameter between 1/10 and 1/5 of the diameter of the respective venturi section and be four or more (an even number). Flow rates ranged from 0.5–4 g/s to 12.5–100 g/s for upstream/throat diameters of 6/2.6 and 16/12 mm, respectively, and differential pressures of 34–2,190 and 33–2,140 Pa, respectively.

The classical venturi has also been used for slurry and two-phase flows for which some data are available. Baker (1991a) has reviewed work on this application of the venturi. There appears to be growing interest in the use of venturi meters in these applications, although the behavior of the flow and the appropriate models of the multiphase flow may not be straightforward. One such example is the performance of the venturi meter in an oil-in-water emulsion in Figure 6.6 (Pal and Rhodes 1985).

The initial cost of the venturi is higher than the orifice because it is a much larger device to make, and precision in manufacture may not be easy. Various manufacturers offer ready-made designs (Figure 6.5).

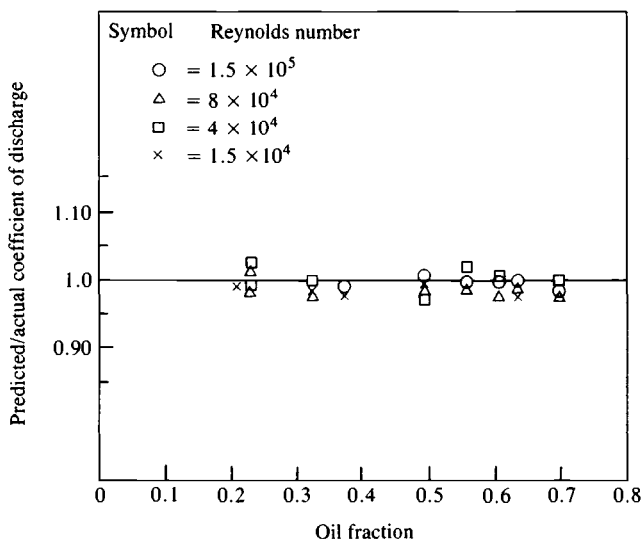


Figure 6.6. Performance of a venturi meter in an oil-in-water emulsion (Pal and Rhodes 1985; reproduced with permission from BHR Group).

The nozzles tend to be more stable than the orifice for high temperatures and high velocities, experiencing less wear (due to the smooth inlet contour as opposed to the sharp orifice upstream edge) and being less likely to distort. They are particularly applicable to steam flows, where they have been widely used.

6.7 CHAPTER CONCLUSIONS

This chapter has been, necessarily, brief because there appears to be less significant new material in the literature to add to the advice and guidance of the standards. However, there does appear to be an increased interest by the oil and gas industry in using venturi meters for both single and multiphase (especially wet gas) metering, and as a result some new work, which appears to be yielding some unexpected results, is taking place (Jamieson et al. 1996).

- It should be noted that the coefficient for the ISA 1932 nozzle has been slightly altered in the standard.
- Jamieson et al. (1996) observed discharge coefficients for a 150-mm (6-in.) diameter venturi in high pressure air (70 bar) at Re up to 8×10^6 and throat velocities up to 125 m/s, which were several percent higher than those obtained in water calibration. The reason appears to lie in the tapping chambers and the possible acoustic effects there.
- Computational fluid dynamics (CFD) work is being developed (cf. Sattary and Reader-Harris 1997) for venturi and nozzles. For venturi with $\beta = 0.4, 0.6$, and 0.75 , CFD tended to underestimate relative to experimental and ISO by up to about 1.2%. Sattary and Reader-Harris (1997) suggest that the difference is due to the actual method of measuring static pressure compared with the CFD model.

They also looked at compressibility effects. There is likely to be growing interest in developing the CFD work to explore the likely ultimate precision of these instruments.

- d. An inside-out nozzle or short venturi in the form of a bullet-shaped target meter could yield a highly predictable meter with direct electrical output via a force sensor. This could be combined with wall differential pressure measurements to provide a condition-monitoring function.

Critical Flow Venturi Nozzle

7.1 INTRODUCTION

This meter uses a fascinating effect that occurs when gas flows at very high velocity through a nozzle. As the gas is sucked through the nozzle, the velocity increases as the cross-section of the nozzle passage decreases toward the throat. At the throat, the maximum achievable velocity is sonic – the speed of sound. Downstream of the throat, the velocity will either fall again returning to subsonic or will rise and become supersonic. In normal operation of the nozzle, the supersonic region is likely to be small and to be followed by a shock wave that stands across the divergent portion of the nozzle and causes the gas to drop, very suddenly, from supersonic to subsonic. The existence of a shock wave does not mean that one will hear a “supersonic bang”! Such bangs are usually caused by moving shock waves carried forward by high speed aircraft.

The fascinating effect of sonic conditions at the throat is that changes in the flow downstream of the throat have no effect on conditions upstream. The sonic or critical condition, as it is called, appears to block any information that is trying to penetrate upstream. A simple picture of this is that the messengers carrying such information travel at the speed of sound and so are unable to make any headway over the fast flowing gas stream.

Two important effects of sonic conditions are that

- i. the mass flow rate is a function of the gas properties, the upstream stagnation temperature and pressure, and the area of the throat, provided that the nozzle is actually running at critical conditions (which can be ascertained by checking the downstream pressure);
- ii. the nozzle acts as a flow controller, creating steady conditions upstream even though conditions downstream are unsteady.

Figure 2.6(a) shows a simple illustration of a convergent-divergent nozzle with a throttle valve downstream. Figure 2.6(b) shows the variation of pressure through the nozzle, and the critical flow conditions are those bounded by curves *c* and *i* for which the flow becomes sonic at the throat.

For a fuller description of the fluid flow behavior, see Baker (1996). Two essential equations apply when the velocity at the throat is equal to the speed of sound (Mach number equals unity) and the conditions are critical. The first equation gives

the pressure ratio for choked flow [Equation (2.18)]:

$$\frac{p_*}{p_0} = \left(\frac{\gamma + 1}{2} \right)^{-\gamma/(\gamma-1)} \quad (7.1)$$

where p_* is the pressure at the throat, p_0 is the stagnation pressure, and γ is the ratio of specific heats c_p/c_v .

The second equation gives the ideal mass flow q_m for choked flow from Equation (2.18), making use of the relationships

$$\mathbf{R}/\mathbf{M} = c_p - c_v$$

and

$$\sqrt{\frac{\gamma}{c_p(\gamma - 1)}} = \frac{1}{\sqrt{\mathbf{R}/\mathbf{M}}}$$

to obtain

$$q_m = \frac{A_* p_0}{\sqrt{T_0(\mathbf{R}/\mathbf{M})}} \sqrt{\gamma} \left(\frac{2}{\gamma + 1} \right)^{\frac{1}{2}(\gamma+1)/(\gamma-1)} \quad (7.2)$$

where $\mathbf{R}$ is the universal gas constant (8.3143 kJ/kmolK), $\mathbf{M}$ is the molecular weight, c_p and c_v are the specific heats at constant pressure and constant volume, A_* is the throat cross-sectional area, and T_0 is the stagnation temperature.

Thus if we can measure upstream stagnation pressure and temperature p_0 and T_0 , and if we know the throat area A_* and the gas properties, we appear to have a mass flowmeter. The flow must be choked. Also this equation is limited to a perfect gas and assumes an idealized and one-dimensional flow.

We are interested in the use of this device for real gases. For precise work, we have to recognize that gases are NOT perfect. We also find that boundary layer growth and curvature of the flow through the nozzle result in a flow that is not truly one-dimensional.

Thus as with many flowmeters, there is a coefficient of discharge, to be discussed later. First, we consider a practical flowmeter installation.

7.2 DESIGN DETAILS OF A PRACTICAL FLOWMETER INSTALLATION

The critical flow venturi nozzle is fully specified by the most recent version of the standard document BS EN ISO 9300: 1995. The reader is strongly advised to obtain a copy of the standard for reference if planning to build a facility using this device and to view this chapter as complementary to the standard.

Figure 7.1 shows a diagram of a critical flow venturi nozzle installation. Note that the temperature tapping can be upstream of flow conditioners because the temperature change is likely to be negligible. However, the pressure tapping must be sensing a true inlet stagnation value. There may also be a downstream measure of pressure to ensure that critical conditions are maintained.

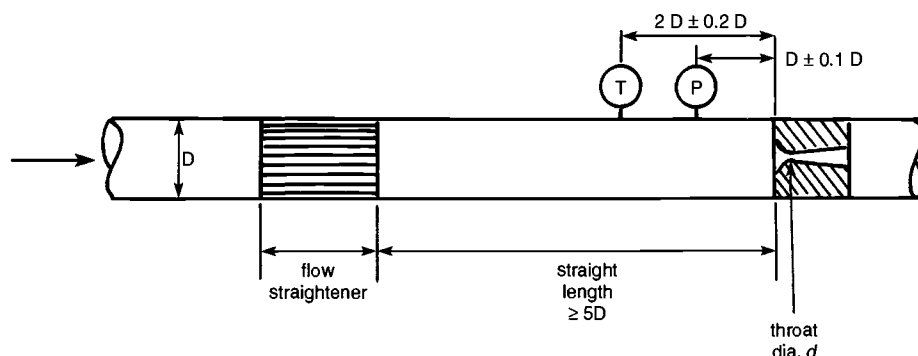


Figure 7.1. Details of venturi nozzle installation.

If the upstream pressure allows only limited variation, then the range of q_m will be limited, and only by using a bank of nozzles of different areas can we achieve a satisfactory calibration range (Figure 7.2). Caron (1995) mentioned a test stand that used a bank of sonic nozzles in binary sequence. Nozzle 2 has twice the flow rate of nozzle 1, etc.

Nozzles may have a convergent portion only; in which case there will be a substantial pressure loss at exit, and the pressure ratio of outlet-to-inlet stagnation must be less than 0.528 for air. On the other hand, the convergent-divergent venturi nozzle in Figure 7.3 may achieve a pressure ratio (exit/inlet) for critical conditions at the throat of $p_2/p_0 \simeq 0.9$. The one shown is the toroidal throat venturi nozzle. An alternative, not favored by some, is the cylindrical throat type where a constant area throat of one diameter in length separates inlet contraction and outlet diffuser.

The diagram in the standard provides the details of design finishes and tolerances that differ slightly from, but cover, those in Figure 7.3. It is also important to note other features of the nozzle in Figure 7.3. The inlet pipe diameter, D , must be greater than $4d$, and referring to Figure 7.1 with a pipe diameter of 50 mm upstream, this would put a maximum on the throat diameter of

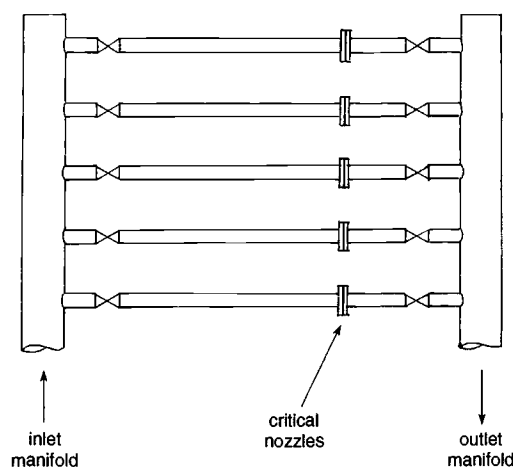


Figure 7.2. Bank of critical flowmeters.

12.5 mm. Also the preferred distance to the upstream pressure tapping is between $0.9D$ and $1.1D$. Other spacings are allowed provided that it can be shown that the tapping gives a correct measure of the nozzle inlet stagnation pressure. The standard allows for a larger space upstream with its own requirements.

The standard also sets out requirements on temperature measurement that cover the position in Figure 7.1 but allow a greater spacing if it gives, reliably, the stagnation temperature at nozzle inlet. The standard also warns that care needs to be taken if the surroundings of the installation differ in temperature by more than 5°C from the gas.

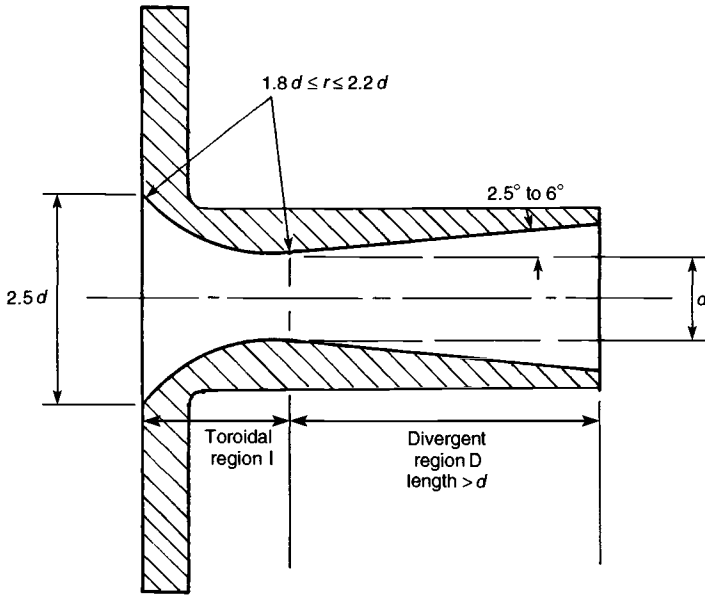


Figure 7.3. Details of toroidal throat venturi nozzle.

Where there is the possibility of condensate, drain holes are allowed provided there is no flow through them when flow measurement is taking place, and they are spaced upstream of the pressure tapping by at least D and not in the same axial plane as the pressure tapping.

The standard requires that the nozzle be made from material capable of finish to the required smoothness, resistant to corrosion in the intended application, and dimensionally stable so that temperature corrections can be used for throat area.

7.3 PRACTICAL EQUATIONS

The practical use of critical flow venturi nozzles is set out in BS EN ISO 9300: 1995. Apart from some nomenclature differences, this section gives the method set out in the standard. Using the expression of Equation (7.2), we introduce a discharge coefficient C to allow for the deviation from the ideal equation, and we also introduce a coefficient for the gas C_* , the critical flow function, obtainable from tables and formulas, and for a perfect gas written as

$$C_{*i} = \sqrt{\gamma} \left(\frac{2}{\gamma + 1} \right)^{\frac{1}{2}(\gamma+1)/(\gamma-1)} \quad (7.3)$$

With these coefficients, we can rewrite the equation as

$$q_m = \frac{A_* C C_* p_0}{\sqrt{T_0 (\mathbf{R}/\mathbf{M})}} \quad (7.4)$$

Alternatively, the equation may be written in terms of p_0 and ρ_0

$$q_m = A_* C C_R \sqrt{p_0 \rho_0} \quad (7.5)$$

where

$$C_R = C_* \sqrt{Z_0} \quad (7.6)$$

and

$$\frac{p_0}{\rho_0} = \frac{Z_0 \mathbf{R} T_0}{\mathbf{M}} \quad (7.7)$$

In this case, we require values of C_* and Z_0 , the compressibility factor at stagnation conditions.

The design should, preferably, ensure that the values of p and T are obtained where the Mach number is small enough to assume that they are equal to p_0 and T_0 . Otherwise, the standard gives equations to obtain the corrected values.

In order to ensure that the nozzle operates critically, the standard requires that the actual maximum outlet pressure $p_{2\max}$, when compared with the ideal value p_{2i} , for which critical conditions are achievable satisfies

$$p_{2\max} - p_* = 0.8(p_{2i} - p_*) \quad (7.8)$$

Using tables such as those in Baker (1996), it is possible to obtain values of p_{2i} for various values of outlet area A_2 , assuming that the gas behaves sufficiently closely to an ideal gas to use the ratio of the specific heats γ . Figure 7.4 shows the values of $p_{2\max}/p_0$ for $\gamma = 1.4$. The standard provides a plot for other values of the isentropic exponent.

Note that γ is the ratio of specific heats for an ideal gas, and κ is referred to in the standard as the isentropic exponent and applies to real gases. I have avoided using the latter in this chapter. For further information, consult Bean (1971) who relates the value (cf. Miller 1996, Sullivan 1979) of κ to γ via an expression (which he does not derive) that recognizes that Z is constant for an ideal gas but varies for a real gas. Miller (1996) also gives plots of κ . Where there is a divergence between κ and γ , the standard indicates which to use.

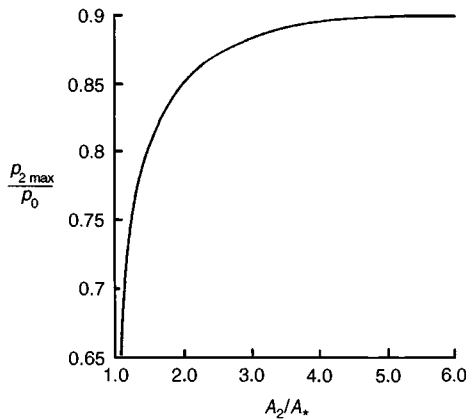


Figure 7.4. Maximum values of back pressure for $\gamma = 1.4$ to ensure critical operation based on the criterion in BS EN ISO 9300: 1995.

It is relatively simple to relate the area ratio between the throat and the outlet based on design angles of the diffuser. If the half-angle of the diffuser is θ , the radius of toroid is r and the length of the diffuser is l beyond the toroid, the increase in diameter is (Figure 7.5) $2[r(1 - \cos \theta) + l \tan \theta]$ so that the outlet diameter

$$d_2 = d + 2[r(1 - \cos \theta) + l \tan \theta]$$

or

$$\frac{d_2}{d} = 1 + \frac{2r(1 - \cos \theta)}{d} + \frac{2l \tan \theta}{d} \quad (7.9)$$

from which the expression in the standard for A_2/A_* can be simply obtained.

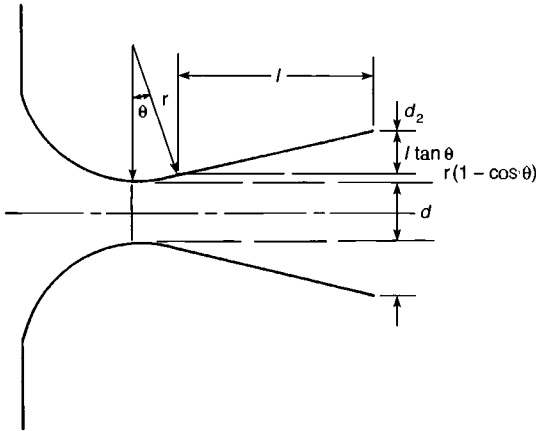


Figure 7.5. Geometrical calculation to obtain the outlet diameter from the dif-fuser geometry.

7.4 DISCHARGE COEFFICIENT C

The data on which the discharge coefficient is based include work by Smith and Matz (1962), Arnberg et al. (1973), Brain and MacDonald (1975), and Brain and Reid (1978, 1980). Figure 7.6 shows the scatter in the data. The general equation that gives a fit to this data is

$$C = a - b \text{Re}_d^{-n} \quad (7.10)$$

and the value of the constants, including the preferred ISO values, are given in Table 7.1. Figure 7.6 shows that the values fall well within an uncertainty band of $\pm 0.5\%$ at 95% confidence level around the BS EN ISO 9300 preferred equation.

The values of b and n are affected by the nature of the boundary layer in the throat. Transition is likely to cause a change in these parameters. Miller (1996) also suggested that the temperature gradient between gas and nozzle may affect b .

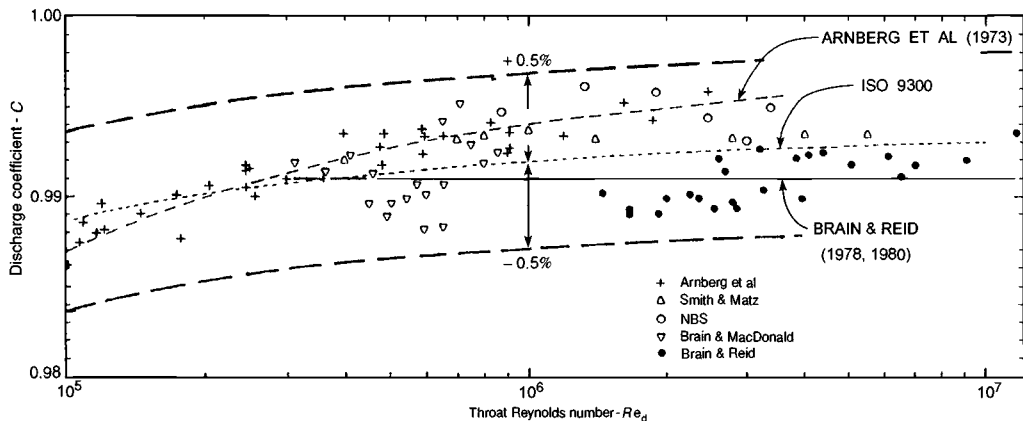


Figure 7.6. Discharge coefficient for toroidal throat venturi nozzles (after Brain and Reid 1978 with permission from NEL).

Table 7.1. Values of constants in equation (7.10) for toroidal throat venturi nozzles^a (equations are plotted in Figure 7.6)

Re_d Range	<i>a</i>	<i>b</i>	<i>n</i>	Reference
$4 \times 10^4 - 3 \times 10^6$	0.99738	3.058	0.5	Arnberg et al. (1973)
$3 \times 10^5 - 10^7$	0.99103	0.0	—	Brain and Reid (1978, 1980)
$10^5 - 10^7$	0.9935	1.5250	0.5	BS EN ISO 9300:1995

^a Other values are available for cylindrical throat venturi nozzles but are not given here.

Nakao et al. (1996) gave variations of the discharge coefficient with Reynolds number and came up with discharge coefficients for a venturi with throat diameter of 0.301 mm of

$$C = 1.006 - 3.783\text{Re}_d^{-0.5} \quad (7.11)$$

and for a venturi with throat diameter of 0.501 mm of

$$C = 1.007 - 3.195\text{Re}_d^{-0.5} \quad (7.12)$$

with uncertainties of about $\pm 0.1\%$. These lie outside the $\pm 0.5\%$ limits in Figure 7.6.

Park (1995) provided experimental evidence in support of the standard but made the point that, if the inlet radius is small, the discharge coefficient may decrease because of separation in the inlet causing a divergence from the standard.

7.5 CRITICAL FLOW FUNCTION C_*

Various gases have values of C_* tabulated in the standard (cf. Miller 1996). Values are obtainable for nitrogen, oxygen, argon, methane, carbon dioxide, air for -50 to 100°C , and steam for 100 – 600°C , and for 0 – 100 bar. A sample of these is given in Table 7.2 for air. These should be compared with the value for a perfect gas with $\gamma = 1.4$ of $C_{*i} = 0.685$.

For gas mixtures, a method (cf. Johnson 1970, 1971 and Miller 1996) is given in the standard. This first uses an expression for the critical flow function for a real gas

$$C_R = C_* \sqrt{Z_0} = a_c f + b_c \quad (7.13)$$

Table 7.2. Sample values of C_* for air (Miller 1996 and BS EN ISO 9300:1995)

Temp. (°C)	Pressure (bar)				
	0	10	20	50	100
0	0.6850	0.6887	0.6925	0.7038	0.7224
50	0.6849	0.6870	0.6889	0.6949	0.7039
100	0.6847	0.6858	0.6869	0.6900	0.6946

where b_c represents the contribution of methane, and $a_c f$ represents the contribution of other components. The square root of the compressibility factor is given by

$$\sqrt{Z_0} = a_z f + b_z \quad (7.14)$$

a_c , b_c , a_z , and b_z are tabulated against temperature and pressure, and f is obtained from

$$f = X(\text{C}_2\text{H}_6) + 2X(\text{C}_3\text{H}_8) + 3X(\text{C}_4\text{H}_{10}) - \frac{1}{2}X(\text{N}_2) + X(\text{CO}_2) \quad (7.15)$$

Limits are set to the mole fraction X of each component, and the value of f is zero for methane. The uncertainty resulting from this formula is estimated for methane at 70 bar as $\pm 0.25\%$ and for a maximum value ($f = 0.2$) as $\pm 0.45\%$.

Weberg (1990) has developed a new critical mass flow correlation for 0–180 bar and 280–340 K. This was based on critical mass flow rates computed from the AGA-8 (1992) equation and from Aly and Lee's (1981) ideal gas specific heat correlation. Average deviation of the correlation from the data was 0.2% (cf. Starling 1994 on hydrocarbon mixture equations of state).

7.6 DESIGN CONSIDERATIONS

Referring to Figures 7.3 and 7.7, some details are given from the standard (cf. ISO 9300 for full details). Wall smoothness, curvature, and axial distances from inlet plane are defined. The inlet plane is that plane perpendicular to the axis, which cuts the inlet toroid at a diameter of $2.5d$.

Swirl and other upstream disturbances (cf. Yoo et al. 1993) may need to be eliminated with a straightener. Drain holes to remove condensation may also be necessary to avoid an accumulation of water reducing the upstream plenum.

These points are summarized here.

Region I

- Average roughness $\leq 15 \times 10^{-6}d$
- Free from deposits of any sort
- Radius of toroid (in plane through the axis) $1.8d \leq r \leq 2.2d$
- Contour toroidal to $\leq 0.001d$

Region D

- Diffuser half-angle between 2.5° and 6°
- Length of diffuser from toroid at least d
- No discontinuities exceeding 1% of local diameter
- Average roughness $\leq 10^{-4}d$

Inlet Plane

- Defined by $2.5d \pm 0.1d$ intercept

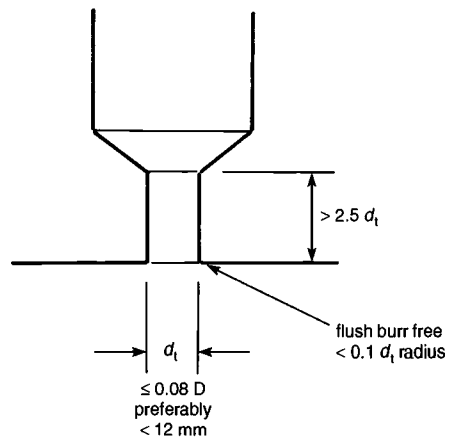


Figure 7.7. Details of pressure tap.

Swirl free

- Inlet flow to be swirl free, if necessary, by use of a straightener at $>5D$

 β Ratio

- either $\beta = d/D$ to be less than 0.25 for a circular inlet pipe or there must be a large space with $>5d$ in all directions from the axis.

Drain Holes

- $<0.06D$ at greater than one diameter upstream of p_0 tapping and away from the plane of tapping

The design of the pressure tapplings is shown in Figure 7.7 where d_t is the diameter of the tapping. The inlet tapping should be situated $0.9\text{--}1.1D$ upstream of the inlet plane, or in a large inlet chamber it should be within $10d \pm 1d$ from the inlet plane.

The downstream pressure tapping (if required) should be less than 0.5 times the conduit diameter from the nozzle exit plane. In some cases, downstream pressure will be known (e.g., when the nozzle exhausts to the atmosphere).

The inlet temperature tapping should be positioned to give a reliable measure of stagnation temperature.

The accuracy of the method will depend heavily on the accuracy of measurement of p_0 . Takamoto et al. (1993a) showed that cross flows can cause errors due to the lack of a measure of stagnation. If we cannot assume that the flow is negligible in the upstream pipe, then p_0 cannot be directly measured at the wall tapping, and we shall have to deduce it from p_1 using

$$\frac{p_0}{p_1} = \left(1 + \frac{\gamma - 1}{2} M_1^2\right)^{\gamma/\gamma-1} \simeq 1 \quad (7.16)$$

where M_1 will be small, but the additional step is likely to reduce the precision of the final measurement. Similar considerations apply to T_0 . It is, therefore, preferable to ensure that stagnation conditions do exist in the inlet plenum.

Pereira et al. (1993) suggested making the nozzle throat and diffuser in two parts with an O-ring between and discussed the effect of a step between throat and diffuser, which they found was not critical within certain limits.

7.7 MEASUREMENT UNCERTAINTY

The equation for the uncertainty summation [cf. Equation (5.26) for alternative equivalent symbols] is

$$\epsilon(q_m) = \pm \left[\epsilon(A_*)^2 + \epsilon(C)^2 + \epsilon(C_*)^2 + \epsilon(p_0)^2 + \frac{1}{4}\epsilon(\mathbf{M})^2 + \frac{1}{4}\epsilon(T_0)^2 \right]^{1/2} \quad (7.17)$$

If we apply some of the possible uncertainties mentioned earlier, we have

$\epsilon(C) = \pm 0.5\%$	based on the values for the ISO equation
$\epsilon(C_*) = \pm 0.45\%$	as given for a value for $f = 0.2$ (but may be too optimistic)

$\epsilon(A_*) = \pm 0.8\%$	for a 5-mm diameter throat would require 0.010-mm uncertainty in throat diameter
$\epsilon(\mathbf{M})$	guess $\pm 0.5\%$
$\epsilon(p_0)$	guess $\pm 0.1\%$
$\epsilon(T_0)$	guess $\pm 0.2\%$

With these values, we obtain an overall uncertainty of

$$\epsilon(q_m) = \pm 1.18\%$$

Caron (1995) gave some details of the critical nozzle test stand designed and used by the Ford Company to calibrate air-flow sensors. The ASME Standard toroidal design was selected, and it was reckoned that after calibration an uncertainty of $\pm 0.25\%$ in the coefficient could be achieved. The paper gives the following values for uncertainties (the area is calibrated out):

Source	Uncertainty	Sensitivity	
		Coefficient	Product
Pressure	$\pm 0.05\%$	1	$\pm 0.05\%$
Temperature	$\pm 0.37\%$	0.5	$\pm 0.185\%$
Discharge coefficient	$\pm 0.25\%$	1	$\pm 0.25\%$
Critical flow function	$\pm 0.0125\%$	1	$\pm 0.0125\%$
Combined uncertainty			$\pm 0.315\%$

Because this omits, due to calibration, the measurement of the throat area and uses higher precision values, it is not out of line with the value obtained earlier.

7.8 EXAMPLE

Using the data given in this chapter, we shall calculate the mass flow of air q_m for a throat diameter d of 6 mm, inlet stagnation pressure p_0 of 2 bar, and inlet stagnation temperature T_0 of 50°C , and we shall calculate the outlet area necessary to allow operation with a back pressure p_2 up to 1.6 bar.

In this example, we are assuming a venturi nozzle conforming to the preceding requirements. We need to calculate the value of Re_d , and for this we require the throat velocity. At critical conditions, $M = 1$, and the velocity is that of sound speed (cf. Baker 1996)

$$c = \sqrt{\gamma(\mathbf{R}/\mathbf{M})T_*} \quad (7.18)$$

Because at $M = 1$ using Equation (2.15),

$$\begin{aligned} \frac{T_*}{T_0} &= \frac{2}{\gamma + 1} \\ c &= \sqrt{\frac{2\gamma}{\gamma + 1} \frac{\mathbf{R} T_0}{\mathbf{M}}} \end{aligned}$$

and because $T_0 = 323.15 \text{ K}$, and for air $\gamma = 1.4$ and $\mathbf{R}/\mathbf{M} = 8.3143/29.0 \text{ kJ/kgK} = 0.2867 \text{ kJ/kgK}$, we obtain $c = 328.8 \text{ m/s}$.

Taking $\nu = 1.3 \times 10^{-5} \text{ m}^2/\text{s}$, we can obtain the Reynolds number for the throat as

$$\begin{aligned}\text{Re}_d &= 328.8 \times 0.006 / 1.3 \times 10^{-5} \\ &= 1.52 \times 10^5 \text{ (within ISO limits)}\end{aligned}$$

Note from Figure 7.6 that small variations of Re_d should have little effect on the value of the discharge coefficient.

Using Table 7.1

$$\begin{aligned}C &= 0.9935 - 1.525 \text{Re}_d^{-0.5} \\ &= 0.9896\end{aligned}$$

Using Table 7.2 at $T_0 = 50^\circ\text{C}$ and $p_0 = 2 \text{ bar}$,

$$C_* = 0.6853 \quad (\text{cf. } 0.685 \text{ using the perfect gas value})$$

From Equation 7.4,

$$\begin{aligned}q_m &= \frac{A_* C_* p_0}{\sqrt{\mathbf{R}/\mathbf{M} T_0}} \\ A_* &= \pi d^2/4 = 2.827 \times 10^{-5} \text{ m}^2\end{aligned}$$

So

$$q_m = 0.01260 \text{ kg/s}$$

To obtain the outlet area, we first obtain the value of the outlet Mach number from Equation (2.45) for ideal flow through the nozzle

$$\begin{aligned}\frac{p_2}{p_0} &= 0.8 = \left(1 + \frac{\gamma - 1}{2} M_2^2\right)^{\gamma/(1-\gamma)} \\ M_2^2 &= (0.8^{-0.4/1.4} - 1) \frac{2}{0.4} \\ M_2 &= 0.574\end{aligned}$$

Then from Equation (2.18),

$$\begin{aligned}\frac{A_2}{A_*} &= \frac{1}{M_2} \left(\frac{2}{\gamma + 1} + \frac{\gamma - 1}{\gamma + 1} M_2^2 \right)^{\frac{1}{2}(\gamma+1)/(\gamma-1)} \\ \frac{A_2}{A_*} &= 1.22 \quad A_2 = 3.45 \times 10^{-5} \text{ m}^2 \\ d_2 &= 6.63 \text{ mm}\end{aligned}$$

This outlet area has assumed a curve like c in Figure 2.6(b). We have ignored what happens between curves c and d . In fact, the flow ceases to be without loss. If the pressure at outlet lies between c and d , the flow will follow d out of the throat before

passing through a sudden pressure increase known as a shock wave. Downstream of this sudden change, the flow will follow a curve below, but not dissimilar to, *c*.

For our purposes, therefore, it will be advisable to ensure that the outlet diameter is larger than 6.63 mm (as in Figure 7.4) so that the throat is definitely running at sonic velocity. ISO 9300 recommends designing for an outlet pressure recovery 80% of the ideal [Equation (7.8)].

7.9 INDUSTRIAL AND OTHER EXPERIENCE

The use of the nozzle in saturated steam with a dryness fraction down to 84% requires a wet steam correction factor (Amini and Owen 1995, cf. ASME/ANSI MFC-7M 1987). However, Amini and Owen suggest that a more practical approach might be to precede the nozzle with a steam-water separator, assume the steam to be dry saturated, and accept an uncertainty in the mass flow of 3%. An uncertainty of 1% arises from the lack of perfect separation.

The Gas & Fuel Corporation of Victoria, Australia, has a gas meter test facility with 11 critical nozzles (Wright 1993), has been in use for 10 years and has a range of 6–5,600 m³/h for meters from 80 to 450 mm, and another facility with six nozzles, which has been in use for 5 years and has a range of 2.4–30 m³/h. The coefficient has $\pm 0.5\%$ uncertainty. The flow rate sequence is that the flow rate of each nozzle is close to the sum of flows through the next three smaller nozzles. This is achieved by using a flow rate ratio for each nozzle compared to the one below of 1.839:1. Wright claimed that the use of two nozzles to match the higher flow rate requires more nozzles to cover the range. Wright also mentioned a Wheatstone bridge technique to calibrate nozzles, and an American quantity called standard time, t_s , used by the American Meter Company and equal to the time in seconds to pass one cubic foot at an inlet pressure of 24.696 psia (10 psig) and an inlet temperature of 60.0°F.

Nakao et al. (1996) described a calibration system for sonic nozzles that used the weight of a collecting vessel to obtain the flow rate. Four lines were available for the venturi nozzles, and the gas was drawn through with a vacuum pump until the required flow conditions were established. The flow line was then changed to the vessel, which had been evacuated and weighed, and the flow continued until a sufficient mass had been collected, when the diversion time ended. Care was needed to allow for dead volumes and the change-over mechanisms. They claimed an overall uncertainty for the rig of about $\pm 0.1\%$.

Bosio et al. (1990) observed a lack of repeatability during gravimetric primary calibration of nozzles in natural gas containing approximately 3 mg/m³ of hydrogen sulfide. They identified the problem as deposition of sulfur in the throat. Chesnoy (1993) also discussed the problem of sulfur deposition in the nozzle throat, causing change in nozzle performance. To reduce the possibility of sulfur formation either the content of H₂S needs to be reduced or the oxidant needs to be eliminated. If this is not possible, work supported by Chevron suggested that the position of the deposition moved downstream due to temperature increase (Mottram and Ting 1992). Chesnoy claimed that K-Lab's primary calibration was within $\pm 0.3\%$ between 20 and 100 bar. A zinc oxide bed had been used to treat the inlet gas successfully up to 100 bar (absolute), but other actions were under consideration above this pressure.

Bignell (1996a) described comparison techniques for the performance of a number of small sonic nozzles, and Kim and O'Neal (1995) used critical flow models to estimate two-phase flow of HCFC22 and HFC134a through short tube orifices.

Nozzles with throat diameters of 10–1,000 μm have been reported (Takamoto 1996). Recent calculations of discharge coefficient using finite elements (Wu and Yan 1996) gave very impressive agreement with ISO.

7.10 ADVANTAGES, DISADVANTAGES, AND APPLICATIONS

The advantages of this device are that it allows mass flow measurement of gas and ensures that, provided the flow is choked and, therefore, operating in its correct regime, flows upstream are not affected by changes in downstream conditions.

It is, however, very inflexible in that flow rate can be increased only by changing the upstream stagnation values or by increasing the throat area. This leads to the requirement for a bank of carefully sized nozzles to span the working range.

Its primary application is in gas flow measurement of high precision, particularly in flow calibration.

7.11 CHAPTER CONCLUSIONS

The critical flow venturi nozzle appears to have the potential for very high precision. It has been suggested that the ISO uncertainty figure is too great (cf. Gregor et al. 1993).

It is a meter where upstream conditions should be capable of control and precise replication so that the profile and turbulence distribution approaching the meter is always the same.

There are clearly sources of reduced accuracy (e.g., deposits) and possible humidity or small quantities of contaminants in the flow.

It also should be amenable to CFD modeling with increasing precision. The precise positioning, design, and consequent effect of pressure and temperature tappings may be amenable to such modeling. Such an approach may suggest, where possible, a most repeatable and most stable overall layout for the upstream pipe, conditioners, etc.

Another direction in which experimental and theoretical studies could go is in the miniaturization of nozzles.

CHAPTER 8

Other Momentum-Sensing Meters

8.1 INTRODUCTION

The previous chapters have been mainly concerned with devices, the designs of which have been defined by various standard specifications. In this chapter, we consider other mainly proprietary meters, which depend on the changing flow pattern and which essentially sense momentum of the flow. The output is in some cases given by a differential pressure measurement and in others by a position or force measurement. This means that the user must be aware of the possible effect of density on the results. Viscosity will also affect some readings.

The somewhat arbitrary order of consideration and the uneven amount of detail in this chapter reflect the current importance of the devices, the available literature, and my experience. Thus the variable area meter takes up a large proportion of the chapter.

The meters to be considered in this chapter are

- ❑ variable area (VA) meters, which depend on gravity to oppose the movement of the float and consist of two main types:
 - those with a tapered tube and a float with a fixed metering edge and
 - those with an orifice with a fixed metering edge and a moving tapered plug;
- ❑ spring-loaded profiled plug in an orifice;
- ❑ target (drag plate) meter sometimes spring loaded;
- ❑ venturi-type meters claiming a low loss, such as Dall, Epiflo, Gentile and Low-Loss;
- ❑ wedge, V-cone meters;
- ❑ meters using a bypass with an oscillating vane, a Pelton, or a VA meter in it;
- ❑ slotted orifice meter;
- ❑ pipework features used as meters such as inlets and bends;
- ❑ averaging pitots under various names (Annubar, Torbar, etc.); and
- ❑ laminar (viscous) flowmeters.

8.2 VARIABLE AREA METER

The variable area (VA) meter is sometimes known as a Rotameter, but this is a trade name of a particular company and I shall, therefore, refer to it as the variable area

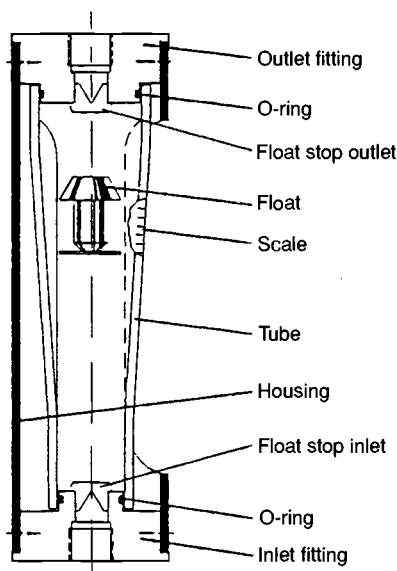


Figure 8.1. Variable area flowmeter (reproduced with permission of Bailey-Fischer and Porter).

meter. In the following sections on the VA meter, I have avoided history, theory, or other items, which have been consigned to Appendix 8.A. The reader may also find practical value in the very useful treatment by Hogrefe et al. (1995). One way to view the VA meter is as a variable orifice meter with a fixed pressure drop resulting from the weight of the float.

8.2.1 OPERATING PRINCIPLE AND BACKGROUND

The VA meter consists of a float that rises up a conical tube due to the upward flow (Figure 8.1). *Float* is a misleading term because it rises not from any buoyant effect but as a result of the upward drag of the fluid, which, in turn, results in a pressure difference across the float. As the flow increases, the float rises higher in the tube to a point where the annulus formed between

the float and the conical tube wall is sufficient to allow the drag on the float to balance the float's weight. The flow is then deduced from the height to which the float has risen. (In Germany, the term *swimmer* is used rather than *float*, which may be more appropriate, but I shall stick with *float* in this book.)

8.2.2 DESIGN VARIATIONS

Coleman (1956) provided useful information on the development of the float and tube. However, we will concentrate mainly on the current practice. Figure 8.2 shows some of the shapes of floats and of a tube.

Tubes

Tubes are often made of borosilicate glass. In addition to the plain taper tubes, there are tubes with three longitudinal beads molded into the glass (beaded or fluted glass) to guide the tail of the float (rib-guided). For very low flow rates a conical tube with a plumb-bob float or a tri-flat tube with a spherical float is used. For plain tapered tubes, a central guide is often used (rod-guided). If the fluid or the operating conditions are unsuitable for glass, a metal tube is used with a fixed orifice and tapered plug or a tapered tube and disk-head float. Figure 8.2(a) shows the cross-section of a tube with guide beads. In manufacture, the induction-heated glass tube is vacuum drawn onto a metal mandrel so designed that several tubes are made at once in-line.

Floats

There are differing views as to whether centrally guided floats with possible friction effects offer more repeatable results than unguided floats [Figure 8.2(b)]. An

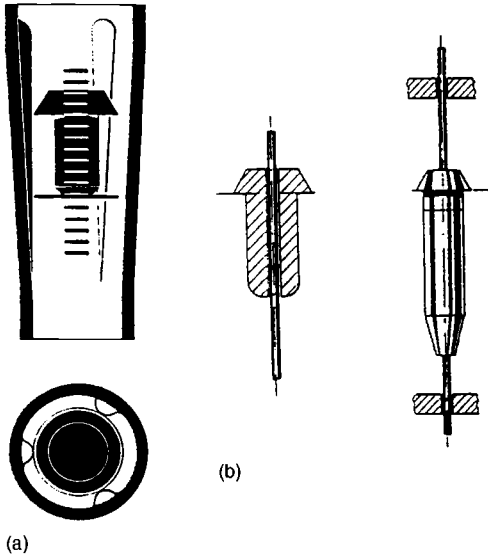
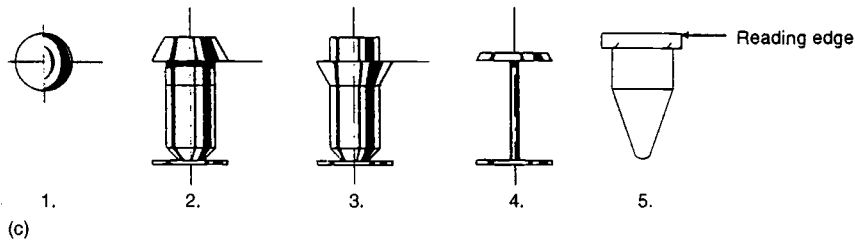


Figure 8.2. Tubes and floats: **(a)** Cross-section of a tube with guide beads; **(b)** Float with guide rod; **(c)** Typical float shapes (1, ball float; 2, viscosity immune float; 3, viscosity nonimmune float; 4, float for low pressure losses; 5, rotating float) (reproduced with permission of Bailey-Fischer and Porter).



alternative is for a tail-guided float, where a ring is held by webs so that the ring moves on the tube beads and keeps the float central in the tube. Figure 8.2(c) shows some float shapes. The development history included attempts to find a viscosity-independent design. This is most nearly achieved with a sharp edge at the maximum diameter. The five floats in the figure are a ball float, sharp edged float to reduce viscosity dependence, a viscosity nonimmune float, a float shaped for low pressure loss, and a rotating unguided float.

8.2.3 REMOTE READOUT METHODS

The most common remote readout method used is magnetic sensing with a metal tube. A diagram of a typical mechanism is shown in Figure 8.3. However, designs using newer field-sensing methods are being introduced.

ISA RP16.4 (1960) deals with extension devices that translate the float motion into a useful secondary function for indicating, alarming, transmitting, etc. An extension tube may be used, and magnetic or electrical sensing coils appear to have been most commonly used.

Liu et al. (1995) described a sensing method for variable area flowmeters that used the capacitance between two conducting strips on the inner surface of the tapered tube and the float of good electrical conducting material. The capacitance

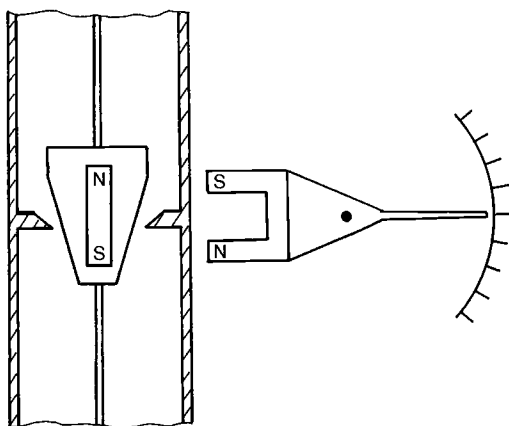


Figure 8.3. Diagram of sensing for metal tubes.

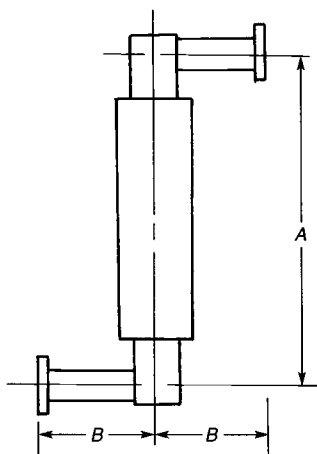


Figure 8.4. Connection dimensions for glass tube type variable area meters (after ISA RP16.1.2.3 1959).

between the strips will vary as the float rises in the tube. They claimed that analysis and experiment have shown that this sensor system can be used for nonconducting fluids. However, tests reported were for a prototype in a laboratory rig and wider testing would appear necessary.

Parker (1990) described an optical transducer. In this, an array of nine infrared LEDs was mounted on one side of the flow tube, and a corresponding array of photo-transistors was mounted on the other but displaced by half the LED pitch. The reading on each detector, or pair of detectors, is taken for each LED, and the responses are normalized. When a float is present, the transmission will vary, and a position-finding algorithm is claimed to give a resolution of 1 part in 2048.

8.2.4 DESIGN FEATURES

ISA RP16.1.2.3 (1959) gives the recommended main dimensions of the connections for the meters, and some of these are shown in Figure 8.4 for the glass tube type, and a sample of the dimensions are given in Table 8.1.

ISA RP16.1.2.3 (1959) gives a uniform terminology, a small part of which has been reproduced in Figure 8.5. It also gives some guidance on precision and on safe working pressures of borosilicate glass tubes.

The manufacturer's instructions on where to read the float level should be obtained (ISA RP16.5 1961a). If no other information is available, the following may be the case:

Table 8.1. Sample of dimensions for glass tube type VA meters (after ISA)

Pipe Size	A	B
$\frac{1}{2}$ " (12.7 mm)	$16\frac{1}{2}$ " (420 mm)	$3\frac{1}{2}$ "–4" (89–102 mm)
1" (25.4 mm)	$17\frac{1}{2}$ " (445 mm)	4"– $4\frac{1}{2}$ " (102–114 mm)
2" (51 mm)	21" (533 mm)	5"– $5\frac{1}{2}$ " (127–140 mm)
4" (102 mm)	28" (711 mm)	6"–7" (152–178 mm)

- ❑ For plumb-bob type floats, their position is usually taken as the largest uppermost diameter.
- ❑ For guided floats, the level may be given by a shoulder on the float.
- ❑ For ball floats, the level is taken as the mid-position (equator) of the ball (but note that others may specify the top of the ball).

These positions may vary among manufacturers.

The meter should, where possible, be installed with a bypass arrangement so that the line can be flushed without affecting the meter and also so that the glass can be replaced without closing down the line. Sudden pressure caused, for instance, by water hammer, liquid close to boiling point, or entrained gas can cause damage and should be prevented.

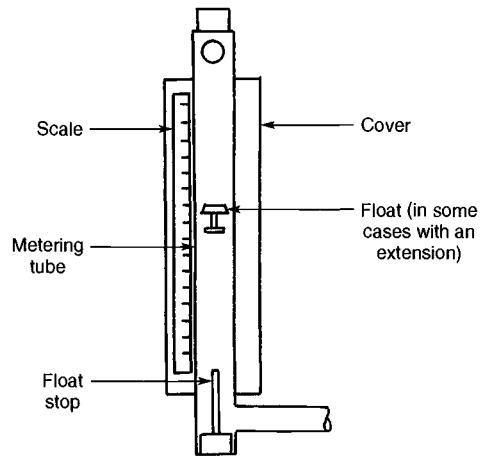


Figure 8.5. Diagram with a selection of the terminology used (after ISA RP16.1.2.3 1959).

8.2.5 CALIBRATION AND SOURCES OF ERROR

If the instrument is to be used as a repeatability measure (ISA RP16.6 1961b), it can usually be calibrated on a liquid such as water, and corrections can be made for density. Corrections for viscosity may also be possible. However, in some cases, the use of a different liquid for calibration may not be acceptable for viscosity reasons. If the absolute accuracy of the flowmeter is required, calibration should be with the actual fluid. If in doubt, refer the question to the manufacturer.

VDI/VDE 3513 (1978) suggests various reasons for error:

- ❑ positioning of the scale, or removing, replacing, or changing of the scale for different ranges;
- ❑ backlash of the float indicator;
- ❑ nonlinearity of the scale, which may accentuate error if incorrectly positioned;
- ❑ friction, which affects the free movement of the float; or
- ❑ unsteadiness of the float.

This does not allow for changes caused by

- ❑ density and viscosity beyond the scale allowance;
- ❑ temperature effects on the conical tube and the float;
- ❑ nonvertical installation, deposits on the equipment, corrosion to the tube or float, vibration, or flow pulsation.

8.2.6 INSTALLATION

The meter must be set vertically, with the inlet at the bottom and the outlet at the top. Head (1946–7) claimed that there was no need for long pipes at inlet or for flow straighteners. Coleman (1956) suggested that, if the tube inlet is less than the annulus area, there was increased sensitivity to inlet flow pattern. He suggested,

therefore, that the maximum tube diameter should not be more than 1.41 times the float diameter ($1.414^2 = 2$ so that this figure ensures that the annulus area is less than the float cross-sectional area). Because of the entry arrangements and the low accuracy at the bottom of the scale, one might expect that installation would have a negligible effect. This appeared to be Head's (1946–7) view. However, some manufacturers recommend at least 5D upstream installation length and may suggest a downstream straight length, which may, in some cases, be greater than the upstream length. A safe rule may be, in the absence of guidance, to take at least 5D of straight pipe both up- and downstream. Installation is more critical with short stroke VA meters where turbulence leads to float instability and reading problems, but modern detector systems can partially overcome this by averaging. Installation into a large diameter gas pipeline can result, at low flows, in vertical oscillation of the float (bounce) often with increasing amplitude and limited only by the float stops.

8.2.7 UNSTEADY AND PULSATING FLOWS

Dijstelbergen (1964) reported extensive mathematics and experimental work on the behavior of the VA meters in a pulsating flow and claimed that “the response of the instrument and the value of the error in the mean float position with pulsating flow can be predicted.” Chatter and bounce can be reduced with guided floats or float rod extensions with a pneumatic damper. Nevertheless, Dijstelbergen's experience of applying these meters in pulsating flow would suggest the need for caution.

8.2.8 INDUSTRIAL TYPES, RANGES, AND PERFORMANCE

Upward flow is, of course, specified, and the meter should be mounted to minimize strain on the glass tube.

Typical uncertainty claimed for these instruments is 1–3% of full-scale deflection as supplied, and probably 0.5–1% repeatability.

Ranges for liquid may be from 0.001 l/min to 100 m³/h, and ranges for gas, from 0.1 l/min to 1,800 m³/h. A meter, typically, has a 10:1 turndown.

Meter tubes are made of glass, acrylic, and special transparent materials where visual reading is used. There is a move to use metal to avoid breakage, etc., for safety reasons.

Transparent tubes are now used mainly for inert liquid and gas applications. Where a metal tube is used, sensing is usually through a magnetic link. For magnetic sensing, the float may be in a tapered tube, or the float may be tapered and, with a fixed orifice, form a variable annulus. The float will probably contain a magnet, and the sensing head will form as closely coupled a magnetic circuit as possible (Figure 8.3). These designs are giving way to newer field-sensing methods. The remainder of the meter may be of aluminum, nylon, stainless steel, PVC, etc.

Temperature can range from –180 to 400°C for metal tubes and rather less for others.

The manufacturer may give advice on flushing the line before the meter is installed and may recommend filters (magnetic if particles are ferromagnetic) for use with devices using metal tubes and magnetic position sensing.

8.2.9 COMPUTATIONAL ANALYSIS OF THE VARIABLE AREA FLOWMETER

Using a commercial meter, Buckle et al. (1992, cf. Leder 1996) have studied the flow around the flowmeter by LDA methods and CFD calculations (Figure 8.6). The work gave quite good agreement between the computation and the experimental results apart from the region of the recirculation zone above the float, which, nevertheless, was shown well by the experimental results. One reason given for the lack of agreement was the small rotation of the float. However, in their subsequent paper (1995), while noting the need to allow for rotation, Buckle et al. concluded that the discrepancies were probably due to asymmetry in the experimental flow. Unfortunately, this work was done at a maximum Reynolds number of 400 so that all the flows were in the laminar regime.

8.2.10 APPLICATIONS

These meters have been used on a wide variety of fluids. Tubes have been made with heating elements, insulating tube arrangements, and armored inspection glass with a pressure-balanced inner tube. Where metal tubes are used, the sensing is via a magnetic coupling, and this can link to pneumatic or electrical signal transmission. The meter has also been used as a bypass for an orifice plate.

Applications include chemicals, pharmaceuticals, refining, food and beverages, power plants, water and wastewater, pulp and paper, mechanical engineering, and plant construction.

Fluids include air, acetylene, ammonia, argon, butane, chlorine, natural gas, helium, steam, oxygen, and various liquids.

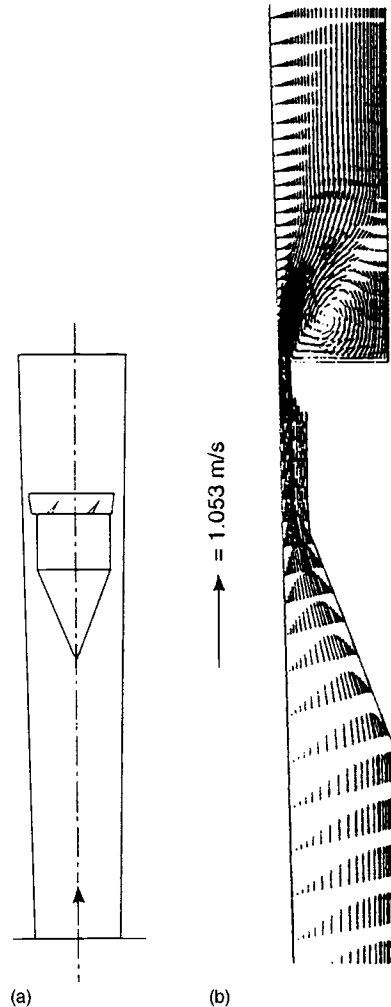


Figure 8.6. Computational results for the flow in a variable area meter: **(a)** Flowmeter geometry; **(b)** Computed velocity vectors (from Buckle et al. 1992; reproduced with permission from Elsevier Science).

8.3 SPRING-LOADED DIAPHRAGM (VARIABLE AREA) METERS

One design is shown in Figure 8.7. Wide turndown and linear response are often claimed as the particular strengths of this device.

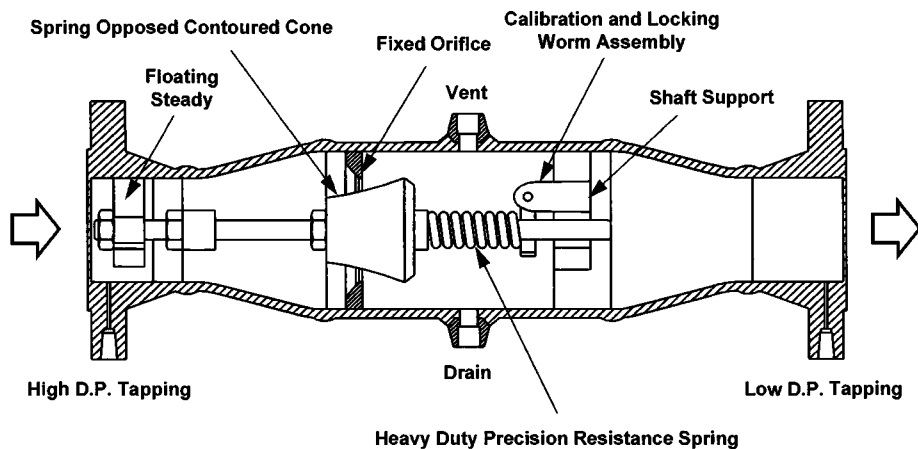


Figure 8.7. Longitudinal section of Gilflo spring-loaded diaphragm meters (courtesy of Spirax-Sarco Ltd.).

The Gilflo-B has a spring-loaded profiled plug in an orifice, which is forced back by the flow, causing a variable area orifice. The differential pressure is sensed between inlet and outlet of the device, and this turns out to be close to linear in relation to flow, resulting in a wide operating range (20:1 up to 100:1 or more) for gas and steam, liquid natural gas, cryogenic, and other liquids. Because the spring-loaded orifice was claimed to give an approximately linear response against pressure difference, a 10:1 flow turndown with a good differential pressure cell was more realistic, and with smart technology 100:1 might have been realizable. However, with linearizing electronics, the user should ascertain the likely accuracy over the range. This design has been succeeded by a wafer design with linearizing electronics.

Other variable area spring-loaded meters have been used for measuring the flow rate of water, paraffin, gasoline, oil, tar, distillates, etc. They can handle both high and low flows. The manufacturers' claims for factory calibration may be of the order of 1–2% and may be based on either rate or maximum flow, although occasionally claims are made for very large turndown such as 100:1 with ranges up to 30,000 l/m, uncertainty of order $\pm 1.0\%$ of full-scale deflection, and repeatability of order $\pm 0.3\%$, and with versions for liquid, steam, and gas. Such claims should be treated with caution. A meter capable of performing to this specification based on rate would be outstanding. On the other hand, if not based on rate, the envelope of measurement uncertainty should be provided for the whole range (cf. Turner 1971).

Whitaker and Owen (1990) also obtained data for a spring-loaded variable area orifice meter in a horizontal water-air flow with void fraction up to 40%. For tests in the range 2–20 l/s, the meter retained its linearity, but the meter factor changed with increasing void fraction by about 25%. Suggestions appear to imply that the meter plug/orifice homogenizes the flow, removing the need for upstream conditioning.

An alternative approach is for the diaphragm or flap to rotate on a hinge and to close due to a spring. Increasing flow forces the flap open, and its angle gives a register of flow rate. Figure 8.8 shows such a design, which has been particularly

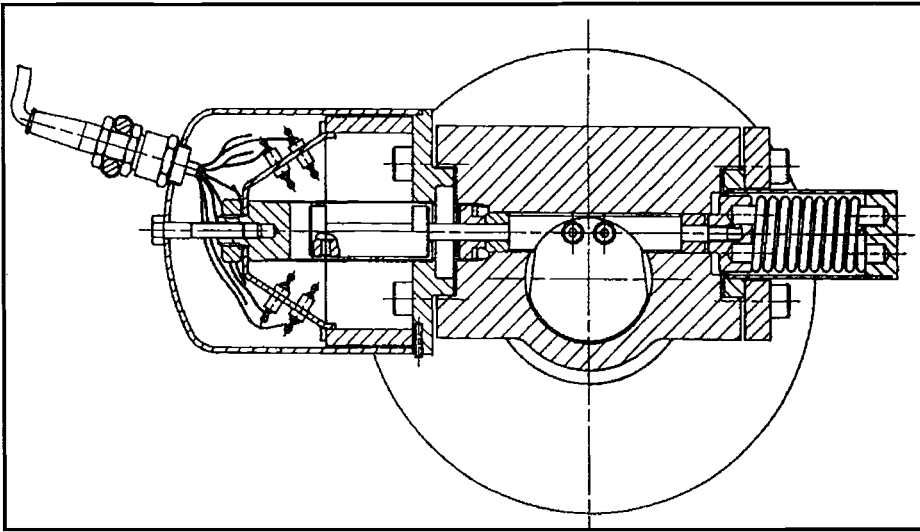


Figure 8.8. Cross-section of spring-loaded rotating diaphragm or flap-type meter (courtesy of Spirax-Sarco Ltd.).

applied to steam flow. The manufacturers recommend that the meter is sized to operate at line velocities up to 35 m/s, which, with meter sizes of 40–100 mm, gives flow rates ranging from 15 kg/h at 1 bar gauge to 9,000 kg/h at 17 bar gauge. The average continuous turndown of the meter is about 25:1.

The measurement uncertainty claims for the meter are $\pm 2\%$ rate. The pressure drop is about 0.5 bar, and its material of construction is iron and cast steel. The outlet signal is electrical from a rotary transformer, which converts the rotation of the flap to an electrical signal.

In Figure 8.9, a typical line arrangement of fittings is shown for steam service. Note that

- ❑ the separator with trap is set to allow condensate through but not steam;
- ❑ the strainer is set horizontally to avoid condensate collecting or particulate dropping into the line;
- ❑ the upstream and downstream spacings, in a reduced line size, are 6D and 3D, respectively; and
- ❑ a nonreturn valve at the end of the line may be necessary.

Hussein et al. (1992) used temperature and pressure measurements, a condensate separator with a condensate flowmeter, and finally a steam flowmeter to obtain the mass flow rate and steam quality. The flow rate was obtained to about 1%, and the dryness fraction to better than 0.3%. The flowmeter was a spring-loaded variable area Spirax-Sarco meter. The separator efficiency was about 92%.

Hussein and Owen's (1991) work was mentioned in Chapter 5. They found that the *Shell Flowmeter Engineering Handbook's* correlation is the best compromise for $x > 95\%$ and for the variable area meter. They wisely suggested that an upstream separator was advisable.

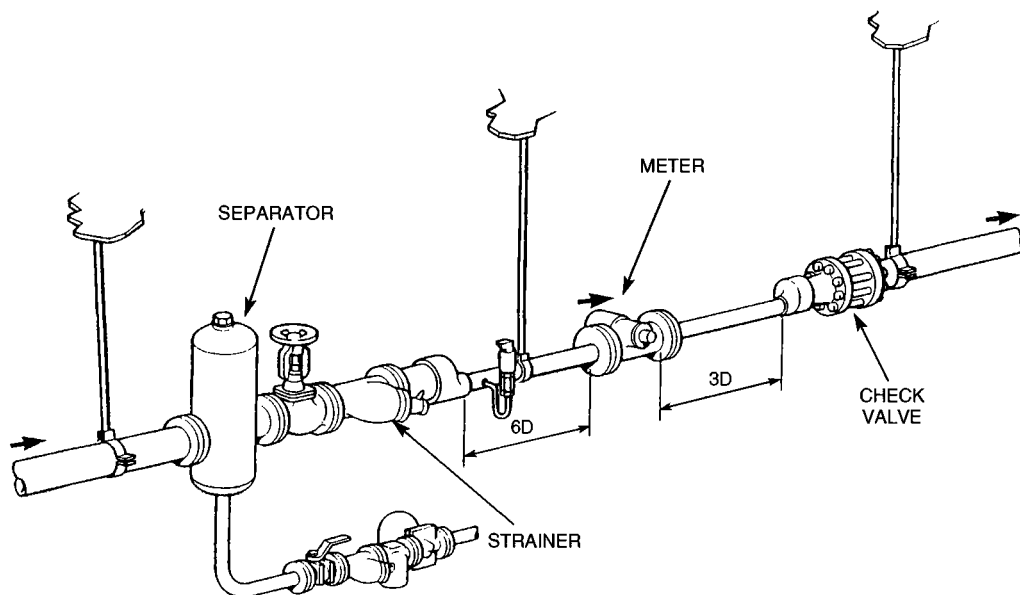


Figure 8.9. Diagram of the line arrangements for measurement of steam flow (courtesy of Spirax-Sarco Ltd.).

8.4 TARGET (DRAG PLATE) METER

The target meter is essentially an inside-out orifice plate meter (cf. Scott 1982, who describes an annular orifice meter) and has similarities to the variable area meter. A drag plate is held in the center of the pipe, and the flow past it exerts a force on the plate which is usually measured pneumatically or with electrical strain gauges (Figure 8.10).

Hunter and Green (1975) reported early work on such a flowmeter in which they demonstrated that with a drag coefficient C_d based on the mean velocity in the gap around the target the variation in C_d over a Reynolds number range of 2,000–250,000 is from about 0.97 to 1.83. They proposed a curve fit of the form

$$C_d = C_0 + C_1 \left(\frac{a}{A} \right) + C_2 \left(\frac{a}{A} \right)^2 + C_3 \left(\frac{a}{A} \right)^3 + C_4 \left(\frac{a}{A} \right)^4 \quad (8.1)$$

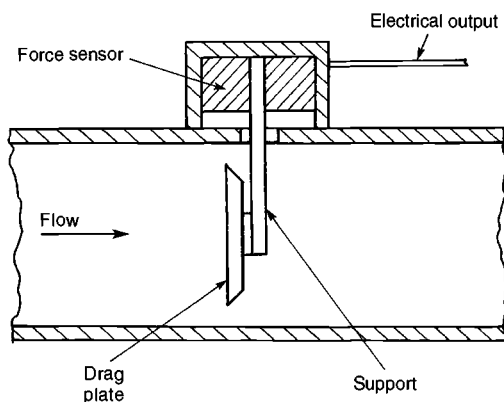


Figure 8.10. Target (drag plate) meter.

where a is the area of the target, and A is the cross-section of the pipe, but they did not appear to propose a term to allow for change in Reynolds number.

The target meter (Ginesi 1991) for $Re > 4,000$ has a drag force proportional to the square of the velocity. The force on the plate

may be obtained from Equation (8.A.3)

$$F = A(p_1 - p_2) = K \frac{1}{2} \rho q_v^2 A / A_2^2 \quad (8.2)$$

where A_2 is, now, the annular area around the target. We may assume that the value of the coefficient will vary with Reynolds number.

In the laminar regime (Ginesi 1991) the device is usable, but results are not so predictable.

Because it allows gas or solids, entrained in the fluid, to pass, the meter has been used for two-phase flows. It needs to be used with care and understanding in this application.

Uncertainty is between $\frac{1}{2}$ and 2% of full scale (Ginesi 1991). The target can be sized for the flow so that for a 2-in. (50-mm) meter full span could be 20 or 200 gal/min (5.5 or 55 m³/h), whereas turndown is limited to 4:1 or 5:1. The expansion of the target due to temperature change will change the response [e.g., with stainless steel 100°F (56°C) can cause about 0.1% error]. The target can also be subject to coating and buildup and will be affected by edge sharpness. Installation requirements may generally be the same as for an orifice plate. Although the accuracy is not as high as other meters, reliability, repeatability, turndown, and speed of response are important characteristics, which may be achieved in some designs.

Figures for a commercial device using force balance arm with either electrical or pneumatic transmission claimed that flows from 0.4 to 1350 m³/h for water and from 12 to 40,500 m³/h for air could be measured by meters of this type with diameter range 25–300 mm and uncertainty of $\pm 3\%$ FSD.

Yokota et al. (1996) appeared to suggest that in unsteady flow measurement up to 10 Hz the uncertainty was within 2%.

Peters and Kuralt (1995) described a flowmeter that was essentially a target flowmeter with such a small gap that the flow was laminar in the gap and the force was proportional to flow. It is likely to be of limited use (cf. Wojtkowiak et al. 1997).

8.5 INTEGRAL ORIFICE METERS

For small diameter pipes and small flows of very clean liquids and gases, it may be possible to use an integral orifice. These devices, essentially, in some arrangement form a part of, or are closely coupled to, the differential pressure cell and can achieve repeatability (Miller 1996) of $\pm 1\frac{1}{2}$ –6% (cf. Cousins 1971).

8.6 DALL TUBES AND DEVICES THAT APPROXIMATE TO VENTURIS AND NOZZLES

The devices in this section all depend on Equation (5.1), with appropriate coefficients defined by the manufacturers.

The Dall tube looks somewhat like a venturi in its largest version [Figure 8.11(a)]. An early description of this device is given by Dall (1962) who also refers to a segmental hump venturi. However, it has certain design features that result in a higher differential pressure and better diffusion than a venturi of equivalent size. These features follow.

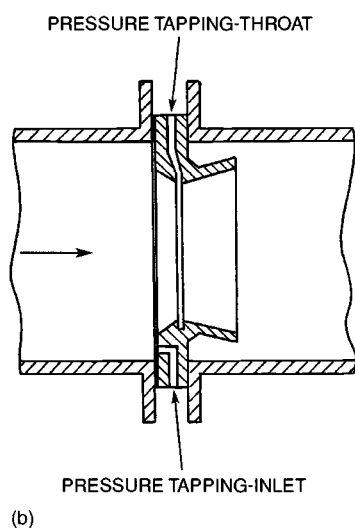
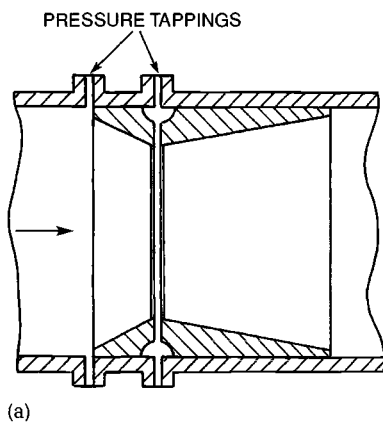


Figure 8.11. Dall tubes (approximate diagrams): **(a)** Dall venturi; **(b)** Dall orifice.

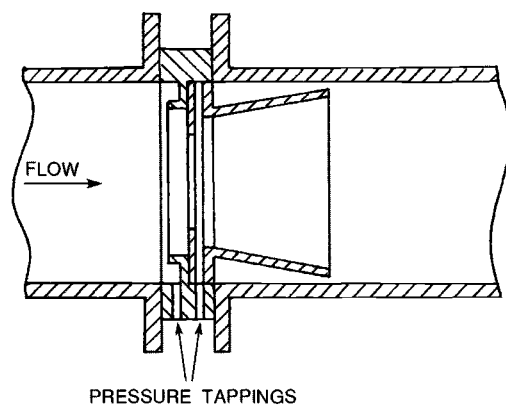


Figure 8.12. Epiflo tube (approximate diagram).

- i. The inlet pressure is raised as the flow is almost brought to rest on the inlet shoulder.
- ii. The throat groove probably results in a toroidal vortex, which sits in the groove, causes the flow to contract and curve to pass around it, and so creates a slightly higher velocity and lower pressure than would otherwise occur.
- iii. The vortex will help the flow to negotiate the changing area at the throat and to reattach in the diffuser.

However, one drawback of this enhanced performance is that any gas in liquid flowing through the Dall tube may tend to get caught in the vortex and to cause substantial performance changes. Cousins (1975) discusses these and other effects caused by slot width, surface roughness, cone angle, and some geometrical details. In a subsequent discussion of this paper, it was suggested that the coefficient could shift by as much as 8% with changing Re from 0.4×10^6 to 1.0×10^6 . The reason for this may have been cavitation or some other mechanism that prevented reattachment of the flow in the diffuser.

A Dall nozzle and orifice have also been made, and Figure 8.11(b) is an approximate diagram of the orifice. In each case, their performance data will need to be obtained from the manufacturer.

Lewis (1975) reported on another device (Epiflo Figure 8.12) in which, at inlet, flow is brought to rest in an upstream-facing annulus. Downstream of this is an orifice followed by a slot and then a short diffuser. The downstream tapping is in the slot. Thus it appears to have some of the features of the Dall tube.

Figure 8.13 shows a device of a venturi type but which has a rounded (bell-mouth) inlet and a wider diffuser angle than ISO allows. It is sometimes referred to as a Low-Loss tube. The manufacturer may claim $\pm 1\%$ uncalibrated and $\pm 0.25\%$

calibrated with a repeatability of $\pm 0.1\%$ and a turndown of 10:1. Sizes range from 13 to 1,200 mm. Installation is claimed to require 12D upstream and 2D downstream for all but partially shut valves and elbows in different planes.

Vincent Gentile, Jr., described a tube which, in one form, is illustrated in Figure 8.14 and is known as a Gentile tube. It is claimed that it differs from other differential devices in that the pressure difference is due to dynamic rather than static pressures. It is also reversible (Scott 1982).

8.7 WEDGE AND V-CONE DESIGNS

Figure 8.15 shows a diagram of a wedge meter. They are designed with an asymmetrical constriction, which may be positioned at the top of the pipe, thus allowing any solids to pass along the pipe. The constriction will result in an increased velocity and reduced pressure. It is likely that a vena contracta will form downstream of the constriction so that the downstream tapping will sense the static pressure near the vena contracta (cf. Dall 1962 who refers to a segmental hump venturi).

It is, therefore, apparent that the orifice equation with a suitable coefficient will be appropriate, but the size of the coefficient and its variation with Reynolds number will depend on the manufacturer's calibration data. Uncertainty after calibration of $\pm 0.5\%$ is suggested.

Oguri (1988) suggested that a wedge flowmeter was suited to solid-liquid and gas-liquid flows. This is probably the reason for its preference over a segmental orifice for dirty or solids-bearing fluids – it is less likely to be damaged by them. It is claimed to have a long life without maintenance and without fouling. It may have a smaller

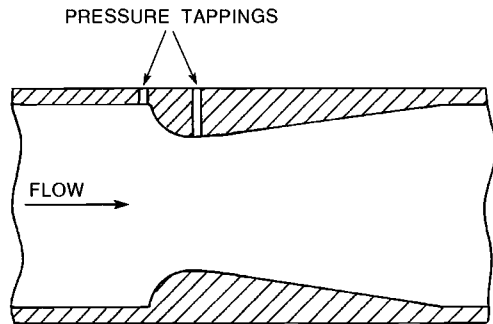


Figure 8.13. Low-Loss flow tube (approximate diagram).

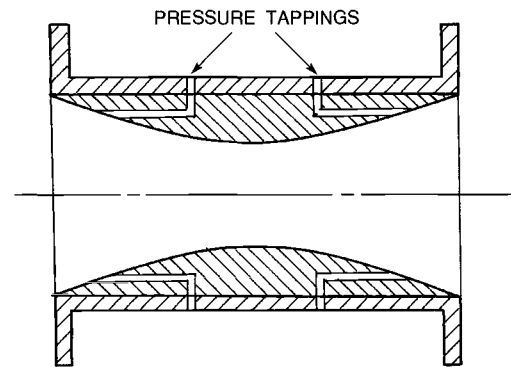


Figure 8.14. Gentile tube (approximate diagram).

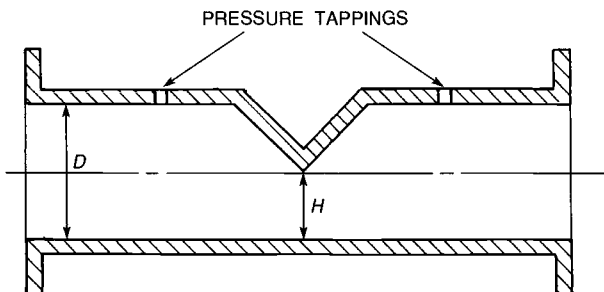


Figure 8.15. Wedge meter (approximate diagram).

pressure loss than the segmental orifice because of the slightly greater fairing of the wedge.

It is available in sizes from about 13 mm to about 600 mm and for a Reynolds number range up from 500.

The V-shaped restriction to contract the flow is characterized by H/D (Ginesi 1991). H is the height of the opening below the restriction (Figure 8.15). There are no critical dimensions or edge sharpness to be retained, and the element can withstand wear and tends to keep clear of buildup due to the slanted upstream face. It is claimed to retain a square law down to about $Re = 500$, regardless of the state of the flow (laminar, transition, or turbulent). With few published recommendations and no standardized geometry, the meter has to be calibrated, and suggestions of 0.5% uncertainty have been made. For $Re < 500$, the calibration should be on a fluid of the correct viscosity. The nonclogging feature with the low-flow capability and ruggedness have made this a realistic choice for fluids such as dewatered sludge, black liquor, coal, flyash, and taconite slurries. Ginesi claims that it is less sensitive to installation than other differential pressure meters. The worst case reported by him, 1.74%, was with three short radius bends in different planes at 5D upstream. To eliminate deviation, 20D is claimed as sufficient.

The commercial device may be offered as a wedge that can be inserted in schedule 40 pipes of small sizes. The manufacturer may supply a mounting tube, constructed to allow direct connection to a pressure transducer, into which upstream and downstream lengths can be screwed. With calibration ± 0.5 to $\pm 0.75\%$ of rate is claimed with repeatability of order $\pm 0.2\%$. Materials are stainless and PVC, the latter as a wafer design. It may also be available as a spool piece. The pressure tappings are presumably symmetrical because the device is claimed as bidirectional. An element is offered for water and gas with similar performance. H/D is in the range 0.2–0.5 (Figure 8.15).

The V-cone meter (venturi cone, which may also be referred to as a McCrometer tube) was introduced in the late 1980s and uses a conical flow restriction centrally mounted in the metering tube (Ginesi 1991, cf. Ginesi 1990; Figure 8.16). Pressure difference is measured across the cone. The beta ratio is equivalent to the open flow area of the orifice, but a wider range is possible for the V-cone. Sizes range from lines of 1–16 in. (25–400 mm), it has good rangeability, and it has resistance to wear, is insensitive to vibration, and is suitable for dirty liquids and gases.

It is claimed to have a turndown of 25:1 or more, presumably by using two pressure transducers, with uncertainty of better than 0.5% of rate and repeatability better than 0.1%. The differential pressure is measured between an upstream tapping and the trailing face of the cone. Ifft (1996) appears to claim that for a $\beta = 0.5$, the performance was $\pm 0.5\%$, and even with a gate valve 0–50% closed and other configurations, shifts were limited to 2%.

It is claimed 3D or less installation length upstream of the spool piece and 3D to 5D downstream results in minimal installation effects. This is explained by the suggestion that the profile entering the meter is flattened by the cone. The effect of a distorted profile on the upstream tapping is not mentioned. Ifft and Mikklesen (1993) suggested that a single elbow and double out-of-plane elbows with $R/D = 1.5$ had little or no effect (cf. Prahu et al. 1996).

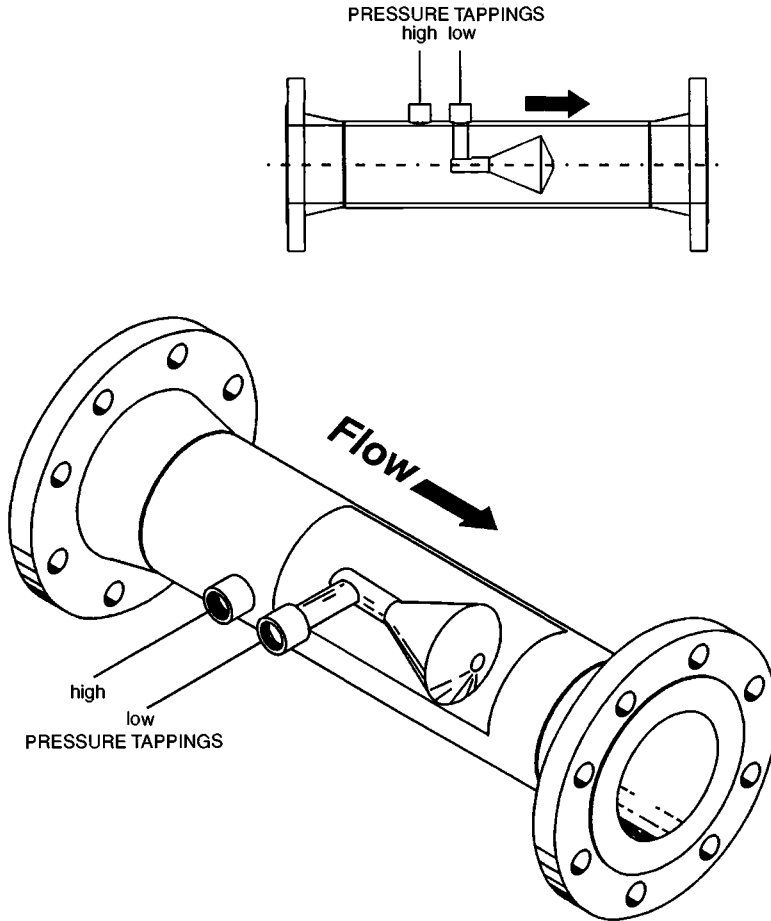


Figure 8.16. V-cone meter (reproduced with permission of McCrometer, Inc., Hemet, CA).

8.8 DIFFERENTIAL DEVICES WITH A FLOW MEASUREMENT MECHANISM IN THE BYPASS

The pressure difference created across an orifice can be used to create a bypass flow, which in turn can be measured by some other flowmeter. One which has been used is a small Pelton wheel. The jet created by the bypass impacts on the wheel, causing it to rotate and allowing the ferrites in the blades to cause pulses in the sensing coil. Claims are for $\pm 1\%$ of full scale on linearity or $\pm 0.5\%$ with 40:1 turndown using linearizing electronics and repeatability $\pm 0.25\%$ for 90% of range.

The oscillating vane meter uses an orifice with the vane in the orifice structure as a bypass (cf. Ginesi 1991). It may be available as a wafer design to go between flanges. The flow through the orifice creates a pressure drop that causes flow through the oscillating vane. The orifice appears to be concentric apart from one segmental region in which the vane is situated. An electric pickup or proximity switch senses the oscillation. Turndown is claimed as 10:1, with an uncertainty of $\pm 0.5\%$ of rate and a repeatability of $\pm 0.2\%$. The meter is for liquid applications. Sizes are

from 25 to 76 mm (1–3 in.). Flow range is from 1.4 to 218 m³/h (5–800 gal/min) with a temperature range of about –25 to 100°C and pressure up to 21 bar. Installation recommendations are compared here with those for an orifice plate with $\beta = 0.3$:

	Oscillating Vane (Upstream/Downstream)	Orifice $\beta = 0.3$
Single bend	10/5	10/5
Two bends in same plane	15/5	16/5
Two bends in perpendicular planes	20/5	34/5
Reducer	10/5	5/5
Control valve	30/5	—
Straightener	5/5	—

Some of the claims give cause for caution when compared with the orifice values.

Other bypass elements used with differential meters are variable area, thermal, and fluidic (Boucher et al. 1991). By including a secondary orifice in series with the bypass, the flow in each should be approximately proportional.

8.9 SLOTTED ORIFICE PLATE

Morrison et al. (1994a, b) suggested using an orifice that had radial slots arranged on two rings instead of a single central hole. The inner ring of slots extended from about 0.1R to about 0.45R, and the outer extended from about 0.55R to about 0.9R. There were 8 slots in the inner ring and 24 in the outer, making an effective $\beta = 0.43$. They concluded from tests in a 50.8-mm line that the slotted orifice is less sensitive to upstream flow conditions than a standard orifice of the same effective beta ratio.

8.10 PIPEWORK FEATURES – INLETS

If a well-formed inlet is provided to a pipe from a stilling tank or large gas vessel, then the pressure drop from the tank or vessel into the pipe can be used to obtain the flow rate. Equation (6.1) is simplified since $\beta = 0$, the flow coming from a large, essentially still, container, and so $E = 1$. The coefficient is also very close to unity.

The *Shell Flowmeter Engineering Handbook* (Danen 1985) gives values for the coefficient. The volumetric flow rate will be given by

$$q_v = C(\pi/4)d^2\sqrt{2\Delta p/\rho} \quad (8.3)$$

where d is the throat diameter (at the downstream tapping) and may equal D , the pipe diameter downstream; Δp is the differential pressure between inlet and throat; and ρ is the density. The coefficients of discharge for various designs may be obtained from the *Shell Flowmeter Engineering Handbook* (Danen 1985). For example,

for a well-formed bell-mouth intake (cf. Figure 8.17), the values of C may be

Re number	5×10^3	10^4	10^5	3×10^5
C	0.914	0.940	0.987	0.991

The flow rate can be obtained from these values and dimensions. The use of such an inlet will clearly depend on the condition of the installation and the upstream flow. The latter should be undisturbed and have had sufficient stilling time. Even so, the method should be used with caution.

Ito et al. (1985) obtained values of about 0.95 at $Re = 20,000$ to nearly 0.99 at $Re = 600,000$, and also gave a useful list of references.

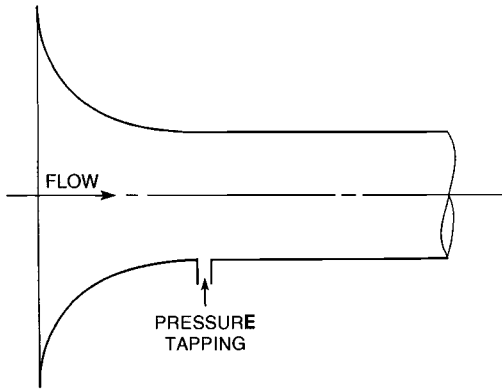


Figure 8.17. Bell-mouth intake.

8.11 PIPEWORK FEATURES – BEND OR ELBOW USED AS A METER

In Chapter 4, the elbow meter is described as an additional means of in situ calibration. However, Xu-bin (1993) reported that the device is widely used in China for the measurement of primary coolant flows in pressurized water reactors, and that it has also been used for measurement of steam, gases, clean liquids, slurries, sludges, and corrosive liquids. Xu-bin also points out that it is easy to install and maintain, can cope with large pipe sizes, and is less prone to damage.

Bean (1971) gives a full derivation of the relationship between pressure and flow. Here we give a simple derivation, which results in the same answer. Figure 8.18 shows the geometry of the bend meter. If we make the grossly simplifying assumptions that

- V is uniform across the bend (clearly this is not the case and there will be flow profile and secondary velocity effects as the flow negotiates the bend) and
- V^2/R is the centrifugal acceleration for all the fluid in the bend (it won't be since the radius of each fluid element differs, and their velocities differ)

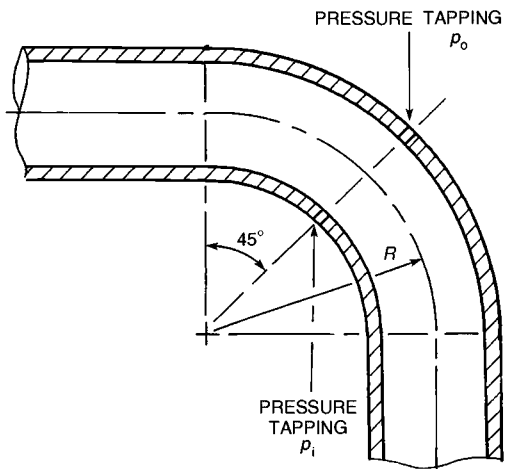
we can write that (Figure 8.18)

$$\begin{aligned} \text{Force across the bend on unit area} \\ &= \text{Mass of fluid of unit cross-section} \\ &\quad \times \text{Acceleration} \end{aligned}$$

or

$$p_o - p_i = D\rho \times V^2/R$$

$$(8.4) \quad \text{Figure 8.18. Bend used as a flowmeter.}$$



Thus

$$V = \sqrt{\frac{R}{D} \frac{\Delta p}{\rho}} \quad (8.5)$$

We are interested in the volumetric flow, and we introduce a flow coefficient to allow for the approximations to real flow. We obtain

$$q_v = KA \sqrt{\frac{R}{D} \frac{\Delta p}{\rho}} \quad (8.6)$$

Bean gave the coefficient as

$$K = 1 - 6.5/\sqrt{\text{Re}} \quad (8.7)$$

for $10^4 \leq \text{Re} \leq 10^6$ and $R/D \geq 1.25$ and suggested that an uncertainty of $\pm 4\%$ should be allowed for. (Note that there appears to be a disagreement with Xu-bin's equations.)

The use of pipe bends is also explained in the *Shell Flowmeter Engineering Handbook* (Danen 1985) where the coefficient includes the $\sqrt{R/D}$. $\sqrt{R/D}$ for a normal 100-mm short radius bend schedule 40 is approximately unity (Danen 1985).

This method should be used with extreme caution because the condition of the bend will be unknown and the uncertainty is unlikely to be as good as 4%.

8.12 AVERAGING PITOT

The Annubar was claimed to be the first averaging pitot (Britton and Mesnard 1982), but many others have followed. The averaging pitot is, strictly, neither averaging nor pitot. In Chapter 18 we shall discuss pitot tubes, which are tubes coaxial with the flow. Although the averaging pitot has a series of upstream-facing holes, they are unlikely to provide a true average of the axial velocities across the pipe. The averaging pitot consists of a bar that spans the pipe and has holes facing upstream and downstream [Figure 8.19(a)]. It is claimed to cope with disturbed profiles due to upstream fittings. The flow will be brought to rest locally on the leading edge of the bar and at the upstream holes unless interconnecting flows occur. The resulting pressures in each of the holes, which will approach the local stagnation pressures, will be linked by the manifold that runs inside the bar. The rear-facing hole or holes will experience a pressure that approximates to the static pressure in the flow. These two pressures, the approximate average stagnation and the approximate static, are then carried by the internal tubes out of the bar and into a suitable pressure transducer where the pressure difference $p_u - p_d$ will be calibrated against the velocity. Essentially, we assume that this pressure difference will approximate to the dynamic pressure that was obtained from Equation (2.10). The calculation of the actual upstream-sensed pressure will depend on internal flows in the connecting manifold from the center holes with higher stagnation pressure to, and possibly out of, the holes nearer the pipe wall where the stagnation pressure will be lower. In addition, the squared relationship

$$\Delta p = \frac{1}{2} \rho V^2 \quad (8.8)$$

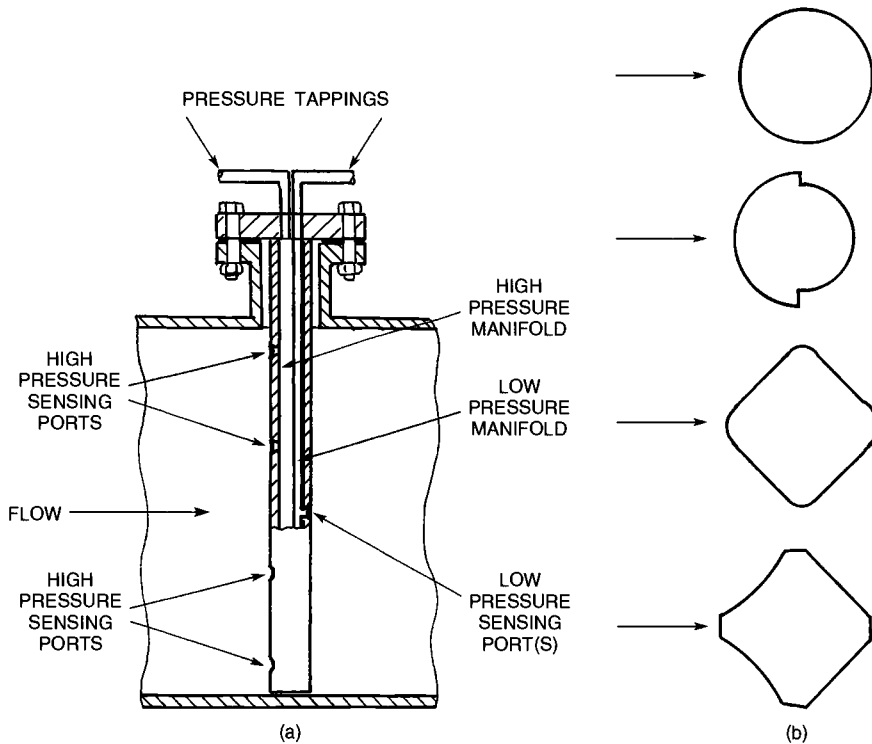


Figure 8.19. Averaging pitot tube/bar: **(a)** Diagram of a typical meter; **(b)** Cross-sections of various designs.

may result in an underweighting of the outer pressures where most of the flow passes. Calculation may suggest that the bar is a near-centerline measurement device with correction for profile shape from the outer tappings.

Figure 8.19(b) shows some of the cross-sections of the commercially available bars. Because flow around a circular cylinder has a changing flow pattern depending on the boundary layer, other shapes have been used to create a less varying pattern. Britton and Mesnard (1982) gave data comparing round- and diamond-shaped sensors, which clearly demonstrated the benefit of the latter over the former. While the coefficient for the round cylinders varied by about 10% for a Reynolds number range of 2×10^5 to 10^6 , the diamond cylinder varied by about 2%. The final version of the cross-section for manufacture was a square section with flattened corners set so that it presented a corner to the flow direction. In addition to the coefficient, a blockage factor that allows for the reduced pipe cross-section at the probe is required. They claimed a measurement uncertainty of $\pm 1\%$ and a repeatability of $\pm 0.1\%$. It has been suggested that a hexagonal bar with faces perpendicular to the flow overcomes the problem of a circular bar, in forcing the flow separation at the edge vertices, as well as the fabrication problem of diamonds and ellipses, which require a weld at the pipe entry point. By using a hexagon, if necessary just at the actual taps, the integrity of a round bar is retained. No doubt different manufacturers will have their own reasons for cross-sectional superiority! The cross-sectional dimension of the bar may be in the range 10–50 mm.

Typical performance claims for these devices are $\pm 1\%$ of reading with $\pm 0.1\%$ repeatability for turndown of 10:1 or more and they are suitable for pipe sizes of 25 mm (1 in.) to 12 m (about 40 ft). Manufacturers should be consulted on flow ranges. Flows of 10–30,000 m³/h for liquids and 200–600,000 m³/h for gases may be possible. Materials are stainless steel, Monel, Hastelloy, titanium, Inconel, PVDF, etc. Temperature range may be from -100 to 450°C , and pressure may be up to 70 bar for some designs. It is claimed by some manufacturers that installation should be with similar constraints to those for the orifice plate, and the connections should also be similar to avoid condensation problems, etc. However, if we compare some claims, we obtain the following:

	Orifice, β			Manufacturers		
	0.2 (Upstream/Downstream)	0.5	0.65	A	B	C (Worst Case)
Single bend	10/4	14/6	22/7	7/3	24/4	10/5
Two bends in same plane	14/4	20/6	32/7	9/3	11/4	15/5
Two bends in perpendicular planes	34/4	40/6	54/7	17/4	—	28/5
Reducer	5/4	6/6	11/7	7/3	9/4	10/5
Expander	16/4	18/6	25/7	7/3	9/4	10/5
Valve*	18/4	22/6	28/7	24/3	27/4	28/5

These results suggest that, in the absence of clear guidance from the manufacturer, taking the spacing for an orifice with $\beta = 0.65$ may be the wisest precaution. It should be remembered that the accuracy of any insertion device is subject to the uncertainty with which the pipe cross-sectional area can be measured. In the case of averaging-pitots, there is a blockage caused by the bar, the effects of which must be obtained from calibration.

It is not clear how manufacturers overcome the uncertainty of the pipe ID and cross-section into which these devices are inserted. If installation of the pipe and bar are at the same time, no doubt this measurement will be possible. The manufacturer will then, only, need to know the allowance for blockage caused by the bar. However, if the pipe is in position, the measurement is considerably more problematic.

In some cases, it is possible that these devices may also create vortex shedding, a phenomenon that we shall encounter and use in the vortex-shedding flowmeter. Unfortunately, such shedding causes lateral forces on the bar and, if these are close to the natural frequency of the bar, may cause vibrations of an unacceptable level. The manufacturer should be asked about this when considering such a device.

We shall discuss pitot tubes in Chapter 18 and shall find a very different shaped device. Cutler (1982), in a letter commenting on averaging-pitots, questions whether multihole sensors do, in fact, generate an average differential pressure. Cutler made the important point that, for fully developed flow profiles, a single-point measurement may suffice. We show in Chapter 2 that one measurement at about

* Orifice distances are based on a fully open globe valve.

three-quarters radius will suffice for turbulent profiles. He gives an example of a device consisting of upstream pointing and downstream pointing tubes.

A meter of this type has been marketed as a flow/no-flow indicator. Impact (upstream-facing) and suction (downstream-facing) ports supply a differential pressure that is displayed on a gauge attached to the pipe or may be used to obtain an indication of flow rate via differential pressure measurement.

8.13 LAMINAR OR VISCOUS FLOWMETERS

The viscous flowmeter for gases uses very small flow passages so that the pressure drop is due to the viscous as opposed to the inertial losses and thus is proportional to the flow rate. The reader is also referred to Chapter 15 where capillary thermal mass flowmeters are described. These are frequently used in conjunction with a laminar flow bypass for higher flow rates. The design with differential pressure measurement appears to have been developed to measure gas flows for research into internal combustion engines where there is a high level of pulsation and unsteadiness. As observed in Chapter 5, pulsation causes an overestimate of the flow if an orifice meter is used. The research engineers, presumably, realized that if the flow behavior could be dominated by viscous effects, the mean differential pressure across such an element would be proportional to the mean flow without errors due to a squared term.

The laminar flow in a circular tube can be shown to obey the equation (Baker 1996)

$$\bar{V} = \frac{1}{8} \frac{\Delta p D^2}{4\mu L} \quad (8.9)$$

where V is the mean velocity in the tube, Δp is the pressure drop along the tube, D is the diameter of the tube, μ is the dynamic viscosity, and L is the length of the tube. From this, we obtain the volumetric flow rate

$$q_v = \frac{\pi D^4}{8 \cdot 16} \frac{1}{\mu L} \Delta p \quad (8.10)$$

In practice, we shall probably need to include, for a particular device, a calibration coefficient C . For very small flow passages viscous effects ensure that, for most practical purposes, the equation will be valid for time-varying flows.

Figure 8.20 is a diagram of such a device with a laminar flow element through which the flow passes, and with pressure measured in the upstream inlet chamber and in the downstream chamber.

Flow ranges for such devices are claimed to be from 0–2 ml/min to 0–2,000 l/min with a pressure difference up to about 10 mm H₂O and a Reynolds number of about 500. The meter is designed to be unaffected by installation and pulsation. Turndown of 20:1 or, with a mass flow computer, of 100:1 are claimed.

It is claimed to be suitable for any dry, clean gas. Calibration is claimed to 2% of reading for 100:1 turndown with 0.2% repeatability and pressure in the range 0–7 bar. Construction is of aluminum, brass, and epoxy. The design allows capacity to be changed by altering the number of parallel passages.

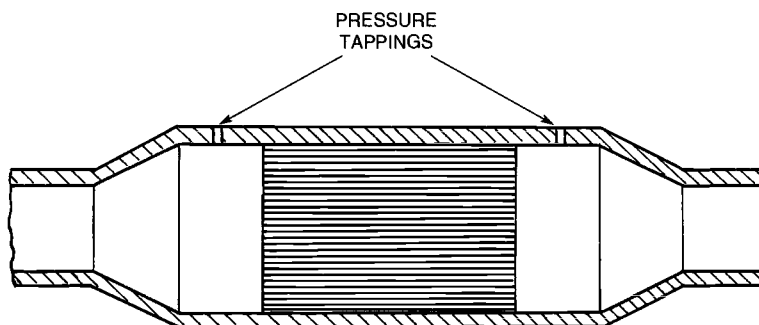


Figure 8.20. Diagrams of a viscous flowmeter.

Weigand (1994) referred to the Hagan-Poiseuille law (Poiseuille 1842) for flow in a capillary

$$\Delta p = Aq_v\mu L/D^4 + B\rho q_v^2 \quad (8.11)$$

where A and B are constants, L is the length of the capillary, and D is its diameter. Other symbols have usual meaning.

Stone and Wright (1994) gave consideration to this meter and obtained the first term in Equation (8.11) and accounted for the second term:

- due to entry length effects, which introduced onto Equation (8.10) a factor of $(1 + C_1\text{Re})$ where $C_1 \approx 0.03$;
- due to entry and exit losses, which they estimated as being of order 0.6% of total pressure drop;
- due to compressibility of the gas (Fanno line cf. Baker 1996) giving an error of about 1%.

If B is small compared with A and L/D^4 is large, Equation (8.11) becomes

$$\Delta p = A'\mu q_v \quad (8.12)$$

where $A' = AL/D^4$ (Bean 1971). This is essentially Equation (8.10) in an alternative form. If we include the factor in (a), we obtain

$$q_v = B(\Delta p/\mu)(1 + C_1\text{Re}) \quad (8.13)$$

which is similar in form to the equation obtained by Jones (1992) except that Jones also had a negative constant term.

Weigand claimed that the laminar flow elements that he describes can operate over turndown ratios of up to 100:1. The main requirement is that viscosity is known. The laminar flow element (LFE) consists of a housing and capillary element. The housing of stainless steel or aluminum provides inlet and outlet connections, structural support, and connections (e.g., to measure differential pressure). The capillary element may consist of one capillary tube with internal diameter as small as 0.228 mm, a bundle of tubes, or a matrix array consisting of a series of triangular passages made from 0.025-mm stainless steel stock and triangles with dimensions of 0.58 mm or less. The length of the capillaries were 76 mm (25 mm for the largest

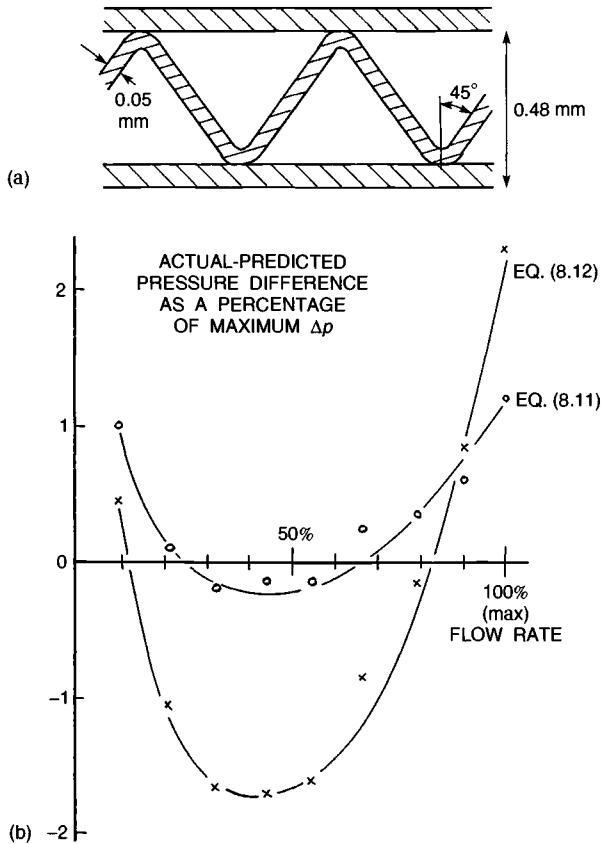


Figure 8.21. Test of viscous flowmeter (after Stone and Wright 1994): **(a)** Section through flow passages of meter; **(b)** Comparison between experimental data for pressure drop and equations of the form of Equation (8.11) (o) and Equation (8.12) (x). Maximum flow rate was about 32 l/s, and maximum Δp was about 250 mm water.

devices and maximum flow rates to reduce pressure loss) and were rigidly held to ensure geometric integrity and consequent repeatability. The 1.5-mm-diameter sensing ports are at the inlet and outlet and within 1.5 mm of the capillary element. The differential pressure is usually less than 254 mm water. Single tubes are used for 0–5 cm³/min up to 0–230 cm³/min. Bundles are used up to 1,300 cm³/min, and arrays, up to 425 m³/min. The normalized flow equation is given as

$$\text{SCFM} = \text{ACFM} \mu_g / \mu_{\text{std}} \quad (8.14)$$

where SCFM is the standardized flow rate, ACFM is the actual flow rate from calibration, μ_g is the viscosity of the calibration gas at flowing conditions, and μ_{std} is the viscosity of the reference gas at standard conditions.

The entrance and exit of the capillary cause a slight nonlinearity. Repeatability is claimed as of order $\pm 0.1\%$. Mass can be obtained using the density. For moist air, there is a humidity correction. Provided the temperature is less than about 26°F and the relative humidity less than 80%, the correction is less than 0.4%. The meter

should be kept as warm as the flowing air. By combining temperature measurement, pressure measurement, and differential pressure measurement and using a microprocessor, gas mass flow rate is obtained. An uncertainty of $\pm 1\%$ of reading for 10:1 turndown is claimed to be achievable.

Abe and Yoshinaga (1991) also described what appears to be essentially a laminar flowmeter with etoile-like resistors but with the center blocked. Their slits were recommended to be 0.2 mm wide and 40–60 mm long. With two in parallel, they claimed flow rates up to about $0.5 \times 10^{-3} \text{ m}^3/\text{s}$ were possible, but nonlinearity appeared to set in below this figure. The device may be used as a flowmeter or a means to measure viscosity.

Stone and Wright (1994) tested a meter with passages as shown in Figure 8.21(a) with length 76.2 mm, about 18,000 passages and a flow area of 3507 mm². The hydraulic mean diameter was 0.347 mm, and the mean velocity at rated flow was 1.73 m/s. Figure 8.21(b) shows the flowmeter characteristic compared with a quadratic assumption as in Equation (8.11) and a linear assumption as in Equation (8.12). Both curves were obtained using regression analysis and not through evaluation of the constants.

8.14 CHAPTER CONCLUSIONS

This chapter has been used to “sweep up” a number of types of meter, among which the variable area meter is the most common. The great benefit of the transparent-tube designs of the variable area meter is the immediate visual reassurance that flow of a certain level is taking place. However, in many applications, a metal tube may be considered necessary for safety reasons. Recent work has started to analyze the flow and to identify reasons for the instability that has been observed since the first use of the meter. There may be value in pursuing these studies further to provide an answer to the reasons for limits of accuracy for the meter:

- ☐ Does upstream disturbance including swirl actually affect performance?
- ☐ Why is it necessary to quote an uncertainty as a percentage of full flow?
- ☐ Does turbulence level affect the meter reading?

Many of the devices depend on proprietary information on performance. My laboratory did tests on one of them under certain flow conditions, but even so I am not able to provide the data here because it is commercially confidential. The user is, therefore, heavily dependent on the information provided by the manufacturer.

The averaging-pitots clearly have a useful role where a full-bore meter is not possible, but whether the claims for installed accuracy are reproduced in practice is dependent mainly on manufacturers' data.

It seems likely that devices such as those in this chapter that use one or other of

- ☐ contoured tube walls,
- ☐ various shapes of insert, and
- ☐ existing pipe features

will continue to have a place. Several of the designs described have been introduced relatively recently.

If a manufacturer is considering the introduction of new designs, the user will be dependent on manufacturer's data for most of the designs that fall outside a standard's specification. The quality of the manufacturer's total operation will, therefore, be a key consideration. The effect of installation should be plausible compared with similar devices from elsewhere. The biggest uncertainty will be in the special conditions that are unforeseen: small traces of gas in a liquid becoming trapped in flow vortices, deposition of small quantities of a solid or liquid film, and materials problems. Increasingly, however, the question will need to be asked, "Is there a linear meter that would be more appropriate?"

APPENDIX 8.A

History, Equations, and Accuracy Classes for the VA Meter

8.A.1 SOME HISTORY

In 1873, G. F. Deacon produced a design that resembled a variable area meter. This was very similar to an earlier device patented in 1868 and made by E. A. Chamberoy of Paris (Coleman 1956). Something similar was devised by Sir J. Alfred Ewing (1924–5) in about 1876 and was initially known as the Ewing flowmeter. Further experiments were carried out by Awbery and Griffiths (1926–7). They found that the float, if a ball, started chattering at high flows and caused an overreading. Three solutions were suggested:

- i. Inclining the tube so that the ball rolled on the wall. This was Ewing's preferred solution.
- ii. Using a float of a different shape. Awbery and Griffiths (1926–7) mentioned that the meter used a cylindrical float with helical channels to cause slow rotation.
- iii. Stretching a fine wire along the tube close to the wall, to prevent the chattering motion (Advisory Committee on Aeronautics 1916–17). Awbery and Griffiths verified this in their observations.

One possible reason for this instability (Leder 1996) is a saddle point (free stagnation point) that forms in the wake downstream of the ball and off-axis and that rotates and induces a large scale rotating helical vortex together with small scale turbulence motions.

Ewing suggested that the range of the instrument could be extended by incorporating more than one float. The largest would be on top and would record low flows, and the smaller ones underneath would rise when the flow rate was sufficient. Ewing also used a dye trace to show the nature of the flow. An alternative could be to have two floats of similar size but with the top one of lower density.

Awbery and Griffiths showed that their experimental results collapsed onto a single curve for a given diameter ratio of D/d , where D is the diameter of the tube at a certain height, and d is the diameter of the ball. They plotted $\log[q_v D / (D^2 - d^2)]$ against $\log[d^3(\rho_f - \rho)/\nu^2 \rho]$, where q_v is the volumetric flow rate, ρ is the density of the fluid, ρ_f is the density of the float, and ν is the kinematic viscosity of the fluid.

British Patent 2428 granted to G. Joslin of Colchester (1879) covered a device for ascertaining rate of consumption of gas, which resembles a VA meter, and this was followed in 1908 by a patent granted to K. Kuppers of Aachen for a tube and float very similar to Joslin's. Felix Meyer of Aachen obtained the patent and started manufacturing the tubes. In 1921 Trost Brothers Ltd. of England took up Meyer's agency for the United Kingdom, and in 1931 Schutte and Koerting secured the manufacturing rights for the United States. Coleman (1956) gave a very useful historical review from which some of these notes are taken.

Head (1946) refers to a downdraught instrument in which the float is lighter than the fluid.

8.A.2 EQUATIONS

The basic theory of the flowmeter depends on the upthrust on the float resulting from the pressure loss across it

$$p_1 - p_2 = K \frac{1}{2} \rho V^2 \quad (8.A.1)$$

where p_1 and p_2 are the upstream and downstream pressures, K is a loss coefficient, ρ is the density and V is the velocity in the annular passage past the float. Replacing velocity by volumetric flow rate q_v such that

$$V = q_v / A_2 \quad (8.A.2)$$

where A_2 is the annular area around the float, we obtain

$$p_1 - p_2 = K \frac{1}{2} \rho q_v^2 / A_2^2 \quad (8.A.3)$$

This equation assumes that the dynamic head at inlet is negligible, and that the pressure difference across the float is caused by the loss of the dynamic head downstream of the float. Dijstelbergen (1964) uses a contraction coefficient C_c for the ratio A_2/A_x , where A_x is the area of the tapering tube at height x .

The immersed weight of the float is given by

$$W = V_f(\rho_f - \rho)g \quad (8.A.4)$$

where V_f is the volume of the float, ρ_f is the density of the material of the float, and g is the acceleration due to gravity. Thus, the balance is given by

$$K \frac{1}{2} \rho q_v^2 A_f / A_2^2 = V_f(\rho_f - \rho)g \quad (8.A.5)$$

where A_f is the maximum cross-sectional area of the float.

This is usually rewritten as

$$q_v = \frac{C_c}{\sqrt{K}} A_x \sqrt{\frac{2V_f(\rho_f - \rho)g}{A_f \rho}} \quad (8.A.6)$$

with $1/\sqrt{K}$ replaced by C (e.g., Martin 1949, Coleman 1956; cf. Head 1946–7).

Coleman (1956) gave values of C for three float shapes (Figure 8.A.1), which suggested that at full flow the value of C ranged from 0.6 for a float of low Re sensitivity to about unity. The plots also showed a decrease in C at low flow rates, and this would broadly reflect the fact that the loss coefficient for an orifice increases with reduced Re (Miller 1990). Coleman also plotted the effect of viscosity on the various float shapes and confirmed the implications of Figure 8.A.1, that the plumb-bob is the most viscosity dependent.

Schoenborn and Colburn (1939) gave a curve of C against Re/C , where Re is the Reynolds number based on D_{eq} . They called it the equivalent diameter but defined it as the difference between D and d . This curve is reproduced in Figure 8.A.2. It was obtained for a large high pressure design using a steel tube with extension rod observable through a graduated glass window. This curve could be used as a first approximation if no other is available. They used a balance of forces of the weight of the float in liquid to the upward force due to the pressure reduction across the float to provide the basis for their experimental correlation. They also noted that the curve for large meters appeared to be less fluid-dependent than for smaller instruments.

Although it may be difficult to predict flow rates, it is possible to convert from one fluid to another at the same reading. Schoenborn and Colburn (1939) gave an equation similar to that given by ISA (1961b):

$$\frac{q_{v2}}{q_{v1}} = \frac{C_2[(\rho_f - \rho_2)/\rho_2]^{1/2}}{C_1[(\rho_f - \rho_1)/\rho_1]^{1/2}} \quad (8.A.7)$$

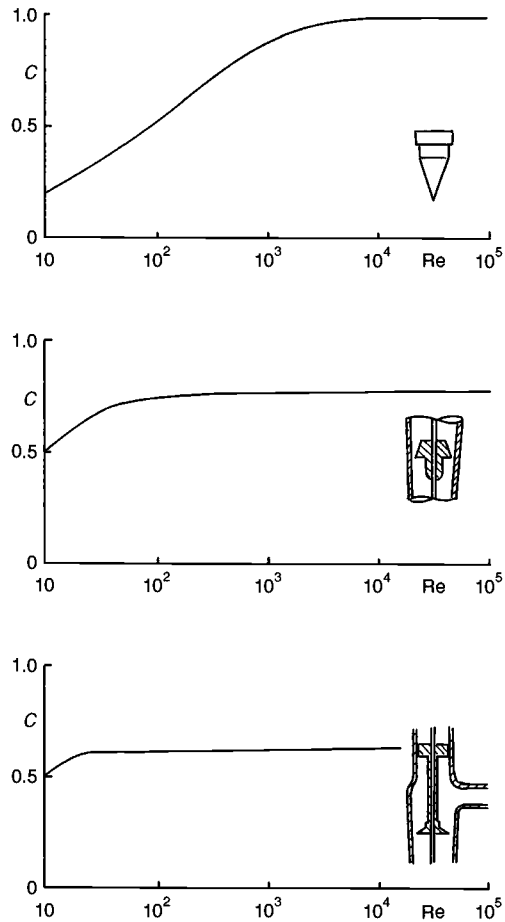


Figure 8.A.1. Effect of float shape on flow coefficient (after Coleman 1956).

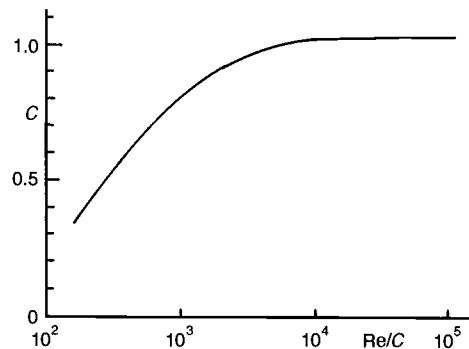


Figure 8.A.2. Plot of C against Re/C for a typical variable area meter (after Schoenborn and Colburn 1939).

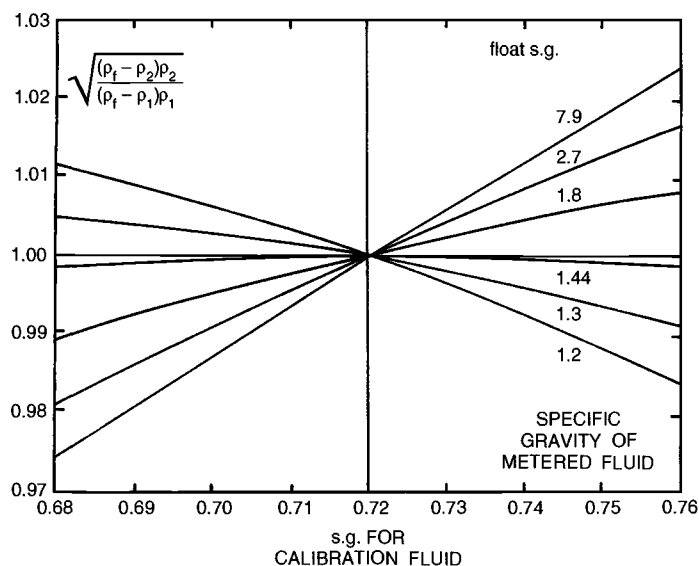


Figure 8.A.3. Density compensation float comparison curves (after Coleman 1956).

The difference is that ISA (1961b) omits C_2/C_1 , thereby making the assumption that the ratio is unity. Coleman (1956) also used this equation with C_2/C_1 equal to unity, noting that variation of viscosity is ignored. He referred to the fact that if Equation (8.A.6) is rewritten for mass flow rate and differentiated for variation in fluid density, it can be shown that a float density, of twice the fluid density gives a zero change and, therefore, is least sensitive to variation in density (cf. Head 1946). Figure 8.A.3 shows values of the density correction factor with change of fluid density and for various float densities, based on calibration for aviation gasoline of specific gravity 0.72. Head (1946) also raised the possibility that, with a downdraught design, the requirement from differentiating the volume flow equation of a float of zero density could be achieved approximately. Head also suggested that float automatic compensation might be used to allow for temperature variation and other variables. Several authors have pointed out that the value of C is constant if $\mu/\sqrt{\rho(\rho_f - \rho)}$ is constant.

8.A.3 ACCURACY CLASSES

To define the accuracy of the variable area flowmeter, VDI/VDE 3513 (1978) recommended an accuracy class and divided the reading into two parts:

- First part is three-quarters of the actual flow rate M ;
- Second part is one-quarter of full-scale or upper range value E .

The summation error F in flow rate, in actual flow rate units, is then given by

$$F = \left(\frac{3}{4}M + \frac{1}{4}E \right) \frac{K}{100}$$

where K is the accuracy class number selected from a range from 0.4 to 10. For example, taking a full-range flow of 100 units, the actual error will be

$$\text{at full scale} \quad \left(\frac{3}{4} \times 100 + \frac{1}{4} \times 100 \right) \frac{K}{100} = K \text{ units}$$

$$\text{at } 1/10 \text{ full scale} \quad \left(\frac{3}{4} \times 10 + \frac{1}{4} \times 100 \right) \frac{K}{100} = 0.325 K \text{ units}$$

If we take a value for K of 2.5, then the error at full flow will be 2.5 units in 100 units (2.5%), whereas at 1/10 full flow it will be 0.812 units in 10 units (8%).

Positive Displacement Flowmeters

9.1 INTRODUCTION

The material in this chapter is based on the paper by Baker and Morris (1985) on positive displacement (PD) meters, which in turn has been updated with more recent industrial and published material and extended to gases. P.D. Baker's (1983) paper provided some additional useful information on these meters. The main types of liquid meter were given by Barnes (1982), Hendrix (1982), Henke (1955), and Gerrard (1979) (cf. Mankin 1955). The reader should refer to API (1992) and similar documents.

At least four of the meter designs to be discussed have been around for over 100 years. The nutating disk flowmeter for liquids was developed in 1850. The rotary piston meter appeared in the late nineteenth century (Baker 1998).

The measurement of gas has depended, from an early date, on two types of positive displacement meter: the wet gas meter of high accuracy and credited to Samuel Clegg (1815), and the diaphragm meter of lower performance but greater range for which William Richards (1843) should take the credit.

9.1.1 BACKGROUND

The concept of carrying known volumes of fluid through a flowmeter is a short step from the use of a discrete measure such as a bucket or measuring flask. Thus in each of the designs described later, the flow enters a compartment that is as tightly sealed as is compatible with relative movement of adjacent components. A knowledge of how many of these compartments move through the flowmeter in one rotation of the shaft leads to a knowledge of the flow rate for a certain rotational speed. A simple theory is developed in Appendix 9.A.1. Clearly every leakage path will reduce or increase the amount carried and cause an uncertainty in the measurement. We can make an approximation of the leakage flows in the simple model. In addition, pressure and temperature will distort the volume and may cause small errors, and we may need to develop compensation for these. Compensation for pressure can be provided in the form of a double-walled meter in which the precision-measuring chamber is unpressurized since the pressure is the same inside and out, and the pressure drop to the outside world is taken across the second and outer wall, the shape of which will not affect the accuracy. The temperature compensation, if necessary, can be incorporated into the calibration adjustment by a temperature-sensitive element.

Very often these devices provide a direct sales ticket that is produced by the rotation of the meter and the starting and stopping of the flow. On the other hand,

it is very difficult to make small adjustments as needed in calibration where the signal is from a mechanical rotation. We, therefore, need to introduce complicated mechanical calibrators where a frequency output is not acceptable. The combination of mechanical counter and ticket machine can cause a substantial drag on the meter and may cause slippage errors in certain types of calibrators.

Many of the points made in this chapter relate primarily to liquid measurement, have some relevance to gases, and should be noted by those mainly interested in gas measurement.

9.1.2 QUALITATIVE DESCRIPTION OF OPERATION

The essential of any positive displacement meter, as indicated earlier, is that a discrete and well-defined portion of the fluid is carried from inlet to outlet without loss or mixing with the remainder of the fluid. This applies to all the fluid that passes through the meter. The skill and ingenuity of the designer is shown, therefore, in achieving this as simply and precisely as possible. If, for instance, we know the volume of each compartment of a reciprocating pump, then knowing how many cylinder-full portions are transmitted in a revolution, we can relate the amount passed to the speed or revolutions of the pump. This is essentially the principle of a metering pump. Taking this a step farther, we could reverse the reciprocating pump. Instead of providing power externally to drive the pump, the fluid moves the pistons, and the valves ensure that each portion is transmitted, without loss, to the outlet. Of course there will be some loss due to imperfect sealing between the pistons and the cylinders, but provided this can be kept very small, we have the basis of a very accurate instrument. The reciprocating principle may not be satisfactory for high throughput meters, and so other designs have been devised. It will be easiest to explain the working of each in turn with respect to diagrams.

The theory of the device is worked out in Appendix 9.A.1. Here we need only note that the simplest equation of flow through such a device is

$$q_v = \text{Volume per revolution} \times \text{Rotational speed} - \text{Leakage} \quad (9.1)$$

Leakage is unlikely to be a constant value for one revolution and is more likely to be speed-dependent. The output from all meters is a rotating shaft, and this needs to be related to the flow rate and total passed. If the shaft passes through the pressure containment of the meter, then a rotary-shaft-sealing arrangement will be needed. In one type of meter, an externally lubricated packing gland is used to isolate the dynamic shaft seal from the product (Baker 1983), thereby increasing the life of the packing gland. The external lubricant must, of course, be chemically compatible with the product being metered. In some cases, the problem is avoided by using a magnetic drive coupling.

In the past, calibration and transmission methods have tended to be by mechanical linkage. Such a linkage loads the rotor and may alter the characteristic of the meter. It is also difficult to design a mechanical system to adjust the calibration of the meter. Some ingenious methods have been adopted and will be described later in the chapter, largely for historical interest and for those who have dealings with such devices that remain in service. Today the possibility of using electrical or optical

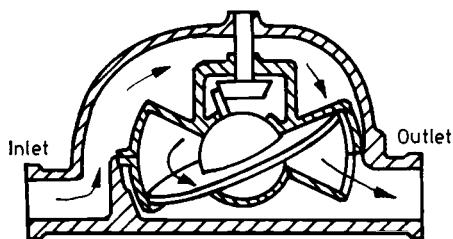


Figure 9.1. Nutating disk meter (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

transmission and still retaining intrinsic safety means that an output pulse train can be used, and signals can be manipulated electronically.

9.2 PRINCIPAL DESIGNS OF LIQUID METERS

This section reviews the main designs and their special operational features and,

where possible, reviews the published information that is available about them.

9.2.1 NUTATING DISK METER

In the nutating disk meter (Figure 9.1), the disk is constrained by the central spherical bearing and by the transmission to nutate. Nutation is a movement rather like a spinning top that is slowing down. The disk is prevented from rotating by a partition separating the inlet and outlet streams. The incoming flow causes the disk to nutate and in doing so the disk forms a closed compartment trapping a fixed volume of liquid. There are effectively two such compartments, one above and one below the disk.

The disk element has the shape of the planet Saturn. The important difference is that the disk has a slit in its "rings," which makes a close fit with a radial partition. Flow enters, say, under the disk on one side of the partition and causes the disk to rise and fill the region under the disk, and so to nutate. When it has nutated half a turn, the entry port becomes blocked by the disk, and the other entry port above the disk has started to allow fluid above the disk. At the same time, the exit port for the liquid below the disk opens to discharge. Thus in terms of Equation (9.1), in one rotation, the cavity on one side of the disk is both filled and discharged. It is unlikely that the leakage can be kept very small, and the overall accuracy is not expected to be very high. Although rather difficult to describe, it is a simple meter of low cost, rugged and capable of uncertainty within $\pm 1.5\%$. (Henke 1955).

Typical industrial designs, some of which may be for water flow measurement, may have a built-in filter. Claims for measurement uncertainty may be of order 1% for flow rates of 10–100 l/min with minimum pressure of 0.1 bar, viscosity up to 1,000 mPa, and temperature up to 80°C. Materials may be stainless steel, polypropylene, or ethylene-tetra-fluor-ethylene. Modern designs may include a microcomputer to convert flow rate into the correct units.

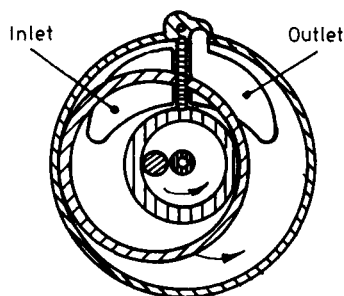


Figure 9.2. Oscillating circular piston meter (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

9.2.2 OSCILLATING CIRCULAR PISTON METER

The oscillating circular piston meter (Figure 9.2) is similar to the nutating disk meter in that the rotation of the piston is constrained by a partition, so that the piston, in fact, rocks. Its center is constrained

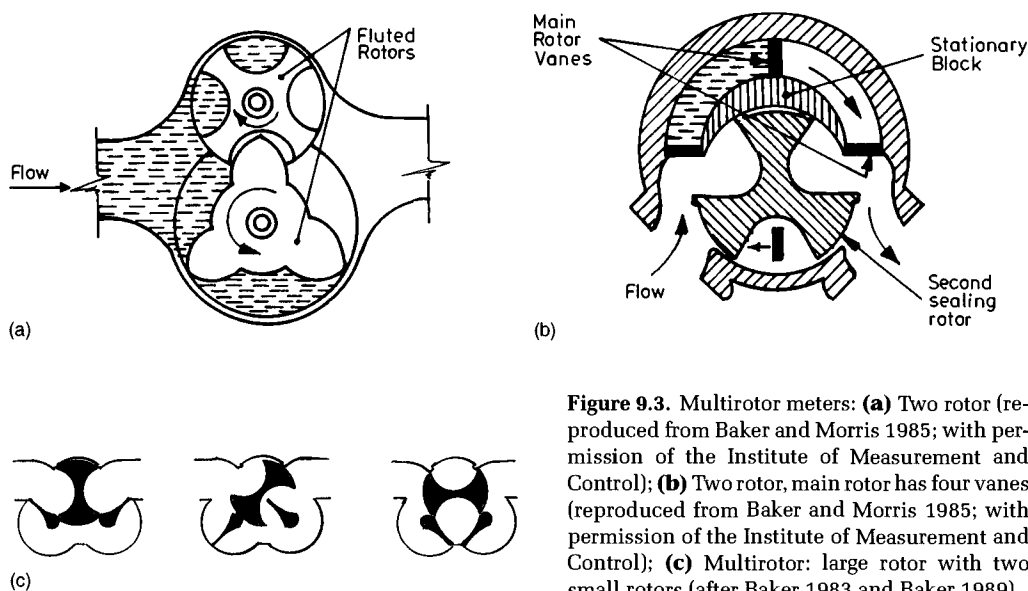


Figure 9.3. Multirotor meters: **(a)** Two rotor (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control); **(b)** Two rotor, main rotor has four vanes (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control); **(c)** Multirotor: large rotor with two small rotors (after Baker 1983 and Baker 1989).

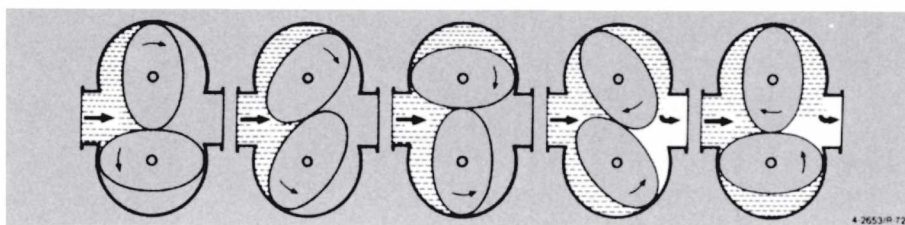
to move in a circle by the transmission so that the radius of the cavity is essentially the sum of the radii of the rotary piston and the circle on which its center moves. When the center of the rotary piston is at the top of its travel, the piston forms a closed compartment with the cavity. One rotation of the shaft will cause the rotary piston to return to its starting place and so to discharge the volume of one compartment. In some designs, the leakage may not be negligible. As well as use with water, these meters may be made in designs suitable for liquid foods and other nutritional fluids (Anon. 1966a). Hamblett (1970) suggests that they can achieve a measurement uncertainty within $\pm 0.25\%$ for flow rates from 0.018 to $110 \text{ m}^3/\text{h}$ (4 – $25,000 \text{ gal/h}$). This accuracy is rather higher than would be expected. He also lists the various types of registers and controls that are suitable for a range of applications.

9.2.3 MULTIROTOR METERS

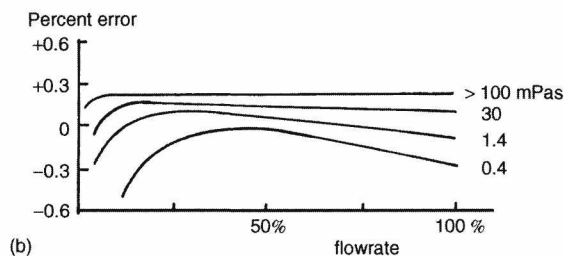
There are several designs of meter with multiple rotors. Figure 9.3(a) has two rotors that seal against each other; each rotor carries liquid through the meter as it rotates. Figure 9.3(b) shows another two-rotor meter, but in this case the main rotor consists of four vanes that form metering compartments. The second rotor is a sealing rotor that returns the vanes to the inlet side of the meter. The second rotor may be two- or three-lobed, and its rotation will be precisely linked to that of the main rotor. It transmits a net fluid flow across the meter equal to the volume of the vane that it returns to the inlet. The mechanism, its design and engineering, is complex and, unless manufactured to a very high standard of precision, may lead to high pressure loss and interference between rotors and housing. Figure 9.3(c) shows a design with a large rotor and two small rotors.

9.2.4 OVAL GEAR METER

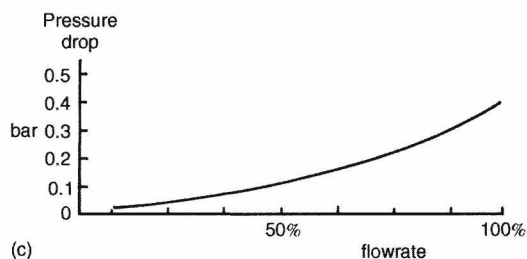
The oval gear meter (Figure 9.4) is a special form of a multiple rotor meter in which each oval rotor is toothed, and sealing between the rotors is enhanced by the



(a)



(b)



(c)

Figure 9.4. Oval gear meter: **(a)** Working principle (reproduced with permission of Bopp and Reuther); **(b)** Error curves for a range of viscosity values (after Haar Messtechnik); **(c)** Typical pressure drop for operating range (after Haar Messtechnik).

resulting labyrinth. Each rotor transmits fluid from inlet to outlet and forms a closed compartment when its major axis is aligned with the flow direction. The volume passed per revolution of each rotor is four times the volume between the rotor and the oval housing when the rotor is confining liquid [Figure 9.4(a)]. In place of the leakage paths between the rotors of multirotor meters, there will be, for this meter, extremely small leakage where the rotors mesh, and the tolerances for the other surfaces are likely to be to a high standard, giving a very small value for the overall leakage.

Small examples of this device are found in oil-flow lines in hydraulic systems, whereas larger high precision stainless steel versions are also available. They are claimed to have a low pressure drop, to have low maintenance, to be of simple construction, and to achieve accuracies within $\pm 0.25\%$ (Henke 1955). Meters are available with a stress-free measuring chamber for high pressure applications.

The meter can be subject to viscosity variation. Figure 9.4(b) shows the error, and Figure 9.4(c) shows the pressure drop for a typical commercial design. Measurement uncertainty is in the range ± 0.1 –2%. Sizes are from 6 to 400 mm, with flow ranges from 0.1 l/m or less up to 1,200 m³/h, but many are for smaller flows. Turndown is of order 10:1 (Endress and Hauser 1989), and operation is possible with, and largely unaffected by, viscosity in the range 0.4–20 cP. However, the possibility of a calibration shift (of order 0.5%) should be checked with the manufacturer. One manufacturer suggests that operation is possible up to very high viscosities. Pressure

loss is 0.5–1 bar, and pressure may be up to 40 bar or higher for certain designs, with temperature up to 290°C.

Transmission may be through a permanent magnet coupling or through magnets embedded in the oval wheels that generate a voltage in an external inductive sensor. Calibration gearing may also pass the output to a totalizing counter. Operation may be bidirectional.

Materials may be, for the body, stainless steel, cast iron, or aluminum and, for the gears, carbon steel, brass, or chemical-resistant plastic.

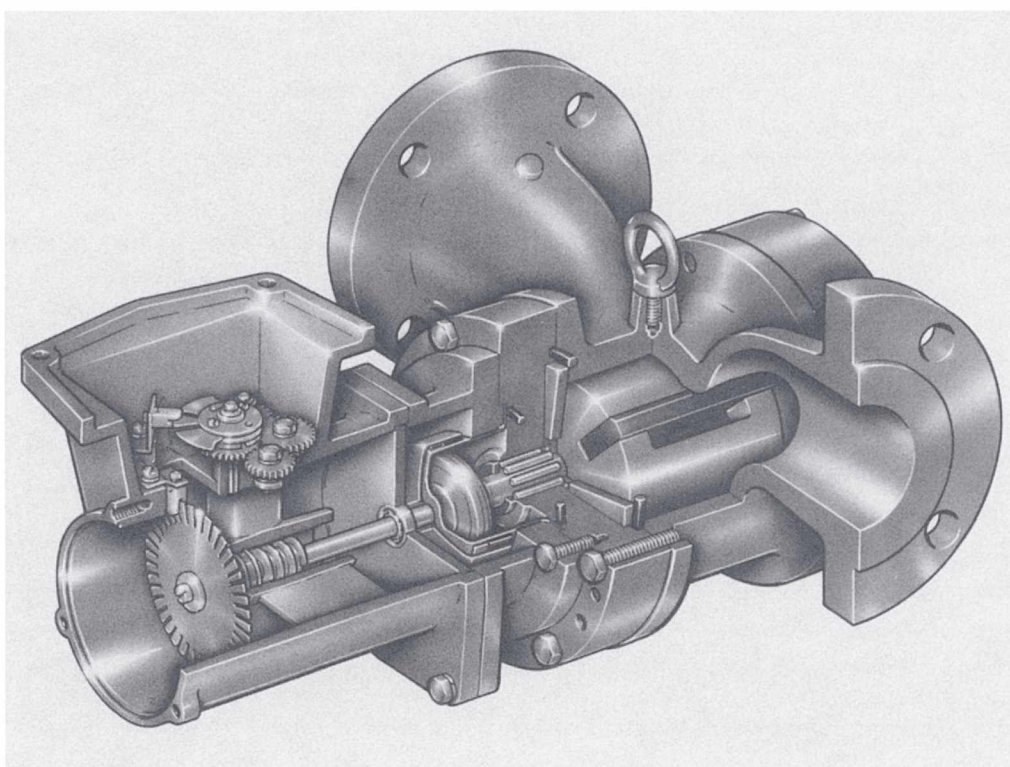
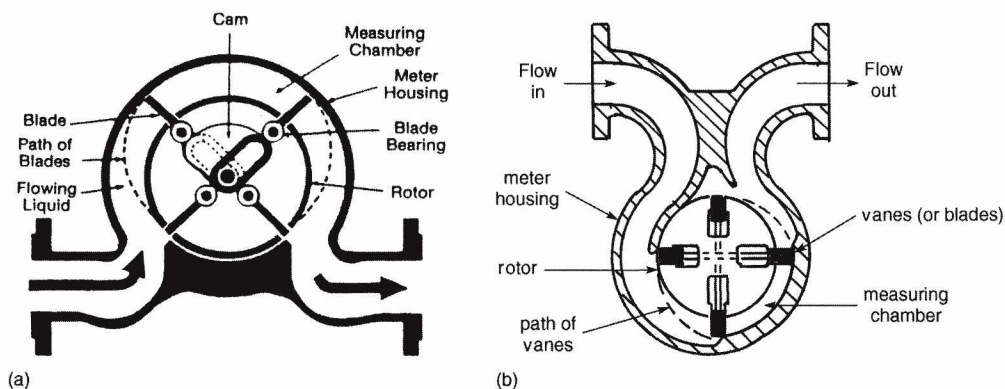
Applications are in adhesives and polymers, but also in many basic utility flows in the water industry and in hydraulic test stands. One manufacturer includes solvents, paints, and adhesives; dispersions, polymerisates, and polycondensates; glucose and alcohols; organic and inorganic liquids; gasoline, fuel oils, lubricants, and raw and intermediate liquid products; and other liquid chemicals.

9.2.5 SLIDING VANE METERS

Sliding vane meters (Figure 9.5) are, probably, the most accurate of this family of meters. Closed metering compartments are created by means of sliding vanes that move out from a slotted rotor to meter the fluid through a passage of constant shape. The movement of the vanes is caused, in the example in Figure 9.5(a), by an internal stationary cam. The inside edges of the vanes have rollers that follow the cam. The vanes are designed to have a small clearance on all fixed surfaces. The movement of the vanes, in the example of Figure 9.5(b), is achieved by an external cam, which the outside edge of the blade, where it meets the end faces, follows. There is therefore some surface contact, which can result in wear and a need for calibration checks.

The elaboration of Equation (9.1) in Appendix 9.A.1 is focused on these devices. The measuring volume per revolution is essentially the volume between the two concentric cylinders, which make up the outer case and the rotating inner wheel with vanes. It is reduced by the volume of the vanes themselves. These designs can be highly accurate, and much of the following discussion is concerned with the limits of accuracy. For instance, they are expected to measure with great precision the transfer of valuable liquids, controlled by agencies such as H.M. Customs and Excise (1995). Such precision is likely to be demonstrated by short-term repeatability, such as three successive measurements of 500 liters within 0.1%, and long-term repeatability over a 6-month period within 0.15%.

Wallis (1975) gave figures from his own tests indicating that these figures are achievable with this type [which he refers to as a rotary piston meter and, as shown by his illustration, is in our category of sliding vane meter, Figure 9.5(b)]. He divided the errors in a meter with a double-walled case into three classes – short, medium, and long term. Experimental tests with kerosene at the U.K. National Gas Turbine Establishment, in which the flow was diverted into calibration vessels, suggested that the following figures were applicable: leakage, 0.005% of rated flow; short-term errors, 0.005–0.05% for different flow rates (affected by environmental and fuel temperatures and hence by weather and conditions); medium-term errors, 0.05%; and long-term errors, 0.2%. This gave a year-to-year repeatability of within 0.15%. Systematic (long-term) errors (e.g., in calibration equipment) were difficult to estimate but might be expected to add an additional 0.2% so that the absolute accuracy was estimated at $\pm 0.35\%$. Another paper (Wallis and James 1985) suggested that the performance of positive displacement flowmeters can be represented by an ideal meter



(c)

Figure 9.5. Sliding vane meter: **(a)** Smith type showing blade (vane) path (reproduced from Baker 1983; with permission of the author); **(b)** Avery Hardoll type (after the manufacturer's literature); **(c)** VAF J5000 (reproduced with permission of VAF Instruments and Industrial Flow Control Ltd.).

of arbitrary but constant volume in parallel with a small leakage that is independent of flow rate.

Manufacturers claim very high performance in line with these published figures of 0.1–0.3%. A repeatability of 0.01–0.05% for flow ranges from about 3 m³/h up to 2,000 m³/h with turndown ratio of 20:1 and size range of 64–152 mm (2.5–6 in.) is typical. One penalty of this type of meter is its bulk and weight (60–136 kg). In

some designs, the highest flow rates are achieved by ganging the meters together in a double or triple capsule. Pressure loss should be kept to within 70–100 kPa (10–15 psi) by suitable choice of meter size.

Materials may be cast iron for the bodies, and the rotors may be made of similar materials or other materials such as aluminum. The same is true for outer covers. Low friction ball bearings may be used. Static seals may be high nitrile, whereas dynamic seals may also be fluorocarbon. Working pressure is about 10 bar, and temperature ranges from about -25 to 100°C . Temperature changes cause less than $0.0015\%/^{\circ}\text{C}$, which is small compared with the theoretical values due to selection of materials. The measuring chamber is fitted with pressure-balanced end covers to ensure that there is no pressure difference across the end covers of the measuring chamber and so to avoid distortion. Figure 9.5(c) shows a device that may also be available for 10- to 200-mm pipe bore sizes.

9.2.6 HELICAL ROTOR METER

Two radially pitched helical rotors trap liquid as it flows through the flowmeter causing the rotors to rotate in the longitudinal plane (Figure 9.6). Flow through the meter is proportional to the rotational speed of the rotors. It can be used on high viscosity liquids, but increased slippage may occur with low viscosity flows and reduce the accuracy. The rotors form a seal with each other and with the body of the flowmeter so that these parts must be manufactured to a high degree of precision. One manufacturer (Gerrard 1979) claimed that its positive displacement meter with helical elements had an accuracy of $\pm 0.5\%$ over a 150:1 flow range and a repeatability of $\pm 0.1\%$. Performance of 0.1% rate uncertainty and 0.01% repeatability with low effect from viscosity are now claimed. Some designs may withstand pressures up to 230 bar, compensate for temperature changes, and allow the passage of small solids. The use of Equation (9.1) will require a knowledge of how many closed compartments there are created by the two helical rotors and how many of them pass in one rotation.

One manufacturer confirms the precision level, suggesting that a higher pressure rating is possible, with maximum flow of about $13\text{ m}^3/\text{h}$, an operating temperature up to about 290°C , and viscosity up to 10^6 cP .

Applications are to polymers and adhesives, fuel oils, lubricating oil, blending, hydraulic test stands, high viscosity, and thixotropic fluids. This type of meter has also been developed for multiphase flows in North Sea oil applications (Gold et al. 1991).

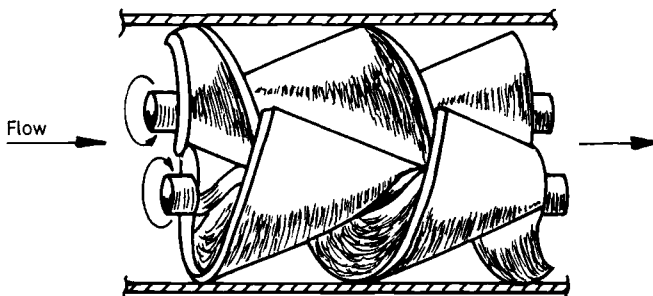


Figure 9.6. Helical rotor meter (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

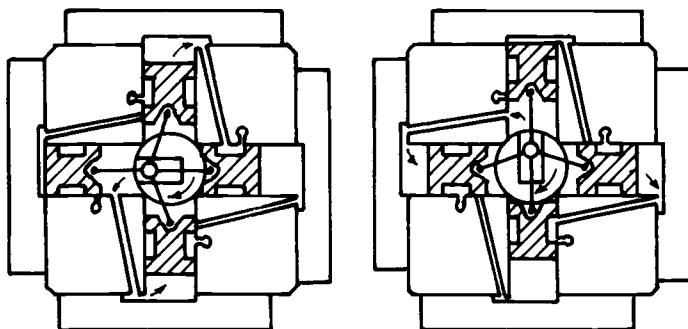


Figure 9.7. Reciprocating piston meter (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

9.2.7 RECIPROCATING PISTON METERS

One design in which four pistons trap liquid as it passes through the flowmeter is shown in Figure 9.7. The crankshaft rotates with a rotational speed proportional to the flow through the meter. The liquid that passes through each cylinder in one rotation of the shaft is equal to the swept volume of the cylinder.

Meters of this type have high accuracy claims and may also have a high pressure drop. The Aeroplane and Armament Experimental Establishment (Anon. 1966b) developed a three-cylinder meter for low flows. Accuracies of $\pm 0.5\%$ were quoted for kerosene flows of 9–182 l/h (2–40 gal/h) with $\pm 2\%$ for gasoline at rates below 32 l/h (7 gal/h). Increased slippage made high accuracy difficult to achieve at low flows, particularly for less viscous liquids.

One commercial producer quotes up to 33 m³/h (120 gal/min) with $\pm 0.5\%$ of reading and with an 1,800:1 turndown and another down to 1 l/h. Viscosity may be possible up to 30,000 cP or greater and temperatures up to about 280°C. Magnetic couplings may be used to reduce drag and avoid a rotating seal.

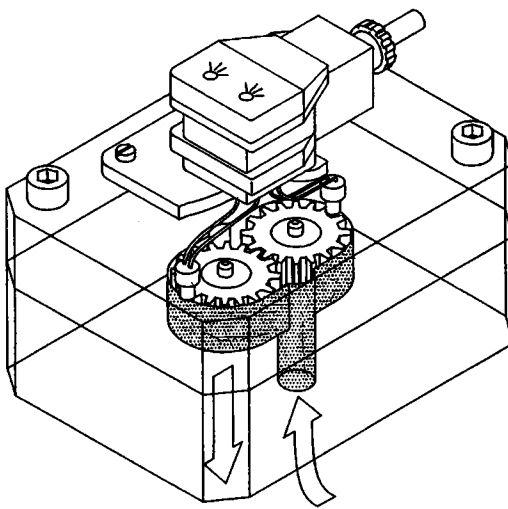
Endress et al. (1989) mentioned accuracy of ± 0.5 to 1% of rate, turndown of 10:1, maximum differential pressure of 5 bar, diameter range of 25–100 mm, and maximum pressure of 40 bar. Others may offer pressure ratings up to 200 bar.

Materials of construction may be predominantly stainless steel with seals of Viton, Teflon, neoprene, Nordel, Buna-N, etc. Filtration to 10 μm may be required, and magnetic coupling may be appropriate.

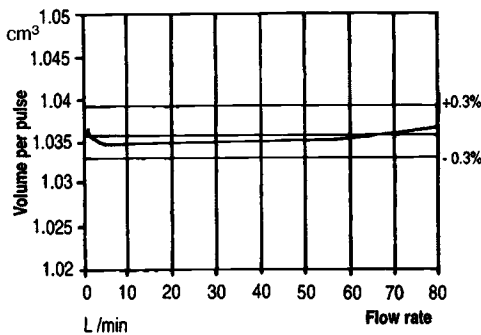
Applications include gasoline pumps, pilot plants, chemical additives, engine and hydraulic test stands, fuel consumption analysis, blending and ratio control, and mercaptan odorant injection.

9.2.8 PRECISION GEAR FLOWMETERS

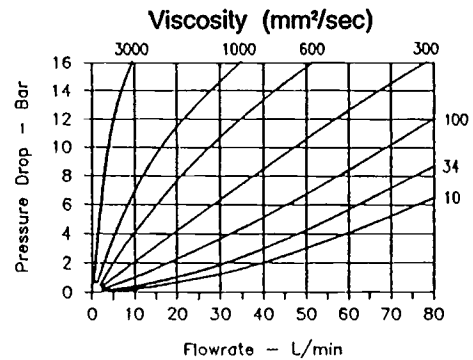
Figure 9.8(a) shows an example of a commercial device. The spaces between the gears and the chamber wall form the fluid transfer compartments. In this version, rotation is sensed by two electromagnetic sensors operating through a pressure-resistant and nonmagnetic element in the housing. Two sensors can be arranged to allow better resolution than one and to determine flow direction. Measurement uncertainty [Figure 9.8(b)] of $\pm 0.1\%$ rate is claimed. It is also claimed that rapid flow reversal can be followed (e.g., 80 l/min in about 0.01 s). Meters for flow ranges of as low as 0.001 l/min and as high as 1,000 l/min may be available with temperature ranges of -30 to 150°C and pressure up to 300 bar or more. Figure 9.8(c) shows typical pressure



(a)



(b)



(c)

losses for various viscosities. Plain bearings or ball bearings may be used depending on the liquid monitored.

Applications may include gasoline, paraffin, kerosene, diesel, mineral oil, hydraulic oil, ink, dyes, paint, greases, polyurethane, polyol-iso-cyanate, glue, paste, cream, resin, and wax. For the auto industry, applications are hydraulic systems, fuel consumption, paint spraying, batch processes, adhesive coatings, hydraulics; and for the chemical industry, applications are process plant, batching, leakage, and mixing ratios.

At least one design is available using a servomotor to maintain zero pressure drop across the meter (cf. Conrad and Trostmann 1981, Katz 1971 regarding servo-control).

9.3 CALIBRATION, ENVIRONMENTAL COMPENSATION, AND OTHER FACTORS RELATING TO THE ACCURACY OF LIQUID FLOWMETERS

This section is primarily concerned with sliding vane meters. It should be remembered that the drive from these meters, when mechanical, may have to be used on

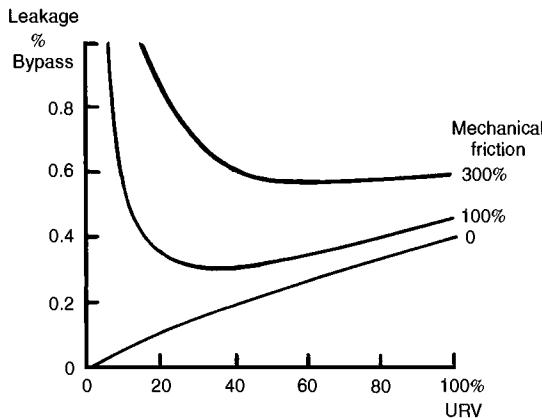


Figure 9.9. Typical effect on accuracy of leakage due to mechanical friction for a sliding vane positive displacement meter (reproduced from Baker 1983; with permission of the author).

a mechanical calibration device that loads the drive and causes a drag on the rotors. Manufacturers will clearly aim to reduce this as far as possible by the use of low friction ball bearings, etc. The mechanical methods of calibration adjustment are discussed first, and the theory behind them is elaborated in Appendix 9.A.5. Baker (1983) gave the characteristics of a good calibrator as

- ability to drive a high torque,
- long service life and low repair or replacement cost,
- fine adjustment and adequate range of adjustment, and
- constant ratio of output to input for whole cycle.

Baker (1983) also gives the effect of changes in mechanical friction in Figure 9.9 showing that greater friction leads to greater leakage past the vanes (cf. Keyser 1973 on the prediction of calibration shift).

9.3.1 CALIBRATION SYSTEMS

Disk and Friction Wheel

In this system a friction wheel runs on the surface of a disk, the two shafts being perpendicular in the same plane (Figure 9.10). By adjusting the radius at which the friction wheel runs on the disk, a change in shaft rotation ratio can be obtained. This system is usually incorporated into an epicyclic gearbox to allow most of the torque to be transmitted via meshing gears. The disadvantage is clearly the problem of ensuring that slip does not occur while, at the same time, the disk is not worn. The advantages are that the output is proportional to the input, and the system may be used for temperature compensation.

Clutch System

In this method (Figure 9.11), a system of clutches transfers the rotation from one shaft to another by means of an offset shaft. The advantage is that frictional transmission by a friction wheel is replaced by the tight frictional lock of the

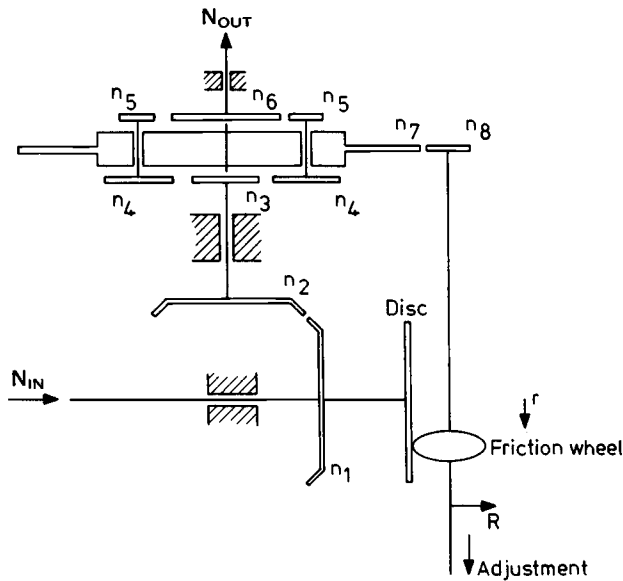


Figure 9.10. Epicyclic calibration adjustment system (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

clutch system enabling a high torque drive. The disadvantage is that the output is not proportional to the input but varies during each rotation of the shaft (Appendix 9.A.5). It is also less well suited to temperature compensation than the previous system. For more information, the reader is referred to Baker and Morris (1985).

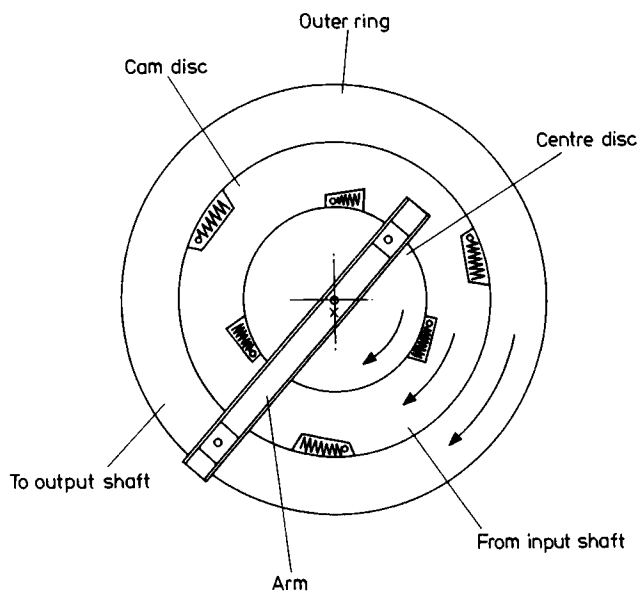


Figure 9.11. Clutch calibration system (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

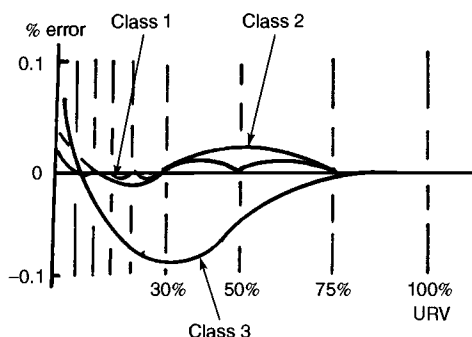


Figure 9.12. Calibration classes (based on information in Avery Hardoll brochures).

Gear Box

The output drive from the rotor shaft passes through a gear box, which allows considerable freedom for interchanging the gears to obtain a required calibration ratio. The disadvantage of this method is that it is not simple to change calibration, but the advantages are that, once done, there should be no further change and also the output is directly proportional to the input. It is not suitable for temperature compensation. Some gear train may be necessary, in any meter, to adjust the rotational speed to a

required volume flow per revolution. This will precede the calibrator, which can provide a fine adjustment. It may not be easy to provide a geared adjustment better than about 0.5%, and the calibrator will need to improve on this for a meter of this quality. The calibrator mechanism may also require that the output of the gear train has an under- or overreading to allow for the calibrator's physical behavior (Baker 1983).

Whichever method is used, the output will pass typically through a calibration mechanism to a mechanical counter which may include a batch recorder, ticket printer, and preset counter. In addition, a pulse transmitter may be included on the output shaft, and even a data link through an optical fiber.

Electronic

Electronic calibration adjustment, shown in Figure 9.12, is achievable using calibration data storage methods where class 3 has calibration at one flow rate, class 2 has calibration at four flow rates, and class 1 at eight flow rates.

9.3.2 CLEARANCES

There are four general types of clearance in these meters, which will be sources of leakage:

- clearance between moving blades and stationary members,
- clearance between retracting blades and rotors,
- clearance between two rotors moving relative to each other, and
- clearance between rotors and end housings.

A typical mesh size for a filter for one type of precision positive displacement meter was 0.250 mm, which is, presumably, of the same order of size as the clearances. If deposits form on the meter wall, they may affect this value, and the meter performance should be monitored carefully.

9.3.3 LEAKAGE THROUGH THE CLEARANCE GAP BETWEEN VANE AND WALL

If the rotor is stationary, the leakage is obtained from the following equation (derived in Appendix 9.A), which is also the same equation as Baker (1983):

$$q_{\text{slip}} = \frac{\ell}{12\mu} \frac{(p_u - p_d)}{L} t^3 \quad (9.2)$$

Ideally, the liquid in the gap would be carried at the same speed as the rotor blade, and so the leakage when the vanes are moving is

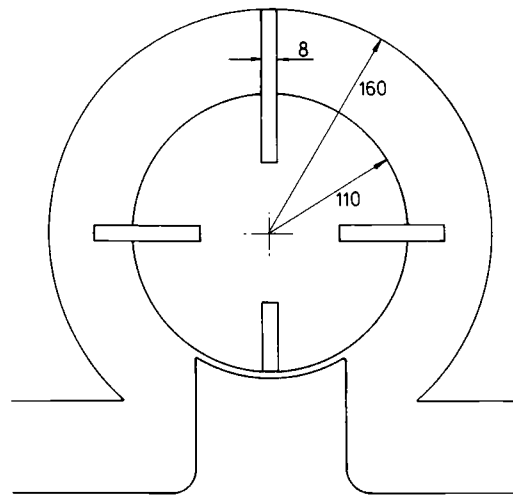
$$q_{\text{leakage}} = \left\{ \frac{Ut}{2} - \frac{1}{12\mu} \frac{(p_u - p_d)}{L} t^3 \right\} \ell \quad (9.3)$$

We can introduce values into this equation for the meter dimensions in Figure 9.13. Assuming an axial length of 225 mm and clearances of 0.1 mm, this gives a value of the leakage as

$$q_{\text{leakage}} = 0.156tN - 3.4 \frac{(p_u - p_d)}{\mu} t^3 \text{ m}^3/\text{s} \quad (9.4)$$

Baker and Morris (1985), who used these values in their estimate of errors due to changes in clearance, showed that changes are of order 0.1% for a 0.1-mm clearance.

Baker (1983) appeared to suggest clearances of 0.003–0.005 in. (75–125 μm), less than the filter mesh size mentioned earlier. However, Baker (1983) appeared to provide experimental values that gave a higher value than those obtained by Baker and Morris [Figure 9.14(a)]. This is an area of necessary compromise in design. Consistent clearance requires high quality production methods. However, a meter



All dimensions in mm

Axial length	225 mm
Clearances	0.1 mm
Wall thickness	10 mm

Figure 9.13. Dimensions of a typical sliding vane meter to obtain leakage (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control).

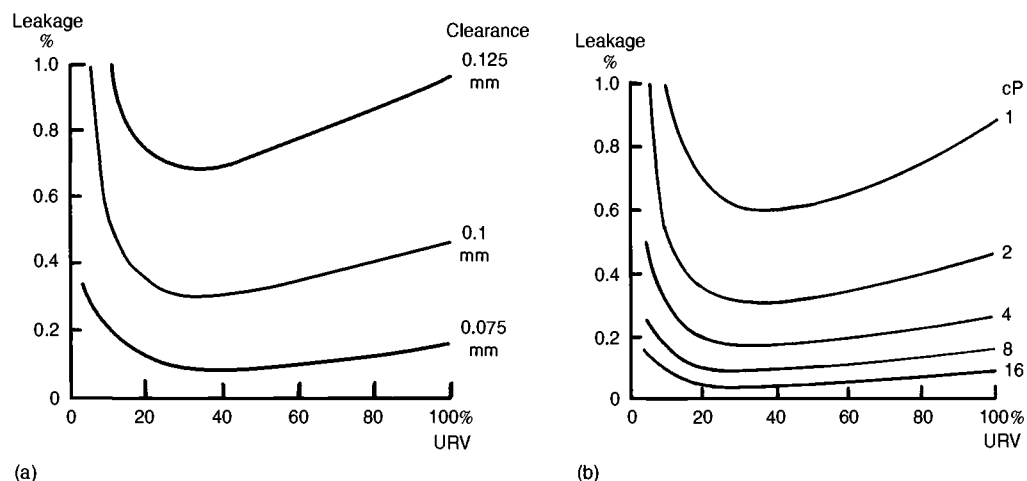


Figure 9.14. Typical effect on accuracy of leakage for a sliding vane positive displacement meter (reproduced from Baker 1983; with permission of the author): **(a)** With clearance width; **(b)** With viscosity.

with extremely tight clearances will have reduced service time before inevitable bearing wear causes some clearances to close, subsequently resulting in increased mechanical friction and possibly severe damage to the meter.

The other point to note from Equation (9.4) is that the larger the viscosity, the smaller the second term, which is responsible for the part of the leakage past the vanes in the meter, which is not proportional to speed of rotation. This appears to be borne out by Figure 9.14(b) (the figure does not distinguish the direction of leakage which is shown in the equation by a negative sign). Baker (1983) made the point that the positive displacement meter is the ideal meter for liquids with viscosity above about 4 cP such as No. 2 Fuel Oil or 40° API gravity crude oil.

9.3.4 SLIPPAGE TESTS

The existence of clearances will result in leakage of fluid past the vanes. This, in turn, will mean that the fluid metered may vary as a result of variation in leakage. This is known as slippage and will normally result, particularly at low rotational speeds, in more fluid passing than is registered, an effect that is anathema to the customs and excise authorities (cf. Plank 1951). In rare cases, back slip can occur, which means that the meter registers more fluid than has passed, and the customs and excise benefit. Thus slippage tests are often required, particularly where fuel is dispensed, because high slippage may result in low measure of liquid quantity and is sometimes exploited by using a slow flow rate to fill a vessel, thereby causing an underregistration of the liquid transferred. The last term in Equation (9.4) is not dependent on N , the rotational speed; hence, if N decreases, q_v will not decrease as fast, and there may be a small value of q_v when $N = 0$.

At $N = 1$ revolution per second the final term in Equation (9.4) is 0.01% of the speed-dependent first term. If on a slippage test the meter were to revolve as slowly as 0.01 revolutions per second, the relative size of the final term becomes about 1% and any changes in $(p_u - p_d)$, t , or μ will also be emphasised.

Back Slip

Although slip occurs when liquid passes through the meter in greater quantities than the rotation indicates, back slip is the apparent rotation of the meter while less liquid passes than expected. The former can be due to high bearing friction at low flows, increased clearances, increased pressure drop, etc. The reasons for the latter are less clear, although its occurrence during calibration tests appears to be observed occasionally. Possible causes of the observed behavior may be categorized as back slip due to meter effects, system effects, and calibration vessel.

Back Slip Due to Meter Effects

- a. The back slip observations are for very small quantities of liquid passed (15–20 liters). For these small quantities, the clutch type calibrator's varying rotational speed could result in an apparent high rotation. Errors could be $\pm 3\%$.
- b. A meter in which two adjacent vanes had a slight volume difference could also exhibit rotational variation as in (a).
- c. The temperature compensation system, where fitted, or other temperature effects due to the slow movement of liquid may cause errors.
- d. The digital readout may not allow measurement to within a liter.

- e. The low flow rate and rotation may markedly alter the liquid flows on sliding and stationary members. The bearing and clearance drag may be changed.

Back Slip Due to System Effects

- f. The quantity of liquid in the pipework may not be precisely the same before and after the slip test. For instance, 20 liters of liquid in a slip test fill 2.6 m of a 100-mm diameter pipe.
- g. In tests using manual and automatic valves, one may be closed at the start of the test and both closed at the end, slightly altering the pipe volume.
- h. Upstream valve control may cause pressure changes leading to volume changes.
- i. Gas may come out of solution during the test.

Back Slip Due to Calibration Vessel

- j. The measurement uncertainty of the vessel is not **better than** about $\pm 0.2\%$ for so short a run.

It appears that various effects occasionally may come together to cause the observation of back slip.

The observations of **back slip** appear likely to result from uncertainties in operation of the meter at **low rotation**. Variation during rotation can become important if flows of about 15 liters are involved because this may result, for a 100-mm meter, in less than two rotations. Thus the nonuniform calibrator transmission or variation in vane clearance may occasionally result in an increased recorded rotation for a given volume passed.

9.3.5 THE EFFECTS OF TEMPERATURE AND PRESSURE CHANGES

The effects of temperature and pressure variation are given in Equations (9.A.9) to (9.A.18), using values in Figure 9.13. These values are, therefore, for a moving vane type meter but will give some indication of the effect on others. Making use of values of α_m of 0.23×10^{-4} for aluminum, 0.11×10^{-4} for mild steel and cast iron, and of α_l of 9×10^{-4} for oil, Table 9.1 gives values for temperature changes of 10°C and pressure changes of 1 bar taking E for mild steel as 207 GN/m^2 . One manufacturer gives the effect of changes in pressure and temperature on the meter excluding the effect of liquid changes, and this is reproduced in Figure 9.15.

9.3.6 THE EFFECTS OF GAS IN SOLUTION

Another effect not considered so far will be the presence of dissolved gas in solution. The dissolved gas may cause no change in liquid volume, although it may have a large volume in gaseous form. The mass of gas involved will often be small enough to neglect.

However, under sudden pressure reduction, the gas may come out of solution. This might occur during a slippage test, if the test is controlled on a valve upstream of the meter, and might cause the rotor to turn due to the presence of the gas alone.

The problem can also arise when metering liquified petroleum gas (LPG) if the back pressure on the meter is insufficient.

Table 9.1. The effects of temperature and pressure changes

Description	Metal/ Liquid	$\Delta q_v/q_v$ (%)	$\Delta t/t$ (%)
For temperature change of 10°C			
Flowmeter	Al	0.07	—
expansion	M.S.	0.03	—
Differential	Al	—	37
expansion	M.S.	—	18
	Al/M.S.	—	19
Liquid-density change	Oil	0.9	—
For pressure change of 1 bar			
Flowmeter			
expansion	M.S.	0.003	—
Leakage gap radial	M.S.	—	1.2
Leakage gap axial	M.S.	—	0.4

9.4 ACCURACY AND CALIBRATION

Accuracy achievable from these meters will be highly dependent on the particular design. Values have been suggested earlier. At the top end, the sliding vane meters can achieve an accuracy that may be limited by the means of calibration. It is highly likely that precision devices such as the oval gear meter will approach the level of performance of the sliding vane meters.

The calculations of the effect of the pressure and temperature in Table 9.1 indicate that close tolerance control is necessary to achieve the required performance. However, it should be remembered that, although liquid density may change the reading of mass flow measurement, the requirement in many cases is for volume measurement.

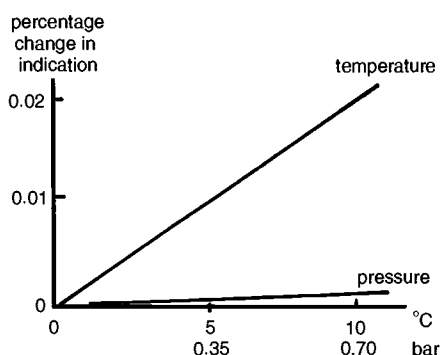


Figure 9.15. Sensitivity of a sliding vane meter to pressure and temperature change excluding the effect of liquid changes (based on Avery Hardoll data).

measurement.

The effect of changes in Δp and μ are indicated by Equation (9.4), and this clearly illustrates the errors due to change in clearances, pressure drop, or viscosity. Wear in a meter will cause t and Δp changes, whereas temperature variation will cause changes in μ .

Apart from the published values already mentioned, Hayward (1977b) developed a technique for measuring the repeatability of flowmeters, which involved comparing readings from two identical flowmeters in series. Tests at National Engineering

Laboratory, Scotland, suggested that repeatabilities better than 10 ppm (0.001%) could be obtained with vane-type flowmeters provided that the drift in readings, which seemed to arise in the shaft encoder, could be eliminated.

Hayward (1979) commented that the rangeability was about 20:1, linearity was $\pm 0.05\%$, and accuracy when newly calibrated was $\pm 0.2\%$ of volume over the range. He suggested that these meters perform best in the maximum flow rate range of 0.002–0.05 m³/s.

Barnes (1982) underlined the superior accuracy, repeatability (within 0.05%), and reliability of positive displacement flowmeters but drew attention to inaccuracies that arose if the liquid contained free or entrained gases. Data from fuel oil, gasoline, and propane were presented to show that the accuracy was highest when the meters were used to measure flow at about 25% of the rated flow capacity. At both higher and lower flows, increased slippage occurs and causes the meter to underregister. The meters were also relatively insensitive to changes in viscosity. Barnes suggested that if a meter initially calibrated for a liquid of 1 cP was used with a liquid of 100 cP, the resulting shift in accuracy was of order 1.2%.

Reitz (1979) quoted the following figures: repeatability within 0.02%, accuracy $\pm 0.25\%$ over a 20:1 flow range, and pressure loss 3 psi (0.2 bar) for liquids of viscosity 1 cP. He suggested that fluids with viscosities in the range 0.1 cP (gas) to 10⁶ cP (liquid) could be metered in this way. He noted that errors arise from excessive wear, which increases slippage and causes underregistration, and entrained gas bubbles, which occupy part of the measuring chamber and cause the meter to give too high a reading.

Kent Meters (Scanes 1974) quoted an accuracy of $\pm 0.5\%$ for a domestic oil meter designed for rates as low as 0.1–10 l/h. The temperature range was quoted as -10°C to 35°C , although temperatures as high as 90°C had apparently been recorded at meter boxes sited in direct sunlight.

One manufacturer gives meter calibration curves for the company's range of meters (Figure 9.16) and claims linearity in the range 0.1–0.3% with repeatability for all but one of their product sizes in the 0.01–0.02% range.

9.5 PRINCIPAL DESIGNS OF GAS METERS

As indicated in Section 9.1, the wet gas meter and the diaphragm meter have been in use for over 100 years, and both still provide service for many applications. However, the predominant use of the wet gas meter has changed to that of a secondary calibration standard, whereas mechanical meters offer an important option for commercial gas metering.

9.5.1 WET GAS METER

Figure 9.17 shows the wet gas meter, which uses a water (or other suitable liquid) bath as the gas seal to create closed compartments for the transfer of the gas. It is a high precision meter capable of an operating range of about 10:1, but it is rather bulky. It was originally developed for the measurement of gas usage in industrial premises but recently has found a niche as a transfer standard for gas measurement.

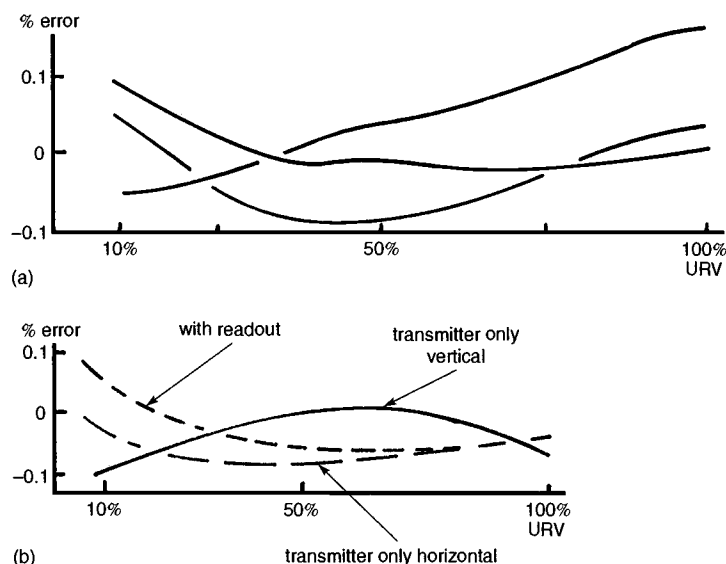


Figure 9.16. Typical calibration curves for a sliding vane PD meter (based on Avery Hardoll data): **(a)** for various models; **(b)** for various registration loads on the meter shaft output.

It needs careful setting up to achieve the highest accuracy. Dijstelbergen (1982) made the following points.

- With careful use, the calibration curve is stable within 0.1%.
- The use of water has been replaced by Caltex Almag to avoid the problem of gas take-up of the water.
- The meter can be used as a secondary standard at atmospheric conditions.

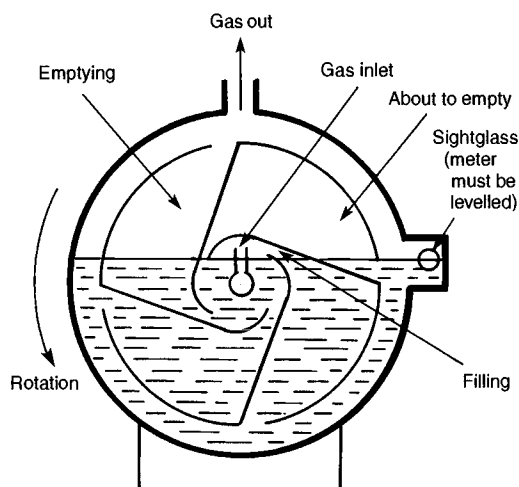


Figure 9.17. Wet gas meter.

9.5.2 DIAPHRAGM METER

Figures 9.18(a, b) show a diagram of the diaphragm meter. It is essentially a piston meter in which the compartments A, B, C, and D form the cylinders and the bellows provide a common piston for A and B and similarly for C and D. The flow into and out of the compartments is controlled by the sliding valve. In Figure 9.18(a), A has emptied and B has filled, and the valve is moving to allow A to fill and B to empty. At the same time C is filling from the inlet chamber due to the valve's position, and D is emptying into the outlet manifold. In Figure 9.18(b), gas is entering the inlet chamber above the valves and is routed into compartment A, while gas leaves compartment B. At the

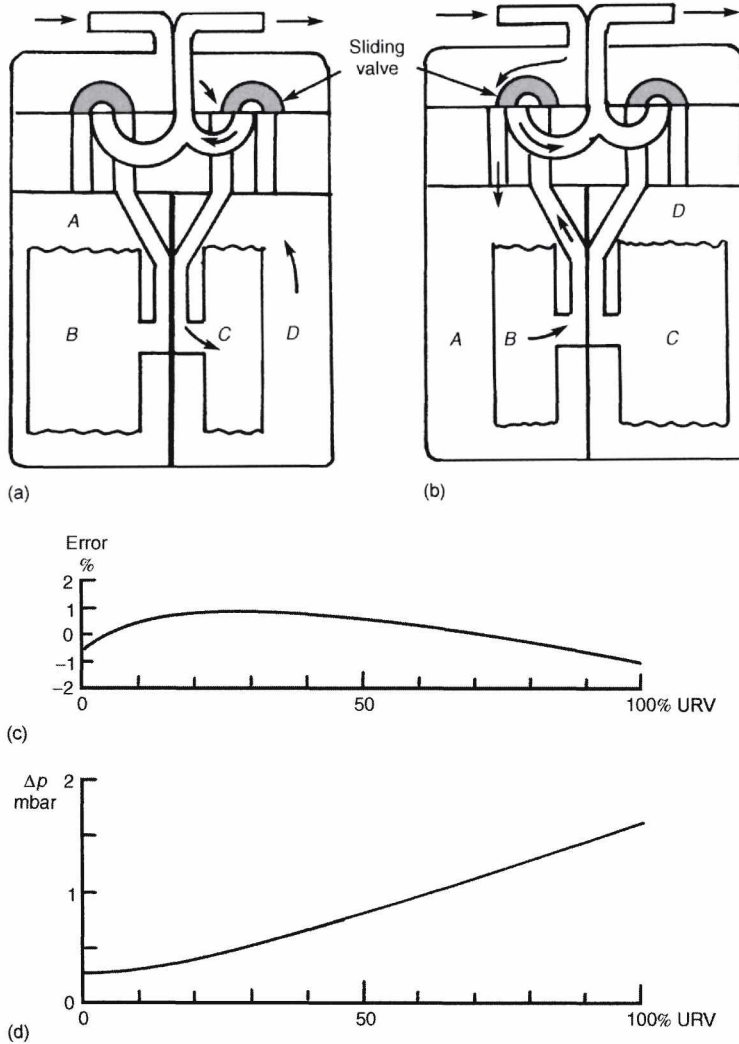


Figure 9.18. Diaphragm meter. **(a)** A, empty; B, full; C, filling; D, emptying. **(b)** A, filling; B, emptying; C, full; D, empty. **(c)** Typical error curve (reproduced with permission of GMT and IMAC Systems Ltd.). **(d)** Typical pressure loss curve (reproduced with permission of GMT and IMAC Systems Ltd.).

same time compartments C and D have reached the change-over position so that C has filled and D has emptied, and the valve is changing over to allow C to empty and D to fill. Dijstelbergen (1982) made the following points.

- If the bellows are made either of a special type of leather or of fiber-reinforced rubber, they may dry out with low water content gases, causing overregistration.
- Most meters are manufactured in accordance with OIML recommendation (Organisation Internationale de Métrologie Légale) and should operate within 2% at high flows and 3% at low flows.
- Rangeability may be up to 150:1.

The typical measurement uncertainty is $\pm 1\%$ [Figure 19.18(c)]. The maximum allowable inlet pressure on some designs is about 0.1 bar above atmospheric, but it may be up to 1 bar on other special versions. The coupling from the mechanical meter to the output may consist of a magnetic coupling to the display. Operating temperature range is typically -20 to $+60^\circ\text{C}$. The meters operate to DIN 3374 and BS 4161. The pressure loss through the meter is about 1.5 mbar [Figure 19.18(d)].

9.5.3 ROTARY POSITIVE DISPLACEMENT GAS METER

The rotary positive displacement gas meter is, sometimes, referred to as a Roots meter, but this name, in a particular style, is the registered trademark of a particular manufacturer's design (Dresser). Figure 9.19(a) shows a diagram of another design of this meter. Two rotors mesh and rotate within an oval body contour so that leakage is at a minimum between the rotors, but the gas is transferred at the outer point of the oval. In recent years they have performed to an increasingly high specification and offer one of the best options for high precision gas flow measurement. There is a possible problem due to pulsating flow resulting from the use of these meters (Dijstelbergen 1982), and rapidly changing flow rates may damage them or downgrade their accuracy. The two rotors have a very small gap between (described by one manufacturer as less than a human hair), and the manufacture is to tight tolerances on impellers and meter body. It is, therefore, necessary for the gas to be free of particulate matter and for the meter to be mounted so that the tight tolerances are not distorted. The gas flow can be distributed to give an even distribution over the full length of the rotor entry.

Maximum flows are from $28 \text{ m}^3/\text{h}$ at 0.5 in. (13 mm) water gauge to $360 \text{ m}^3/\text{h}$, with a pressure loss of 0.5 to 1 in. (13–25 mm) of water gauge. Figure 9.19(b) shows a typical performance characteristic with a performance well within 0.5% over a 10:1 turndown. In fact, the claims are that 80:1 may be achievable. Temperature range is from -40 to 65°C .

Transmission can be through timing gears on the end of the shafts, with direct counter driven or with magnetic couplings.

The rotors may be hollow extruded aluminum with self-cleaning tips and a body of cast aluminum. The surfaces may be anodized.

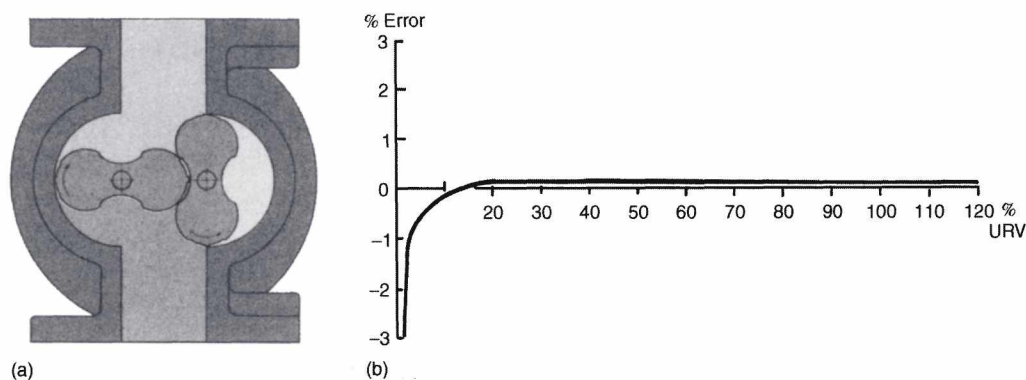


Figure 9.19. Rotary positive displacement gas meter (reproduced with permission of Romet Ltd. and IMAC Systems Ltd.): **(a)** Diagram of meter; **(b)** Plot of the performance characteristic for the meter.

The CVM meter is of similar design to Figure 9.3(b) with a main rotor that consists of four vanes or moving walls that form metering compartments. The second rotor is a sealing rotor that returns the vanes to the inlet side of the meter. The second rotor may be two- or three-lobed, and its rotation will be precisely linked to that of the main rotor. According to Dijstelbergen (1982), it is suitable for high pressure metering, does not cause pulsations in the flow, handles flows up to $1,200 \text{ m}^3/\text{h}$, and a repeatability of 0.05% may be achievable.

9.6 POSITIVE DISPLACEMENT METERS FOR MULTIPHASE FLOWS

In a patent (1981), Arnold and Pitts proposed temperature and pressure sensing and a gamma-ray densitometer for sensing density inside a positive displacement flowmeter (Figure 9.20), but they gave no experimental data. However, the patent presents an attractive device, which has the attribute of forcing the flow into a closed volumetric space regardless of the flow components. A major disadvantage is the possibility of solids causing it to seize up and block the line, requiring designs to have large clearances and consequent loss of precision.

An alternative approach has been taken by Gold et al. (1991) who used a helical rotor meter (Figure 9.21). In this design, two rotor sections are used in series, and between them is a section where the condition of the multiphase flow is measured.

Gold et al. (1991) described their solution, which controls the flow being measured and so avoids the problem of slip between phases. The helical rotors are also claimed to be more resilient, with suitable bearings, to the rigors of multicomponent flow, reducing the possibility of damage due to transients. Between the two rotors,

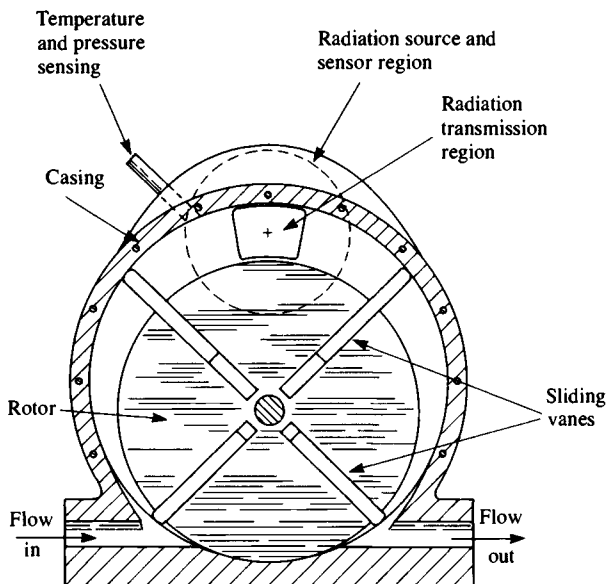


Figure 9.20. Diagram based on Arnold and Pitts' patent (1981) (reproduced from Baker 1991a; with permission of the Council of the Institution of Mechanical Engineers).

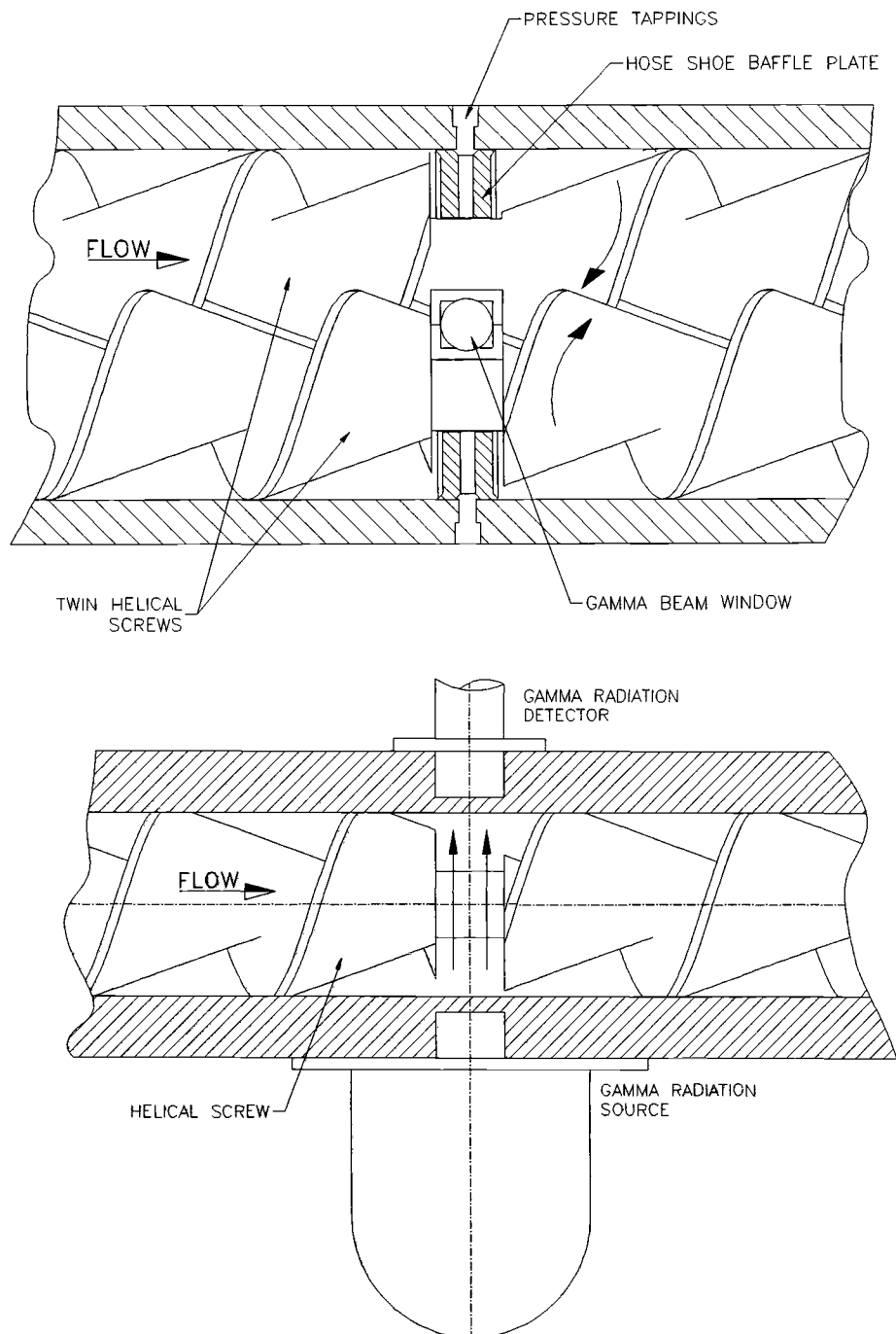


Figure 9.21. Helical multiphase meter (Gold et al. 1991, Priddy 1994; reproduced with permission of ISA Controls Ltd.; design is the property of BP International).

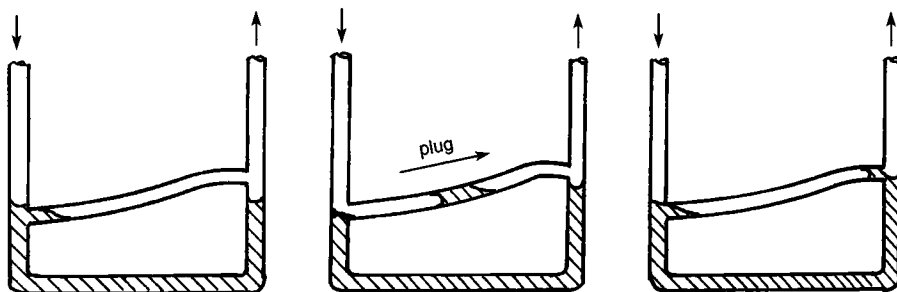


Figure 9.22. Meter using liquid plugs to measure low flows.

the density is measured. The preferred orientation is with flow vertically downward. The bearings are simple low friction journals but with thrust bearings that give long service.

The meter was seen as suitable for both surface and subsea service and aimed at a specification of $\pm 5\%$ rate for total flow and $\pm 5\%$ rate for oil flow rate with $\pm 10\%$ for both gas and water. The field trial meters were designed for up to 20,000 barrels per day ($130 \text{ m}^3/\text{h}$) with at least 10:1 turndown and $\pm 5\%$ uncertainty. A nominal 4-in. (100-mm) bore meter was specified. The body had about 0.3-m outer diameter. The meter length was 1.2 m, and with nucleonic densitometer it weighed about 450 kg. The scatter on test appears to be a maximum of about 20% in mid values of void fraction. Priddy (1994) reported tests at Prudoe Bay, Alaska, which are extremely promising in terms of the specific aims.

9.7 METER USING LIQUID PLUGS TO MEASURE LOW FLOWS

Figure 9.22 shows a diagrammatic representation of a meter that uses liquid plugs of a sealing fluid such as distilled water to measure the flow of gas. The plugs are carried through the tube as shown, and the passage and velocity are sensed optically. Accuracy claims are $\pm 1\%$ at the optimum rate, and the range is 8–3,000 ml/h. In this device, dry gases pick up moisture. It is not clear how the size of plug, which presumably varies, is allowed for. It may be that the optics provide a compensation.

9.8 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

Applications

The applications will vary to some extent for the various designs; in particular, the accuracy and cost of the meters will make them more suitable for particular applications. In general, they operate on clean or well-filtered fluids. The clearances are so fine that dirty or aggressive fluids may prevent operation and cause total blockage in the line. For the same reason, the instruments must be correctly mounted to avoid mechanical distortion, which could affect the clearances.

In particular, for the sliding vane meter, one manufacturer claimed: blending, batching, dispensing, inventory control, custody transfer of oils, solvents, chemicals, paints, fats, and fertilizers.

For high viscosity fluids, the flow passage may create too great a pressure drop. For some oils, the temperature of the liquid may need to be sensed and compensation used.

Presumably, the design developed for multiphase flows will eliminate the need for well-filtered fluids.

Advantages

The advantages are seen in high precision instruments, and their accuracy after calibration is possibly only limited by the calibration facility used. Most are little affected by viscosity. The most accurate ones among them, therefore, provide a device that is highly suitable for custody transfer and fiscal transfer applications.

Disadvantages

The meters may create pulsations and may be subject to damage from rapidly varying flows. Damage to the meters causing them to stop rotating will also cause total line blockage. They may have a high initial price and may require careful maintenance with high associated costs. The meters may create a substantial pressure loss.

9.9 CHAPTER CONCLUSIONS

What is likely to be the future of such meter designs? No doubt other clever designs will be introduced such as that selected as a utility gas meter (Kim et al. 1993b, O'Rourke 1993, 1996), which uses an undulating membrane (Figure 9.23). The sensor consists of such a membrane, which traps discrete volumes of gas as they move through the flow chamber in pulses. Movement is sensed by piezoelectric elements bonded to the membrane. The design is patented. Tests suggest that it is capable of operating over a range of $0.007\text{--}7\text{ m}^3/\text{h}$ with uncertainty of $\pm 1\text{--}2\%$ for -40 to 140°F (-40 to 60°C), and with a life of 20 years (10-year battery life).

More generally, these essential workhorses of the industry will continue to benefit from improvement in materials, bearings, and manufacture, so that they can be used with more difficult fluids.

Electronic control will also ensure that their standard applications are possible with greater automation and precision.

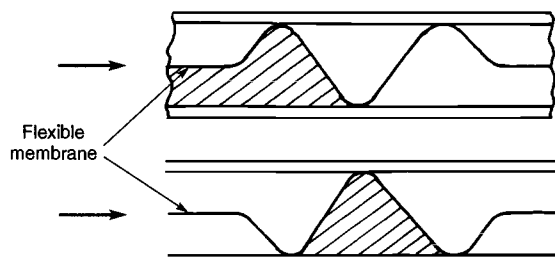


Figure 9.23. Undulating membrane meter (after Kim et al. 1993 and O'Rourke 1993, 1996).

The successful use of positive displacement meters in such a hostile environment as oil-well flows suggests that other difficult applications may be within range.

Total blockage of the line in the event of failure is, perhaps, the greatest disadvantage, and ways of dealing with this through an automatically controlled bypass, in the event of blockage, may overcome this problem.

APPENDIX 9.A

Theory for a Sliding Vane Meter

The theory of the meter was set out by Hahn (1968). In his paper he considered the motion of the vanes and calculated the radial velocity and acceleration if the internal controlling cam gave a sinusoidal motion. He noted that the resultant forces between vane and wall contributed to the wear of the vanes. Avoidance of jerky motion and shock stresses is clearly important.

He also calculated the volumetric flow rate (cf. Baker and Morris 1985 whose work was in ignorance of that due to Hahn). Hahn pointed out that the radial motion of the vanes results in a sinusoidal fluctuation superimposed on the volume versus rotation characteristic and that the amplitude of this fluctuation can be of order 0.05–0.1% for various designs. This, in turn, would be shown in an irregularity of the rotation of the shaft of as much as 1.25%. However, in practice there will be a balance between varying flow, pressure variation, and rotor velocity variation. Hahn also gave characteristics showing variation of within 0.2% over a range of 14:1 (slight drop of 1.5–2% with increasing flow) and changes of +0.15% for a change in viscosity from 0.75 to 4 cP.

9.A.1 FLOWMETER EQUATION

The ideal behavior of the meter is given by the product of transferred volume, the number of volumes transferred per revolution, and the rotational speed. For the actual behavior allowance has to be made for the leakage that takes place past the moving rotors. The bulk volumetric flow for a meter of the sliding vane type is given by

$$q_v = [\pi (r_o^2 - r_i^2) - \delta] N \ell_{ax} - \text{Leakage flow} \quad (9.A.1)$$

where ℓ_{ax} is the axial length of the chamber. The flow in each gap (Figure 9.A.1) will be governed by

$$\frac{p_u - p_d}{L} = -\mu \frac{d^2 u}{dy^2} \quad (9.A.2)$$

where μ is the fluid viscosity at y in the gap, t is the clearance, p_u and p_d are the upstream and downstream pressures, L is the length of the clearance gap or, in the case of the vanes, the thickness of the vanes, and μ is the viscosity. This results in a

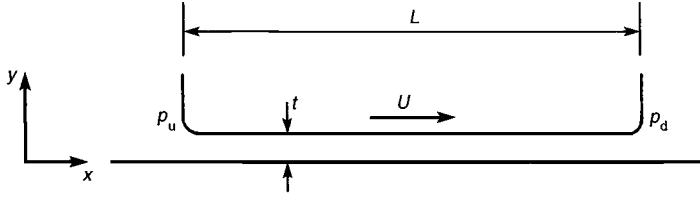


Figure 9.A.1. Geometry of leakage path between fixed and moving parts of the meter (after Baker and Morris 1985).

velocity distribution of

$$u = \frac{1}{2\mu} \frac{(p_u - p_d)}{L} (t - y) y + \frac{Uy}{t} \quad (9.A.3)$$

and a volumetric flow through the gap of

$$q_v = \left\{ \frac{1}{12\mu} \frac{(p_u - p_d)}{L} t^3 + \frac{Ut}{2} \right\} \ell \quad (9.A.4)$$

where ℓ is the axial length of the gap.

If the rotor is stationary, this quantity becomes the slippage;

$$q_{\text{slip}} = \frac{\ell}{12\mu} \frac{(p_u - p_d)}{L} t^3 \quad (9.A.5)$$

Ideally, the liquid in the gap would be carried at the same speed as the rotor blade:

$$q_{\text{ideal}} = Ut\ell \quad (9.A.6)$$

From Equations (9.A.4) and (9.A.6)

$$q_{\text{leakage}} = q_{\text{ideal}} - q_v \quad (9.A.7)$$

so

$$q_{\text{leakage}} = \left\{ \frac{Ut}{2} - \frac{1}{12\mu} \frac{(p_u - p_d)}{L} t^3 \right\} \ell \quad (9.A.8)$$

From Equations (9.A.1) and (9.A.8) and assuming that the leakage can be taken as the sum of leakage through clearances t_j of length in the direction of leakage L_j and in the direction perpendicular to the leakage ℓ_j and with velocity of moving component U_j ,

$$q_v = [\pi (r_0^2 - r_i^2) - \delta] N \ell_{\text{ax}} - \sum_j \left\{ \frac{U_j t_j}{2} - \frac{1}{12\mu} \frac{(p_u - p_d)}{L_j} t_j^3 \right\} \ell_j \quad (9.A.9)$$

An expression for the pressure drop across the vanes is obtained from the torque balance on the rotor. The turning force due to $(p_u - p_d)$ will act on a vane and will precisely balance the frictional torques caused by bearings, vane clearance spaces, and readout mechanism:

$$\frac{1}{2} (p_u - p_d) (r_0^2 - r_i^2) \ell_{\text{ax}} = T_0 + \frac{T_1 N}{N_{\text{max}}} \quad (9.A.10)$$

where T_0 is a constant torque and $T_1 N/N_{\max}$ is a speed-dependent torque. Part of the torque $T_1 N/N_{\max}$ will be due to the clearance drag torque:

$$\mu \frac{U_j}{t_j} L_j \ell_j r_j$$

where r_j is the radius from the rotational axis of the j th clearance.

9.A.2 EXPANSION OF THE FLOWMETER DUE TO TEMPERATURE

If α_m is the coefficient of linear expansion of the metal, then a change in temperature ΔT_m of the metal of the flowmeter will cause a change in length

$$\frac{\Delta \ell}{\ell} = \alpha_m \Delta T_m \quad (9.A.11)$$

so that

$$\frac{\Delta q_v}{q_v} = 3\alpha_m \Delta T_m \quad (9.A.12)$$

giving the fractional change in measured volume per revolution of the flowmeter rotor.

Differential Expansion

If the expansion of the rotor differs from that of the casing due either to different temperatures or to different materials, then the clearances will be designed for the greatest relative movement between these members. The change in the value of the clearance t will be

$$\frac{\Delta t}{t} = \alpha_m \Delta T_m \frac{r_0}{t} \quad (9.A.13)$$

between uniform temperature conditions and conditions when the temperature of the internal rotor and the casing differ by ΔT_m . For the case where the temperatures are the same but the coefficients of expansion differ, this becomes

$$\frac{\Delta t}{t} = \Delta T_m \frac{r_0}{t} (\alpha_{m1} - \alpha_{m2}) \quad (9.A.14)$$

Liquid Density Change

The change in the bulk flow due to expansion of the liquid will be

$$\frac{\Delta q_v}{q_v} = \alpha_\ell \Delta T_\ell \quad (9.A.15)$$

where α_ℓ is the coefficient of volume expansion of the liquid.

Because many liquids are bought and sold by volume, this change is of a different significance from those given before. The volume passed by the meter may, therefore, be important. In some cases, the change in liquid density may be ignored.

9.A.3 PRESSURE EFFECTS

Provided that the meter is supplied with an outer pressure vessel, there should be no net pressure causing distortion of the inner meter. However, this is not invariably so. The meter will then suffer from two distortions, hoop stress and axial stress. For convenience, we will consider these effects separately and ignore the effects of one on the other.

Bulk Flow Error

If Δp is the difference between internal and external pressure and E is Young's modulus of elasticity, then the change in the outer radius will be

$$\frac{\Delta r_0}{r_0} = \frac{r_0}{E} \frac{\Delta p}{t_0} \quad (9.A.16)$$

where t_0 is the thickness of the outer casing. The axial stress will cause an elongation of

$$\frac{\Delta \ell_{ax}}{\ell_{ax}} = \frac{r_0}{2E} \frac{\Delta p}{t_0} \quad (9.A.17)$$

where ℓ_{ax} is the axial length of the meter's measuring chamber. Together these lead (neglecting Poisson's ratio effects) to an increase in measuring volume, which causes an error in the volumetric flow rate of

$$\frac{\Delta q_v}{q_v} = \frac{r_0}{2Et_0} \left\{ \frac{5r_0^2 - r_i^2}{r_0^2 - r_i^2} \right\} \Delta p \quad (9.A.18)$$

Leakage Gap

In certain designs, the hoop stress will cause an increase in the clearance of

$$\frac{\Delta t}{t} = \frac{\Delta r_0}{t} = \frac{r_0^2 \Delta p}{tt_0 E} \quad (9.A.19)$$

whereas the axial stress will cause an increase in end clearance at each end of

$$\frac{\Delta t}{t} = \frac{\Delta \ell_{ax}}{t} = \frac{r_0 \ell_{ax}}{4tt_0} \frac{\Delta p}{E} \quad (9.A.20)$$

Clearly these calculations are simplifications of the actual expansion that will occur, but they allow some estimate of likely errors to be made.

9.A.4 METER ORIENTATION

Bearings at Each End of Shaft

Assuming that the shaft behaves like a beam simply supported at each end and that the rotor weight is evenly distributed along the shaft, then the maximum deflection at the point midway between the bearings will be given by

$$\text{Deflection} = \frac{5Mg\ell_{ax}^3}{96\pi Er_s^4} \quad (9.A.21)$$

where M is the mass of the rotor, g is acceleration due to gravity, and r_s is shaft radius.

Bearing(s) at One End of Shaft Only

Assuming that, in this case, the bearing(s) hold the shaft without deflecting, the shaft bending will be equivalent to a uniformly loaded encastré beam. The end deflection of this, which will be the maximum value, is given by

$$\text{Deflection} = \frac{Mg\ell_{ax}^3}{2\pi Er_s^4} \quad (9.A.22)$$

It is apparent that, other things being equal, the deflection of the end bearing shaft will be nearly 10 times greater than the shaft with bearings at each end. This will require greater clearances and, consequently, allow greater leakages. To avoid this problem, it may be recommended that meters be mounted with shafts vertical.

9.A.5 ANALYSIS OF CALIBRATORS

The rotation of these meters gives the volumetric flow. If every rotation causes a pulse, it is a simple matter with modern electronics to scale the pulse rate to give any required calibration factor or combine the electrical output from a temperature sensor to give a temperature compensation. However, in many applications, mechanical calibration and temperature adjustment is required as a result of existing installation constraints, and three main methods have been used or are possible.

Gearbox

The gearbox with interchangeable gears will not be discussed in this section. The drawback of the gearbox is clearly the limited adjustment possible. We are concerned with two calibrators, which may introduce special features into the meter performance.

Epicyclic System

A schematic diagram of this system is given in Figure 9.10. The drive from the flowmeter shaft enters with rotation N_{IN} . The drive then divides. The main drive is via gears n_1, n_2, n_3 , and so to the epicyclic unit. The other is via the disk and adjustable friction wheel to n_8, n_7 . The recombined drives leave the epicyclic unit via n_6 with rotation N_{OUT} . The integrity of the unit depends heavily on the wheel and disk contact. There must be no slip between these components, but they must also not wear and must be adjustable (r the radius of frictional contact being changed by the adjustment).

The ratio of N_{OUT}/N_{IN} is given by

$$\frac{N_{OUT}}{N_{IN}} = \frac{r}{R} \frac{n_8}{n_7} \left\{ 1 - \frac{n_3 n_5}{n_4 n_6} \right\} - \frac{n_1 n_3 n_5}{n_2 n_4 n_6} \quad (9.A.23)$$

where R is the radius of the friction wheel.

Putting typical values of the numbers of teeth (n) into Equation (9.A.23)

$$\begin{aligned} n_2 &= n_1 \\ n_4 &= \frac{20n_3}{15} \\ n_6 &= \frac{20n_5}{15} \\ n_7 &= 8n_8 \end{aligned}$$

then

$$\frac{N_{\text{OUT}}}{N_{\text{IN}}} = \frac{r}{R} \frac{1}{8} \left\{ 1 - \left(\frac{15}{20} \right)^2 \right\} - \left(\frac{15}{20} \right)^2 = 0.055 \frac{r}{R} - 0.563$$

If $R = 10$ mm and we let $r = 10$ mm,

$$\frac{N_{\text{OUT}}}{N_{\text{IN}}} = -0.508$$

If $r = 11$ mm,

$$\frac{N_{\text{OUT}}}{N_{\text{IN}}} = -0.503$$

or a change in calibration of 1%.

We may also write down the modified version of Equation (9.A.23), assuming that a small amount of slip occurs between disk and wheel equivalent to a rotation N_s in N_{IN} . Then

$$\frac{N_{\text{OUT}}}{N_{\text{IN}}} = \frac{r}{R} \frac{n_8}{n_7} \left(1 - \frac{n_3}{n_4} \frac{n_5}{n_6} \right) \left(1 - \frac{N_s}{N_{\text{IN}}} \right) - \frac{n_1}{n_2} \frac{n_3}{n_4} \frac{n_5}{n_6} \quad (9.A.24)$$

In this case, a relation between the torque in T_{IN} and the torque out T_{OUT} , the friction force between wheel and disk F , is given by

$$T_{\text{IN}} = \frac{N_{\text{OUT}}}{N_{\text{IN}}} T_{\text{OUT}} + \frac{N_s}{N_{\text{IN}}} rF \quad (9.A.25)$$

At the start of rotation if N_{OUT} were zero and N_{IN} were nonzero, then

$$T_{\text{IN}} = \frac{N_s}{N_{\text{IN}}} rF$$

and

$$\frac{n_s}{N_{\text{IN}}} = 1 - \frac{\frac{n_1}{n_2} \frac{n_3}{n_4} \frac{n_5}{n_6}}{\frac{r}{R} \frac{n_8}{n_7} \left(1 - \frac{n_3}{n_4} \frac{n_5}{n_6} \right)}$$

Using the values given earlier,

$$\begin{aligned} \frac{N_s}{N_{\text{IN}}} &= -9.2 \\ T_{\text{IN}} &= -9.2 rF \end{aligned}$$

If slip does not occur, then

$$\begin{aligned} T_{\text{IN}} &= \frac{N_{\text{OUT}}}{N_{\text{IN}}} T_{\text{OUT}} \\ &= -0.508 T_{\text{OUT}} \end{aligned}$$

With these relationships and the coefficient of friction, the required contact force between wheel and disk can be estimated from a knowledge of the torque required to operate the counter unit.

Clutch System

The clutch (or cam) system consists of three concentric disks and rings (Figure 9.11). The central ring is driven by the input shaft and carries a series of clutches that lock on the center disk and the outer ring, causing them to rotate at least as fast as the central ring. An arm rotates on a separate shaft, which is not concentric with that of the disk and rings. In this arm are two sliders, one fixed to, and rotating with, the center disk and the other fixed to, and rotating with, the outer ring. Because of the eccentricity of the arm shaft, the center disk and the outer ring are forced to rotate at different speeds. The difference in speeds can be adjusted by manually altering the amount of eccentricity. Thus, the speed at which the output shaft revolves relative to the input shaft is adjusted to achieve compensation for use of the measuring element under a particular set of operating conditions. It can be shown that if Φ is the angular position of the arm and θ_c and θ_o are, respectively, the angular positions of the center disk and outer ring [Figure 9.A.2(a)]

$$\Phi = \theta_c - \tan^{-1} \left(\frac{\Delta \sin \theta_c}{R_c + \Delta \cos \theta_c} \right) \quad (9.A.26)$$

and

$$\Phi = \theta_o + \tan^{-1} \left(\frac{\Delta \sin \theta_o}{R_o + \Delta \cos \theta_o} \right) - \pi \quad (9.A.27)$$

where Δ is the distance between the centers of rotation of the arm and the disks.

These are plotted in Figure 9.A.2(b), and it can be seen that at $\Phi = 0, \pi$, and 2π , the angular positions differ by π . The angular velocities will be equal when, for a given value of Φ , the slopes of θ_c and θ_o are equal.

In Figure 9.A.2(c), the relationship between output rotational speed N_o and input rotational position θ_i (which for constant flow rate is proportional to time) is plotted. It is important to note the nonuniform rotation of the output and the mean value in relation to actual rotational speed at various points in the cycle.

9.A.6 APPLICATION OF EQUATIONS TO A TYPICAL METER

Using the meter with dimensions shown in Figure 9.13 and Equation (9.A.9), we obtain for the main coefficients, assuming a total leakage length of 325 mm,

$$\begin{aligned} q_v &= 0.00918N \\ &\quad -0.0000156N \quad (\text{leakage}) \\ &\quad +0.034 \times 10^{-10} \frac{\Delta p}{\mu} \quad (\text{leakage due to pressure change}) \end{aligned}$$

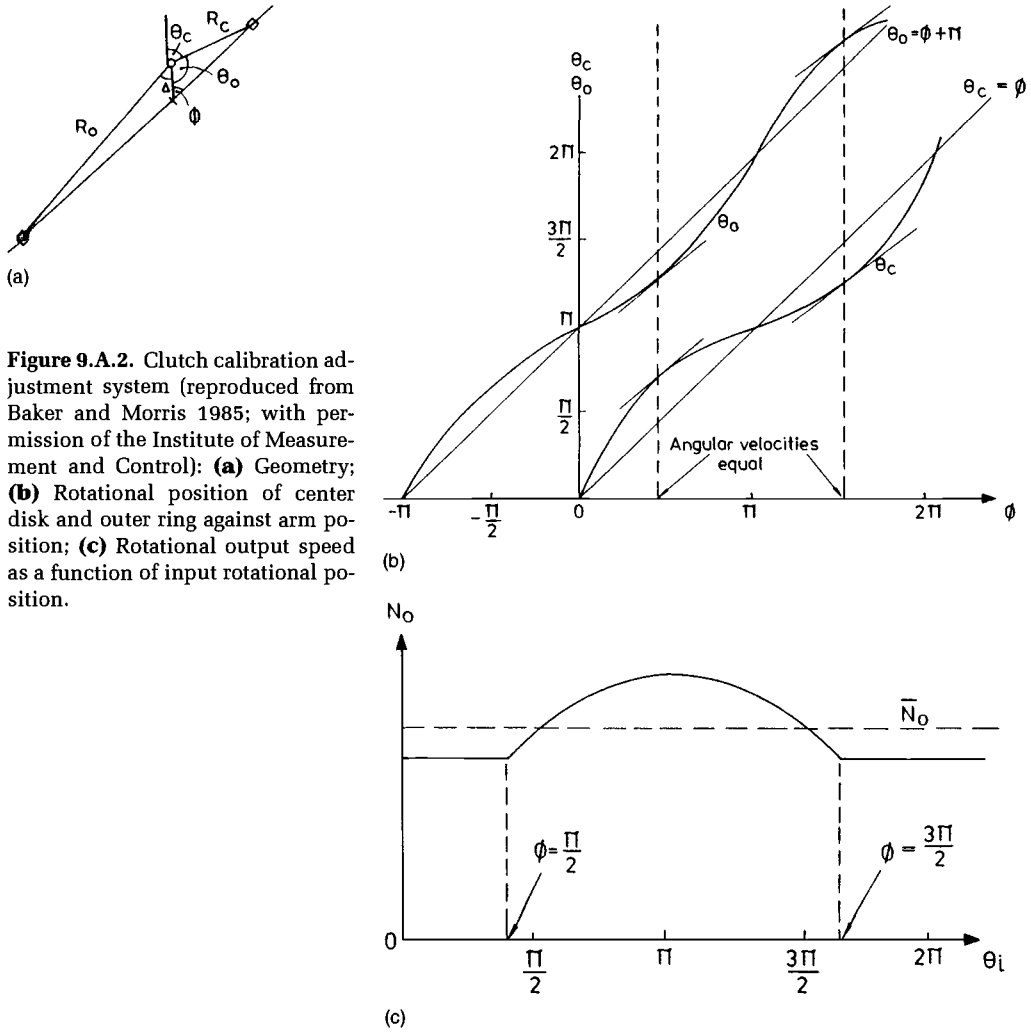


Figure 9.A.2. Clutch calibration adjustment system (reproduced from Baker and Morris 1985; with permission of the Institute of Measurement and Control): **(a)** Geometry; **(b)** Rotational position of center disk and outer ring against arm position; **(c)** Rotational output speed as a function of input rotational position.

Taking

$$\Delta p = 20000 \text{ N/m}^2$$

$$\mu = 0.1 \text{ Poise}$$

$$q_v = 0.00916N + 0.677 \cdot 10^{-6} \text{ m}^3/\text{s}$$

The sensitivity to t and Δp is given by

$$q_v = 0.00918N - 0.156tN + 3.4 \frac{\Delta p t^3}{\mu} \text{ m}^3/\text{s}$$

Taking the datum values of $t = 0.1 \text{ mm}$, $\Delta p = 0.2 \times 10^5 \text{ N/m}^2$ and $\mu = 0.1 \text{ poise}$, the effect of changes of $\pm 0.5 \Delta p / \mu$ for $N = 1$ revolutions per second and $t = 0.2 \text{ mm}$ is about $\pm 0.05\%$ (Baker and Morris 1995).

Turbine and Related Flowmeters

10.1 INTRODUCTION

10.1.1 BACKGROUND

Spirals, screws, and windmills have a long history of use for speed measurement. Robert Hook proposed a small windmill in 1681 for measuring air velocity and later one for use as a ship's log (distance meter). A Captain Phipps, in 1773, created a ship's log, using the principle that a spiral, in turning, moves through the length of one turn of the spiral. Many centuries earlier, it appears that a Roman architect, Vitruvius, suggested a more basic form of this device.

In 1870 Reinhard Woltmann developed a multibladed fan to measure river flows (Medlock 1986). The device was a forerunner of the long helical screw-type meter still called after him and used widely for pipe flows in the water industry. The first modern meters, of the type with which we are mainly concerned, were developed in the United States in 1938 (Watson and Furness 1977; cf. Furness 1982). These were attractive for fuel flow measurement in airborne applications. They consisted of a helically bladed rotor and simple bearings. Improved sleeve bearings were developed for longer life with hardened thrust balls or endstones to withstand the axial load. An alternative, developed over several years and patented by Potter (1961), was to profile the hub of the rotor so that the pressure balance across the rotor, rather than the thrust on the bearings, held it against the axial drag forces. This allowed the rotor to run on a single journal bearing.

10.1.2 QUALITATIVE DESCRIPTION OF OPERATION

The turbine consists of a bladed rotor that turns due to the flow in the pipe. In most of the designs to be discussed, the rotor is designed to create the minimum disturbance as the oncoming flow passes around it. Ideally, it cuts perfectly through the fluid in a helix so that every revolution of the helix represents one complete axial length of the screw, and hence a calculable volume of the fluid. In practice, there are drag forces, which slightly retard the rotation. These result from frictional drag on the blades, the hub, the faces of the rotor, and the tip of the blades; bearing drag, also a frictional effect; and drag due to the means by which the rotation is measured, which is usually a magnetic drag. These drag forces affect the otherwise ideal relationship of constant fluid volume for each rotor revolution and lead to nonlinearity.

The rotor has to be held in the stream, and so supports are invariably necessary to position the bearings centrally in the pipe. Virtue is made of necessity, and these supports are used to provide some flow straightening, since the turbine meter is of a type particularly susceptible to swirl. The oncoming fluid will therefore need to flow over the upstream support, and the naturally occurring profile, where the velocity of the fluid is lower at the pipe wall than at the center of the pipe, will be redistributed into the annular passage past the blades, and, perhaps most importantly, there will be a reduction in any swirl component due both to conservation of angular momentum during the redistribution and to the straightening effect of the supports. The velocity in this passage will vary a small amount with radial position, and a well-designed turbine rotor will generally have helical blades to match the axial and tangential velocity of the rotor at each radial position.

The main variants from this description are the Pelton and vane or paddle wheel designs. In the former, a tangential jet of fluid hits the buckets or blades of the wheel and causes them to rotate. This design has two alternative virtues: it can be made as a simple insertion probe to obtain an estimate of the flow rate, or it can be used in very small flows to register where other designs fail. Other designs using larger paddle wheels and angled propellers are also available (cf. Baker 1991b, 1993).

10.1.3 BASIC THEORY

Many turbine meters use flat section blades. Ideally these will cut smoothly through the flow in a perfect helix. Using this basic concept, the value of V_z can be obtained from the frequency of blade passing f (Figure 10.1). Thus,

$$V_z = W / \tan \beta \quad (10.1)$$

and

$$f = N \tan \beta V_z / 2\pi r \quad (10.2)$$

where V_z is the axial velocity of the flow approaching the blades, W is the blade velocity, β is the blade angle, N is the number of blades, and r is the radius of the blades at the measurement point.

Equation (10.2) can be used to obtain

$$f = K q_v \quad (10.3)$$

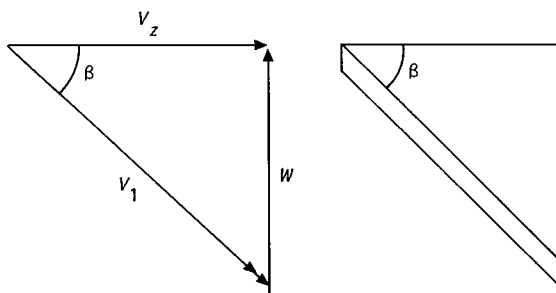


Figure 10.1. Relationship between velocities and angles for ideal flow over blades.

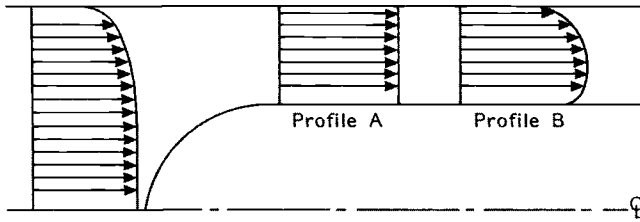


Figure 10.2. Velocity profiles at inlet to, and in the annulus of, a turbine meter.

which provides the basis for the constant, which in various units and dimensions is known as meter registration, meter constant, meter coefficient, calibration factor, etc., and is measured in pulses per unit volume or radians per unit volume. We shall call this the K factor, in (pulses per cubic meter), as in Chapter 1.

Tan (1973; cf. Fakouhi 1977) gives an equation of the form

$$\frac{f}{q_v} = a_0 + \frac{a_1}{q_v} - \frac{a_2}{q_v^2} \quad (10.4)$$

where the second term is affected by the viscosity and leakage, and the third term is affected by the mechanical and aerodynamic drag. Adjusting these two terms has been found to affect the familiar hump in the turbine characteristic (cf. Newcombe et al. 1972 who gave an equivalent equation for gas meters).

However the theory is not, unfortunately, as simple as this because the blades do not cut the fluid perfectly, and the value of r and β must change to accommodate the profile across the pipe.

Suppose that we consider the variation of blade angle for a uniform profile A (Figure 10.2), then it can be seen that, with no variation in axial velocity across the annulus, the correct blade angles will be given by

$$\tan \beta / r = \text{constant} \quad (10.5)$$

If the midspan blade angle is 41° as in Xu's papers (1992a, 1992b) then the values at hub and tip will be 30° and 49° as in Xu's meter, suggesting that this was the method applied in the design which he used. If, however, profile B existed, then the flow will not meet the blades at the correct angle, and the designer may adjust the blade twist to allow for this. The reader is also referred to Hochreiter's (1958) paper where he allows for blade obstruction.

It thus becomes apparent why the flowmeter will be susceptible to incorrect installation, since this will cause a flow profile that results in the wrong incidence angle for some parts of the annulus. It also suggests why a turbine wheel, which is optimized for flow profile range, will give a better performance than one with, say, constant angle blades. To develop a fuller theory, it is necessary to take account of the lift and drag on the blades, bearings, etc. Figure 10.3 sets out the main angles and forces for a cascade of blades. The analysis is given in Appendix 10.A.1.

Tsukamoto and Hutton (1985) predicted the complete characteristic of a turbine meter with reasonable accuracy allowing for viscosity, inlet flow profile, and meter geometry. Blows (1981) developed a model of a turbine flowmeter, which allowed the effects of viscosity and tip clearance to be examined.

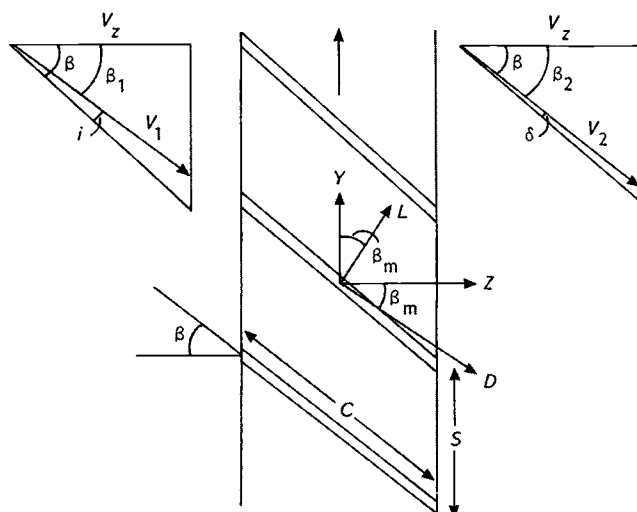


Figure 10.3. Cascade angles, dimensions, and forces.

The equation for the driving torque [Equation (10.A.1)], given by Tsukamoto and Hutton (1985) (Figure 10.4) and equivalent to the equation used by Blows (1981), is equated to the drag torque. Tsukamoto and Hutton (1985) gave four components of the drag torque:

- T_b , the bearing drag;
- T_w , the hub disk friction drag;
- T_t , the blade tip clearance drag; and
- T_h , the hub fluid drag.

Blows (1981) discussed pickup torque and thrust pad drag torque but dismissed the former as difficult to calculate and likely to be significant only in small meters. He suggested that the thrust pad force might vary, not only in sign, but by a factor of 100; hence, because effects of this significance are not observed, it could also

be ignored. The consensus appears to be that the dependence of all drag forces on speed, apart from aerodynamic drag on the blades and the hub fluid drag, is a power less than two and therefore of decreasing significance with increasing speed.

Tsukamoto and Hutton's (1985) results were impressive, giving a characteristic showing the hump and the effect of the blade boundary layer transition at different blade positions at different radii, and the linear range when the boundary layer is turbulent (Figure 10.5). This hump has been attributed to skin friction and to unsteady wake flow caused by straighteners. Tsukamoto and Hutton (1985) also

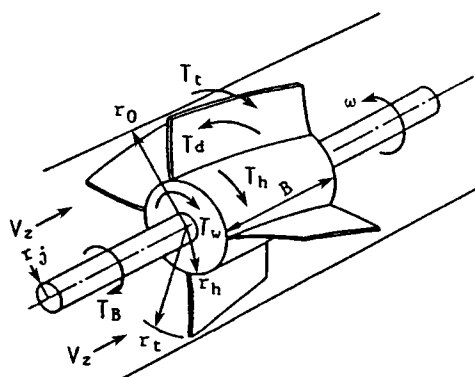


Figure 10.4. General view of a turbine rotor and torques on the rotor (based on Tsukamoto and Hutton 1985; with the authors' permission).

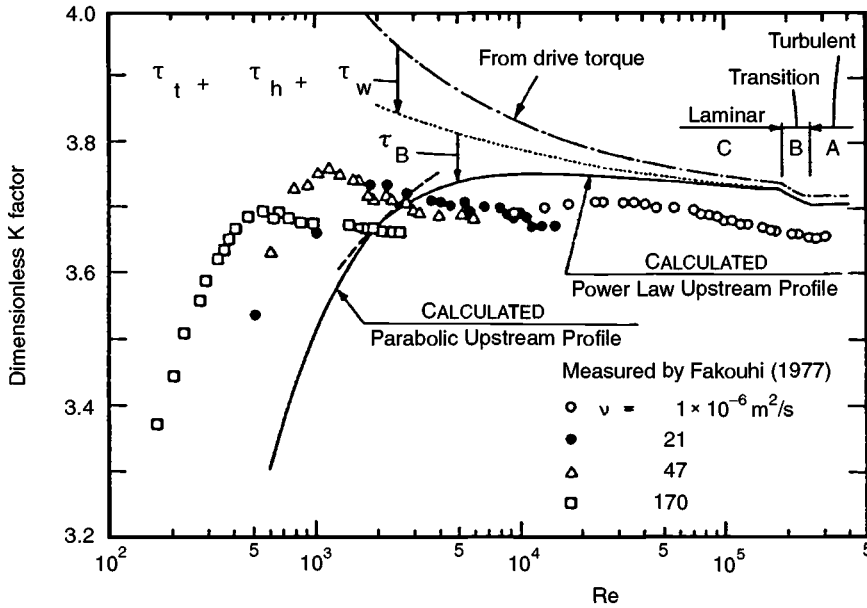


Figure 10.5. Comparison between calculated and measured meter performance. Relevant dimensions: $r_0 = 18.288$ mm, $r_t = 18.008$ mm, $r_h = 9.703$ mm, $B = 14.224$ mm, $t_b = 1.245$ mm, $L = 95.280$ mm, $N = 6$, $r_j = 2.477$ mm, $\circ$, $\nu = 1 \times 10^{-6}$ m²/s, $\bullet$, $\nu = 21 \times 10^{-6}$ m²/s, $\triangle$, $\nu = 47 \times 10^{-6}$ m²/s, $\square$, $\nu = 170 \times 10^{-6}$ m²/s (based on Tsukamoto and Hutton 1985; with the authors' permission).

obtained good qualitative agreement for the effect of change from uniform to fully developed turbulent inlet profile and for tip clearance between Tan (1973), Jepson and Bean (1969), and their own theory.

Thompson and Grey (1970) derived expressions for rotor hub fluid drag and blade tip clearance drag and introduced the bearing drag and the pickup drag. Using calculated annulus flow profiles, they demonstrated that, even with helical blades, the profile results in a variation of angle of attack for a 50-mm meter of $+7^\circ$ to -8° and because the actual incidence angle will be greater than the angle of attack, it is possible that blade stall may be occurring. The effect of this on the driving torque is best shown in a plot of torque against radius. It is also useful to note that change of fluid for the same velocity results in a change in Reynolds number and hence in profile, although the effect on angle of attack is comparatively small. The annulus profiles and the blade driving torque are shown in Figures 10.6 and 10.7.

Jepson and Bean (1969) in their earlier work essentially reproduced Equation (10.A.1) with a slightly different constant. In their analysis, they made three simplifying assumptions.

- They assumed that the exit angle was the blade angle and for helical blades would be given by [cf. Equation (10.5)]

$$\beta = \tan^{-1} \left(\frac{2\pi r}{K_h} \right) \quad (10.6)$$

where K_h is a constant.

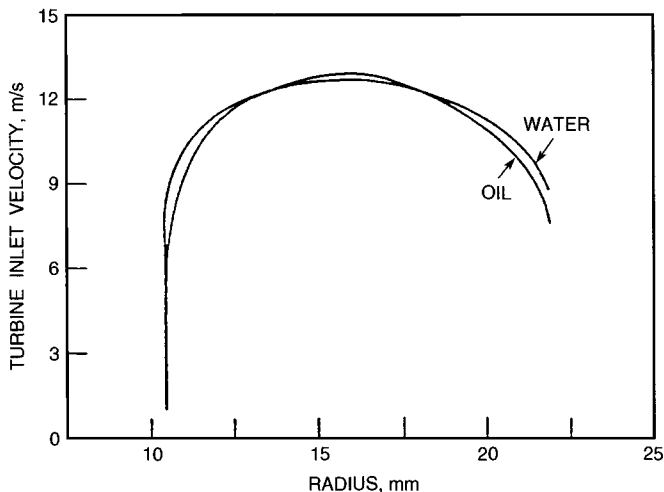


Figure 10.6. Turbine inlet velocity profiles for water and oil (Thompson and Grey 1970; reproduced with permission from ASME).

- b. They assumed that the retarding torque caused by bearing friction was negligible compared with the aerodynamic forces, and they justified this on the grounds that it is constant. They pointed out that the assumption was only valid for commercial meters with low resisting torques.
- c. They did not consider retarding torques apart from bearing friction and blade aerodynamic drag.

They used this simple approach to examine the effect of flow profile and, knowing the profile downstream of a cone exit, to calculate meter error with distance from the cone exit.

Hutton has made the point that a basic difficulty in applying blade theory to turbine meters is that their blades are not very like flat plates because the parallelogram

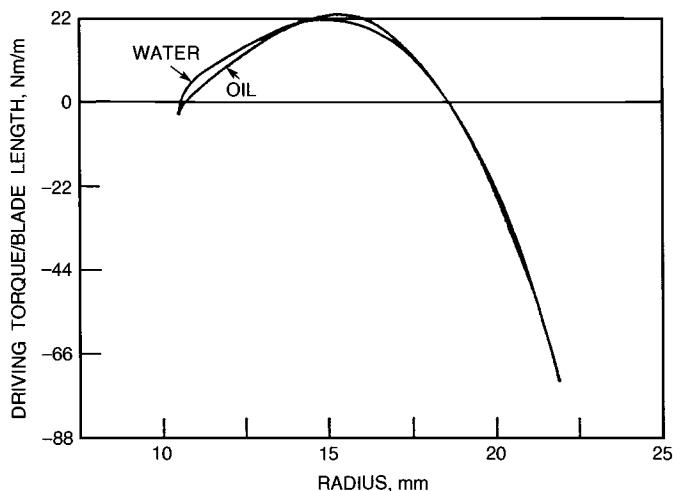


Figure 10.7. Driving torque per unit blade length for water and oil (Thompson and Grey 1970; reproduced with permission from ASME).

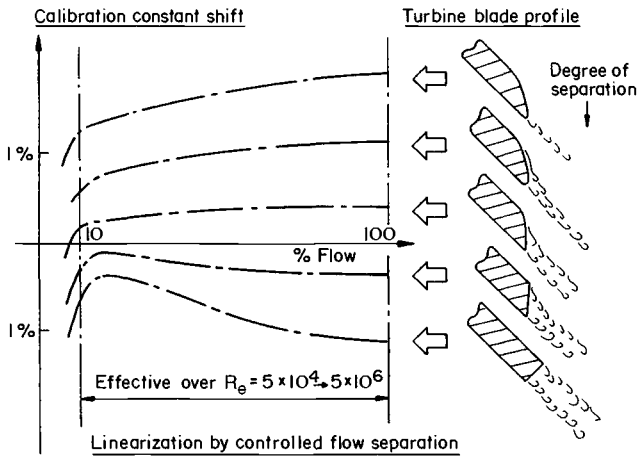


Figure 10.8. Effect of trailing edge on calibration (after Griffiths and Silverwood 1986; reproduced with permission of Flow Automation (UK) Ltd.).

cross-section caused by the chamfer on leading and trailing edges results in an S-shaped camber line. The effect of the trailing edge is confirmed by Griffiths and Silverwood (1986) in Figure 10.8.

Xu (1992a) presented a very interesting study of the flow around a typical turbine meter blade and calculated the flow at leading and trailing edges. The flow at the leading edge, as might have been expected, separates on one or both sides depending on the incidence angle. The vortex shedding at the trailing edge does not appear to be greatly influenced by incidence angle. Of particular interest is the variation of lift and drag with incidence angle. In Figure 10.9, Xu's values are compared with those given by Wallis (1961) and Jepson (1964). Xu's results are for $Re = 29,500$ based on the half thickness of the blade and for blade stagger angles of tip 49° , midspan 41° , hub 30° .

Xu (1992b) applied his results to the performance of a 100-mm turbine meter with and without swirl. He measured the inlet velocity profile, which he then appeared to apply to a theoretical calculation of the meter characteristic starting essentially from Equation (10.A.8) (apart from a sign difference). The agreement he showed between prediction and measurement was impressive.

Tan and Hutton (1971) found that a full diameter rotor had a rising characteristic, whereas a turned down rotor had a falling characteristic; they concluded that the tip clearance could account for this.

Hochreiter (1958) used nondimensional groups to correlate experimental data but found that, for a Reynolds number based on rotational speed below 6,000–10,000, scatter may have resulted from poor lubricational properties of gasoline.

10.2 PRECISION LIQUID METERS

10.2.1 PRINCIPAL DESIGN COMPONENTS

Figure 10.10 gives an example of a commercial design that introduces most of the essential features of these meters. The first essential component is the rotor itself,

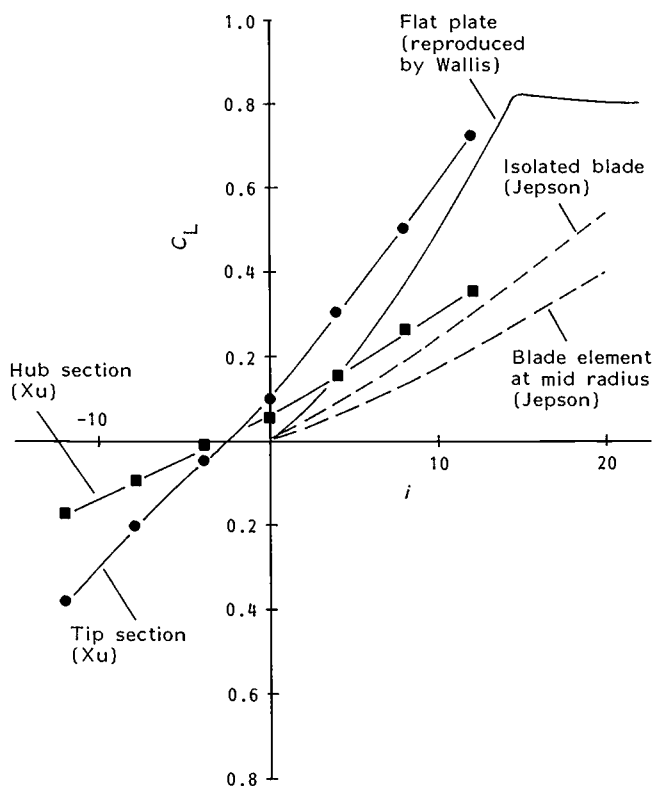


Figure 10.9. Comparison of lift coefficient as given by various authors (Wallis 1961, Jepson 1967, Xu 1992a).

which is like a small turbine. This is formed either from straight or twisted (helical) blades. For good design, it is suggested that the hub radius be approximately half the tip radius; the blades, when viewed axially, just block the cross-section of the pipe; and the axial length of the blades be approximately equal to the blade span, which is the length of the blade from hub to tip.

It is important to remember that unlike the power turbine, the meter aims to cut through the fluid without disturbing the flow. This clearly is an impossible ideal; nevertheless, the power extracted from the fluid is very small, and the incidence of the flow onto the blades is consequently also very small. The reason for the popularity of helically twisted blades is that the relative angle of approach of the fluid onto the blades varies with radius. Consequently, flat plate blades will not allow a constant angle of attack and will lead to unnecessarily large incidence angles, both positive and negative, and thus will impart unnecessary flow disturbance and drag.

The rotor is usually machined from solid, and the flat plate blades often have leading and trailing edges that are milled in a constant end plane. The cross-section of the blade at a constant radius will therefore be essentially a parallelogram. Sharp edges should be avoided because they may cause cavitation. Griffiths and Silverwood (1986) describe how small changes in the shape of the trailing edges can cause a change of the blades' aerodynamic drag and consequently of the meter calibration (Figure 10.8).

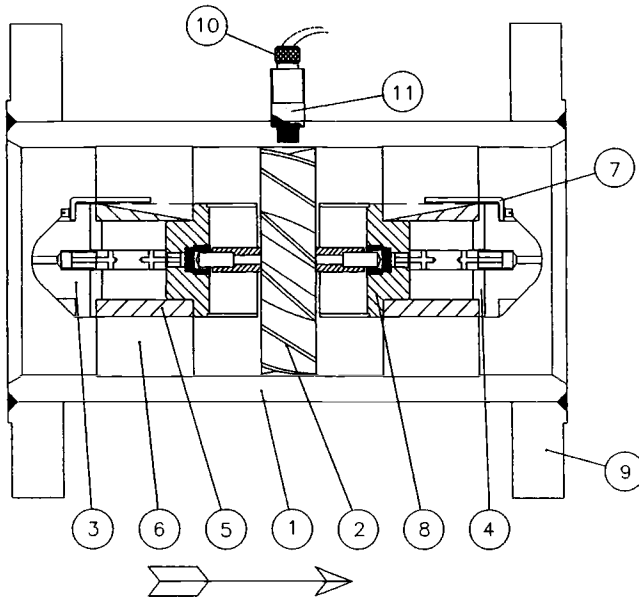


Figure 10.10. Precision commercial turbine flowmeter design for liquids for 100–500 mm diameter (1, body; 2, rotor; 3, support nut; 4, support plate; 5, support tube; 6, support vane; 7, lock tab; 8, end support assembly; 9, flange; 10, pickup assembly; 11, pickup stub) [reproduced with permission of Flow Automation (UK) Ltd.].

The hangers or support vanes that position the bearings centrally in the pipe are commonly used as flow straighteners to reduce swirl. Their shape varies considerably. In Figure 10.10, they are flat plates, and in another design, they are parallel tubes. The rotation of the rotor is sensed by a pickup, which most commonly senses a change in the magnetic material permeability in its vicinity as a blade passes (magnetic reluctance change). Alternatives to this are to use the eddy current effect or dynamo effect as a blade passes through the magnetic field of the sensor (magnetic inductive) or to use a modulated high frequency signal (modulated carrier or radiofrequency). The capacitance effect can also be used to sense the proximity of a passing blade to the sensor head. Optical fibers have been tried, but if the optical path passes through the fluid, there is a danger of the windows becoming fouled. For increased integrity, some meters have two sensors spaced at about 120° , the pulse trains from both are sensed, and the phase angle is constantly checked.

In small meters, the internals are held in position by circlips, the removal of which allows the internals to be withdrawn. In other designs and larger sizes, the bearing assemblies may be removed from the body by releasing a taper locking nut.

10.2.2 BEARING DESIGN MATERIALS

Minkin et al. (1966) attributed calibration unpredictability to bearing variation as a major factor.

Clean Liquids

Open ball bearings of stainless steel or other compatible material are suitable for liquids with lubricating properties in the temperature range -50 to 250°C , but a

filter with a mesh size small enough to prevent solid particle ingress into the bearings should be used.

Liquids in General

Tungsten carbide or high chrome/high cobalt journal bearings possibly with carbide pinions and a stellite sleeve are suitable in the range -50 to 400°C unlubricated. The bearings are lubricated by the metered fluid and made up of tungsten carbide with a hardness of Rockwell C-94 and a surface smoothness of $0.05\text{ }\mu\text{m}$. Tungsten carbide with cobalt bonding is most common, but the cobalt is leached out by acidic solutions used for cleaning. Stellite may then be used as a less durable alternative for more corrosive fluids. A more recent alternative is nickel-bonded carbide bushes with titanium carbide shafts. An alternative design uses rotor pins of hard metal in sapphire bearings with thrust plate and rings made with curved contact surfaces for line contact. Polytetrafluoroethylene (PTFE) is used in some applications. Ceramic bearings (e.g., Al_2O_3), may also be an option.

One manufacturer has used a ball-and-sleeve design. The shaft was tipped with an ellipsoid and ran in a ceramic sleeve so that line contact was achieved. Typical bearing life was claimed as 12–24 months when used with 50–75% duty cycle in clean particulate-free gasoline. For ball race bearings correctly lubricated, a life of 4,000–6,000 hours was claimed, and for journal bearings 20,000 hours or more was claimed, depending upon the properties of the measured fluid and rate of flow.

For some larger sizes (above about 80 mm) only an upstream bearing is used; it consists of a tungsten carbide (or other suitable materials) bush and shaft. Special bearings are available for cryogenic applications (cf. Rivetti et al. 1987). Hydrostatic bearings may be suitable for dirty fluids. House and Johnson (1986) applied hydrostatic bearings and obtained a remarkable performance (turndown ratio for water of 1,000:1) and suitability for dirty fluids. Wemyss and Wemyss (1975) described the development of the Hoverflo bearingless meter. Some manufacturers provide application charts.

Ball bearings have been found satisfactory for cryogenic applications in rocket testing because of their ability to tolerate overspeed and operation with gas. However, journal bearings, although less linear, may be more repeatable.

10.2.3 STRAINERS

Typical values for strainers are given by manufacturers with maximum allowable particulate sizes (Table 10.1).

10.2.4 MATERIALS

Because the signal is usually obtained from magnetic sensing, it is common to make the tube of austenitic stainless steel to allow the magnetic field to penetrate. However, carbon steel is used for flanges and sometimes also for the bodies; in this instance, provision is made for the pickup assembly to function despite the magnetic body.

Table 10.1. Typical values for strainers from manufacturers' literature with maximum allowable particulate sizes

Meter Diameter (mm)	Particulate Sizes (μm)		Mesh	Hole Size (mm)
	Ball Bearings	Journal Bearings		
13	60	100	150	0.100
20	90	142	100	0.150
25–80	100	185	80	0.175
100–150	142	251	60	0.200
200–300	251	401	40	0.300

10.2.5 SIZE RANGES

Meter sizes are typically:

for flanged, 6–500 mm,
for threaded, 6–50 mm.

In some designs, the bore reduces so that at the propeller the bore radius is a minimum. Flow ranges are typically 10:1 turndown (but may be up to 30:1) for flow rates of 0.03–7,000 m³/h. Overspeed up to 1.5 times the maximum flow rate may be permitted for short periods, but if it takes place for prolonged periods, the bearing life is reduced.

Pressure losses at maximum flow rate, with water as the flowing medium, range up to 0.2 bar (2×10^4 N/m²) in 12-mm sizes and up to 0.25 bar (2.5×10^4 N/m²) in 200-mm sizes.

Maximum pressures range up to 25 bar for hygienic designs, up to 240–400 bar for threaded designs, and according to the flange rating for flanged designs.

Temperature ranges are typically –50 to 150°C but may be as wide as –265 to +310°C.

Some manufacturers give plots of the effect of change of viscosity on the performance of their meters (Baker 1991b). The bearing, as well as the blade design, will have an influence on this. Acceptable viscosity limits for smaller sizes (less than 80 mm) may be up to 15 mPas (15 cP) and for sizes above 100 mm up to 50 mPas (50 cP). These viscosity limits prevail as a result of increasing nonlinearity with reducing Reynolds number.

10.2.6 OTHER MECHANICAL DESIGN FEATURES

Internals may be made of stainless steel, but other materials such as Teflon and polythene are also used.

In order to increase the pulse rate from impellers with typically six to eight blades, a shroud ring with holes or high permeability soft iron alloy studs is used. This increases the number of pulses due to the holes changing the magnetic reluctance and also allows viscosity compensation to improve low speed performance. The shroud can also give greater strength and reduced blade vibration.

In some small designs of turbine meter, the blades have a T shape so that at the tip they are longer (longer chord) than at the hub. One reason given for this is to improve insensitivity to viscosity change, which may result from a greater driving torque achieved by increasing the blade lift at the maximum radius. It may also flatten the characteristic hump at low speed. I am not aware of published data to confirm these tendencies.

Some designs can be used bidirectionally, but, of course, they require calibration in each direction.

Response time constants measured with water or with a liquid of similar density range from 0.005 to 0.05 s for a 50% flow rate change or up to 0.17 s to reach 63% of a step change final value.

After use with corrosive fluids, the meter should be cleaned with solvents. However, maximum rotational speeds should be carefully observed.

10.2.7 CAVITATION

Cavitation is caused when the local pressure drops below the vapor pressure of the liquid and results in a two-phase flow of increased volume and, therefore, causes overreading. Figure 10.11 demonstrates this effect where the vapor pressure of the hydrocarbon is less than 0.007 bar. It may also cause damage to the internals. A simple rule is that the pressure downstream of the meter should be at least 1 bar above atmospheric pressure. More generally, the rule given is that the back pressure should be at least twice the pressure drop across the meter plus the absolute vapor pressure times 1.25. This recognizes that, in addition to the pressure loss through the meter at rated flow of between 0.3 and 0.7 bar and the pressure needed to accelerate the flow past the turbine wheel of 0.3–1.0 bar, there is also the minimum pressure level to prevent cavitation.

Bucknell (1963), from his work on liquid oxygen and hydrogen, suggested that a minimum safe margin for most meters was a back pressure of four times the meter permanent pressure loss above the vapor pressure.

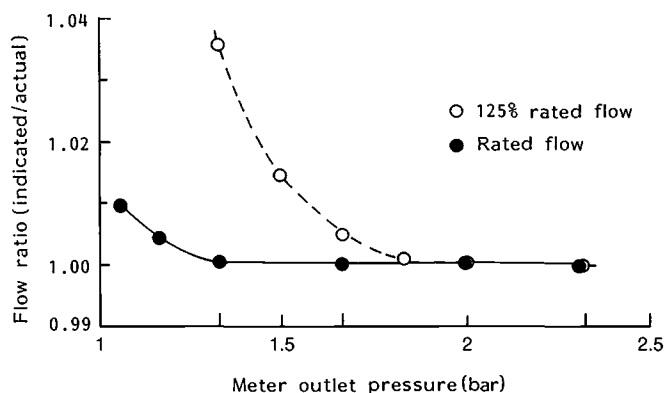


Figure 10.11. A typical example of cavitation effects on calibration factor (Shafer 1962; reproduced with permission from ASME).

10.2.8 SENSOR DESIGN AND PERFORMANCE

A direct mechanical drive to register may be used in large meters. However most commonly, to obtain rotor speed, blade passing is sensed by the change in the magnetic field around the sensor. The terms given to the various methods suffer from some variation, and the following are suggested for consistency (Olsen 1974):

- a. *Inductive*: Magnets are embedded in the hub or blades, and a pickup coil with a soft iron pole piece senses their passage;
- b. *Variable reluctance*: A permanent magnet with pickup coil is positioned in the body of the flowmeter near the propeller/rotor, which senses the variation of the flux due to the passage of each blade or shroud ring stud of highly permeable magnetic rotor material;
- c. *Radiofrequency (RF)*: An oscillator applies a high frequency carrier signal to the coil in the pickup assembly, and the passing of the rotor blades modulates the carrier. At very high frequencies of signal transmission, the skin effect is such that the electromagnetic field is essentially reflected by the passing blades with negligible drag effect on the wheel.
- d. *Photoelectric*: A light beam is interrupted by the passage of the blades. Optical fiber methods, which are intrinsically safe, may also be used where the light reflects off the blade tips. The problem with this method is that the windows tend to become fouled, and so light transmission through the liquid is not to be recommended in many applications. Place and Maurer (1986) proposed an optical system that overcame this problem where the blade passage rocked a magnetic element and the rocking was sensed optically;
- e. *Magnetic reed switch*: Contacts are opened and closed by magnets in the rotor or some rotating part of the meter.

The resulting signal must be amplified with care and shielded from extraneous noise due to voltage sources or magnetic fields because spurious pulses can introduce significant errors.

Vibration may also cause microphonic effects. Screened cable should be used, and the distance from the flowmeter to the preamplifier should be as short as possible and typically not greater than 2 m.

Typically the pulse is amplified from about 15 mV root-mean-square (RMS) to 8 V amplitude (increases with flow rate) and is of 0.5- to 20-ms duration. The pulse rate ranges from 50 Hz to 3 kHz, and impedance from 300 to 1,500 ohms. A maximum transmission cable length has been given as 1,000 m by one manufacturer. The signal is then applied to a signal converter, an electronic gearbox, to convert pulse rate into cubic meters per hour and total into cubic meters. A high pulse density is desirable where a meter is to be calibrated against a volumetric standard such as a prover loop where generally it is recommended that 10,000 pulses or more should be collected per unit reference volume. Eide (1991) described a 12-in. Brooks Compact Prover and a 3-in. Brooks turbine meter modified to give 27,000 pulses/m³, which were used to calibrate pipe provers. The uncertainty of the water draw was better than $\pm 0.003\%$, and the master meter's uncertainty was 0.02%.

Ball (1977) demonstrated that below about 1/3 FSD, the drag due to a magnetic pickup could cause differences compared with an RF pickup and at 1/6 FSD could indicate an error of 4% rising further with decreasing speed.

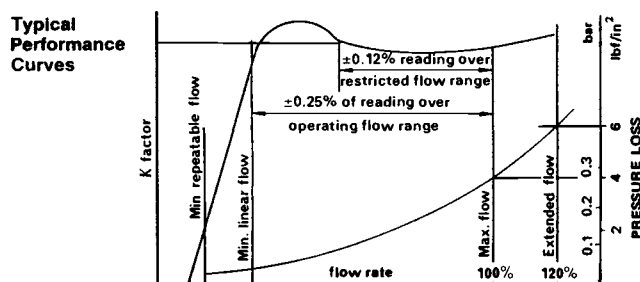


Figure 10.12. Characteristic for a precision liquid meter after KDG Instruments (reproduced with permission).

10.2.9 CHARACTERISTICS

The typical characteristic of the precision turbine meter is shown in Figure 10.12. There are four regions to note:

- i. the highest accuracy region;
- ii. an extended range that should be used only briefly and with care to avoid excess bearing wear;
- iii. the normal range of operation that includes both the high accuracy region and also a lower flow region which is often shown as having a “hump”; and
- iv. a low flow region where drag forces are dominant and where the performance of the meter is less reliable.

Most of these regions and their causes are obvious. The one that is often puzzling is the hump region in (iii). Tsukamoto and Hutton (1985) have confirmed theoretically that a hump in the characteristic can be caused by movement of the point of transition of the blade boundary layer near the leading edge of the blades from turbulent in the high accuracy region to laminar. Griffiths and Silverwood (1986), as indicated earlier, have suggested that changing the profile of the blade trailing edge by filing off the corner leads to a removal of the hump and an increase of speed of rotation of over 1% due, presumably, to changes in the point of separation and reduction of drag (Figure 10.8).

Pressure loss arises largely from the pressure reduction at the rotor and bearing supports and the failure to recover fully the total pressure downstream for an area ratio $m = A_1/A_2$, which will give a maximum pressure loss of

$$\Delta p_{\text{loss}} = \frac{1}{2} \rho V_1^2 K$$

For pressure change due to area change only (neglecting friction), the loss factor $K = (m^2 - 1)$, and for values of $m = \frac{4}{3}$, $\rho = 10^3 \text{ kg/m}^3$ and $V_1 = 10 \text{ m/s}$ this gives a pressure change of 0.39 bar ($3.9 \times 10^4 \text{ N/m}^2$). Experimental values of K range from 0.4 to 0.9 depending on the geometric details of the meter. For $K = 0.5$, $\rho = 10^3 \text{ kg/m}^3$ and $V_1 = 10 \text{ m/s}$, the pressure loss is 0.25 bar.

10.2.10 ACCURACY

Typical values of measurement uncertainty given by manufacturers are ± 0.12 – 0.25% for the high accuracy range and ± 0.25 – 0.5% for the lower part of the normal range.

This is sometimes given as a linearity. The meters are capable of calibration to achieve these values and possibly slightly better than ± 0.1 to 0.5% over a turndown ratio of 6:1 at least and up to 18:1 for thrust-compensated designs. With achievable uncertainty of $\pm 0.15\%$, they can meet the requirements of oil pipeline and custody transfer applications.

Repeatability is typically given as $\pm 0.05\%$ for meters of less than or equal to 50 mm and $\pm 0.02\%$ for meters of greater than or equal to 75 mm. Such precision values do not take into account bias errors often present in calibration equipment. Mattingly et al. (1977) obtained day-to-day laboratory repeatability of ± 0.07 – 0.12% . Shafer (1962) made the important point that before calibration the meter needs to be soaked in the calibration fluid and run for some time beforehand. He obtained impressive performances of 0.1–0.2% agreement as typical of year-to-year repeatability.

These meters are widely used as transfer standards, and for best performance the meter is calibrated with upstream and downstream pipework and an upstream straightener. Twin meter packages of turbine and other meter types have been used for transfer standards, but Minkin et al. (1966) used a twin turbine meter arrangement to give extra confidence.

Withers et al. (1971) tested 5×75 mm meters on water, kerosene, gas oil, and spindle oil and obtained repeatability to better than $\pm 0.05\%$ (95% confidence) on all. However the characteristics were very varied when used with different liquids and temperatures. The temperature effect was shown to be due more to viscosity than to expansion. Despite the apparent variation, the authors claimed to be able to determine a universal curve for each meter in terms of Reynolds number.

Minkin et al. (1966) reported extensive tests on two groups of meters ranging from about 20 to 50 mm diameter in which they showed the following results.

- a. The calibration factor for full scale had a maximum deviation from the mean over 2 years of 0.6%.
- b. The calibration factor for water and liquid hydrogen for a good meter (horizontal and vertical orientation) varied by less than 1% over 3 years apart from one occasion with 2% change.
- c. The orientation of meters affected lower range values more than full scale.
- d. The effect of thermal expansion resulted in a calibration factor ratio for $C_{\text{water}}/C_{\text{liquid hydrogen}}$ of 0.991–0.994 theoretically with wider experimental variation (cf. Shamp 1971 who reckoned registration for liquid nitrogen at -190°C was 1.5–1.9% less than for water at about 15°C).

It is also necessary to calibrate in the same orientation as installation.

10.2.11 INSTALLATION

Installation is one of the major causes of meter error. The ideal long straight length of upstream pipe required to achieve a fully developed (usually turbulent) profile is seldom possible even in calibration facilities. It may be worth considering the

To achieve the maximum performance from a turbine flowmeter it is essential that straight pipe sections are fitted immediately upstream and downstream of the flowmeter position. For liquid flows the dimensions indicated below should be considered a minimum and for gas flow they should be multiplied by 4.

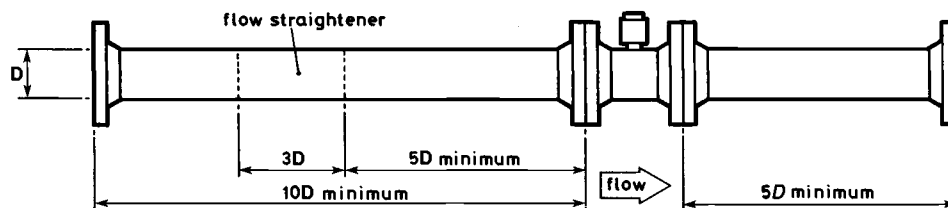


Figure 10.13. Installation for a precision liquid meter as suggested by Quadrina Ltd. (reproduced with permission).

calibration of the flowmeter together with surrounding pipework and straightener, to eliminate, as far as possible, the effect of upstream flow effects. Because of the susceptibility of turbine meters to asymmetry and swirl, generated by upstream fittings, it is common for manufacturers to provide recommendations for upstream pipe configurations, including the use of flow conditioners. In most cases these will, presumably, be to avoid additional error. These are sometimes accompanied by a diagram. A typical example is shown in Figure 10.13. This suggests that an upstream straight length of 10 diameters is required, with the downstream end of a flow straightener, 3D long, positioned at least 5D upstream of the meter. It is always slightly surprising that some other manufacturers, who provide a useful diagram like this one, suggest that without a flow straightener the user should still allow only, say, 10D of straight pipe upstream of the flowmeter.

ANSI/API have suggested from experience that an upstream length of 20D and a downstream length of 5D provided effective straightening in many installations. In addition, maximum allowable misalignment of the meter bore and the preceding pipework has been given as 5%. Values are given in Table 10.2 for upstream distances for particular fittings for a commercial instrument, which have also been given by API.

Wafer construction has been available, allowing installation between flanges. Presumably great care is needed in the positioning of this meter to obtain satisfactory results. Published data suggest that some of these figures for upstream

spacing may be overly optimistic and that the uncertain effects of an upstream valve indicate the need to avoid this configuration and, if unavoidable, to use a straightener.

Larger hub-to-tip ratios appear to be less sensitive to profile, and the larger the tip clearance the less the sensitivity. Salami (1984) confirmed that a commercial meter with a hub-to-tip ratio of about 0.5 and a tip clearance of 10% was almost insensitive to changes from uniform to turbulent profile (cf. Tan 1976 and Jepson and Bean 1969).

Table 10.2. An example of upstream spacings from manufacturer

Fitting	Upstream Distance
Reducer	15D
Swept elbow	20D
Two swept bends in the same plane	25D
Two swept bends in perpendicular planes	40D
Valve	50D

Table 10.3. Some values from tests by Mattingly and Yeh (1991) on installation effects

Fitting	Upstream Distance (Diameters)	Change in Meter Factor (%)
Without tube bundle flow conditioner		
Single elbow	30	Less than 0.1
Two swept bends in perpendicular planes	25	+1
With tube bundle flow conditioner		
Single elbow	15	−0.1
Two swept bends in perpendicular planes	10	+0.2

Millington et al. (1986) reported data from tests on 24 meters, the dimensions of which ranged from about 11 mm to about 34 mm, with hub-to-tip ratios between about 0.4 and 0.5; tip clearance between 1 and 4%; flat, twisted, and T-shaped blades; and integral straighteners of about 1D to about 1.6D. Their results suggested that a single bend ($R/D = 1.5$) caused an error of about 0.3% at 5D upstream, but that a double bend causing swirl could cause serious errors at less than 10D (cf. Brennan et al. 1989 and Mattingly et al. 1987 on the effects of swirl).

Mattingly and Yeh (1991) reported tests on installation effects for turbine meters at a Reynolds number between 10^4 and 10^5 in a 50-mm water flow facility. In addition, they showed, for swirl caused by the double elbow out-of-plane configuration, that the swirl angle dropped from about 18° to about 6° after 40D and to about 3° after 80D (cf. Hutton 1974 and Salami 1985). Some of the values from their plotted data are given in Table 10.3.

Mattingly and Yeh concluded that the tube bundle flow conditioner may introduce errors. As an example, they gave the case of a meter 10D downstream of a single elbow. In this case, the conditioner caused the meter factor to shift from -0.2% to -0.8% . This suggests that the conditioner/flowmeter combination should be tested together (cf. Baker 1993 and Mottram and Hutton 1987 for other published work on installation).

10.2.12 MAINTENANCE

Problems can arise from defective bearings, jammed bearing or rotor, damaged blades, defective pickup, break in the transmission path or amplifier failure, or extraneous signal interference.

Servicing should take place at least annually. If possible, lines should be drained and the flowmeter removed for inspection for foreign matter and to check for free rotation of bearing and proper tip clearance. If necessary, the meter should be washed with clean fluid or an appropriate cleaner. If the impeller touches the casing, it may indicate a damaged or worn bearing. It will then probably be necessary to return it to the manufacturer for replacement. Electrical continuity should be checked, and the correct behavior of the sensor should be confirmed.

The nature of the output pickup signal waveform will often reveal the existence of vibration and nonuniform rotation (worn bearings) when examined with an oscilloscope.

10.2.13 VISCOSITY, TEMPERATURE, AND PRESSURE

Temperature differences between calibration and operation cause dimensional changes, viscosity change, density change, and velocity pattern shifts (Gadshiev et al. 1988). Manufacturers (cf. the standard documents) may provide correction factors (cf. Hutton 1986 concerning a universal Reynolds number curve).

Linear range decreases progressively above 1 cSt and virtually disappears between 50 and 100 cSt. Helical blades are affected much less by viscosity change than constant angle blades. Lower flow rates, smaller meter sizes, and high viscosity all lead to decrease in the range of linear operation. Other causes are gum, varnish, and other deposits on the bearings. A running-in period will usually overcome this problem. Care is needed in specifying liquid type when purchasing a turbine meter for a particular application since the viscosity of, for instance, JP-5 fuel can vary from 0.8 to 10 cSt or more (cf. Ball 1977 who gave a polynomial series for the effect of viscosity).

Pressure and temperature will have an effect on the liquid volume and will also affect the dimensions of the meter.

10.2.14 UNSTEADY FLOW

The effect of pulsation in liquid flow is much less than in gas flow because the density of the rotor is closer to that of the fluid. There may, however, be a tendency to overestimate the flow rate. Grey (1956) gave values of time constant for a range of meter sizes and showed that, in the range 12–150 mm sensing element diameter, the value was in the range 1–5 ms at full flow rate and 2–9 ms at half flow rate. Increased blade angle was also noted to increase the time constant. Higson (1964) suggested that the equation

$$\frac{q_2 - q_t}{q_2 - q_1} = \exp(-t/t_R) \quad (10.7)$$

where q_1 is the initial flow rate, q_2 is the final flow rate, q_t is the flow rate at time t , and t_R is the relaxation time, gives the transient behavior of a meter. He showed that, for the 20-mm meter used, change to 63% of the step change took about 0.6 revolutions and 6–110 ms. To return to equilibrium takes about $4t_R$.

One assumes, although to the author's knowledge there is no experimental flow data, that when the wheel slows down as the flow in the pipeline stops, the liquid or gas between the blades will be carried round in the wheel with little interchange with the upstream or downstream fluid.

10.2.15 MULTIPHASE FLOW

Baker (1991a) contains a list of references relating to the use of the meter in multiphase flows. The turbine meter has been used in experiments to model nuclear plant operating a long way from normal conditions (e.g., Ohlmer and Schulze 1985). Under the resulting flows, high precision is not necessarily required. Much of the published data for water-air flows do more to warn of the unpredictability than to offer hope of high precision and some data appeared to exhibit a hysteresis effect in vertical upward water-air flows. A small increase in air content from about 4% to about 5% caused an error increase of about 15%, and this was not immediately removed by reducing the air content back to 4%. A possible explanation was that a

vortex structure within the meter had trapped air and caused it to be drawn into the bearing. Upstream jet mixing may reduce errors. Meter speed has also been found to increase with increased solids concentration for the same volumetric flow rate. I consider that the safe rule is to avoid using the meter in multiphase flows.

Mark et al. (1990a, 1990b) sought to interpret the data from the pulse spacing in a turbine meter for void fraction measurement in a two-phase flow. They claimed that fluctuation in the time between pulses was caused by the presence of the second phase. They termed this variation signal turbulence and claimed that, at any flow rate, it was directly proportional to void fraction.

Johnson and Farroll (1995) claim that, even though errors in turbine water and air flows can be as high as 12.5% for a void fraction of 25%, the use of measurement of the fluctuations in the turbine rotor speed can be used to measure void fraction and water flow rate to an improved precision. The reader is referred to the original article for more details but should be cautious when making general deductions from this approach.

10.2.16 SIGNAL PROCESSING

Electrical noise may be the most troublesome element in turbine meter systems. Two sensors are used where increased reliability and self-checking is required. However, the signal-to-noise ratio should be high so that the counter is not affected by the noise.

Footprinting of turbine meters can be used as a means of real-time measurement monitoring and control. This has been used as a means to extend calibration intervals (Gwaspari 1990). The use of the full-frequency spectrum, for so long discarded as noise, opens up the possibility of condition monitoring particularly where the theoretical understanding is sufficient to link the spectrum distribution with mechanical and electrical causes (Higham et al. 1986, Turner et al. 1989; however, cf. Cheesewright et al. 1998).

10.2.17 APPLICATIONS

Typical applications are

- in the process industries – providing precision, corrosion resistance, intrinsic safety, good temperature and pressure rating, and ease of installation;
- in oil (including crude) pipelines – where shroud rings may be used with nickel rivets to increase the count rate by 4:1; when duty payable on hydrocarbons; may be able to cope with abrasive sand-laden crudes (Morse 1976);
- in a range of mechanical engineering test rigs;
- in the drinks and dairy industry;
- in cryogenics – with ball and journal bearings in liquid hydrogen, oxygen, nitrogen, argon, carbon dioxide, normal and superfluid helium; rapid changes in density which occur due to temperature changes; cryogenic fluids since they are clean and of low viscosity. (The most appropriate bearing is one with stainless steel balls, nonmetallic cage, and preferably self-lubricating properties. Rotors for cryogenic service are usually of a nickel alloy to ensure compatibility with liquid oxygen);

- in sanitary products – where the exacting requirements for precision measurement are appropriate for turbine application;
- as a secondary standard – to provide a transfer standard for the calibration of other meters;
- as a reference system – consisting of two meters in series so that the reference system is self-checking. (Of course the meters need to be isolated from each other by flow conditioning in the line between them);
- for high pressure – with very high pressure capabilities; and
- for high and low temperatures.

(cf. Baker 1993 for application references).

10.2.18 ADVANTAGES AND DISADVANTAGES

The compactness of the turbine meter gives it a considerable advantage over the positive displacement meter. The pressure drop may be twice as great for a positive displacement meter. Short-term repeatability is excellent, being better than 0.02%, whereas long-term repeatability may be only 0.2% over a period of 6 months, so that, for custody transfer, regular proving under operating conditions is necessary. Although viscosity effects for large meters used with crude oil are less than for smaller meters; nevertheless, a 200-mm meter will be affected by as much as $\frac{1}{2}\%$ for changes from 1 to 30 cSt and will need to be reproved for large changes due to fluid change or temperature. Above about 50 cSt, it will probably be necessary to use positive displacement meters. It is best for light products with viscosities which change little.

Other advantages are pulsed output, reliability over extended periods, and rapid response. Disadvantages are particles affecting bearing and wheel and sensitivity to installation and swirl, which may affect calibration.

10.3 PRECISION GAS METERS

10.3.1 PRINCIPAL DESIGN COMPONENTS

Figure 10.14 shows a typical gas turbine meter. Even though many of the previous comments concerning liquid meters are relevant, it is clear that there are significant design differences. The most obvious is the large hub and comparatively small flow passage. The reason for this is to impart as large a torque as possible on the rotor by moving the flow to the maximum radius and increasing the flow velocity. The second difference is the frequent use of a worm and gear output drive, resulting from the requirement by some national authorities for a mechanical display. It is common, however, to include an electrical output as well as the mechanical register. Bonner and Lee (1992) recorded some of the important innovations introduced into the design in the early 1960s; these innovations included flat helically twisted blades with overlap and blade tips that extend into a recess in the outer wall of the flow passage.

Lee et al. (1982) described a design in which the main rotor was followed by a second rotor to sense the condition of the installed flowmeter. The speed of the second wheel as a ratio of the first would change if the flow leaving the first wheel was deflected more by rotational constraints on the first wheel. It was possible for

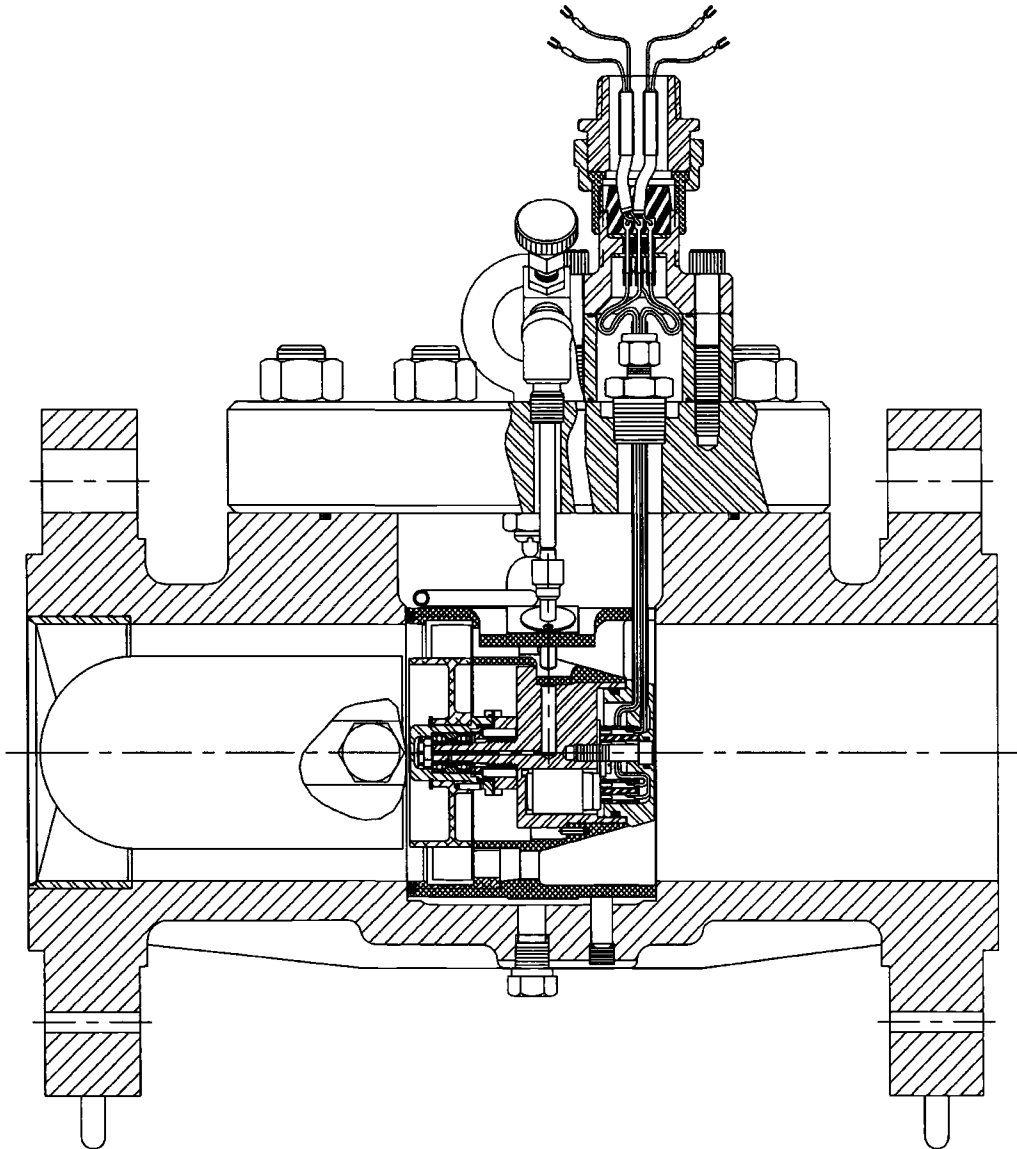


Figure 10.14. Diagram of 100 mm gas turbine meter (reproduced with permission of Daniel Europe Ltd.).

changes in the upstream flow also to affect the ratio. Self-correction was claimed for this device.

10.3.2 BEARING DESIGN

Shielded ball bearings are sometimes used. External lubrication is necessary in some designs using instrument oil. Special lubricants are available for use with oxygen. Under normal use, lubrication should be sufficient two or three times per year. Some designs, such as sealed ball bearings, do not need external lubrication. Sealed ball bearings are suitable for gases contaminated with solid particles.

10.3.3 MATERIALS

Rotor material is typically Delrin or aluminum, usually the latter for sizes greater than 150 mm. Sometimes stainless steel is used.

10.3.4 SIZE RANGE

For a 25-mm diameter, the range is from about 0.8–10 m³/h; for a 50-mm diameter, the range is from about 5–100 m³/h (with a minimum flow response of 1.2 m³/h); and for a 600-mm diameter, the range is from about 1,000–25,000 m³/h. One type offers a 30:1 turndown ratio. Numbers of blades are typically 12 to 24, and maximum pulse frequencies can be 3 kHz. The maximum pressure rating is up to 100 bar. These figures can vary significantly between manufacturers.

The pressure loss in the meter at maximum flow rate is about 5.5 mbar for the 50-mm-diameter meter and 14 mbar for the 600-mm-diameter meter. Pressure loss is, of course, directly related to the density, and therefore to the pressure, of the flowing gas. Reference should be made to manufacturers' data for given conditions.

10.3.5 ACCURACY

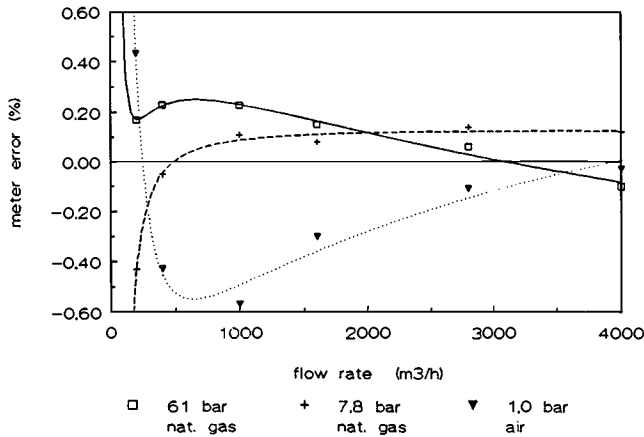
A typical specification for uncertainty is $\pm 2\%$ from the minimum flow rate $Q_{\min}$ to 20% of the maximum flow rate $Q_{\max}$, and $\pm 1\%$ from 20% $Q_{\max}$ to $Q_{\max}$. Some linearity claims of $\pm 0.5\%$ are made.

The linear performance may be as good as $\pm 0.5\%$ on about 20:1 turndown with repeatability of $\pm 0.02\%$. Maximum flow velocities can be up to 30 m/s. Other data available to me show remarkable calibration stability with shifts of order $\pm 0.2\%$ in 9 years with about 10⁸ m³ of natural gas having been transported through the turbine meters at 8-bar line pressure.

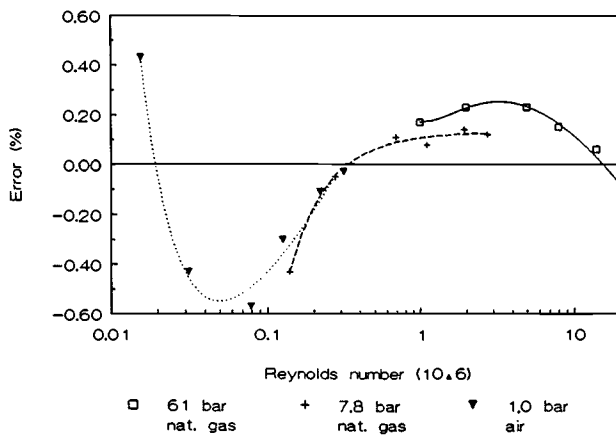
van der Grinten (1990) gave an error curve for gas turbine meters, which allowed for drag of gas between blades, annulus boundary layer, and friction in the bearing. Figure 10.15 demonstrates this curve fit and also shows the behavior of the meter with changing pressure and gas.

The turndown of the simple turbine meter increases proportionally with the square root of the gas density ratio. At a pressure of 20 bar, the turndown can be as high as 100:1 compared to 15:1 at a few millibars gauge working pressure (Griffiths and Newcombe 1970). Watson and Furness (1977) claimed that for low pressure nitrogen a flow range of 5:1 would be achieved, whereas for high pressure natural gas the range might be as much as 30:1.

van der Kam and Dam (1993) found that turbines can operate within an envelope of $\pm 0.5\%$ down to about 25% maximum flow, and of $\pm 1\%$ in the lower range. They also found that a pressure range of 1–10 bar caused errors well within 0.5% compared with 1% in older turbines. The Reynolds number dependence of turbines may, in some cases, allow the change in density to be related to curves of Reynolds number dependence (cf. Lee and Evans 1965). Their data demonstrate a repeatability within 0.1%. Another report (Erdal and Cabrol 1991) on calibration of 6 in. $\times$ 6 in. turbine meters suggested repeatability of about 0.24%, linearity of about 0.42%, and day-to-day repeatability of about 0.05% or less; for periods in excess of 4 years, de Jong and van der Kam (1993) suggested shifts in calibration of +0.2–0.3%, and Koning et al. (1989) claimed shifts of order 0.1% in 10 years. Gasunie's experience suggests that



(a)



(b)

Figure 10.15. Meter error of a turbine gas meter (van der Grinten 1990; reproduced with permission from Nederlands Meetinstituut): **(a)** Versus the flow rate at working conditions; **(b)** Versus the Reynolds number based on the inner meter diameter.

the shift is of order a few hundredths of a percent per year. van der Kam and de Jong (1994) claimed that 50:1 range with an error curve band of less than 0.5% over the whole range was no longer exceptional.

10.3.6 INSTALLATION

Work carried out by British Gas Engineering Research Station confirmed that these meters are extremely insensitive to flow disturbance, and in most practical cases no upstream or downstream straight pipe is required (Fenwick and Jepson 1975; cf. Harriger 1966). This is presumably due to various effects:

- the reduction of any swirl in the flow at the larger radius annulus due to conservation of angular momentum and also due to the flow straighteners;
- the flow development in the small annulus, which follows a large contraction; and

- c. the integrating effect due to the linear relationship between lift coefficient and incidence for small incidence angles.

They suggested that flow straighteners should only be used upstream if there is a possibility of persistent swirling flow at inlet (cf. Mottram and Hutton 1987).

For inlet flow disturbance, van der Kam and Dam (1993) suggested that the inlet flow conditioner could be quite effective in removing swirl, so that swirling flow from a double elbow out-of-plane (with a 40° swirl angle) may not cause errors of more than 0.3% (cf. Bellinga and Stronk 1974 on the development of a straightener). Also change in pipe size before the meter is insignificant. In extreme cases, a tube bundle may deal with the problem. Roughness does not affect performance. Temperature effects are small for a 20°C variation but are difficult to examine due to the lack of a suitable rig. The turbine meter is not suitable for wet or dirty flows. Gases should be clean and free of liquids and dust; otherwise, a filter with 5- μm filtration quality or better should be used. The upstream pipework should be cleaned before installing the meter (cf. Bonner 1993 and ISO 9951).

According to Harriger (1966), a short-coupled installation with as little as 4D upstream (2D straightener and 2D straight pipe) is allowable. However, swirl and pulsation can cause significant errors. Meters have a built-in flow straightener, which removes some of the swirl. Straightening vanes may be necessary if pipe fittings are at 5D or less upstream. The meter should be carefully centered and protuberances should be avoided for at least 5D upstream. Downstream the pipework should be full bore, but no other restrictions are placed on it. van der Kam and van Dellen (1991) for a 300-mm gas turbine meter suggested that 10D upstream length was sufficient to keep the meter within the limits allowed and 15D if swirl was present.

Excessive speeds can damage the meter, but 20% excess may be allowable for short periods.

Temperature should be measured downstream within 2D. One manufacturer gives a temperature range of -10 to 50°C .

If, as a result of changes in the process conditions, liquid condensate is produced in the pipework, means of drainage should be provided.

10.3.7 SENSING

The alternative means of measuring the rotation of the wheel are the mechanical system involving a gear train, which can result in retarding forces due to gear losses, losses in the magnetic coupling, and loads from the meter register and calibration adjustment, or the electromagnetic design in which the loads are much reduced. For a high frequency signal, a proximity switch operating off the aluminum blades, metal strips contained in the hub, or a follower disk on the main shaft, with magnetic probe or proximity switch, allow up to 3 kHz. For low frequencies of 1–10 pulses per revolution, a reed switch or slot sensor are used.

10.3.8 UNSTEADY FLOW

The gas turbine meter is affected by varying or pulsating flows. The increasing flow creates higher incidence angles on the turbine blades, and the turbine wheel accelerates fast. However, when the flow decreases, the blades presumably stall with low

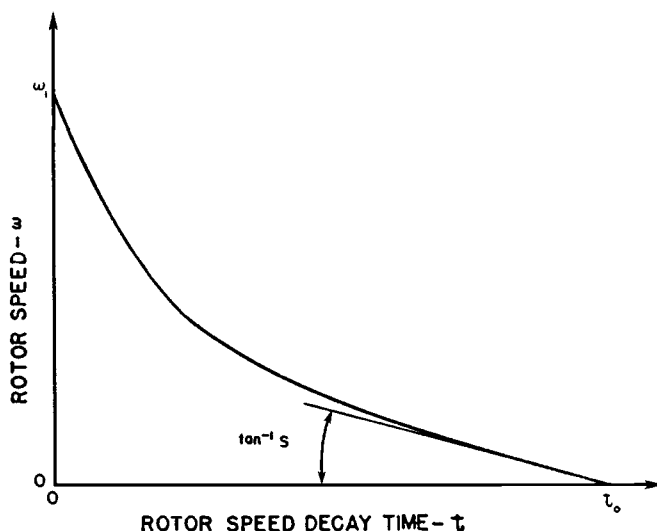


Figure 10.16. Rotor speed decay curve for no flow [Equation (10.A.20)] of a given meter during coasting process of the spin test (Lee and Evans 1970; reproduced with permission from ASME).

retarding force and hence low deceleration. The effect is to give an overestimation of the total flow. The turbine bearings may be damaged if subject to fluctuating flows for extended periods. Head (1956) gave a pulsation factor, which for turbine meters was

$$\frac{q_i}{q_v} = (1 + \alpha b \Gamma^2) \quad (10.8)$$

where q_i is the indicated flow rate, q_v is the actual flow rate, $\alpha = \frac{1}{8}$ for sinusoidal variation of flow, b is taken as unity for nonfollowing meters, and Γ is full flow amplitude relative to average flow. Head gives $\Gamma = 0.1$ as the practical threshold for significant error.

The transient analysis is given in Appendix 10.A.2. From this analysis, a speed decay curve for no flow is obtainable (Figure 10.16). We can obtain from this curve the rotor coast time to standstill and the final slope S of the curve, which is related to the ratio of the nonfluid drag and the inertia and hence to the condition of the bearings. However, de Jong and van der Kam (1993) were doubtful as to its value for high pressures. The reader is also referred to the paper by Lee and Evans (1970) who describe how they obtained speed of decay curves using an externally applied mechanical friction loading method and also gave typical values of inertia (e.g., for a 150-mm low pressure meter with plastic rotor $I = 0.242 \times 10^{-3} \text{ kg m}^2$ and for a high pressure meter with aluminum rotor $I = 0.486 \times 10^{-3} \text{ kg m}^2$). They also gave $\eta = 0.2$ as used in Appendix 10.A.2.

Lee et al. (1975) obtained the effect of sinusoidal pulsation on errors. Assuming the worst situation where rotor inertia is too great to follow pulsation, they obtained for a pulsation index of 0.1 about 0.5% error and for 0.2 about 2% error where the

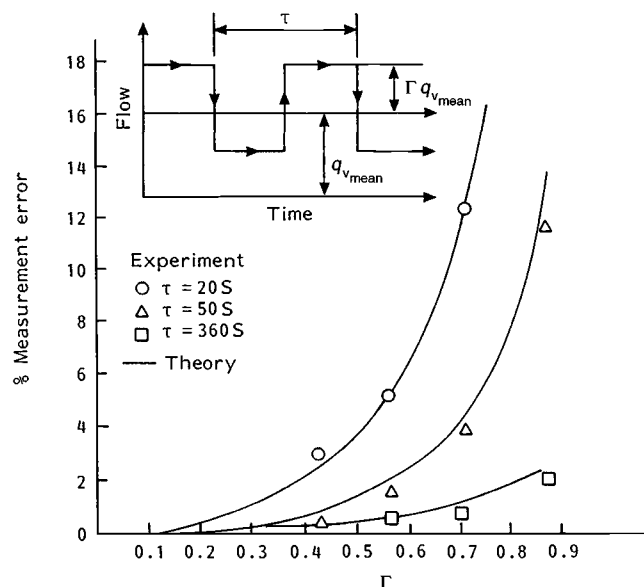


Figure 10.17. Effect of modulating flow on the performance of a 4-in. (100-mm) turbine meter (Fenwick and Jepson 1975; reproduced with permission of British Gas).

pulsation index is assumed to be given by

$$\Gamma = \frac{V_{\max} - V_{\min}}{V_{\max} + V_{\min}}$$

Figure 10.17 is adapted from Fenwick and Jepson (1975) and shows the effect on a turbine meter of a square wave pulsation. McKee (1992) found errors rising from zero for a 2% variation to 1.5% for 6% and above (cf. Atkinson 1992 who developed a computational analysis of the meter error for near-sinusoidal flow pulsations and Cheesewright et al. 1996 who queried the lack of reports of flows with pulsating waveforms).

Fenwick and Jepson (1975) gave as an example a 100-mm meter subjected to on-off pulsations with a period of 60 s resulting in a 40% overregistration (cf. Grenier 1991).

I worked with some data from tests where a gas conversion process required variations in natural gas flows from high to low flow and back with sudden change and significant errors, which were predicted by the methods of Jepson and others.

10.3.9 APPLICATIONS

One manufacturer claims that its meters can be used with all nonaggressive gases and fuel gases including town gas, natural gas, refinery gas, coke-oven gas, propane, butane, liquid gas/air mixtures, acetylene, ethane, nitrogen, carbon dioxide (dry), air, and all inert gases.

It is unusual for the turbine meter to be applied to flows of oxygen for the following reasons.

- a. Any lubricant used must be compatible with oxygen.

- b. A maximum flow rate of order 10 m/s may be specified for oxygen flows in pipework that may oxidize, and this tends to be rather low for a gas turbine meter.

Pfrehm (1981) described a system adapted from long accepted and accurate liquid measurement techniques to the mass measurement of ethylene gas. It consisted of a meter, density measurement, flow computer, and piston bidirectional pipe prover. The meter is claimed to provide accuracies of $\pm 0.2\%$ with linearity from 20% or less to 100% of maximum flow.

10.3.10 ADVANTAGES AND DISADVANTAGES

Newcombe and Griffiths (1973) made the following suggestions.

- a. Changes in friction and blade form due to mechanical deterioration or damage reduces the meter's rangeability and gives a slow registration. Filtering limits the rate of deterioration, but regular proving tests are required. Spin down tests can indicate bearing deterioration.
- b. Rapid flow variations cause the meter to read fast. For a 10 minute off/10 minute on cycle, one meter read 3% fast.
- c. Swirl will affect registration unless a straightener is fitted.
- d. Registration shifts between low and high pressure operation and, due to high bearing friction, can be as large as 2%.
- e. Security of gas flow is not affected if the meter fails.

To these, van der Kam et al. (1990) add reliability, high accuracy, dual measurement, and back-up system in a device with proven technology.

10.4 WATER METERS

10.4.1 PRINCIPAL DESIGN COMPONENTS

Figures 10.18(a, b) show diagrams of water meters.

A common design of this meter allows for a complete mechanical insert to fit into a flanged iron section of pipe with a specially made central containment section. The insert consists of flow straighteners, which also incorporate the upstream bearing; the rotor, which is usually of helical design and may have a substantial clearance between the blade tips and the casing; the rotor shaft, the motion of which is transmitted via a right angle worm gear assembly into a watertight gearbox; and a rudder (or trimming vane) upstream of the rotor, which provides a mechanical calibration adjustment by altering the angle at which the inlet flow stream hits the rotor blades. Water may be allowed to enter the gearbox via a filter, and the calibration rudder adjuster within the gearbox is accessed through the calibration port provided in the top cover. When the cover is fitted over the gearbox to form a watertight joint, the output from the gearbox is transmitted via a magnetic coupling to the totalizing register of the meter.

Figure 10.18(b) shows a meter in which the shaft is perpendicular to the pipe, providing a simpler transmission but a more tortuous flow path and, presumably,

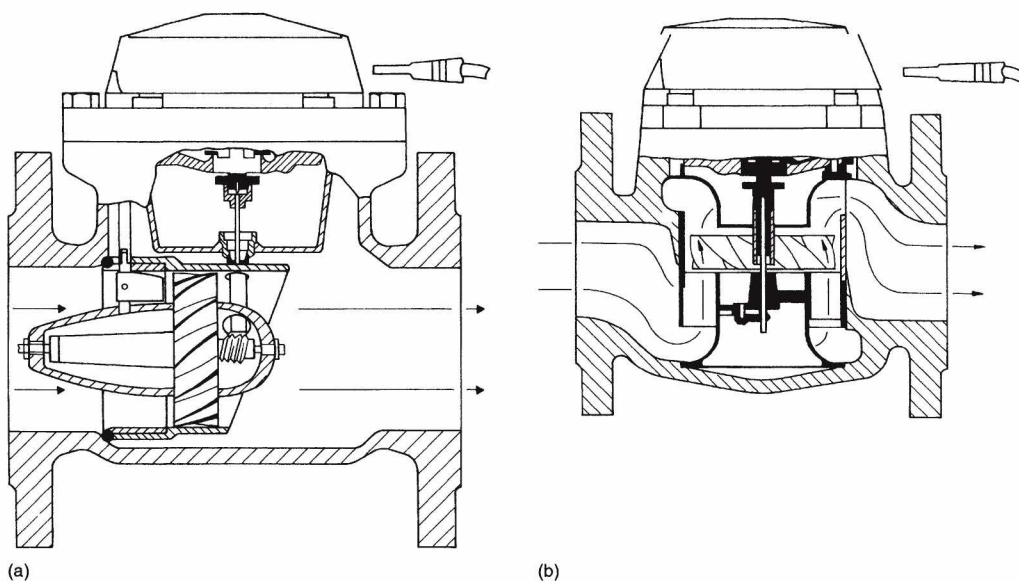


Figure 10.18. Diagrams of water meters (reproduced with permission of Meinecke): **(a)** Gas and Water Meter Manufacturers meter; **(b)** Gas and Water Meter Manufacturers meter with shaft perpendicular to the pipe.

resulting in a higher pressure loss. Special designs are available (Section 10.4.8), and Figure 10.19 shows one installed on a hydrant.

10.4.2 BEARING DESIGN

Manufacturers appear to give little information on this. Stainless steel shafts and in some cases stainless steel bearings are used.

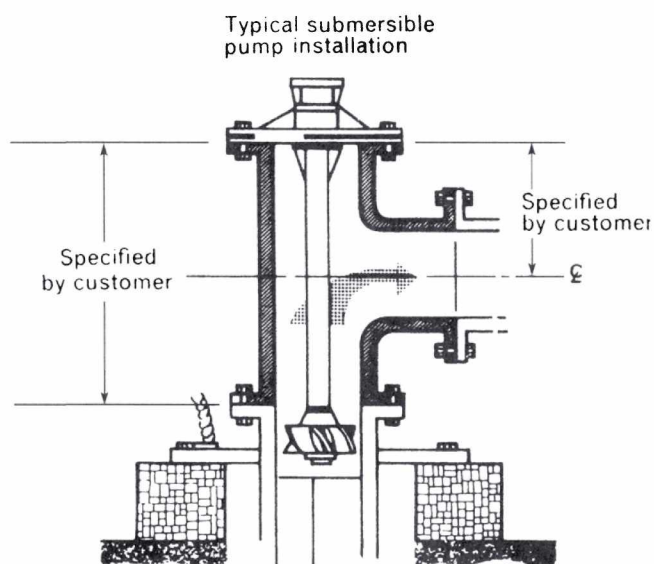


Figure 10.19. Hydrant meter (reproduced with permission of KDG Instruments).

10.4.3 MATERIALS

For temperatures up to 40°C, polythene propellers are suitable, but for meters for hot water, the internals are of hot-water-resistant plastic up to 150°C, and for temperatures up to 200°C, the measuring element may be red bronze and the wheel may be stainless steel. Other materials used are brass and stainless steel.

10.4.4 SIZE RANGE

Meters range from 15 to 500 mm. One manufacturer has introduced a dual system, which combines meters of very different capacity and uses a valve to control the flow through the larger meter.

10.4.5 SENSING

This is usually via a gear train but may also use optoelectrical methods, reed switches, etc.

10.4.6 CHARACTERISTICS AND ACCURACY

Figure 10.20 shows a typical operating envelope for a water meter. For the top of the range (q_{trans} to q_{max}), $\pm 2\%$ is required for a turndown between 11:1 and 5:1, depending on the size of the meter. For the bottom about 3:1 to 7:1 (q_{min} to q_{trans}), the requirement is $\pm 5\%$. Some values from ISO 4064 and BS 5728 specifications follow:

Size (mm)	50	65	80	100	150
q_{max} (m ³ /h)	30	50	80	120	300
q_n (m ³ /h)	15	25	40	60	150
q_{trans} (m ³ /h)	4.5	7.5	8	12	30
q_{min} (m ³ /h)	1.5	2	1.2	1.8	4.5
Starting flow (approx)	0.34	0.68	0.68	0.91	1.59
Head loss at q_{max} (bar)	0.20	0.04	0.09	0.08	0.09

where q_{max} is the maximum peak flow rate, q_n is the recommended continuous flow rate, q_{trans} is the flow rate at which the uncertainty specification changes, and q_{min} is the minimum flow at which the performance is defined.

Some commercial designs outperform these requirements, and the reader should check for ISO updates.

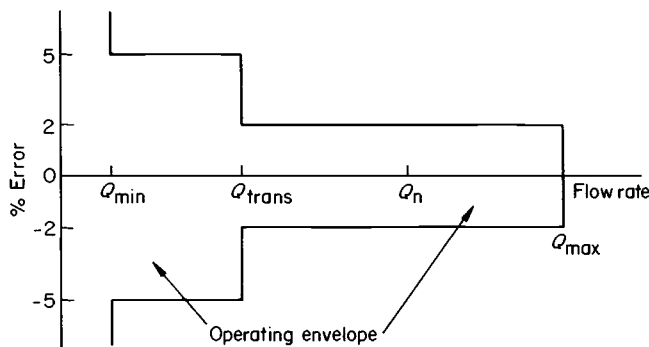


Figure 10.20. Operating envelope for a water meter based on ISO/BS specifications.

10.4.7 INSTALLATION

A typical requirement for installation is 5D upstream. In addition, air and solid matter should be avoided.

10.4.8 SPECIAL DESIGNS

Irrigation and mainline designs are inserted through openings in the pipe, manholes, and saddles or by a bolted flange in a T piece. Accuracy is unlikely to be high. Sizes can be as great as 1,000 mm in diameter.

10.5 OTHER PROPELLER AND TURBINE METERS

10.5.1 QUANTUM DYNAMICS FLOWMETER

One design used a patented twin-turbine configuration whereby the downstream slave turbine drove the shaft on which the indicator turbine bearings rotate freely. Under normal conditions, the rotational speeds of the indicator and slave turbines are closely matched. At high flow rates, the rotational speeds of the indicator turbine and the slave turbine/shaft assembly will begin to diverge. An integral flow-straightening device upstream of the indicator turbine reduces swirl and hence the effect of upstream flow distortion. With a special design of RF pickup, claims of $\pm 0.01\%$ linearity for a turndown of between 350:1 and 500:1 have been made. In gaseous flow, these rangeabilities were claimed to correspond to a mass turndown in excess of 1,000:1.

10.5.2 PELTON WHEEL FLOWMETERS

Pelton wheel flowmeters operate like a hydraulic Pelton wheel by using the impulse due to the fluid jet momentum. To achieve this, the flow is constricted to a small outlet and hence forms a high speed jet. A typical design is shown in Figure 10.21.

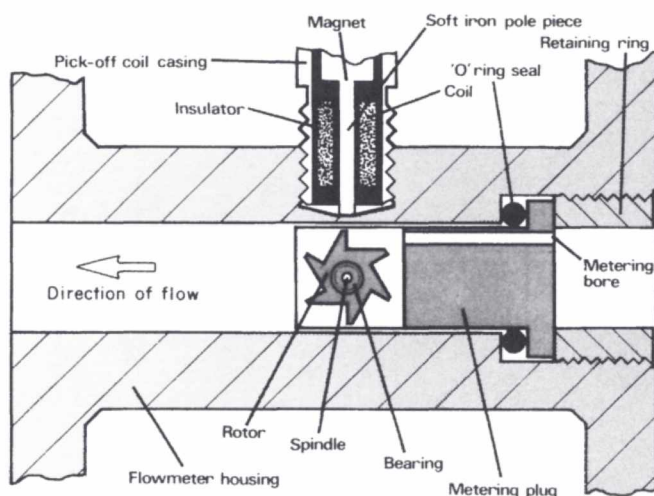


Figure 10.21. Pelton-wheel-type flowmeter (reproduced with permission of Nixon Instruments).

It is designed for use with liquids. An upstream filter with a 60-mesh insert may be recommended. A typical low flow range is 0.18–1.8 l/min with a repeatability claimed as $\pm 0.25\%$. The passing of the rotor tips is sensed by a reluctance pickup.

Other designs on the Pelton wheel principle have also been used to produce meters capable of flow rates as low as 0.06 l/h to 0.75 m³/h with repeatability of $\pm 0.3\%$ of FSD. Alternatively, they have been used as a bypass flowmeter, which operates across an orifice to allow higher ranges.

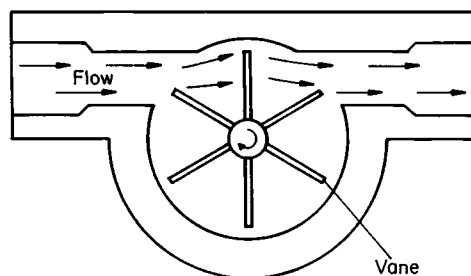


Figure 10.22. Vane flowmeter (reproduced with permission of Fisher Rosemount Ltd.).

10.5.3 BEARINGLESS FLOWMETER

This meter has a floating rotor. After the inlet, the liquid divides and flows upward and downward past the twin rotors, which are on a common shaft. The flow lifts the rotor off its seating because there is lower static pressure in the flow than under the rotor. The flow at both ends of the shaft passes the blades on the rotor wheels before leaving through two channels that join at the exit. Sensing magnets are sealed into the rim of the lower rotor, and their passage is sensed by an inductive coil. The basic instrument has a size range of about 15- to 80-mm diameter with flow range from about 0.2 to 5.2 m³/h.

10.5.4 VANE-TYPE FLOWMETERS

Vane-type flowmeters cannot be claimed as high precision instruments. The description “inferential,” sometimes applied to turbine flowmeters, is probably appropriate for these meters because the flow of fluid through the meter is inferred from the effect on the rotor of its interaction with a proportion of the fluid flow.

Figure 10.22 shows a diagram of the principle of operation. The liquid hits the vanes and imparts momentum to them. The clearance between the vanes and the casing distinguishes this meter from the positive displacement meter and allows the fluid to slip past the vanes on the return.

The flow ranges for this type of meter may cover 0.45 to about 12 m³/h with a maximum pressure drop of about 1.4 bar and a maximum allowable kinematic viscosity of about 48 cSt.

Other devices of a similar type are the angled vane flowmeter with three to four blades on a shaft angled at about 45°, designed with a removable body, and a further design operating on the same principle but in the converse way to a paddle wheel. The fluid passing pulls the ridged wheel with it.

10.6 CHAPTER CONCLUSIONS

The considerable range and diversity of this family of designs are shown in this chapter. The one drawback is that with rotation there is often the potential problem of wear to consider, especially for nonlubricating liquids and dry contaminated gases.

However, against this must be set their extraordinary versatility. Very high accuracy is achieved in some; very large turndown ratio is achieved in others; and suitability for a wide range of fluids and a very wide price range are also features of the family. New designs with extraordinarily great turndown and others with self-checking ability indicate that the family is far from dying out.

The problem of bearing wear has been controlled but requires that all types of meter are subject to regular maintenance. van der Kam and de Jong (1994) claimed that the gas turbine meter, calibrated at the operational pressure, can be one of the most accurate gas meters available at present and could be used more for custody transfer. Gasunie have suffered very few mechanical failures (blade failure or damaged bearings) and van der Kam and Dam (1993) reckon the turbine meter still the best option and capable of further development.

Great care must be exercised in ensuring that the installation does not invalidate the calibration. Transient behavior in gases results in an overregistration of the flow. New sensor technologies may lead to lower drag and higher integrity of signal, particularly in adverse radiation environments.

Intelligent designs that build in a self-checking or condition-monitoring facility are likely to be of particular value for metering high value fluids in fiscal and custody transfer applications.

Theoretical analysis of flow through the meter should take account of the very small deflection compared with the substantial deflection that occurs in a power turbine. Ferreira (1988) attempted some computational work on the flows within some turbine meters and compared these with LDA measurements. The results were interesting and original, but the nature of the flow and the very small deflections, which result in the case of a well-designed rotor, suggest that future computations should take account of these features in a perturbation-type analysis.

APPENDIX 10.A

Turbine Flowmeter Theory

10.A.1 DERIVATION OF TURBINE FLOWMETER TORQUE EQUATIONS

Blade Aerodynamics

There are two distinct approaches to the blade forces in the literature. Using cascade theory, Tsukamoto and Hutton (1985) give the driving force as

$$T_d = \rho N \int_{r_h}^{r_t} r V_z^2 s (\tan \beta_2 - \tan \beta_1) dr \quad (10.A.1)$$

which equates the torque to the change in angular momentum of the fluid passing through the blade row. Replacing β_1 by $\beta - i$ and β_2 by $\beta - \delta$, and noting that both

i and δ will be very small, we may rewrite Equation (10.A.1) as

$$T_d = \rho N \int_{r_h}^{r_t} r V_z^2 s [\tan(\beta - \delta) - \tan(\beta - i)] dr \quad (10.A.2)$$

$$= \rho N \int_{r_h}^{r_t} r V_z^2 s \left(\frac{\tan \beta - \delta}{1 + \delta \tan \beta} - \frac{\tan \beta - i}{1 + i \tan \beta} \right) dr \quad (10.A.3)$$

Ignoring terms in $i\delta$, etc. ($1^\circ = 0.0175^\circ$),

$$T_d = \rho N \int_{r_h}^{r_t} r V_z^2 s \frac{(i - \delta)(1 + \tan^2 \beta)}{1 + (i + \delta) \tan \beta} dr \quad (10.A.4)$$

or if we ignore $(i + \delta)$ compared with unity,

$$T_d = \rho N \int_{r_h}^{r_t} r V_z^2 s (i - \delta)(1 + \tan^2 \beta) dr \quad (10.A.5)$$

The alternative airfoil theory approach is to use lift and drag coefficients. These are given per unit length of the blade by

$$L = \frac{1}{2} \rho c K C_L \left(\frac{V_z}{\cos \beta_m} \right)^2 \quad (10.A.6)$$

$$D = \frac{1}{2} \rho c C_D \left(\frac{V_z}{\cos \beta_m} \right)^2 \quad (10.A.7)$$

where K is the factor that allows for the change in lift coefficient between an isolated airfoil and a cascade. The torque on the rotor is now given by

$$T_d = \frac{1}{2} \rho N \int_{r_h}^{r_t} \frac{r V_z^2}{\cos \beta_m} c (K C_L - \tan \beta_m C_D) dr \quad (10.A.8)$$

This is essentially the term given by Blows (1981) for the aerodynamic lift torque minus the aerodynamic drag torque. Next, Equations (10.A.2) and (10.A.8) are shown to be equivalent.

Equating the axial force to the force due to change in static pressure:

$$\begin{aligned} Z &= (p_1 - p_2)s \\ &= -\frac{1}{2} \rho (V_1^2 - V_2^2)s + (p_{01} - p_{02})s \end{aligned}$$

Equating the tangential force to the rate of change of angular momentum,

$$Y = \rho V_z^2 s [\tan(\beta - \delta) - \tan(\beta - i)]$$

Substituting for V_1 and V_2 , we obtain

$$\begin{aligned} Z &= -\frac{1}{2} \rho V_z^2 s [\tan^2(\beta - i) - \tan^2(\beta - \delta)] + s(p_{01} - p_{02}) \\ &= -\rho V_z^2 s [\tan(\beta - i) - \tan(\beta - \delta)] \tan \beta_m + s(p_{01} - p_{02}) \end{aligned}$$

We may write the lift in terms of Y and Z as

$$L = Y \cos \beta_m + Z \sin \beta_m$$

and, substituting for Y and Z ,

$$L = \rho V_z^2 s [\tan(\beta - \delta) - \tan(\beta - i)] \cos \beta_m - \rho V_z^2 s [\tan(\beta - i) - \tan(\beta - \delta)] \frac{\sin^2 \beta_m}{\cos \beta_m} + s(p_{01} - p_{02}) \sin \beta_m$$

So

$$L = \rho V_z^2 s [\tan(\beta - \delta) - \tan(\beta - i)] \sec \beta_m + s(p_{01} - p_{02}) \sin \beta_m$$

and for the drag

$$D = Z \cos \beta_m - Y \sin \beta_m$$

and substituting for Y and Z

$$D = \rho V_z^2 s [\tan(\beta - \delta) - \tan(\beta - i)] \sin \beta_m + s(p_{01} - p_{02}) \cos \beta_m - \rho V_z^2 s [\tan(\beta - \delta) - \tan(\beta - i)] \sin \beta_m$$

So

$$D = s(p_{01} - p_{02}) \cos \beta_m$$

Substituting into Equations (10.A.6) and (10.A.7), we obtain

$$\begin{aligned} KC_L &= 2 \frac{s}{c} [\tan(\beta - \delta) - \tan(\beta - i)] \sec \beta_m \cos^2 \beta_m + \frac{s}{c} \left(\frac{p_{01} - p_{02}}{\frac{1}{2} \rho V_z^2} \right) \sin \beta_m \cos^2 \beta_m \\ &= \frac{2s}{c} [\tan(\beta - \delta) - \tan(\beta - i)] \cos \beta_m + C_D \tan \beta_m \end{aligned} \quad (10.A.9)$$

where

$$C_D \tan \beta_m = \frac{2s}{\rho c} \left(\frac{p_{01} - p_{02}}{V_z^2} \right) \sin \beta_m \cos^2 \beta_m$$

which shows that Equations (10.A.2) and (10.A.8) are equivalent.

The theoretical expression given by various authors for the lift coefficient is

$$C_L = 2\pi \sin \alpha$$

or in fuller form by Batchelor (1967)

$$C_L = 2\pi \sin \alpha (1 + 0.77 t_b / c)$$

Lift curves are reproduced in Figure 10.9, and drag curves can be found in Wallis (1961). The angle α is that between the far field and the line of zero lift of the blade, which for a flat plate is the angle with the plate. It appears to be conventional to take the direction of the far flow field for a cascade as the mean direction between inlet and outlet and α for a cascade as the angle between this mean direction and the blade angle (cf. Thompson and Grey 1970). Thus for a cascade of closely spaced flat plates, the exit angle relative to the blades will be zero, whereas the inlet angle

will be equal to the far field. Thus the value of α will be half the incidence angle i . So following convention,

$$C_L = 2\pi \sin i/2 \quad (10.A.10)$$

for the closely spaced flat plates, and, in this limit, according to Weinig (1932):

$$K = \frac{s}{c} \frac{2}{\pi \cos \beta} \quad (10.A.11)$$

Thus

$$KC_L = \frac{s}{c} \frac{4 \sin i/2}{\cos \beta} \simeq \frac{s}{c} \frac{2i}{\cos \beta} \quad (10.A.12)$$

and Equations (10.A.5) and (10.A.8) are identical provided $\delta = 0$ and $C_D = 0$.

For values of s/c not tending to zero, we shall find it useful to use the values of K obtained by Weinig (1932) and given by Wislicenus (1947), and it will be necessary to write the lift coefficient in terms of the true incidence angle i . From Figure 10.3,

$$\frac{\tan(\beta - i) + \tan(\beta - \delta)}{2} = \tan \beta_m$$

so that

$$\beta_m \simeq \beta - \frac{i + \delta}{2}$$

but

$$\beta - \alpha = \beta_m$$

so

$$\alpha = \frac{i + \delta}{2}$$

and

$$C_L = 2\pi \left(\frac{i + \delta}{2} \right) = \pi(i + \delta) \quad (10.A.13)$$

and hence

$$T_d = \frac{1}{2} \rho N \int_{r_h}^{r_t} \frac{r V_z^2}{\cos \beta_m} c [K\pi(i + \delta) - C_D \tan \beta_m] dr \quad (10.A.14)$$

If we make the assumption that both δ and C_D may be neglected in the case of a well-designed turbine wheel, we have

$$T_d = \frac{\pi}{2} \rho N \int_{r_h}^{r_t} \frac{r V_z^2 c K(r) i(r)}{\cos \beta_m} dr \quad (10.A.15)$$

Finally, it is useful to consider the actual values of C_D and δ . We can first of all obtain from Equations (10.A.5), (10.A.8), and (10.A.10)

$$\frac{1}{2} \frac{c}{\cos \beta_m} K \pi i = \frac{s}{\cos^2 \beta} i$$

$$K \simeq \frac{2}{\pi} \frac{s}{c} \frac{1}{\cos \beta_m}$$

Then applying this to C_D and δ and making use of Equations (10.A.5) and (10.A.14),

$$\frac{1}{2} \frac{cK\pi\delta}{\cos \beta_m} - \frac{1}{2} \frac{c}{\cos \beta_m} C_D \tan \beta_m = -\frac{s\delta}{\cos^2 \beta_m}$$

Hence,

$$C_D = 4 \frac{s}{c} \frac{\delta}{\sin \beta_m}$$

From Wallis (1961), as i tends to zero, C_D tends to 0.017. For $\beta = 45^\circ$ and $s/c \simeq 1$,

$$C_D \simeq 0.017 \simeq 4\sqrt{2}\delta$$

so that

$$\delta \simeq 0.003^\circ \simeq (1/6)^\circ$$

So if we neglect the drag terms (Equation 10.A.14),

$$K\pi\delta - C_D \tan \beta_m \simeq 2 \frac{s}{c} \frac{1}{\cos \beta_m} \delta - \frac{4s}{c} \frac{\delta}{\sin \beta_m} \tan \beta_m$$

we are actually neglecting

$$-2 \frac{s}{c} \frac{\delta}{\cos \beta_m} \simeq -0.0085$$

We can compare this with the size of the drive torque

$$K\pi i = 2 \frac{s}{c} \frac{i}{\cos \beta_m}$$

Thus the neglected term is δ/i of the drive term.

Drag Terms

Tsukamoto and Hutton (1985) identified four drag torques:

- T_b , bearing drag;
- T_w , hub disk friction drag;
- T_t , tip clearance drag;
- T_h , hub fluid drag.

In the first three, the torque is obtained from an equation of the form

$$T = \text{Shear stress} \times \text{Area} \times \text{Radius} \times \text{Reynolds number} \\ \times \text{Function of Reynolds number}$$

For instance, for the bearing drag, we obtain

$$T_b = \left(\frac{\mu r_j \omega}{t} \right) (2\pi r_j B)(r) \left[\frac{\rho(r_j \omega) t}{\mu} \right] f(\text{Re}) \\ = \pi \rho (r_j \omega)^2 r_j^2 B (\text{Re})^n$$

where

$$f(\text{Re}) = \begin{cases} \frac{2}{\text{Re}} & \text{for } \text{Re} < 1,000 \\ \frac{0.016}{\text{Re}^{0.25}} & \text{for } \text{Re} > 1,000 \end{cases}$$

where Re is the Reynolds number based on the radial gap and circumferential speed. Blows (1981) obtained the same expression if $\text{Re} < 1,000$. From the work of Tsukamoto and Hutton (1985), the dependence on ω ranges from $T \propto \omega$ to $T \propto \omega^{1.8}$. At low speeds, the dependence is in the range ω to $\omega^{1.5}$, whereas at high velocities the dependence tends to $\omega^{1.75}$ to $\omega^{1.8}$.

The hub fluid drag is

$$T_h \propto \frac{1}{2} \rho W_h^2 C_h$$

where W_h is the relative velocity at the hub and C_h is a constant for a particular design. It will therefore behave like an adjustment to the main aerodynamic drag term coefficient.

Flowmeter Equation

We may now equate

$$T_d = T_b + T_w + T_t + T_h$$

and we may note that, with increasing speed, the terms T_b , T_w , and T_t will become less important compared with T_d and T_h . If we combine the T_h drag with the aerodynamic drag, we obtain from Equation (10.A.8)

$$\int_{r_h}^{r_t} \frac{r V_z^2 c}{\cos \beta_m} (K C_L - C'_D \tan \beta_m) dr = 0$$

where C'_D has incorporated T_h . If we then substitute for $C_L = \pi i$ and approximate C'_D as constant,

$$\int_{r_h}^{r_t} \frac{r V_z^2 c}{\cos \beta_m} K \pi i dr = \int_{r_h}^{r_t} \frac{r V_z^2 c}{\cos \beta_m} C'_D \tan \beta_m dr \quad (10.A.16)$$

By iteration, this equation enables the wheel speed to be obtained for a given flow profile in the annulus (cf. Newcombe et al. 1972 for an expression for gas meters).

10.A.2 TRANSIENT ANALYSIS OF GAS TURBINE FLOWMETER

Lee and Evans (1970) used a simple analysis to derive the behavior of a meter by observing the spin down as an indicator of changed bearing friction. The transient equation (Lee et al. 1975, cf. Bonner 1977) is

$$\frac{I d\omega}{dt} = T_d - T_r \quad (10.A.17)$$

where the driving torque T_d may be obtained from Equation (10.A.1) for one radius and unit blade length as

$$T_d = \rho N r V_z^2 s (\tan \beta_2 - \tan \beta_1) \quad (10.A.18)$$

or, since $V_z \tan \beta_1 = r\omega$ and putting

$$T_d = \rho N r s V_z \frac{(V_z \tan \beta - r\omega)}{(1 + \eta)}$$

where η allows for the blade deviation factor. Also putting $T_r = \rho A r^3 C_f \omega^2 / 2 + T_n$, where the first term is fluid drag and the second term T_n is nonfluid drag assumed constant, Lee et al. (1975) then obtained

$$D_1 \frac{dW}{d\tau} + VW + D_2 W^2 = V^2 - D_3 \quad (10.A.19)$$

where

$$D_1 = \frac{(1 + \eta)I}{r^2 \rho A V_0 T}$$

$$D_2 = \frac{(1 + \eta) \tan \beta C_f}{2}$$

$$D_3 = \frac{(1 + \eta) A T_n}{r \tan \beta \rho A^2 V_0^2}$$

and $V = V_z / V_0$, $W = r\omega / V_0 \tan \beta$ and $\tau = t / T$, the fundamental period of the pulsating flow. This equation is soluble on a computer. For certain cases, an exact solution is possible. Lee and Evans (1970) give the speed decay curve for no flow ($V = 0$) as

$$\tau = \frac{D_1}{D_2} \left(\frac{1}{(D_3/D_2)^{1/2}} \right) \left(\tan^{-1} \frac{W_1}{(D_3/D_2)^{1/2}} - \tan^{-1} \frac{W}{(D_3/D_2)^{1/2}} \right) \quad (10.A.20)$$

This curve is shown in Figure 10.16. We can obtain from this curve, as shown by Lee and Evans (1970), that the rotor coast time to standstill is

$$\tau_0 = \frac{D_1}{D_2} \left(\frac{D_2}{D_3} \right)^{1/2} \tan^{-1} \left[W_1 \left(\frac{D_2}{D_3} \right)^{1/2} \right] \quad (10.A.21)$$

and just before this point

$$D_1 \frac{dW}{d\tau} = -D_3 \quad (10.A.22)$$

so that, from the slope S , we have

$$S = -\frac{D_3}{D_1} \quad (10.A.23)$$

Vortex-Shedding, Swirl, and Fluidic Flowmeters

11.1 INTRODUCTION

This chapter deals with measurement techniques based on instabilities in fluid behavior. In preparing this chapter, I benefited from an unpublished note in 1988 by Dr. M. V. Morris, of Cranfield University, who drew my attention to some early references on this type of meter.

11.2 VORTEX SHEDDING

The detailed behavior as fluid flows past a cylindrical bluff body is given in Appendix 11.A.1 in relation to the size of the Reynolds number based on the diameter of the cylinder. Figure 11.1 shows the development of the flow past such a bluff body until, at a Reynolds number of about 40, the familiar shedding phenomenon starts. However, small changes of detail in the pattern of the shedding are illustrated by the diagram for certain Reynolds numbers. Frequency changes occur at these Reynolds numbers. These changes could result in nonlinearity and poor repeatability in a meter.

Figure 11.2 is a simple diagram to suggest how, as a vortex rolls up downstream of a noncircular bluff body, it first forms, fills with rotating fluid, and then draws in fluid with opposite rotation, which eventually cuts off the vortex and allows it to move downstream. In Appendix 11.A.2 a simple argument is used to suggest that the frequency of shedding f is about

$$f = V/2\pi w \quad (11.1)$$

where V is the velocity past the body of width w . An important parameter, the Strouhal number, is given by

$$S = fw/V \quad (11.2)$$

This relationship has been applied to long slender cylinders. Our application is to the vortex-shedding flowmeter where the flow is in a pipe and the width of the shedder (bluff body) is between a quarter and a third of the pipe diameter, and thus is not a long slender cylinder. However, as we shall see, this length/width ratio is advantageous in terms of the strength of vortex shedding and leads to coherent shedding, whereas longer and more slender shedding bodies result in weaker and less coherent shedding.

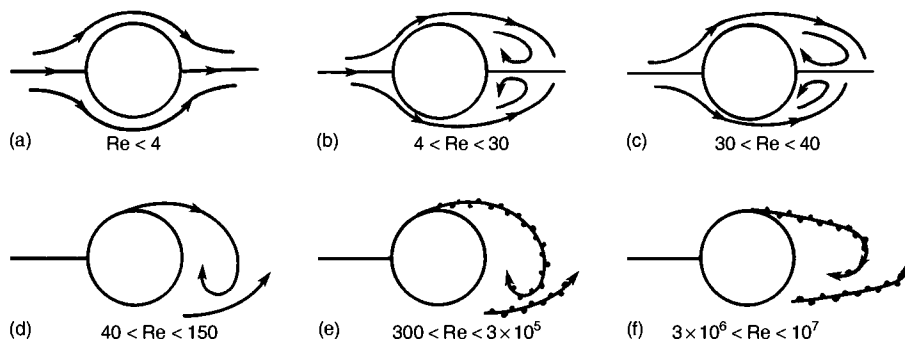


Figure 11.1. Flow patterns around a cylindrical body: **(a)** For $Re < 4$; **(b)** Initiation of separation for $4 < Re$ and up to about $Re = 30$; **(c)** Lack of separation symmetry behind cylinder for about $30 < Re < 40$; **(d)** Stable and viscous shedding for $40 < Re < 150$; **(e)** Stable shedding with turbulent shear layer for $300 < Re < 3 \times 10^5$; **(f)** Stable shedding with turbulent boundary and shear layers for $3 \times 10^6 < Re < 10^7$.

11.3 INDUSTRIAL DEVELOPMENTS OF VORTEX-SHEDDING FLOWMETERS

The use of this instability for velocity measurement was recognized at an early stage (cf. Roshko 1954) and has led to its use in flowmeters and probes. (cf. Medlock 1976).

Most modern meters use bluff bodies that are not circular in cross-section and that have a sharp shedding edge to remove the shifts resulting from changes in the boundary layer. Coherent shedding appears to take place along the length of the bluff body if the length (in this case the diameter of the pipe) is in the range of about 2.5 to 4 bluff body widths (Zanker and Cousins 1975). By *coherence* we mean that the vortex is shed from end to end of the bluff body on the same side at the same time.

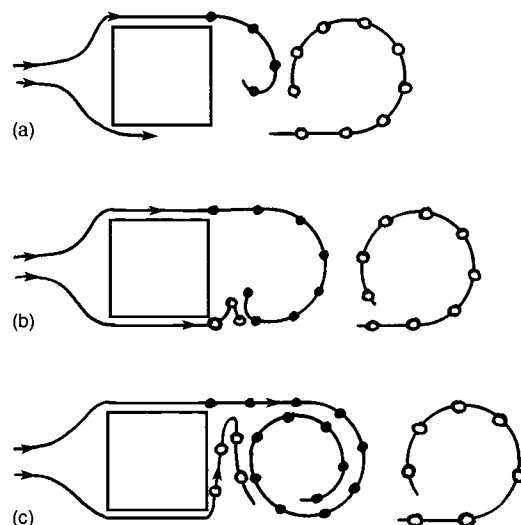


Figure 11.2. Diagram of vortex formation behind a bluff body: **(a)** Start of formation of new vortex; **(b)** Opposite vorticity starting to be entrained; **(c)** Opposite vorticity breaks and separates vortex.

For a cylindrical bluff body, it is found that the strength of the force caused by the vortex shedding is greatest when D/w is about 2.8 because the shedding becomes more coherent and enhanced. By this means, the shedding becomes approximately two-dimensional. For a flowmeter where the shedder spans the pipe, it is advantageous if its width is about a quarter of the pipe diameter. The wall of the tube then acts as end plates for the shedder and encourages coherent and strong vortex shedding. Coherence and stability of the shedding range has been achieved down to about $Re = 10^4$, where the Reynolds number is based on bluff body width (cf. Takamoto and Komiya 1981 who described a positive feedback system to extend the range below $Re = 2,000$).

In flow measurement, strong pressure pulsations caused by the shedding will enhance the signal, resulting in easier detection of shedding.

Three-dimensionality in the flow stream (e.g., turbulence, velocity gradients, and swirl) has detrimental effects on the quality of the vortex shedding (Zanker and Cousins 1975, cf. El Wahed and Sproston 1991 and Robinson and Saffman 1982). Except in carefully controlled conditions, a slender cylinder in the flow will create vortex shedding, which varies in strength and phase along its length (cf. Saito et al. 1993).

11.3.1 EXPERIMENTAL EVIDENCE OF PERFORMANCE

The vortex flowmeter, therefore, consists of a bluff body, usually of noncircular cross-section, spanning the pipe and having a width of a quarter to a third of the pipe diameter (Figure 11.3). Zanker and Cousins (1975) provided a useful review of early work on vortex flowmeters (cf. Tsuchiya et al. 1970 and Cousins et al. 1973) and the various effects that take place.

As the body becomes thicker (w increases) and the velocity past the body, therefore, increases for a fixed volumetric flow rate, Equation (11.1) suggests that the

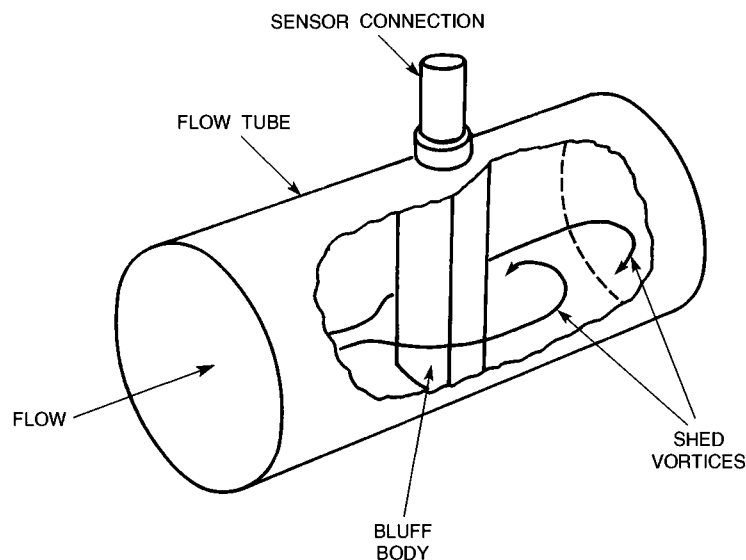


Figure 11.3. Diagram of vortex flowmeter.

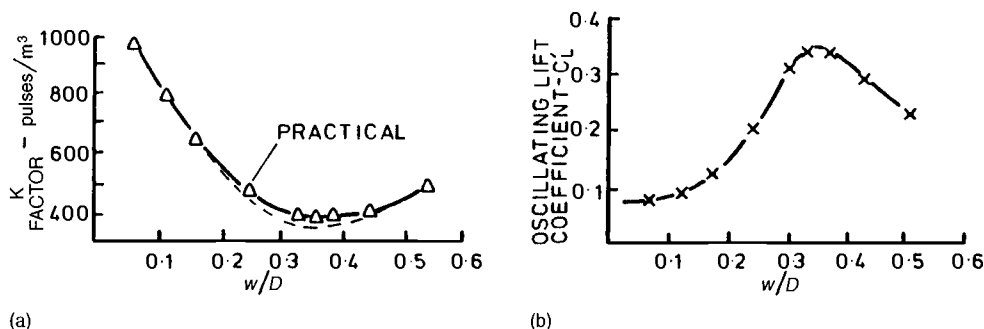


Figure 11.4. Pulses per unit volume and lift force for cylindrical bodies (from Zanker and Cousins 1975; reproduced with permission from NEL): **(a)** Pulses per unit volume against blockage ratio; **(b)** Lift coefficient against blockage ratio.

frequency and hence the pulses per unit volume of fluid may change. We shall refer to this ratio as the K factor (see Section 1.4). Figure 11.4(a) for a cylindrical bluff body shows this factor reducing to a minimum at a w/D of about 0.35. When w is of order $0.1D$, the strength of the lift coefficient is low [Figure 11.4(b)], but when it has the value $0.35D$, it is at a peak. Zanker and Cousins suggested that this strength was due to two effects that fortunately occur together:

- coherent shedding due to the shorter length of the bluff body between the end plates of the wall of the tube;
- the accelerating flow through the smaller space between the body and the tube wall, leading to a more uniform profile and less dependence on the upstream flow profile.

An important reason for the choice of the minimum is that small variations in w/D have a minimal effect on the frequency.

The constancy of the meter factor is shown in Figure 11.5 for a Reynolds number range of nearly 1,000:1 with air and water where the scatter was usually within $\pm 0.5\%$ (Zanker and Cousins 1975). White et al. (1974) gave a calibration curve

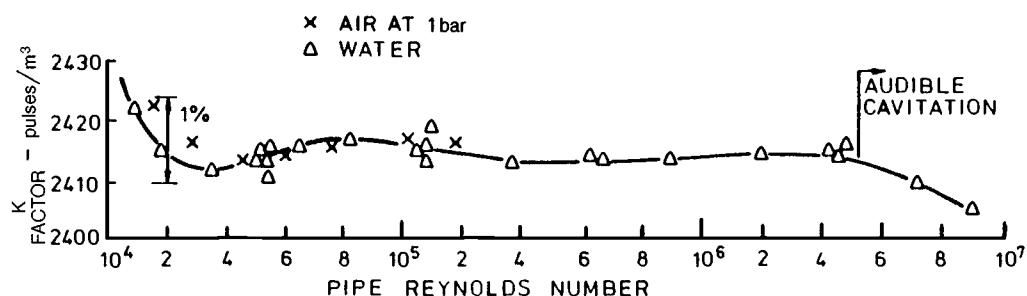


Figure 11.5. Typical calibration curve for a 3-in. (75-mm) vortex meter (from Zanker and Cousins 1975; reproduced with permission from NEL).

for a 3-in. (75-mm) meter in air over a 360:1 turndown (with a lower Reynolds number of 10^4) and showed that scatter was of order $\pm 1\%$ over the range. A major problem in achieving this size of turndown is how to sense at the varying level of signal. If velocity changes by 100:1, then pressure head variation due to the velocity will change by $10^4:1$. Goujon-Durand (1995) confirmed that a minimum Reynolds number of $Re = 30,000$ is probably necessary to avoid serious increase in nonlinearity. Takahashi and Itoh (1993) showed a slight decrease in Strouhal number (about 2%) from $Re = 20,000$ to $Re = 50,000$ for heavy oil, water, and air.

Inkley et al. (1980) obtained data to show the small effect of temperature change through viscosity and density on the meter. Change of liquid and liquid viscosity appears to have caused calibration shifts of order 0.5%. If the meter is designed close to the minimum in Figure 11.4(a), any variation due to temperature expansion will be minimized. The major change was due to cavitation. The authors agreed with White et al. (1974) that a universal curve should be obtainable and foreshadowed the work of Takamoto and Terao (1994) and, no doubt, others.

11.3.2 BLUFF BODY SHAPE

Figure 11.6 shows the effect of body shape on the optimum value of w/D and also shows that the K factor decreases from the value for circular bluff bodies. The truncated triangular or trapezium shape appears to be best if length to width is in the range 1.2 to 1.5. The rectangular body with length to width about 0.6 may have strong shedding.

Lucas and Turner (1985) confirmed that a splitter plate attached to the bluff body improved signal quality, which, presumably, is the reason for the T and truncated triangular body shapes.

El Wahed and Sproston (1991) experimented with various bluff body shapes in air. The sensing used a modulated signal from naturally occurring flow-induced streaming currents. The shedder shapes are shown in Figure 11.7. The shedders had end plates to create two-dimensional conditions. The main conclusions were that the best position for the sensor was attached to the front face of the bluff body and that:

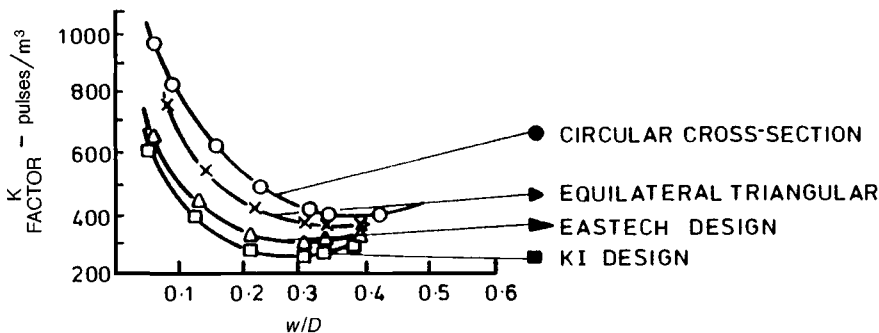


Figure 11.6. Optimum w/D for various body shapes (from Zanker and Cousins 1975; reproduced with permission from NEL).

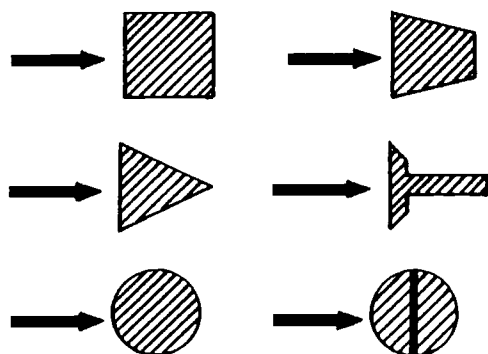


Figure 11.7. Shedder shapes tested by El Wahed and Sproston (1991) (reproduced with permission of Elsevier Science Ltd.).

- the cylinder with a slit was slightly better than the solid cylinder (cf. Igarashi 1986);
- the rectangular shedder produced a high signal-to-noise ratio suggesting that the body produced a large drag, which in turn encouraged the vortex to sweep across the rear face, reinforcing and stabilizing the shedding; and
- the T-shaped shedder was the best, suggesting that the tail helps control the shedding and produces a high signal-to-noise ratio, which is approximately constant throughout the entire range.

Figure 11.8 shows the cross-section of various bluff body shapes that have been used in flowmeters. Bentley and Nichols (1990) suggested that, for a thin plate, the vortex field is much farther downstream than for a square. The shape in Figure 11.8(g) may result in a considerable amplification of the vortex strength and hence of the pressure variation.

There appear to be mixed views on the use of composite shapes (cf. Majumdar and Gulek 1981 and Herzl 1982). The conditions for optimum shedding from dual bodies (Bentley and Benson 1993) relate to

- the thickness-to-width ratio of the two bodies;
- the spacing of the bodies; and
- the positions of maximum vorticity.

For dual bluff bodies, the movement of the boundary layer through the gap is essential for vortex enhancement. There are two options (Bentley et al. 1996) for position of a second bluff body: either the second body is far enough away for there to be shedding from both, or the second is less than a critical spacing, which

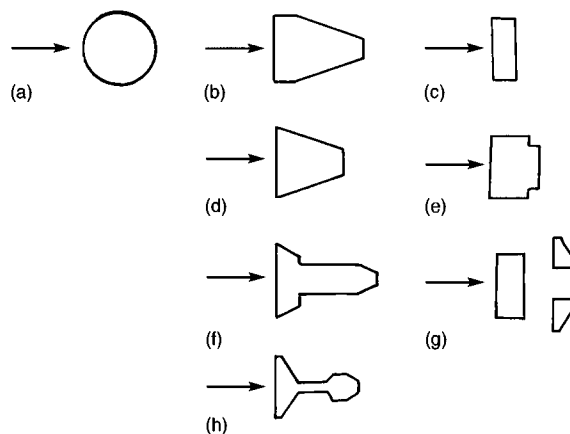


Figure 11.8. Diagrams of cross-section of bluff bodies (some after Yamasaki 1993 and some from manufacturers' brochures).

leads to shedding taking place downstream of both, with greater shedding stability. Bentley et al. identified certain spacing, which appears to be preferable, but the small size of the spacing between bodies may present a problem in small meter sizes.

It is interesting to note that the dual bluff body is essentially providing a feedback path between the two bodies of fixed dimensions and so, presumably, defining the volume needed to fill the feedback path and the time in a more stable way, hence the greater stability. Tests by Bentley and co-workers suggested that this is still further enhanced by introducing a throat in the gap, by making the downstream body a double wedge with the vertex pointing upstream.

Some manufacturers' designs allow the replacement of the sensor without closing down the line. One that uses a dual bluff body (essentially a triangular shape followed by a piezoelectric sensor mounted outside the flow line on a symmetrical wing airfoil that senses the downstream pressure variations) provides for the piezo crystal's accessibility during operation.

Turner et al. (1993) proposed a new body shape (cf. Zanker and Cousins 1975) for vortex-shedding flowmeters, consisting of a cylinder split across its lateral diameter, with a concave rear surface. They claim a high signal-to-noise ratio and an almost constant Strouhal number over a wide Reynolds number range. Miao et al. (1993) described a T-shaped vortex shedder consisting of a trapezoidal bar of fixed shape, with a rear plate whose length can be varied. The optimum length of plate was found to be in the range 1.56–2.0 times the width of the vortex shedder. They obtained a turndown ratio of about 17 : 1.

One manufacturer (at least) offers a triple body. The outer bodies (cylinders) create the vortex street, and the center body senses the vortices. It should give similar performance for flow in both directions.

11.3.3 STANDARDIZATION OF BLUFF BODY SHAPE

Takamoto and Terao (1994) have taken a basic design of bluff body of the truncated triangle type (Figure 11.8b). Figure 11.9 shows the essential dimensions, and the paper sets out the tolerance on these and the tolerance with which the body should be set. Thus, a selection of the dimensions and tolerances (rounded to reduce significant figures in some cases) are given in Table 11.1. The alignment refers to the orientation of the bluff body relative to the axis of the pipe. The perpendicularity is relative to the pipe diameter. This is a very interesting development, which appears to stem

Table 11.1. Selection of dimensions and approximate permitted error for a standard flowmeter^a

Dimension	w/D	H/D	h/D	L/D	θ	Alignment	Perpendicularity
Value	0.28	0.35	0.03	0.912	19°	0°	0°
Maximum error	0.1%	0.7%	6.6%	0.15%	0.4°	0.5°	0.3°
Effect on K-factor (%)	0.13	0.09	0.13	0.08	0.05	0.05	0.06

^a Refer to the original paper by Takamoto and Terao (1994) for precise values.

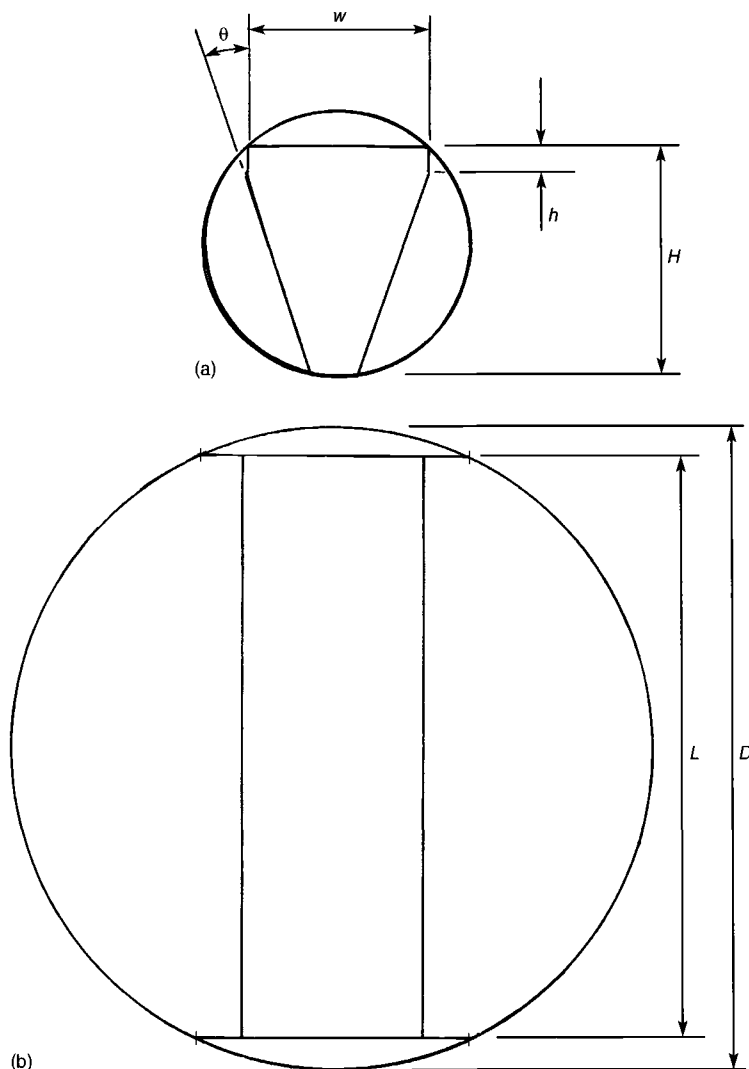


Figure 11.9. Diagram to show the main dimensions for a proposed standard (after Takamoto and Terao 1994): **(a)** Cross-section of bluff body; **(b)** Cross-section of pipe.

mainly from Japan. It is interesting to note that the flow area is close to that for an orifice of $\beta = 0.65$. In their tests, the value of H/D ranged from 0.20 to 0.40 for $w/D = 0.28$, and the Strouhal number varied by about 8% for this range. Their results also suggest that $H/D = 0.35$ results in a Strouhal number, which only varies by about 1% for a range of Reynolds number from 2×10^4 to 10^6 .

11.3.4 SENSING OPTIONS

A major limitation of the meter is the transducer; consequently, various sensing methods have been tried, and all have advantages and disadvantages. Some of these methods include:

- ❑ Ports in the sides of a bluff body as in Figure 11.8(a, b) leading to a transverse duct, which allows movement of fluid to be sensed by an internal hot wire or a thermistor;
- ❑ Temperature variation to sense changes by attaching thermistors to the upstream face of a bluff body such as that in Figure 11.8(b) or Figure 11.8(d);
- ❑ Pressure on lateral (electrical capacitance) diaphragms that deflect [Figure 11.8(e)];
- ❑ Pressure on a small diaphragm in the side of the bluff body [Figure 11.8(f)];
- ❑ Pressure sensed through ports in the sides of the bluff body [as for Figures 11.8(b, d)], or downstream of the bluff body in the fluid or in a symmetrical wing airfoil. The pressure changes create very small bending of a sensitive element (at low flows the tip may be moving by as little as 3×10^{-8} m). This movement can either be measured by strain gauge or by capacitive methods.
- ❑ Flexure of whole or part of the bluff body [Figure 11.8(h)] measured with strain gauges;
- ❑ Flexure of the whole bluff body sensed by strain gauges in the end supports of the body;
- ❑ A shuttle ball in an internal cavity in the bluff body linked by ports to the fluid, with movement sensed by magnetic induction [Figure 11.8(b)]; and
- ❑ A beam of ultrasound, crossing the flow downstream of the bluff body, modulated by the vortex shedding.

Pressure is the most common variable to be sensed, evidenced by: ports in the bluff body, pressure sensor built into the bluff body surface, flexible diaphragm, forces on the body, movement of a shuttle in the body. Thermal sensors are also used, built into the body and affected by fluid flows. Fluids should be clean and noncorrosive.

A new design of meter using ultrasonic sensing appears to operate successfully. Optical sensing may offer a further alternative (e.g., in the form of optical fibers in the bluff body).

Takahashi and Itoh (1993) described a design that used the alternating lift on the shedder, which is sensed through two piezoelectric elements [Figure 11.10(a)]. By using two sensors, effects due to vibration of the pipeline are reduced. A stable zero is created by using the strength of the vortex-shedding signal (proportional to ρV^2) and creating a low flow cutoff when this does not tally with the value of V from the vortex-shedding frequency. Another arrangement is shown in Figure 11.10(b); here the vortices lead to a small flexural movement of a section of the shedder bar, and a piezoelectric element inside the sensor senses this minute movement. Figure 11.10(c) shows the position of a sensor bar, which can be inserted into or placed downstream of the shedder body.

Herzog (1992) used optical fibers within the bluff body of a vortex-shedding flowmeter to sense the frequency. One fiber appears to have been subject to the transverse forces through ports in the body, whereas the other was totally enclosed. The bending of the first fiber caused changes to the quality and intensity of the light, and both speckle pattern and intensity were used to obtain the frequency (cf. Hisham Marshad and Irvine Halliday 1994 and also Wen Dong-Xu 1990).

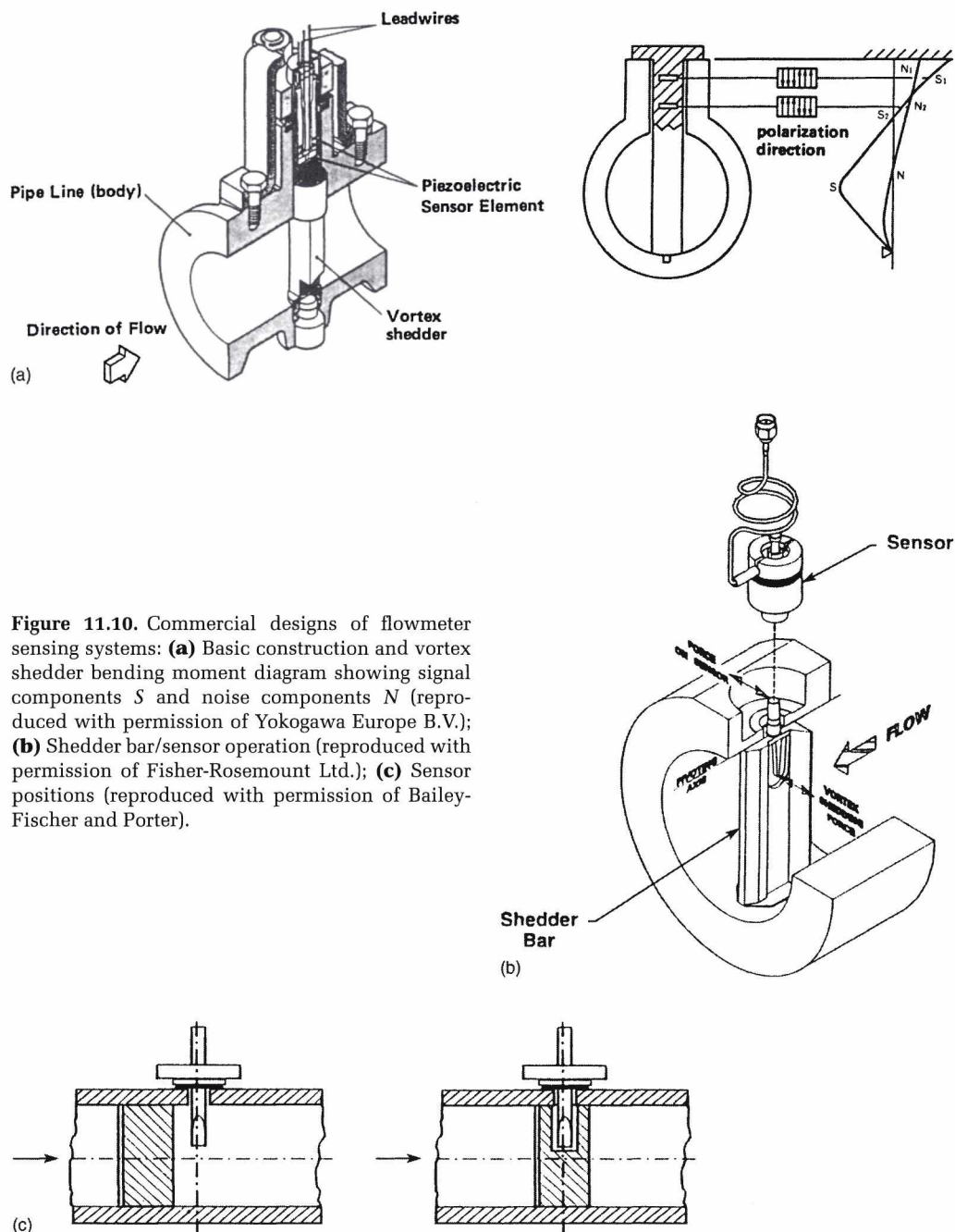


Figure 11.10. Commercial designs of flowmeter sensing systems: **(a)** Basic construction and vortex shedder bending moment diagram showing signal components S and noise components N (reproduced with permission of Yokogawa Europe B.V.); **(b)** Shedder bar/sensor operation (reproduced with permission of Fisher-Rosemount Ltd.); **(c)** Sensor positions (reproduced with permission of Bailey-Fischer and Porter).

Sproston et al. (1990) experimented with electrostatic sensing, and in later tests El Wahed and Sproston (1991) used electrostatic sensors as shown in Figure 11.11, either in the walls, behind the body, or on the front face of the body. The mechanism, on which this sensing is based, is that charge is produced in flows (flow-induced streaming currents), and so its variation can be used to measure flow variation. The experiments were in air.

Piezoelectric elements may be sensitive to pipe vibration and degrade the signal-to-noise ratio. Stiffening the meter body, using filtering and a low cutoff (Kawano et al. 1992) may partially overcome these problems.

Figure 11.12 shows a block diagram for a signal converter.

11.3.5 CROSS CORRELATION AND SIGNAL INTERROGATION METHODS

Coulthard and Yan (1993a) suggested the use of the vortex wake as a tracer for use with cross-correlation techniques. The bluff bodies that were used presented a low blockage to flow and appeared to give similar accuracy to those of vortex-shedding frequency measurement. This presumably indicates that the cross-correlation technique is capable of the same discrimination as the frequency measurement electronics. Coulthard and Yan (1993b) investigated the effect of various bluff body shapes but found little difference between them (however, cf. Terao et al. 1993).

Amadi-Echendu et al. (1993) described a further piece of work to analyze the raw signal from a flowmeter, in this case a vortex

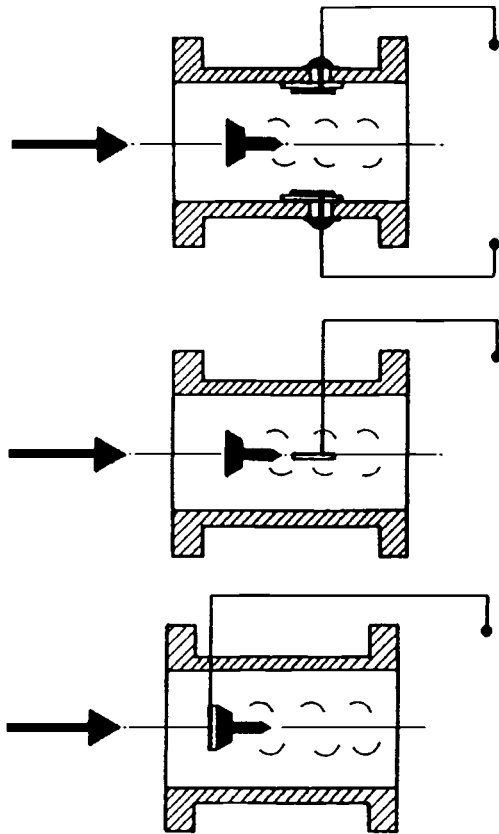


Figure 11.11. Electrostatic sensor positions (El Wahed and Sproston 1991; reproduced with permission of Elsevier Science Ltd.).

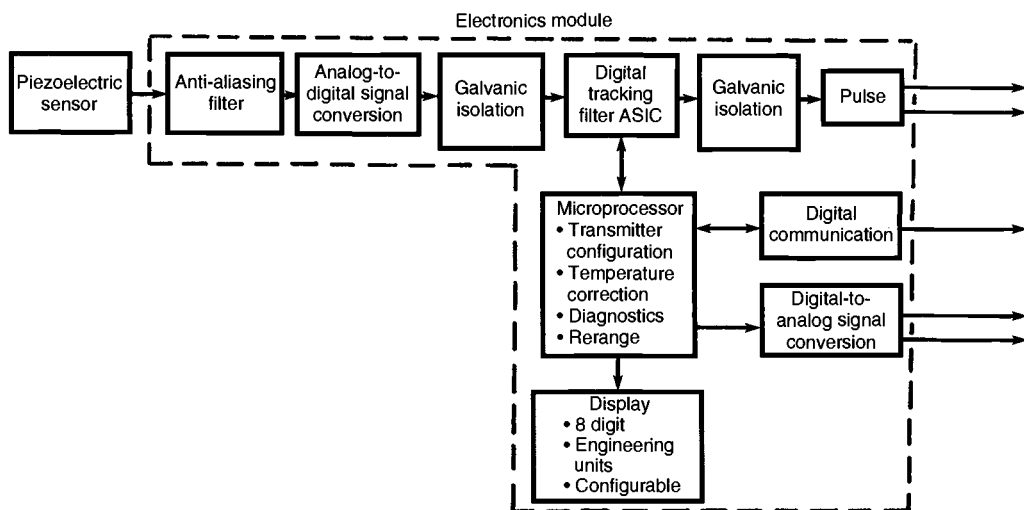


Figure 11.12. Block diagram of electronics module for commercial flowmeter (after Fisher-Rosemount Ltd.).

flowmeter, to ascertain information beyond pure flow information. It is not clear from this paper the extent to which this will be possible. Clearly condition monitoring from this source would be very valuable. However, there is a need to link the frequency data as closely as possible to the physical effects so that the changes can be intelligently used (cf. Section 10.2.16 and a note in Chapter 21).

11.3.6 OTHER ASPECTS RELATING TO DESIGN AND MANUFACTURE

Itoh and Ohki (1993) claimed (as does one manufacturer) a vortex flowmeter design, which obtains mass flow rate by using the lift on the central body divided by the frequency of pulsation. This depends on the assumption that the lift coefficient and the Strouhal number are both constant so that the lift is proportional to ρV^2 , and the frequency is proportional to V . As a result, the ratio is proportional to ρV (cf. Yamasaki 1993).

Miller et al. (1977) gave some impressively consistent K factors from the production process showing random uncertainty in K of order $\pm 0.8\%$ for 50-mm meters, $\pm 0.4\%$ for 75-mm meters, and $\pm 0.3\%$ for 100-mm meters.

11.3.7 ACCURACY

Typical performances of meters currently available are given as: repeatability, about $\pm 0.2\%$ rate; linearity, better than $\pm 1\%$ for a turndown ratio of up to 40:1 for liquids and up to 30:1 for gases. For liquids at higher flow rates, the uncertainty appears to be in the range 0.5–1% of rate. At low flow rates, it is the same range of values but as a percentage of full-scale deflection. The change will be at about $Re = 20,000$, with a minimum value of Re of about 4,000 and upper limit of about 7×10^6 . The type of transducer may affect the low flow limit.

For gases and steam, the uncertainty appears to be in the range 1–1.5% of rate, but with $Re < 20,000$ likely to increase uncertainty.

Temperature variation is likely to affect precision, and compensation may be included (e.g., 0.3%/50°C).

Cousins has commented that it is necessary to collect a large number of pulses to achieve good repeatability due to variation of 10–20% in the shedding period. As a consequence, long integration times should be expected.

11.3.8 INSTALLATION EFFECTS

Table 11.2 shows the change in error for increasing installation lengths (Takamoto et al. 1993b). Some of the results suggest that the error does not always decrease with spacing. The worst case has been used, from the results of four meters.

Mottram (1991) commented that “The difficulties of deciding what should be the minimum straight pipe length between a given type of flowmeter and a particular upstream pipe fitting are notorious. Even when the geometry is standardized, as is the case for the sharp edged orifice, the issue is still controversial. This is due to the typical shape of the envelope enclosing the experimental results for the calibration shift plotted against the straight pipe length separating the meter from the upstream fitting. Minimum straight lengths may differ by a factor of two.” Mottram’s tests were for a vortex meter with an area ratio approximately equivalent to a $\beta = 0.6$ orifice

Table 11.2. Results from Takamoto et al. (1993b) that appear to give the worst case uncertainty (rounded up to the first decimal place) at various spacings

	Bend 90° (%)	2x Bend (%)		Reducer (%)	Expander (%)	Gate Valve Open (%)	
		Same Plane	90° Plane			100%	50%
5D	1.9	2.1	2.1	0.9	4.2	0.3	5.1
10D	0.6	0.7	1.1	0.7	1.4	0.4	1.5
20D	0.4	0.4	0.8	0.3	0.3	0.4	0.5
30D	0.3	0.2	0.6	0.3	0.3	0.3	0.4

area ratio. We compare in Table 11.3 Mottram's and Takamoto et al.'s (1993b) values for the vortex installation lengths and those from ISO 5167 for orifice meters with $\beta = 0.6$ and $\beta = 0.75$.

Mottram gave a general rule that vortex sensitivity was similar to that for a high beta ratio orifice plate. This would appear to be a reasonable working rule of thumb from his results and is in close agreement with Ginesi and Annarummo (1994) who suggested using the 0.7 beta ratio lengths. However, Mottram's comment that minimum lengths may differ by a factor of two is borne out by Takamoto et al.'s (1993b) results.

Further results from Mottram and Rawat (1988) (using a rectangular bluff body and hot-wire anemometer probe to detect the vortices) suggested that pipe wall roughness, presumably resulting in profile changes, can cause 3% variation in calibration (cf. Witlin 1979).

If we compare manufacturers' requirements, they are fairly consistent in suggesting 5D downstream although at least one suggests 10D, but for upstream spacing,

Table 11.3. Installation lengths (diameters) between upstream fitting and meter for changes within $\pm 0.5\%$ compared with spacing for orifice

	2x Bend			Reducer	Expander	Gate Valve Open	
	Bend 90°	Same Plane	90° Plane			100%	50%
Orifice – zero additional uncertainty							
$\beta = 0.6$	18	26	48	9	22	14	—
Vortex ^a	55		60				30
Vortex ^b	13	12	43	14	16	5	16
Orifice – ± 0.5% additional uncertainty							
$\beta = 0.75$	18	21	35	11	19	12	—
Orifice – zero additional uncertainty							
$\beta = 0.75$	36	42	70	22	38	24	—

^a Mottram's (1991) meter with blockage equivalent to $\beta = 0.6$.

^b Takamoto et al.'s (1993b).

Table 11.4. Installation requirements (in diameters) as indicated in manufacturers' literature compared with orifice plate requirements

	Orifice $\beta = 0.6$ ($\pm 0\%$)	Vortex	Orifice $\beta = 0.75$ ($\pm 0.5\%$)
90° bend or T	18	10–20	18
2 bends			
in same plane	26	20–25	21
in perpendicular planes	48	30–40	35
Reducer	9	10–20	11
Expander	22	10–40	19
Valve ^a	26	20–50	18
Downstream spacing	7	5–10	4

^a Globe valve fully open.

they give values as in Table 11.4. These are, again, compared with orifice spacings, for $\beta = 0.6$ ratio orifice (zero additional uncertainty) and $\beta = 0.75$ ratio orifice ($\pm 0.5\%$ additional uncertainty).

Apart from the valve, the values in Table 11.4 suggest that to take the greater of the spacings for a $\beta = 0.6$ orifice with zero additional uncertainty and for a $\beta = 0.75$ orifice with $\pm 0.5\%$ additional uncertainty might provide a rule of thumb for other fittings. There appears to be a more severe effect due to reducers and expanders, about which some manufacturers are clearly cautious. Further, although this is in line with Takamoto et al. (1993b), it does not cover the more pessimistic results of Mottram (1991), and for safety we may need to accept his guidance that the installation distance should be that for zero additional uncertainty with $\beta = 0.75$ (or 0.8), accepting Mottram's factor-of-two observation.

Manufacturers have warned against the protrusion of gaskets into the flow upstream and adjacent pipe bores of different size to the meter. The inlet pipe should be free of weld beads and thermometer pockets and should be smooth; the pipework should be aligned with the meter. Takahashi and Itoh (1993) suggested $\frac{1}{4}\%$ error for installation in a pipe of different bore to the meter, and one manufacturer gives the effect as less than 0.6% for a step of 4–10% greater than the flowmeter ID resulting from the pipe schedule.

Cousins (1977) commented on the severe effect of swirl in changing the calibration and even in suppressing the shedding and suggested at least 50D upstream spacing from the source of the swirl for a swirl angle of 10° and more spacing for larger angles. However, Laneville et al. (1993) showed that, for swirl indices ($\Omega =$ injected swirling flow rate/total flow rate) less than 0.3, the vortex-shedding signal quality from a vortex flowmeter with trapezoidal body is equivalent to the zero swirl case, and the frequency is still within about 2% of the swirl free value for a spacing 18D downstream of the swirl generator. These apparently differing results will call for caution.

Pressure taps downstream should be 2D to 7D, and temperature taps should be 1D to 2D downstream of the pressure ones. A flow straightener of 2D length

separated by 2D from the disturbance should have 5D to the meter. Mottram (1991) obtained good results using a Mitsubishi conditioner with a different spacing. For a bend, 90° offset bends, orifice plate with $\beta = 0.645$, and fully open globe valve, his results were within a band of $\pm 1\%$ with 5D between the fitting and the conditioner, 2.5D between the conditioner and bluff body, and 2.5D downstream of the bluff body.

Other mounting constraints are not usual, but pipework must ensure that the meter runs full.

11.3.9 EFFECT OF PULSATION AND PIPELINE VIBRATION

It is known that vortex meters may be affected by pulsatile flow, particularly where the pulsating frequency or its harmonics are close to the shedding frequency. The meter may lock on to the pulsation frequency and give an error (cf. Amadi-Echendu and Zhu 1992). Some sensing techniques may be more susceptible to error due to pulsation than others. Some may be designed to select the shedding frequency by eliminating pulsation or vibration using a push-pull arrangement.

Hebrard et al. (1992) confirmed that pulsations can cause large metering errors. Pulsations of an amplitude of only a few percent may be sufficient to produce metering errors, particularly when the frequency is near the fundamental or first harmonic of the shedding frequency. The lock-on frequency is most likely to occur when the pulsation frequency is twice the vortex-shedding frequency, and lock-on can occur (Al-Asmi and Castro 1992) even down to pulsation amplitudes of about 2%. Outside the lock-on range, there can still be significant differences between the shedding frequency in steady and pulsating flows, and free stream turbulence may increase the problem.

Manufacturers may suggest that disturbances upstream of an unsteady nature should be given a wide berth. Malard et al. (1991) found that there was an increase of about 2% and a decrease in the signal-to-noise ratio when the turbulence intensity increased from 3.3 to 10.3%. Their work also confirmed the errors due to pulsatile flows.

As indicated earlier, sensors are designed to minimize the effect of pipeline vibration, and the manufacturer should be consulted for information. However, process noise may cause zero flow signals but may be diagnosed by looking at the raw signal (Ginesi and Annarummo 1994).

11.3.10 TWO-PHASE FLOWS

Hussein and Owen (1991) found that the correction factor for the vortex meter for wet steam of high dryness factor is closest to the value $x^{-1/2}$ (Figure 11.13). BS 3812:1964 is quite close. Pressure was shown to have little effect on correction factors. Hussein and Owen (1991) gave the wise advice that an upstream separator is advisable, and that there are several types giving a value of $x > 95\%$ with reasonably low pressure loss.

The possibility that the meter might be developed to sense the droplets of wet steam is an interesting challenge for the signal processing experts.

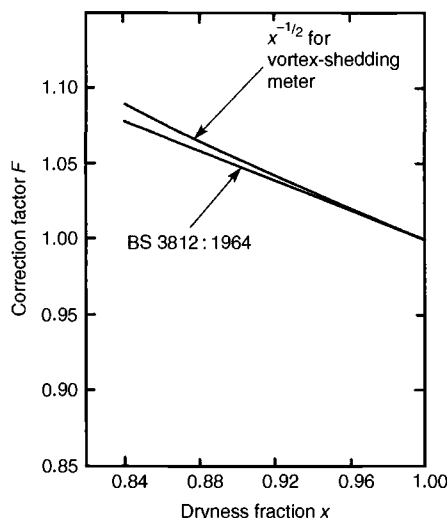


Figure 11.13. Comparison between wet steam correction factors for vortex meters (after Hussein and Owen 1991).

The meter has virtually the same calibration for liquid and gas flows and might therefore be expected to behave well in two-component flows. However the published work highlights some severe problems.

The first is that the components are separated by the vortex motion. Hulin et al. (1982, 1983) found that shedding in vertical water-air flow was stable up to about 10% void fraction. They also found that the vortices trap bubbles [Figure 2.9(b)] with an associated decrease in the strength of the pressure fluctuations. Their later work on water-oil flows shows some contrasts. The separation is much more complete for water-air due presumably to the density difference, and this leads to a much narrower vortex emission bandwidth. The stability at oil concentrations greater than

30% was notable, and the cores of the vortices were continuous oil.

The second is that the bubbles may trigger certain sensors. Baker and Deacon (1983) obtained results suggesting that, even though vortex-shedding continued with increase in void fraction in a vertical upward water-air flow, bubble impact may cause additional pulses. Above air volume concentrations of 1%, the meter error increased by more than 1% for each additional 1% air concentration.

A third problem, to which Washington (1989) attributed poor results in wet gas, may be slip between the components. A homogeneous second component in the flow may be measured but will usually result in a loss of accuracy due, in part, to the fact that the instrument measures the volumetric flow.

11.3.11 SIZE AND PERFORMANCE RANGES AND MATERIALS IN INDUSTRIAL DESIGNS

As indicated earlier, the low end is limited by Reynolds number behavior, and the effect of Equation (11.1) is that, as the size increases, the frequency becomes very low and makes the meter unusable.

Diameter ranges from about 12 to 200 mm or more are quoted by various manufacturers. Flow rates through the meters are up to about 9 m/s for liquids and 75 m/s for steam and gas. Flow ranges for liquids may be from a minimum of less than 1 l/s to a maximum of about 600 l/s and for gases 15 l/s or less to 3,000 l/s or more.

Range is of order 20:1 on gas and steam and 10:1 for liquids (up to 60:1 is sometimes given). This creates a constraint with the minimum Re , which may inhibit its application where the precise range is unknown or spans two meter sizes. In others, a smaller-sized meter may be necessary to cope with the range. Overranging up to 20% is generally allowable, but the low flow end may be more due to the strength of the signal.

Minimum Reynolds number is variously given as 3,800–5,000, although meters may be of lower accuracy below 10,000 and upper limits of 500,000 and up to or greater than 1,000,000 may be given.

Viscosity should be below 8 cP (the viscosity of cooking oils), although it may be possible to measure with reduced range up to 30 cP (Ginesi and Annarummo 1994). A low flow cutoff is usually set somewhat below $Re = 10,000$, and this may be a disadvantage for some applications.

The temperature range may be from less than -200° to over 400°C . Pressure may be up to 150 bar or more depending on the transducer. Pressure drop on a meter of nominal internal diameter equal to the pipe is normally less than about 40 kPa (6 psi) on water flow.

Pulse rates will vary enormously depending on the size of the meter. For a small meter of about 20 mm diameter, the rate will be of order 200 pulses/l, whereas for a large meter of 300 mm diameter the rate will be of order 0.02 pulses/l. Response time may be of order 0.2 s or three shedding cycles for a change to 63% of input. All this may be affected by resolution problems when combined with poor period repeatability.

Material of the bluff body is most commonly stainless steel but may be titanium or another material.

Vortex meters, like turbine meters, are limited at the upper end of their range by possible cavitation. Casperson (1975) suggested the possibility of hysteresis effects due to cavitation and the need to avoid operating in this regime. To avoid cavitation in liquid flows, some manufacturers give a formula of the following type for the minimum back pressure 5D downstream:

$$p_{\text{gmin}} = A\Delta p + Bp_v - p_{\text{atmos}}$$

where Δp is the pressure drop across the meter, p_v is the saturated liquid vapor pressure, and p_{atmos} is the atmospheric pressure. A appears to have a value of about 3.0, and B has a value of about 1.3. Δp is of order 1 bar at 10 m/s.

These meters may, also, be subject to compressibility effects.

11.3.12 COMPUTATION OF FLOW AROUND BLUFF BODIES

Some early work obtained good agreement between numerical and experimental results for Re up to 1,000 for confined two-dimensional flows around rectangular cylinders (Davis et al. 1984). More recent work by Matsunaga et al. (1990) showed some impressive computer-generated flow patterns around a trapezoidal body in a two-dimensional flow. Comparison of Strouhal number obtained computationally and experimentally agreed quite well at $Re > 1,000$ but appear to diverge for lower Re and were less good for Re of order 10,000. Figure 11.14 shows the vortex-shedding mechanism as interpreted by the authors.

Johnson (1990) computed the unsteady laminar flow around a cylinder and obtained agreement within 10% for the Strouhal number in comparison with experimental values for part of the range. Computer simulations by El Wahed et al. (1993) again obtained good agreement with the experimentally acquired Strouhal number and indicated that the T-shaped shedder produced the largest vortex.

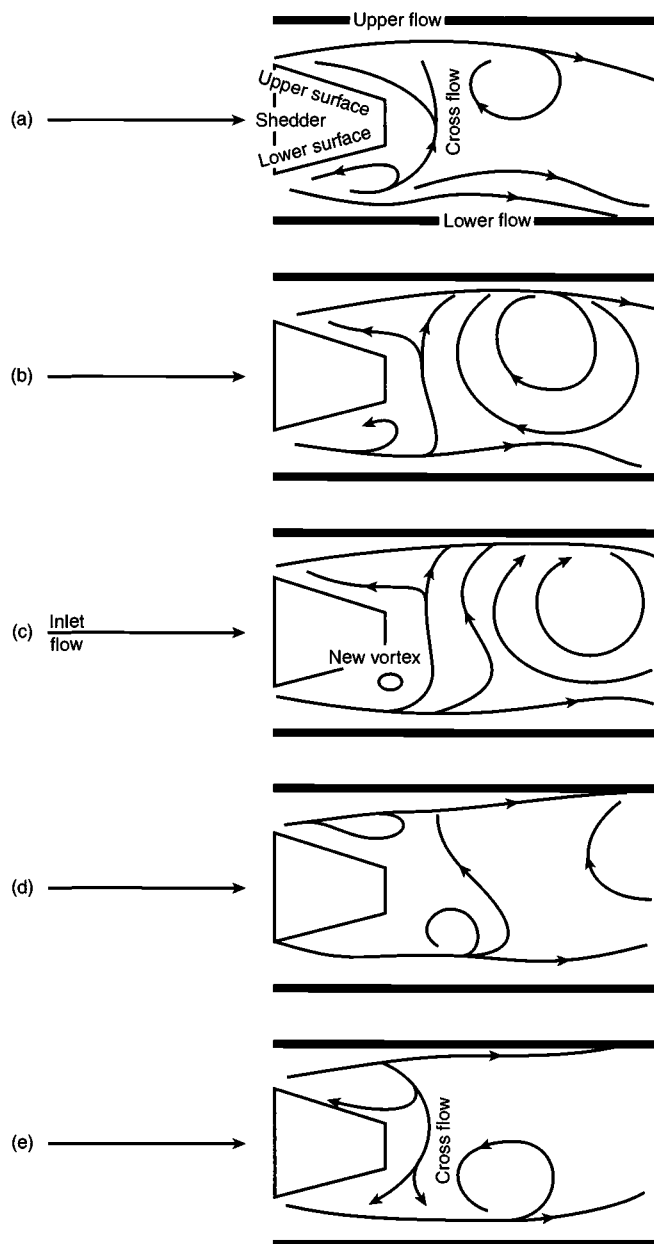


Figure 11.14. Flow pattern behind shedder: **(a)** $t = 1.2615$ s; **(b)** $t = 1.2660$ s; **(c)** $t = 1.2690$ s; **(d)** $t = 1.2735$ s; **(e)** $t = 1.2765$ s (after Matsunaga et al. 1990; reproduced with permission of Elsevier Science Ltd.).

11.3.13 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

Furness and Jelffs (1991) sought to make a case for using vortexmeters to measure flows in and out of refineries rather than depending on tank transfer. Proving, however, may be problematic due to the poor period repeatability.

Siegwarth (1989) discussed tests on specially designed vortex-shedding meters for application to very high flow rates of liquid oxygen in the space shuttle main engine ducts. The details of this work may be of interest to those engaged in vortex flowmeter design.

The claims of manufacturers for applications include

- ❑ liquid flows (e.g., liquid N₂, CO₂, O₂, clean liquids, distilled water, glycol, some acids, low viscosity hydrocarbons, benzene, diesel, hydraulic oils, creosote, and tar),
- ❑ gas flows [e.g., steam (superheated and saturated) and various gases including compressed air, methane, N₂, and CO₂], and
- ❑ cryogenic applications, but not multiphase applications.

In general Yamasaki (1993) saw the meter as an alternative to the orifice meter, provided it could cover a similar range of fluid parameters. (One manufacturer claims that costs are about 50% of those for orifice plates.) The sensing method is the main constraint to extending parametric range. However, temperature range now allows its use with superheated steam and liquid natural gas (LNG).

11.3.14 FUTURE DEVELOPMENTS

There is a need to extend the range of the vortex meter, to improve application data (e.g., installation effects), and to develop the signal processing.

Zanker and Cousins in their work as long ago as 1975 foresaw the dual bluff body, and, no doubt, there is further development possible in the design of these bodies. They also mentioned annular ring/coaxially mounted bluff bodies, and Takamoto and Komiya (1981) also proposed such a ring-shaped bluff body that sheds ring vortices.

Miau and Hsu (1992) surveyed the flow around axisymmetric disks and rings as vortex shedders, showed the pattern of vortex loops behind a disk, and were particularly interested in the flow characteristics near the pipe wall, suggesting that an advantage of the axisymmetric designs is that the sensing can be on the pipe wall. Axisymmetric vortex-shedding rings intuitively offer an elegant axisymmetric version of the meter. Tai et al. (1993) discussed optimum size and positioning and signal-to-noise ratio. Cousins and Hayward (1993, cf. Cousins et al. 1989) described developments of the vortex ring flowmeter and discussed sensing options. They made the point that ring vortices are far more stable than those from straight bluff bodies. They suggested that a ring/pipe diameter ratio of 0.4 is about optimum. They found that the signal-to-noise ratio varies markedly across the pipe and down the centerline, peaking at 3 to 4 ring widths downstream. They found that the inner ring vortices give a clearer signal. The provisional design, which appears to be optimal is a double ring with a tail; it appears to give a repeatability of $\pm 0.1\%$ or better with only 0.3 velocity head pressure loss.

The combination of electrostatic sensing as described by El Wahed and Sproston (1991) with ring vortex shedders could be very attractive. The sensor could be in the wall of the tube or even at various points to give greater intelligence to the meter and forms of self-monitoring.

11.4 SWIRL METER – INDUSTRIAL DESIGN

This section is concerned with the industrial design of the swirl meter, which has some fluid behavioral similarities to the vortex whistle (Chanaud 1965). In the whistle, initial swirl is created by tangential flow into an upstream plenum from which the helical vortex emerges.

11.4.1 DESIGN AND OPERATION

A diagram of this type of flowmeter is shown in Figure 11.15. As the flow enters the meter, inlet guide vanes cause it to swirl. As the flow moves through the contraction, the angular momentum created by the guide vanes is largely conserved, and so the angular velocity will increase. The vortex filament develops into a helical vortex, as shown in the diagram, which moves to the outside of the tube. The frequency with which the helix passes the sensor provides a measure of the flow rate. The flowmeter section then expands, and the swirling jet at the wall surrounds a region of reverse flow on the axis. This motion is removed before the fluid leaves the meter by means of a deswirler. Dijstelbergen (1970) explained the precession as being due to the increasing pressure in the diffusing section causing a force on the exiting swirl, which is converted to a lateral momentum change. Ricken (1989) gave further details of a model to describe the behavior of the meter.

Heinrichs (1991) described a sensing system for a swirl meter in which the rotation is sensed by using two diametrically opposed pressure ports. In the measuring tube of the swirl meter, a helix-shaped vortex appears, rotating with a frequency proportional to the volumetric flow rate. In this way, the differential pressure eliminates common mode signals and hence the problem of a high line pressure and flow

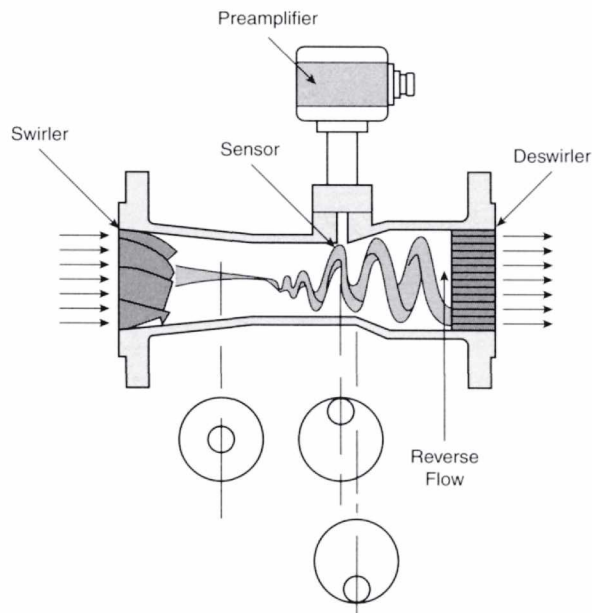


Figure 11.15. Swirl flowmeter (after Bailey-Fischer and Porter with their permission).

fluctuations. The push-pull arrangement suggests that the range may be extended. Capacitive sensors seem to be less suitable, whereas the results for piezoelectric ones are more promising.

11.4.2 ACCURACY AND RANGES

An industrial design has been available for liquids, gases, or steam with vortex rotation frequency obtained using piezosensors. Measurement uncertainty of better than 1% of rate is claimed by the manufacturer over the upper 80% of the range for the smaller sizes, with better than 2% for the 10–20% part of the range. For sizes above 32 mm, uncertainty of better than 1% of rate is possible over the upper 90% of the range.

Flow ranges are, for liquids, from 0.2 to 2 m³/h up to 180 to 1800 m³/h and, for gases, from 5 to 25 m³/h up to 1000 to 20,000 m³/h. Values for steam should be obtained from the manufacturer. Turndown for some sizes for liquids is up to about 30:1, and for gases it is between 5:1 and 20:1. Maximum frequencies range for liquids from 100 Hz for a 20-mm-diameter meter to 13 Hz for a 400-mm meter and for gases from 1200 Hz for meters of 20 mm down to 150 Hz for a 400 mm diameter meter. A 10% overranging is allowed in gases, but in liquids cavitation must be avoided. Maximum velocity is 6 m/s for liquids and 50 m/s for gases.

For liquids at the top of the flow range, pressure loss can range from about 200 to 400 mbar or more, and for gases the range is from 25 to 70 mbar or more. The flowmeter is available in designs suitable for hazardous area use.

11.4.3 MATERIALS

The material in contact with the liquid is predominantly stainless steel, except for the thermistor sensor in gases, which is glass-insulated and epoxy-potted, and the piezosensor, which is stainless steel or some other material.

11.4.4 INSTALLATION EFFECTS

For bends, contractions, and valves upstream, the recommendation is for 3D straight pipe before the inlet flange; for bends and expansions downstream, it is 1D. However, a contraction downstream is not permitted. For large radius bends, etc., the requirements may be further reduced.

The manufacturer appears to claim low sensitivity to fluid pressure, temperature, and density. Fluid temperature is –40 to 280°C.

Maximum viscosity is 70 mPas. Deposits can cause errors, but it may also tend to self-clean. Cavitation must be avoided with liquids by ensuring adequate static pressure.

11.4.5 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

The meter is recommended by the manufacturer for applications where absolute reliability with minimum maintenance is essential and is claimed to be suitable for wet or dirty gases and liquids. Specific applications are water, sludge water, condensate, acids, solvents, petrochemical raw materials, gasoline, air, CO₂, natural gas, ethylene, and steam.

11.5 FLUIDIC FLOWMETER

The third instability we consider is that for the fluidic flowmeter. A diagram of a typical design is shown in Figure 11.16. The fluidic flowmeter uses the Coanda effect. If a two-dimensional diffuser has too wide an angle, the flow will separate and attach to one wall of the diffuser. It may attach to either wall and can be displaced from one to the other by a suitable disturbance. The fluidic flowmeter introduces such a disturbance, which causes the flow to oscillate between the diffuser walls at a frequency proportional to the velocity of flow. There are several parallels with the vortex meter. Types of sensors used in vortex flowmeters could be applied to this device. There is likely to be room for design optimization. The flow range may be wide, and certainly the device looks promising for low flows.

The mechanism by which the flowmeter operates is that the flow, having attached to one wall, passes into a feedback channel. The fluid when it exits from this channel knocks the main jet to the other side of the diffuser, and the process starts again. Thus the period is related to the time for the flow to move from the entry of the flowmeter to the entry of the feedback channel and communicate, via that channel, to the entry. This time will be inversely proportional to the velocity of flow, and so the oscillating frequency will be proportional to the flow rate.

11.5.1 DESIGN

Boucher and Mazharoglu (1988) developed one design of fluidic target meter, which operated down to Re of about 70 but with considerable change in Strouhal number at low Re . Hysteresis may have a small effect in starting and stopping (cf. Boucher 1995).

Sanderson (1994) described a new fluidic water meter using electromagnetic sensing. This is a particularly appropriate method with a wide dynamic range and a linear response to flow rate. It is also usable in a constant magnetic field design due to the fluctuating nature of the flow. The sensing electrodes were made of stainless steel, and earthing electrodes were also provided. The magnetic materials considered for this device included sintered ferrite and bonded and sintered neodymium. The general configuration is shown in Figure 11.17. Sanderson's preferred arrangement of magnetic sensor was for the measurement to take place in the main jet with

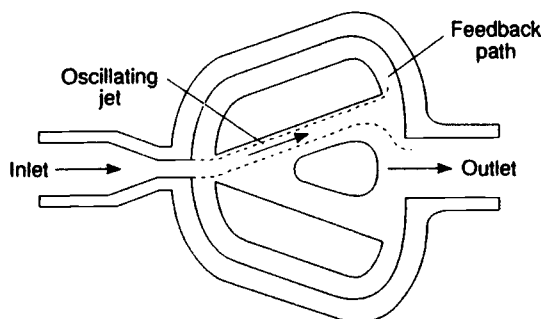


Figure 11.16. Diagram of a fluidic flowmeter (reproduced with permission of Professional Engineering Publishing).

magnetic material in the diffuser walls giving a field strength of about 0.3 T. The two pairs of electrodes were in the side walls acting 180° out of phase in a push-pull arrangement. Over most of its range, the meter was linear to within $\pm 2\%$ (from about 5 to 500 ml/s). Further linearization was possible within the electronic circuitry. The meter operated from a 3.6-V lithium battery with a 10-year lifetime.

Sakai et al. (1989) described a dual sensing system gas flowmeter for domestic use in Japan. It was designed to operate from 3,000 l/h down to 150 l/h with a piezoelectric differential pressure sensor and seems to have achieved $\pm 2.5\%$ for 150–600 l/h and $\pm 1.5\%$ above 600 l/h. For the range 3–150 l/h, an integrated circuit thermal sensor was used. This had a heated area with temperature sensor areas on each side of the integrated circuit chip. The characteristic of this sensor appears to have a varying coefficient, and it is not clear what final precision was claimed for low flows. Sakai et al. saw the prospect of using one lithium battery of 5,000 mAh, which would last 10 years in continuous service by lowering the consumption of the circuits.

Measurement accuracy of a preproduction gas meter on field tests in Japan is claimed (Yamasaki 1993) to be within $\pm 1.5\%$ of reading with a 30:1 turndown ratio. Upper flow limits are defined by the allowable loss in the meter. Computer simulation was used as an aid in the design.

Nishigaki et al. (1995) used LDA methods to obtain the flow patterns in a fluidic gas flowmeter and concluded that the patterns were almost the same across a Reynolds number range from 291 to 2,160.

The development of a meter for various Japanese gas companies (Aoki et al. 1996, Sato et al. 1996) has included numerical analysis. It has a more complicated array of internal bodies with a U-shaped initial target. A 60:1 turndown appeared to be achieved.

11.5.2 ACCURACY

Okabayashi and Yamasaki (1991) found that accuracy could be improved by

- i. optimizing the shapes of the side walls and the position of the target and
- ii. incorporating a flow straightener to make the jet two-dimensional.

They claimed a linearity of $\pm 0.8\%$ of reading within the range 0.15–5.0 m³/h with a repeatability of $\pm 0.2\%$ of reading. The frequency appeared to be about 20 Hz/m³/h. Sensing was by a piezoelectric sensor (using a polymer piezoelectric film) that does not consume power and an amplifier with very low current consumption.

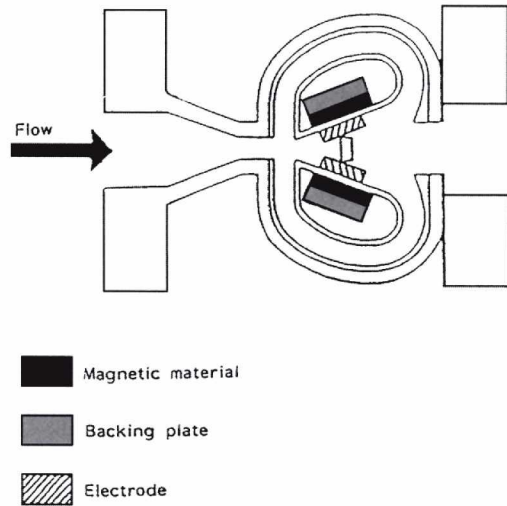


Figure 11.17. Fluidic flowmeter with magnetic sensing (Sanderson 1994; reproduced with permission of Elsevier Science Ltd.).

11.5.3 INSTALLATION EFFECTS

It seems likely that these devices will be virtually insensitive to upstream disturbance because the inlet duct within the meter will have changed section and will be equivalent to several diameters. There is also likely to be flow straightening.

11.5.4 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

The prime application of these devices is for utility flows where they meet requirements of

- small size,
- large turndown,
- low power, and
- long life without maintenance.

With no moving parts and compact size, these devices should show advantages over existing meters for utility (water and gas) flows. However, the sensing range will require capability for large turndown if the fluid turndown range is to be realized.

11.6 OTHER PROPOSED DESIGNS

A wake oscillation was observed by Mair (1965). With an axially symmetric and streamlined body followed by a disk, the wake gathers in the cavity and puffs out regularly. With a torpedo-shaped body of diameter D , a coaxial disk of diameter d , and spacing x downstream of the flat rear of the body, Mair found that with $x/D \sim 0.5$ and $w/D \sim 0.8$ the vortex was trapped between the body and the disk and resulted in a low drag, whereas for $x/D \sim 0.3$ and $w/D \sim 0.8$, a high drag and oscillation occurred, which he attributed to breathing of the cavity.

Parkinson (1991) described a combination of a venturi nozzle with a folded fluidic flowmeter (Figure 11.18). Boucher et al. (1991) also designed a folded version of the fluidic flowmeter to create a bypass for a venturi meter (cf. Wang et al. 1996). In this way, the fluidic principle can be applied to pipes in the range 40–150 mm with turndown ranges of 24:1 to 40:1. The sensing in the fluidic flowmeter was by means of a miniature air-flow sensor on a silicon chip based on thermal convection.

Shakouchi (1989) demonstrated an alternative geometry of fluidic meter where a rectangular jet issues into a larger space. The oscillating vane meter was discussed in Section 8.8.

11.7 CHAPTER CONCLUSIONS

The vortex meter offers a very attractive alternative to the orifice plate for some applications and gains over the orifice plate by its frequency output and larger turndown. Its application to dry saturated steam may prove to be one of the most important contributions of this device. It would be interesting to explore means of sensing, by impact or by entrainment in the shed vortices, the amount of water in the flow and thus to obtain a dryness factor.

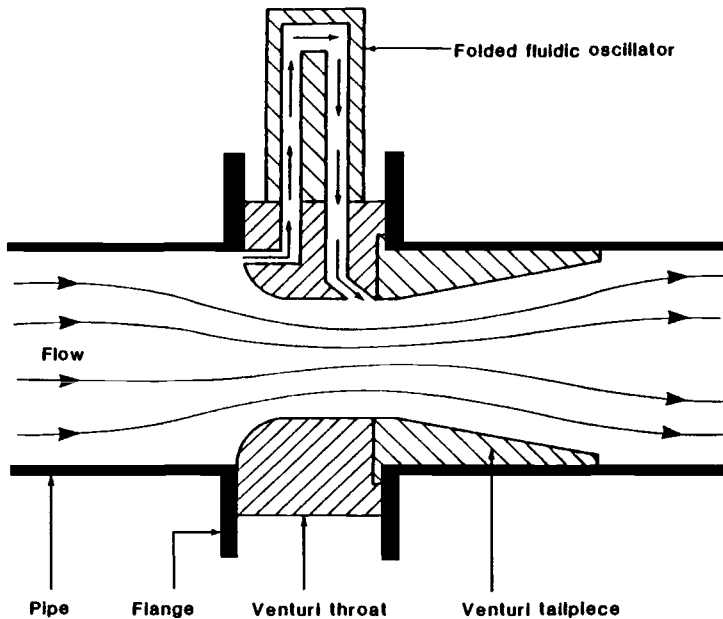


Figure 11.18. Combined venturi nozzle with a folded fluidic flowmeter (from Parkinson 1991; reproduced with permission of the Institute of Measurement and Control).

The prediction of performance, the effect of inlet flow and turbulence level, and the means of sensing are areas for development. These will require consideration of optimum geometry and, in the case of the vortex flowmeter, the axisymmetric bluff body.

Much of the computational fluid dynamics (CFD) work has concerned two-dimensional solutions. The axisymmetric vortex body should lend itself to this, but the straight vortex body in a circular pipe may call for more of an understanding of how upstream distortion and turbulence distribution feed into the creation of the shed vortices. There may be some useful analytical work that would shed light on this and might lead to a weight function for the meter.

The sensing systems are somewhat limited. Pressure, force, etc., have problems on turndown. Temperature sensing may offer scope for some applications. However, the electromagnetic solution is very attractive for liquids, and as well as application to fluidic flowmeter, I have, for some time, felt that the vortex flowmeter could be combined with a magnetic flow tube to give self-checking possibilities.

An elegant solution for gases is still awaited, and the development of suitable sensors is an unresolved problem. Should thermal be checked? Does the electrostatic option provide a solution?

At a more pedestrian level, but of great practical value for the vortex meter, would be further comparative installation effects of commercial meters, together with tests on the effect of pipe size difference, nonalignment, weld beads, etc., and also vibration. This might be combined with the development of a standard vortex-shedding meter as foreshadowed by Takamoto and Terao (1994).

The fluidic meter appears to have found a particular niche in the utilities, and several are being, or have been, developed. They appear to offer a large turndown, provided a suitable sensing system is available. One such system is the magnetic system developed (Sanderson 1994) for water flows.

The CFD analysis of the fluidic flowmeter is also, mainly, two-dimensional, but the effect of end walls and the turbulence level of the inlet flow may need further assessment, and it seems unlikely that the internal geometry has been finally optimized.

At the opposite extreme would be the investigation of intrinsic flow noise due to vortex shedding in existing pipework, which could be sensed and calibrated as a flowmeter system. If such noise exists, an intelligent sensor could learn its calibration and then provide for a utility measurement.

The continuing development of these meters in terms of range, reliability, low maintenance, accuracy, and price should ensure that they establish a respected place in the options for flow measurement.

APPENDIX 11.A

Vortex-Shedding Frequency

11.A.1 VORTEX SHEDDING FROM CYLINDERS

Flow around a cylindrical body develops as in Figure 11.1. At a low Reynolds number [Figure 11.1(a)], the flow pattern on each side of the cylinder is almost symmetrical (for $Re \ll 1$ there is symmetry cf. Tritton 1988). The symmetry between the flow approaching the cylinder from upstream and that leaving it downstream is removed with increasing Re . When Re exceeds about 4 (Tritton 1988), eddies, which are more or less symmetrical [Figure 11.1(b)], are formed behind the cylinder and, as shown in Figure 11.1(c), start to become unsymmetrical before a Reynolds number of about 40 is reached. It is from this point that we obtain the remarkable phenomenon of vortex shedding.

The lack of symmetry in Figure 11.1(c) breaks down into a periodic flow as illustrated in Figure 11.1(d). At this stage, the flow is stable and viscous (Roshko 1954). The boundary layer on the cylinder and the shear layer in the wake are presumably laminar. The shedding is very regular in this region. In the range omitted from Figure 11.1, namely $150 < Re < 300$ (or $200 < Re < 400$ according to Tritton), transition from laminar to turbulent takes place in the shear layer, and the transition is accompanied by less regular shedding. Once the regime in Figure 11.1(e) is reached, the shedding becomes regular once more. A further transition in the region $3 \times 10^5 < Re < 3 \times 10^6$ occurs as the boundary layer changes from laminar to turbulent causing irregularity (Tritton 1988). In this case, Tritton illustrates the laminar separation followed by turbulent reattachment and then, again, the occurrence of separation. The regime in Figure 11.1(f) has regular shedding and a later separation point. Tritton comments that markedly periodic shedding remains a characteristic of the flow up to the highest values of Reynolds number at which observations have been made, about 10^7 .

11.A.2 ORDER OF MAGNITUDE CALCULATION OF SHEDDING FREQUENCY

Using a numerical solution, Abernathy and Kronauer (1962) showed that two shear layers result in the build-up of large vortex regions. Their diagrams also showed that one shear layer gets caught in the other shear layer and eventually results in a region where vortices of opposite sense cancel each other and break the shear layer (cf. Robinson and Saffman 1982 on the stability of Karman vortex arrays).

Figure 11.2 is a simple diagram of this mechanism (cf. Figure 11.14). In Figure 11.2(a), one vortex has just separated and the next is beginning to roll up. The roll-up is caused by the influence of each new vortex on those that have gone before. In Figure 11.2(b), the large vortex roll has curled back on itself and has started to entrain the vortex sheet of opposite sense on the other side of the body. In Figure 11.2(c), this entrainment has advanced to the point where the upper vortex sheet is nullified and the roll-up of the vortex is stopped as the whole vortex roll is set free (cf. Gerrard 1966).

A simple model to obtain a value for the roll-up time of the large vortex is given by Figure 11.A.1. The vorticity shed will be of order V/δ , where δ is the shear layer thickness. The rate of shedding will be approximately the mean velocity in the shear layer $V/2$ multiplied by the layer thickness δ . Thus vorticity shedding rate is

$$\frac{V}{\delta} \times \frac{\delta V}{2} = \frac{V^2}{2}$$

The vorticity has to fuel a vortex with a diameter approximately equal to the size of the bluff body. The total vorticity within this roll will be

$$\begin{aligned} \int \omega \cdot d\mathbf{a} &= \int \nabla \times \mathbf{V} \cdot d\mathbf{a} \\ &= \oint \mathbf{V} \cdot d\mathbf{s} \\ &= \pi w V \end{aligned}$$

The time to fill a vortex of diameter w with $\pi w V$ is approximately

$$\tau = \frac{\pi w V}{V^2/2} = 2\pi w/V$$

Thus an order of magnitude value for the frequency again is

$$f = \frac{V}{2\pi w} \quad (11.A.1)$$

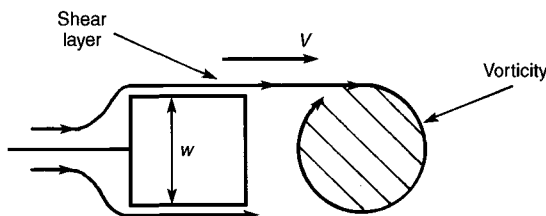


Figure 11.A.1. Simple model of vortex shedding.

A useful parameter is the Strouhal number given by

$$S = fw/V \quad (11.A.2)$$

and using the approximation of Equation (11.A.1) we obtain a value of $S = 1/(2\pi) = 0.16$. Goldstein (1965) gave 0.18 for a circular cylinder at a Reynolds number in the range 300–100,000.

Because of the reduced cross-section of the flowmeter due to the presence of a bluff body, the actual velocity past the body $V_{\max}$ will be greater than that in the upstream pipe V . For an incompressible fluid, area $\times$ velocity is constant. The area ratio will be given approximately by

$$\frac{A_{\min}}{A} = \frac{\pi D^2/4 - wD}{\pi D^2/4} \quad (11.A.3)$$

Thus for an incompressible fluid

$$\frac{V_{\max}}{V} = \frac{1}{1 - 4w/\pi D} \quad (11.A.4)$$

and so combining Equations (11.A.2) and (11.A.4) and assuming the appropriate velocity for Equation (11.A.2) is $V_{\max}$, we obtain

$$\frac{f}{q_v} = \frac{4SV_{\max}/V}{\pi w D^2} \quad (11.A.5)$$

$$= \frac{4S}{\pi D^3} \frac{1}{(w/D)(1 - 4w/\pi D)} \quad (11.A.6)$$

and so the approximate Strouhal number of 0.16, which we obtained based on the flow past the body will apparently be increased if based on the velocity in the main pipe according to Table 11.A.1. For comparison, Takamoto and Terao (1994), for a w/D of 0.28, give S in the range 0.245–0.265 for Re from 10^5 to 10^6 . The limitations of this simple approximation are shown by the effect of change in length of the bluff body from $0.4D$ to $0.2D$ found by Takamoto and Terao.

Zanker and Cousins (1975) assumed that the roll vortex was slightly larger than the bluff body diameter by between about 10 and 50%. This created a vena contracta confining the flow downstream of a sharp-edged bluff body to a smaller area near the wall in which the maximum velocity increases more. Adding their factor K to Equation (11.A.6), we obtain the same equation as they did:

$$\frac{f}{q_v} = \frac{4S}{\pi D^3} \frac{1}{(w/D)(1 - 4Kw/\pi D)} \quad (11.A.7)$$

It is arguable that a further K should appear as a multiplier in the denominator to allow for the larger vortex, which has to be filled. However, this is probably stretching a simple model too far, but interestingly it appears that for a w/D of

Table 11.A.1. Approximate variation of Strouhal number based on mean pipe velocity for various body sizes

w/D	$V_{\max}/V$	S corrected
0.1	1.15	0.18
0.3	1.62	0.26
0.5	2.75	0.44

about 0.3, this extended form of the denominator is less affected by the value of K than for, say, 0.1 or 0.5.

One can see how the change in the upstream conditions could influence the shedding frequency. Upstream disturbances will alter the vorticity in the flow and so reduce or increase the frequency of shedding and possibly the regularity. These changes are illustrated by Mottram (1991) for the effect of a more peaky profile due to pipe roughness and an increased Strouhal number. In this case, both the increase in local velocity at the body and the increase in vorticity will have effects. Other results discussed in this chapter show, surprisingly, that this type of meter is quite susceptible to the effect of a reducer upstream, presumably again because of the effect on the turbulence spectrum.

Terao et al. (1993) made two interesting observations.

- Vortices moved faster downstream than the mean velocity.
- Areas of intense vorticity existed at the wall.

The first of these two must affect the use of vortex shedding with cross-correlation techniques. The second suggests that three-dimensional computational solutions must include this observation (cf. Majumdar and Gulek 1981 for an early paper on shedding from various prisms).

Electromagnetic Flowmeters

12.1 INTRODUCTION

The possibility of inducing voltages in liquids moving through magnetic fields was known by Faraday in 1832, but the first flowmeter-like device was reported by Williams in 1930. The first real advance in the subject came from the medical field where Kolin (1936, 1941) introduced many ideas that are now standard practice.

The industrial interest in electromagnetic flowmeters (sometimes referred to as EM or magnetic flowmeters) grew in the 1950s with

- ❑ the Tobiflux meter (Tobi 1953) in Holland for rayon viscose, sand and water, and acid slurries;
- ❑ Foxboro, to whom the patent was assigned in 1952;
- ❑ the first commercial instruments in 1954 (Balls and Brown 1959).
- ❑ nuclear reactor applications;
- ❑ the work that resulted in an essential book by J. A. Shercliff (1962).

In this chapter, we shall concentrate on the application of the flowmeter to fluids that are of low conductivity, such as water-based liquids (Baker 1982). The flowmeter has also been used with liquid metals (Baker 1969, 1970b, 1977), and a few designs have been built for use with nonconducting dielectric liquids (Al-Rabeh et al. 1978). The reader is referred to the original papers because space prevents their inclusion here. Three papers by Wyatt (1961, 1977, 1982), a pioneer in blood flow measurement research, are referenced for those interested.

12.2 OPERATING PRINCIPLE

We start with the simple induction, which occurs when a conductor moves through a magnetic field. Figure 12.1 shows a copper wire cutting the flux of a permanent magnet. The wire is moving in a direction perpendicular to its length and perpendicular to the magnetic field with velocity V , and the result is a voltage generated between its ends of value $B l V$ where l is its length and B is the magnetic flux density.

Figure 12.2 shows a diagram with the essential features of an electromagnetic flowmeter. The liquid flows in a circular cross-sectional tube. A magnetic field is created across the pipe, usually by coils excited by an alternating current. The tube itself

must be made from nonmagnetic material so that the magnetic field can penetrate the tube. In this diagram, we may imagine filaments of liquid spanning the tube from one electrode to the other and moving through the magnetic field at different speeds generating voltages between their ends as was the case with the copper wire. To avoid these being shorted out, the tube is lined with an insulating material. The voltages in the liquid are measured between electrodes, which are set in the wall of the tube.

Referring now to Figure 12.3, we have depicted the case where more than one wire is moving through the magnetic field. Wire P is in a region of strong magnetic field B and has a velocity V . Wire Q is in a region where the field is about the same size, but it only has a velocity $V/2$. Wire R, although moving with velocity V , is in a region of weak magnetic field, say, $B/4$. Thus the magnetic induction in each will be different:

$$\begin{aligned} \text{In P} \quad \Delta U_P &= BIV \\ \text{In Q} \quad \Delta U_Q &= BIV/2 \\ \text{In R} \quad \Delta U_R &= BIV/4 \end{aligned} \quad (12.1)$$

Now if the ends of the wires are connected, currents will flow because the potentials differ, and the result will be that ΔU_P will be reduced by the ohmic loss. Carrying this same argument back to Figure 12.2 results in a complicated picture of voltages and circulating currents. Despite this apparent complexity, the actual operating equation of the flowmeter is quite simple for a range of conditions. The voltage between the electrodes is given by

$$\Delta U_{EE} = BDV_m \quad (12.2)$$

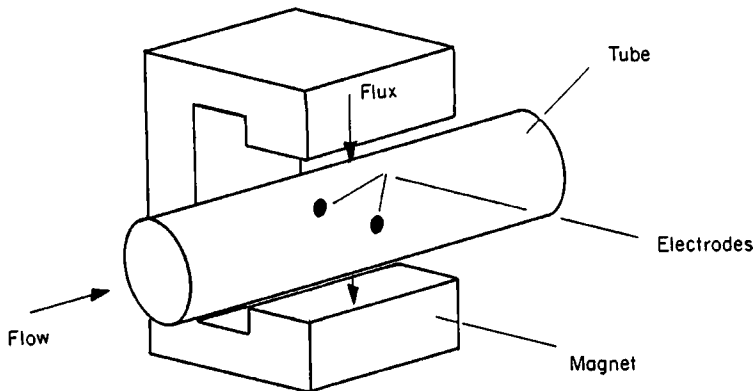


Figure 12.2. Simple flowmeter.

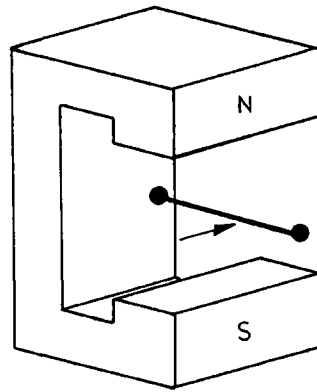


Figure 12.1. A wire moving through a magnetic field generates a potential difference between its ends.

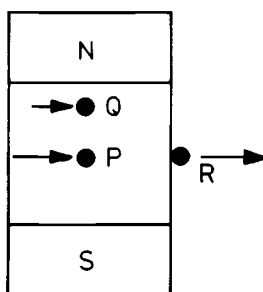


Figure 12.3. Three wires moving through a magnetic field of varying spatial strength at different speeds.

where B is the magnetic flux density in tesla, D is the diameter of the tube in meters, and V_m is the mean velocity in the tube in meters per second.

This is the basic equation for the flowmeter, and it can be shown to be valid only when

- the magnetic field is uniform and
- the velocity profile is axisymmetric.

In modern designs, these requirements are seldom satisfied, and so designs have been developed to give an output signal that is as little affected by the flow profile as possible, in compact designs with magnetic fields far from uniform.

12.3 LIMITATIONS OF THE THEORY

Uniform Field – It can be shown that in theory the only field that is uniform is one from a magnet of infinite extent.

Axisymmetric Profile – All fully developed pipe flow profiles are axisymmetric. But it is not always convenient to allow the necessary upstream pipe length to ensure a fully developed profile.

Shercliff (1962) suggested a means of predicting the effect of distorted profiles using what he called a weight function (Appendix 12.A). He calculated the shape of this function for a uniform magnetic field meter [Figure 12.4(a)]. In its simplest form, this indicates how important the flow is in any part of the pipe cross-section and how great its influence will be on the flow signal. The velocity at each point in the cross-section of the flowmeter is multiplied by the weight function. Thus the weighting function of 2 near the electrodes will cause the flow in that part of the pipe to have twice the influence on the signal. If all the flow went past that point, and the flow everywhere else was stationary, the signal would be twice as big as it should be.

This important concept has been extended by Bevir (1970) and Hemp (1975) and has proved a very useful method of assessing the performance of this and other types of flowmeter.

However, Figure 12.4 compares the distribution due to Shercliff with others. Figure 12.4(b) shows the distribution for flowmeters with rectangular-shaped coils, a common design. Figure 12.4(c) shows the distribution for flowmeters with diamond-shaped coils with their corners along the axis of the pipe and toward the electrodes. These are quite similar to some commercial designs, and the plots of weighting functions can be seen to give a much more uniform distribution, particularly in the diamond-shaped case.

Another approach that has been used is to introduce large electrodes that have an integrating effect on the signal and again improve the theoretical performance. Figure 12.4(d) shows an example of a large electrode meter, and it can be seen that

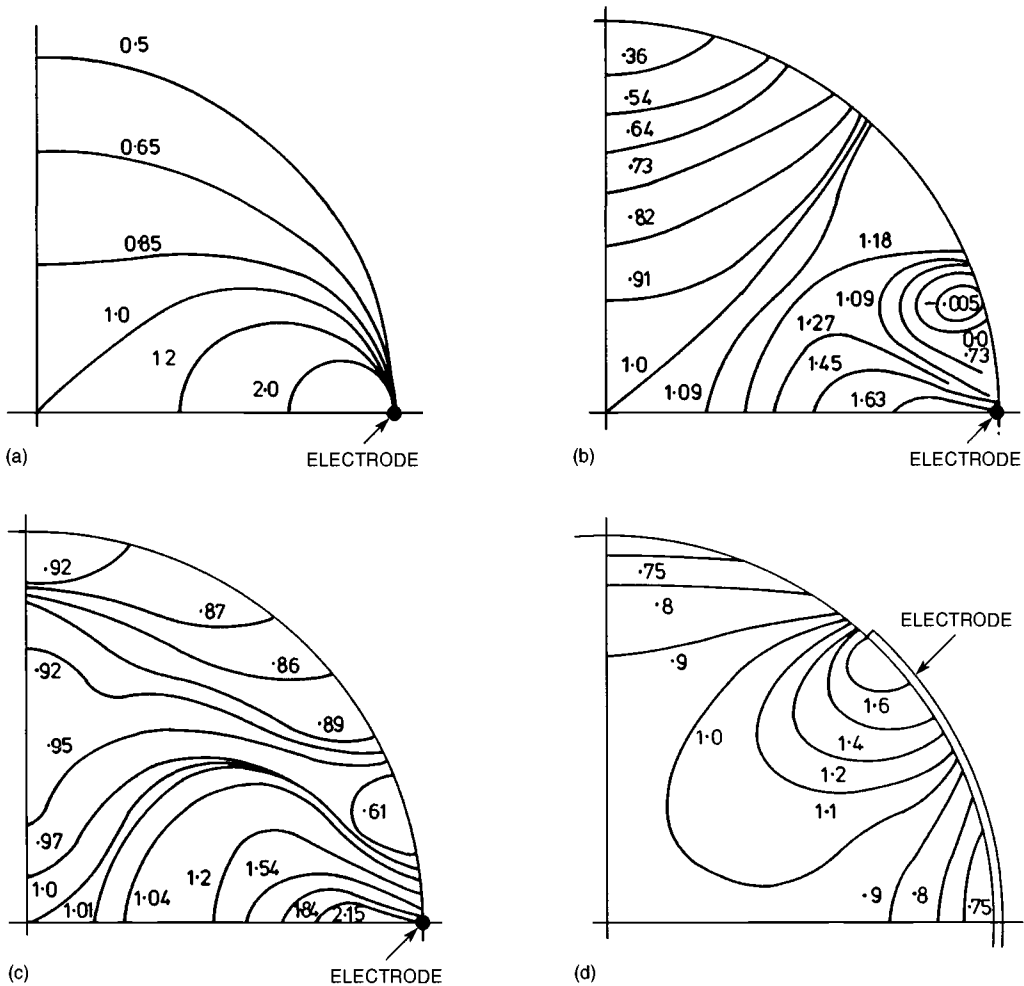


Figure 12.4. Weight functions: **(a)** For a uniform field point-electrode flowmeter (Shercliff 1962); **(b)** For a point-electrode flowmeter with rectangular-shaped coils and circular yoke (Al-Rabeh and Baker 1979); **(c)** For a point-electrode flowmeter with diamond-shaped coils and circular yoke (Al-Rabeh and Baker 1979); **(d)** For a large electrode flowmeter with rectangular-shaped coils and rectangular yoke (Al-Khazraji and Baker 1979).

the weight function over most of the cross-section varies only between about 0.75 and 1.6 (cf. Al-Khazraji and Hemp 1980).

For further details on the development of this meter and for additional references, see Baker (1982, 1983, 1985) and Hemp and Sanderson (1981). A thorough review of operating problems was provided by Cox and Wyatt (1984).

The effect of short magnetic fields causing nonuniformity of field results in the kind of shorting taking place between P and R in Figure 12.3, which was called end-shorting by Shercliff. If we define a value S , the flowmeter sensitivity, by

$$S = \frac{\text{Voltage generated}}{BDV_m} \quad (12.3)$$

we find that $S = 1$ for Equation (12.2), but for a field length equal to the pipe diameter, it falls to about 0.8.

12.4 DESIGN DETAILS

Figure 12.5 shows the main components of an industrial flowmeter with point electrodes. The industrial flowmeter is a combination of two elements – the sensor or primary element and the transmitter or secondary element.

The sensor or primary element consists of the metering tube with insulating liner, flanged ends, coils to produce a magnetic field, and electrodes. In addition, some means to produce a reference signal proportional to the magnetic field is usually provided. On AC-powered systems, this has, typically, been a current transformer, or search coil. On DC systems, it may be appropriate to measure the excitation current while constant.

The transmitter or secondary element, which is sometimes known as the converter,

- a. amplifies and processes the flow signal;
- b. eliminates spurious electromotive forces;
- c. ensures that the transmitter is insensitive to electricity supply variation, radio interference, etc;
- d. must have the required level of safety; and
- e. communicates with the user's control computer in the most efficient way.

12.4.1 SENSOR OR PRIMARY ELEMENT

The metering tube (Figure 12.6) will usually be nonmagnetic to allow field penetration. It may be available in a range of diameters from about 2 mm to about 3000 mm (10 ft). Flow ranges may be from 0 to 28,500 m³/h or more. The inside surface is insulated to prevent signal shorting. Electrodes are positioned at opposite ends of a diameter perpendicular to, and central in, the magnetic field. They are typically small (5–20 mm diameter) and are sometimes referred to as point or button

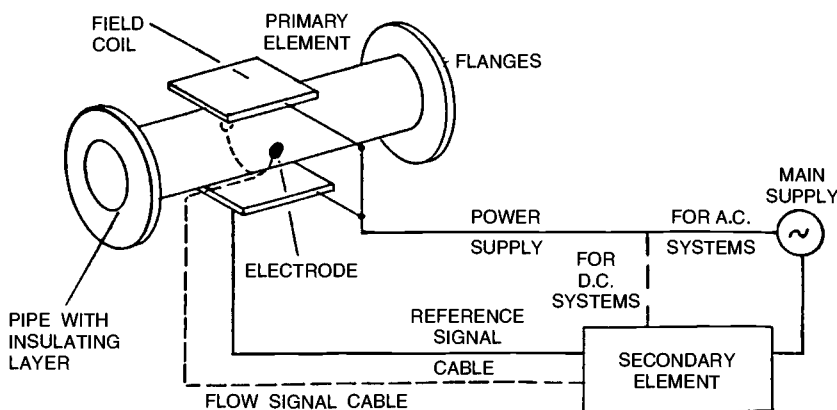


Figure 12.5. Main details of an industrial meter.

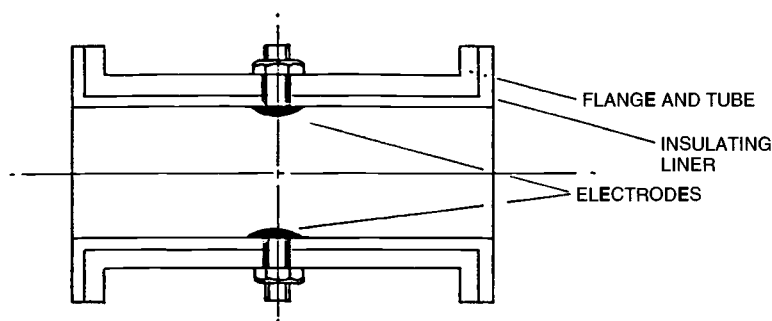


Figure 12.6. Metering tube.

electrodes. Large electrodes have been used in a few designs, and these may subtend about 90° and approach one diameter in length.

Typical materials for insulation are given in Table 12.1. The material is often molded around the tube end forming a composite with the flanges. Too high velocities [i.e., greater than 4 m/s (15 ft/s)] may lead to liner wear, which can be reduced with liner protection (Ginesi and Annarummo 1994). Grounding/earthing rings have been used in the past, where the pipes were lined or nonconducting, to provide a reference point for the measuring electrodes. In modern designs, reference plates or electrodes, which are not necessarily earthed, may be provided. An earth link could, where the liquid is carrying an electric current, provide a track to

Table 12.1. Liner materials with an indication of approximate temperature limits that should be checked with the manufacturer

Material	Application	Temperature Limits ($^\circ\text{C}$)
Natural rubber	Wear and chemical resistance	-20 to 70
Neoprene	Good chemical and wear resistance in presence of oil and grease	0 to 100
Teflon	Abrasion and chemical resistance	0 to 90
Ebonite		
Fluorocarbon } Polyurethane } Elastomer }	Slurry	
Polyurethane		
	Wear and impact resistance	-50 to 70
Polytetrafluoroethylene (PTFE)	Wear resistance, chemically inert, good for foodstuffs	-50 to 200
Ceramic 99.9% alumina (Al_2O_3) with cermet ($\text{Pt}-\text{Al}_2\text{O}_3$) electrodes moulded in before firing		

earth and cause galvanic damage. In this case, the meter may be allowed to float electrically and would be made safe via an isolating transformer.

The electrodes normally have to make contact (cf. noncontacting electrodes, Section 12.6) with the fluid and therefore break through the tube lining (Figure 12.6). They are sometimes formed as a dome-headed screw and bolted in, pulling down onto the liner material, the electrical tag connection to the leads being bolted on finally. Because they are in contact with the fluid, the material of the electrode must be chosen with care. Some of the materials used are nonmagnetic stainless steel (for nonaggressive liquids), platinum-iridium, Monel, tantalum, titanium, zirconium (for aggressive liquids), and Hastelloy-C. Stainless steel may be recommended for slurries. Note also the ceramic combination of liner and electrode.

In paper pulp and possibly other applications, the impact of paper or other material on the electrodes can cause noise. According to one manufacturer, a porous ceramic coating over the electrode may reduce this problem.

Because the electrode must make contact with the liquid, various means have been tried for electrode cleaning including

- ❑ scraping (a scraper or brush may pass through the center of the electrode and be turned to scrape the surface) (cf. Rose and Vass 1995),
- ❑ burn-off (the passing of a large enough current to remove surface deposits while the rest of the electronics is disconnected),
- ❑ ultrasonic cleaning (vibrating the electrode with ultrasound to cause local cavitation and hence cleaning),
- ❑ electrodes removable in service, and
- ❑ bullet-nosed electrodes.

The method should be chosen in relation to the likely nature of the deposit. In many cases, electrodes appear to be self-cleaning: the flow past them keeps deposits at bay. Coatings on meter internals can have more, the same, or most commonly less conductivity than the bulk fluid. In modern DC systems, the input impedance may be high enough to make the effect of deposits negligible.

The magnetic field is usually produced by a pair of coils and a laminated yoke (Figure 12.7). Typical power consumption used to be 10–100 W, but the lowest

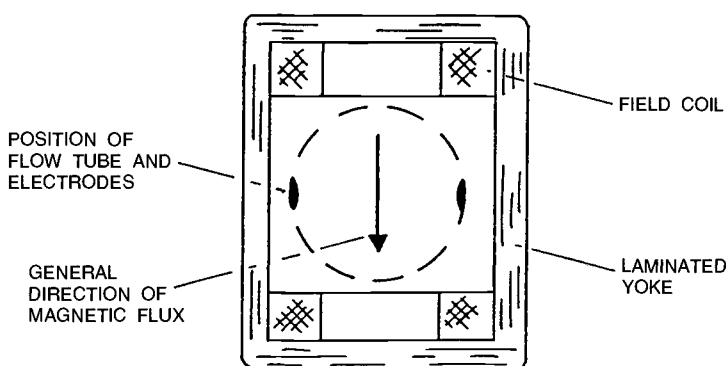


Figure 12.7. Magnetic field coils and yoke.

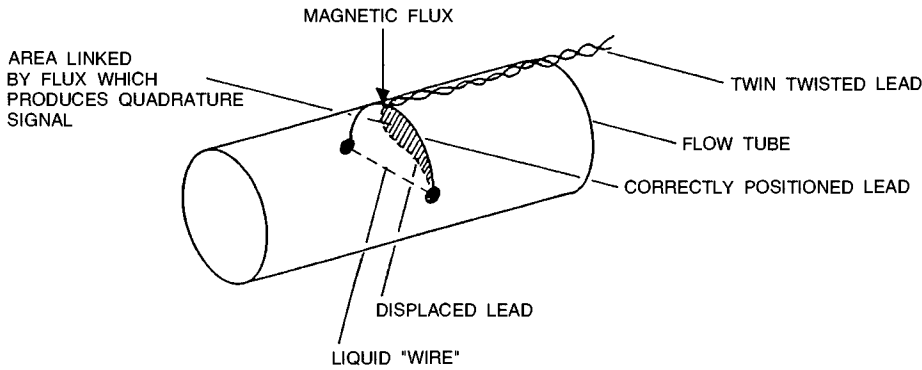


Figure 12.8. To show flux linking of signal leads.

powers now can be as little as 0.5 W and, presumably, on average a lot less than this since long life on battery power is now possible.

As a result of using AC excitation, there is a danger of inducing a transformer-type signal, by the changing magnetic flux linking a loop of the electrode leads and fluid combination. Figure 12.8 indicates a poorly laid lead and the resulting area, which is linked by the changing flux. This area does not need to be large to result in a signal of a size comparable with the flow signal. It will be a quadrature signal (the phase will be 90° out of phase with the induced flow signal) given approximately by

$$\text{Quadrature voltage} \sim 2\pi fBA$$

where f is the frequency, B is the flux density, and A is the area of the resulting loop projected onto a surface perpendicular to the field direction. For example, if f is 50 Hz, B is 0.02 T, and A is 10^{-4} m^2 (1 cm^2), then the quadrature voltage will be about 0.6 mV. This should be compared with a flow signal for 0.1 m diameter tube and 5 m/s of 10 mV. Careful design, and mechanical adjustment of the position of the leads in manufacture, can reduce this quadrature to some extent, but it is finally dealt with in the converter. The use of DC excitation overcomes this problem, by sensing the flow signal when the field is briefly constant in time.

The primary element installation should not cause undue stresses in the tube and should ensure that the tube is always full. It is usual to mount the metering tube so that the electrode diameter is in a horizontal plane to avoid bubbles at the top of the tube open circuiting the electrodes.

The flow tube is most commonly of stainless steel, to allow the magnetic field to penetrate. Maximum pressure for sensors may be as much as 1,000 bar.

Designs may include options for use in adverse and hazardous environments.

12.4.2 TRANSMITTER OR SECONDARY ELEMENT

Various types of secondary element are now available. The longest serving AC types have used 50 or 60 Hz. Because mains power is used, the magnetic field and, hence, also, the flow signal are high. The common but more recent designs use a low frequency square wave of various patterns, which thereby allows quadrature signals to decay before the flow signal is sampled. This type is referred to here as the square wave excitation (or DC) system, but it comes under a variety of names depending on

the manufacturer (cf. Brobeil et al. 1993). The description DC is used with caution because early meters did, indeed, attempt to operate on true DC without success. Field strengths may be lower in DC designs, but electrical noise from slurries and electromechanical effects will be the same as for AC. Most recent designs of DC meter may, therefore, have special high power supply modules to address this.

Bonfig et al. (1975) described one of the first successful designs, which is referred to as keyed DC field. Hafner (1985) described another system, switched DC, which had features for noise reduction (e.g., both passive and active shielding), electrochemical effects, periodical zeroing of the amplifier, multiple sampling of the signal, increased field frequency (up to 125 Hz), digital filtering, and the use of flow noise analogue circuitry. With low power [down to 1.5 W (as well as size and weight)], the design achieved intrinsic safety requirements and operated off a battery. Microprocessor control also provided self-testing, temperature compensation, interchangeable primaries and secondaries, and communication. In addition, electrodes provided earthing and checking for empty tube. Herzog et al. (1993) dealt with switched DC design, which included a periodic integration of the electronic reference point and discussed a third electrode circuit for partially filled tubes.

The output is usually 0–10 or 4–20 mA, and two or three range switches are provided to allow a full-scale output reading to be achieved with flows from 1 to 10 m/s. However, this is being superseded by the power of microprocessor technology, which allows a move toward smart/intelligent instrumentation with autoranging, digital transmission and a much greater range of possibilities.

Figure 12.9 gives a block diagram of a typical AC circuit. The demodulator uses the reference signal to remove quadrature voltages, and the circuit obtains the ratio of flow signal to reference signal.

The block circuit of Figure 12.10(a) is typical of the approach used in the DC systems. The sampling at times T_n , T_{n+1} , and T_{n+2} is illustrated in Figure 12.10(b), which

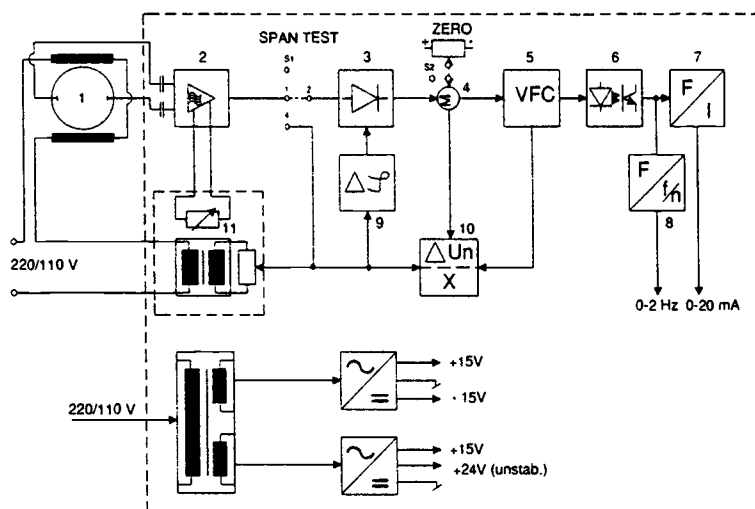


Figure 12.9. Diagram of converter circuit for AC system: 1, sensor; 2, input amplifier; 3, demodulator (phase-selective); 4, summation point; 5, voltage/frequency converter; 6, optocoupler (galvanic separation); 7, frequency/current converter; 8, pulse output (option); 9, reference; 10, division selection (coil current compensation); 11, current converter, reference source, measuring range module (reproduced with permission from Endress and Hauser).

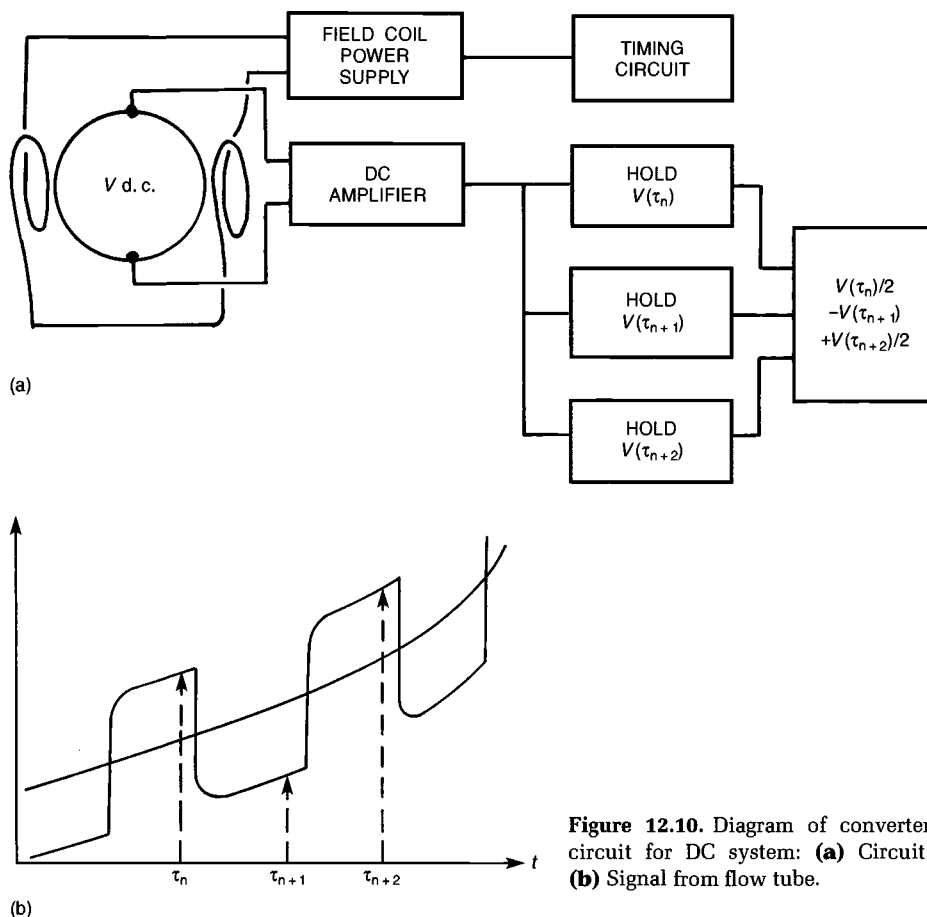


Figure 12.10. Diagram of converter circuit for DC system: **(a)** Circuit; **(b)** Signal from flow tube.

exaggerates the drift of the baseline of the square wave signal due to electrochemical and other effects, and the reason for three samples is then apparent.

The manufacturer's limits on length and specification of signal cable should be carefully followed. The cable is usually twin-twisted-shielded for AC excitation. However, with low power operation possibly using a battery, two-wire operation may be offered.

Zero drift may be found in some instruments but is usually small and is probably due to incomplete suppression of unwanted (particularly quadrature) voltages. The DC systems usually claim to eliminate this, although with a low flow cutoff, this may be hard to confirm. A low flow cutoff is usually set at 1% of upper range limit (Ginesi and Annarummo 1994) or possibly lower.

The overall accuracy of the converter may be claimed to be about 0.2% for quite wide tolerance of mains voltage, quadrature signal, temperature fluctuation, etc. Very low flows can be measured, but with decreasing precision.

Commercially available transmitters will increasingly offer:

- response – of order 0.1 s,
- turn down – up to 1000:1,
- flow ranges – part of 0.005–113,000 m³/h,
- pulse rates – from 0.01 to 10 l/pulse.

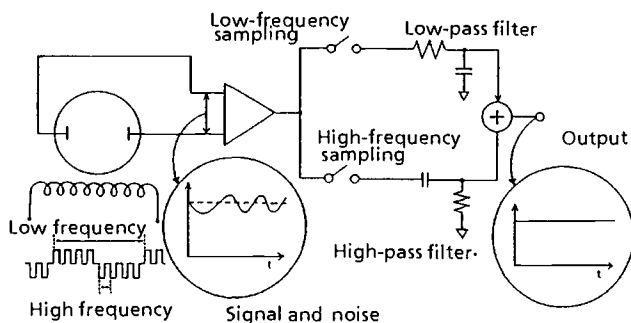


Figure 12.11. Dual-frequency operation (reproduced with permission of Yokogawa Europe B.V.).

Manufacturers list some of the following features:

- ❑ sensor power supply and transmission carried on intrinsically safe two-wire cable for both power and signal;
- ❑ digital signal transmission through AC modulation of analogue signal;
- ❑ Interelement Protection, IP65 for transmitter;
- ❑ dual frequency (Figure 12.11) giving benefits of both high and low frequencies: the frequencies are processed through separate channels before being brought together to give both low-flow stability and low noise (cf. Matsunaga et al. 1988);
- ❑ interference immune transmission;
- ❑ self-checking or multiply checked data;
- ❑ empty pipe detector and alarm using an electrode to sense tube empty conditions (Ginesi and Annarummo 1994);
- ❑ grounding electrode;
- ❑ check for fouling of main electrodes;
- ❑ flow in both directions measurable with suitable electronics;
- ❑ auto ranging.

Application-specific integrated circuits (ASICs) offer advantages such as automatic system monitoring with diagnostics for reverse flow and other errors and alarms, dual ranging, and serial interface communication (Vass 1996).

12.5 CALIBRATION AND OPERATION

Because of manufacturing variation between meters, they require calibration, and this is usually done by the manufacturer. One manufacturer, for instance, offers a 13-point calibration for a master meter. This is known as *wet calibration*. The term *dry calibration* is used in connection with electromagnetic flowmeters to imply calibration by measurement of the magnetic field and deduction of flow signal from this. The relationship between the field at a specific point and the overall performance of the meter is not as straightforward as Equation (12.2) might imply, and any dry calibration should be treated with caution at the present time.

The operation of the meter should not, generally, be affected by variation of liquid conductivity, provided that the conductivity is uniform over the region of

Table 12.2. Output resistances of meter tube based on electrode diameter of 0.01 m

	Liquid Conductivity		Resistance (ohms)
	(S/m)	($\mu\text{S/cm}$)	
Best electrolyte	$\sim 10^2$	$\sim 10^6$	1
Sea water	~ 4	$\sim 4 \times 10^4$	25
Tap water	$\sim 10^{-2}$	$\sim 10^2$	10,000
Pure water	4×10^{-6}	4×10^{-2}	25,000,000

the flowmeter. This also presupposes that the conductivity is sufficient to ensure a primary device output resistance of two or more orders of magnitude less than the input resistance of the secondary device. However, severe variations in conductivity may cause zero errors in AC-type magnetic flowmeters. Even though some suggest that DC pulse types should be unaffected by such changes above a minimum threshold (Ginesi and Annarummo 1994), one manufacturer appears to take an opposite view, recommending AC type for two-phase, slurries, low conductivity flows, or inhomogeneous flows with rapidly varying conductivity. However, continuous development of the DC type may ensure that they are equally suitable.

The output resistance of the primary device is approximately given by

$$R \approx \frac{1}{d\sigma} \text{ ohms} \quad (12.4)$$

where d is the electrode diameter, and σ is the conductivity.

From Equation (12.4), we can obtain typical resistances of a meter with electrodes of diameter 0.01 m, as shown in Table 12.2.

A secondary device with a typical input resistance of 20×10^6 ohms could cope with the first three liquid conductivities in Table 12.2, but not with the last one. Manufacturers may limit conductivity minima for certain sizes of meter. For instance, for 25–100 mm, conductivity may only be acceptable down to $20 \mu\text{S/cm}$. However, at least one manufacturer offers a meter capable of use with down to $0.05 \mu\text{S/cm}$ conductivity.

Entrained gases cause errors due to breaking of electrical continuity, nonuniformity of conductivity, and uncertainty as to what is being measured. The flowmeter should be positioned where these are negligible.

12.6 INDUSTRIAL AND OTHER DESIGNS

It is not my intention to detail all the features of industrial designs nor the common operational problems (see, for instance, Baker 1982) but rather to give the main types that reflect the theoretical developments. Figure 12.12 shows typical industrial designs. After early designs, which were long in an attempt to create a uniform field, all are now more compact and Figures 12.4(b,c) provide a means of performance comparison with Figure 12.4(a) for the uniform field design.

Internal coils recessed into an enlarged cross-section of the flow tube have been used in the past to avoid the need for nonmagnetic pipes. If used today, they are rare.

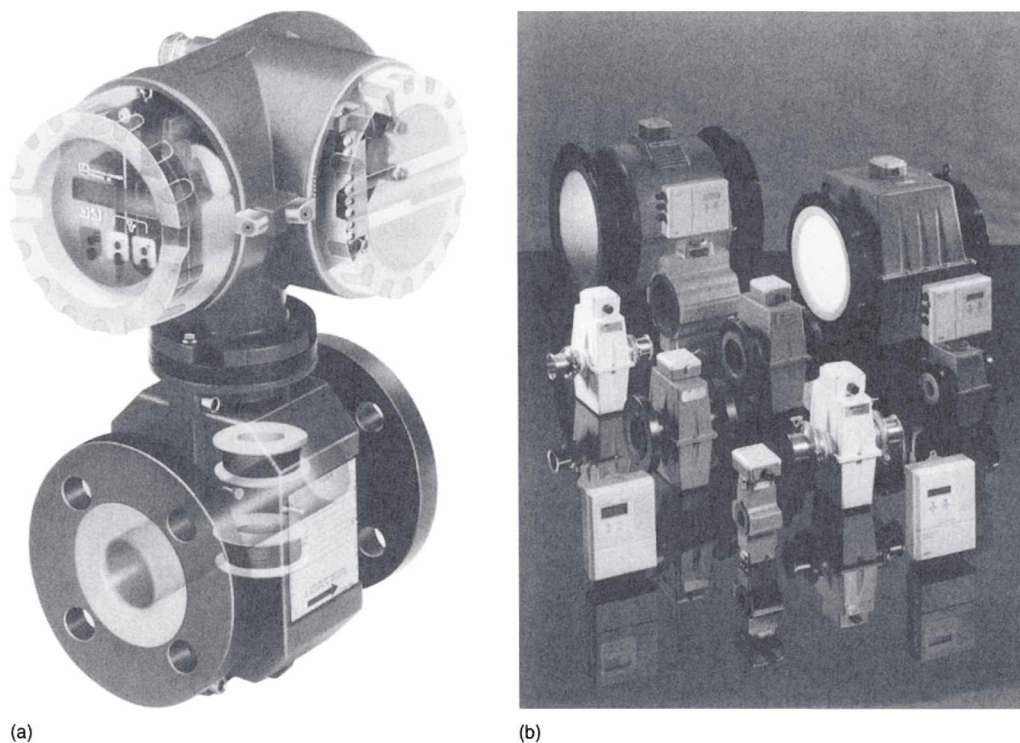


Figure 12.12. Industrial designs (reproduced with permission from Endress and Hauser); **(a)** Showing main components; **(b)** Various sizes.

Manufacturers have developed electrodes with shapes and sizes optimized for individual applications. Large electrodes have been tried (Figure 12.13) and should result in a more uniform weight function [Figure 12.4(d)]. Yoshida et al. (1993) suggested using short axial length arc-shaped electrodes subtending about 45° for partially filled pipes.

However, noncontacting electrodes offer additional advantages by permitting a continuous lining to the flow tube and avoiding crevices around the electrodes. They may also be less sensitive to fouling (cf. Rose and Vass 1995). Noncontact designs may allow operation with fluids of conductivity down to $0.05 \mu\text{S}/\text{cm}$ compared with normal minimum values of $5 \mu\text{S}/\text{cm}$. Other design advantages indicated earlier

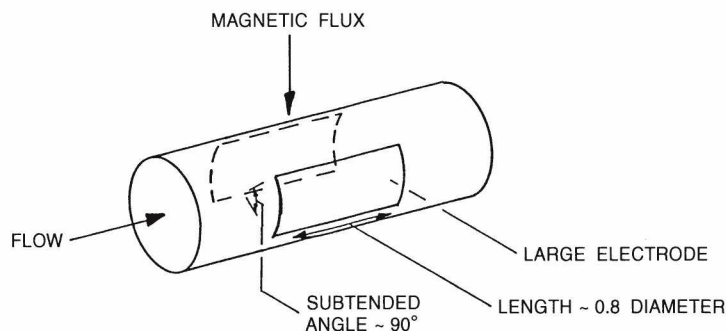


Figure 12.13. Large electrode flowmeter.

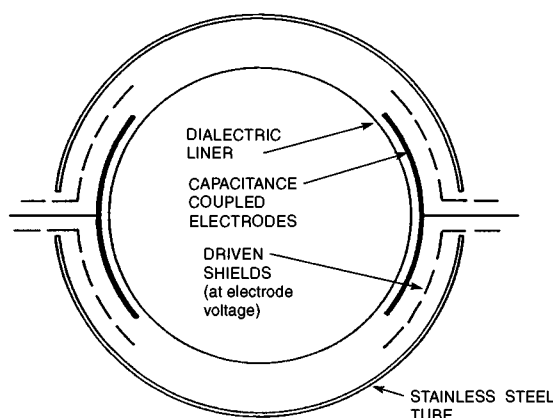


Figure 12.14. Noncontacting electrodes.

do not appear to have been fully exploited in commercially available instruments (Figure 12.14; cf. Hussain and Baker 1985, Brockhaus et al. 1996).

An electromagnetic meter has been proposed (Rose and Vass 1995) and may be available, with three pairs of electrodes – one on the horizontal diameter and the other two pairs at the end of horizontal chords below the diameter. This arrangement is claimed to allow the meter to run partially empty. The field coils are excited in series for a flow signal. However, when used in opposition, it is possible to obtain the level of liquid in the pipe. With this arrangement, when the pipe is full, the mean voltage on the diametral electrodes is zero, but imbalance in the signal increases as the pipe empties. Installation requirements are more stringent with 10D straight pipe upstream and 5D downstream. The pipeline should also have only a small slope to ensure tranquil flow. Minimum conductivity for the commercial version is claimed as $50 \mu\text{S}/\text{cm}$, and uncertainty is claimed as 1.5% span down to 10% fill level.

Power ratings will cover a wide range (from 0.5 W for new designs up to 300 W for a noncontact flowmeter).

Accuracy claims for most designs are of order 0.5% of flow rate, but manufacturers of noncontacting designs may initially be more cautious.

Hofman (1993) proposed the use of ceramic materials to increase stability particularly with temperature change. PTFE and the electrode gaskets can result in changes due to use, temperature change, and pressure over periods of, say, 2 years. With ceramic, by incorporating sintered electrodes, creep is reduced from order 1% to order 0.1% or better. The problem with ceramic is temperature shock, but new ceramics may allow rapid changes of up to 100°C . Hofman also suggested a conical inlet to improve profile at the cost of a small increase in pressure loss. With special materials to ensure magnetic field stability, he claimed, on a 10-mm tube, $\pm 0.1\%$ on a 10:1 turndown (4.2 m/s upper range value with a possible maximum of 12 m/s) for 20–85°C compared with $\pm 0.4\%$ for PTFE over the same range.

12.7 INSTALLATION CONSTRAINTS – ENVIRONMENTAL

Some manufacturers indicate that they prefer vertical installation, with upward flow. This ensures that entrained gases are carried out and do not affect electrode contact.

Table 12.3. Results obtained from Bevir (1972) for the effect on sensitivity of field and liner length

Field Length	Liner Length				
	0.2D	0.5D	1.0D	2.0D	∞ D
0.2D	0.211	0.335	0.380	0.388	0.389
0.5D	0.211	0.493	0.638	0.661	0.662
1.0D	0.211	0.497	0.790	0.865	0.867
2.0D	0.211	0.497	0.790	0.966	0.979
∞ D	0.211	0.497	0.790	0.966	1.000

The electrode to detect if the pipe is not running full should then be positioned at the top of the meter as installed.

12.7.1 SURROUNDING PIPE

Wafer construction flowmeters may be affected by surrounding pipe. Bevir (1972) calculated the effect of length of magnetic field and liner on sensitivity. Using a magnetic field, which represented typical industrial practice, he obtained values for uniform flow profile of which a sample are given in Table 12.3.

Nonconducting pipes abutting the flowmeter will be equivalent to an infinite liner length. Thus a flowmeter with field length of 1D and liner length of 1D will lose 9% in sensitivity when inserted in conducting pipework.

Baker (1973) calculated sensitivity reductions of about 5% for liner length reduction from 2D to 1D for a rectangular coil circular yoke flowmeter of 0.8D coil length.

Although the values in Table 12.3 are not directly related to the effect of steel pipework on the magnetic field distribution, it is reasonable to assume that the effect of such pipework will be less than lengthening the field. Thus for a 1D liner length, we note that increasing field length from 1D has no tabulated effect. If, however, the flowmeter is connected in nonconducting pipe, then field length change from 1D to 2D will cause a sensitivity increase of 13%. Assuming, for example, that the edge field increase due to connection into steel pipe will be of order 5% of the effect of increasing the length of the field, a 13% change (in the 5% component of the field) will be less than 1% on the sensitivity.

12.7.2 TEMPERATURE AND PRESSURE

Fluid density changes must be allowed for in calculating mass flow, but here we are concerned with accuracy as a volumetric flowmeter. Pressure will cause hoop stresses in the pipe resulting in strain, but the effect is likely to be negligible (Baker 1985).

Temperature increase will cause expansion of the flow tube and of the windings. The effect of the latter will depend on the method of compensation for field strength variation. If a search coil is used, this may also be affected by temperature change. Using a coefficient of expansion α of $10^{-5}/^{\circ}\text{C}$, Baker (1985) obtained a maximum error due to temperature of about $-0.3\%/100^{\circ}\text{C}$. However the effect of temperature and pressure on the liner, which may well dominate, has not been considered.

12.8 INSTALLATION CONSTRAINTS – FLOW PROFILE CAUSED BY UPSTREAM PIPEWORK

12.8.1 INTRODUCTION

In this chapter, four examples of rectilinear weight function distributions have been given (a fifth is given in Figure 12.A.1(b)). From these, it should be possible to deduce the performance of different types of flowmeters for various installations.

Rather than use actual fittings, some experiments have used localized jets and orifice plates to distort the flow profile in the meter. Shercliff (1955) used a localized wall jet to demonstrate the validity of the weight function distribution in Figure 12.4(a). Cox and Wyatt (1984) used this method to show the insensitivity of a contactless design to severe profile changes.

It is necessary to remember that in real installations the designer seeks a meter that is affected as little as possible by changes from the calibration profile. These changes, in most cases, represent small perturbations of the datum profile. The electromagnetic flowmeter is insensitive (Baker 1973) to a number of profiles that have reverse symmetry about diametral planes. Thus if such a profile is added to the datum profile, there should be no change of signal.

12.8.2 THEORETICAL COMPARISON OF METER PERFORMANCE DUE TO UPSTREAM FLOW DISTORTION

Al-Khazraji et al. (1978) used an upstream orifice to distort the flow profile (Table 12.4). The diameter of the orifice was half that of the pipe, and the center of the eccentric orifice was at half the pipe radius.

Where experiments were done, they appeared to confirm these trends. These computed values suggest that a disturbance upstream of the flowmeter will have minimum effect if its plane of symmetry is perpendicular to the electrode plane.

Approximate calculations based on eccentric orifice profile and Shercliff's (1962) weight function distribution [Figure 12.4(a)] suggest that errors for a uniform field flowmeter would be $\pm 1\%$ depending on the jet orientation.

Halttunen (1990) used data on flow profiles and combined them with magnetic field data obtained using a Hall probe and a computerized traversing and measurement system. He also gave experimental data from installation tests. The tests appear to have been confined to a single elbow test for one configuration of electromagnetic meter. The agreement is quite good and suggests that this type of combined approach to installation testing may be worth pursuing. The conclusions that can

Table 12.4. Calculated signal change (%) from uniform profile (Al-Khazraji et al. 1978)

Orientation of Eccentric Orifice Peak Velocity	Small Electrode		Large Electrode [Figure 12.4(d)]
	Rectangular Coil [Figure 12.4(b)]	Diamond Coil [Figure 12.4(c)]	
At electrodes	3.3	1.5	0.03
45° displaced	2.4	0.9	−0.03
90° displaced	1.4	0.0	−0.02

Table 12.5. Conclusions that can be drawn from the plots obtained by Halttunen (1990) on signal change due to upstream disturbance

	2D (%)	5D (%)	10D (%)	15D (%)	20D (%)
Single elbow	-1.5 to 0.5	± 0.5	± 0.5	small	small
Double elbow 90°	± 2.5	± 1	± 1	± 1	± 1

Note: that at about 40D error becomes small.

be drawn from the plots are given in Table 12.5. The flowmeter's (electrodes and magnetic field) orientation relative to upstream orientation of fittings may cause an additional $\pm 0.5\%$ at 5D or less spacing. I was pleased to see a meter in these tests named The Shercliff after someone whose contribution to the development of this meter has been so important!

Luntta and Hattlunen (1989) confirmed the effect of a bend but noted that four different meters (400 mm ID) gave very different results when downstream of an eccentric orifice plate and of a reducer.

12.8.3 EXPERIMENTAL COMPARISON OF METER PERFORMANCE DUE TO UPSTREAM FLOW DISTORTION

Scott (1975b) was the first to publish data on electromagnetic flowmeter operation downstream of pipe fittings. As Mr. D. Halmi commented in the discussion of this paper, it is not possible to generalize from such data. This is mainly because of the variation in meter design.

Much of the useful data is inaccessible to the public. For instance, de Jong (1978) has published a few results from his tests, which omit full flowmeter design detail. His most useful conclusions appear to be that, for a gate valve 5D upstream and at least 50% open, errors were within 0.5% of specification.

Tsuchida et al. (1982) gave the requirements in Table 12.6 for minimum upstream straight pipe to ensure uncertainty within $\pm 0.5\%$ of full scale.

Table 12.6. Tsuchida et al.'s (1982) values for minimum upstream straight pipe to ensure uncertainty within $\pm 0.5\%$ of full scale

Pipe Fitting	Minimum Upstream Straight Pipe
Reducer	3D (direct connection)
Expander }	5D
Ball valve }	
Single elbow	10D
Two elbows in perpendicular planes	
Gate valve	
Butterfly valve	
	15D

Table 12.7. Deacon's (1983) values of maximum error over all tests for upstream components

	Spacing	Max. Error (%)
Reducer	2.5D	0.8
Gate valve (at least 50% open)	5D	1.2
One bend	5D	2
Two bends in perpendicular planes		

Deacon (1983), after tests with a gate valve upstream (at least 50% open), one swept bend (radius/ $D = 1.5$), two swept bends in perpendicular planes, and a reducer, was able to conclude that the worst error with 5D spacing was less than 2% from calibration (Table 12.7). For gate valve 50% open and for one bend, the orientation of the flowmeter electrodes relative to the pipe fittings did not affect the magnitude of the error.

These data indicate considerable variation for a particular installation. One bend with 5D spacing caused about 0.3% error for a large electrode flowmeter and up to 2.0% error for a small electrode flowmeter. The small electrode flowmeters exhibited errors that could be, in some cases, a factor of 2 different. For one small electrode flowmeter, opening the gate valve from 50 to 75% *increased* errors by a factor of more than 4 when the orientation caused a jet past the electrodes.

12.8.4 CONCLUSIONS ON INSTALLATION REQUIREMENTS

Tentative conclusions on the likely influence of installation profiles follow.

- The error due to a reducer installed next to the flowmeter is likely to be less than 1%.
- An error of up to 2% should be assumed for other fittings separated from the meter with 5D of straight pipe.
- An error of 1% may occur for other fittings separated from the meter with 10D of straight pipe.
- The orientation of the flowmeter at 5D spacing or greater does not simply correlate with the size of the error.
- At least 3D spacing downstream should be allowed.

Manufacturers may be less conservative, but spacing such as 3D to 5D upstream, with 3D downstream, should be treated with caution. Despite data such as de Jong's (1978), valves should not be installed close upstream of flowmeters and, even if fully open, are likely to have an effect of $\pm 0.5\%$ even if at least 15D upstream. In all cases, it should be recognized that, because of the design differences between magnetic flowmeters, it is not possible to give absolute guidelines that are equally valid for all designs.

A crude rule of thumb to retain errors within $\pm 0.5\%$ would be to use the installation data for a $\beta = 0.2$ orifice plate for "zero additional uncertainty," although the very different operation of the meters makes this very conservative for, say, two bends in perpendicular planes.

12.9 INSTALLATION CONSTRAINTS – FLUID EFFECTS

12.9.1 SLURRIES

The electromagnetic flowmeter has found a special niche (almost a universal solution) for slurry flow measurement and also for some complex non-Newtonian fluids. A slurry will affect the flowmeter because of flow distortion, conductivity variation, and magnetic permeability. The work quoted in Section 12.9.3 enables us to make an informed guess as to the errors, if we have some idea of the size of the conductivity variation due to the presence of slurry. In addition, the distribution in Figure 12.4(b) suggests that a flattened profile, due to a non-Newtonian fluid, say, could cause negative errors. The likely performance on an untried fluid should, therefore, be undertaken with caution.

In a vertical line, which is a common installation position in slurry service, it will also be necessary to allow for any slip that may occur between the conducting fluid and the nonconducting solid. For instance, a settling slurry in a vertical pipe with zero net flow will record the movement upward of water displaced by the settlement downward of the solid matter (as noted by a colleague when working for a flowmeter manufacturer).

A magnetic slurry (Baker and Tarabad 1978) will result in a change of size and distribution of magnetic field. If the flow signal is compensated for change in field current size, then the increase in fluid permeability will cause an uncompensated increase in flux density and flowmeter signal.

Where the flow signal compensation is achieved by a search coil monitoring magnetic flux density, then careful design should ensure that magnetic fluids have a negligible effect on flow signal.

A magnetic slurry may also be deposited in the flowmeter magnetic field. The cure for this may be to run the flowmeter with a lower field strength.

Eren (1995) provided an example of the use of frequency analysis of the output signal generated by DC-type electromagnetic flowmeters. The frequency and amplitude components are dependent on slurry characteristics of the flow. Eren implied that these signals can be related to the density and other characteristics of the slurries.

12.9.2 CHANGE OF FLUID

Although conductivity of the fluid should not affect the flowmeter signal, there has been some suggestion that electrochemical effects of the fluid may do so. There has been some suggestion that this may affect the performance of the modern square wave field excitation flowmeters. However, as noted earlier, Ginesi and Annarummo (1994) commented on the effect of severe variations in conductivity in AC-type magnetic flowmeters but suggested that DC types should be unaffected above a minimum threshold. Few data are available, but Baker et al. (1985) compared the performance of an AC-excited and a DC-field flowmeter for a change in fluid from pH = 7 to pH = 4 and found no significant effects.

12.9.3 NONUNIFORM CONDUCTIVITY

Nonuniformity of conductivity in the flowmeter can cause changes in flowmeter signal due to the changes in the size of the shorting currents which result. Thus a higher conductivity layer near the flowmeter wall will cause higher shorting currents and a reduced flow signal. For certain conditions of turbulent flow, field shape, and

conductivity profile, a signal change of up to 3% may occur (Baker 1970a), and it will be higher for laminar profiles. This is probably most significant in multiphase flow applications with a second nonconducting phase (cf. Ginesi and Annarummo 1994) because the continuous conducting component will generate the same signal as for a uniform fluid provided the fluid is homogeneous.

12.10 MULTIPHASE FLOW

Baker and Deacon (1983) have reported experiments on the behavior of an electromagnetic flowmeter in a vertically upward air-water flow and have shown that volumetric flow of the mixture was measured with errors of less than -1% up to void fractions of 8%. For void fractions greater than this, the negative error increased.

Bernier and Brennen (1983) have analyzed the flowmeter's performance with uniform field and obtained in addition to the following equation for a dispersed second phase without slip:

$$\Delta U_{EE} = BDV_m = \frac{4B}{\pi D} \frac{q_L}{(1 - \alpha)} \quad (12.5)$$

where q_L is the liquid (or conducting phase) volumetric flow rate, and α is the void fraction, the same equation for annular flow. They also show that, for small values of α , the expression is the same for "cylindrical voids" parallel to the flowmeter axis. Their data for bubbly and churn flows fall within $\pm 2\%$ of the predictions up to void fractions of 18%. Murakami et al. (1990) developed an electromagnetic flowmeter with two pairs of electrodes for studying gas-liquid two-phase flow.

The use of the electromagnetic flowmeter for slurry flows has been mentioned earlier in the context of its early development. Peters and Schook (1981) have developed the idea of using the signal fluctuation as a measure of slurry concentration. It is also possible that special design of the magnetic field could be used to obtain concentration information.

Most recently, Krafft et al. (1996) have looked at the passing of gas bubbles through the flowmeter in terms of the electric dipoles that result and have made a start on relating the theoretical signal pattern to experiment (cf. Baker 1977).

12.11 ACCURACY UNDER NORMAL OPERATION

Manufacturers' claims for uncertainty (often referred to as accuracy) tend to lie somewhere near the following:

Uncertainty	Flow range as % of FSD
$\pm 0.3\%$ rate	100–50%
$\pm 0.5\%$ rate	50–15%
$\pm 1.0\%$ rate	15–5%
$\pm 1.5\%$ rate	5–2.5%

In other cases, the value may be given as a percentage of rate or a velocity (e.g., ± 0.015 m/s), whichever is greater.

Repeatability is likely to be of order $\pm 0.2\%$ rate. The analogue output may be less precise than this (e.g., $\pm 0.1\%$ FSD).

One manufacturer, at least, offers meters in the range 2.5–6 mm with uncertainty of order 0.8% rate at the upper end of the flow range.

There has been some discussion in the technical press relating to the accuracy of this instrument over large turndown ranges. There may, therefore, be some research needed to consider the fundamental limits to accuracy related to

- a. flow steadiness and turbulence, and the sensitivity of the meter to profile variation over the Reynolds number range;
- b. the field stability and the usually neglected size of currents flowing in the liquid;
- c. the effect of charge distribution, current size, and conductivity variation; and
- d. the dimensional stability of the flowmeter's components, and compensating systems with variation in ambient conditions.

12.12 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

12.12.1 APPLICATIONS

The applications of this meter for liquid flow measurement are extensive. It is suitable for essentially any conducting liquid, and to my knowledge, it has been unsuccessful on hardly any to which it has been applied. One industrial expert once claimed to me that his only major problem had been icing sugar! Failure is more likely to be due to flow problems or incompatibility. If it is applied to a two- or multicomponent flow, then the continuous component must be conducting, and the signal will, essentially, be due to the velocity of this component. If it is applied to a liquid metal, the physics becomes more complex. A description of this application is beyond the scope of this book, but a review has been given by Baker (1977; cf. Tarabad and Baker 1982).

Applications include viscous fluids, corrosive chemicals, erosive slurries, and start-up and shut-down operations, but the flow tube should run full (some manufacturers offer versions capable of part-filled tubes), and the electrodes should not be open circuited by air bubbles (Ginesi and Annarummo 1994). The tube should, if possible, be vertical with upward flow. If horizontal, the electrodes should be in a horizontal diameter. If the meter is at a low point in the line, then the danger of slurries or other fluids coating the electrodes should be monitored. Coatings can have a conductivity that differs from the fluid and will then behave like a partially conducting wall, as well as changing the internal diameter of the meter. Likelihood of deposition is reduced if the velocity is kept above about 2–3 m/s (6–10 ft/s) through the meter. Electrodes of conical shape reduce deposits, and electrode-cleaning systems can be used. Non-Newtonian fluids may alter the response. Wear of the liner can also result from abrasive slurries from close elbows, etc., and can be reduced by liner protection. The cleaning fluid must also be compatible. Additives can cause nonuniform conductivity.

The AC technology may be preferable for applications with large amounts of entrained air, with slurries that are nonhomogeneous or have nonuniform particle size, high solids content or solids with a tendency to clump, and with pulsating flows. This includes about 15% of the industry including pulp and paper flows. The

DC output can be noisy in these applications, but it is increasingly competitive as an option. Damping of 1–3 s may help.

Effects of radiofrequency interference (RFI) should be virtually eliminated in new meters. However, signal cables must be screened and earthed according to the manufacturer's instructions.

Rose and Vass (1995) discussed adaptations of electromagnetic flow technology to difficult processes:

Chemical	acids, bases, polymers, emulsions, latex solutions
Pharmaceutical	spray coatings, flavoring, hygienic
Mining and minerals	slurries of iron ore, taconite, magnetite, pyrite, copper, alumina
Food and beverage	beer, soda, toothpaste, milk, ice cream, sugar, juices
Water and waste	water, wastewater, raw sewage, primary sludge, digester flows
Pulp and paper	white and black liquor, brown stock, bleaching chemicals, additives
Nuclear fuel processing plant	on both radioactive and nonradioactive liquids (Finlayson 1992)

To this list may be added slag, cement, slurries (abrasive), reagent charging, and special applications like ultra low rates, custody transfer, liquids with steam tracing, blast furnace flows, batching, and erosive liquids.

At high (120 measurements per second) rates, AC metering allows measurement of pulsating flows from pumps, etc.

One manufacturer offers a range of sizes from 2 to 25 mm for milk, and others will offer their own range and specification for hygienic and sanitary applications. The meters are suitable for high speed batch processing of these and other applications and can achieve 0.2% repeatability.

12.12.2 ADVANTAGES

- The theory shows that the response is linear, and the only reason why the meter may not register satisfactorily down to zero flow is zero drift. It is one of the few meters capable of this performance and has been unfairly criticized as a result, for the zero drift that is observable. Modern designs often have a low flow cutoff that avoids this problem.
- Clear bore is of most value in fluids that contain solids or in fluids that can be damaged by passing through constricted flow passages.
- There are no moving parts.
- Sensitivity to upstream fittings is comparable to other meters and only seriously bettered by positive displacement and Coriolis meters or by ultrasonic meters with more than two beams.

12.12.3 DISADVANTAGES

The major disadvantage is its restriction to conducting liquids only, and although laboratory designs have operated on nonconducting liquids (e.g., transformer oil),

to my knowledge only one or two commercial designs have attempted to operate in this regime. However, those interested in this application should consult Al-Rabeh et al. (1978) and Al-Rabeh (1981), although it is unlikely that such a meter could operate with normal gases.

Its sensitivity to upstream distortion has sometimes been suggested as a weakness. In my view, it is one of its strengths. Very few meters operate with less detrimental effect from upstream disturbance than this one. The other feature that was also suggested as a disadvantage from time to time was zero drift, the fact that early designs were prone to large errors at very low flows. Again, there are virtually no flowmeters capable of operating remotely linearly over the range of this one or down to the flow rates of which this is capable. Indeed, at least one commercial meter is claimed to be capable of 1,000:1 turndown.

12.13 CHAPTER CONCLUSIONS

More high-level mathematics has probably been applied to this design of flowmeter than any of the others. The theoretical analysis of the electromagnetic flowmeter is well understood and has been developed by Shercliff, Bevir, Hemp, and others with some contributions from the present author. This analysis allows the response of the flowmeter to various flow profiles to be deduced but also, through the weight function analysis, allows a flowmeter to be designed with a predetermined level of insensitivity to profile change.

The weight function analysis is, itself, very important as it offers an approach that Hemp has exploited for ultrasonic, thermal, and Coriolis meters. A similar approach may be possible in other types of meter.

This meter must be a prime contender in any flow measurement application with a conducting liquid. I have always taken the view that its successful commercial development for nonconducting liquids, although possible, was highly unlikely because the voltages needed to drive the field coils would be incompatible with the intrinsic safety requirements of many nonconducting (oil-based) liquids. However, Al-Rabeh et al. (1978) have given the theory for nonconducting fluids, and Al-Rabeh (1981) tested such a meter. Further work may overcome some of the expected limitations for this meter.

It is unlikely that the limitation of the meter to liquids will be overcome, and we have to look elsewhere for a technique for gas flow measurement with this meter's advantages.

What other developments are likely? The discredited idea of dry calibration could still be developed to provide condition monitoring of the field. For some time I have wondered if the inclusion of a vortex-shedding bar in the meter, to provide another approach to condition monitoring, would be worthwhile, even though it would remove the clear bore. It would provide two independent velocity measures, the electromagnetic steady signal, with a fluctuation superimposed related to the vortex shedding frequency. It might also be possible to include additional electrodes in the shedder to provide a guide to upstream flow distortion.

One very elegant device, which could well offer applications other than that for which it was developed, should be mentioned.

Gray and Sanderson (1970) developed a differential flowmeter using two adjacent rectangular flow channels so that the same magnetic field traversed both and the electrodes were so connected as to give the difference of the two signals due to the two flows and hence the differential flow. They claimed to be able to measure differences of 1–10 cm³/min in a flow of 500 cm³/min to $\pm 10\%$ and possibly even to measure down to 0.5 cm³/min.

Another line of development, exemplified by Horner et al. (1996), is the use of rotating fields or other multiple coil arrangements. In their case, two pairs of coils and 16 electrodes appear to have been used to improve performance on nonaxisymmetric flows.

With the developments in design, materials technology, manufacture, and signal processing, this meter has become of increasingly competitive price, usually lower maintenance than those with moving parts, and steadily improving performance. This has included

- ❑ improved accuracy,
- ❑ extended turndown,
- ❑ reduced size and weight,
- ❑ reduced power, and
- ❑ microprocessor-based intelligence.

This has led to the possibility of battery operation with time periods up to 10 years or more and turndown claims of 1000 : 1. The potential of the meter has also encouraged the use of new materials, especially ceramics, to increase stability and, by careful design, to reduce noise from radiofrequency and electrochemical sources.

Independent verification of the remarkable performance claims for these devices would be very valuable.

APPENDIX 12.A

Brief Review of Theory

The theory set out in this appendix has been extended to other flowmeters, and the relevant papers are given elsewhere in this book.

12.A.1 INTRODUCTION

We make use of a simplified form of Maxwell's equations

$$\nabla \times \mathbf{B} = \mu \mathbf{j} \quad (12.A.1)$$

$$\nabla \cdot \mathbf{B} = 0 \quad (12.A.2)$$

$$\nabla \times \mathbf{E} = -\mathbf{B} \quad (12.A.3)$$

and Ohm's law with velocity included

$$\mathbf{j} = \sigma(\mathbf{E} + \mathbf{V} \times \mathbf{B}) \quad (12.A.4)$$

where $\mathbf{B}$ is magnetic flux density vector, μ is permeability, $\mathbf{j}$ is current density vector, $\mathbf{E}$ is electric field vector, σ is conductivity, and $\mathbf{V}$ is velocity vector.

We may ignore $\dot{\mathbf{B}}$ because the effect of this term may be eliminated by careful mechanical and electrical design. This allows the introduction of the scalar electric potential U :

$$\mathbf{j} = \sigma(-\nabla U + \mathbf{V} \times \mathbf{B}) \quad (12.A.5)$$

Taking the divergence of this equation and making use of the fact that the magnetic field produced by external coils will be virtually unaltered by the very small currents in the fluid so that

$$\nabla \times \mathbf{B} = 0 \quad (12.A.6)$$

we obtain the flowmeter equation

$$\nabla^2 U = \mathbf{B} \cdot \nabla \times \mathbf{V} \quad (12.A.7)$$

and boundary conditions for a nonconducting wall of

$$\frac{\partial U}{\partial n} = (\mathbf{V} \times \mathbf{B})_{\perp} \quad (12.A.8)$$

when n is the coordinate perpendicular to the wall. Since the velocity at the wall is normally parallel to the wall, this will become, for cylindrical coordinates,

$$\frac{\partial U}{\partial r} = -V_z B_{\theta}$$

This condition is assumed to exist everywhere on the wall of a conventional flowmeter with small electrodes. For a nonslip condition on velocity, the right-hand side is zero. However, if the electrodes are large, then the simplest assumption is that the fluid at the electrode surface is at the same potential as the electrode. The electrode is assumed to have the same potential at every point on its surface, and the total current flow into the electrode is assumed to be zero. If a contact resistance exists at the electrode surface, this will have to be allowed for in relating the value of the voltage on the electrode and in the fluid.

One important observation from Equations (12.A.2) and (12.A.6) is that $\mathbf{B}$ in the fluid may be expressed as the gradient of a scalar potential Φ ; hence

$$\nabla^2 \Phi = 0 \quad (12.A.9)$$

and we observe that the magnetic field distribution obeys Laplace's equation.

The boundary conditions on the magnetic field will usually be those existing on the pole pieces and at the surface of the excitation coils or the slots into which they fit. Thus, Φ will have a constant positive value on the surface of one pole and an equal but negative value on the other. The size of the surface potential on the remainder of the yoke may be more complicated unless the coil winding slots are thin and the value of Φ can be assumed to be zero on the yoke with discontinuous changes at the poles.

So far we have assumed steady conditions. Often, however, the excitation is sinusoidal, and the voltages induced by transformer effect may not be negligible. We can demonstrate that the analysis so far is valid by noting that if we associate a sinusoidal excitation of known phase with the applied magnetic field, all the equations

relating to flow-induced signal will have the same phase as the applied field, and the electric field derived from Equation (12.A.3) will have a $\pi/2$ phase shift and may be eliminated by suitable electronic design. Additionally, the effect of this spurious field may be reduced by improving the symmetry of the flowmeter.

12.A.2 ELECTRIC POTENTIAL THEORY

It may be shown (Baker 1968) that the solution of Equation (12.A.7) for a circular pipe with axisymmetric flow profile is given by

$$U = \frac{1}{2} \int_0^r [V(\sqrt{\rho r}) + V(a\sqrt{\rho/r})] B_\theta(\rho, \theta) d\rho \quad (12.A.10)$$

where r is the radial coordinate, $V(r)$ is the axisymmetric flow profile, θ is the azimuthal coordinate, and $B_\theta(r, \theta)$ is the azimuthal component of magnetic field, which varies with r and θ and is constant with axial position z . ρ is a dummy variable, and a is the pipe radius.

The potential difference between diametrically opposed electrodes ΔU_{EE} is then given by

$$\Delta U_{EE} = 2 \int_0^a V(\sqrt{ra}) B_\theta(r, \theta) dr \quad (12.A.11)$$

For a uniform magnetic field B , this becomes

$$\begin{aligned} \Delta U_{EE} &= \frac{4B}{a} \int_0^a r V(r) dr \\ &= BDV_m \end{aligned} \quad (12.A.12)$$

where D is the pipe diameter, and V_m is the mean velocity. This equation resulted in the high expectations for the device that offered true bulk flow measurement. However, the assumptions – uniform field and axisymmetric flow profile – were too constraining in practice.

12.A.3 DEVELOPMENT OF THE WEIGHT VECTOR THEORY

Shercliff showed that the response to a point-electrode uniform-field electromagnetic flowmeter, when subjected to an arbitrary rectilinear flow profile, could be represented by a weighting function W given by

$$W = \frac{a^4 + a^2 r^2 \cos 2\theta}{a^4 + 2a^2 r^2 \cos 2\theta + r^4}$$

where θ is zero in the direction of the magnetic flux. This is the function shown in Figure 12.4(a).

Bevir (1970) developed the powerful concept of the weight vector $\mathbf{W}$. Bevir introduced the concept of the virtual current $\mathbf{j}_v$, which is related to a potential function U_v by

$$\mathbf{j}_v = -\sigma \nabla U_v \quad (12.A.13)$$

and is the current density that would exist in the flow tube in the absence of magnetic field and flow if unit current entered by one electrode and left by the other.

He showed that (cf. Baker 1982 for a simplified derivation)

$$\Delta U_{EE} = \iiint_{\text{Flow tube volume}} \mathbf{V} \cdot \mathbf{B} \times \mathbf{j}_V d\tau \quad (12.A.14)$$

where the integration is taken over the whole volume of the flowmeter tube. Bevir introduced $\mathbf{W}$ as

$$\mathbf{W} = \mathbf{B} \times \mathbf{j}_V \quad (12.A.15)$$

and hence

$$\Delta U_{EE} = \iiint_{\text{Flow tube volume}} \mathbf{V} \cdot \mathbf{W} d\tau \quad (12.A.16)$$

Bevir then showed that the necessary and sufficient condition for an ideal flowmeter, one which measured the mean flow regardless of flow profile, was

$$\nabla \times \mathbf{W} = 0 \quad (12.A.17)$$

An important point to note is that Equations (12.A.16) and (12.A.17) possess a generality beyond electromagnetic flow measurement. Bevir (1970) proposed a rectangular section flowmeter that satisfied Equation (12.A.17), and he carried out extreme tests to demonstrate its performance. Using noncontacting electrodes and a special magnetic field distribution, Hemp (with Al-Khazraji 1980 and with Sanderson 1981) has designed a flowmeter, which in theory should be virtually insensitive to flow profile effects.

12.A.4 RECTILINEAR WEIGHT FUNCTION

However, most designs have assumed a rectilinear flow and have used an appropriate weight function W'

$$W'(r, \theta) = \int_{-\infty}^{\infty} W_z dz \quad (12.A.18)$$

where W_z is the axial component of $\mathbf{W}$. This results in

$$\Delta U_{EE} = \iint_{\text{Flow tube cross-section}} r V_z(r, \theta) W'(r, \theta) d\theta dr \quad (12.A.19)$$

This is essentially Shercliff's (1962) idea, which resulted in the well-known distribution for a uniform magnetic field [Figure 12.4(a)].

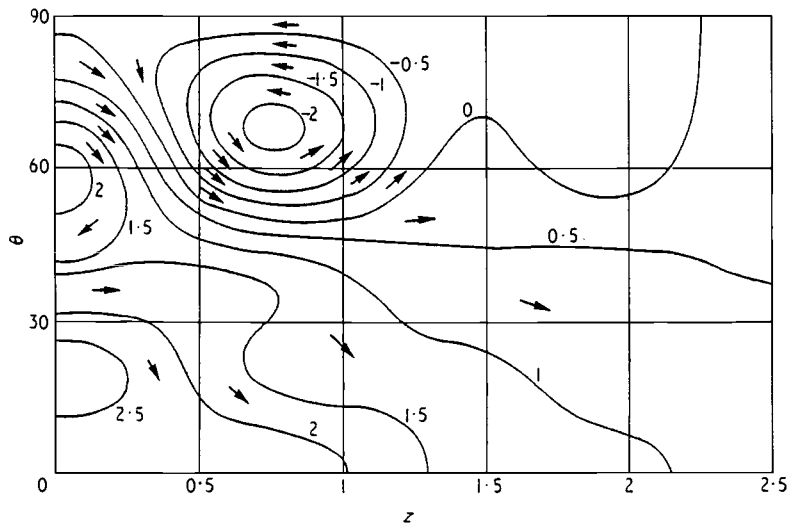
Many weight function distributions have appeared in the literature since Bevir's work, and we have noted three of these:

- For a point-electrode flowmeter with rectangular coils and circular yoke (Al-Rabeh and Baker 1979) [Figure 12.4(b)]. This type of flowmeter is of interest in that it approximates to some industrial flowmeters.

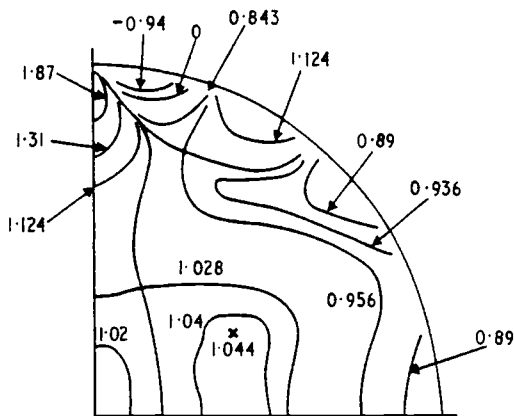
- ❑ For a point-electrode flowmeter with diamond coils [Figure 12.4(c)] and circular yoke, which approximates to one introduced in the 1960s and claimed to have a reduced sensitivity to flow profile distortion.
- ❑ For a large electrode flowmeter with rectangular coils and yoke (Al-Khazraji and Baker 1979), shown diagrammatically in Figure 12.13, for which the weight function is reproduced in Figure 12.4(d).

The improvement obtained in the uniformity of W' by use of diamond coils instead of rectangular coils is very noticeable, and there is even more improvement by using large electrodes. However, a major disadvantage of these is the change in sensitivity resulting from electrode fouling (Al-Khazraji and Baker 1979).

Hemp (1975) attempted to minimize the variation of weight function with the copper strip pattern in Figure 12.A.1(a). Figure 12.A.1(b) shows the resulting weight function distribution, which should be compared with those in Figure 12.4. The axisymmetric weight function was approximately constant for about 87% of the pipe falling to zero at the wall (Figure 12.A.2).



(a)



(b)

Figure 12.A.1. Hemp's (1975) improved magnetic field design: **(a)** Copper strip pattern associated with the improved magnetic field. The number associated with each line is the value of the magnetic field potential function, and the arrows, which are equal in size in each strip, show the direction of the current; **(b)** Weight function diagram for the improved magnetic field (refer to the original paper for further details; reproduced with permission from the author and IoP publishing).

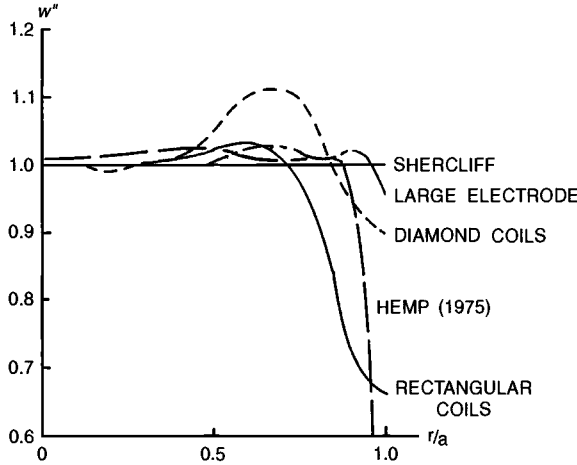


Figure 12.A.2. Axisymmetric weight functions based on Figures 12.4 and 12.A.1.

12.A.5 AXISYMMETRIC WEIGHT FUNCTION

In concluding this section on the theory, it is useful to refer briefly to the axisymmetric weight function. This is given by

$$W''(r) = \frac{1}{2\pi} \int_0^{2\pi} \int_{-\infty}^{\infty} W_z dz d\theta \quad (12.A.20)$$

Its value for a uniform field is constant and unity. However, for most practical fields, it falls in value at the outside of the tube. This trend is illustrated in Figure 12.A.2, which gives plots for Shercliff's uniform field weight function [Figure 12.4(a)] and by interpolation for the others in Figure 12.4. The signal is given by

$$\Delta U_{EE} = 2\pi \int_0^a r W''(r) V_z(r) dr \quad (12.A.21)$$

Thus as flow profiles change with increasing Reynolds number from being more peaked in the center of the pipe to more uniform across the pipe, the signal will fall below direct proportionality.

12.A.6 PERFORMANCE PREDICTION

Apart from the design problem, but related to it, has been the accuracy with which performance could be predicted from theory. This is the problem of dry calibration. Equation (12.A.12) suggested a very simple prediction for a uniform field flowmeter. However, the precision of such prediction has always been uncertain. Bevir et al. (1981) have described a very elegant piece of work in which magnetic field measurements around the tube wall were used to compute the weight function, and very good agreement was obtained between prediction based on pipe flow profiles and

actual meter response. They concluded that sensitivities would be predicted for most flowmeters by their procedures with an error of not more than 0.5%.

Al-Rabeh and Baker (1986) have attempted to predict the performance of a flowmeter from its magnet design. However, although good qualitative agreement on distribution of field was obtained, they found a discrepancy between predicted and actual field size. Thus with care, high accuracy prediction of flow signal may be achieved from actual magnetic field distribution.

Others have developed this approach by measuring and/or predicting the field at the pipe wall and using this to calculate the weight function.

12.A.7 FURTHER EXTENSIONS TO THE THEORY

The reader who is interested in various extensions of the theory is referred to the following papers by Hemp and co-workers.

Hemp and Wyatt (1981) explored the use of a “worst flow” to compare electromagnetic flowmeter performance.

Hemp and Versteeg (1986) developed the analysis of the flowmeter performance based on the magnetic field on the flowmeter surface. This approach would allow ‘dry calibration’ of a meter based on magnetic field measurements on a tube next to the inner surface.

Krafft et al. (1996) discussed the use of the transformer effect signal due to bubbles to obtain both bubble velocity and velocity of the continuous phase of a bubbly flow.

Hemp (1991) developed the theory of eddy currents in electromagnetic flowmeters and suggested a means of self-calibration. Hemp also (1994c) looked at errors in potential measurement due to nonuniform contact impedance at the electrodes.

Ultrasonic Flowmeters

13.1 INTRODUCTION

The first proposal for the use of ultrasound for flow measurement, according to Thompson (1978), seems to have been in a German patent of 1928. It was not until after 1945 that the idea became more widely proposed. But not until the development of piezoelectric transducers in the past 40 years or so have ultrasonic applications become really attractive. Fischbacker (1959) provided an early review of ultrasonic flowmeters in which, essentially, the transit-time, sing-around, and beam deflection methods were mentioned. He also referred to phase-difference measurement, means of obtaining sound speed from time measurement and impedance, and how to obtain density. He saw the advantages of off-axis paths.

Sanderson and Hemp's (1981) review is still a useful source of information on the subject. The ultrasonic flowmeter's attraction as a flow measurement device is its linearity, lack of obstruction to flow, and, in contrast to the magnetic flowmeter, its ability to measure the flow of gases.

In this chapter, we shall consider three main types of ultrasonic flowmeter, and it is important to understand the strengths and weaknesses of each and that they are very different in performance and application.

The transit-time flowmeter, or time-of-flight flowmeter, is the most accurate of the family and is available as a spool piece meter for liquids and gases or as a clamp-on design for liquids only. It can also be retrofitted into a pipe. Measurement uncertainty will be from a fraction of a percent to about 5%. A transducer in direct contact with the fluid is said to be wetted, as compared to one that is fixed to the outside of the pipe or has a protective layer between it and the fluid. Transducers for gases must be in contact with the gas (i.e., wetted, a rather inappropriate word for this context!).

The doppler meter is a very different device, and it may be unfair to call it a flowmeter. Its performance is very sensitive to installation, and it has been oversold to such an extent that many people assume that, when an ultrasonic flowmeter is mentioned, it is one of these. Measurement uncertainty is unlikely to be better than about $\pm 2\%$, and at worst it is indeterminate. Having said this, it has a very important niche in some liquid flow applications where there is an adequate second component to provide reflection of the beam.

The cross-correlation flowmeter should be capable of reasonable precision (between the preceding two devices), requires a disturbed liquid or multiphase flow to operate satisfactorily, is likely to be much more expensive than either of the other types, and is mainly of interest for measurement in multiphase flows.

It is not unusual for other meters that have ultrasonic sensing to be referred to as ultrasonic meters. One such meter is the vortex-shedding meter with ultrasonic sensing.

Beyond these, there is the possibility of using the deflection of the beam to measure flow rate, but to my knowledge this has not been done commercially. There is also the possibility of measuring density and so turning the meter into a mass meter.

It is important not to confuse the transit-time meter with the doppler meter and to appreciate that the transit-time ultrasonic meter is a far more accurate instrument than the doppler meter and should be considered seriously in a wide range of applications, especially involving nonconducting liquids or gases.

We shall treat each type of ultrasonic flowmeter in turn. All are based on the fact that ultrasound is made up of acoustic waves at frequencies above the audible range. As a form of sound, ultrasound waves travel with the speed of sound relative to the medium and consist of pressure perturbations in gases and liquids. In solids, the connecting wave can also move as a shear wave, due to the elasticity of the solid in shear.

In Figure 13.1(a), we note that the ultrasound is carried by the fluid, so that its speed is the sum or difference of its own speed c and that of the fluid V . This is the basis of the transit-time meter, which uses the difference in time of transit in an upstream and a downstream direction. If the sound is crossing the flow, then the apparent speed is obtained from the hypotenuse of the triangle in the third example of Figure 13.1(a).

In flowmeters, the continuous wave [Figure 13.1(b)] is sometimes used. The diagram shows the main features of the wave pattern, with the wave length and the higher and lower pressure regions. The frequency f is related to the sound speed c and wavelength λ by $f = c/\lambda$. The intensity of the transmission varies according to the angle off the perpendicular axis through the transducer. Three patterns are shown in Figure 13.1(c) for three different ratios of the transducer radius a to the wavelength. The higher the frequency and the shorter the wavelength, the more pencil-like the transmitted beam. A narrow beam is preferable to a wide one because it is less likely to cause spurious reflections, and its path will be better defined. For a speed of sound in water of 1414 m/s, a frequency of 400 kHz will result in a wavelength of about 3.5 mm, whereas 1 MHz will result in 1.4 mm. For a speed of sound in air of 343 m/s, a frequency of 40 kHz will result in a wavelength of about 8 mm, whereas 100 kHz will result in about 3.4 mm. Thus for air with $\lambda = 2a$, the diameter of the piezoelectric crystal to obtain the middle distribution in Figure 13.1(c) at 40 kHz will be about 8 mm, whereas the crystal diameters for water are likely to be such that $\lambda \ll a$ and result in a narrow intense beam.

More commonly the ultrasound is sent in a small burst of waves. Figure 13.1(d) shows an idealized diagram of a short burst of waves transmitted across the flow.

However, the sweeping effect [Figure 13.1(e)] will mean that the direction of the beam will be a compromise between high and low flow rates to ensure that it arrives at the receiving transducer. Another feature of the bending of the beam is that there is a tendency for downstream sound to be reinforced and for upstream sound to be diffused and weakened (cf. Baker 1996, page 2).

When ultrasound crosses an interface between two different fluids or between a fluid and a solid, there is both transmission and reflection at the interface. The transmission will be greater if the density times wave speed of the two materials

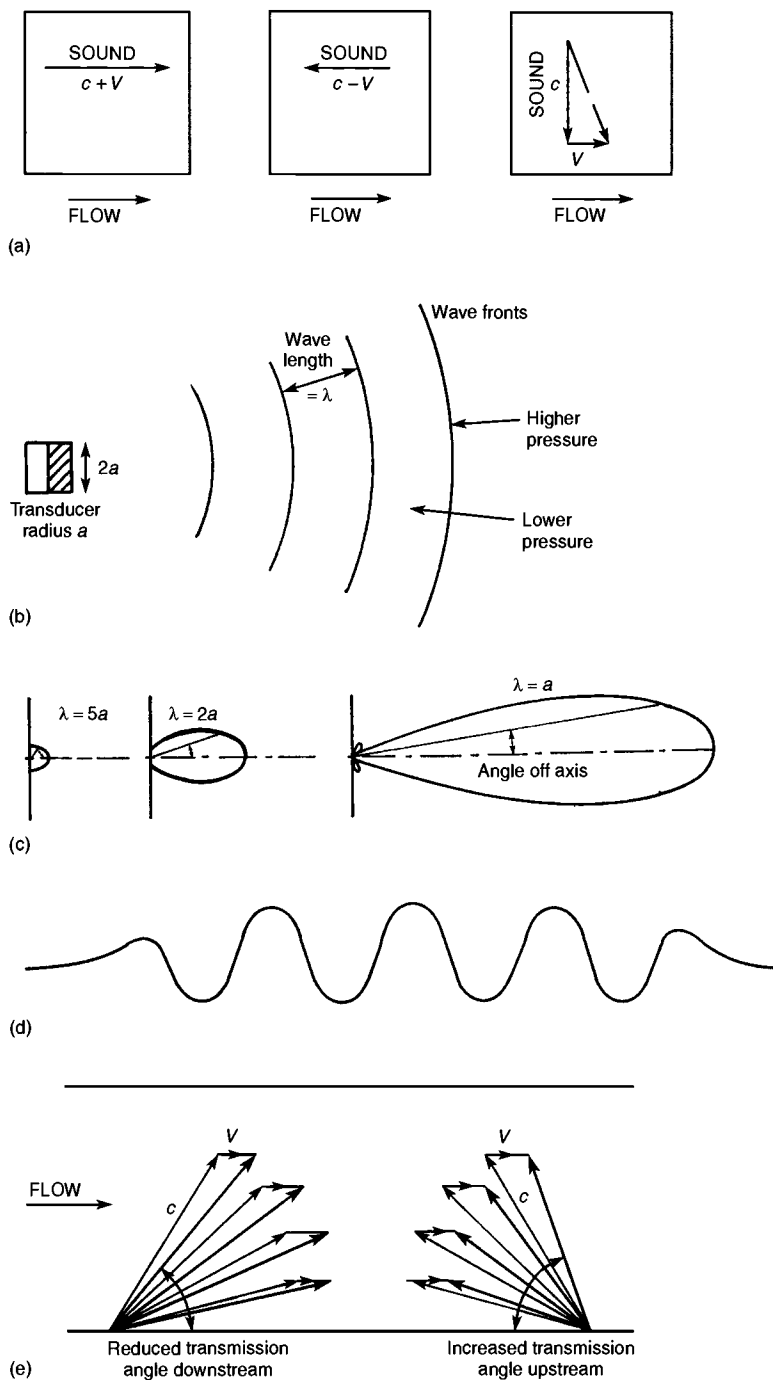


Figure 13.1. (a) Apparent sound speed as viewed by an observer outside the moving medium. (b) Ultrasonic waves move forward with the speed of sound and with a wavelength λ from a source. (c) Radiated intensity variation with angle from a piston transducer set in a plane wall (after Morse and Ingard 1968). (d) Wave packet. (e) Ultrasound waves (like sound waves) can be bent by the flow and reinforced or dispersed.

are similar in size. Thus, it proves difficult to transmit ultrasound from air, through metal, and then into a transducer. On the other hand, it is possible to transmit from liquid through a solid and then to a transducer. The relevant characteristic of the materials is the impedance.

Impedance = Density of material through which
ultrasound is transmitted $\times$ Velocity of
ultrasound in the material

$$Z = \rho \times c \quad (13.1)$$

and using approximate values at 20°C for air

$$Z = 1.19 \times 343 = 408 \text{ kgm}^{-2}\text{s}^{-1}$$

for water

$$Z = 1000 \times 1414 = 1.41 \times 10^6 \text{ kgm}^{-2}\text{s}^{-1}$$

and for steel

$$Z = 8000 \times 5625 = 45 \times 10^6 \text{ kgm}^{-2}\text{s}^{-1}$$

The proportion of ultrasound power transmitted is given by Asher (1983; cf. Asher 1997) as

$$P_T = \frac{4Z_1 Z_2}{(Z_1 + Z_2)^2} \quad (13.2)$$

and that reflected is

$$P_R = \frac{(Z_1 - Z_2)^2}{(Z_1 + Z_2)^2} \quad (13.3)$$

Thus whereas transmission from steel to water is about 12%, that from air to steel is negligible. In discussion at a symposium on flow measurement (*Trans. Soc. Instrum. Eng.*, Vol. 11, June 1959), Bertele of ICI raised the problem of getting enough energy into the gas. As a result, we shall find that transducers for gas flow measurement must be wetted.

The final point to make is that attenuation increases with increasing frequency and tends to be greater in gas than in liquid.

13.2 TRANSIT-TIME FLOWMETERS

In this section, the operation is described, and the key flowmeter equations are given. The simple theory of these devices is given in Appendix 13.A.1–13.A.3, with a brief reference to some fuller solutions.

13.2.1 SIMPLE EXPLANATION

The transit-time flowmeter depends on the slight difference in time taken for an ultrasound wave to travel upstream rather than downstream. Thus waves are launched

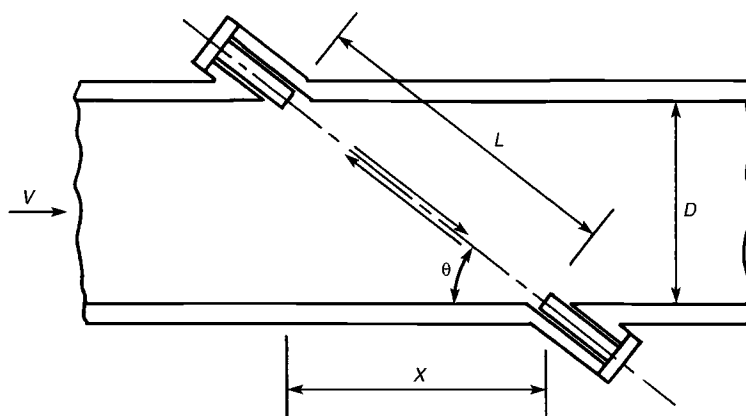


Figure 13.2. Geometry of the transit-time flowmeter.

each way, their time of transit is measured, and the difference can be related to the speed of the flow. Apart from the smallest bore flowmeters, it is usual to send the beam across the flow but not at right angles to the flow, so that there is a component of the fluid velocity along the path of the acoustic beam. Figure 13.2 is a simple diagram of an ultrasonic flowmeter geometry.

13.2.2 FLOWMETER EQUATION AND THE MEASUREMENT OF SOUND SPEED

We can show that for a uniform velocity in the pipe (Appendix 13.A.1), we can obtain an expression in terms of the transit times of upstream t_u and downstream t_d wave pulses, and the time difference Δt :

$$V = \frac{L^2 \Delta t}{2X t_u t_d} \quad (13.4)$$

where L is the path length, and X is the axial spacing between transducers. This is one form of the equation, and others are set out in Appendix 13.A.1.

The reason for the product $t_u t_d$ in Equation (13.4) is to eliminate the speed of sound, which otherwise appears in the equation. By measuring the mean time of transit, and $\sqrt{t_u t_d}$ approximates to this, and by knowing the transit distance, we can obtain the speed of sound.

In the transit-time method, pulses of sound of a few cycles in length are transmitted in each direction (Figure 13.3), and the time taken for them to reach the receiving transducer is measured. Hemp (1982) gave examples of pulses:

Water metering:	50 mm diameter pipe
	Pulse of 6 cycles
	1-MHz carrier
	1.5-mm wavelength
	Wave packet about 9 mm long
Domestic gas:	12-mm diameter pipe
	Pulse about 13 cycles
	2-mm wavelength
	Wavepacket about 26 mm long

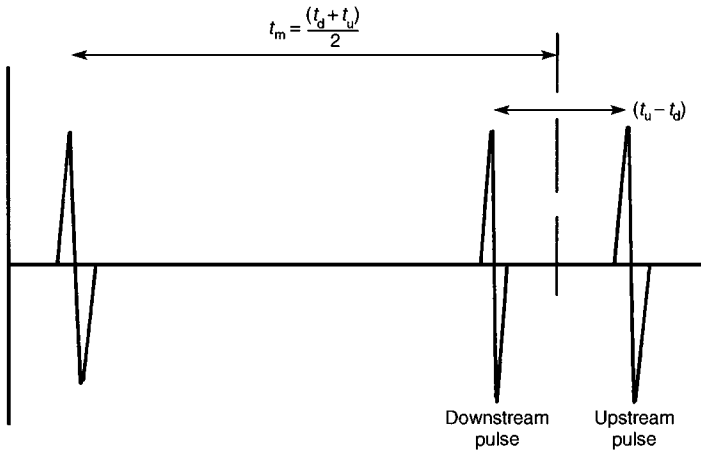


Figure 13.3. Diagram to explain transit-time measurement.

We obtain from Equation (13.4), assuming V is constant across the pipe,

$$\begin{aligned} q_v &= \frac{\pi D^2 V}{4} \\ &= \frac{\pi D^2 L^2 \Delta t}{8 X t_u t_d} \end{aligned} \quad (13.5)$$

It is important to appreciate the value of the times involved in these measurements. If the flowmeter is to discriminate to better than 1%, we may define $E(\Delta t)$ as $\Delta t/100$, and for two meters, we obtain the values in Table 13.1 (Baker 1988/9).

Table 13.1 highlights the problem of the very small time differences we need to measure in the transit-time flowmeter. For the smaller tube, this is to better than 1 ns. However, one advantage of this system is that any echoes or standing waves have time to die down, another advantage is that the same path can be used for each direction of the pulse, and, finally, the individual measurements can be used as a tolerance check to ensure that spurious measurements are rejected (Scott 1984).

An alternative way to eliminate sound speed is the sing-around method (Figure 13.4) discussed by Suzuki et al. (1975, cf. 1972). In this method, each of the paths, upstream and downstream, is operated in such a way that when the pulse is received by the receiving transducer, it triggers a new pulse from the transmitting transducer. Thus the frequency of pulses is dependent on the velocity of sound and

Table 13.1. Transit-time meter times

Diameter (mm)	$(t_u + t_d)/2$ (s)	Δt (s)	$E(\Delta t)$ (s)
100	10^{-4}	10^{-7}	10^{-9}
300	3×10^{-4}	3×10^{-7}	3×10^{-9}

$\theta = 45^\circ$, $V = 1$ m/s, and c is taken as 1430 m/s, the value for water at 4°C .

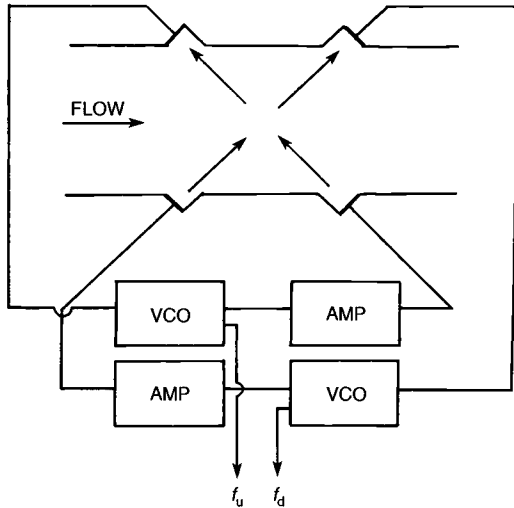


Figure 13.4. Diagram to explain sing-around system.

of the flow. We find that the frequency of pulses is given by

$$\Delta f = 2XV/L^2 \quad (13.6)$$

which has the virtue of relating V to the frequency difference without requiring the value of c , the sound speed. Again we can obtain typical values for the frequencies from Table 13.2 (Baker 1988/9). Making the approximation that V is constant across the pipe, we obtain

$$q_v = \frac{\pi D^2 L^2 \Delta f}{8X} \quad (13.7)$$

The limitation of the sing-around method relates more to the very low frequencies and the long time periods needed to achieve sufficient accuracy. To achieve

1% uncertainty, we shall need to have a measuring period of greater than 1 s for the most favorable of the cases in Table 13.2.

Timing can also be achieved in some cases by obtaining the phase shift between the two arriving waves to obtain the time difference (Baker and Thompson 1975). Other problems such as standing waves and spurious transmission paths that cannot be gated out may need to be considered in relation to any of the methods chosen. Unwanted reflections within the pipe and their effect on the measurement uncertainty may be reduced by the use of higher frequencies. These attenuate reflections, but they also attenuate the signal. Cunningham and Astami (1993) produced curves of upper and lower frequency limits against pipe diameter for various reflection settling times. The upper limit ensures that signal attenuation is not too great, whereas the lower limit is to ensure that attenuation of reflections is sufficient. They reckoned that the received signal should be at least 30% of that transmitted.

An alternative approach, rather than measuring the value of c , is to use the fact that $c^2 = 1/(k_s \rho)$ for a particular liquid and that $c^2 = \gamma p/\rho$ for a particular gas [Equation (13.A.3)]. For a liquid,

$$\begin{aligned} \Delta t &= 2k_s X \rho V \\ &= \frac{8k_s X q_m}{\pi D^2} \end{aligned} \quad (13.8)$$

Table 13.2. Sing-around meter frequencies

Diameter (mm)	V = 1 m/s		V = 10 m/s	
	f_d (Hz)	Δf (Hz)	f_u (Hz)	Δf (Hz)
100	10,117	10	10,162	100
300	3,372	3.3	3,387	33

$\theta = 45^\circ$, $V = 1$ m/s, and c is taken as 1,430 m/s, the value for water at 4°C.

Table 13.3. γ for various gases from Haywood (1968)

Type of Gas (number of atoms)	Examples of Gases	Value of γ	Mean Value of γ^a
Monatomic	Argon	1.67	1.67
Diatomic	Air	1.40	1.40
	Nitrogen	1.40	
	Oxygen	1.40	
3–5 atoms			1.29
3 atoms	Carbon dioxide	1.31	1.17
5 atoms	Methane	1.31	
8–9 atoms			
8 atoms	Ethane	1.19	1.15
9 atoms	Propylene	1.15	

^a Based on number of atoms.

where the adiabatic compressibility would need to be known and, because it varies with temperature, would need to include some temperature compensation. For a gas,

$$\begin{aligned}\Delta t &= \frac{2\rho XV}{\gamma p} \\ &= \frac{8Xq_m}{\pi D^2 \gamma p}\end{aligned}\quad (13.9)$$

thus relating the mass flow rate of a gas to the time difference, the pressure of the gas, and the geometry of the flow tube. Table 13.3 gives values of γ , and a flowmeter using this method would need a selection switch for the type of gas in addition to the pressure measurement facility.

An alternative method to obtain the mass flow is to obtain the density from the value of the impedance discussed earlier. Guilbert and Sanderson (1996a) have described such an ultrasonic mass flowmeter, which combines a fairly standard configuration of ultrasonic flowmeter with an impedance density cell, in which the impedance of the liquid is obtained through the interface between the liquid and a material that divides the cell into compartments. The material selected was polyamide/imide. They also used the time between reflected signals to obtain the sound speed in the divider and hence its impedance (cf. Chapman and Etheridge (1993) who described the evolution of the ultrasonic flowmeters for gas measurement).

13.2.3 EFFECT OF FLOW PROFILE AND USE OF MULTIPLE PATHS

The preceding simple analysis has assumed that the flow profile is uniform. In fact, if we allow for varying velocity along the path of the ultrasonic beam, Equation (13.4) may be rewritten in the form

$$V_m = \frac{L^2 \Delta t}{2X t_u t_d} \quad (13.10)$$

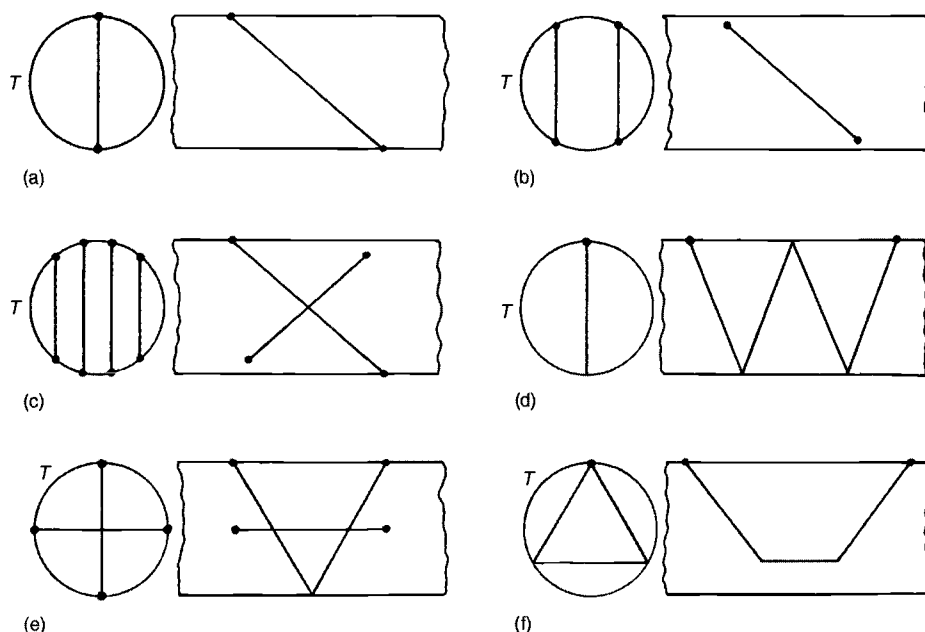


Figure 13.5. Acoustic beam positions (T, top of pipe, indicates possible preferred orientation in liquid flow): **(a)** Diametral path; **(b)** Mid radius paths; **(c)** Four path; **(d)** Reflex mode with W path; **(e)** Reflex mode with two V paths; **(f)** Reflex mode with triangular path.

where V_m is the mean velocity along the path of the beam, since the flowmeter averages velocity along the path of the beam. In general, this average does not correspond to the area average of the velocity required for an ideal flowmeter. Unfortunately, for circular pipes with paths crossing the diameter [Figure 13.5(a)], the mean velocity along the path gives too heavy an emphasis to the flow at the center of the pipe. This implies that change of profile from laminar to turbulent and through the turbulent range will affect the calibration. It is easy to show (Kritz 1955) that the effect of integrating a laminar profile across a diameter of the pipe causes a signal about 33% high due to the overweighting that the velocities near the axis receive, compared with a uniform profile. Through the turbulent range, using the $1/n$ power law (see Chapter 2) for a Re number variation between 10^3 and 10^6 , a variation of about 3% occurs. Apart from Kritz, others have provided calibration factors (Kivilis and Reshetnikov 1965, Del Grosso and Spurlock 1957). Thus for a meter that is constructed this way, there is an inherent variation in calibration factor, which cannot be wholly allowed for in the transmitter circuitry due to Reynolds number dependence on factors other than velocity only. Scott (1984) made the point that a change in viscosity from 1 to 10 cP causes a change in profile, which will account for a 1% error.

One way to improve this is to offset the paths from the axis [Figure 13.5(b)], in which case a position at about the mid radius is found to be a good compromise (Appendix 13.A.2). Baker and Thompson (1975) showed the effect of offset paths (Figure 13.6) for varying Reynolds number on the signal with the paths offset at $0.5R$, $0.505R$, and $0.523R$ from the axis. This shows that a position of $0.523R$ is ideal for the turbulent range, whereas $0.505R$ is probably a reasonable compromise for

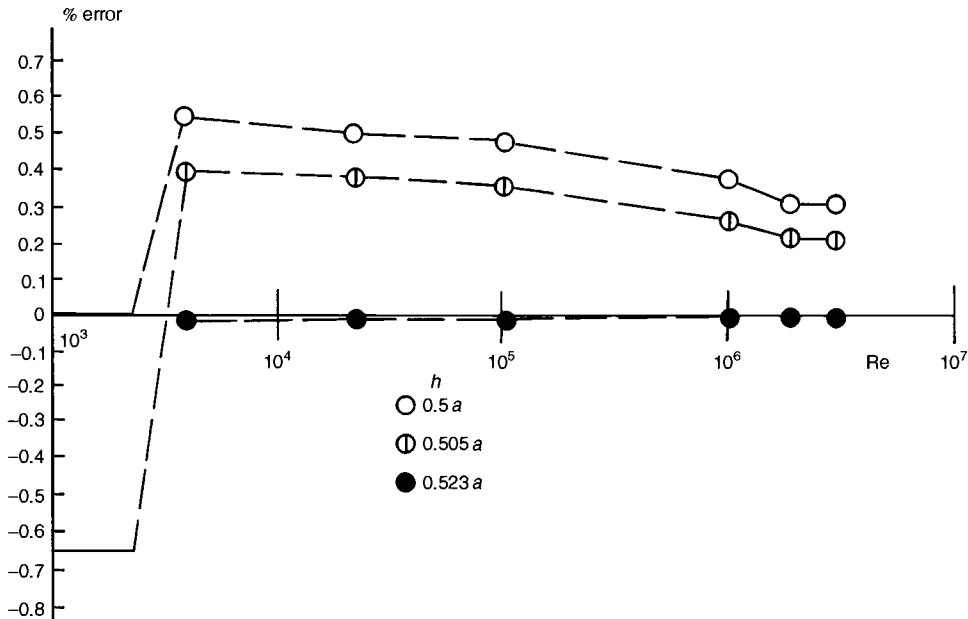


Figure 13.6. Calculated values of the percentage error against Reynolds number for three positions of the acoustic paths (after Baker and Thompson 1975).

the whole Reynolds number range. They also suggested that the meter should have low sensitivity to swirl and to flows that have symmetry of size but opposite sense about the axis.

Hastings (1968, 1970) used four paths [Figure 13.5(c)] and selected their positions so that a Gaussian Quadrature integration (a numerical method for integrating general curves, which is similar but more complicated than Simpson's rule – see Appendix 13.A.2) could be used to give a correct mean velocity signal for profiles that are formed from polynomials of seventh power or less. Again the experimental evidence since these early papers confirms the quality of the integration achieved and the potential accuracy of the design.

Using the velocity measured on multiple paths, Vaterlaus (1995) gave the following positions for multiple paths and the weight to be given to each:

Planes	Distance from center plane	Weight
2	± 0.520	0.5
3	0	0.444
	± 0.774	0.277
4	± 0.339	0.326
	± 0.861	0.174

The value given for two paths agrees well with Baker and Thompson (1975). Taylor and Cassidy (1994) quoted weightings for four planes, which, to three significant figures, agreed with these (cf. Pannell et al. 1990 on a generalized approach to positioning chordal paths).

A variation on the positioning of the paths uses internal reflection of the beams to allow transmitter and receiver to be on the same side of the pipe [Figures 13.5(d, e)]. This is known as reflex mode. This can be either *V* with one reflection or *W* with three reflections. It can also be used to create subtle reflections, which essentially give the path position the benefits of off-axis integration [Figure 13.5(f)] and multiple paths around the pipe to give even better integration of the profile. An example of this results in the paths lying on an equilateral triangle if viewed axially down the pipe, thus using the midradius point but also covering a large area of the tube (Drenth and Huijsmans 1993).

Jackson et al. (1989) developed an ultrasonic flowmeter using one transmitting transducer and a lens system made of Perspex, which produced a diverging ultrasound beam detected by three widely spaced receiving transducers. The chords of the three paths are all designed to be of different lengths and angles. In the test meter, a further path was used across the flow to obtain the sound speed. It is not obvious that this design contributes significantly to the normal design with three complete paths or, indeed, the designs using reflecting beams to obtain off-diametral paths. The advantage of reciprocating paths is that the sound speed is obtained using the mean.

Jackson et al. (1991, cf. 1989) described a three-path meter where the path angles were chosen to obtain information about the flow profile. The paths were in a diametral plane and with chordal paths on each side at angles of 44° and 52° to the diametral plane. For axisymmetric profiles, performance was of order 2%. Asymmetry was detected unless there was a symmetry about the diametral plane, but precise flow measurement with asymmetry was not obtained.

Van Dellen (1991) of Daniel Industries, Inc., demonstrated compensation by using multiple paths. Such designs allow self-checking routines where speed of sound is compared and successive measurements can be used.

Multipath meters are an option for hydroelectric turbine efficiency measurement. Lowell and Walsh (1991) reported on the use of eight eight-path crossed plane meters and concluded that $\pm 0.5\%$ uncertainty of flow rate could be achieved with appropriate relative positioning of acoustic plane and upstream fittings.

Sanderson and Hemp (1981) also mention the use of a wide beam to average the flow rate.

Hemp (1982) has developed the concept of weight vectors (cf. Appendix 13.A.3) and in a more recent paper (Hemp 1997) showed a weight function distribution for point transducers (which, in practice, would presumably need to be small area) in a circular pipe. This indicated that the meter was most sensitive to flow near the transducers.

See Smith and Morfe 1997 on the effect of beam bending due to velocity variation across the pipe.

13.3 TRANSDUCERS

One of the most important features of the mechanical design of the meter is the transducer and its mounting. This must achieve

- efficient transmission and reception of acoustic signals through the interface,

- ❑ negligible acoustic transmission through the body of the flowmeter,
- ❑ accurate and permanent positioning,
- ❑ no adverse effects due to operating fluid, and
- ❑ trouble-free performance.

Transducers fall into a number of categories, and these will be worth bearing in mind in the following discussion:

Factory installed	Wetted transducers – open cavity
	Nonwetted transducers – open cavity
	Nonwetted transducers – filled cavity

Note: For any of these, the spool piece may be factory wet calibrated. Only the first is suitable for gases.

Retrofit	Wetted transducers – open cavity
	Nonwetted transducers – open cavity

Note: Only in situ calibration is possible. Dry calibration by very careful measurement may be acceptable.

Clamp-on	Nonwetted transducers – transmission through pipe wall

Note: Wet calibration is virtually impossible. This transducer is not suitable for gases.

Figure 13.7 shows some typical mounts for transducers. Figure 13.7(a) shows a wetted transducer in an angled mounting block. The crystal emits a pencil of sound through perpendicular interfaces, and it is not therefore refracted. Figure 13.7(b) shows an alternative arrangement with mounts perpendicular to the tube and transducer with an angled end in which the crystal is mounted. If the liquid is aggressive or particle laden, some form of window may be required. In this case, the transducer may be mounted as shown in Figure 13.7(c), where a metal window is used but where all interfaces remain perpendicular to the beam. The cavity is a source of flow disturbance and solid deposit. Both may cause signal failure or flow measurement errors. Figure 13.7(d) has the cavity smoothed off with a filler of some sort such as an epoxy. Ideally, this filler should have an acoustic refractive index similar to the liquid, thus keeping refraction to a very low value, and the bending of the acoustic ray on crossing the interface will be small. However, changes of fluid or temperature may introduce errors. An important advantage of the mounts shown in Figures 13.7(a, b) is that alternative transmission around the tube wall and outside the fluid is reduced, and in practice the transducer mount can be made to absorb it. With the mounting systems in Figures 13.7(c, d), the signal processing must discriminate between the fluid-borne signal and the spurious signals around the wall of the tube. Sanderson and Hemp (1981) gave some details of ways to increase the bandwidth of the piezoelectric plate.

Figure 13.7(e) shows a clamp-on system in which the transducer mount is entirely external to the tube and may be mounted on existing tube. Here the problems of the type found in the mounts shown in Figures 13.7(c, d) are greatly increased.

For very small pipe sizes, the limitations of pipe diameter and consequent short pulse times can be overcome by a flow tube and transducer mounting system of the

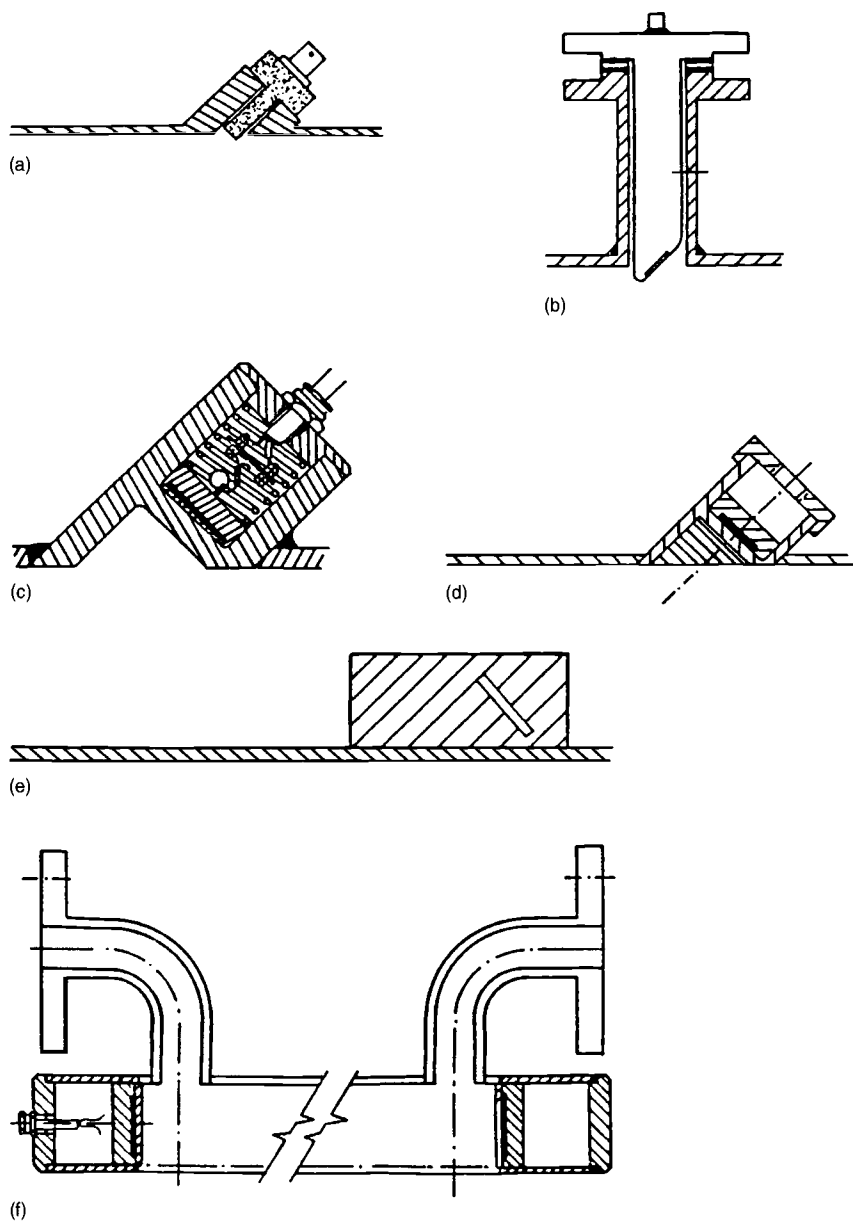


Figure 13.7. Transducer mounts (cf. Baker 1988/9): **(a)** In line – wetted; **(b)** Angled – wetted; **(c)** In line – window protection; **(d)** In line but with cavity filled with epoxy or other filler; **(e)** Clamp-on transducer with crystal embedded in the mount; **(f)** In-line transmission for small tubes with either wetted or protected transducers.

type shown in Figure 13.7(f). The flow enters a section of tube in which the acoustic beam is axial, and hence there is no limitation on the separation of transducers apart from those resulting from the application.

For gases, it is essential that the transducers are in contact with the fluid and so are mounted as shown in Figures 13.7(a, b) or possibly Figure 13.7(f).

To overcome the impedance mismatch, a low impedance transducer was reported (Collings et al. 1993). When used in a meter, it allowed natural gas flows of 0.013–8 m³/h with a temperature range of –13 to 47°C and is claimed to have significant advantages over current domestic meters.

It should also be remembered that for high pressure service the transducers must be flanged to the correct line pressure (rather than screwed). Transducers that project into the tube may avoid attenuation and other problems due to air collection within the pockets. To reduce turbulence caused by the transducer cavities on a 20-mm-ID ultrasonic flowmeter for gas, nets have been placed to create a wall that transmitted the ultrasound (Hakansson and Delsing 1992).

Lynnworth (1988) suggested the use of new types of wetted and clamp-on buffer rods to allow for extreme temperature (e.g., $\pm 200^{\circ}\text{C}$) such as might occur in wetted use in cryogenics or in the flow of quench oil in a carbon steel pipe at 260°C.

13.4 SIZE RANGES AND LIMITATIONS

Single- and twin-path designs may be available for liquids in 80–2,000 mm diameter or greater, although very large sizes may be retrofit applications. The arrangement for small tubes is likely to be for 10–80 mm.

When retrofitting, there is always a tolerance on the pipe dimensions, both diameter and wall thickness, which will affect the deduction of mean flow from ultrasound path velocity, and the measurement needs to take account of the actual separation of transducers (Scott 1984). For instance, in a carbon steel pipe with a 100–1,000 mm diameter, the tolerance may be $\pm 1\%$ on diameter and $\pm 10\%$ on wall thicknesses of 3–5 mm. In addition, erosion, corrosion, and painting of the pipe makes it improbable that the error can be reduced below 0.5%. Manufacturers may offer retrofit for one, two, or four paths, which with precise internal measurement should be capable of high accuracy.

13.5 SIGNAL PROCESSING AND TRANSMISSION

As indicated earlier, the measurement system essentially requires the ability to measure very small time periods, or periods to very high precision. This increment could be of order 10^{-9} (a nanosecond), which Sanderson and Hemp (1981) pointed out was comparable with the (then) propagation delays and rise times of the fastest commercially available logic. Clearly the technology in this area is moving rapidly, and the problems encountered in small time measurement, phase comparison, or sing-around are finding new solutions.

Tables 13.1 and 13.2 have identified the design requirements for the signal processing, namely,

- for transit-time meters to measure timed differences to better than 10^{-9} s (one nanosecond) and
- for sing-around meters to measure frequency differences to better than, say, 0.03 Hz for 1% discrimination in a 300-mm tube with $V = 1$ m/s.

To achieve this, various clock circuits have been used. In an early commercial meter, a ramped voltage system provided a time measure. In addition, there are problems in ensuring that the zero-pass, or whatever part of the wave is used, is consistent between transmitters and receivers. Szebeszyk (1994) discussed some aspects of the design of the ultrasonic flowmeter and transducers, use of first zero crossing of the signal, optimum thickness of matching layers, etc.

Vaterlaus (1995) described a time measurement system with two counters. One counter N_t was clocked during the measuring period by a stable quartz oscillator with a frequency $f = 64$ MHz, whereas the other counted the number of samples N_n . A start pulse initiated the measurement by activating the two counters and triggering the ultrasonic pulse emission. During the measurement period $\tau = 30$ ms, every pulse received caused a new pulse to be emitted. The two counts N_t and N_n were used to calculate the propagation time from N_t/N_n .

It appeared that τ was obtained precisely from N_t/f ; hence, provided τ ended after a complete value of N_n , it was possible to obtain the transit time as $N_t/(N_n f)$ and a resolution of 60 ps in 116 μ s was claimed.

Advantages claimed were:

- Velocity resolution was to 0.8 mm/s.
- Precision depended on the stability of the oscillator.
- Jitter in digital logic was averaged.

Another solution to the timing problem (Pavlovic et al. 1997) appeared to use

- a. a transmitted pulse in two halves, the first at one amplitude and the second at twice the amplitude, presumably allowing more precise timing edges;
- b. the downstream and upstream times were then given by

$$t_d = N_d T + \Delta\tau_d$$

$$t_u = N_u T + \Delta\tau_u$$

where N_d and N_u were the whole number of counter periods of a 2.2-MHz square wave train obtained from a 17.7-MHz oscillator through a divider by 8 and were obtained by standard counter methods;

- c. $\Delta\tau_d$ and $\Delta\tau_u$ were measured through the phase difference between transmission and received signals, and a two-stage measurement system using a coarse and fine approach. An integrator was used to obtain $\Delta\tau_d$ and $\Delta\tau_u$.

The effects of error due to temperature on pipe diameter, recesses, delays in electronics, and transducers were also addressed.

Because such short times are being measured, system designs need to address delays in the cables, transducers, and converter (Scott 1984). Time delays of microseconds could lead to errors if the head is calibrated on water for use on oil ($\pm 1.5\%$), or if the temperature of the water changes by 10°C ($\pm 0.1\%$). Further delays are caused in the window, or in the transmission through the wall, ranging from about 3 μ s to about 15 μ s. These delays will also be dependent on the choice of materials.

Hemp (1988, 1998) has proposed a reciprocity approach to the elimination of zero instability. This requires that the transducers are driven, say, by a voltage pulse,

and that the received signal is then sensed by its current pulse. The path is then reversed.

13.6 ACCURACY

Van Dellen (1991) suggested that the ultrasonic meter could meet custody transfer accuracies, and this appears to be happening (cf. Selvikvag 1997).

Reports indicate that good accuracy can be achieved by dry calibration – the use of flowmeter dimensions to obtain calibration (de Boer and Lansing 1997). Zanker and Freund (1996) even suggest that the difficulties of calibrating a large (30-in. or 750-mm)-diameter meter may make dry calibration preferable. This will, in part, reflect the limitations in the flow calibration facilities at large volume flow rates.

13.6.1 REPORTED ACCURACY – LIQUIDS

Cascetta (1994) compared a clamp-on transit-time ultrasonic meter with an electromagnetic meter in situ in a water distribution network with 400 mm ID. The ultrasonic meter generally overestimated the flow rate, and the difference between the ultrasonic and electromagnetic flowmeter was mainly in the range ± 1 –5%.

Delsing (1991) tested the zero-flow performance of a sing-around ultrasonic flowmeter with multiple paths. The meter was in a 25-mm line, with transducers set at 20° to the flow direction, and of 15 mm diameter. The sound path was 96 mm, giving a period of about $64 \mu\text{s}$ for a sound speed of 1,500 m/s. The single-period resolution was about 15 ns, giving a velocity resolution of about 0.18 m/s. This could be increased by using multiple-period averaging techniques. For instance, using 1,000 sing-around loops, a theoretical period measurement resolution of about 15 ps and velocity resolution of about 0.18 mm/s could allow measurement down to 1.8 cm/s with an uncertainty of $\pm 0.5\%$. The experimental data showed an overall zero-flow error of less than ± 0.6 mm/s, which would allow uncertainty of $\pm 1\%$ for flows as low as 6 cm/s, and Delsing suggested that this would open up the meter's range to 100:1.

Manufacturers of commercial meters tested at NEL (Brown 1996) seemed to claim for uncertainty, in the main, about $\pm 0.5\%$ of rate, but there appeared to be quite a lot of variation in actual performance between designs.

13.6.2 REPORTED ACCURACY – GASES

Holden and Peters (1991) found that, for undisturbed flow, the British Gas 300-mm meter readings lay within 0.4 and -0.8% of the reference turbine meters. For a 150-mm meter, the variation was between 1.4 and -0.35% , and a 500-mm meter appeared to be within about 0.3%.

A 6-in. (150-mm) ultrasonic meter calibration against sonic nozzles (with repeatability of 0.04%, Erdal and Cabrol 1991) gave a day-to-day repeatability of 0.1%.

Grimley (1996) tested Daniel and Instronet meters and concluded that they were capable of accuracies within 1% tolerance and with repeatability better than 0.25%. Pressure change (e.g., 4.5 MPa) caused shifts of 0.4%. It also suggests that, for dry calibration, there will be a need to understand the effect of parameter changes.

13.6.3 MANUFACTURERS' ACCURACY CLAIMS

Typical performance claims for single-path meters for liquids with wetted transducers are $\pm 1\%$ of full scale. Repeatability of $\pm 0.2\%$ full scale may be available. Ability to cope with entrained solids or gas bubbles is increasing. Wetted transducers are available for temperature ranges from -200 to $+260^\circ\text{C}$ and pressures up to 200 bar. Pipe sizes range from less than 50 mm to 2 m. Fluid velocity in the pipe can range from 0.03 to 10 m/s or higher.

For gases, the accuracy is likely to be of order ± 0.5 – 5% of reading, and repeatability will range from 0.25 to 0.5% of reading, with a range of velocity of 0.3–45 m/s. Duct sizes can range up to 10 m. Temperature range may be as much as -50 to $+260^\circ\text{C}$.

13.6.4 SPECIAL CONSIDERATIONS FOR CLAMP-ON TRANSDUCERS

For clamp-on meters, manufacturers appear to be unwilling to claim better than about 3% of rate at higher flow rates, or even 3% of full-scale deflection for low flow rates. However, as in at least one case, the manufacturer may give figures such as

- repeatability if nothing changed, $\pm 1\%$;
- uncertainty ($D < 25$ mm), $\pm 10\%$;
- uncertainty ($25 < D < 50$ mm), $\pm 5\%$; and
- uncertainty ($D > 50$ mm), $\pm 3\%$.

Sanderson and Torley (1985) described an intelligent clamp-on flowmeter and claimed an uncertainty within $\pm 2\%$ of reading. They concluded that the wedge angle of the transducers (estimated uncertainty of $\pm 1^\circ$) could account for much of the error. They gave plots of baseline drift equivalent to about 30 mm/s. They demonstrated the value of Hemp's (1979) reciprocal drive system on baseline stability (e.g., where there is temperature change), which reduced drift to about one-eighth of that for a conventional system. They found that an error of $\pm 2.5\%$ in transducer separation resulted in errors within 0.5%. The meter used additional transducers to locate optimal axial separation of transducers and for self-calibration. The transducers allowed pipe wall thickness to be measured and also internal dimensions of the pipe. They also gave a very useful analysis of the likely errors in the system.

Cairney's (1991) experience of dedicated transducers, where there is a specific pair of clamp-on transducers for each different size of pipe, with a scale card within the portable electronics unit, was that, within the limitations, accuracy and repeatability were good for pipes with OD greater than 35 mm. An example is the measurement of lubricating-oil flows at near ambient temperature in a 60.4-mm OD line with 5.5-mm carbon steel wall. Reynolds number was 50–350 (laminar). Readings were within $\pm 3\%$ of a volumetric tank, where $\pm 3\%$ was typical above 35 mm OD, but below this errors tended to increase. It seems that Cairney also implied that removing and replacing the transducers led to larger errors, typically in the range 5–10%.

Cairney's (1991) experience of multipurpose transducers, which allowed the use of one pair of transducers for a wide range of pipe sizes, was that the unit would operate with almost any homogeneous, Newtonian liquid and on any pipe that

Table 13.4. Pipe material and sizes used by Cairney (1991)

Materials	Outside Diameter (mm)	Wall Thicknesses (mm)
Carbon steel	60.4–61.0	5.55–8.6
	89.1	6.5
	33.3–33.7	3.89–6.21
Stainless steel	33.4 & 33.5	1.74 & 2.86
Aluminum	63.7	6.6
Unplasticized polyvinyl chloride (UPVC)	33.5 & 33.7	5.10 & 2.39
	114.3	5.4

conducts sound. Relevant details are fed into the electronics unit: pipe geometry, material, and fluid properties. The manufacturer claimed $\pm 3\%$ uncertainty without calibration. Cairney tested the meters to check this claim. The tests were all in horizontal pipes with the transducers mounted on the horizontal centerline. Pipe details for his tests are given in Table 13.4.

For most of the tests flowmeter uncertainty was within $\pm 5\%$. Particular notes follow:

- ❑ Flow rate and pipe material had no noticeable effect, and coupling to the pipe was not a problem.
- ❑ Performance was similar in direct and reflex modes.
- ❑ Rust on the inner surface is more likely to affect the cross-sectional area of the pipe than the signal transmission.
- ❑ The ratio of pipe diameter D and wall thickness t should be greater than 15; $D/t > 10$ is acceptable, and $D/t < 10$ may cause problems due to ultrasound transmission around the pipe wall. The low values may increase the uncertainty to order $\pm 10\%$.
- ❑ Removal and replacement of transducers caused, on some occasions, a severe shift in reading traced to an incorrect measure of sound speed. This appeared to be a fault with the electronics. Cairney recommended checking the computed speed of sound. Provided care is taken, the removal and replacement of transducers should contribute only about $\pm 2\%$ to the error.
- ❑ Air bubbles can cause loss of signal strength, which should, therefore, be monitored.
- ❑ Nominal pipe dimensions should not be used. A 1-mm error in a pipe of 50-mm diameter results in about 8% error in cross-sectional area.
- ❑ Incorrect axial spacing of the mounting tracks is not significant, but incorrect angular spacing can cause errors up to $\pm 10\%$.

Cairney concluded that errors should be containable within about $\pm 5\%$.

Commercial devices allow a range of materials for the pipe including steel, stainless steel, cast iron, vinyl chloride, fiber reinforced plastic (FRP), and asbestos, in sizes from 6 mm or less to 9,000 mm. The presence of a lining of tar, epoxy, mortar, or Teflon appears to be surmountable. They may be available in both reflex (one or more internal reflections off the pipe wall) or direct mode.

Sensitivity as small as 0.3 mm/s and zero drift of 3–6 mm/s has been suggested with uncertainty claims of 1.5% reading down to 1 m/s. The meter should be capable of measuring flows in continuous liquids with low attenuation of the beam and no air bubbles, etc.

NEL (1997) reckoned that, with care, clamp-on meters could achieve spool piece precision, but diametral paths were a limitation. An error in the measurement of diameter of, say, 1% would result in a cross-sectional error of 2%. However, the point was also made that temperature change could lead to changes in the refraction angle.

13.7 INSTALLATION EFFECTS

In addition to the following effects, it should be remembered that with these precision instruments, in which the accuracy of mounting transducers and general dimensional stability is important, temperature variation within the measuring region of the flow tube, stresses in pipe fixtures, unsuitable mounting, and even Joe Bloggs who regularly taps the pipe with his spanner as he walks by or climbs on it as a step, may have detrimental effects on precision.

Zanker and Freund (1994) found that less than 1% condensate had little effect (+1%) and that the meter survived in site trials, despite adverse conditions. NEL (1997) reckoned that the maximum air-in-water for satisfactory operation was 0.5% by volume, but that oil-in-water was a lesser problem.

Heritage (1989) reported an important series of tests on the performance of transit-time ultrasonic flowmeters using water as the test fluid. The first part of her work indicated the surprisingly large failure rate of new flowmeters. Out of 11 different flowmeters from seven manufacturers, 3 failed to complete the test program, only 2 performed within the manufacturer's specification, and differences of as much as 1.7% of rate were observed between the analogue and pulse outputs for certain meters. Based on analogue readings, 2 meters were within 2%, 3 were within the range 2–5%, 2 were within the range 5–10%, and 2 were over 10%.

13.7.1 EFFECTS OF DISTORTED PROFILE BY UPSTREAM FITTINGS

Some early reported installation effects (Al-Khazraji et al. 1978) obtained by combining measured flow data for an eccentric orifice discharge at 5.5D with theoretical calculations suggested that a single-path meter could have errors of 5–16%; a twin path meter, $\pm 1.6\%$; and a four-path meter could have errors of less than 1%.

Tables 13.5–13.7 summarize the installation tests of Heritage (1989) on water. In the original paper, the results of two single-path meters are reported separately, but they are combined in Table 13.5. The effect of the gate valve is taken regardless of orientation. This is partly because the differences did not appear to be substantial, and the safe assumption is the range given. The same is true for the two bends, which were in perpendicular planes. However, in this case, the variation between the meters is so great that it appears to outweigh any effect of orientation.

It appears from Tables 13.5 and 13.6 that a single-path meter will probably require at least the spacing for 0.75 beta ratio orifice. For dual-path flowmeters, the performance (Table 13.7) appears to be close to that for an orifice with $\beta = 0.6$ with $\pm 0.5\%$ additional uncertainty.

Table 13.5. Installation tests on water for single-path meter (after Heritage 1989)

Disturbance	Spacing Between Downstream Flange of Disturbance and Upstream Flange of Meter (Error Range in Percentages)		
	5D	10D	15D
Gate valve 50% closed by movement	Not obtained	-1 to -2.5	+1.5 to -3
Swept bend ($r/d = 1.5$) parallel	Not obtained	+4.5 to -6.5	-2 to -3.5
Swept bend ($r/d = 1.5$) perpendicular	Not obtained	-3.5 to -4.5	+3.5 to -3
Two swept bends	Not obtained	+7.5 to -11	-0.5 to -7
Reducer	Not obtained	+2 to -4.5	+2 to -1

Halttunen (1990) used data on flow profiles and applied them to the ultrasonic flowmeter analyses to obtain the effect of flow distortion on the performance. He also gave experimental data from installation tests presumably on water. Each of his methods appeared to give a slightly different datum, a salutary reminder that all such measurements and predictions are subject to some uncertainty. The conclusions that can be drawn from the plots are given in Table 13.8.

Table 13.6. Installation tests on water for single-path clamp-on meter (after Heritage 1989)

Disturbance	Spacing Between Downstream Flange of Disturbance and Upstream Flange of Meter (Error Range in Percentages)		
	5D	10D	15D
Gate valve 50% closed by movement	Not obtained	+3 to -2.5	+1.5 to -0.5
Swept bend ($r/d = 1.5$) parallel	Not obtained	+3.5 to -3	-2.5 to -3.5
Swept bend ($r/d = 1.5$) perpendicular	Not obtained	-2 to -2.5	+2.5 to -3.5
Two swept bends	Not obtained	-5 to -10	+0.5 to -6
Reducer	Not obtained	-2.5 to -4.5	0 to -0.5

Table 13.7. Installation tests on water for dual-path meter (after Heritage 1989)

Disturbance	Spacing Between Downstream Flange of Disturbance and Upstream Flange of Meter (Error Range in Percentages)		
	5D	10D	15D
Gate valve 50% closed by movement	0 to +1.5	+0.75	Not obtained
Swept bend ($r/d = 1.5$) parallel	0 to +1	+0.25	Not obtained
Swept bend ($r/d = 1.5$) perpendicular	+0.75	-0.75	Not obtained
Two swept bends	-1.5 to -3	+0.5 to -1.5	Not obtained
Reducer	+1 to +0.5	+1 to +0.5	Not obtained

The orientation for single-path ultrasonic flowmeters may cause an additional $\pm 1\%$ at 10D or less spacing, and for dual-path ultrasonic flowmeters it may cause an additional $\pm 0.5\%$ at 5D or less spacing.

The following claims of Vaterlaus (1995) appear, possibly, optimistic but probably reflect the (surprising) variation found by Heritage (1989). For a single-path meter, errors should be within 1% if at least:

- 10D is allowed between a reducer and the meter;
- 20D is allowed for an elbow or a T;
- 25D is allowed for two elbows in one plane;
- 40D is allowed for two elbows in perpendicular planes; and
- 50D is allowed for a partially open valve or for a pump.

Table 13.8. Halttunen's (1990) installation data

	5D	10D	20D	40D	80D
Ultrasonic flowmeter – single path					
Single elbow	-5%	-2.5%	-2%	—	—
Double elbow 90°	-7%	-5%	-3%	-1%	-1%
Ultrasonic flowmeter – dual path					
Single elbow	-1%	-0.5%	small	—	—
Double elbow 90°	$\pm 1.5\%$	$\pm 1\%$	$\pm 1\%$	+0.5%	+0.5%

Hakansson and Delsing (1992) tested the effect of upstream disturbance on a 20-mm-ID ultrasonic flowmeter for gas. Against a reference flow from a 100D upstream straight pipe, the results indicated that the laminar/turbulent flow change at Re in the range 2,500–4,000 caused a calibration shift of about 11% and that for the Re range 4,000–11,000 calibration dropped by about 1%. They reckoned that, within the turbulent region,

- ❑ for a single bend in the same plane as the ultrasound beam, about 40D upstream straight pipe (from the transducer path) is required to contain the errors to within $\pm 1\%$ of the reference. Note that longer lengths may not significantly reduce this value at certain Re values.
- ❑ for two bends in perpendicular planes upstream, 80D upstream straight pipe does not contain the errors to within $\pm 1\%$ of the reference.

Multipath (apparently two paths, and not necessarily off-axis) flowmeters were tested on water to obtain profile effects due to a bend. A 10D upstream length was considered to cause a calibration shift of 1% (Johannessen 1993).

Holden and Peters (1990, 1991) concluded from tests on a four-path ultrasonic flowmeter operating on high pressure gas, in both fully developed flows and also with upstream disturbances, that 10D upstream and 3D downstream is sufficient for the data to fall within $\pm 1\%$. The meter can also deduce swirl and turbulence in a disturbed flow and may operate satisfactorily with less than 10D upstream at the lower end of the flow range.

The effect of bends, step changes in diameter, and pressure reduction were investigated by a joint industry project on multipath meters. van Bloemendaal and van der Kam (1994) concluded that, in well-developed conditions, uncertainty of 0.6% should be achievable without calibration but with careful determination of meter dimensions and zero setting. They reckoned that an additional $\pm 0.5\%$ should be allowed for 10D minimum spacing from a bend and that small changes in pipe diameter had negligible effect but that swirl and noise from pressure reductions could have severe effects resulting in 2–2.5% additional uncertainty.

Lygre et al. (1992) described a five-path gas ultrasonic flowmeter. The paths were positioned so that three are on one diagonal plane, whereas the other two lie between the three but on a reverse diagonal plane. It was designed to operate with 10D upstream and 3D downstream. They claim that, for a bend upstream, the installation length may be reduced to 5D. Their results suggested an uncertainty within $\pm 0.5\%$ over a 4:1 turndown.

From these sources of data, the following overall conclusions are suggested.

- a. A single-path meter will have a calibration shift of up to 33% for changes from laminar to turbulent and 3% for changes in the turbulent range.
- b. For a single-path meter to remain within 2% of calibration, the following spacings should be allowed:

- 15D for a reducer
- 20D for a bend or a T
- 40D for two elbows in perpendicular planes.

For distances less than these, the following values are conservative:

Fitting	Error	Spacing
Reducer	$\pm 5\%$	10D
Bend	$\pm 4\%$	15D
Two bends	$\pm 7\%$	15D

- c. A two-path meter has a calibration shift of 0.7% (NEL 1997) or less for changing Reynolds number and should have the following spacings to remain within 1% of calibration:

10D for a reducer
 10D for gate valve (50% closed) or bend
 20D for two swept bends

- d. A four-path meter should retain its performance to within $\pm 1\%$ in liquid or gas with 10D upstream and 3D downstream.

The values appear to be reasonably confirmed as being on the cautious side by Grimley (1997). However, Grimley's results have some puzzling changes in baseline, which may result from the meter calculation method, and appear to make performance better than 0.5% uncertain. His work suggested, also, that thermowells at 5D or less may cause errors of order 0.6%.

NEL's (1997) rule of thumb that single-, two-, and multipath meters need 20D, 10D, and 5D upstream between the meter and a fitting perhaps needs a bit more detail. The summary of values, which take the more pessimistic end of the published data, suggests that to achieve less than 1% on a single-path meter would require similar installation lengths or more than those for an orifice plate for the maximum beta ratio and "zero additional uncertainty." The two-path meter, to remain within 1%, would require those for an orifice with $\beta = 0.6$ with $\pm 0.5\%$ additional uncertainty, whereas the four-path meter is probably much better than these figures suggest. The assumption made is that there should be similar effects for gas and liquid.

13.7.2 UNSTEADY AND PULSATING FLOWS

Mottram's (1992) key comment "if you can't measure it, damp it!" is balanced by his advice that ultrasonic transit-time meters are probably not affected. However, there are some dangers of which the user should be aware. Hakansson and Delsing (1994) found that there were at least three error-creating effects due to pulsating flows in ultrasonic flowmeters:

- Aliasing where the sampling frequency picks samples that do not give a true average – This error should be avoidable provided it is recognized.
- Flattening of the flow profile due to the pulsating flows – This error is fundamental to pulsating flows and the incorrect averaging of the diametral path ultrasonic flowmeter and, therefore, is difficult to avoid.
- Extreme flows during the fluctuation exceeding the upper and lower cutoffs of the flowmeter and giving a spurious mean – This error should be avoidable with care.

13.7.3 MULTIPHASE FLOWS

I have observed, in some field data from an ultrasonic flowmeter, a behavior that could result from entrapment of air. A meter in a low head flow, where air was entrained with the water, periodically failed. The possibility that in such a flow the transducer cavities could cause small local vortices that would entrap the air and block the ultrasonic beam offered a possible explanation.

Lenn and Oddie (1990) have looked at how the scattering of ultrasound from secondary components in the flow, both liquid and solid, could be detected, and they considered that it was possible to measure component size concentrations and velocity using a variety of signal-processing techniques, particularly at low concentrations.

A commercial transit-time device has been claimed to be capable of operating in flows of drilling mud, although my experience is that it is usually recommended that commercial devices should not be applied to other than single-phase flows.

Brown (1997) appears to have suggested that ultrasonic meters may be able to cope with oil-gas and oil-water flows, but Johannessen (1993) put a limit of 10% oil-in-water and reckoned that they were unlikely to cope with more than 0.5% by volume of air in liquid, in agreement with NEL's (1997) summary.

13.8 GENERAL PUBLISHED EXPERIENCE IN TRANSIT-TIME METERS

13.8.1 EXPERIENCE WITH LIQUID METERS

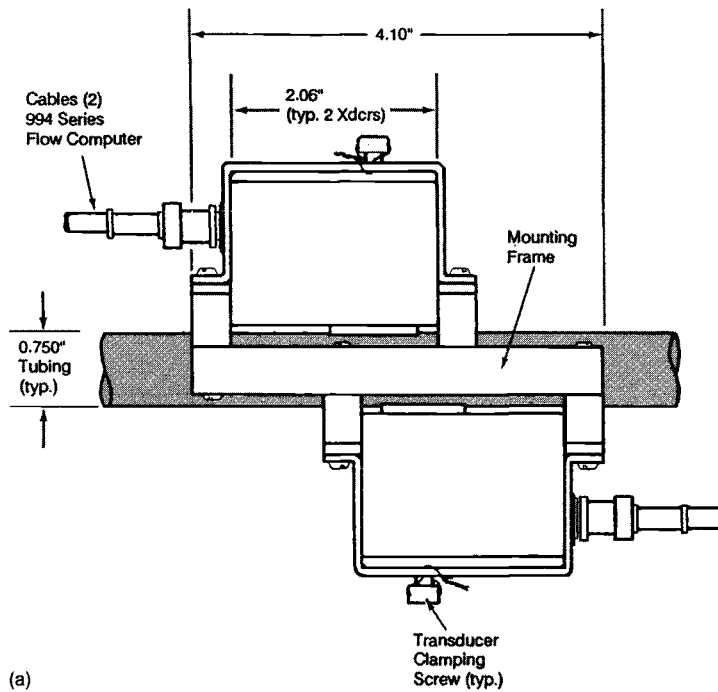
Boer and Volmer (1997) described a five-path liquid meter claimed to be for use in maintenance-free custody transfer applications. It had two paths to measure swirl and inlet and outlet cones to condition the profile.

Reynolds (1994) discussed water industry specifications, and data sheets have been produced for the industry on ultrasonic flowmeters.

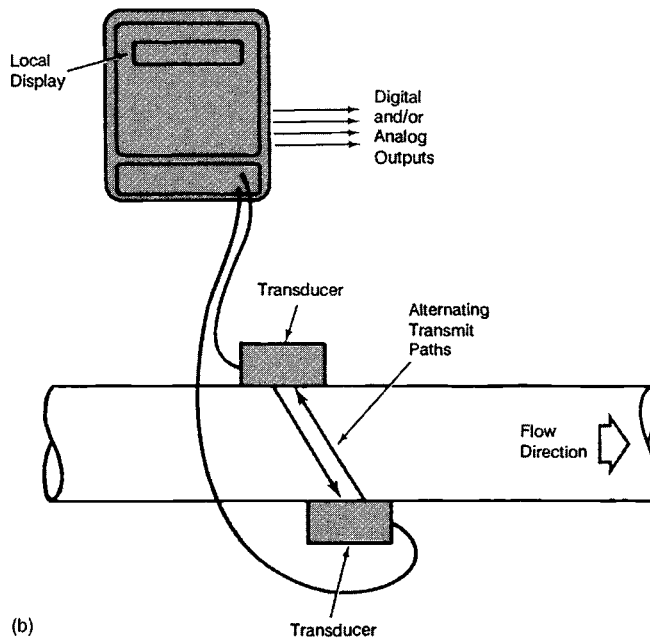
Transit-time meters have been used in the electricity supply industry (Cairney 1991). Transducers were grouted into the wall of an octagonal section culvert. It was also possible to use radioactive-isotope dilution methods for in situ calibration with an uncertainty of $\pm 1\%$ or better with a 95% confidence level. The ultrasonic flowmeter was reckoned to have an overall error of $\pm 1.5\%$. A difference of 5% from the manufacturer's calibration was put down to error in the set-up procedure.

In the Ataturk Hydro Power Plant in Turkey, two systems were used to compare flows for loss due to bursting (Vaterlaus 1995). The pipes were up to and over 6 m in diameter. Taylor and Cassidy (1994) discussed the use of ultrasonic meters at BC Hydro in Canada.

Baumol (1994) described the main features of an ultrasonic meter including velocity up to 30 m/s (± 100 ft/s), an active zero flow, sensitivity of 0.3 mm/s (0.001 ft/s), and uncertainty of up to $\pm 0.5\%$ of flow rate. Figure 13.8 shows diagrams of an ultrasonic clamp-on meter, and Figure 13.8(c) shows a reflect mode installation. Signal strength fluctuation indicates changes in the quality of the flow, like aeration. Applications given are fuel mass metering, hydraulic fluid flow metering, leak detection, engine lubricant, ground support of all aircraft fluid systems, rocket fuel and

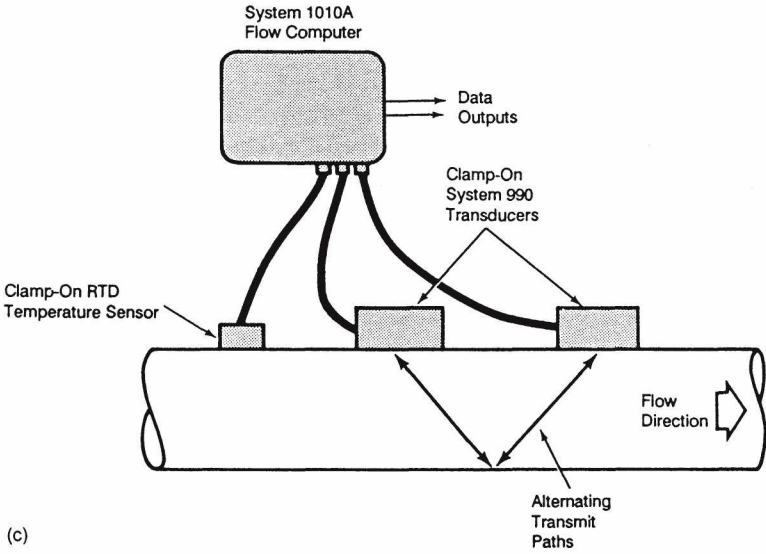


(a)

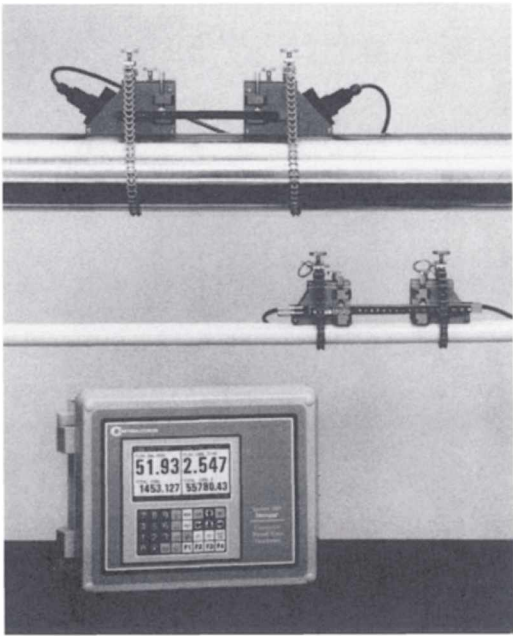


(b)

Figure 13.8. Clamp-on ultrasonic flowmeter (Baumoel 1994; reproduced with permission of ISA and Controlotron Corporation): **(a)** Diagram of the clamp-on meter; **(b)** Direct mode installation; **(c)** Reflect mode installation; **(d)** Photo of clamp-on flowmeters.



(c)



(d)

Figure 13.8. (continued).

oxidizer metering, and space vehicle coolant flow metering. The sound speed for hydrocarbon fluids was shown to drop over a range from about 0–40°C (30–100°F) by about 12% and differs for different hydrocarbons. With the relationship between temperature and density, the meter can deduce the type of fuel in the line, whether there is water in the line, and the mass flow of the fluid. Some units have

been in use for over 20 years, and Baumöel claimed that no repair or recalibration was required.

The results of tests to compare a portable transit-time flowmeter (manufactured by Tokyo Keiki in September 1988), which had two sets of clamp-on transducers, with a two-path profiling system were described by Lynch and Horciza (1995). The transducers were fixed magnetically to the carbon steel pipe. The outside diameter was measured, and the wall thickness was obtained with a sonic gauge. The performance appears to have been remarkably good within 0.6% of a calibrated venturi. Four portable meters (three from Tokyo Keiki 1 MHz and one single path from Panametric 0.5 MHz) were tested in V and Z modes and appear to have agreed with an eight-path chordal meter to within 1%, suggesting a convenient and cost-effective means of testing small hydro plant and turbine performance. Lynch and Horciza appear to have had some problems with poor earthing arrangements on one of the installations.

Also see Falvey (1983) who, working under very different conditions in a river delta, experienced refraction due to salinity and temperature gradients.

13.8.2 GAS METER DEVELOPMENTS

British Gas's development of a multipath ultrasonic flowmeter was discussed in several papers (cf. Nolan et al. 1985), and site tests were reported. Four-path meters were tested ranging from 150-mm (6-in.) to 1,050-mm (42-in.) in diameter. In particular, tests of a 300-mm (12-in.) meter were reported. The signal was used to derive, without any empirical constants, a flow rate that was initially checked on a traceable calibration stand and found to be well within $\pm 1\%$ of rate over nearly 12:1 turn-down. On site, it appeared to give a performance of order $\pm 0.5\%$ or better. The meter has now been developed and sold by Daniel Industries worldwide.

An accuracy of 0.5% was attainable for a custody transfer application with a multipath ultrasonic meter (Beeson 1995). NorAm installed the first custody transfer meter in North America in 1994 at a cost of about four times that of a single-path meter. One problem experienced was that a small oscillating flow caused a small flow indication, dealt with by increasing the low flow cutoff. The installation is the most critical step under line pressure. Having removed outer paint and used ultrasonic gauging for the pipe wall thickness, the outside diameter was measured, and the internal diameter calculated. Two collars were welded in the correct positions, valves were installed on the collars, and the line was hot-tapped. Probes were then inserted in addition to pressure and temperature sensors. Beeson also recommended on-board isolation to avoid ground loop problems and commented on failure of new units. Faults were in probes and electronic circuits.

Dry calibration is possible, but the speed of sound needs to be verified by measurement of the gas constituents from a gas chromatograph, and the gas in each case must be thoroughly mixed. A standard, which consisted of two multipath meters in series with a bank of sonic nozzles calibrated against a primary standard weigh tank, was reported. It appears that the meters ranged from 0.4 to 0.75 m (16 to 30 in.).

The development, testing, and production of the new Siemens ultrasonic gas meter, which has had DTI approval, was briefly described by Sheppard (1994) (cf. Chapman and Etheridge 1993). The aspects of communication, automatic meter

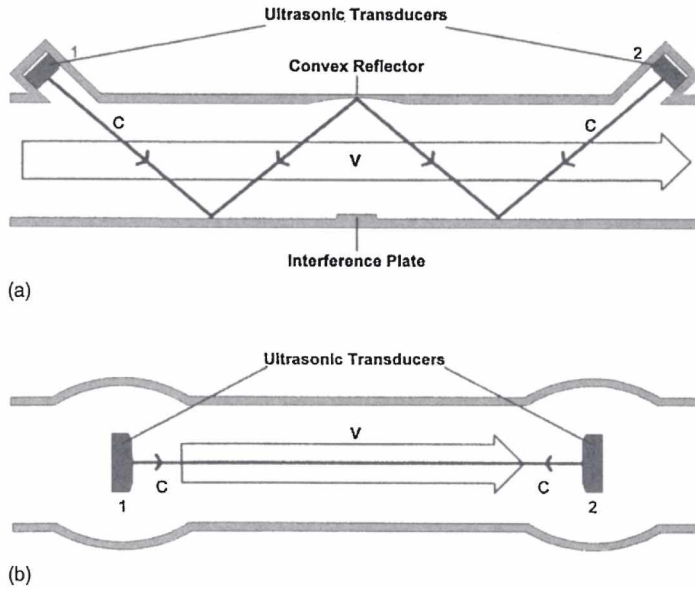


Figure 13.9. Ultrasonic gas meter (Sheppard 1994; reproduced with permission of Elsevier Science Ltd.): **(a)** W beam type; **(b)** Dog-bone type.

reading, and temperature compensation were also discussed. The meter uses a W path configuration [Figure 13.9(a)] where the beam is reflected within the tube three times. It is claimed that this gives good integration across the flow. The timing resolution at the low flow rates defined by British Gas (40 l/h) is of order a few nanoseconds. Low power requires a two-stage timing design and resulted in a 10-year life on one D cell battery. The use of an application-specific integrated circuit (ASIC) allowed a fully electronic meter at a cost comparable with the diaphragm meter. By observing Hemp's reciprocity requirements, drift is compensated, and stability is achieved. The flow is sampled randomly but with a basic 0.5-Hz frequency. The specification is $\pm 1.5\%$ from 6,000 l/h down to 80 l/h, and $\pm 3\%$ from 80 l/h down to 40 l/h. It is notable that the production requirements were considered from the outset of the design of the meter. Kochner et al. (1996) described the tube as having a rectangular cross-section, 30 mm $\times$ 6.3 mm, with the transducers recessed. The small width and W path increase velocity and improve averaging. The reflector supplies some focusing to the beam, and an interference plate suppresses single reflection (V) path signals. The meter housing acts as a plenum chamber between mains and the meter inlet, which has a strongly converging horn-type inlet. A calibration factor adjusts the measured velocity to the mean velocity in the duct. A transition from laminar to turbulent flow occurs in the duct at about 1,000 l/h, which gives a Reynolds number, based on hydraulic diameter of the duct, of about 1,000. The meter needs to cope with contaminants and is calibrated on an automated flow rig.

Bignell et al. (1993a, cf. 1993b) also described an ultrasonic domestic gas flowmeter. They aimed to discriminate to 0.005 m³/s with a time difference of 5 ns to achieve this. They used an inverted signal periodically to deal with the long "tail" of

transmission along the duct due to slower moving modes, as well as mode control bodies in the flow. Application-specific integrated circuits were used, and an optical port allowed communication via an infrared carrier using a portable box or remote reading. The operation was for -10 to 50°C within about $\pm 1.5\%$.

Bignell (1994) also discussed a secondary standard gas meter. Again, he discussed the effect of small tubes on the various modes of the ultrasonic beam. He described a meter with axial transmission through a dog-bone shape [cf. Sheppard 1994 and Figure 13.9(b)] with the transducers in the enlarged “ends” around which the inlet and outlet flow passed to and from the “bone” between the ends. The tube length was 0.56 m with a diameter of 11.95 mm. The transducers were polyvinylidene fluoride (PVDF) with 125 kHz, which was considered low enough to avoid absorption and high enough to be away from ambient noise and to give a rapid zero crossing with good timing precision. Modes of transmission and attenuation of downstream plane wave compared with the upstream wave are noted, and one suspects that this may be due to the dispersion of sound due to profile of the flow. Bignell claimed that he had achieved a $300:1$ turndown and saw the possibility of achieving $1,000:1$ in future.

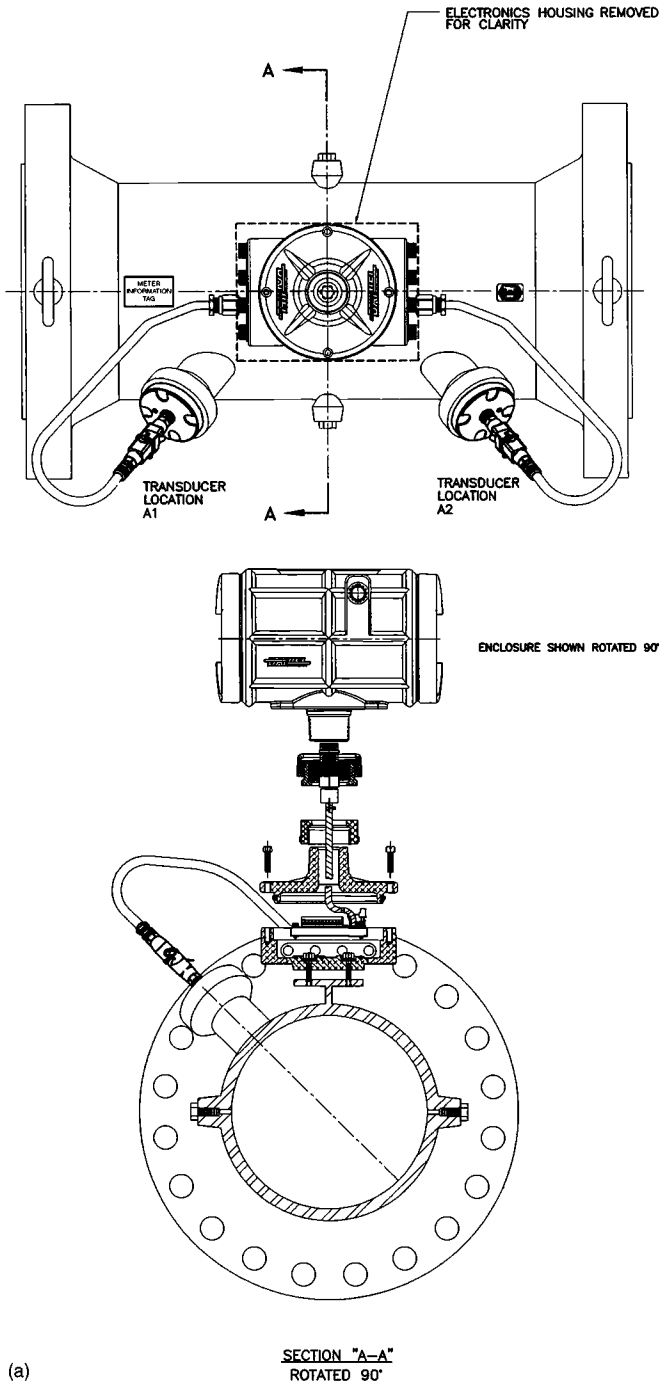
A meter with a single path is shown in Figure 13.10(a), dual paths are shown in Figure 13.10(b), and a multipath gas meter is shown in Figure 13.10(c). It is claimed that for a flow range of 0.3 m/s (1 ft/s) to about 20 m/s (70 ft/s), the four-path design is capable of an uncertainty with calibration of $\pm 0.5\%$ rate plus an additional factor depending on size of meter, etc., of 0.2 – 0.1% full scale for sizes 100 – 600 mm and a temperature range of -20 to 40°C . The whole operates on a clock resolution of 10 ns with nearly $1,000$ measurements per minute and for a range of temperature of -20 to 40°C . Van der Kam and Dam (1993) described the replacement of orifice meters in export stations of Nederlandse Gasunie and suggested that for wet or dirty flows the four-path ultrasonic would possibly be best of all.

A novel five-path meter has been reported (McCarthy 1996), which was developed in cooperation between Gasunie and Instramet. Trials on an export line resulted in a performance within 0.2% of installed turbine meters.

To overcome the beam bending, Mylvaganam (1989) used a ray-rescue offset angle, which appeared to be taken as the half-width of the transmission lobes of the transducer sound energy pattern. He pointed out that this will also prevent standing waves in small tubes. He also briefly described the transmission of ultrasound, which is achieved in a chirp pulse where the frequency is ramped up in the time of the pulse, which resulted in pulse compression at the receiver through correlation (Cook and Bernfeld 1967). At low flows a continuous wave was used. Mylvaganam appeared to claim 3 – 5% uncertainty with a 95% confidence level. Other solutions have been discussed in this chapter.

For very high flow rates (Mach number of greater than 0.2), Guilbert and Sanderson (1996b) have calculated the shape of wall for a reflex meter to focus the beam onto the receiving transducer. The result is a slight hump about 1 cm maximum in a duct of 10 cm with transducer spacing about 10 cm.

Noise in the ultrasonic range is a serious problem that can be caused by valves, compressors, etc., and noise suppression algorithms have been tested to overcome the problem (Kristensen et al. 1997).



(a)

Figure 13.10. Ultrasonic gas meters (reproduced with permission from Daniel Industries, Inc.): **(a)** Drawing of single path; **(b)** Drawing of dual path; **(c)** Drawing of multipath.

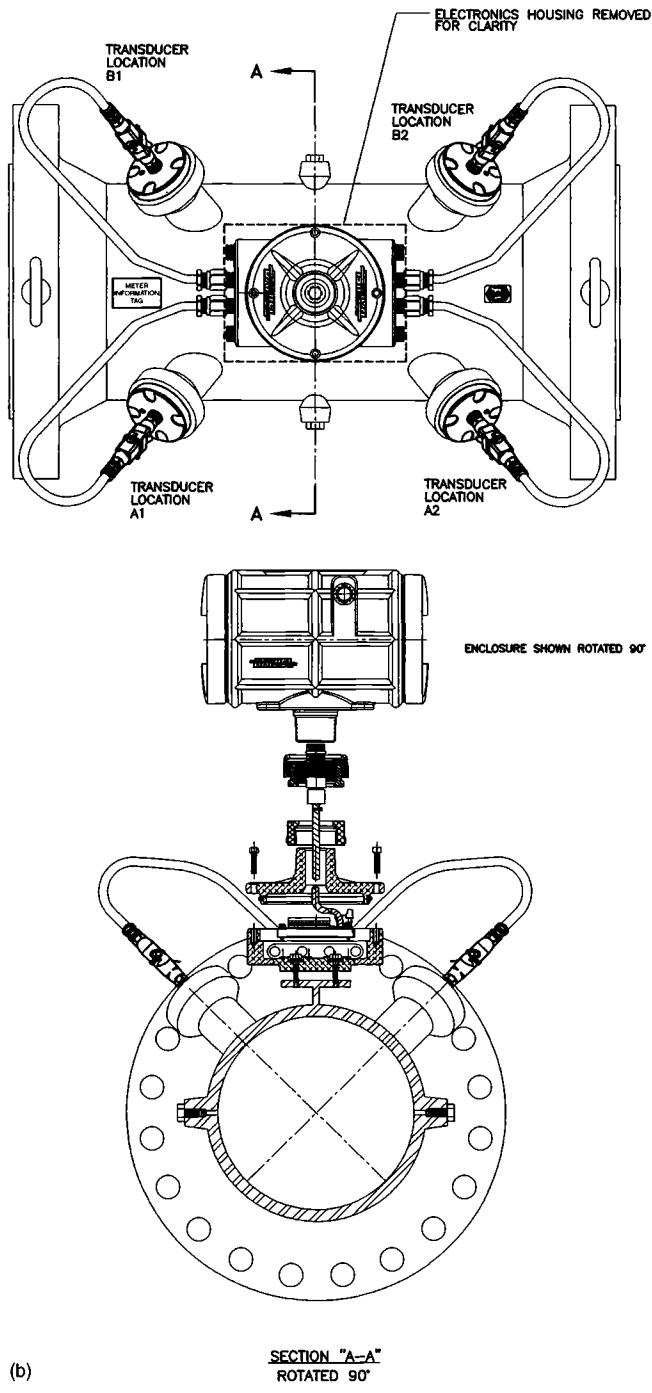
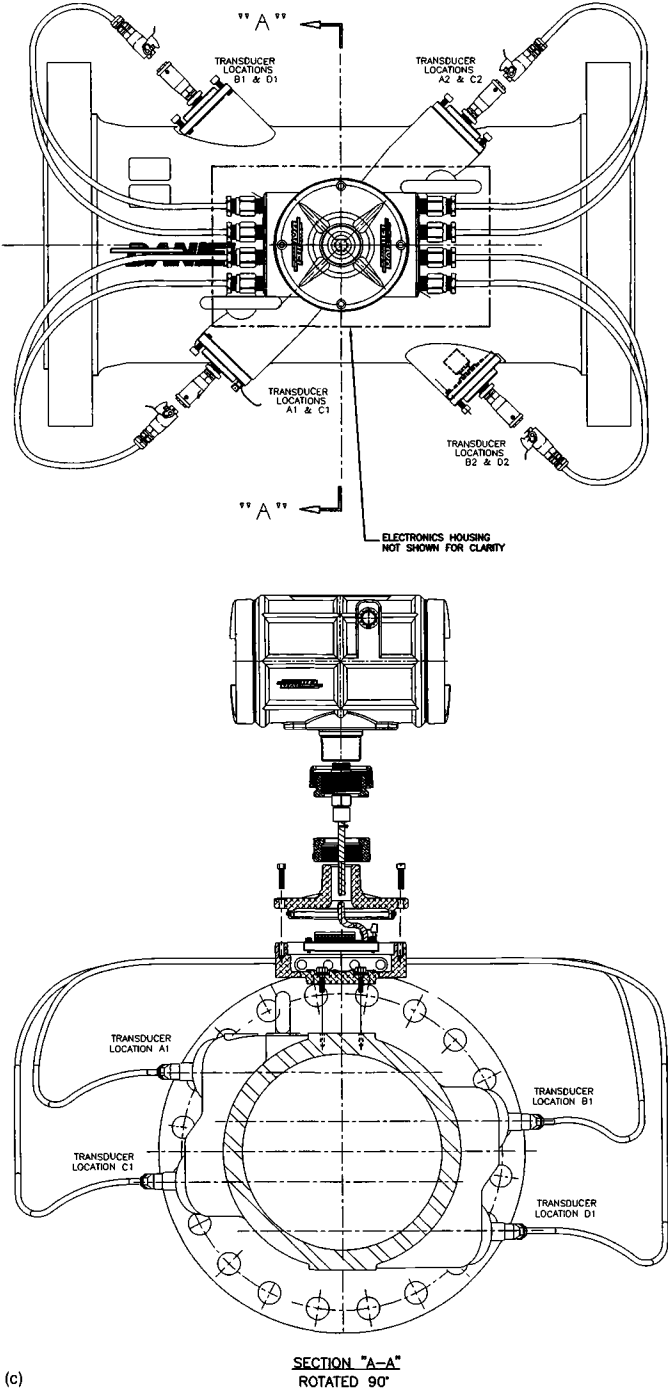


Figure 13.10. (continued).



(c)

Figure 13.10. (continued).

13.9 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

Manufacturers suggest that the meters are suitable for any homogeneous liquid that admits an ultrasonic wave (e.g., service water, seawater, and drainage), but not if bubbles or turbidity are present. However, one manufacturer cites applications such as city water, sewage water, industrial water, river water, sea water, and oil.

With its introduction into some national regulations for fiscal measurement of oil and gas, this meter is becoming more widely used in gas export stations (Sloet and de Nobel 1997) and flow line measurement (Agricola 1997). They can also provide self-monitoring (Sakariassen 1997).

Particular applications include:

- ❑ Flare-gas flow measurement (Raustein and Fosse 1991 of the Norwegian Petroleum Directorate) where ± 5 –10% measurement uncertainty appears to be common—This may be (Mylvaganam 1989) in a pipe of, say, 36-in. (0.91-m) diameter with velocities ranging from 0.3 to 80 m/s. The low flare is activated for 95% of the time, but this constitutes only about 10% of the gas flared. The pressure is only about 0.7 bar.
- ❑ Water flow in steam power plants, and hydroelectric plants (Erickson and Graber 1983).
- ❑ Pipeline management by clamp-on transit-time meters (Baumoel 1996) combined with leak detection.
- ❑ Nuclear fuel processing plant (Finlayson 1992) for both nonradioactive and radioactive liquid flows.
- ❑ Fiscal gas metering (Hannisdal 1991).
- ❑ A general-purpose gas meter capable of use with air flows, nitrogen, argon, and chlorine.

Advantages include (Beeson 1995, Mylvaganam 1989) its nonintrusiveness, lack of moving parts, compactness, high rangeability, ease of installation, cost savings and improvements in maintenance with self diagnostics, accuracy unaffected by build-up of contaminants, fully pigable, bidirectional flow capability, no line pipe restriction, and unaffected by pulsation flow. NorAm found that a single-path flowmeter could be installed in a day, and hot-tapping can be used under pressure. Self-checking using signal strength and quality alert the instrument technician to problems.

de Vries et al. (1989) claimed that transducers could be installed in underground gas pipes from ground level, so that, using reflection mode, a measurement uncertainty of 2% could be achieved with a repeatability of 0.2%.

Problems when applied to gases include low efficiency of launching waves due to impedance differences, absorption increases with frequency, and beam bending in large flow lines with high speed flow.

Adverse effects may be present with pulsating flow, and Beeson (1995) suggested using asynchronous sampling techniques.

Many would share Cairney's (1991) sentiment that "it is a pity that [the transit-time clamp-on flowmeter] is still associated with Doppler meters"

13.10 DOPPLER FLOWMETER

13.10.1 SIMPLE EXPLANATION OF OPERATION

The doppler flowmeter depends on the doppler frequency shift, which occurs when sound bounces off a moving object as shown in Figure 13.11. In the doppler meter, the waves need to reflect off something moving with the flow. If they reflect off a stationary object, then they retain their wavelength and frequency. If, however, they reflect off a moving object, the wave fronts will hit the moving object with a time interval that is not the same as their period in a stationary medium. As a result, the reflected wave will have a new period, frequency, and wavelength. It is almost invariably a clamp-on design for liquids. Figure 13.11 is a diagram of the effect.

The arrangement for typical commercial doppler meters is also shown in Figure 13.11. The transmitting and receiving transducers may be in the same block and held on the outside of the tube. On the other hand, they may be in separate blocks (Figure 13.11) and can then be positioned on the same side or on opposite sides of the tube.

We are now faced with one of the major uncertainties in the operation of these devices. What is the velocity of the reflecting object compared with the axial mean velocity of the flow in the pipe? This depends (Figure 13.11) on

- what the reflecting surface is,
- where the reflecting surface is, and
- the velocity of the object and its relative velocity to the flow in size and angle.

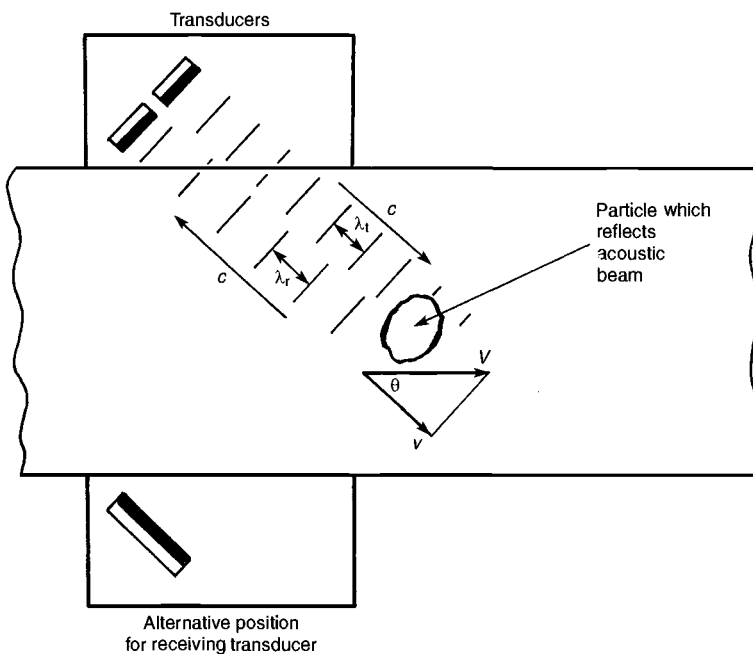


Figure 13.11. Diagram of doppler flowmeter showing the doppler effect and typical arrangements of the transducers for a commercial device.

We can start by taking the simplest assumption that the particle is moving with the flow and that path, therefore, makes an angle with the axis of θ . So in terms of V , the velocity in the pipe is

$$\Delta f = 2 f_t \frac{V}{c} \cos \theta \quad (13.11)$$

We can obtain a typical value of Δf assuming a transmission frequency f_t of, say, 5 MHz and a flow rate of water of 10 m/s. Δf will then have the value, assuming 1414 m/s for sound, of about 50 kHz. Clearly, this will not be one sole frequency coming back from the fluid, but rather a range of frequencies, and some method will be built into the converter to identify the favored frequency.

At least one manufacturer appears to offer a profile measurement capability. This is usually achieved by range gating the returning signal so that the distance of penetration into the pipe is known and hence the velocity at that point.

Uncertainty claims may be $\pm 2\%$ of full scale, although great care and understanding of the flow is needed in gaining confidence in the reading from these devices.

Poor mounting can cause spurious reflections, and pipe vibration may give false flow signals. Transmit and receive may be in one transducer block or may be in separate blocks allowing positioning on opposite sides of the pipe (Figure 13.11).

13.10.2 OPERATIONAL INFORMATION

One manufacturer suggests that installation should allow 6D upstream and 4D downstream. Transducers should be mounted adjacent to each other for large pipes but on opposite sides for small pipes. I noted the following comment: "... the pulses are reflected from a large area of the flow profile, giving a good representation of mean velocity. Repeatability of measurement is to specification, and with 'on-site' calibration checked, volumetric accuracy is assured." I would be less certain than this particular manufacturer, until I had the experience of a prolonged set of calibrations and a usage log that showed consistency.

Ranges are from, say, 0.3 to 6 m/s, with a temperature range of -20 to 80°C .

13.10.3 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

Cairney (1991) commented on doppler flowmeters saying that "these were the first type of ultrasonic flowmeter produced commercially. Experience has shown that they were oversold as an all-purpose flowmeter. When there were particles in the fluid measured they have been of some use, but most fluids flowing through power station pipelines are clean, which makes them far less effective."

Manufacturers suggest that these meters can be used for mining slurries, coal slurries, sewage, sludge, raw water, sea water, pulp (paper), acids, emulsion paint, fruit juice, yogurt, citric acid, glucose, contaminated oil, cement slurry, lime slurry, and industrial effluent.

13.11 CORRELATION FLOWMETER

13.11.1 OPERATION OF THE CORRELATION FLOWMETER

If two beams cross the flow at a known distance apart L , as in Figure 13.12(a), and the received signals are compared, as in Figure 13.12(b), to find a similar pattern, the

pattern for channel B will be found to be displaced a time τ_m from that for channel A. From this, it is simple to deduce that the flow has taken time τ_m to move a distance L or

$$V = \frac{L}{\tau_m} \quad (13.12)$$

Correlating the two signals requires some complex electronics. The fluctuation in the signal must be adequate, and for this reason, it will probably need to be created artificially by an upstream disturbance in the flow, if heavy turbulence, bubbles, second phase, etc., are absent naturally. The mathematical concept of cross correlation is defined by the equation (Keech 1982)

$$R_{yx}(\tau) = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T y(t)x(t + \tau)dt \quad (13.13)$$

where $x(t)$ and $y(t)$ are the upstream and downstream signals, respectively. The value of time delay τ_m corresponding to the maximum of $R_{yx}(\tau)$ provides a measure of the flow transit time between the two beams spaced L apart. The calculation of the integral in Equation (13.13) is achievable with modern computation. The problem is to do it cheaply and fast. For instance, sufficient precision may be achievable by using only the polarity of the signal, and by cutting the signal into sufficiently small bits of fixed size, but by varying polarity, the integration may be simplified and speeded up without significant loss of precision. (However, see Yang and Beck 1997.)

It has found a particular niche in multiphase flows where the disturbance to the signal will be substantial. However, Keech (1982) suggested that some form of validation is needed because the system can give incorrect estimates of velocity: spurious noise can give a correlation, decay of disturbances at low flows, other movement such as cavitation-generated shock waves or intense pressure waves, and oscillatory flows. Multichannel devices may overcome some of these effects.

13.11.2 INSTALLATION EFFECTS

Paik et al. (1994) used clamp-on transducers with 1-MHz crystals, 35° wedges, and 2D or 3D spacing in some tests of installation effects. Their baseline tests were with 120D upstream straight pipe. The errors found by them are shown in Table 13.9.

There may be other experimental studies of installation effects of which I am not aware, but because this is the only data provided in this book, it should probably be used with caution.

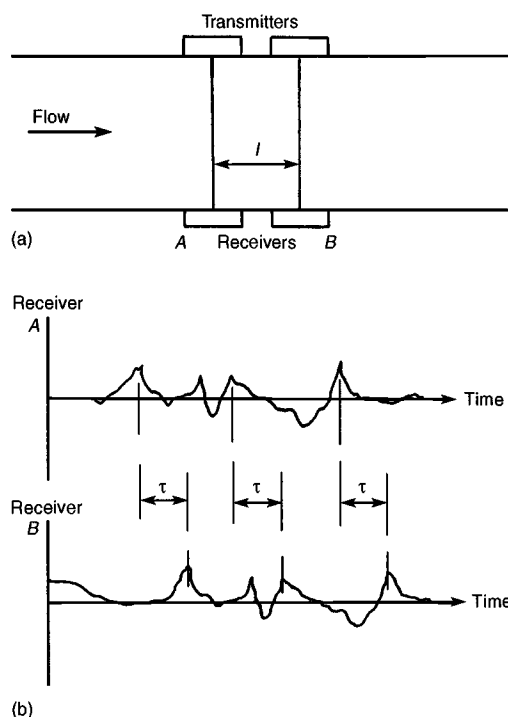


Figure 13.12. Correlation flowmeter: **(a)** Diagram of the geometry; **(b)** Traces from channels A and B.

Table 13.9. Error in signal due to upstream fittings

Fitting	Diameters Upstream			
	5	10	20	30
Single elbow	-7%	-4%	(-2%)	(-2%)
Double elbow out-of-plane	-2 to -4%	+5%	+2 to 6%	0 to 4%
Double elbow in-plane	-6%	-5%	-2%	-2%
() possible scatter.				

13.11.3 OTHER PUBLISHED WORK

King (1988; cf. King et al. 1988 and Sidney et al. 1988a, b) mentioned the development of an ultrasonic cross-correlation meter for multiphase flow at the National Engineering Laboratory, Scotland, in collaboration with Moore, Barrett, and Redwood. This development undoubtedly exploits one of the most important applications of this technique, and increasingly sophisticated electronics should allow more information about the flow components and their velocities to be deduced.

Coulthard and Yan (1993c) provided one of the many reviews of correlation flowmeters. They review results for two-phase mixtures:

Solids and liquids	Satisfactory operation
Oil and gas	At void fractions up to 25% and velocities up to 7 m/s scatter of results was $\pm 4.5\%$ without mixing and $\pm 2\%$ with mixing

Single-phase gas and liquid are also usable but may need a bluff body to enhance the signal.

Xu et al. (1994) described a clamp-on ultrasonic cross-correlation flowmeter for liquid-solid two-phase flows, with brief indications of how the various components operate. Calibrations of these meters against an electromagnetic meter gave differences of about $\pm 5\%$ for a 0.4% by weight paper pulp flow and about $\pm 3\%$ for a crude oil flow.

Recent work by Battye (1993) suggested measurement uncertainty over 10:1 of $\pm 2\%$ with 0.5% scatter, but that temperature variation in the acoustic path, if not allowed for, might reduce linearity. Other work by Kim et al. (1993a) suggested $\pm 2.2\%$ or for clamp-on $\pm 2.8\%$. A very recent paper by Yang and Beck (1997) claimed that, with a highly intelligent cross-correlator, a measurement accuracy of 1% could be achieved.

Lemon (1995) discussed a technique referred to as acoustic scintillation flow measurement; it appears to refer to, or be related to, cross-correlation flow measurement using the disturbances caused to the ultrasonic beams in traversing the flow.

It should be remembered that even though cross-correlation has been used with ultrasonics and included here for that reason, it has much wider application. Chen et al. (1993) reported work using two pairs of radiation sensors. Their application was to medium-consistency pulp in the paper industry. Flows approach Newtonian

at high rates but at low flows behave more like a plug with a laminar annulus of continuous pure liquid between plug and wall. As flow increases, the annulus flow becomes turbulent with lower consistency suspension. Eventually complete plug disruption takes place. Flow rates in industry are typically 0.3–3 m/s. The correlation can be based on inhomogeneities, flocs in the annulus, or temperature changes, and precision of 2.5% may be possible.

Beck and Plaskowski (1987) have given a full statement on this topic.

13.11.4 APPLICATIONS, ADVANTAGES, AND DISADVANTAGES

The cross-correlation ultrasonic flowmeter appears to require a significant disturbance in the flowing fluid. Where the disturbance is due to multiphase flow, the meter comes into its own and provides one of the few types that can satisfactorily deal in a nonintrusive way with such flows. Its current price may make it less attractive for normal applications, but it is more in line with the overall costs of applications such as oil well monitoring.

13.12 OTHER ULTRASONIC APPLICATIONS

Flowmeters could be designed to make use of the beam sweeping caused by the flow. These have been proposed, but I am not aware of any that use only this principle. Figure 13.13 shows a beam crossing the flow approximately at right angles but being swept sideways due to the fluid motion. On the receiving side of the flow passage, two or more sensors measure the strength of the received signal at each sensor and so allow the electronics to deduce the deflection due to the flow. The angular deflection β is given by

$$\tan \beta = \frac{V}{c} \quad (13.14)$$

and the deflection on the receiving wall will be

$$z = \frac{VD}{c} \quad (13.15)$$

Ultrasonic sensing is also used in open channel flowmeters, flumes, and weirs to sense the level of the upstream surface from which is deduced the flow rate (Herschy 1995).

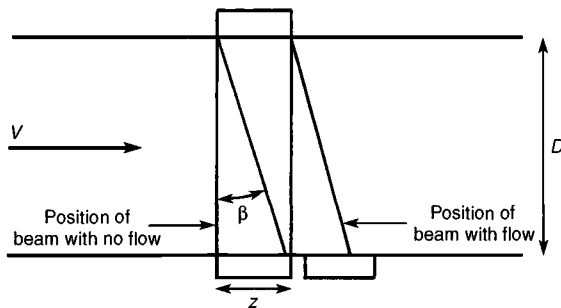


Figure 13.13. Flowmeter concept using beam sweeping.

In vortex meters, one method of sensing the oscillation of the shedding is ultrasound, which is disturbed by the vortices and provides a signal that can be used to extract the frequency of shedding.

Coulthard and co-workers have also suggested using the vortex-shedding combined with a correlation-sensing system to obtain flow measurement. It should be noted in this technique that vortices may move faster than the fluid.

Teufel et al. (1992) described a new technique for measuring the profile in pipes and applied it to oscillating flows. The object of the paper was to compare LDV and the new ultrasonic velocity profile sensor. They saw the techniques as complementary, and it is possible that the second may offer an industrial approach where profile determination is important.

Olsen (1991) described tests of what appeared to be an anemometer using sonic and ultrasonic signals and allowing velocities in a wind tunnel to be measured in three directions simultaneously.

Joshi (1991) describes a surface-acoustic-wave (SAW) flow sensor. The oscillation frequency is temperature dependent, and so if the element is heated above ambient and placed in the flowing gas, the frequency will be flow-rate dependent.

Guilbert et al. (1996) used a pulse of heat through the pipe wall to mark the liquid, and the passing of the marked liquid was sensed by an ultrasonic beam using the transit time across the pipe for cold and hot liquid.

13.13 CHAPTER CONCLUSIONS

One feature of ultrasonic flowmeters is the large amount of information available from the system, but not completely used, which will give indication of operating changes or problems. In my earlier book (1988/9), I suggested that ultrasonic technology might develop a clamp-on meter (possibly a master meter) that sensed:

- ❑ wall thickness,
- ❑ the quality of the inside of the tube,
- ❑ the turbulence level,
- ❑ profile from a range-gated doppler system,
- ❑ flow measurement from a transit-time system,
- ❑ correlation to provide information about a second phase,
- ❑ density from the impedance and sound speed, and
- ❑ condition (self) monitoring.

Such a device could measure diameter and other pipe details and could program a much simpler device for permanent installation at the site.

Some of these features now appear together. Sanderson and Torley (1985) combined wall thickness and some dimensional and condition monitoring, with intelligent control. Range-gated doppler has been exploited in medical applications. Birch and Lemon (1995) discussed the use of doppler on multiple paths with range gating to obtain profiles of flows in open channels and discharges. A commercial device may allow operation in both transit-time and doppler modes. Correlation has been

used for multiphase oil and gas flow measurement. Guilbert et al. (1996) have used the device for mass flow measurement. Lynnworth (1990) has found that the phase velocity of flexural waves in the wall of small diameter pipes was dependent on the density of the liquid inside. As the technology becomes more widely accepted, the microprocessor control is likely to become more powerful and to introduce multiple facilities.

The extraordinary success in developing instruments for domestic gas measurement, which are within the tight budget for the utilities, operate off a battery with 10 years or more of life, and have sophisticated external communication ability, will, with little doubt, provide precursors of much more in this area.

Perhaps the ultrasonic flowmeter is the realistic rival to the Coriolis meter in gas mass flow measurement, at lower cost, and of greater versatility.

Much development work continues as shown by the preceding references. Bragg and Lynnworth (1994, cf. Lynnworth 1989, 1994) proposed the use of a single port with two axially spaced transducers. As an example, they suggested that a spacing of 60 mm axially for a velocity of 1 m/s and a sound speed of 343 m/s would result in a 1- μ s transit-time difference or a 10-ns resolution (i.e., 1%). To avoid cross-talk between the transducers, they investigated O-rings of various materials and found that silicone was highly attenuating.

The application at which this is aimed is firstly noncombustible gases, in which category some stack gases fall.

They also suggested that two angled transducers in the same port can transmit and receive a signal that forms a triangular path in the pipe cross-section and thus senses swirl. Lynnworth et al. (1994) also developed a profile-measuring method using the one port with either a traversing reflector or a series of fixed reflectors.

Ultrasonics combined with other methods (e.g., Guilbert et al. 1996) such as thermal marking may provide new scope.

Another interesting development is the insertion meter of Rawes and Sanderson (1997) discussed in Chapter 18, which could revolutionize in situ calibration in liquids and gases.

This is such a powerful technology that it is starting to be seen as capable of meeting fiscal demands for metering liquid hydrocarbons; therefore, limits of its precision and reasons for baseline instability need to be understood urgently.

APPENDIX 13.A

Simple Mathematical Methods and Weight Function Analysis Applied to Ultrasonic Flowmeters

13.A.1 SIMPLE PATH THEORY

Transit-Time

Figure 13.2 provides a diagram on which to base the mathematics. We can work in terms of trigonometric functions or dimensions.

We start with the downstream-going wave and note that its speed will be greater than the sound speed by the component of the flow in the direction of the path. This component is $V \cos \theta$. The distance to be traveled is $D \operatorname{cosec} \theta$, where D is the pipe diameter so that the time taken for the downstream wave to travel from the transmitter to the receiver is

$$t_d = \frac{D \operatorname{cosec} \theta}{c + V \cos \theta} \quad (13.A.1)$$

A similar expression for the upstream wave time t_u is

$$t_u = \frac{D \operatorname{cosec} \theta}{c - V \cos \theta} \quad (13.A.2)$$

We can now obtain the difference between these two wave transit times (Figure 13.3)

$$\begin{aligned} \Delta t &= t_u - t_d \\ &= \frac{D \operatorname{cosec} \theta}{c - V \cos \theta} - \frac{D \operatorname{cosec} \theta}{c + V \cos \theta} \\ &= \frac{D \operatorname{cosec} \theta / c}{1 - \frac{V}{c} \cos \theta} - \frac{D \operatorname{cosec} \theta / c}{1 + \frac{V}{c} \cos \theta} \end{aligned}$$

Combining the two fractions on the right-hand side over a common denominator and ignoring terms in $(V/c)^2$, we obtain

$$\Delta t = \frac{2VD \cot \theta}{c^2} \quad (13.A.3)$$

Thus we see that, provided we know the speed of sound, we can find the value of V from the time difference and the geometry of the meter. The requirement to know the value of c is not difficult if we measure, as well as the time difference, the actual transit times each way and take the mean t_m . We can make the approximation

$$\begin{aligned} t_m^2 &= (t_u - \Delta t/2)(t_d + \Delta t/2) \\ &= t_u t_d + \Delta t(t_u - t_d)/2 - (\Delta t)^2/4 \\ &= t_u t_d + \Delta t(\Delta t)/2 - (\Delta t)^2/4 \\ &= t_u t_d + (\Delta t)^2/4 \\ &\approx t_u t_d \end{aligned} \quad (13.A.4)$$

where, using values in Table 13.1, we have neglected a term of about 10^{-6} . We can now replace c^2 by $L^2/t_u t_d$ and $\cot \theta$ by X/D . We can then rewrite Equation (13.A.3) by

$$\Delta t = \frac{2VX t_u t_d}{L^2} \quad (13.A.5)$$

From this, we can obtain Equation (13.5).

Sing-Around

An alternative way to eliminate sound speed is the sing-around method (Figure 13.4) discussed by Suzuki et al. (1975, cf. 1972). In this method, each of the paths, upstream and downstream, is operated in such a way that when the pulse is received by the receiving transducer, it triggers a new pulse from the transmitting transducer. Thus the frequency of pulses is dependent on the velocity of sound and of the flow. We can obtain the period between pulses from Equations (13.A.1) and (13.A.2), and the pulse frequency will be the inverse of the period, or for the downstream-going pulses

$$f_d = \frac{c + V \cos \theta}{D \operatorname{cosec} \theta} \quad (13.A.6)$$

A similar expression for the upstream pulse train is

$$f_u = \frac{c - V \cos \theta}{D \operatorname{cosec} \theta} \quad (13.A.7)$$

If we now take the difference between these frequencies, we obtain

$$\begin{aligned} \Delta f &= \frac{2V \cos \theta \sin \theta}{D} \\ &= V \sin 2\theta / D \end{aligned} \quad (13.A.8)$$

We can write this in terms of dimensions, since $2 \cos \theta \sin \theta = 2DX/L^2$, as

$$\Delta f = 2VX/L^2 \quad (13.A.9)$$

which has the virtue of relating V to the frequency difference without requiring the value of c , the sound speed. Again we can obtain typical values for the frequencies from Table 13.2 (Baker 1988/9). Making the approximation that V is constant across the pipe, we can obtain Equation (13.7).

13.A.2 USE OF MULTIPLE PATHS TO INTEGRATE FLOW PROFILE

The preceding simple analysis assumed that the flow profile is uniform. In fact, if we allow for varying velocity along the path of the ultrasonic beam, Equation (13.A.3) may be rewritten in the form of an integral

$$\begin{aligned} \Delta t &= \frac{2 \cot \theta}{c^2} \int_{\text{along chord}} V(y) dy \\ \Delta t &= \frac{2 V_m Y \cot \theta}{c^2} \end{aligned} \quad (13.A.10)$$

The flowmeter obtains the average velocity, V_m , along the path of the beam.

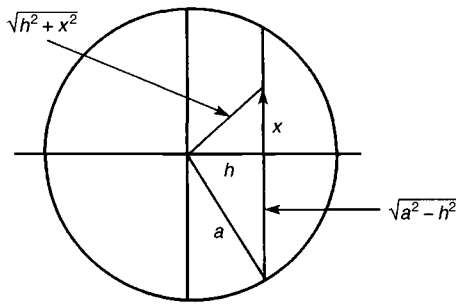


Figure 13.A.1. Geometry of the pipe cross-section for chordal integration.

For integration along the chord shown in Figure 13.A.1,

$$\Delta t = \frac{2 \cot \theta}{c^2} \int_{-\sqrt{a^2-h^2}}^{+\sqrt{a^2-h^2}} V(\sqrt{h^2+x^2}) dx \quad (13.A.11)$$

If the path is on the diameter, two extremes of laminar profile and uniform profile yield a Δt ratio for equal volumetric flows of 4 : 3. Thus an ultrasonic flowmeter with diametral sensing has an intrinsic calibration shift across the laminar-turbulent profile change approaching 33%.

Baker and Thompson (1975, 1978) showed the improvement possible by using approximately midradius transducers and recalculating Equation (13.A.11) for values of $h \approx a/2$. By using the approximate expression for the turbulence velocity profile of

$$V(r) = \frac{(1+2n)(1+n)}{2n^2} V_m \left(1 - \frac{r}{a}\right)^{1/n} \quad (13.A.12)$$

where n varies with Reynolds number as

Re	4×10^3	2.3×10^4	1.1×10^5	1.1×10^6	2.0×10^6 to 3.2×10^6
n	6.0	6.6	7.0	8.8	10

the signal variation can be obtained as in Figure 13.6.

The basis for using more than two paths lies in the mathematics of numerical integration. Linked to the name of Gauss, it can be shown that a polynomial of the form

$$f(t) = c_0 + c_1 t + c_2 t^2 + c_3 t^3 + \cdots + c_{2n+1} t^{2n+1}$$

can be precisely integrated as

$$\int_{-1}^{+1} f(t) dt = A_0 f(t_0) + A_1 f(t_1) + A_2 f(t_2) + \cdots + A_n f(t_n) \quad (13.A.13)$$

Or a polynomial of degree $2n+1$ or less can be correctly integrated by an expansion such as in Equation (13.A.13) provided the constants A_n and the positions t_n are correctly chosen. It is also possible to show that an integral can be rearranged to the interval -1 to $+1$. Thus with four positions and with suitable weighting, the integral of a polynomial of up to order seven can be correctly evaluated.

It is as a result of considerations such as these, and possible refinements based on likely profiles in the flow, that weightings for multiple paths (cf. Vaterlaus 1995) have been obtained.

13.A.3 WEIGHT VECTOR ANALYSIS

Hemp (1982) has developed the ideas from electromagnetic flowmeter theory for ultrasonic flowmeters. As in the case of electromagnetic flowmeters, the condition for an ideal flowmeter, one where the signal is independent of flow profile, is

$$\nabla \times \mathbf{W} = 0 \quad (13.A.14)$$

and Hemp obtained a flow signal expression using his reciprocal theory of

$$\pm (U_I^{(2)} - U_I^{(1)})I = 2 \int \mathbf{V}_s \cdot \mathbf{W} dV \quad (13.A.15)$$

where the negative sign is for electromagnetic and/or magnetostrictive transducers and the positive one for electrostatic and/or piezoelectric, I is the driving current, $U_I^{(2)}$ and $U_I^{(1)}$ are the received voltages, and $\mathbf{V}_s$ is the undisturbed flow. The weight vector is then given in terms of ρ_m the mean density in the meter and $\mathbf{v}_0$ the acoustic field for two distinct cases (1) and (2) when opposite transducers transmit as

$$\mathbf{W} = \rho_m \left[(\mathbf{v}_0^{(2)} \cdot \nabla) \mathbf{v}_0^{(1)} - (\mathbf{v}_0^{(1)} \cdot \nabla) \mathbf{v}_0^{(2)} \right] \quad (13.A.16)$$

Hemp suggested that a flowmeter in a rigid-walled duct with plane wave modes, a meter with radial transmission, and a meter with a large area transmission (e.g., the industrial design for small pipes) are approaching ideal configurations.

In addition to the fundamental weight vector [Equation (13.A.16)], it is possible to define weight vectors for phase shift and amplitude shift (in sine wave operation) and for pulse operation (Hemp 1998).

13.A.4 DOPPLER THEORY

Figure 13.11 is a diagram of the effect. A wave approaches a particle moving with velocity component v in the direction of the acoustic wave, which has a velocity c , and is reflected. The velocity remains at c relative to the medium after reflection because sound speed is not affected by frequency (within reasonable limits). But the period between two waves being reflected will not be λ_t/c . The wave will hit the particle at a velocity $c - v$, and so peaks will hit the particle every $\lambda_t/(c - v)$ seconds. But successive peaks will make contact with the moving particle at different particle positions, so the second peak will hit the particle $\lambda_t/(c - v)$ seconds after the first but will then need to travel $\lambda_t V/(c - v)$ to reach the starting point of the first peak. This additional travel takes $\lambda_t V/c(c - v)$ seconds. So the time between peaks is τ given by

$$\tau = \frac{\lambda_t}{c} \frac{1 + v/c}{1 - v/c} \quad (13.A.17)$$

Expanding Equation (13.A.17) using the binomial theorem, the frequency of the reflected wave f_r becomes (ignoring $\frac{v^2}{c^2}$)

$$f_r = \frac{c}{\lambda_t} \left(1 - \frac{2v}{c} \right) \quad (13.A.18)$$

Hence, the frequency shift Δf is given by

$$\begin{aligned} \Delta f &= f_t - f_r \\ &= 2f_t \frac{v}{c} \end{aligned} \quad (13.A.19)$$

Mass Flow Measurement Using Multiple Sensors for Single- and Multiphase Flows

14.1 INTRODUCTION

The measurement of mass flow, a fundamental requirement for any fluid, has been an elusive goal due to the problems of developing a suitable flowmeter. Despite this, the availability of mass flowmeters has increased greatly over the last 15 years. This is partly due to the increasing value of products (Hall 1990), but it is also due to an increasing realization that volumetric flow measurement is often inappropriate. In addition, the advent of the Coriolis flowmeter has stimulated engineers to find other mass flowmeters. General reviews of mass flowmeters were given by Sproston et al. (1987), Betts (1990), and Medlock and Furness (1990).

Mass flow measurement is commonly categorized as direct (true) or indirect (inferential). However, it may be useful to allow a few more than two categories.

- a. True (direct) mass flow measurement by a single instrument is rare. It appears that to achieve it we need to use one of the fundamental acceleration laws. We can do this by creating:
 - the force (or torque) resulting in a linear (or angular) acceleration (Chapter 16 describes an example) or
 - the force that produces Coriolis acceleration (Chapter 17).
- b. Fluid-dependent thermal mass flow measurement uses the temperature rise resulting from heat addition but is affected by other parameters such as the specific heat of the fluid (Chapter 15).
- c. Multiple differential pressure flowmeters used in a dedicated system (Section 14.2) depend on the nonlinearity of the flowmeter equation.
- d. Indirect (inferential) mass flow measurement combines volumetric flow rate or momentum flow rate with a density measurement (Section 14.3). (An alternative is to combine volumetric and momentum flow measurements.)
- e. Multiphase flow measurement almost always requires multiple measurements (Section 14.4).

14.2 MULTIPLE DIFFERENTIAL PRESSURE METERS

Before turning to the one method that is commercially available, for completeness we shall look at two methods that have been suggested but, to my knowledge, have not been used commercially.

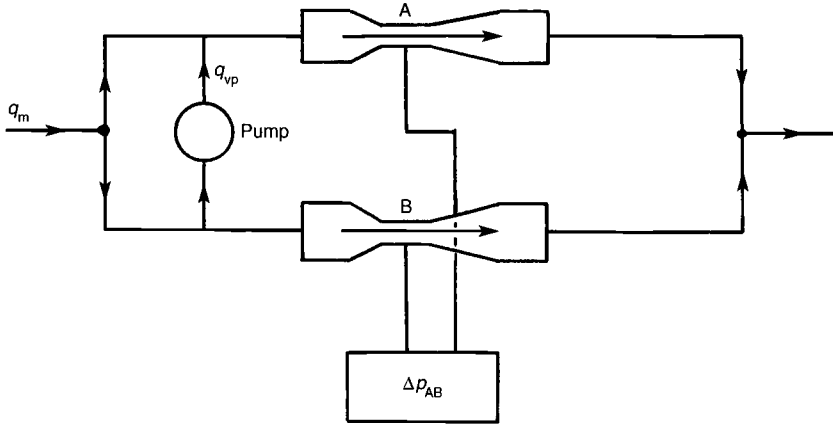


Figure 14.1. Twin venturi system.

Twin Venturi System

The twin venturi system depends on the use of two identical venturi meters (or other differential pressure devices) that are arranged in parallel balanced paths as shown in Figure 14.1. By means of a metering pump, which transfers q_{vp} from one line to the other, the flows may be unbalanced. If the pump is off, so that the flow is split equally between the two paths and is, therefore, the same in each meter, the differential pressure across each meter is also the same. If the pump is started, then one venturi meter has a lesser and the other a greater flow passing through it. The pressure difference between the throat pressures Δp_{AB} is then measured, and the mass flow rate q_m is deduced.

We first note that the pressure drop to the throat of each venturi is given by

$$\Delta p_A = \Delta p_B = K(q_m/2)^2/\rho \quad (14.1)$$

where K is a constant, ρ is the density of the fluid, and q_m is the total mass flow through both venturis, and so $q_m/2$ is the total mass flow through one of the venturis when the flows are equal.

If the main flow q_m splits equally, then if the pump is started and transfers fluid as shown, and provided that the upstream pressures at inlet to the venturis are not affected by the transfer, the equation for the pressure drop to the throat of A will be

$$\Delta p_A = K(q_m/2 + \rho q_{vp})^2/\rho \quad (14.2)$$

and to the throat of B

$$\Delta p_B = K(q_m/2 - \rho q_{vp})^2/\rho \quad (14.3)$$

Thus the differential pressure will be

$$\begin{aligned} \Delta p_{AB} &= K(q_m/2 + \rho q_{vp})^2/\rho - K(q_m/2 - \rho q_{vp})^2/\rho \\ &= K\left(\frac{q_m^2}{4} + \rho q_m q_{vp} + \rho^2 q_{vp}^2\right)/\rho - K\left(\frac{q_m^2}{4} - \rho q_m q_{vp} + \rho^2 q_{vp}^2\right)/\rho \\ &= 2Kq_m q_{vp} \end{aligned} \quad (14.4)$$

Hence

$$q_m = \frac{\Delta p_{AB}}{2Kq_{vp}} \quad (14.5)$$

Thus the precision of q_m depends on Δp_{AB} , K the venturi constant assumed to be the same for each meter, and q_{vp} , which is set by the pump. However, there are clearly various effects such as the incompressibility of the fluid and the disturbing effect of the transfer flow, which would need careful consideration.

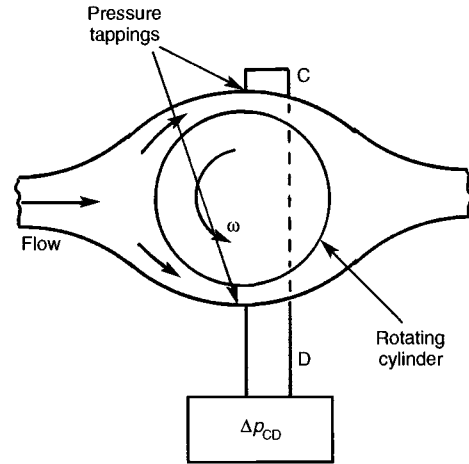


Figure 14.2. Brand and Ginsel's system.

Brand and Ginsel's System

This is illustrated in Figure 14.2. It has similarities to the twin venturi method.

The flow on each side of the rotating cylinder passes through a contraction. An imbalance in the flow is introduced by rotating the central cylinder and the pressures at the contraction throats are again compared.

Medlock (1989) refers to an MSM1 flowmeter made by Professor Dr. W. J. D. van Dijk and associates at the Technical University of Delft. The MSM1 meter (Massa Stroon Meter) was $\frac{3}{4}$ in. and consisted of a solid cylindrical rotor spinning at a constant angular velocity within the meter casing. The rotor induced circulation, and so half of the annulus passed more fluid, and the other half passed less. A differential pressure was established between the two throats in the same way as in the differential venturi system, and the mass flow rate was found to be proportional to the differential pressure and the rotational speed.

14.2.1 HYDRAULIC WHEATSTONE BRIDGE METHOD

A Wheatstone bridge system has been successfully developed as a commercial instrument.

14.2.2 THEORY OF OPERATION

The theory of operation of this type of flowmeter is illustrated by Figure 14.3. Four matched orifices are arranged in a hydraulic Wheatstone bridge network with a constant volume pump producing a recirculating flow. The process mass flow is q_m , and the pump volumetric flow is q_{vp} . The pressure drop is given by Δp_{BD} in Figure 14.3(a) for the situation when $q_m > \rho q_{vp}$ (high flows). If the flow q_m were to divide equally between the upper and lower halves of the Wheatstone bridge, then the pressure drop through each would be [as for Equation (14.1)]

$$\Delta p = K(q_m/2)^2/\rho$$

Applying this relationship to the four orifices in the bridge, when the pump creates an unbalanced flow, as in Figure 14.3(a), we obtain the following

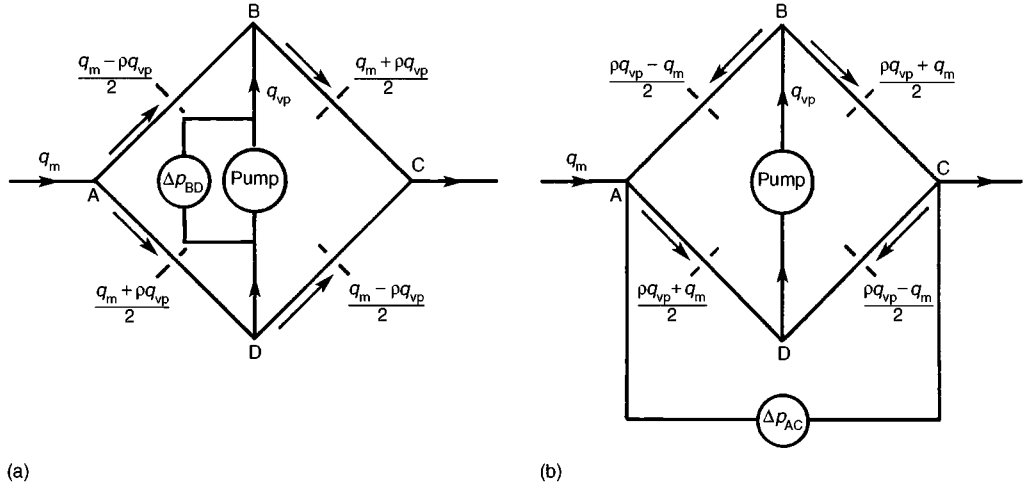


Figure 14.3. Wheatstone bridge flowmeter: **(a)** High flows; **(b)** Low flows.

expressions:

$$\Delta p_{BD} = \Delta p_{AD} - \Delta p_{AB} \quad (14.6)$$

$$= K(q_m/2 + \rho q_{vp}/2)^2/\rho - K(q_m/2 - \rho q_{vp}/2)^2/\rho \quad (14.7)$$

$$= K q_m q_{vp} \quad (14.8)$$

$$q_m = \frac{\Delta p_{BD}}{K q_{vp}} \quad (14.9)$$

Similarly, for the case of Figure 14.3(b), the pressure difference is given by Δp_{AC} when $q_m < \rho q_{vp}$ (low flows), and we obtain

$$\begin{aligned} \Delta p_{AC} &= \Delta p_{AB} + \Delta p_{BC} \\ &= -K(q_m/2 - \rho q_{vp}/2)^2/\rho + K(q_m/2 + \rho q_{vp}/2)^2/\rho \end{aligned} \quad (14.10)$$

$$= K q_m q_{vp} \quad (14.11)$$

$$q_m = \frac{\Delta p_{AC}}{K q_{vp}} \quad (14.12)$$

14.2.3 INDUSTRIAL EXPERIENCE

The meter consists of four precisely matched orifices forming the hydraulic Wheatstone bridge. A recirculating pump establishes reference flow in the bridge. At zero measured flow rate, pressure drop across the bridge is zero. As measured flow passes through the meter, it upsets the balance in the bridge such that a differential pressure signal, which is linear and proportional to true mass flow rate, is generated.

The resulting flowmeter is claimed to give true mass flow rate and may be unaffected by changes in fluid, temperature, density, and viscosity. It can be of stainless steel construction and capable of operation at elevated temperatures and with low lubricating liquids.

Typical flow ranges may be

- 0–2.3 kg/h (0–5 lb/h) for microflows with a turndown of 50:1 and a minimum flow of 0.05 kg/h;
- 0–900 kg/h (0–2,000 lb/h) for low flow ranges and 0.5 kg/h minimum flow for low ranges; and
- 0–23,000 kg/h (0–50,000 lb/h) for high flow ranges with about 100:1 turndown.

A wide range version in the low flow range may be capable of 500:1 turndown. Best accuracy may be of order 0.5% of rate plus 0.02% FSD, and repeatability 0.25% of rate although in some designs an additional amount based on FSD may be added. Pressure drop may be in the range from about 0.1 to 2 bar. Depending on the design, fluid temperature may be in the range from about –18 to +150°C and pressure up to about 67 bar gauge.

14.2.4 APPLICATIONS

This device is particularly suited for fuel flow measurement applications such as engine emission control and fuel economy testing. Other applications may be fuel efficiency testing in boilers. It produces an output signal that is linearly proportional to mass flow rate.

14.3 MULTIPLE SENSOR METHODS

If the fluid has unknown properties, we shall probably need to combine the flow measurement with a density measurement. However, if the fluid has known parameters, we may be able to use equations that describe the fluid's state. The following are examples of both.

- A volumetric meter (e.g., turbine) with density, or pressure and temperature measurement. This is a relatively common practice. A combination of electromagnetic flowmeter with a gamma density-sensing cell has been available commercially. The mass flow will be given by

$$q_m = \rho VA \quad (14.13)$$

where ρ is the fluid density, V is the mean velocity through the meter, and A is the cross-section of the meter. There is likely to be a correct position at which to measure the density. There will, therefore, be an overall uncertainty stemming from the various components

$$\frac{\delta q_m}{q_m} = \frac{\delta \rho}{\rho} + \frac{\delta V}{V} + \frac{\delta A}{A} \quad (14.14)$$

or the equivalent root-mean-square expression. Similar error calculations will be appropriate to the other methods mentioned here.

- A vortex meter that combines frequency measurement with measurement of lift on the shedding body. Itoh and Ohki (1993) claim a vortex flowmeter design that obtains mass flow rate by the use of the lift on the central body divided by the

frequency of pulsation. This depends on the assumption that the lift coefficient and the Strouhal number are both constant, so that the lift is proportional to ρV^2 , and the frequency is proportional to V . As a result, the ratio is proportional to ρV .

- Ultrasonic flowmeters. The speed of sound is related to the density

$$c^2 = \gamma p / \rho \quad (14.15)$$

where γ is the ratio of the specific heats, and ρ is the density of the fluid. A similar equation for liquids is

$$c^2 = 1 / (k_s \rho) \quad (14.16)$$

where k_s is the adiabatic compressibility. Because c^2 appears in the equation for time-of-flight ultrasonic meters, it can be combined through these equations to give velocity times density and hence mass flow. This assumes that for gases γ is constant, which is a good approximation for air. The value of γ is dependent on the number of atoms in the gas molecule. The value of p will be needed. For a liquid, the value of k_s will be needed (Baker 1976).

- Alternatively for ultrasonic flowmeters the impedance of the fluid is related to the density and may be used to obtain its value (Guilbert and Sanderson 1996a).
- A differential pressure flowmeter with the density of the fluid obtained either directly or, say, for a gas via pressure and temperature measurements (cf. Figure 5.20).
- A combination of velocity measurement to obtain V , with momentum ρV^2 , allows the value of ρV to be found.

Medlock used the term *hybrid* (1989) to describe meters combining momentum sensing and volume sensing, such as the venturi tube with an electromagnetic flowmeter built into it or venturi and turbine meters (Frank et al. 1977, Reimann et al. 1982) or the turbine meter that also senses drag on the wheel or on a separate drag plate (Reimann et al. 1982) or gauze (Cole 1985). Wong et al. (1981) suggested the use of the pressure drop in a swirl generator in combination with a venturi.

14.4 MULTIPLE SENSOR METERS FOR MULTIPHASE FLOWS

14.4.1 BACKGROUND

There is a growing need to measure the flow of fluids, which consist of more than one component. The nature of the second component may or may not be known, and the proportions or indeed the existence of the second or even third component may not be known. The reasons for this growing need are the increasing value of such fluids, being used in an ever widening variety of processes, and the increasing requirement to conserve energy. The fluids involved range across wide industry areas as diverse as food processing and subsea hydrocarbon extraction, but at present the technology appears mainly to be driven by the latter.

Against this background, most flowmeters have been designed to measure the single-phase flow of a Newtonian fluid but, despite this, are being called on,

sometimes unknowingly, to handle more complex fluids. Baker (1991a) attempted to draw together the references and the most useful data on the likely response of bulk flowmeters when placed in such flows (cf. Baker 1988, 1989). More recently, Whitaker (1993) has reviewed the latest developments.

These complex flows (discussed briefly in Chapter 2) have been extensively studied, and flow pattern maps have been developed to indicate the conditions under which the various flow regimes occur (Butterworth and Hewitt 1977, Hetsroni 1981). However these may be limited in their application to particular fluid combinations and pipe sizes. To measure the flow in any of these regimes requires a large number of variables to be interpreted from measurements. Mixing or conditioning of the flow may be possible in some applications but may not greatly improve the situation. Separation of the components is also possible in some applications, and some of the designs mentioned in Section 14.4.3 attempt this, but again this is not possible or convenient in many applications.

Some of the earliest data appear to be those that relate to a nuclear plant under emergency conditions. Experiments were undertaken, and special designs and combinations of meter were tried to obtain a system that would signal fault conditions in the nuclear reactor duct flows. The expected changes in the flows were considerable, and so precise flow measurements were not always essential.

Dykesteen (1992) identified the development of small oil wells in the North Sea, where a platform would be too costly, as one reason for needing a multiphase meter. Subsea reservoir management will need to take place without the use of a separator, and where different satellite wells are operated under different licences, individual flow measurement will be required. The multiphase meter should, if possible, provide component fractions, component velocities, and component densities.

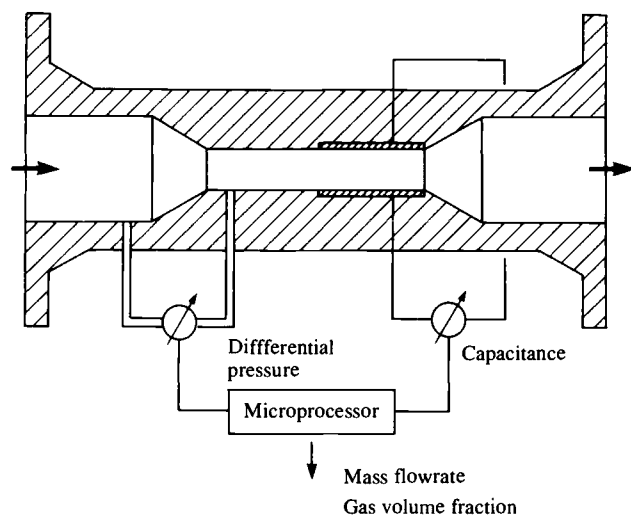
14.4.2 CATEGORIZATION OF MULTIPHASE FLOWMETERS

Dykesteen (1992) suggested that multiphase meters operated in one of three ways:

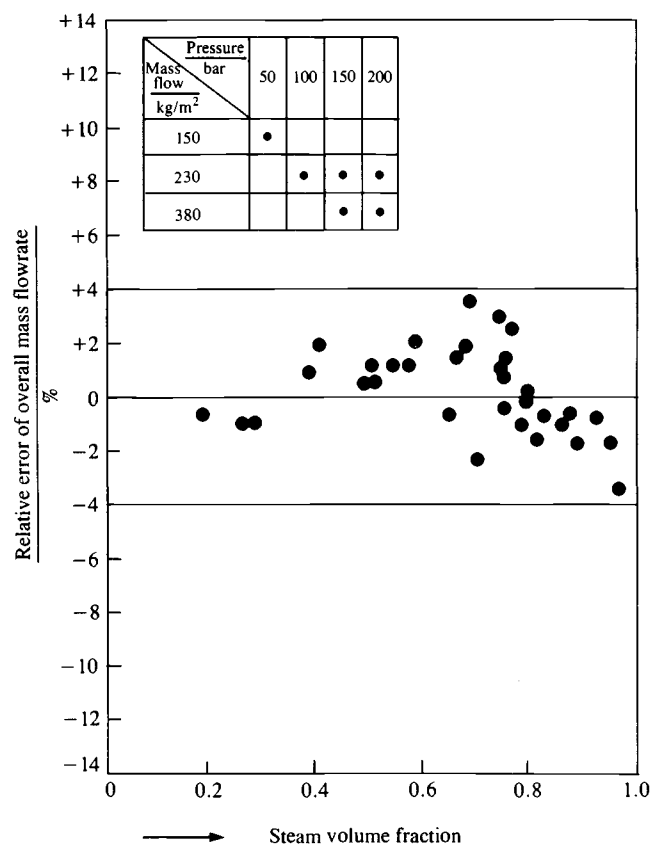
- i. by total separation of components;
- ii. by measurement of the multiphase flow rate but separation of a sample for component measurement; or
- iii. by in-line measurement of all parameters.

For applications other than oil production, (iii) has usually been the aim. An example is the early work in the nuclear industry such as the combination of turbine and target meters for loss-of-coolant nuclear applications (cf. Goodrich 1979 and other papers at the same conference). A more recent example is the venturi meter, which has been combined with both capacitance and gamma densitometer (Kratzer and Kefer 1988, Smorgrav 1990) with some impressive results [Figures 14.4(a,b)]. One advantage of the venturi is that it appears to homogenize the flow. Other examples are Arnold and Pitts' (1981) proposal for a positive displacement meter with a gamma densitometer, temperature and pressure sensing (Figure 9.20), and Priddy's (1994) tests of a positive displacement design (Figure 9.21).

Xiaozhang (1995) suggested the use of a magnetic flowmeter with large electrodes, differential (pressure) flowmeter, and capacitance/impedance transducer for a



(a)



(b)

Figure 14.4. Mass flowmeter due to Kratzer and Kefer (1988) (reproduced with permission of Cranfield University): (a) Geometry; (b) Error in steam-water flows.

multiphase flowmeter, particularly for oil well flows, but it would be necessary for the continuous phase to be conducting.

14.4.3 MULTIPHASE METERING FOR OIL PRODUCTION

The following list, which uses Dyksteen's (1992) categorization, is a compilation from papers by King (1990), Whitaker (1993), Whitaker and Millington (1993), and others (cf. Kjolberg and Berentsen 1997, who tested five meters from Fluenta, Framo, ISA, Kvaerner, and MFI, and Klemp et al. 1997 on field developments).

Separation of Components

The Texaco SMS subsea metering system (Figure 14.5) combines an inclined gravity separator, a microwave sensor in a side stream sampling loop for water cut, a differential pressure liquid flow rate meter, a vortex-shedding meter on the gas outlet, and temperature and pressure measurements (cf. Dean et al. 1990a, 1990b).

Measurement of the Multiphase Flow Rate, but Separation of a Sample for Component Measurement

Some early results were reported by Hall and Shaw (1988) and were for a multiphase flowmeter on the Rolf facility in the Danish section of the North Sea. This system depended on turbine meter, gamma densitometer, pressure, temperature, and flow separation. Typical error values for favorable conditions were 25 %, but some very severe errors were recorded in the water measurements. A commercial device similar to this claimed pipe size up to about 600 mm, pressure up to about 65 bar, temperature up to 400°C, and flows up to about 8,000 m³/h.

A system was reported (Anon. 1988, Kinghorn 1988) using a correlation flowmeter and gamma densitometer to monitor the main flow after jet mixing and took

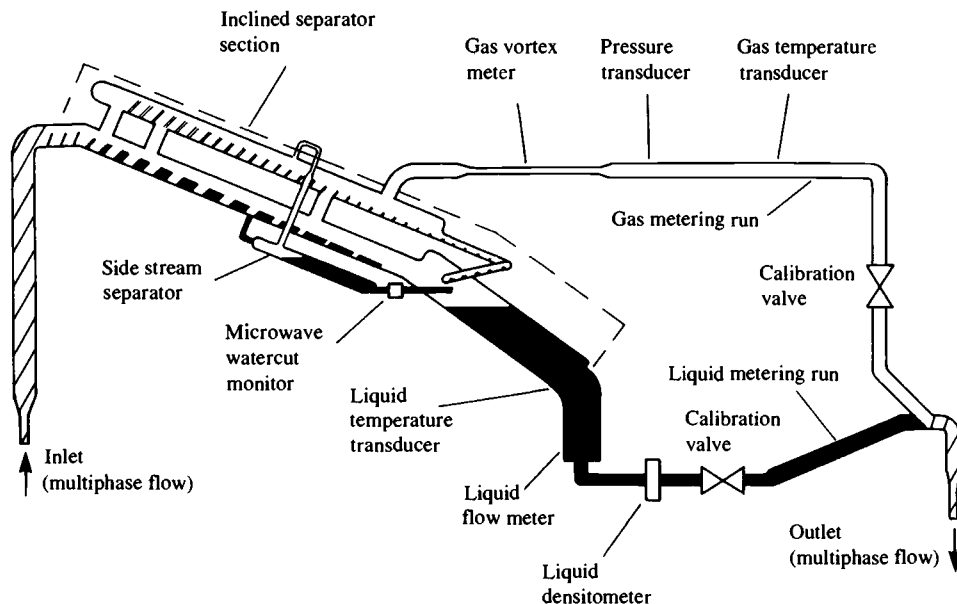


Figure 14.5. Subsea metering system (Dean et al. 1990a; reproduced with permission from the NEL).

a sample of the flow, separated the liquid and gas components in the sample and measured the water content of the liquid component (cf. King 1988; Millington and King 1988, who used a jet mixer/turbine meter package).

S. G. S. Redwood (published reports appear to associate this development with NEL and Imperial College, London, England) used a static mixer, with venturi or pressure loss across the mixer, for total flow rate. Density was obtained with a gamma densitometer at the venturi throat. In addition, a small sample of the mixture was extracted and separated, and a Coriolis meter was used to obtain the water cut (Anon. 1994). King (1988) also mentioned the use of ultrasonic cross correlation (cf. Sidney et al. 1988a, 1988b).

Tuss (1996) described tests of an Agar multiphase meter consisting of a positive displacement meter to measure the total volume of the flowing stream, a venturi tube to measure gas liquid flow rates, and a water cut monitor to measure water-liquid fraction. A gas bypass loop measures the gas volume flow. About 10 % uncertainty of the reading for each component appears to be claimed.

In-Line Measurement of All Parameters

Fluenta, with work from the Christian Michelson Institute (Dykesteen et al. 1985), developed a meter. It obtained the velocity from cross correlation of signals from spaced capacitance sensors and used these sensors with gamma densitometer to obtain the makeup of the fluid. The velocities of all the components were assumed to be equal (cf. Frantzen and Dykesteen 1990). The Fluenta MPFM 900 is a full-bore, noninvasive instrument using capacitance to determine the water cut and the addition of gamma ray absorption to obtain phase fractions (Millington et al. 1993). The flow rates are obtained from cross correlation of the axially spaced capacitance sensors with measures of large and small bubble transits.

Torkildsen and Hanssen (1996) described the Framo meter as consisting of

- static flow mixer for homogenizing the flow;
- multi energy gamma meter for measurement of the oil, water, and gas fractions; and
- venturi meter for measurement of the total flow with differential pressure, pressure, and temperature transmitters.

Torkildsen et al. (1997) commented on the need to be aware of the possibility that flow regime and salinity may affect the meter performance.

The ISA Controls/BP meter (Priddy 1994) (Figure 9.21) uses a twin helical screw positive displacement meter with gamma density meter spanning the flow between the two sets of screws.

The CSIRO, Australia, developed a multiphase meter based on gamma rays to obtain mass, cross correlation of gammas to obtain velocity and dual-energy gammas to obtain mass fraction of oil in liquid. In addition, pressure and temperature measurements were made, and specific gravities, etc., of the components were known (Watt 1993). A measurement uncertainty of between 5 and 9% appears to have been achieved.

The Kongsberg/Shell meter uses capacitance to obtain the gas-liquid interface in the channel, the oil-water fraction, and, by cross correlation, the slug velocity and to estimate the individual flow rates.

The Multi-Fluid International (MFI) meter is suitable for flows with continuous oil component. The composition is obtained with microwave and gamma radiation, and velocity is measured using cross correlation between two dielectric sensing sections. Okland and Berentsen (1994) described tests of the MFI, which suggested that after optimization the MFI could achieve uncertainty on oil flow rate within 5%.

Using a combination of venturi, capacitor, and single-beam gamma densitometer, Fischer (1994) showed that, provided the mixture was homogeneous, the mass flow rates of the components of oil-water-air could be obtained within about $\pm 25\%$.

Another method that has been looked at is pulsed neutron techniques (Anon. 1993). Ekrann et al. (1988) included flow measurement as part of a capacitive system presumably by using a correlation method.

The trend appears to be toward a meter that allows full-bore measurement of all components, although removal of a small sample may allow a higher precision while the technology develops.

Whitaker and Millington (1993) concluded that, from available evidence, the then state of the art should have allowed phase flow rates to be measured to approximately 10% over reasonably wide operating envelopes. Whitaker (1993) also appeared to conclude that 10% or so bulk measurement uncertainty was unlikely to be bettered without new technologies and more recently (Whitaker 1996) confirmed the $\pm 10\%$ of reading for four meters: Agar, Fluenta, Framo, and MFI.

See also North Sea Flow Measurement Workshops 1992, 1993, and 1994 for further papers on multiphase flow meters and mixers.

14.5 CHAPTER CONCLUSIONS

This chapter has been concerned with a wide range of mass flow measurement that falls outside the meters discussed in Chapters 15–17. Apart from the hydraulic Wheatstone bridge method that has had a particular market niche, the other multi-meter methods are likely to be in competition with the increasingly versatile Coriolis meter. The most promising multimeter development is likely to include ultrasonics.

It is in the area of multiphase flows that the major developments are likely to occur, backed by the considerable cost advantages that result from subsea multiphase flow measurement. These are likely to have improved component ratio measurement and clever data handling to obtain the most information. We shall, therefore, concentrate on these, always remembering that other areas of industry beyond the oil industry will benefit from such developments.

14.5.1 WHAT TO MEASURE IF THE FLOW IS MIXED

It would be satisfactory to have a flowmeter that could interpret all the variables that occur in a multicomponent flow. Such a device might identify the position on a flow pattern chart for well-documented fluids. It could also deduce the mass flow of each component. The state of the art has not reached that point yet, and it is likely to be

several years before it does. It is therefore important to consider what variable we are seeking in the measurements that are being made now. The most useful measure is clearly mass flow of the separate components. But the sophistication needed to achieve this for gases and liquids, without errors due to water, sand, and wax, is high, and most methods will need to compromise.

Beg and Toral (1993) reported on the use of pattern recognition techniques to obtain superficial gas-liquid flow rates. They reckoned that the techniques could be extended to oil-water-gas flows by using pressure sensing and an orifice plate with capacitance and gamma ray. They introduced a new hydrodynamic system for scaling pipe diameter and fluid properties and used neural network pattern recognition techniques (cf. Toral et al.'s 1990 computer package).

The use of mixing to create an almost homogeneous flow also has drawbacks. The flow downstream of a mixer is extremely disturbed, and such a position would certainly not be recommended for precise conventional flow measurement. In many cases, the flow also starts to separate as soon as it is mixed. The flowmeter would need to be calibrated in these flow conditions, and such a calibration would need to take account of the effect of varying component ratios. The possibility of using the pressure loss across the mixer was noted earlier.

14.5.2 USABLE PHYSICAL EFFECTS FOR DENSITY MEASUREMENT

Because it is unlikely that direct mass flow measurement will be suitable for multiphase flows, we need to consider the indirect measurements that could be combined with a flow rate measurement. These are

- ❑ weighing elements;
- ❑ vibrating elements;
- ❑ acoustic methods (sound speed, acoustic impedance);
- ❑ electromagnetic field methods (magnetic resonance, electrical capacitance, static charge); and
- ❑ nuclear radiations (X-ray, microwave, gamma).

It becomes apparent that, for two-component flows, some of these are unsuitable because, for instance, bubbles and particles do not follow vibrating movements. Optical (transmission problems) and thermal (heat transfer across boundaries) methods have not been listed because they, also, are unlikely to be suitable (cf. King 1990).

However, Dyksteen et al. (1985) suggested that the measurement of resistance and capacitance could achieve multicomponent ratio measurement. One manufacturer has offered a full-bore device for measuring water in oil from 0 to 80% using capacitance methods for pipe diameters up to 300 mm and with the possibility of a ceramic-lined device for pressures above 16 bar. Microwaves are used by another manufacturer for the measurement of water in oil by absorption. Nucleonic measurement has also been offered commercially as part of a three-phase unit for oil, water, and gas. Frantzen and Dyksteen (1990) report a multiphase fraction meter combining capacitance and gamma radiation, which looks very promising. Gainsford (1990) described meters to obtain water fraction in oil, as well as oil, water, and gas fractions. It appears that microwave or gamma radiation were used for the former and a combination for the latter.

Work in Shell laboratories (van Santen et al. 1995) also suggested that dual-energy absorption techniques with gamma- or X-rays may offer an on-line monitoring method for individual fractions of oil, water, and gas in pipelines but may be subject to errors from change in water salinity and also possibly from changes in distribution in space and time.

14.5.3 SEPARATION OR MULTICOMPONENT METERING

The problems relating to the accurate measurement of multicomponent flow are so great that separation has been the only viable route in most cases. Whether separation of the components continues to be used will depend on the following conditions.

- a. The components are separable. This may be difficult in truly two-phase fluids and in fluids where the finished product contains several components such as liquid foods.
- b. The environment is conducive to separation. Subsea separation may be more difficult than multicomponent flow metering.
- c. The ultimate level of accuracy required can be achieved without separation.
- d. A complete system for multicomponent flow, pumps, valves, etc., is available.

Separation of the components and subsequent measurement of each component may offer the quickest route to an accuracy that approaches fiscal requirements, but in the longer term competitive methods that do not separate may result from elegant applications of more advanced physics and clever computational interpretation of signals.

14.5.4 CALIBRATION

National Engineering Laboratory, Scotland, now has the facilities to test multiphase flowmeters for hydrocarbon flows (cf. King 1990).

It is tempting to suggest, as for single-component calibration rigs, that the test section of such a rig should be preceded by a long settling length to obtain fully developed flow. However, there are good reasons why such an approach is less appropriate for a multicomponent rig.

- i. The settling length may be impracticably long.
- ii. The orientation of the pipe will affect the flow, and a long settling length will prohibit other than horizontal orientations.
- iii. It is virtually certain that actual installation positions will not have fully developed flow conditions.
- iv. The enormous range of flow conditions requires that the flowmeter accuracy is not dependent on particular flow conditions.

A more appropriate design for such a rig is likely to be a test section preceded by a short upstream section and a manifold for component injection. By selection of the injection manifold, flow regimes could be constructed upstream of the flowmeter. The whole unit including manifold, upstream section, and test section would be short enough to rotate to any orientation so that both horizontal and vertical flows could be simulated. The outlet flow through a flexible pipe would enter a separator

before recirculation. Such a rig may also have a small enough total volume to allow live crude to be used as one component for oil-related flows.

Other combinations of flow components for other industries may lend themselves to other rigs using more conventional calibration methods such as volume and mass measurements of the total flow passed.

See also Mazzoni et al. (1994) for a description of a multiphase test facility.

14.5.5 ACCURACY

The errors in the papers reviewed on multicomponent/multiphase flows range from $\pm 1\%$ for an ultrasonic correlation flowmeter in paper pulp flow, for example, to as much as 25% for one of the offshore oil flow measurement systems. This range of error is symptomatic of the problem for the flowmeter designer. For control and distribution, a measurement uncertainty of order $\pm 10\%$ may be acceptable. However, high accuracy is required for fiscal and custody transfer applications, and here the level that can be achieved by the designer is far from clear. Is even $\pm 5\%$ achievable in multicomponent oil flow measurement?

Thermal Flowmeters

15.1 INTRODUCTION

There are broadly two concepts of thermal flowmeter now available for gas mass flow measurement, one of which is also applicable to liquids. I shall follow the useful terminology in ISO committee draft (ISO/CD 14511:1998) for the two types. The first is the capillary thermal mass flowmeter (CTMF), which has broad applications in the control of low flows of clean gases, but which can also be used with a bypass containing a laminar element to allow higher flow rates to be measured. The arrangement of heaters and coils between the various manufacturers differs, but the basic approach is the same, with heat added to the flowing stream and a temperature imbalance being used to obtain the flow rate.

The second is the full-bore thermal mass flowmeter (ITMF), which is available as both insertion probe and in-line type. It has a widely used counterpart in the hot-wire anemometer for measurement of local flow velocity, but, as the need for a gas mass flowmeter has become evident, it has been developed as a robust insertion probe for industrial usage and then as the sensing element in a spool piece flowmeter. It has been produced by an increasing number of manufacturers in recent years as a solution to the need for such a mass flowmeter.

15.2 CAPILLARY THERMAL MASS FLOWMETER – GASES

In various examples of the CTMF, the gas flows through a very small diameter tube that has heating and temperature-measuring sensors. For larger flows, this is used with a bypass laminar flow element in the main gas stream. In many cases, the mass flowmeter is in one unit with the control electronics and the gas control valve. However, in this chapter our interest is in the meter itself.

15.2.1 DESCRIPTION OF OPERATION

In this design of thermal flowmeter, the gas flows through a very small diameter tube, with a sufficient length-to-diameter ratio to ensure laminar flow, and on the outside of which there are heating and temperature-sensing windings. The heater winding transfers heat through the wall of the tube to the gas. If there is a gas flow, the heated gas is carried downstream. The downstream temperature sensor will then sense a higher temperature than the upstream sensor. This differential can then be used to deduce the mass flow. A diagram is shown in Figure 15.1(a). The equation

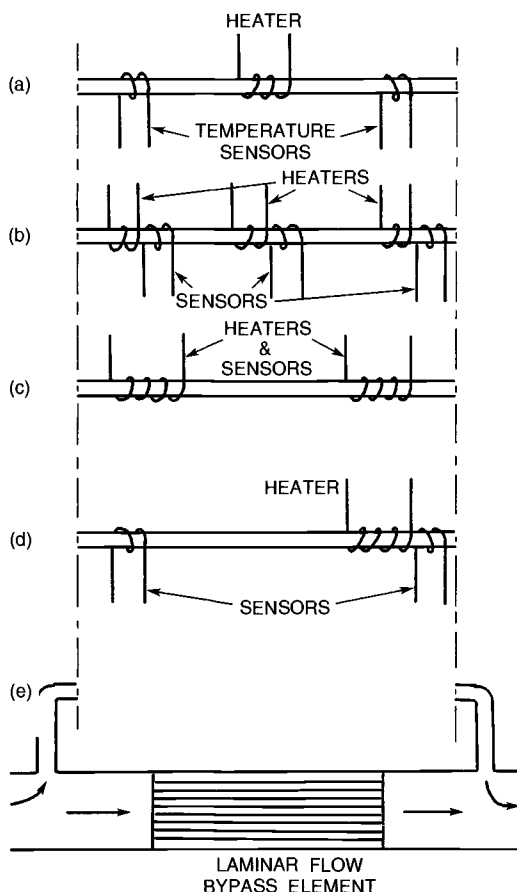


Figure 15.1. Diagram to show the various configurations of CTMFs: **(a)** Central heater with upstream and downstream temperature sensors; **(b)** Three heaters with sensors to ensure that they remain at constant temperature; **(c)** Two combined heaters and resistance thermometers; **(d)** One downstream heater with two sensors; **(e)** Bypass with laminar flow element.

bypass is then proportional to the flow through the capillary. In some designs, a control valve is incorporated. In others, an air-actuated valve may also be offered by the manufacturer.

Figure 15.2 shows an industrial CTMF with bypass through n identical elements, where n can be selected for the appropriate flow range. It also shows the valve, operated from a second amplifier, which controls the flow.

In another design [Figure 15.3(a)], the means of heating the tube is through a current passed along the tube wall, which, in turn, heats the fluid. Two heat sinks, one at each end, conduct away the heat that has passed up the tube. The fluid cools the tube, and the resultant temperature difference is measured in two places with the thermocouples TC_1 and TC_2 ; the flow rate can be deduced from the difference. As with the other designs of thermal flowmeter, this one can be used with a bypass. The distributions of temperature for zero flow and for a small flow are shown in Figure 15.3(b). It can be seen that the temperature difference between the two

for this is given in the steady state by

$$q_m = Q_h / (K c_p \Delta T) \quad (15.1)$$

where q_m is the mass flow rate to be measured in kilograms per second, Q_h is the heat input in Joules per second, K is a constant, c_p is the specific heat at constant pressure in Joules per kilogram Kelvin, and ΔT is the measured temperature difference in Kelvins. A knowledge of K , Q_h , and c_p allows q_m to be deduced from ΔT .

In an alternative mode of operation [Figure 15.1(b)], the sensed temperature is used to adjust the heat into three heaters to keep them at a constant temperature. The energy required in this mode to restore equilibrium to the windings can be used to obtain the instantaneous rate of flow. This results in a sensor with a short response time.

Figure 15.1(c) shows a design that uses the temperature difference between two coils positioned along the tube axis, both of which are heated. The coils also act as resistance thermometers. When flow occurs, the downstream coil is cooled less than the upstream coil, and the differential temperature is used as a measure of the flow.

Figure 15.1(d) shows a design with two temperature-sensing coils and one heater coil.

In order to accommodate a greater flow rate than is possible through the capillary, a laminar flow bypass is used [Figure 15.1(e)]. The total flow rate through capillary and

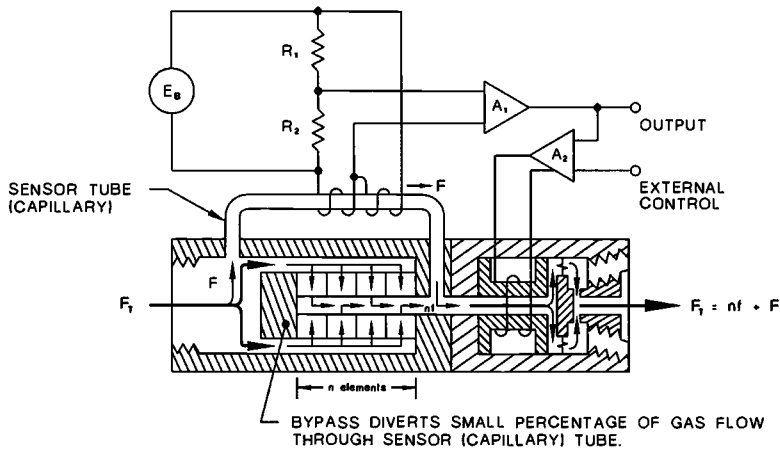
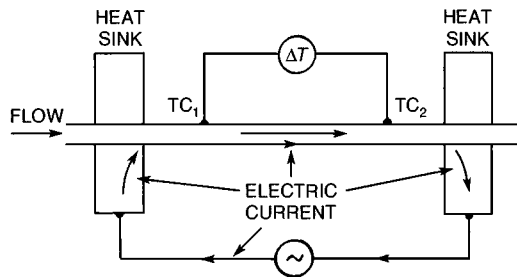


Figure 15.2. Industrial capillary thermal mass flowmeter showing schematic of flow paths and electrical diagram (reproduced with permission of BOC Edwards).

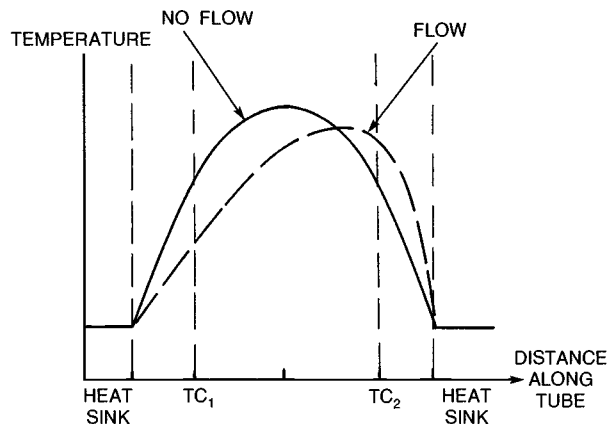
thermocouples will be zero at zero flow but will increase with flow. The equation derived for this flowmeter is reproduced by Hemp (1995a) (cf. Appendix 15.A.4).

All these varieties are dependent on a knowledge of c_p , which depends on the gas and varies with pressure as shown in Table 15.1.



(a)

Figure 15.3. Heated tube thermal flowmeter (after Hastings): (a) Schematic diagram; (b) Temperature distribution under static and flowing conditions.



(b)

Table 15.1. Variation of c_p with pressure for four gases

Gas	Approximate % Variation	
	0–10 bar	40–100 bar
H ₂	+0.1	+1.6
Air	+2	+16
O ₂	+1	+18
CH ₄	+2	+31

It is important for the calibration of the meter to be close to, or referenced to, operating conditions; otherwise, it will be necessary to look up the data for the gas and pressure at operation. Temperature variation has only a slight effect on the value of c_p . For example, for air it is 0.01%/°C, and for methane it is 0.11%/°C. Viscosity variation has a negligible effect unless the gas is near liquefaction. Calibration may be referenced to primary volumetric standards at

atmospheric conditions and corrected normal temperature and pressure or standard temperature and pressure.

15.2.2 OPERATING RANGES AND MATERIALS FOR INDUSTRIAL DESIGNS

Rangeability is up to 50:1. Flow ranges are from 0 to 3 ml/min or lower to 0–100 l/min or even up to 1,000 m³/h for special bypass designs. A 100-mm diameter meter would typically give a range up to about 250 m³/h and a 200-mm diameter meter up to about 1,000 m³/h. Note that, for air at ambient conditions, 1,000 m³ has a mass of about 1.2 kg.

These meters may be able to operate for some designs up to about 200 bar or even, in some cases, 300 bar pressure. Temperature operating range may be 0–65°C with a coefficient of less than 0.1%/°C. Pressure drop on the line is up to 0.02 bar for the maximum flow rate.

Materials in contact with the gas include 316, 316L, and high alloy ferritic stainless steel; anodized aluminum, brass, and Viton; Buna-N; Kalrez; and Teflon or silicone O-rings.

Power supply requirements usually relate to 15 and 24 V DC versions; when combined with a controller, these may draw up to about 180 mA resulting in power consumption of order 3.5 W.

Flowmeters are likely to be factory-set to a preselected gas, pressure, and flow range. Correction factors range from about 0.55 for ethane to about 1.40 for argon against air. These may also be referred to nitrogen.

15.2.3 ACCURACY

Uncertainty is typically claimed as $\pm 1\%$ full-scale deflection or FSD (or for a recent design $\pm 0.75\%$ rate $\pm 0.25\%$ FSD) but may be $\pm 1.5\%$ for the top end of the flow range and $\pm 4\%$ FSD in some other cases, depending on design and the temperature and pressure range. Repeatability is typically 0.2–0.5% FSD, but in some cases it may be quoted as being of order $\pm 0.25\%$ of rate and in at least one case as 0.05% FSD. In addition, there may be pressure and temperature coefficients.

15.2.4 RESPONSE TIME

An exponential curve is usually used to indicate the response of a flowmeter. The exponential curve is given by

$$S = S_0(1 - e^{-t/\tau})$$

The time constant τ gives the time for the flow signal S to reach about 60% of its final value S_0 . Table 15.2 gives the times to approach the final value.

τ is of order 250 ms to over 1 s. A standard instrument may take 1–5 s to settle to within 2% of final value for a 0–100% command change. However, manufacturers may claim a response of about 1 s for some designs.

Thermal drift may be reduced against ambient temperature changes by insulating the unit, but with the consequence of a longer time constant (Sullivan no date). A solution is to enclose the sensor in a temperature-controlled environment. Sensors may also suffer from radiofrequency interference and may be incompatible with corrosive gases. Typically, a 30-min warm-up time may be needed.

Table 15.2. Time to approach the final value after a flow change

To Approach Within	Response Time
5%	3τ
2%	4τ
1%	4.6τ
0.5%	5.3τ

15.2.5 INSTALLATION

There is unlikely to be any effect from upstream pipework for the basic device without bypass because the bore of the tube is so small that the profile will be readjusted before reaching the elements. For bypass designs, there may be an effect, and it may be necessary to include 20D upstream straight pipe (Gray et al. 1991). Designs should ensure that flow is truly laminar in the bypass, by suitable matching of size and pressure drop.

There may be significant effects on sensitivity that can seriously impair accuracy under certain conditions of installation (e.g., mounting angle), gas density, and pressure when outside normal operating envelopes. Zero offset appears to be affected by meter orientation and may result from free convection when the meter is near vertical. Rarefied gas flows may also create problems.

Temperature effects on meter zero are up to 0.15% FSD/°C and on span are less than $\pm 1\%$ FSD over a 10–50°C range. Pressure sensitivity is of order $\pm 0.4\%$ per bar up to about 14 bar. Power supply sensitivity is of order $\pm 0.1\%$ FSD per percent power supply voltage variation.

Typical weight is of order 0.5–2.5 kg. For bypass designs, the weight may be up to 20 kg. Meters should have been subjected to burn-in and cycling prior to shipment.

Gray et al. (1991) discussed environmental factors that affect mass flow controllers (MFC). The MFC is tested with a vacuum diagnostic system to which the MFC is connected. The test chamber is evacuated to below 50 mtorr, and the valve to the vacuum pump is then shut. The initial flow rate is set, and the pressure is measured as it rises. From this, the time to achieve controlled flow, actual flow rate, error, and deviations from flow are obtained. They tested various effects and concluded that, to minimize effects, temperature extremes must be avoided and surroundings should not be allowed to raise the temperature of the gas boxes, pressure should be carefully regulated, and vibration should be avoided. Where possible, tubing distances should be minimized, and MFCs should be mounted in a normal upright orientation.

15.2.6 APPLICATIONS

CTMFs are most commonly applied to low flows of clean dry gases above their dew-points (e.g., gas blending, semiconductor industry). Gases quoted by manufacturers include air, acetylene, ammonia, argon, arsine, nitrogen, butane, carbon dioxide, carbon monoxide, chlorine, ethane, ethylene, fluorine, Freon 11, Freon 12, Freon 13, helium, hydrogen, hydrogen sulfide, krypton, methane, neon, nitrous oxide, oxygen, propane, propylene, silane, and xenon. Examples of problem gases that have been used with this type of meter include

- BCl_3 (boron trichloride) corrosive gas;
- SiCl_4 (silicon tetrachloride) corrosive and low vapor pressure gas;
- CCl_4 (carbon tetrachloride) low vapor pressure gas;
- SiH_4 (silane) typical of gases that react and create deposits, resulting in clogging of flow controllers; and
- WF_6 (tungsten hexafluoride) similar to the handling problems of silane.

The meter may be used with a wide range of gases for processes involving crystal growth, thermal oxide, diffusion, chemical vapor deposition, sputtering, ion implantation, and plasma etching, and the gases involved will each require a correction factor against the calibration gas, possibly nitrogen, which may vary between as low as 0.17 for Freon-C318 (C_4F_8) or perfluoropropane, and as high as 1.543 for krypton. The manufacturer should be asked for the compatibility of any particular gas and the correction factor (Sullivan no date). In selecting a flowmeter, particular care should be taken where the inlet pressure is less than atmospheric or has a vapor pressure less than atmospheric.

15.3 CALIBRATION OF VERY LOW FLOW RATES

One method of calibration of these flowmeters for low flow rates is by means of a special piston prover. A piston moves within a precision glass tube. The piston is connected to a counterweight by means of a tape that turns an encoder wheel. The temperature and pressure of the gas are measured while flow is taking place.

Positive displacement devices or piston meters may achieve calibration to an accuracy of about 0.2% of reading over a range from 0.02 to 30 l/min (Sullivan et al. no date). The volume flow at standard temperature and pressure (STP), subscript s, may be obtained from the actual, subscript a, using

$$q_{vs} = q_{va} \frac{T_s}{T_a} \frac{p_a}{p_s}$$

15.4 THERMAL MASS FLOWMETER – LIQUIDS

15.4.1 OPERATION

The diagram in Figure 15.4 shows the layout of a meter with parallel tubes running between a heating block and a heat-sink block. At the midpoint of each tube is the

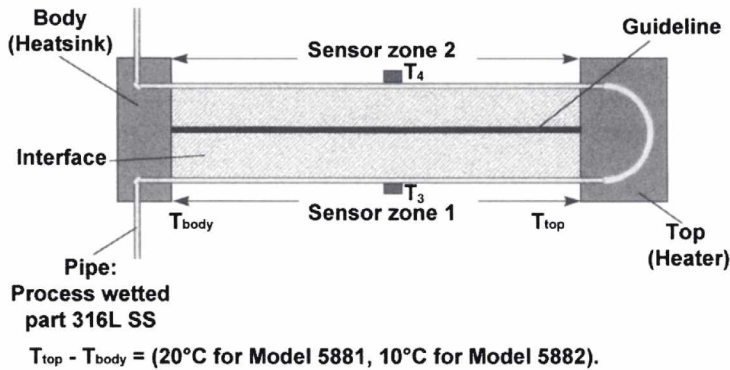


Figure 15.4. Liquid mass flowmeter (reproduced with permission from Fisher Rosemount Ltd.).

temperature sensor. The heating causes an increase in temperature of 10 or 20°C. When the flow takes place, heat is convected, and the difference between T_3 and T_4 will vary with the mass flow rate. It appears that the heat is then conducted through the body of the meter. The liquid may increase in temperature by as much as 15°C within the meter, but there is negligible change in liquid temperature outside the device.

Huijsing et al. (1988) described a new meter aimed at fuel consumption measurement in automobiles and light aircraft and also industrial processes. The external mufflers (for heat transfer and temperature measurement) of other designs were replaced with internal copper blocks, perforated with multiple axial holes through which the liquid flowed, resulting in much greater thermal contact with the liquid.

15.4.2 TYPICAL OPERATING RANGES AND MATERIALS FOR INDUSTRIAL DESIGNS

For liquids, minimum measurable flow is about 0.002 kg/h with a maximum measurable flow of between 0.1 and 0.5 kg/h, depending on the design, and a turndown of up to about 15 : 1. Accuracy is about 0.5–1% FSD and repeatability 0.2% of rate for the upper end of the range and 0.1% FSD for the lower part of the range. Repeatability is of the order of $\pm 0.5\%$ FSD per month.

A bypass arrangement allows greatly increased flow range up to 100 kg/h, but the measurement uncertainty is likely to increase to about 2% FSD. However, if extreme range turndown ratios are claimed, it is likely that the manufacturer will give a performance envelope for the meter.

The ambient temperature range may be from 0 to 65°C, and pressure ratings can be as high as 400 bar. Maximum viscosity may be about 100 cP. Pressure loss at operating conditions is claimed as very small.

The response time for sensors is likely to range from 0.5 s up to about 10 s for the largest meters, but the response for controllers should be obtained from the manufacturer.

15.4.3 INSTALLATION

Pipe size connections range from $\frac{1}{16}$ to $\frac{1}{4}$ in. (about 1.5–6 mm) with materials of stainless steel and seals of Buna-N, Viton, PTFE, or Kalrez. Process temperature change can cause $+0.02\%/^{\circ}\text{C}$, and ambient temperature effects can cause up to $+0.03\%/^{\circ}\text{C}$.

15.4.4 APPLICATIONS

It may be suitable for measurement of very low flow rates of toxic, corrosive, and volatile liquids (liquefied gases under pressure), catalysts in petrochemicals, pigments in paints, reagents in pharmaceuticals, flavorants, enzymes, antibiotics in food, odorants into natural gas, titanium chloride closing into furnaces for hardening metals, corrosion inhibitors in pipes, and reagents in fermenters. It also may offer a means for controlling and measuring very low flows in laboratories concerned with microfiltration, measurement of porosity of rocks, fuel consumption, lubrication, gas/vapor mixtures, and chromatograph flows, as well as controlling the injection of additives (catalyzers) into industrial reactors.

15.5 INSERTION AND IN-LINE THERMAL MASS FLOWMETERS

The in-line thermal mass flowmeters (ITMFs) represent an alternative design concept to the CTMF, and both insertion and in-line versions appear to be aimed at the same goal – mass flow measurement of gases in pipes and ducts.

In Figure 15.5 there is a diagram of an arrangement for an ITMF. The heaters and temperature sensors for either insertion or in-line designs may be contained in one or two probes, may use the same element for both functions, and, where two probes are used, will often, as in Figure 15.6, be in the same cross-section of the duct.

This flowmeter (Kurz 1992) uses a heated element in a flowing stream, the convective heat transfer of which is dependent on the temperature difference between the probe and the fluid, on the flow rate, and on characteristics of the fluid. A second temperature probe is used to measure the temperature of the fluid. Two types have been used: the constant power and the constant temperature anemometers. The temperature of the heated probe in the former varies with fluid velocity and is used as the sensing parameter. The response is slow due to the thermal inertia of the probe, and the zero is unstable because natural convection becomes important at low flows. In the latter type, the probe is kept at constant temperature through a feedback control circuit, and the power needed to retain this constant temperature is used to obtain the velocity. As a result, the response is much faster, typically 1 s for a 67% change. The resistance of the sensing elements is connected into a modified Wheatstone bridge (Figure 15.6, cf. Figure 15.A.1), in which the voltage across the bridge is amplified and fed to the top of the

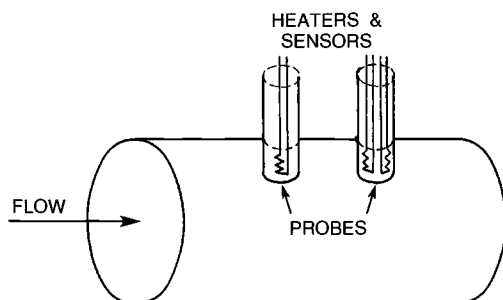


Figure 15.5. Diagram of an ITMF with either insertion probes or in-line design. In the latter, the probes are often in the same cross-section.

the probe, and the zero is unstable because natural convection becomes important at low flows. In the latter type, the probe is kept at constant temperature through a feedback control circuit, and the power needed to retain this constant temperature is used to obtain the velocity. As a result, the response is much faster, typically 1 s for a 67% change. The resistance of the sensing elements is connected into a modified Wheatstone bridge (Figure 15.6, cf. Figure 15.A.1), in which the voltage across the bridge is amplified and fed to the top of the

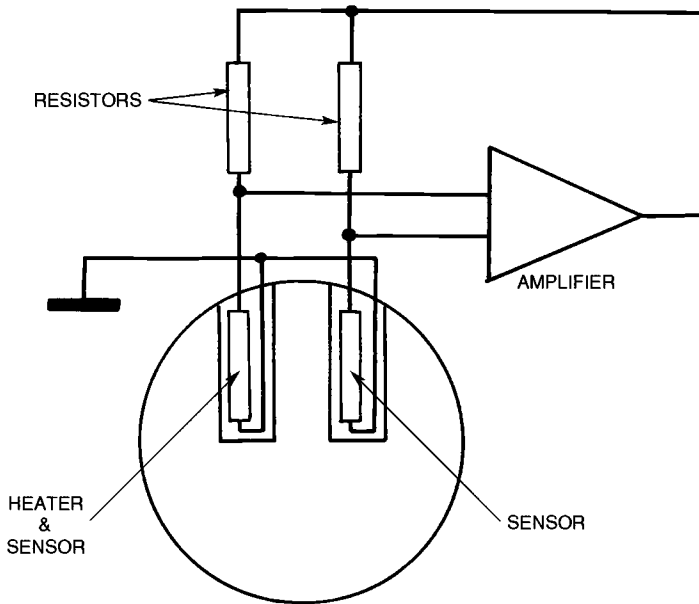


Figure 15.6. Schematic of in-line thermal mass flowmeter showing Wheatstone bridge arrangement.

bridge to retain a constant temperature difference. By using a three-wire sensor, the lead resistance can also be compensated for.

The relationship for these flowmeters, from Equation (15.A.8), is

$$q_h = k' \left(1 + K q_m^n \right) \Delta T \quad (15.2)$$

where q_m is the mass flow rate, q_h is the heat supplied, and ΔT is the temperature difference between heated and unheated probes. k' is a constant that allows for heat transfer and temperature difference at zero flow, and K is a constant incorporating the area of the duct into which the probe is inserted, as well as gas and heat transfer constants. In Equation (15.A.8), n was equal to $\frac{1}{2}$. This may be approximately correct for low velocities but may be more like $\frac{1}{3}$ for higher velocities. Factors affecting the equation, as well as free convection at low velocities, are some heat transfer from the probe to its base, usually small, and thermal radiation, which is small compared to forced convection for normal temperatures.

15.5.1 INSERTION THERMAL MASS FLOWMETER

An example of an ITMF is shown in Figure 15.7(a) and consists of two pairs of elements. In each pair, there is a resistance thermometer; coupled to these are, in one pair, a heated rod and, in the other, a similar unheated body as a mass equalizer. Flow past the heated element results in a heat loss, and the combination of this and the differential temperature allows the flow rate to be deduced.

Insertion probes may be inserted into and withdrawn, without process shut-down, from a pipeline of 30- to 500-mm diameter. They may also be available with multiple sensors for installation across a pipe [Figure 15.7(b)].

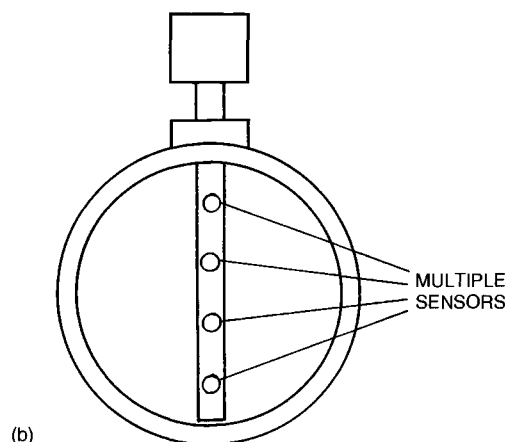
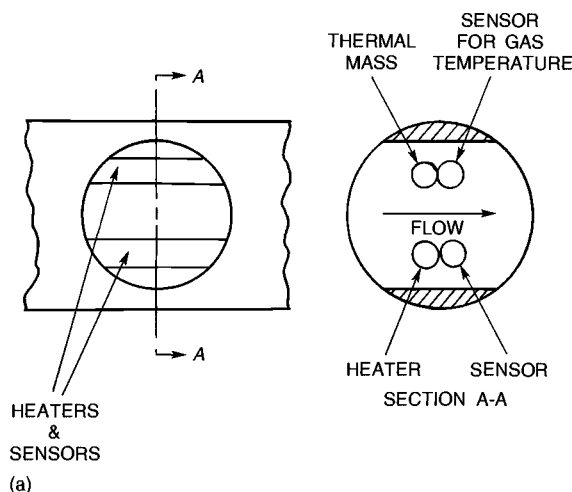


Figure 15.7. Diagrams of single- or multisensor probe: **(a)** Showing individual sensor; **(b)** Showing multisensor probe in a pipe (after the design of FCI).

The uncertainty may be quoted, typically, as being of order $\pm 1\%$ FSD or $\pm 3\%$ rate, whichever is less, for turndown ratios of order 10 : 1. However, gas velocities up to about 60 m/s or more have been claimed with turndown up to 800 : 1. Manufacturers who claim wider turndowns will presumably provide a calibration that relates uncertainty to various points in the characteristic.

Various designs of these meters have been used for boiler applications (preheater and combustion air), combustion gas flows, hydrogen coolant flow, gas pipeline transmission flows, methane flows, nitrogen purges, flare gas flow, ventilation, heating and air conditioning, and superheated and dry saturated steam.

Lai et al. (1991) discussed the use of thermal mass air-flow sensors (essentially a hot-wire anemometer) for measuring the air flow into gasoline engines. A recommended arrangement is to have the mass air-flow sensors in a bypass of the intake air passage of an engine. Possible errors may arise due to variation in the profile at inlet caused by filter blockage.

15.5.2 IN-LINE THERMAL MASS FLOWMETER

Figure 15.8 shows a commercial ITMF with heated and unheated temperature-dependent resistors. Another approach that may be available uses two stainless steel wires in the flow stream in the throat of the transducer – one heated with a current, and a bridge circuit to adjust the current to keep the temperature between them, measured by their resistances, constant. There may also be designs that heat the fluid and sense its temperature from outside the tube, leaving an unobstructed bore to the tube.

15.5.3 RANGE AND ACCURACY

Meters of 25–200 mm may be available as flanged or, in some sizes, as wafer designs. Turndown is of order 50:1 or higher with uncertainty of 2% of rate and within a band of order 4% down to 1% FSD. Uncertainty may be quoted in the range $\pm 1.5\%$ to $\pm 2.5\%$ FSD. Repeatability may be given as 0.25–3% of rate or as 0.25–1% FSD, the lesser value being taken. The manufacturers give correction factors for various gases.

Flow ranges are from 2–125 kg/h to 300–8,000 kg/h or greater. Response time is about 0.2 s. Pressure drop is up to 10 mbar. Operating pressure is up to 40 bar. The temperature range is -50 to 300°C or greater, depending on the design. Hygienic versions are available. There may also be temperature effects of order $\pm 0.3\%$ rate or $\pm 0.08\%$ FSD/ $^{\circ}\text{C}$.

15.5.4 MATERIALS

Materials of construction are typically stainless steel with seals of Viton rubber and encapsulated sensor elements, consisting of platinum resistance thermometers or thermistors, in stainless steel or glass.

15.5.5 INSTALLATION

Installation orientation may affect the performance, and manufacturers' instructions should be followed. Installation requirements recommended by one manufacturer are given in Table 15.3. For a flow conditioner of the perforated plate type based on the Mitsubishi design, this manufacturer recommended 8D upstream spacing with wafer and insertion-type meters, and with larger in-line meters, but allowed positioning of the flow conditioner at the inlet flange of the in-line design for

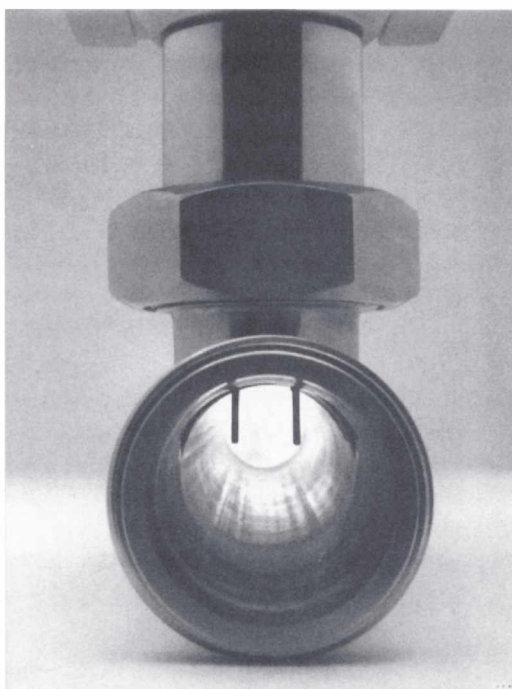


Figure 15.8. Photograph of an in-line thermal mass flowmeter (reproduced with permission from Endress and Hauser).

Table 15.3. Installation requirements for a wafer or insertion thermal mass flowmeter compared with those for a $\beta = 0.55$ orifice plate (after Endress and Hauser)

Upstream Fitting	Thermal Flowmeter		Orifice Plate ($\beta = 0.55$)	
	Upstream ^a	Downstream ^b	Upstream	Downstream
90° bend or T-piece	20D	5D	16D	6D
2 × 90° bends same plane	25D	5D	22D	6D
2 × 90° bends perpendicular planes	40D	5D	44D	6D
Reducer	20D	5D	8D	6D
Expander	20D	5D	20D	6D
Control valve	50D	5D	24D ^c	6D

Note: Upstream distances should be doubled for light gases such as helium and hydrogen.

^a Reduce by about 5D for flanged in-line flowmeter.

^b Reduce by about 3D for flanged in-line flowmeter.

^c Globe valve fully open.

25- to 100-mm diameters. The manufacturer reckoned that, for a control valve downstream, there should be 10D separation (8D to flange of in-line flowmeter). Gaskets, flanges, and pipes should be correctly sized and aligned. The table compares the ITMF with a $\beta = 0.55$ orifice plate and shows that the requirements are about the same apart from the reducer and the valve (not recommended upstream), which are more stringent for the ITMF. For the in-line version, the agreement is even closer.

The influence of temperature variation can be as small as 0.05%/K rate and of pressure 0.2%/bar rate.

15.5.6 APPLICATIONS

This may be an alternative meter for installations where orifice or vortex meters are commonly used for gas mass metering. Examples are chemical engineering and processing technology, semiconductor process gas measurement and control, combustion air measurement for large utility plant, mass flow rate for stack effluent, air sampling in nuclear plant and environmental facilities (Kurz 1992), flare gas systems, leak detection, food industry (e.g., CO₂ metering in breweries), internal works gas consumption measurement, dosing plants, nitrogen and oxygen flows in steel works, burner control, combustion and preheated air, flue gas, methane, moist and corrosive gas from sludge digester, compressed air in hospitals and other medical applications, argon and helium flows in laboratory environments, waste air measurement, and large ducts. These meters are unlikely to be suitable for steam, but the manufacturer should be consulted.

Haga et al. (1995) used a thermal flowmeter for fuel vapor flow measurement and used electrical heaters to ensure that the piping was of a temperature to prevent liquefaction of the vapor. A liquid recovery container and a filter were provided at the inlet of the flowmeter. To prevent the remaining vapor in the piping from being liquefied, air purge was used. They gave expressions for specific heat and density as a function of pressure, temperature, and vapor pressure. Haga et al. claimed that it had been possible to measure the vapor flow in real time for a moving vehicle and that it offered a practical meter.

15.6 CHAPTER CONCLUSIONS

This technology, which is of more modest price than the Coriolis, is likely to be exploited because it also (with ultrasonics) offers a means of mass flow measurement and possibly greater versatility for gases. Development could take place in various directions:

- ❑ improved theoretical understanding and design of the CTMF to minimize pressure, temperature, and installation effects and to improve accuracy;
- ❑ further analysis of the ITMF designs to understand accuracy limitations, environmental, and flow profile effects (it should be remembered that the accuracy of this instrument depends in part on the measurement of a small temperature difference within a gas of variable temperature);
- ❑ improved theoretical predictions of a meter which is insensitive to flow profile (Hemp 1994a/5a);
- ❑ use of multiple probes;
- ❑ use of nonintrusive heating and temperature measurement; and
- ❑ self-checking with transit time to downstream probes.

Casperson (1993) described a device that used a thermocouple in a pulsed mode. Current pulses heat and cool the junctions, and the subsequent decay is monitored. With flow, the time constant will change. Being nonlinear, the device needed calibration.

An important development is microthermal sensors, which allow fabrication of air-flow sensors as well as signal-conditioning electronics on a single chip (van Dijk and Huijsing 1995) with a standardized digital output signal for a microprocessor. The one described is direction-sensitive.

Nguyen and Kiehnscherf (1995) described a flow sensor in which the channel with an area of 0.6 mm^2 and length 10 mm was etched, and the sensor contained polysilicon heaters and measurement resistances on the chip. The chip is bonded to Pyrex glass to complete the channel. Four modes of operation were built into the electronics to allow constant temperature and constant power operation. In this way, they tested with both liquid (0–500 ml/min or 0–10 ml/min) and gas (0–500 ml/min) (Figure 21.2).

With such standardized components, the possibility of bypass metering with multiple and replaceable bypass meters is an interesting development.

APPENDIX 15.A**Mathematical Background to the Thermal Mass Flowmeters****15.A.1 DIMENSIONAL ANALYSIS APPLIED TO HEAT TRANSFER**

The heat flux density (Kay and Nedderman 1974) is given by

$$q_h = f(\Delta T, k, c_p, \rho, \mu, D, V) \quad (15.A.1)$$

where ΔT is the temperature difference between the heat source and the fluid, k is the thermal conductivity of the fluid, c_p is the specific heat at constant pressure, ρ is the fluid density, μ is the dynamic viscosity of the fluid, D is the pipe diameter, and V is the fluid velocity.

One simple form of Equation (15.A.1) for CTMFs is

$$Q_h = K \Delta T c_p q_m \quad (15.A.2)$$

where Q_h is the total heat input into the winding, and a knowledge of K , ΔT , and c_p allow q_m to be deduced.

We can identify dimensionless groups, which must have a functional relationship with each other. For this purpose, a number of heat transfer groups have been suggested

$$\text{Nusselt number (Nu)} = \frac{q_h d}{k \Delta T} \quad (15.A.3)$$

is the ratio of heat transfer to the rate at which heat would be conducted through the fluid under a temperature gradient $\Delta T/d$;

$$\text{Prandtl number (Pr)} = \frac{\mu c_p}{k} \quad (15.A.4)$$

is the ratio of kinematic viscosity to thermal diffusivity;

$$\text{Reynolds number (Re)} = \frac{\rho V d}{\mu} \quad (15.A.5)$$

is the ratio of inertia forces to viscous forces in the flow.

These give a general relationship for heat transfer:

$$\text{Nu} = f(\text{Pr}, \text{Re}) \quad (15.A.6)$$

A special case of this equation is that due to King for the hot-wire anemometer:

$$\text{Nu} = A + B \text{Pr}^{0.5} \text{Re}^{0.5} \quad (15.A.7)$$

where $A = 1/\pi$ and $B = (2/\pi)^{0.5}$. The hot-wire or hot-film anemometer provides a tool for the measurement of local fluid velocity.

15.A.2 BASIC THEORY OF ITMFS

King (1914) investigated the relationship between heat transfer rate and flow velocity. Using Equation (15.A.7), the following expression gives heat loss per unit length of a cylinder of diameter d at a temperature of ΔT above the fluid temperature in which it is immersed:

$$q_h = k\Delta T + (2\pi k c_p \rho d V)^{0.5} \Delta T \quad (15.A.8)$$

where k is the thermal conductivity of the fluid, c_p is the specific heat of the fluid at constant pressure, ρ is the density, and V is the velocity assumed to be perpendicular to the cylinder (cf. Ower and Pankhurst 1966, who used c_v in place of c_p and also provided details and early references).

As the velocity increases, the second term becomes more important, and the first term, which is due to conduction and natural convection, becomes more important at very low flow rates. It is clear from this equation that it is necessary to measure q_h and ΔT and hence to obtain V assuming that the constants of the wire and the fluid are known. However, the relationship between q_h and ΔT is more usually obtained by calibration.

This, in turn, suggests that any device designed to measure velocity from this formula will need a means of measuring the temperature of the stream as well as the heat loss from a heated wire. This may be achieved by calibrating the probe for the proposed application.

The other important observation is that, when the density is not constant, the combination of velocity and density with the area of the meter will give mass flow rate.

King showed that in air the formula held down to a value of

$$Vd = 0.0187 \quad (15.A.9)$$

where V is in centimeters per second, and d is in centimeters. At very low flow rates, the natural convection in the region of the wire becomes significant in comparison with the forced flow. Collis and Williams (1959) gave a Reynolds number, below which convection becomes important, as

$$\text{Re} = \left(\frac{g d^3 \Delta T}{T \nu^2} \right)^{1/3} \quad (15.A.10)$$

where ν is the kinematic viscosity, T is the ambient temperature of the air, and ΔT is the difference between the temperature of the wire and the ambient air.

Simplifying Equation (15.A.8) and introducing the heating current I through a resistance R , we can write

$$q_h = I^2 R = C + DV^{0.5} \quad (15.A.11)$$

where C and D are functions of temperature. But if the resistance of the wire or cylinder is kept constant, then the temperature of the wire or cylinder will also be constant and so C and D will be constants of the meter for a particular fluid.

To maintain the constancy of the resistance, a bridge circuit such as that shown in Figure 15.A.1 is used where the null indicator ensures that the resistances are all

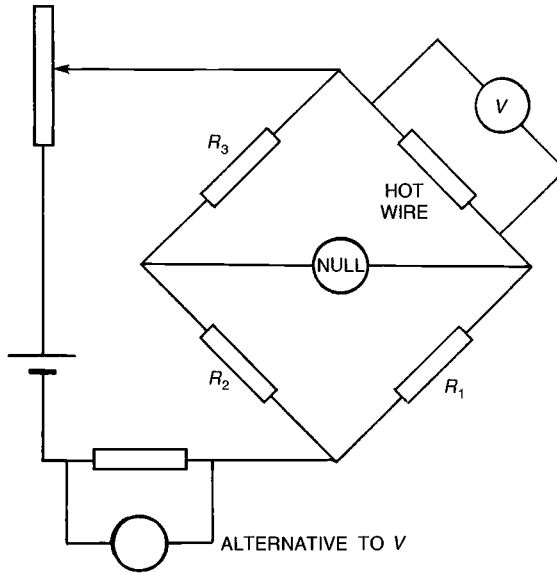


Figure 15.A.1. Constant temperature resistance circuit (after Ower and Pankhurst 1966).

equal, and the current, and hence the power, is measured by means of the resistor in series with the bridge. The advantages of using a constant resistance compared with a constant current are that it is possible to maintain a constant temperature and avoid thermal inertia, and that the equation is simplified as in Equation (15.A.11).

Using Equation (15.A.8) and allowing for temperature variation, we can rewrite it as

$$q_h = k(1 + Kq_m^{0.5})\Delta T \quad (15.A.12)$$

where K is a constant incorporating the area of the duct into which the probe is inserted and the gas and heat transfer constants, q_m is mass flow rate, q_h is heat supplied, and ΔT is temperature difference between heated and unheated probes.

This provides the basic relationship for ITMFs with either q_h or ΔT constant.

15.A.3 GENERAL VECTOR EQUATION

The governing equation for flow with heat transfer under steady conditions is

$$\mathbf{V} \cdot \nabla T = \frac{k}{\rho c_p} \nabla^2 T + \frac{q_a}{\rho c_p} \quad (15.A.13)$$

where q_a is the added heat flux. This can be obtained from Hemp (1994a, Equations 2 and 5)

$$\mathbf{q}_h = -k\nabla T \quad (15.A.14)$$

and

$$\nabla \cdot \mathbf{q}_h = -\rho c_p \mathbf{V} \cdot \nabla T \quad (15.A.15)$$

with the addition of the final term, which relates to the local heat addition. If all heat is added across boundaries with the flow, then the last term disappears. Hemp uses Equations (15.A.14) and (15.A.15) to develop his weight function theory for thermal diffusion flowmeters. This is an elegant theory, and the interested reader is encouraged to refer to Hemp's papers. However, as he comments, the work at present is probably mainly of theoretical interest.

We can write Equation (15.A.13) for flow in a two-dimensional channel as

$$V \frac{\partial T}{\partial z} = \frac{k}{\rho c_p} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial z^2} \right) + \frac{q_a}{\rho c_p} \quad (15.A.16)$$

We can simplify this for flow in a tube where we neglect temperature gradients in any but the axial direction. Referring to Figure 15.A.2 and rewriting the gradient and the Laplacian as finite difference expressions, we obtain

$$V \frac{(T_2 - T_1)}{2L} = \frac{k}{\rho c_p} \frac{(T_1 - 2T_c + T_2)}{L^2} + \frac{q_a}{\rho c_p} \quad (15.A.17)$$

If we take the area of the duct to be A and the heat source to be at the center, which has a temperature T_c , we can rewrite the equation in terms of mass flow:

$$q_m(T_2 - T_1) = \underbrace{\frac{2kA}{c_p} \frac{(T_1 - 2T_c + T_2)}{L}}_{\text{convection term}} + \underbrace{\frac{2LA}{c_p} q_a}_{\text{heat input term}} \quad (15.A.18)$$

The first and last terms constitute the simplest approximation for the behavior of the thermal gas flowmeter. Heat is convected away by the flow, and the difference of temperature between inlet and outlet is related to the heat input, the specific heat, and the mass flow. This assumes that the first term on the right-hand side can be neglected. However, in the case of no flow, the heat added is conducted away, and the central term cannot be neglected because it gives the size of the conduction.

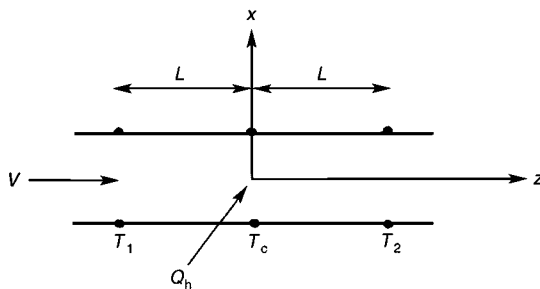


Figure 15.A.2. Diagram of tube to illustrate the use of finite differences.

15.A.4 HASTINGS FLOWMETER THEORY

Figure 15.A.3 shows the heat flows in the Hastings flowmeter based on an approximate one-dimensional theory. From the heat flows in the diagrams, we can derive the simple equation of the meter. We consider first the wall, and specifically an element of length δz . Here we have a heat addition as a result of the electric current I , which flows down the tube wall of resistance R per unit length. Thus the heat input is $I^2 R \delta z$. From this element, heat flows along the wall, of amount δQ_h , which can be related to the temperature gradient in the wall

$$Q_h = -kA \frac{dT}{dz} \quad (15.A.19)$$

where A is the cross-section of the wall, and k is the thermal conductivity of the wall material. Heat also flows out as heat loss from the meter, μT , μ being a dimensional constant to give the heat loss based on the wall temperature, and it flows into the fluid, $q_\perp$. From this we have an equation

$$I^2 R \delta z = \mu T \delta z + q_\perp \delta z + \delta Q_h \quad (15.A.20)$$

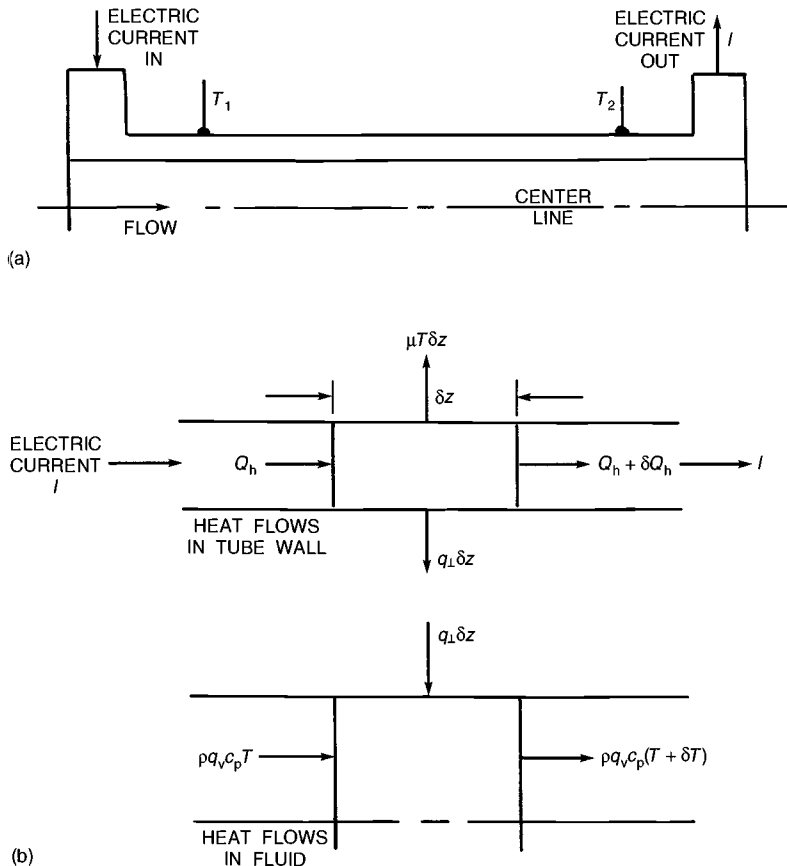


Figure 15.A.3. Geometry and heat flows for the Hastings flowmeter: **(a)** Schematic diagram; **(b)** Heat flows within the meter.

$q_{\perp}\delta z$ then flows into the fluid, which is assumed to be at approximately the same temperature as the wall at each point along its length. This heat causes a rise in the temperature of the fluid. The rate of heat addition $q_{\perp}\delta z$ is equal to the product of the temperature rise of the fluid, the specific heat of the fluid at constant pressure, and the mass flow rate. Thus we can write

$$q_{\perp}\delta z = \rho q_v c_p \delta T$$

or

$$q_{\perp} = \rho q_v c_p \frac{dT}{dz} \quad (15.A.21)$$

Finally, we can combine Equations (15.A.20) and (15.A.21) and use Equation (15.A.19) to eliminate Q_h

$$\frac{-dQ_h}{dz} - \rho q_v c_p \frac{dT}{dz} - \mu T + I^2 R = 0$$

or

$$kA \frac{d^2 T}{dz^2} - \rho q_v c_p \frac{dT}{dz} - \mu T + I^2 R = 0 \quad (15.A.22)$$

This equation is quoted by Hemp (1995a), and he also quotes a solution due to Blackett and Henry (1930), which for low flow rates can be written in a linear form

$$T_2 - T_1 = K q_v \quad (15.A.23)$$

Hemp suggests that the nonlinear complete solution is probably invalid outside the linear range due to the assumptions about uniform temperature across any cross-section of the flowmeter. For other solutions and more details, the reader is referred to Hemp (1994a, 1995a), Blackett and Henry (1930), Komiya et al. (1988), Brown and Kronberger (1947), and Widmer et al. (1982).

15.A.5 WEIGHT VECTOR THEORY FOR THERMAL FLOWMETERS

Hemp (1994a, 1995a) presented an extension of the weight vector theory to thermal flowmeters. Again the ideal weight vector must satisfy

$$\nabla \times \mathbf{W} = 0 \quad (15.A.24)$$

For a typical thermal diffusion flowmeter, the signal

$$T_B - T_C = \int \mathbf{V} \cdot \mathbf{W} dv \quad (15.A.25)$$

and the weight vector is given by

$$\mathbf{W} = \rho c_p T^{\text{CB}} \nabla T^{\text{A}} \quad (15.A.26)$$

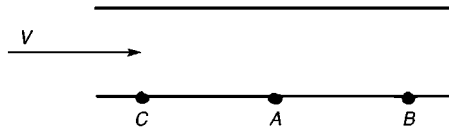


Figure 15.A.4. Diagram for Hemp (1994a) theory.

where T^A is the temperature distribution resulting from unit heat flux injected into A (Figure 15.A.4), and T^{CB} is the temperature distribution when unit heat flux enters at C and leaves at B. It appears that the usefulness of Hemp's theory is limited at present by the nature of the cases for which it is valid and the flow rates that are low enough to ensure linear conditions.

Angular Momentum Devices

16.1 INTRODUCTION

I am indebted to Medlock (1989) for notes on three early devices that attempt to use change of angular momentum to obtain mass flow.

Katys (1964) used an electric synchronous motor with a special rotor supported internally in the pipe and a stator outside the pipe. Fluid passed through vanes in the rotor and acquired angular momentum. From the power used or the torque needed to drive the rotor, it was claimed that the mass flow could be deduced.

According to Medlock, the Bendix meter “measures the torque required to impart angular momentum to the liquid. One end of a calibrated spring is driven at a constant speed and the other end is connected to a freely rotatable turbine. The torsion developed in the spring is a measure of mass flow rate.”

The twin rotor turbine meter (Potter 1959) attempted to measure mass flow by means of two in-line turbine rotors of different blade angles, which are joined by a torsion spring. Despite discussions with Medlock about this meter, I am not convinced that this is a true mass flow meter and suggest that the valid derivative of the Bendix meter is the fuel flow transmitter described later.

An early design with many similarities to the current commercial designs was a device due to Orlando and Jennings (1954) shown in Figure 16.1. A constant speed motor drives an impeller imparting swirl to the liquid.

The angular momentum, which is removed from this liquid per second as it leaves the driven rotor and moves into the tethered rotor, is

$$X = \omega R^2 q_m \quad (16.1)$$

where ω is the angular velocity of the rotor, and R is the radius of the annulus in which the flow takes place. The force on the spring restraining the tethered rotor will be equal to the loss in angular momentum per second and will be indicated by the angular deflection of the tethered rotor. The mass flow will therefore be given by

$$q_m = s\theta/(\omega R^2) \quad (16.2)$$

where θ is the angular deflection, and s is the spring constant. It is important to note the following.

- i. The flow will not precisely follow the vanes unless sufficient length is allowed to force the flow to be axial relative to the blades.

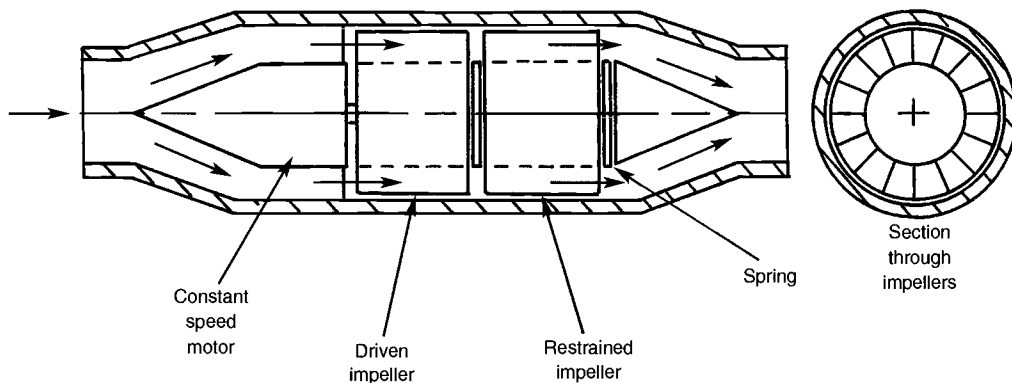


Figure 16.1. Orlando and Jennings device.

- ii. The flow profile in the annulus will modify the effective value of R .
- iii. The motor speed must be known.

16.2 THE FUEL FLOW TRANSMITTER

Much of the material in this section comes from a lecture by Rowland (1989). The main features of a particular commercial device are shown in Figure 16.2 and in exploded view in Figure 16.3. This design eliminates the electrical drive by extracting power from the liquid flow to drive the turbine and provides one example of commercial meters of this type.

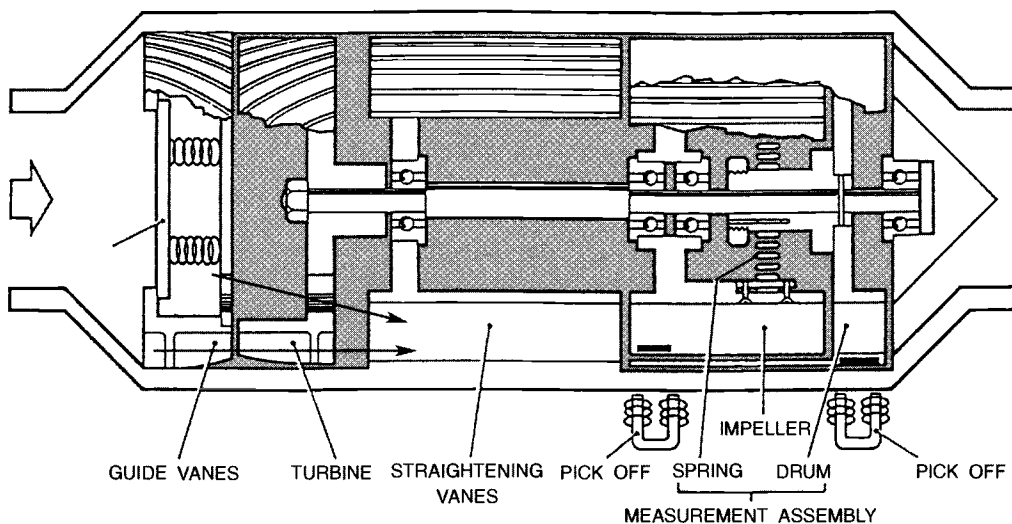


Figure 16.2. Diagram of the fuel flow transmitter (reproduced with permission of GEC-Marconi Avionics).

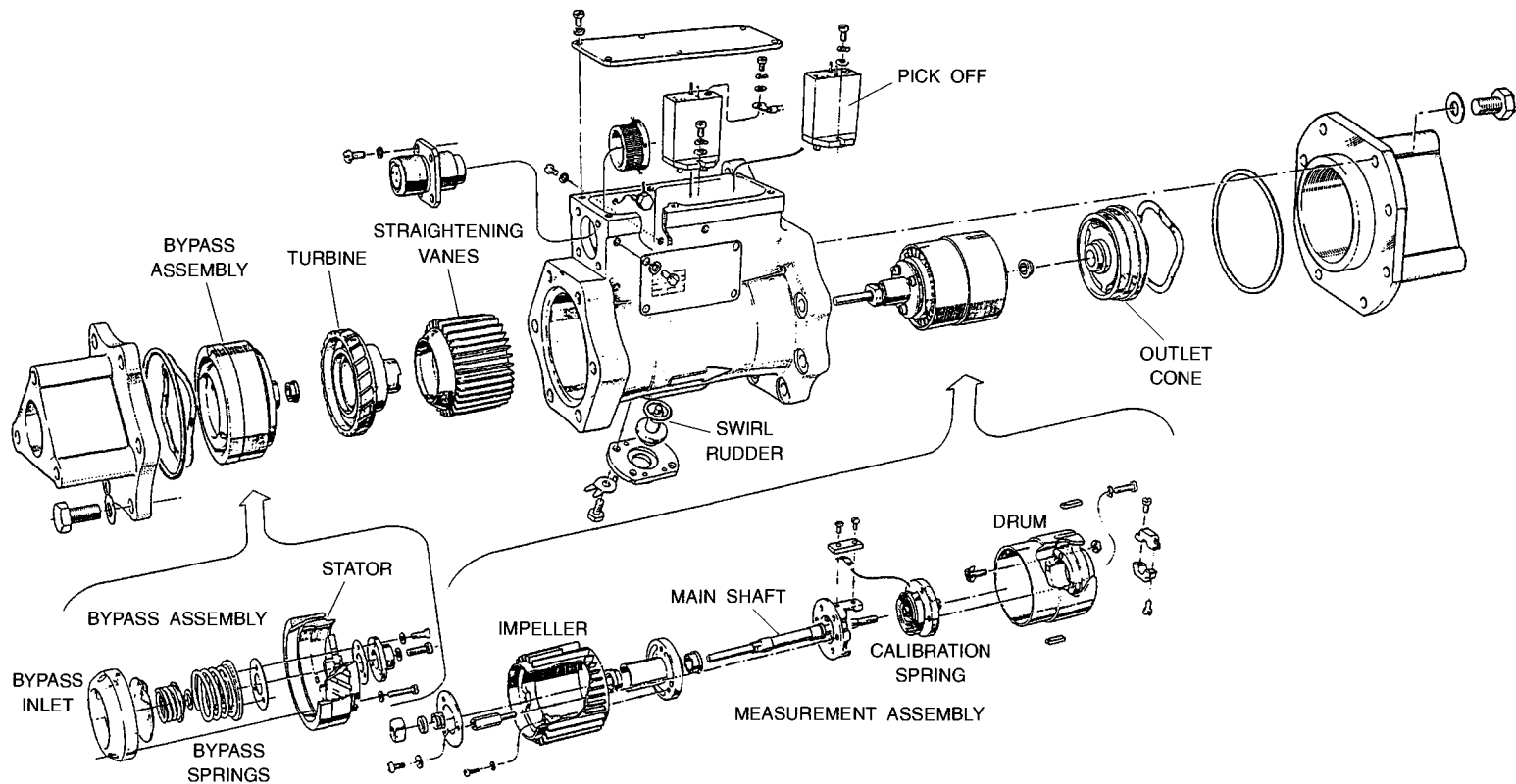


Figure 16.3. Exploded diagram of the fuel flow transmitter (reproduced with permission of GEC-Marconi Avionics).

16.2.1 QUALITATIVE DESCRIPTION OF OPERATION

As the flow enters the unit, a bypass assembly controls the amount of liquid passing through the power turbine in order to avoid the assembly's running too fast or too slow. The bypass mechanism consists of a spring-loaded bypass valve that contains the speed of the unit to within the range 70–400 rpm. At the lowest flow rates, all the liquid passes through the turbine, but as the flow rate increases, the build-up of pressure across the turbine progressively opens the valve and allows an increasing portion of the total liquid flow to bypass the turbine.

Depending on the opening of the bypass valve, a proportion of the flow passes through a stator with guide vanes that create swirl in the flow. The flow then enters the power turbine, which drives a shaft with the measurement assembly on the other end. However, having extracted power from the flow in this way, it is essential that the remaining swirl after the power turbine is completely removed from the liquid.

After the turbine and bypass valve, therefore, the flows combine to pass through the straightening vanes, before passing into the driven measurement assembly. The meter proper then follows and in this particular design, unlike the example described earlier, measures the torque needed to impart the angular momentum to the liquid. Within the rotating measurement assembly, a tethered impeller imparts to the liquid the angular momentum. The ratio of blade length to mean space between the impeller vanes is 8 : 1. Clearances are kept to a minimum to reduce leakage, and the meter is insensitive to installation attitude.

In the process, the impeller experiences a torque that causes the restraining spring made of NiSpan-C902, a material whose change of stiffness with temperature can be adjusted by heat treatment, to tighten. To avoid overtightening the spring, the bypass control of speed also reduces the necessary deflection of the spring. This can be seen from the fact that, without the bypass, the angular velocity would be approximately proportional to the mass flow rate and so the torque would be proportional to the square of the angular velocity. By allowing some of the fluid to bypass the drive turbine, the angular velocity is reduced, and the speed need not rise in proportion to the mass flow rate. The relationship between flow rate and rotational speed is shown in Figure 16.4.

The bearings are precision ball races. Friction errors and vibration errors can result from spurious torques at the impeller bearings. The fuel acts as a lubricant and must therefore be clean (filtered).

The deflection is measured by means of a pair of electromagnetic pickup coils placed on the outside of the nonmagnetic meter body. A pair of small powerful magnets displaced 180° apart are attached to the circumference of both drum and impeller. Signals are therefore generated for each 180° of rotation of the assembly.

16.2.2 SIMPLE THEORY

We can obtain an expression for the mass flow rate. We write the spring constant as s . The torque causes the impeller to be displaced by an angle θ from the rest of the measurement assembly. The displacement in the range 0–160° is measured by recording the time of transit of a marker on the assembly drum and a marker on the impeller. A knowledge of the time delay τ and the angular velocity of the assembly ω will then allow the angular displacement to be deduced. Maximum calibration

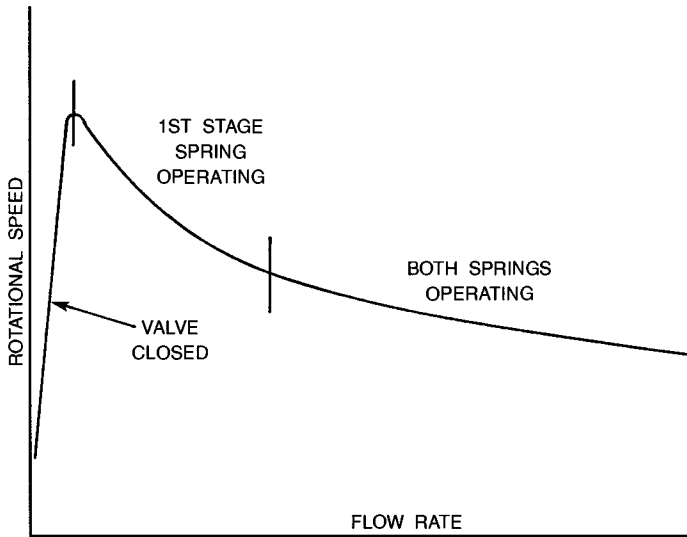


Figure 16.4. Relation between flow rate and rotational speed for the fuel flow transmitter (reproduced with permission of GEC-Marconi Avionics).

spring hysteresis is approximately 0.05% of FSD. Thus

$$q_m = s\theta/(\omega R^2)$$

where the effective radius of the flow annulus is taken as R . This is affected by the velocity profile in the impeller and the shape of the channel between the vanes. Hub-to-tip ratio is generally greater than 0.7.

The time difference between the markers is

$$\tau = \theta/\omega \quad (16.3)$$

so that

$$q_m = s\tau/R^2 \quad (16.4)$$

and we, thus, note that the mass flow may be obtained as directly proportional to τ , provided s is constant, and our simple description of operation is adequate.

16.2.3 CALIBRATION ADJUSTMENT

The meter has three methods of calibration adjustment (Figure 16.5):

- i. Zero adjustment achieved by moving the impeller pickup circumferentially out of alignment with the drum coil so as to introduce an apparent zero flow adjustment.
- ii. Rate adjustment achieved when the meter is partially disassembled by altering the clamping point of the spring and hence its effective length. This will alter the value of s in the preceding equations.
- iii. Swirl adjustment by means of a small rudder in the flow upstream of the measurement assembly creates a small local flow deflection and affects the performance throughout the characteristic of the meter.

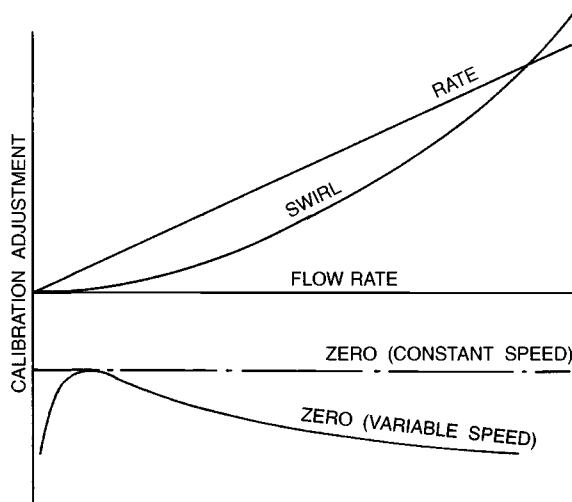


Figure 16.5. Calibration adjustments (reproduced with permission of GEC-Marconi Avionics).

16.2.4 METER PERFORMANCE AND RANGE

The meter is adjusted to rotate at 150 rpm at maximum flow and between 150 and 300 rpm at other flow rates. Figure 16.4 indicates an increase in rotational speed at lower flows resulting from the action of the bypass valve to maintain spring wind-up. The assembly is fitted with a stop to prevent, at low flow rates, a reverse oscillation causing spurious readings. It has a flow range of about 115–4,500 kg/h. The temperature range is -40 to 150°C , and the maximum line pressure is about 130 bar with a maximum pressure drop of about 0.5 bar.

Accuracy appears to be claimed as $\pm 1\%$ of rate over a 7 : 1 flow range.

It is desirable to install the meter with a straight upstream length of pipework to avoid swirl and profile distortion effects. Because this is seldom possible on an aircraft, the calibration should be done with a representative piece of inlet pipework.

Referring to Equation (16.4), we can review the likely sources of uncertainty in this meter that will influence the claims for its accuracy. It is unlikely that the impeller completely imparts the assumed angular velocity to all the fluid. In addition, there will be some leakage and drag due to the gap between rotor, tube, and vanes, although the shroud may reduce some of these. These may be approximately proportional to the mass flow rate so error may be of second order. The spring may introduce nonlinearity or hysteresis and may give problems if not correctly installed. The time difference may be of order 100 ms, and so measurement will be to within about $100\ \mu\text{s}$. There are also the characteristic changes due to the means of zero adjustment. These may be incorporated into a calibration characteristic if a flow computer is available to adjust the flow signal accordingly.

The material of the body is high strength aluminum alloy.

16.2.5 APPLICATION

The meter appears to have primarily one application to date. This is for the measurement of fuel flow in aircraft.

16.3 CHAPTER CONCLUSIONS

The fuel flow transmitter is an elegant solution but probably suffers, in an age when mechanical precision is being displaced by solid state, from being a mechanical rotating device. If the mechanical integrity can be assured and operating life is long enough, one would expect that sensing of the rotation, torque, and other factors, with intelligent secondary instrumentation would make it highly accurate.

It is surprising that it has not been exploited for more applications, and even for gas flow measurement. Presumably the main reason is the preference for a meter without moving parts.

The logical next step to overcome this would be to move to an oscillating or vibrating device to achieve the same result. Such devices have been suggested but may offer little advantage over the Coriolis, which we shall look at next. Any developments in this device will also be in competition with the rapidly developing ultrasonic methods that clearly meet many of today's requirements in terms of no moving parts and predominantly electrical sensors.

The future development of a device such as this, therefore, is unlikely to be extensive.

Coriolis Flowmeters

17.1 INTRODUCTION

This is one of the most important recent meter developments, and its application in the next few years is likely to impinge on many areas of industry. Its importance is recognized by the main national and international advisory bodies (cf. Mandrup-Jensen 1990 who described initial work in Denmark on pattern approval and ISO developments).

17.1.1 BACKGROUND

Plache (1977) makes the interesting point that mass cannot be measured without applying a force on the system and then measuring the resulting acceleration. This is a point that I have long considered a possible requirement, and it is certainly supported in the Coriolis meter.

Possibly the first application of the Coriolis effect for mass flow measurement was proposed by Li and Lee (1953). The meter is shown in Figure 17.1. The T-piece flow tube rotated with the outer casing and was linked to it by a torque tube. As the flow increased, so the T-piece experienced a displacing torque due to the Coriolis acceleration, and this was measured from the displacement of the T-piece relative to the main body.

In the Li and Lee meter, the liquid was forced to move radially, and therefore a force was applied to it through the tube. This force, in turn, was balanced by an equal and opposite one applied by the liquid to the tube. The force caused the tube to twist, and the small rotation was sensed to obtain the mass flow rate.

It is interesting to note that Stoll (1978) made no mention of the technology that was to revolutionize mass flow measurement. Medlock (1989) referred to several designs that, although not identified as dependent on the Coriolis effect, have a very similar basis. Macdonald (1983) described the vibrating vane flowmeter, Vsesojuzny (1976) described the vibrating nozzle flowmeter shown in Figure 17.2 (cf. Hemp's 1994b recent work on vibrating elements). Finnof et al. (1976) described the vibrating tube flowmeter shown in Figure 17.3.

Tucker and Hayes (1982) analyzed the error in oil flow rate for a vibrating pendulum two-phase flowmeter invented by Rivkin (1978) and its suitability for North American oil well applications. Reimann et al. (1982) tested a true mass flow meter (TMFM) on a steam-water loop. This instrument created an angular momentum in the fluid and then measured the torque needed to remove it. John et al. (1982) built and tested a TMFM for up to 50-kg/s flows in air-water. They found that it was nearly

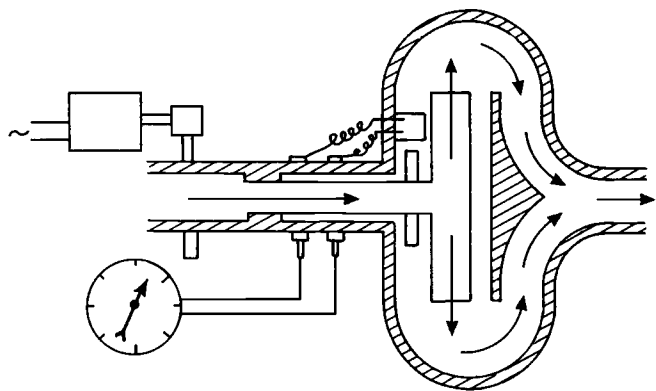


Figure 17.1. Li and Lee (1953) fast response true mass-rate flowmeter (reproduced with the permission of ASME).

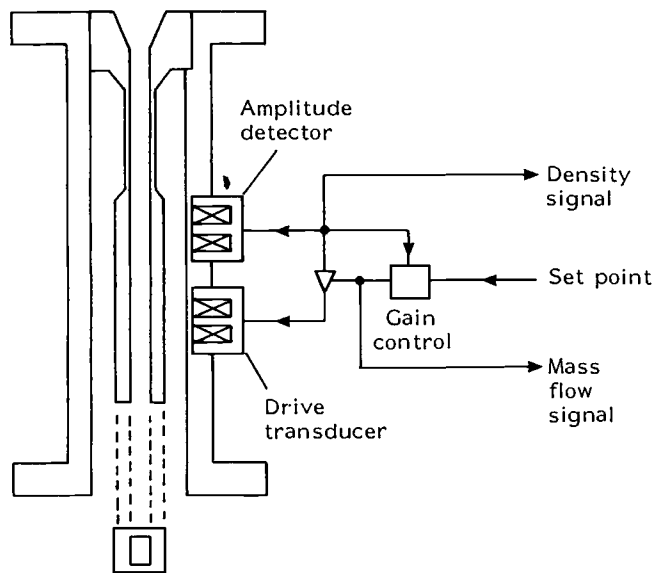


Figure 17.2. Vsesojuzny (1976) vibrating nozzle.

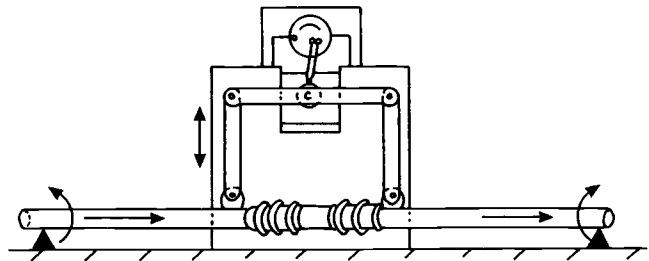


Figure 17.3. Finnof et al. (1976) vibrating tube.

independent of the flow regime and had an overall measurement uncertainty of less than $\pm 1.5\%$ of full-scale deflection up to 20% mass fraction of air, although they observed a shift in the characteristic of 2.5% for mass fractions of air above 1%. This was an impressive performance. However the instrument is unlikely to be attractive for most applications due to the requirement to drive a rotor and the consequent design and maintenance problems.

Dimaczek et al. (1994) suggested the use of a radial turbine type wheel to create a Coriolis effect meter for dosing gas-solid flows. They claimed that a measurement uncertainty of $\pm 1\%$ rate was achieved, and that it was used for flows of coal dust, quartz sand, feldspar, plastic granulates, and foodstuffs.

Decker's (1960) design can be explained in terms of a modern Coriolis meter.

It appears to have been the meter developed by Finnof et al. (1976, cf. Smith and Cage 1985), which was first successfully launched into the market in 1976 (Plache 1977, cf. 1980) according to Medlock (1989). Tullis and Smith (1979) reported early tests on meters of $\frac{1}{8}$, $\frac{1}{4}$, and 1 in., which were promising and led to optimism for wide application of the meters to liquids, gases, and two-phase flows.

17.1.2 QUALITATIVE DESCRIPTION OF OPERATION

My explanation of the operation of these flowmeters will be based on the U-tube design (Figure 17.4) rather than on the most recent straight tube design, which I expect to become increasingly common. The theory is essentially the same for both, but I find it easier to explain for the U-tube flowmeter. We consider one of these vibrating tubes, and we consider the period of vibration during which it rotates upward. A simple theoretical treatment is given in Appendix 17.A.1.

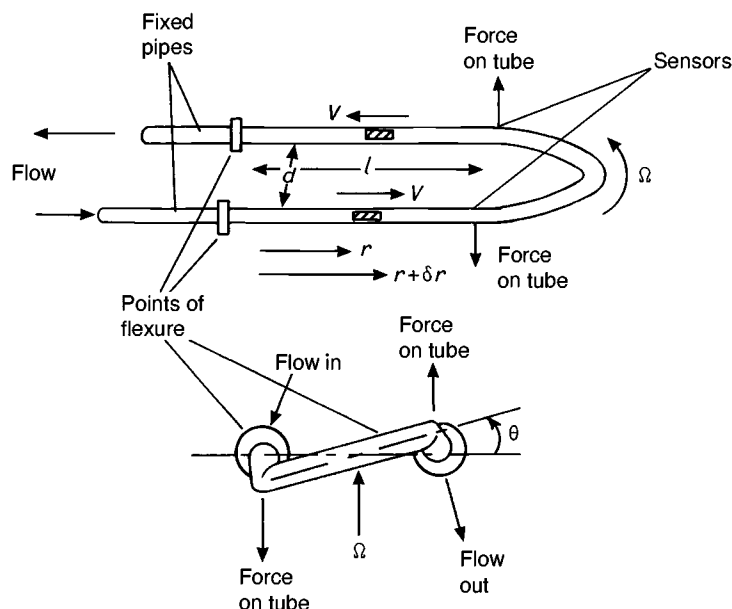


Figure 17.4. Diagrams of the U-tube meter to show motion of the vibrating tube and consequent forces during the upward movement (after MicroMotion).

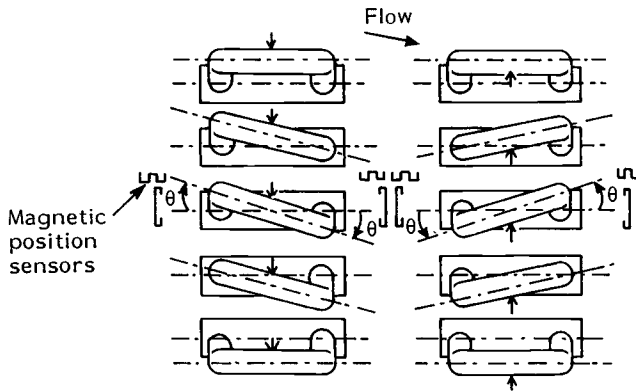


Figure 17.5. Variation of twist with vibration for U-tube meter (reproduced from MicroMotion's brochure with permission).

Consider first a portion of the fluid moving outward in half of the U-tube. It will have a certain angular momentum because of the rotation of the tube upward. As it moves outward and the tube rotates upward, the angular momentum will increase because the same portion of fluid will be at a greater radius from the hinge and so will have to move at a greater velocity upward. To achieve this, it is necessary to apply a force on the portion of fluid in an upward direction. As a result, the portion of fluid will apply an equal downward force on the pipe. Now consider the other half of the U-tube where the fluid is moving inward and another portion of fluid. As this portion moves inward, its angular momentum will be reduced because the upward velocity of the tube will be less. The tube will therefore have to apply a force downward on the portion of fluid, and this will, in turn, apply a force upward on the tube. The result of all this is a force twisting the tube so that the near side is forced down and the far side is forced up.

The principle of operation is based on the Coriolis acceleration, resulting from the flow of fluid through the flow tube. However, the motion is vibration rather than pure rotation. The forces are essentially the same, but are alternating. The vibration takes place in such a way that one end is essentially stationary, and the other end is vibrating through an arc. This is illustrated in Figure 17.5.

As the tube rises, the forces are at their maximum as it passes the midplane, and at the top of the motion the forces become zero and the twist will cease. As the U-tube descends, the forces reverse, and the twist reverses so that again at the midplane the twist is at a maximum. It is then apparent that if the transit time of the two halves of the U-tube past the midplane are measured, the difference will be related to the twist in the tube and therefore to the mass flow through the tube.

If we, now, unbend the U-tube so that it forms a straight tube with the vibrating drive at the center, each half will "rotate" about its end. If we consider the center rising, the inlet half will be distorted downward by the Coriolis forces, and the outlet half will be distorted upward. The sensors at the quarter and three-quarter positions will sense the lag, rather than the twist, caused by the Coriolis forces.

17.1.3 EXPERIMENTAL INVESTIGATIONS

Adiletta et al. (1993, cf. Cascetta et al. 1989a, 1989b) described a prototype that sought to overcome the limitations of commercial instruments' dependence on tube deformation by

- a. a system of rigid tubes,
- b. a separate elastic suspension, and
- c. electromagnetic drive and capacitance transducers to infer the twist.

Their design appeared to be affected little by external vibrations.

Sultan (1992) described a single-tube meter of 28-mm OD built for laboratory experiments. It had a drive coil in the center of the 1410-mm-long straight pipe and two symmetrically positioned detector coils. For his rig, a change of water temperature of about 14°C caused a zero flow calibration shift equivalent to about 0.4 m/s (maximum flow rates tested were about 5.6 m/s). He accounted for this as being due to differential thermal expansion and the consequent differential characteristics. This will lead to frequency change. Tube temperature change will also cause a change in the modulus of elasticity and hence a frequency change. Sultan also showed that a pump in the flow circuit caused a zero fluctuation due to vibration equivalent to a flow of about ± 0.1 m/s.

Kolahi et al. (1994) described a prototype meter with similarities to the MicroMotion meter having two U-tubes. The design was very versatile in terms of component changes. Optical, capacitive, and inductive sensors could be used. It was also possible to tune it so as to amplify the torsional amplitude by up to 100 times.

Hagenmeyer et al. (1994) reported a design of a single-tube, compact Coriolis meter operating in a hoop mode and claimed that experimental results were promising.

17.2 INDUSTRIAL DESIGNS

The MicroMotion meter was the first to appear in 1981 and resulted from patents filed in 1975, 1977, and 1978 (cf. Willer 1978 and Smith 1978). The initial device consisted of a single U-tube. A recent paper by Tsutsui and Yamikawa (1993) seemed to be using one tube plus a resonance vibrator as in the early MicroMotion meter.

Twin tubes were introduced in model D in 1983. The main features of this meter are a single inlet splitting into the two tubes, a drive system to cause the tubes to vibrate, a pair of sensors to detect the movement of the tubes past the midplane, and a strong point around which the tubes vibrate in opposition. In some meters, the outer cover forms a pressure vessel so that, if the tubes fail, the fluid will be contained. Failure should not be due to fatigue, because manufacturers should design so that tubes will not fail due to cyclic stressing. Tube failure almost always occurs due to corrosive stressing.

Several designs followed MicroMotion and a selection of these, some no longer available, are shown diagrammatically in Figure 17.6 based on my understanding of the manufacturers' brochures. Apart from those shown [Bopp & Reuther, Danfoss, Endress & Hauser, EXAC, Foxboro, K-Flow, Krohne, MicroMotion, Neptune (Schlumberger), and Smith], there maybe other manufacturers of this type of meter,

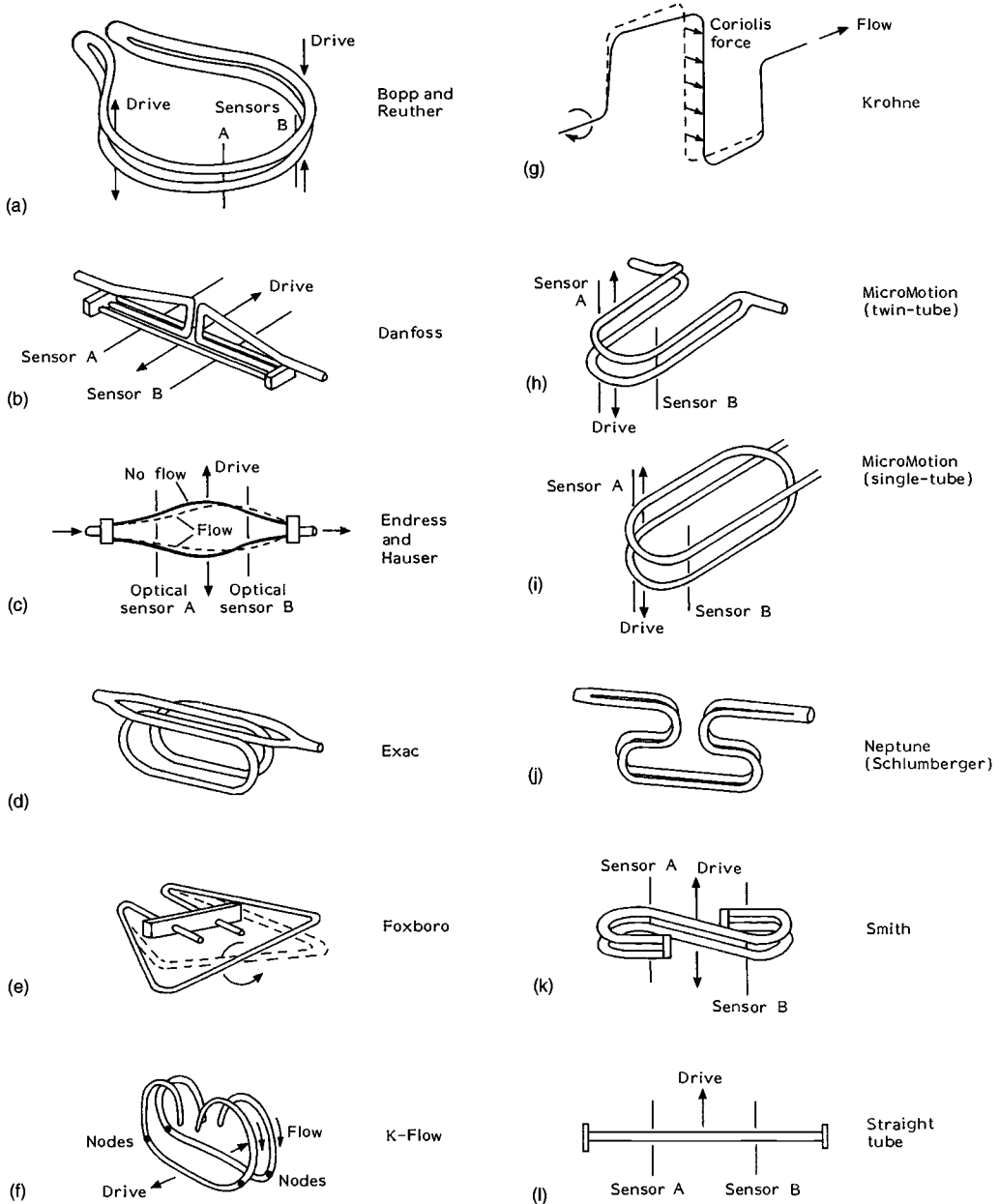


Figure 17.6. A selection of Coriolis-type mass flowmeters available commercially now or in the past: **(a)** Bopp and Reuther; **(b)** Danfoss; **(c)** Endress & Hauser; **(d)** EXAC; **(e)** Foxboro; **(f)** K-Flow; **(g)** Krohne; **(h)** MicroMotion (twin-tube); **(i)** MicroMotion (single-tube); **(j)** Neptune (Schlumberger); **(k)** Smith; **(l)** Krohne, Endress & Hauser, and others.

such as Heinrichs and Yokogawa. It should be noted that some manufacturers offer two options.

Most industrial designs in current operation use two parallel wetted tubes or two tubes in series. The deflection is measured from one tube to the other so that external

vibrations are largely eliminated and the measured shift between the tubes has the effect of adding the flows in the two tubes even if the flow split is not equal.

Single-tube versions (e.g., Hussain and Farrant 1994) are beginning to be introduced, and these are likely to be an increasingly attractive option for the future (Reider and Drahm 1996, Yamashita 1996). At present, these may be offered as a second design by some manufacturers (cf. Stansfeld et al. 1988, Harrie 1991 for an earlier design).

17.2.1 PRINCIPAL DESIGN COMPONENTS

Flow Tubes

The deflection of the tubes is sensed from one tube to the other in twin-tube versions or between tube and outer compensating tube in single-tube versions. This eliminates, to a large extent, external vibrations. The tubes are mounted to isolate them from external vibration and are designed to optimize phase shift between sensor signals and to minimize pressure loss. Wagner (1988) discussed trade-offs for tube selection. For instance, by lengthening the meter tube, vibration of the tube is easier, stresses are reduced, and twist and Coriolis force increase. However, this is at the cost of greater weight and pressure drop.

The problem of corrosion is a reason why some manufacturers use an outer containment vessel around the vibrating tubes in case of tube failure.

Foxboro has introduced a technique called antiphase excitation, which uses a double driver system to produce a rotating motion on the tube essentially at the flattened bottom of the U-tube. In this way, the stresses are greatly reduced because the vibration is away from the pipe joints. This appears to be similar to the Bopp & Reuther and the Krohne designs. Watt (1990) described work on the computation of pipe stresses.

Drive Mechanisms

Drive mechanisms are most commonly electromagnetically excited to oscillate at the chosen frequency. In some cases, this is the natural (resonant) frequency; in others, it is a harmonic of the natural frequency. Higher frequencies are claimed to reduce the effect of external disturbances by introducing free nodes of vibration and by working farther above the most common surrounding frequencies. The control circuit then keeps the oscillation at the chosen frequency. For high viscosity or density, additional power may be required in the drive.

Sensor Types

In meters with twin tubes, the sensor usually measures the displacement between the two tubes, thus eliminating any common spurious vibrations. One method used in several designs has a coil on one tube and a magnet on the other. The relative velocity causes a voltage to be generated in the coil, and, in order to obtain position, this signal is integrated. Another method (Vogtlin and Tschabold, Endress & Hauser) is optical and uses an arrangement of photodiodes and a modulating shutter (Figure 17.7). "Small plates acting as shutters are attached to the tubes ... leaving a defined gap between each pair. The shutter changes as the two tubes oscillate in phase opposition. ... Infrared light is beamed from a light-emitting diode into a sapphire rod with face ground to a 45° angle. Thus the quantity of light is modulated ... and converted into alternating electric current. ..."

Temperature sensing is included in some commercial instruments (e.g., the use of a platinum resistance thermometer (PRT) with resistance set to 100 ohms). Nicholson (1994) noted that the location of the temperature sensor within the meter housing had a significant effect on the precision with which fluid temperature was measured. His results suggest discrepancies ranging up to $\pm 2^\circ\text{C}$. In Krohne's single straight tube design, a strain gauge has been added together with a temperature sensor, in order to obtain information from the meter for subsequent corrections by the electronics.

Kalotay (1994) used the pressure drop through the MicroMotion meter to obtain a value of viscosity.

Secondary Containment

Some manufacturers design the flow tube with an outer containment tube so that, in the event of flow tube failure, the process fluid is contained (cf. Hussain and Farrant 1994 who quote a bursting pressure of 500 bar with in-house testing up to 300 bar).

Meter Secondary

As indicated earlier, the distortion in the tube due to Coriolis forces will cause the two sensors to pass the midplane at different times. The phase shift at full-scale flow may be about 1° . At zero flow, the phase difference should be zero, although normally there is an offset and hence a zeroing routine.

Figure 17.8(a) shows the block diagram of a signal processing system. The drive coil vibrates the tube at its natural frequency. Sensing obtains tube velocity, which is integrated to obtain position and in so doing filtered to remove some noise.

The conversion of the signals is shown in Figure 17.8(b). Position detector signals are slightly offset, and the effect at zero flow can be seen. A reference current is integrated during the pulses so that the voltage on the integrator is proportional to the time period and hence the mass flow.

An alternative is to use a high speed clock to count the time differences. At maximum flow this is of order $120\ \mu\text{s}$. The minimum measurement must therefore be less than 10 ns to discriminate at the lowest flow rates.

For their latest designs, MicroMotion claims to achieve a time difference between the two pick-off signals of about $60\ \mu\text{s}$ at the top of the range and, with a 100:1

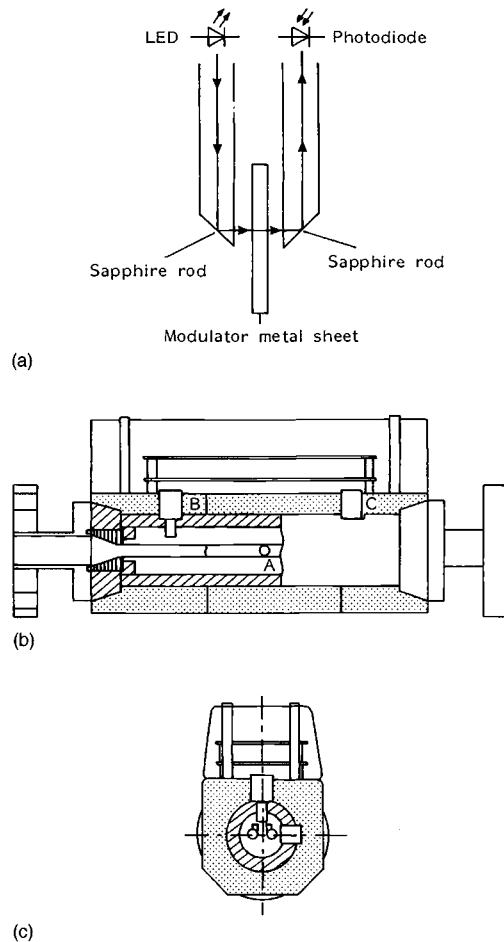


Figure 17.7. Endress and Hauser modulating shutter (reproduced with permission): **(a)** Schematic representation of optical sensor; **(b)** Longitudinal and **(c)** cross-sectional construction of the measuring sensor; A, electromagnetic excitation (coil); B, C, optical detection of the tube oscillations.

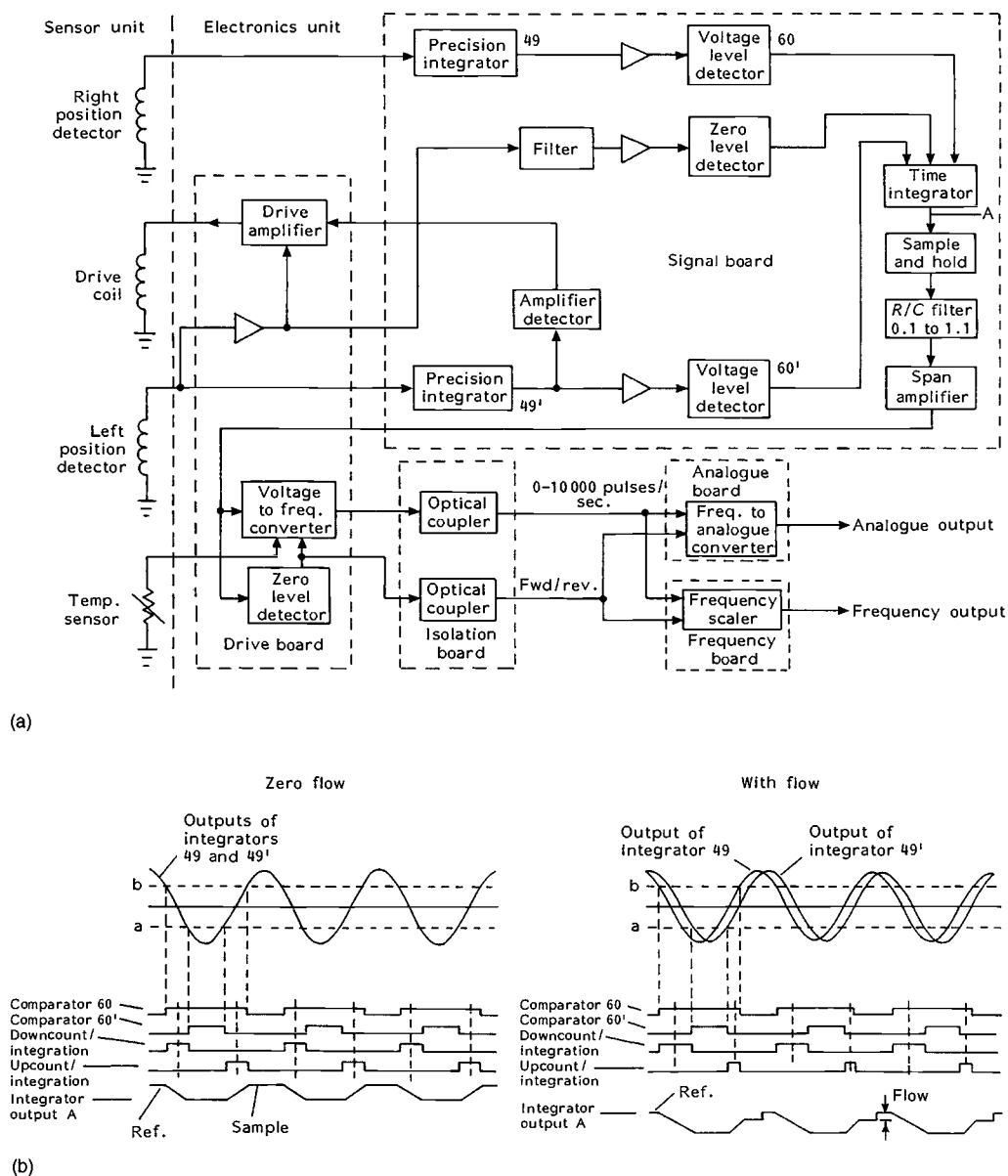


Figure 17.8. MicroMotion signal processing system (reproduced with permission): **(a)** Block diagram; **(b)** Conversion of signals.

turndown, $0.6 \mu\text{s}$ at the bottom of the range. To measure this with an uncertainty of $\pm 0.4\%$ requires a resolution of about $\pm 2.4 \text{ ns}$.

In another signal processing system (Vogtlin and Tschabold undated), the circuit keeps equal the amplitudes of the alternating currents, which energize each of the sensing photodiodes. These are then fed into a differential amplifier. A phase locked loop (PLL), avoiding phase shift, ensures that the tubes vibrate at resonance, and hence the density of the fluid is obtained.

17.2.2 MATERIALS

Materials of construction may be: for wetted parts 316L stainless steel, NiSpan-C, Hastelloy-C22, titanium, and zirconium; for gaskets Viton and Kalrez; and for non-wetted parts 304 or 316 stainless steel.

Manufacturers have had to address the problem of corrosion fatigue, which is a particular problem where materials are undergoing cyclic stress and an aggressive environment. Carpenter (1990) provided a useful introduction to material selection and gave information about alloy composition and examples of corrosion. It is necessary, therefore, in the Coriolis flowmeter, which depends on vibration and is applied to some very aggressive fluids, to take particular care that corrosion fatigue does not occur and that, if it occurs, the results are contained. This has been achieved by the following:

- ❑ Correct choice of materials. Corrosion resistance of 316L stainless steel is a result of its ability to form a protective oxide film due, primarily, to the presence of chromium, nickel, and molybdenum. However, this film can break down in certain fluids and cause pitting, intergranular corrosion, stress corrosion cracking, and corrosion fatigue (MicroMotion information). Hastelloy-C22 may provide resistance to both oxidizing and reducing media and is superior to local attack due to chloride ions because of the formation of a very stable oxide film that is little affected by chloride ions. However, there is a need for care in the manufacture of tubes from Hastelloy due to localized diffusion of carbon at grain boundaries resulting from problems in the drawing and annealing process. In addition, brazing with filler materials may cause intergranular penetration cracking of the Hastelloy (MicroMotion information). Titanium is used by some manufacturers because of its favorable elasticity, low density and compressive strength, coefficient of thermal expansion, and corrosion and abrasion resistance to many different aggressive fluids (cf. Wagner 1988 on the problem of intergranular stress corrosion, pitting, etc.);
- ❑ Control of amplitude of vibration to ensure that the stress cycles keep within the design envelope.
- ❑ Containment of the vibrating tube in a pressure vessel so that, in the event of failure, the process line is intact. 316L stainless steel wetted parts may be standard (except with chlorides and halogens) on most Coriolis flow sensors. This grade of stainless provides corrosion resistance with the majority of fluids used in the process industries.

As described earlier, sensor tubes must be flexible to generate sufficient deflection and phase shift. Thus, relatively thin-walled tubing is required. For example, a typical 25-mm (1-in.) sensor may use 19-mm ($\frac{3}{4}$ -in.) OD tubing with a nominal wall thickness of 1.65 mm (0.065-in.). Unlike most other system components, there is minimal corrosion and erosion allowance. Therefore, it is not recommended that users rely on standard compatibility guides, which often incorporate such allowances.

17.2.3 INSTALLATION CONSTRAINTS

Sensors may, in some cases, have preferred orientations due to the complexity of the meter pipework. For instance, vertical downward flow may sometimes result in a partially empty flow tube. For liquid applications, any trapped gases should be

able to escape, as should any liquid or condensate for gas applications. In addition, pipework arrangements that might lead to trapped air or gas (e.g., an inverted U) should be avoided.

Although they are unlikely to be sensitive to upstream flow profile, Cascetta et al. (1992) have suggested that axial swirl and turbulence spectra may have an influence on the flowmeter performance of some designs (but cf. Hemp 1994b). Nicholson (1994) found some effects due to pipeline configuration but no effects due to compressive or tensile loadings on the meters.

The presence of two-phase flow introduces its own problems due to the splitting of the flow that takes place in many instruments and the possible effect of the tubes on coalescence. Swanson (1988) described an in situ method that provided for recognition and compensation for coating or scale build-up within the meter. In all cases, manufacturers should be consulted on best practice.

Pressure variation can cause the tube dimensions to change with possible effects on the sensitivity and zero stability of the flowmeter. Nicholson (1994) found K-factor shifts due to fluid temperature.

Kiehl (1991) considered the adverse effect of an interference frequency close to the operating frequency of a Coriolis flowmeter, with particular reference to cross-talk between two similar meters working in close proximity in applications such as leak detection, difference measurements, or parallel flow configurations. To avoid this, mechanical decoupling of the meters may be possible by using rubber or plastic hose between the meters or by changing the cross-section with the insertion of a large cross-sectional pipe between the two meters. An alternative available from some manufacturers is to use two different working frequencies separated by a few hertz: Kiehl instances 6.5 Hz.

17.2.4 VIBRATION SENSITIVITY

The meter has been found to be affected by inadequate supporting structures that have a natural frequency of vibration within 20% of the operating frequency and by external vibrations close to the natural vibrational modes of the meter (e.g., side-to-side mode 200 Hz or twisting mode 30 Hz). To avoid effects due to external vibration, some meters operate at higher frequencies than the resonant frequency and at higher harmonics. Nicholson (1994) found only one of the eight meters that he tested to be affected by vibration, and in this one case he was unable to explain the fact that it occurred at a frequency different from the meter's (cf. Adiletta et al.'s 1993 meter designed to reduce the effects of external vibrations).

Vetter and Notzon (1994) showed that small amplitude pulsation ($\pm 2\%$ of the flow) disturbed the operation of the flowmeters when the frequency coincided with the torsional frequency of the measuring tube. The frequency range of concern is 30–1,000 Hz for most Coriolis meters. Reciprocating pumps are likely to have lower frequencies than these, but rotary pumps may cause problems. They suggested that below the $\pm 2\%$ the effects may not be severe, but some of their data above this value indicated serious errors.

17.2.5 SIZE AND FLOW RANGES

Meters are available in sizes from 6 mm or less to 200 mm or more and for flow ranges from 0–3 kg/h up to 0–680,000 kg/h. The weight of the instruments ranges from about 8 kg to more than 635 kg for the largest sizes.

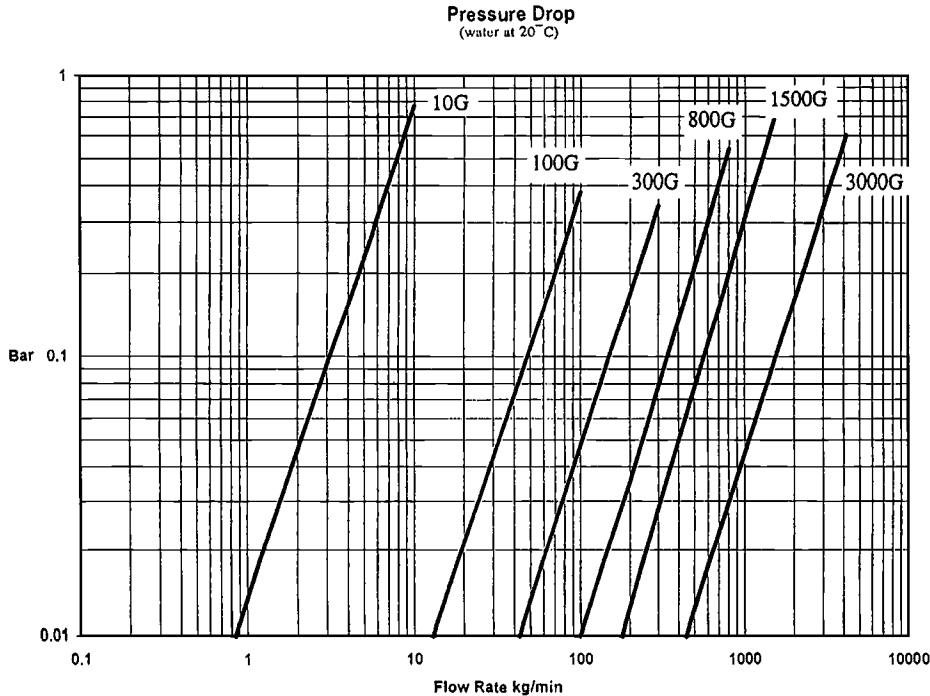


Figure 17.9. Typical flow ranges and pressure drops for straight tube design (reproduced with permission of Krohne Ltd.)

Figure 17.9 shows the typical ranges for one manufacturer with pressure loss. It should be noted that more than one size is suitable for most flow rates, and the manufacturer may be able to fit flanges to suit the particular application. Turndown ratio is typically in the bracket 20 : 1 to 100 : 1. Typical temperature range is -240 to 200°C , and high pressure versions in certain sizes are available up to 393 bars. There may be a minimum static pressure of about 1.2 bar. Zero stability appears to be about 0.01% of FSD. Performance is claimed to be unaffected by pressure variation, temperature variation, viscosity, line vibration, density, or back pressure.

Single-tube meters are available up to 75 mm diameter with overall lengths of about 400 mm to 1.25 m and flow rates from 15–1,200 kg/h up to 1,500–180,000 kg/h. Process temperature range is typically -25 to 130°C and 63 bar. Measurement uncertainty is from ± 0.05 to $\pm 0.3\%$ or more depending on zero stability. The outer tube is likely to become a more important element in the fundamental design and may require weights positioned to give the correct natural frequency, as well as to provide secondary protection, etc. (cf. Hussain and Farrant 1994, Yamashita 1996, and Reider and Drahm 1996).

Koudal et al. (1996) reported on a 3.5-mm ID design for flows up to 450 kg/h that has a special design of carrying plate with damping elements on the nodal line.

17.2.6 DENSITY RANGE AND ACCURACY

Most meters offer the option of density measurement for a typical range of $0\text{--}3,000\text{ kg/m}^3$ ($0\text{--}3\text{ g/cm}^3$). The uncertainty (over the calibration range) is from ± 0.0002 to $\pm 0.01\text{ g/cm}^3$, with a typical value of about 0.001 g/cm^3 . Repeatability is ± 0.0001 to $\pm 0.003\text{ g/cm}^3$.

In tests on water, naphtha, and two different crude oils (Eide and Gwaspari 1996), all the density outputs showed some dependence on either temperature or pressure.

Density is a more problematic measurement in gas, but with careful calibration Pawlas and Patten (1995) claimed that it was possible.

17.2.7 PRESSURE LOSS

To reduce pressure loss in the tube, a flow rate in the lower half of the normal range is recommended in some cases (Figure 17.9 is for straight tube design). For other designs, pressure loss at half range is of order 0.5 bar, and at full range it is nearly 2 bar. With viscosities of 200 cSt or more, the losses may be as high as 5 bar. However, for a single straight tube instrument, the pressure loss is, essentially, the same as for the equivalent length of straight tube, thus introducing no additional pressure loss. Also for abrasive fluids, a low measuring range with velocities below about 1 m/s is recommended.

Cascetta et al. (1992) gave a plot of the pressure loss of a selection of commercial flowmeters [Figure 17.10(a) for flow paths], which is reproduced in Figure 17.10(b).

Nicholson (1994) tested eight commercial 1-in. (25-mm) meters and obtained pressure drops from 1.2 to 2 bar at 4 kg/s and noted that two of the meters gave audible signs of cavitation above 4 kg/s.

17.2.8 RESPONSE TIME

Trigas and Hope (1991) tested three meters from two manufacturers for step changes in the flow rate. A time constant was defined as the time to reach 63.2% [which is $1 - (1/e)$] of its steady state value. The response time was then defined as six times this value. They found that repeatability improved as integration time increased and as flow rate increased. The problems with long time constants are

- a. that the actual flow rate will not be followed precisely and the smoothing effect may cause errors to be introduced, and
- b. that, for batch control, the integration time will essentially cause some of the batch flow to be missed at the start of the batch and will therefore stop the batch too late, allowing additional batch product.

Paik et al. (1990) found, for a standing start and stop system, a time lag for the meter of order 2 s.

17.2.9 ZERO DRIFT

Keita's (1989a, 1989b) computational work suggested that damping due to dissipation, friction, etc., and nonsymmetry of the tubes were contributory causes of zero drift. He also appeared to demonstrate that lack of symmetry between the tube ends can lead to zero errors in water of up to 0.04%. He showed that fluid changes due to density and viscosity can cause zero shifts of over 0.2%. Cascetta et al. (1992) also attributed sensitivity to zero drift to the unavoidable lack of symmetry between the two sensor tubes but also to an unequal split of fluid, which is not homogeneous.

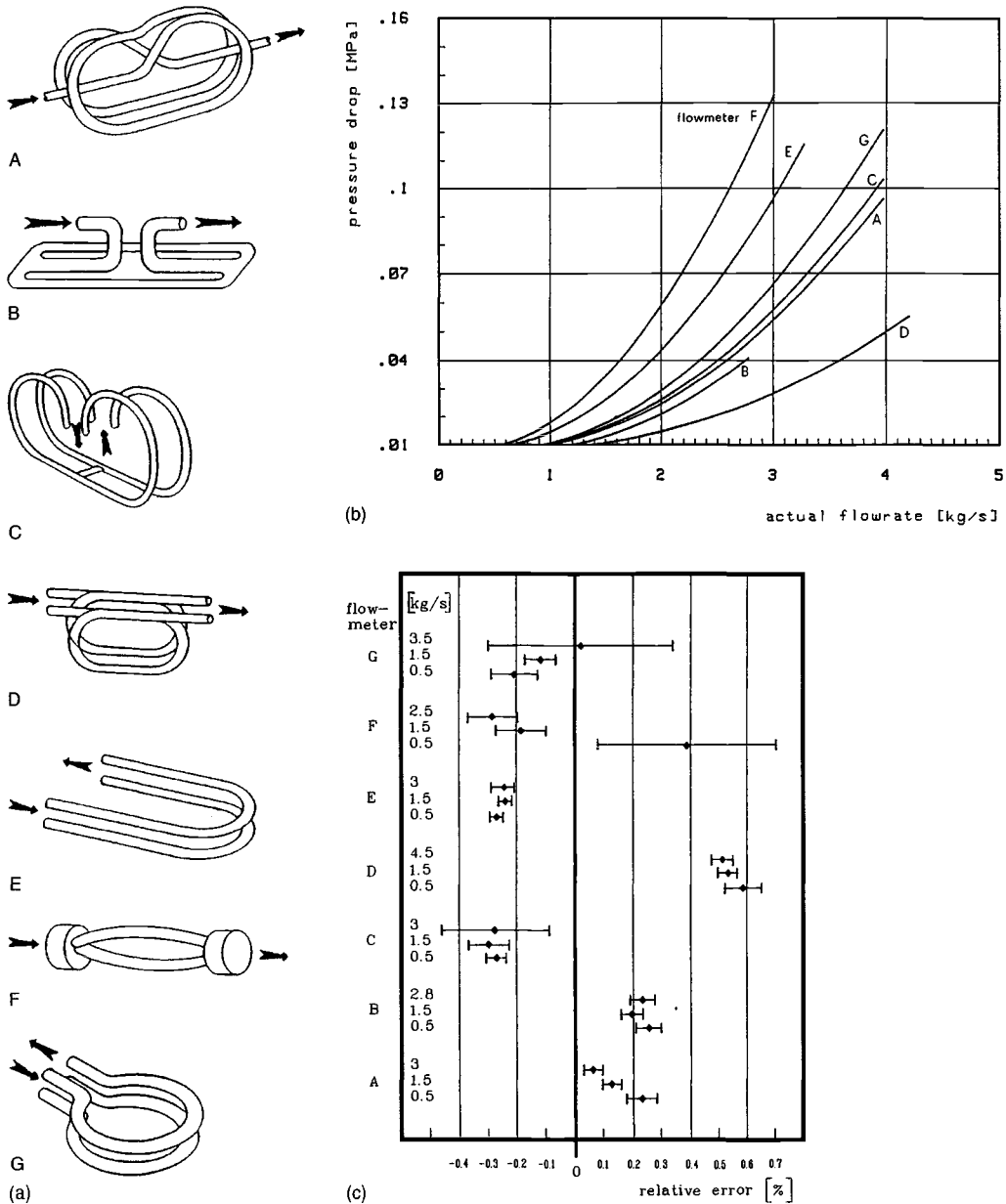


Figure 17.10. Cascetta et al. (1992) plot of data for a selection of commercial flowmeters in a 25 mm test line (reproduced with permission from the Institute of Measurement and Control): (a) Flow paths of the meters A–G; (b) Pressure loss; (c) Accuracy and repeatability of the meters.

A zero shift error causes the uncertainty to be given as of rate at the high ranges and of span at the lower ranges (Ginesi and Annarummo 1994). Zero should be adjusted after installation and fitting and may need adjustment for temperature shifts of 10°C or more. Downsizing may be used with caution to increase velocities and reduce coating as well as to reduce costs, but beware of corrosion, erosion, and pressure drop.

Eide and Gwaspari (1996) aimed to test six different makes of Coriolis meter on water, naphtha, and two different crude oils. In each, there was a five-point calibration. The zero error for most of the meters for most of the time was less than $\pm 0.3\%$ FSD and appeared not to be influenced by increasing temperature and pressure.

17.3 ACCURACY UNDER NORMAL OPERATION

Typical performance for a 20:1 turndown for commercial instruments is claimed as of order or better than:

Uncertainty	$\pm 0.25\%$ rate + zero stability
Zero stability	$\pm 0.02\%$ FSD
Repeatability	$\pm 0.1\%$ rate + zero stability/2

For 100:1 turndown, the uncertainty is likely to be ± 0.5 – 1.0% .

Mencke (1996) reported errors for a large number of Coriolis designs ranging from 2 to 250 mm on various fluids including some non-Newtonian (liquid color) and two component (liquid chalk; water with 76% solids) and obtained for:

100–50% flow range	± 0.01 – 0.35%
50–10% flow range	± 0.01 – 0.50%
10–1% flow range (three meters)	± 0.11 – 0.37%

Uncertainty according to Harrie (1991) is 0.25% on mass and $\pm 0.0002 \text{ g/cm}^3$ on density. Repeatability is given as $\pm 0.05\%$ on mass and 0.0001 g/cm^3 on density. Harrie claimed a turndown ratio of 20:1. The maximum operating pressure was about 145 bar, and the temperature range was -50 to 110°C . He gave typical results on zero offset due to pressure, temperature, and density as:

0–40 bar	0.025% FSD
6.7 – 30°C	0.05% FSD
Air to water	0.05% FSD

Cascetta et al.'s work (1992) provided some very interesting comparisons of the performance of seven commercially available meters [Figure 17.10(a)]. Figure 17.10(c) shows the bias and approximate 95% confidence band (two standard deviations) of the meters for three flow rates as first received. Two points are noteworthy.

- i. Some manufacturers appeared to have significant bias errors on their calibration facilities.
- ii. Three of the makes had serious random errors, and two of these were at full-scale readings.

Cascetta et al. then adjusted the calibration constants and recalibrated, and the result is that for A, B, E, and to a lesser extent D, the calibrations are within $\pm 0.1\%$ over a 4:1 turndown ratio. E achieved $\pm 0.05\%$ for 7:1. F achieved about $\pm 0.2\%$ for about 4:1 turndown, and C and G were within about $\pm 0.4\%$ for 4:1 turndown. It should, however, be remembered that the meters to which these tests refer were probably designed in the mid to late 1980s and will very probably have been upgraded

by now to an improved level of performance. Mencke (1989) appears to confirm the variation found by Cascetta et al. for the performance of these meters.

Nicholson (1994) tested eight commercial designs and found that only one meter was within specification for all test phases, whereas two more were within specification after initial problems. He considered that two others could meet specification after further development, but the three remaining were very erratic.

In tests of six different makes of Coriolis meter on water, naphtha, and two different crude oils (Eide and Gwaspari 1996), the general performance appeared to be of order $\pm 0.5\%$ for most meters but was quite sensitive to pressure and temperature.

Keita (1990) discussed the effect of gas compressibility, which led to a loose coupling between fluid and vibrating pipe, and hence to the possibility of errors. As with densitometers for gases, a correction for gas mass flows may be required.

Nicholson's (1994) tests on gas were generally encouraging, suggesting that at low pressure (ambient to 15 bar) two out of three meters performed with linearity and repeatability within about 1% of liquid value, despite a 10% calibration shift. In the high pressure tests (60 bar), linearity was about 1.5%, and repeatability was within about 0.3% of liquid values. Zero setting at varying pressures had a small effect, and one meter in both high and low tests was unsatisfactory.

Pawlas and Patten (1995) claimed that the Coriolis mass flowmeter could provide a solution for measuring the mass flow rate of gases directly. An increase of 100°C for a stainless steel tube causes a decrease of Young's modulus and the stiffness of the tube and causes about 5% change in the flow signal. A temperature measurement allows a compensation to be made. Elevated pressures cause the tube to stiffen due to the radial stress. The effect ranges from negligible for small sensors to about 0.08% per 100 psi for larger sensors. Compensation is usually by a bias in the calibration constant for larger sensors.

As an example, they suggested the application to custody transfer for natural gas. A vehicle-filling cycle starts at up to 50 lb/min (23 kg/min) with supply pressure of 3,500 psi (240 bar) and the vehicle pressure of zero gauge and continues for about 5 minutes or more at a decreasing rate until the vehicle is filled. The meters tested had errors in the range ± 0.2 –1.8%. At high mass flow rates, the meter cannot cope due to the high flow noise. Their plots suggested that water and compressed air gave similar results (air within about $\pm 1\%$ of the water), and that similar changes resulted from pressure change up to 1,450 psia (100 bar). They suggested that, using water calibration, the performance on air is better than $\pm 2\%$ and in much of the operation within $\pm 0.5\%$, and over wide flow and pressure ranges.

17.4 PERFORMANCE IN TWO-COMPONENT FLOWS

The meter may be suitable for homogeneous two-phase flows and for heterogeneous flows provided the phases are of similar density so that when vibrated the two phases behave as if rigidly connected. Hemp and Sultan (1989) have addressed this question theoretically as mentioned in Appendix 17.A.3. Nicholson (1994) found that as little as 2% air by volume in liquid gave errors ranging from 3 to 58% with two of the eight meters he tested ceasing to operate.

Agarwal and Turgeon (1984) found that operating temperature and specific gravity affected the early versions they were using, but despite this they appear to claim deviations on solvent and coal slurry (39% solids) within -2.7 to 4.8% .

17.4.1 AIR-LIQUID

Grumski and Bajura (1984) reported tests on single-tube and twin-tube MicroMotion meters in air-water and air-glycol two-component flows. The mass flow rate of air was considered negligible compared with the liquid, but the void fraction and flow regime were recorded. The single tube exceeded a 2% change of reading with 1.5% void fraction, whereas the twin tube could accept 7.5%. However, with an inlet causing a more homogeneously mixed flow, void fractions of 20% were obtained with signal change not exceeding 2%. Grumski and Bajura suggested that more and smaller tubes will reduce the damping effect of the air and improve the meter performance.

Birker (1989) gave a plot showing the effect of air in liquid, which suggested that above about 0.5% air-in-water by volume the error in reading rose until by about 4% it had reached about 10% error. Birker appears to have suggested that this was due to a nonuniform distribution of phases in the meter.

17.4.2 SAND IN WATER

Measurement of flows of water with sand particles of predominantly 0.125 mm (80%) and also 0.250 mm (15%), 0.063 mm (4%), and 0.355 mm (1%) were made using a MicroMotion meter (Dominick et al. 1987). It was found that, for a homogeneous steady flow with loading from 3 to 22% of sand, the measurements were within 1% of the reference values. Mass flows were up to about 140 kg/min.

17.4.3 PULVERIZED COAL IN NITROGEN

A MicroMotion meter with a U-tube of 12.7 mm (1/2 in.) and a flow range of 0–200 kg/min (0–1.5 lb/s) was operated on nitrogen at 7.5 bar, and the outputs for mass flow rate and fluid density were recorded (Baucom 1979). The meter was initially calibrated using water, and it was then compared with load cell measurements of coal hopper weight loss. This latter measurement did not allow a high accuracy. The metered flow rates agreed well with the load cell measurements except where slugging of the coal and transport gas caused erratic meter behavior. The meter behaved best for dense phase coal flows, but for low coal-to-gas ratios, the erratic behavior occurred.

17.4.4 WATER-IN-OIL MEASUREMENT

de Kraker's work (1989) gave early suggestion that by using both mass flow and density the water cut in oil production could be obtained. However, Liu and Revus (1988, cf. Liu et al. 1986) have given some impressive results of Chevron's water cut measurement for a wide range of water cuts. The claims appear to suggest results within about ± 1 to 2% of actual water cut. Young (1990) gives a plot of water content measured by an EXAC meter, which suggests a low reading at 50% water cut of up to about 10%. He also used it to obtain percentage of solids in liquid.

17.5 INDUSTRIAL EXPERIENCE

Two important papers were published at the North Sea Metering Workshop in 1988. These represent some of the earliest available field data for operation of these meters in North Sea applications. Occidental (Lawson 1988) installed a MicroMotion D150 meter, rated at ANSI 1500, on the Claymore condensate system. The meter calibration was checked on a SIRA 1.5-in. NB line with 15D upstream using kerosene and gas oil at 30°C. The uncertainty in the calibration was estimated as $\pm 0.2\%$ with a 95% confidence level. The meter was then installed on Claymore in a vertical downward flow with about 4D between the preceding upstream bend and the flange and a similar spacing after the meter. There appeared to be a discrepancy of about 0.35% between the meter and an orifice plate. A final calibration of the meter showed an underreading of about 0.5%. Occidental concluded that more work needed to be done on density measurement.

Chevron (Dean 1988a) installed an EXAC 2-in. class 1500 meter on the Ninian Central Platform on a liquified petroleum gas (LPG) flow. This was in series with a 3-in. turbine meter and a density meter. A bend was immediately upstream of the meter. The main problem appears to have arisen from zero errors, and the resultant data were not very precise. Despite this Dean (1988b) saw advantages for petroleum liquids. Coriolis meters have been used to improve crude oil measurement at individual wells throughout the Little Knife field (Liu et al. 1986). After 5 years of operation, they had proved a significant improvement over conventional flowmeters. Good agreement was achieved consistently with custody transfer meters.

These results should be compared with Rezende and Apple's (1997) claims, 10 years later, that with proper zeroing of the Coriolis meter, the greatest variation from a turbine meter was 0.12%.

Communication between two Coriolis meters can be overcome by using flexible pipework or by separating the operating frequencies (Kiehl and Gartner 1989). Data on a 6-in. MicroMotion meter (McKenzie 1989) reported by the manufacturer suggested linearity of ± 0.15 – 0.25% with repeatability of $\pm 0.05\%$ or better. Frankvoort (1989) gave results of tests of the Endress and Hauser meter for temperature, pressure, and viscosity change and for the effect of a pump running. The data seem to suggest that calibration changes of 0.3% or more are possible as a result of temperature, pressure, and fluid change.

Davis (1990) described tests of a MicroMotion 25-mm meter, which operated for 15 months on the distillate export facility at St Fergus gas terminal. Davis considered that, with in situ proving, a Coriolis system with an overall mass measurement uncertainty of 0.1% was possible. The density measurements from the meter also compared to within 0.1% with those from an independent density measurement.

Grendstad et al. (1991) tested three MicroMotion meters and one Schlumberger meter. The reference flowmeter was a 3-in. Brooks turbine meter calibrated using a compact prover and having a linearity within a 10:1 range of $\pm 0.15\%$. The liquid density was measured to $\pm 0.3\%$ or better for the reference test fluid *N*-butane. Some problems were encountered due to zero setting, stress caused by distortion between flanges, and cross-talk. The meters were then tested on natural gas against sonic nozzles. Cross-talk only gave clear effects for one meter. The results indicated

no effect from an upstream bend. However changing the clamp tightness caused a shift in some meters of 2%. The tests indicated the importance of correct zeroing. Two of the meters were within the $\pm 1\%$ of rate, which was the aim of the tests.

Arasi (1989) found that field performance of a Coriolis meter on an 80% ethane 20% propane mixture resulted in errors ranging from +1 to -2% or more, despite good results on a water calibration, and concluded that for field service regular proving, as for turbine or positive displacement meters, would be necessary.

Myhr (1991) used propane and propylene in tests of the mass meter in series with a turbine meter. Readings 10% lower than the turbine meters were obtained, although the linearity was within $\pm 0.2\%$ for a 10:1 turndown. Stable performance was experienced. Myhr casts doubt on the transferability of a water calibration and suggests that the meters are sensitive to changes in pressure, density, and viscosity.

Erdal and Cabrol (1991) found that a 1.5-in. (38-mm) Coriolis meter on natural gas had a repeatability over a year of 0.47%, and more recent tests by Withers et al. (1996) for meters offshore resulted in performance within 0.4% of water calibration.

17.6 CALIBRATION

Paik et al. (1990) described results from a gravimetric calibration system allowing both standing and flying start and stop. Their claims for the accuracy of the system and for the performance of a Coriolis flowmeter were high.

Mencke (1989) described the process for obtaining limited approval in Germany in the light of the weights and measures regulations for new techniques of flow measurement. This involved test facility calibration and site tests following limited approval. Strawn (1991b) reviewed proving methods and listed test sites approved by the American Petroleum Institute.

Hayward and Furness (1989) reported on the development of a new gravimetric prover based on the substitution method of dynamic weighing but using a bottom entry pipe and a sealed tank. Repeatability of the prototype was not as good as expected.

Rivetti et al. (1989) have described a rig that was in part for their evaluation of Coriolis meters. It had a range of 0.2–7 kg/s at 0.1 MPa or 0.2–5 kg/s at 0.5 MPa, with a usual uncertainty of $\pm 0.1\%$.

Grini et al. (1994) described a calibration system for Coriolis gas flowmeters that depended on measuring the mass change in a gas bottle of nitrogen or some other gas. They claimed an accuracy of better than 0.02%. Standard deviations for the two makes of Coriolis meter tested on nitrogen were 0.09 and 0.15%.

17.7 APPLICATIONS, ADVANTAGES, DISADVANTAGES, AND COST CONSIDERATIONS

17.7.1 APPLICATIONS

Furness in 1988 (cf. 1989) claimed that, in the previous 5 years, the market segment had gone from virtually nothing to over 5% (by sales volume) and was still

Table 17.1. Compatibility of 316L stainless steel (from MicroMotion user information)

Compatible Fluids	
Adhesives	Asphalt
Beer/beer foam	CNG
Ethylene oxide	Fatty acids
Fruit juices	Fuel oils
Isopropanol	Lime slurries
Liquified gases N ₂ , O ₂ , LPG	Magnetic slurries
Milk/cream	Molten sulphur
Nitric acid	Olefins
Paint	Peanut butter
Phosphoric acid	Pie fillings
Polymers	Polypropylene
Potassium hydroxide	Sodium hydroxide
Sour crude	Tar sands
Urethane	
Incompatible Fluids	
Acetic acid (high concentrations and elevated temperatures)	
Ammonium chloride	
Bromine	
Calcium chloride	
Iodine (other than 100% dry)	
Mustard	
Sea water	
Dyes and inks in some cases	
Fluids containing halogen ions	

increasing. Walker in 1992 assessed Coriolis market share as 9% and predicted that it would be over 16% by 1996. Hall in 1990 claimed that more than 75,000 Coriolis meters were in use, measuring flow rates in food, pharmaceutical, chemical, paper, petroleum, and petrochemical industries.

One manufacturer claimed that over 100,000 meters have been installed and applied to flow measurement in polymer/monomer, ethylene oxide, milk, fruit juice and concentrates, liquid chocolate, processed egg, coffee extract, peanut butter, animal and vegetable fats, vegetable oil, titanium dioxide, asphalt, fuel oil, and natural gas. Table 17.1 and Figure 17.11 give some of the constraints due to material compatibility. Grob (1984) confirmed the meter's wide applicability but added little specific information. Agarwal and Turgeon (1984) described the use of early meters in coal liquefaction plant including sour water, solvent, slurry, and high temperature solvent. The meter operated successfully on coal liquids with a high (39%) solids content. Blumenthal (1985) listed application of the MicroMotion meter to the pulp and paper industry: green liquor, black liquor, coating of filler slurries, dyes, fuel oils, tall oil, and shear sensitive fluids.

In some designs, care may be needed in applications involving liquids with high vapor pressure (low boiling points) such as hydrocarbons like propane and

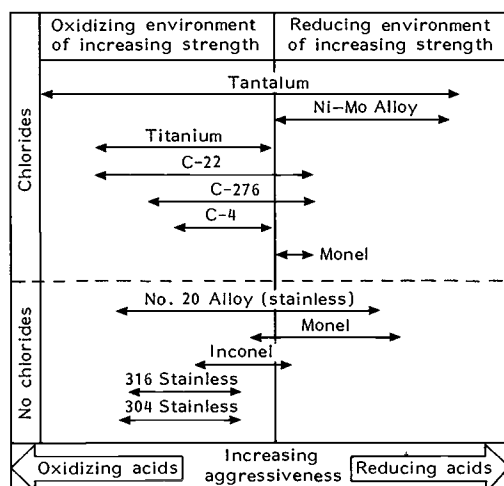


Figure 17.11. Compatibility of materials from Micro-Motion (reproduced with permission).

butane, solvents, liquified gases (e.g., CO_2), and liquids that tend to produce gas. Such liquids might start to gas and form cavities that could lead to measurement errors, and so in applying the meter it will be important to check with the manufacturer that the operating conditions are suitable. This caution is even more important in the case of multiphase and multicomponent flows.

Benabdelkarim and Galiana (1991) claimed that this meter provided an excellent density measurement of cement slurry. The combination of mass flow and density is likely to allow applications such as water-in-oil measurement in crude oil pipelines. Taylor and Nuttall (1993) indicated the benefits of these meters to obtain on-line well test data, and Sims (1992) considered that

they had revolutionized blending and produced reductions in operating costs. Chateau (1991) cites the value of the instrument for the batching of binders in a mixing plant where an accurate knowledge of the mass delivery of the binder is required.

Where density is known, or the liquid is of low value, or repeatability is more important than accuracy, the Coriolis meter is less competitive (Ginesi and Annarummo 1994). However, the ability to deduce not only mass flow, temperature, and density, but also, from these, percent solids and other data, makes the meter particularly useful. Two-phase fluids can cause problems, and single-tube meters may be preferable. In either design, the largest particles must pass through the meter. Pitting may result and can cause problems due to thinner walls and cycling fatigue. The meter should remain full and preferably be in a vertical upward flow. All air must be bled or purged before zeroing, and the piping should avoid air collection. The pump should also be running when zeroing is done. The manufacturer's instructions should be followed, particularly on support, and the meter should be supported by adjacent pipework and not vice versa. No supports should be in contact with the flow measuring tube. No upstream flow effects are expected.

17.7.2 ADVANTAGES

Davis (1990) commented that Total Oil Marine planned to continue the use of Coriolis meters in fiscal measurement applications because they offered significant capital and operating cost savings over equivalent turbine meter systems. In 1993 MicroMotion successfully proved two sizes, which are being used on fiscal duties. They have also been used as transfer standards. (However, in 1991 Hannisdal considered that Coriolis meters were not, then, an alternative for fiscal measurement.)

One manufacturer suggests the following advantages:

- Obstructionless,
- No bearings,
- No probes,

Compact design (space saving),
Low power consumption,
Suitability for steam cleaning,
Suitability for bidirectional flow,
Low maintenance, and
Sanitary design.

Cascetta et al. (1992) added to this list their wide range and high accuracy, comparable to that of the best volumetric flowmeters, and Cascetta and Vigo (1988) added true mass flow measurement of all liquids, slurries, and foams.

17.7.3 DISADVANTAGES

Robinson (1986) recorded some of the early limitations (e.g., reduced performance above about 120°C) and problems (e.g., tube failure) from a survey of process industry users. He also remarked on the problems of cleaning due to pipe configuration. Cascetta et al. (1989a) also suggested other limitations such as difficult maintenance (cleaning and repairing) in the case of clogging of sensor tubes when difficult fluids such as slurries are used; limited temperature range and incomplete temperature compensation; and the effects of stress, creep, etc. In addition, Cascetta and Vigo (1988) listed high pressure loss, high initial cost, spring constant temperature sensitivity, suitability for only very high pressure gases, and calibration needed if density of liquid differs substantially from that of calibration fluid.

They may, also, be in competition with ultrasonic multipath meters for volumetric custody transfer for larger (greater than 100 mm) sizes.

Walker (1992) suggested that the meters still had limitations for some hazardous environments, might need secondary containment, and might be limited by pressure drop and sensitivity to changes in temperature and pressure and to pipe vibration. He also identified cost as a limitation. He was optimistic about overcoming many of these limitations.

17.7.4 COST CONSIDERATIONS

The initial cost of these meters appears to be significantly greater than most other meter technologies. It is therefore necessary for the prospective user to assess the appropriateness of the meter for a specific application in terms of

- a. initial instrument cost compared with value of product being metered;
- b. cost of installation including any necessary pipework changes in orientation;
- c. cost of operation, maintenance, and regular servicing;
- d. cost of recalibration including
 - the adequacy of in situ methods and
 - the cost of removal and test stand calibration, as well as any increased uncertainty between test stand calibration and plant performance.

Liu et al. (1986), from oil field experience, found significant capital and operating cost savings compared with a more conventional approach, and this appears to be confirmed by Robinson's (1986) survey, which suggested that, after 3 years these meters are more economical than positive displacement meters.

17.8 CHAPTER CONCLUSIONS

This chapter was initially written prior to 1994, but material that I have added suggests that the commercial developments have moved ahead of the published material on test programs.

The meter lends itself to the use of sophisticated computer models that take in the fine detail of the construction and allow for fluid behavior, compressibility, homogeneity, and other features. As suggested in Chapter 20, the optimization of this model should also introduce factors relating to the precision of the manufacturing options because these are likely to have an influence on final instrument performance. Combined with new materials, this should lead to production methods that provide an increasingly high quality product with wide applicability.

These meters are, clearly, most at home with single-phase liquid flows, and we should expect to see uncertainty of the $\pm 0.05\%$ level claimed increasingly widely for 30:1 turndown or greater.

For gases, the scene does not appear so clear. On the one hand, the meter is likely to be in competition with ultrasonic and thermal techniques. Even though neither of these offers direct mass flow, both approach it and are likely to be considerably less costly. On the other hand, there are various questions raised in the literature about the accuracy of a gas Coriolis meter. Effects due to compressibility and other features of the gas could mean that, for the highest accuracy, the meter becomes sensitive to type of gas.

For two or more components in the flow, there are clear problems in the use of these meters, although claims are made for their use in certain slurries and other flows. Fundamentally, this is due to the possibility that the components can move relative to each other. Relative motion should be essentially eliminated if the phases are of similar density.

The meter has developed from the U-tube designs that have been so successful, through various configurations to the very significant development of the single-tube meter (cf. Stansfeld et al. 1988 and Harrie 1991 for details of an early design that appears to have been withdrawn). Krohne's single straight tube design (Figure 17.12; Hussain and Farrant 1994), which also incorporates a compensation cylinder in a floating mode, represents a notable achievement that is already being followed by other similar designs (Rieder and Drahm 1996, Yamashita 1996). This trend is very likely to continue, and it should offer a very significant addition to the flowmeter options for new applications.

What will be the next major development? Some of the recent work suggests that the single-tube vibration can be in pipe cross-section rather than in longitudinal bending. Could the next stage be shorter pipes or multiple sensors? Is there a need to return to the vibrating elements within tubes (Hemp 1994b) or traveling wave devices as has been suggested in the past? Could tubes be set into torsional motion with internal segmental arrangements – a cross between the angular momentum devices and these?

Matthews and Ayling (1992) described a meter that had tines, rather like an elongated tuning fork, which stretched the length of the meter. The Coriolis principle is still valid, and Hemp's work on the oscillating element meter is presumably

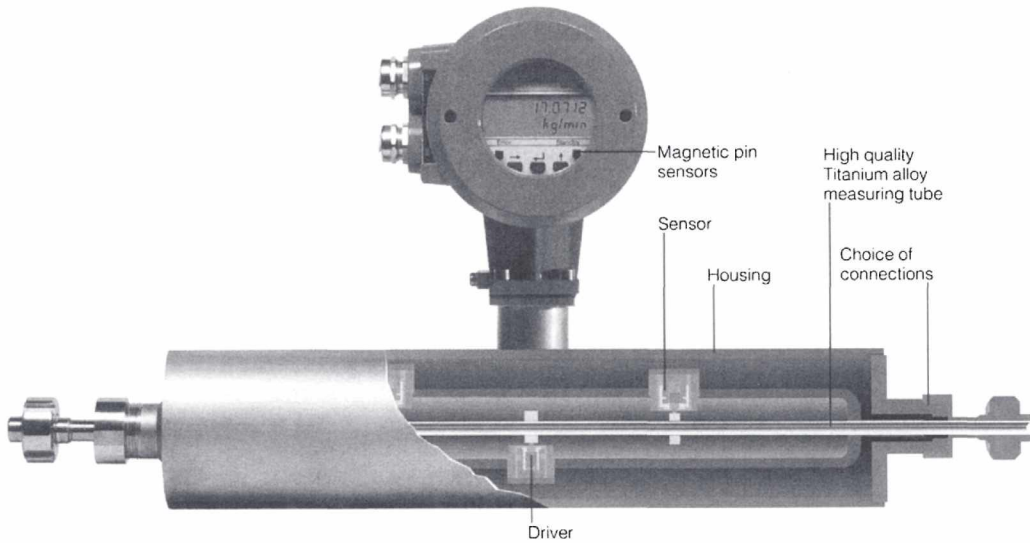


Figure 17.12. Single straight tube Coriolis meter (reproduced with permission of Krohne Ltd.).

relevant. The meter has, as a result, an essentially straight spool piece with the tines projecting into it. The initial work was on a 4-in. (100-mm) meter, and the plans are that larger sizes, capable of use on refined single-phase liquids and gases, should be developed.

Possibly the most significant way in which this meter will develop is in its price. Manufacturers often seem unwilling to quote precise prices. They are certainly high at present. However, increasing competition and improved manufacturing techniques may well bring them down to a level competitive with other types of flowmeter providing high precision liquid measurements.

APPENDIX 17.A

A Brief Note on the Theory of Coriolis Meters

17.A.1 SIMPLE THEORY

If the flow velocity is V (Figure 17.4), and the angular velocity caused by the vibration is Ω upward, then a mass of δm at a radius r will experience an upward velocity $r\Omega$, which increases at $r + \delta r$ to $(r + \delta r)\Omega$. The angular momentum of the mass δm will change from $r^2\delta m\Omega$ to $(r + \delta r)^2\delta m\Omega$, a change of $2r\delta r\delta m\Omega$, neglecting δr^2 . If δm is moving at velocity V , it takes $\delta r/V$ seconds, and so the pipe will experience a force downward due to the Coriolis acceleration of

$$F = 2\Omega V\delta m \quad (17.A.1)$$

The other half of the tube will experience an equal and opposite force. Because the mass of the tube depends on the fluid density $\delta m = \rho A \delta r'$, the force on each half of the tube due to length $\delta r'$ will be equal to $2\Omega \rho A V \delta r'$. Taking the width of the U-tube (or the unwrapped length between sensors) as d , the twisting (or distorting) torque becomes

$$T = 2K\Omega\rho AV dl \quad (17.A.2)$$

where l is the length of each half of the tube, and K allows for the fact that the distorting Coriolis forces will not form a straight integration. The mass flow is $q_m = \rho AV$, and the torque can be related to the mass flow rate by

$$q_m = T/(2K\Omega dl) \quad (17.A.3)$$

We can now introduce the oscillating motion by putting $\Omega = \Omega_0 \cos \omega t$, where ω is the driving frequency, and t is time; hence,

$$T = 2K\Omega_0\rho AV dl \cos \omega t \quad (17.A.4)$$

If, for the special case of the U-tube, we relate the twisting torque to the twist in the tube θ (ignoring damping effects etc.) by

$$I_s \frac{d^2\theta}{dt^2} + K_s\theta = T \quad (17.A.5)$$

where I_s is the inertia, and K_s is the spring constant of the U-tube in twisting oscillation, and assume a solution of the form $\theta = \theta_0 \cos \omega t$, we obtain an expression for θ_0 by equating T in Equations (17.A.4) and (17.A.5)

$$\theta_0 = \frac{2K\Omega_0\rho AV dl}{K_s - I_s\omega^2} \quad (17.A.6)$$

where θ will be in phase with Ω . It is then apparent that, while the twisting torque is at its maximum at the midpoint, the twist is zero at the extremities of the oscillation. This is illustrated by Blumenthal (1984) and shown in Figure 17.5. The angle of twist is of order 1/100 degrees.

The free vibration frequency in twisting occurs when $K_s = I_s\omega^2$ or

$$\omega_s = \sqrt{K_s/I_s} \quad (17.A.7)$$

We can now obtain a relationship between q_m and θ_0

$$q_m = \frac{K_s(1 - \omega^2/\omega_s^2)\theta_0}{2K\Omega_0 l d} \quad (17.A.8)$$

(Damping would prevent the bracket in the numerator from becoming zero when $\omega = \omega_s$.) We can go a step farther and relate the twist θ_0 to the time difference for transit of the two sides of the U-tube. Because the velocity of the U-tube at the sensor is approximately $\Omega_0 l$, and the displacement due to the twist is $\theta_0 d$, the time of transit

of the two sides will differ by

$$\tau = \theta_0 d / (\Omega_0 l) \quad (17.A.9)$$

Hence the mass flow is given in terms of the transit time difference by

$$q_m = \frac{K_s \tau (1 - \omega^2 / \omega_s^2)}{2Kd^2} \quad (17.A.10)$$

A similar equation will give the relationship for a straight tube. It will be noted that the actual value of angular velocity Ω has been eliminated from this expression. The amplitude of the vibration is typically between $60 \mu\text{m}$ and 1 mm , and the frequency is in the range $80\text{--}1,100 \text{ Hz}$. The higher frequency tends to be well above most common mechanical vibrations. In most meters, two tubes oscillate in antiphase acting like a tuning fork. When flow occurs, the forces created result in phase shifts in the motion of the two halves of the oscillating tube, and the passage of the halves of the tube past fixed points will therefore be displaced in time, and the time difference will be measurable.

An equation similar to Equation (17.A.6) can also be used for the oscillation without flow, and for this the natural frequency of the tube will be given by

$$\omega_u = \sqrt{K_u / I_u} \quad (17.A.11)$$

where K_u is the spring constant of the tube in normal oscillation, and I_u is the inertia in that plane. Because I_u is proportional to the mass of the tube and therefore related to the density of the fluid, the natural frequency can be used to obtain density of the fluid. In particular, the frequency is little affected by flow so that the density can be obtained with flow (Raszillier and Raszillier 1991).

17.A.2 NOTE ON HEMP'S WEIGHT VECTOR THEORY

Hemp (1994b) has developed a weight vector theory for the Coriolis meter [cf. appendices in electromagnetic (Section 12.A.3), ultrasonic (Section 13.A.3), and thermal (Section 15.A.5) chapters]. The flowmeter signal is given by

$$\Delta\phi = \int \mathbf{v} \cdot \mathbf{w} dv \quad (17.A.12)$$

where $\Delta\phi$ is the phase difference between the total velocities at the two sensing points. The weight vector is then given by

$$\mathbf{w} = -\rho \left[(\mathbf{v}^{(2)} \cdot \nabla) \mathbf{v}^{(1)} - (\mathbf{v}^{(1)} \cdot \nabla) \mathbf{v}^{(2)} \right] \quad (17.A.13)$$

where $\mathbf{v}^{(1)}$ and $\mathbf{v}^{(2)}$ are oscillatory velocity fields set up in the stationary fluid by the driving transducer and by equal and opposite unit alternating forces applied at the sensing points. (See the last paragraph of the fourth section of Hemp 1994b.)

Hemp commented that the theory had been applied at that date (1994b) only to two relatively simple flowmeter configurations. The first was for a U-tube meter

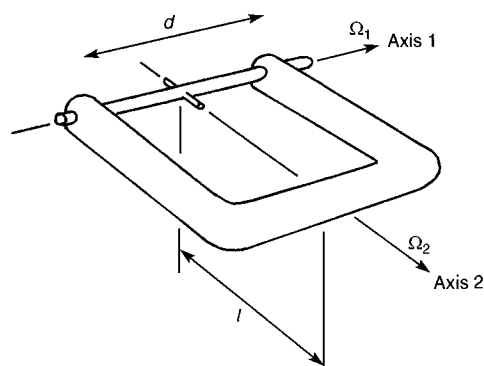


Figure 17.A.1. Geometry of Hemp's U-tube meter (after Hemp 1994b).

(Figure 17.A.1), where the tube was rigid and rotation took place about two shafts: one at the "top" of the U (axis 1) and the other "vertically" through the center of the U and parallel with the sides of the U (axis 2). The oscillatory velocity fields $\mathbf{v}^{(1)}$ and $\mathbf{v}^{(2)}$ were, respectively, those that accompany a rotation of angular velocity Ω_1 about axis 1 due to the driving force and a rotation of angular velocity Ω_2 about axis 2 due to forces of unit size applied to the sensing points and in the direction of the Coriolis forces.

Hemp obtained the value of $\mathbf{W}$ as $\pm \frac{1}{2} \rho d \Omega_1 \Omega_2$ for the "sides" of the U-tube and parallel with the sides, and as $-\rho l \Omega_1 \Omega_2$ along the straight "bottom" of the U and in the same direction as the rotational vector. Hemp commented that because $\mathbf{W}$ is constant and parallel to the tube axis in each straight section and in the direction of flow, perfect averaging of the flow is achieved. The effect of flow in the bends had not been calculated.

The second configuration was for a rigid elliptical cylinder that might be mounted across the diameter of a pipe. Hemp and Hendry confirmed that the weight vector for a vibrating element flow sensor of this sort would mainly be sensitive to flow near the element and, in a pipe, would be prone to velocity profile effects.

Hemp also stated that the theory was limited to small amplitude vibrations. He suggested that, for short vibrating tube Coriolis meters and in vibrating element sensors, the assumption was probably adequate, but that, for traditional meters with long tubes, amplitudes could be of order one-tenth of a tube diameter and might introduce errors into the theory. The theory may also need to be extended to allow for compressibility and turbulence.

As with all Hemp's theories, this is very elegant. It appears to confirm, for the U-tube meter, that installation effects resulting from profile distortion (as opposed to vibration, very high turbulence, or pulsation) are likely to be, in general, of minor concern in this type of flowmeter (cf. Hemp and Hendry 1995) but of major concern in vibrating element sensors.

17.A.3 THEORETICAL DEVELOPMENTS

Durst and Raszillier (1990) computed the perturbation of the flow in a pipe that is rotating and considered the relevance of flow field and forces that result to Coriolis meters. They produced very interesting 3-D plots of the flow perturbation and raised the possibility that the secondary flows might be a source of asymmetry in the pipe oscillations. Raszillier and Durst (1991) derived the equation of motion for a straight-tube meter, whereas Sultan and Hemp (1989) developed a theory for a U-tube meter. They obtained impressive agreement with experimental data, although they

indicated that there were various limitations of the theory and that further areas of work were still needed. Raszillier and Raszillier (1991) used dimensional arguments to show the possible interdependence of velocity and density. John Hemp (private communication July 1994) commented that the very useful results of this reference include general expressions for time delay and frequency, showing how they depend on various meter parameters such as Young's modulus and length, and estimates of the order of effects of fluid density on time delay and natural frequency. Raszillier et al. (1993) suggested that, by influencing the vibration spectrum, the sensitivity of the instrument may be increased. Raszillier et al. (1994) investigated the effect of a concentrated mass at the middle of the Coriolis pipe segment as required for the purpose of symmetric excitation of the vibration. The flowmeter factor was found to be almost independent of the mass up to fairly high values compared with the mass of the fluid-filled pipe segment, although the frequency of the fundamental mode is strongly influenced. Lange et al. (1994) examined the effect of detector masses on the calibration of Coriolis flowmeters. They showed that the position of the detectors must be chosen with care because it may have implications for the accuracy to which the calibration of the meter remains independent of the fluid density. The calibration becomes dependent on fluid density if the detector masses are not negligible compared with the pipe mass. They suggested the most appropriate positions for the first symmetric and the first antisymmetric vibration modes (cf. Sultan 1992).

Hemp's (1988) reciprocity concept for flowmeters to reduce zero drift errors may also be applied to Coriolis meters.

Watt (1991) used a simple Coriolis force model and applied it to obtain the tube distortion for an Endress and Hauser *m*-point type tube using a finite element computer analysis. His work confirmed that the highest stresses were around the midpoint where the tube is driven and at each end. He makes the point that the Coriolis distortion of the tube will modify the Coriolis forces compared with a tube in pure bending.

Hemp and Sultan (1989) explained the concept of effective mass of bubbles in a liquid and calculated the expected error. Compared with Grumski and Bajura's (1984) measurements, the theory overestimated the error possibly due to interaction between bubbles.

Pawlas and Pankratz (1994) discussed their CFD analysis of flow in Coriolis tubes and outlined their plans for an iterative approach to linking structural analysis of the rotation of the flow tube with fluid mechanics analysis to obtain a spatially varying rate of rotation and hence the pressure distribution on the tube wall, in part due to Coriolis forces.

Hemp (1996) used dimensional analysis (cf. Raszillier and Raszillier 1991) to explore the design requirements if the meter was to be used with low mass flow rates (e.g., when applied to gases near normal temperature and pressure). His findings suggest that the most appropriate type of sensing transducer in this case is the displacement transducer and that the likely requirements, keeping maximum strain unchanged and retaining similarity in layout, will be a reduction in tube diameter to one-third or so and wall thickness to about one-quarter while retaining the same length of pipe.

Keita (1994) provided a very useful discussion of the various equations used to predict the behavior and the sources of calibration shift in Coriolis meters for gas flow measurement. He concluded that the main contribution to the shift in the meter factor was due to pressure, that the compressibility effects were very small and partly compensate for the pressure effect, and that, despite the high resonance frequency of straight-tube flowmeters, they were suitable for gas flow measurement.

Probes for Local Velocity Measurement in Liquids and Gases

18.1 INTRODUCTION

We resort to probes for flow measurement for three main reasons:

- i. To provide a low cost method of flow monitoring;
- ii. To provide an in situ calibration;
- iii. To obtain fine detail of the flow in the pipe (velocity profile, swirl, and turbulence).

For our present purposes, we shall not consider (iii). It is relevant to fluid mechanics research, mainly uses hot-wire and laser doppler anemometers, and is, therefore, outside the scope of this book. We will concentrate on work relating to bulk flow measurement. In situ calibration techniques are covered in Chapter 4. In this chapter we shall consider the various devices that have been used and are commercially available either as a local flow monitor or as the probe for in situ calibration. However, before considering these, we need to understand what is being measured.

- i. Ideally the probe measures the local velocity and obtains a representative value of the mean velocity from the shape of the velocity profile. Alternatively, it may measure the complete profile.
- ii. In practice the probe may be in error because:
 - it was calibrated in an approximately uniform profile or on the pipe axis but will need to measure velocity profiles that result in a significant variation of velocity across the sensing head;
 - it will alter the flow by its presence; and
 - the flow pattern will continue to change as the probe head is inserted farther and farther into the pipe and as it approaches the opposite side of the pipe.In some cases, these flows may be analyzed theoretically, but in most cases one would expect to calibrate the probe, and this calibration will be different at the pipe axis and at the pipe wall.
- iii. The probe will reduce the flow area of the duct by its presence, and the area will continually decrease as the probe is inserted farther. This blockage will need to be allowed for. The blockage will also differ between incompressible and compressible fluids.

Our object in using a probe is to obtain the best measure of the local flow rate that can be obtained. We shall need to remember both the limitations of the probe

and the uncertainty of the relationship between the reading and the mean velocity. We also need to remember that, to deduce the mean flow rate in the duct, there is uncertainty in the measurement of the cross-section of the pipe and the blockage caused by the probe.

The variation on one probe, according to tests by the National Engineering Laboratory in Scotland, suggested that blockage at the center for pipe sizes from 250 to 750 mm ranged between about $\pm 5\%$ and the change as the probe moves from the center to the wall can be of order 10%.

For any new probe design, there is clearly a need for some careful work as to

- what velocity the probe is measuring in an ideal uniform flow;
- how the signal will change in a real, nonuniform pipe flow, and when close to the wall; and
- how the flow in the duct will be altered by the probe and how the local probe measurement would relate to the flow in the duct if no probe were present.

In Section 2.3 and, specifically, in Table 2.2, we noted that the velocity in a turbulent flow at about the 0.76 radius point (0.24 of a radius in from the wall) is practically equal to the mean flow in the pipe. So this suggests that the probe should be set at this point. However, this may result in reduced precision due to the shape of the flow profile in this region (b), the level of turbulence, and the proximity of the wall.

18.2 DIFFERENTIAL PRESSURE PROBES – PITOT PROBES

Using Equation (2.10), but introducing a factor k for nonideal conditions, we can obtain the pressure rise when flow comes to rest in the entrance of a pitot tube (Figure 18.1) as

$$\Delta p = \frac{1}{2} k \rho V^2 \quad (18.1)$$

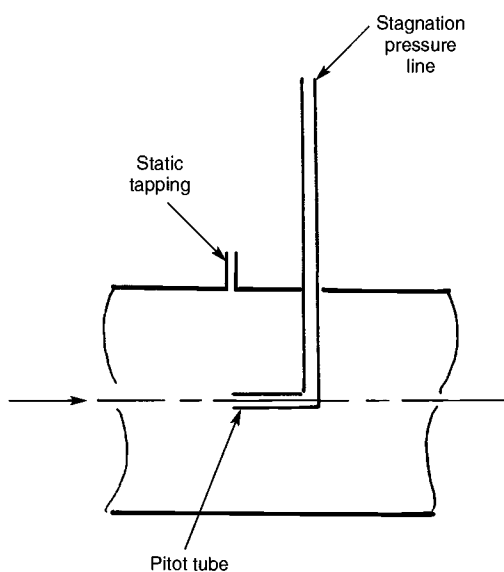


Figure 18.1. Separate pitot and static tubes.

For ideal conditions $k = 1$. Δp is the difference between the stagnation pressure at the mouth of the pitot tube and the static pressure; in this case, it was obtained with a wall tapping in the same cross-sectional plane as the mouth of the pitot tube. The static pressure at the wall tapping will be the same as the value at the pitot tube assuming that there is no cross flow in the tube. The measurement assumes that any pressure effects caused by gravity between the pitot tube entry and the wall tapping will be equal for both the total and the static measurements and will, therefore, cancel out.

Figure 18.2 shows two forms of pitot-static tube used to obtain the dynamic pressure or velocity head. Figure 18.2(a) is a

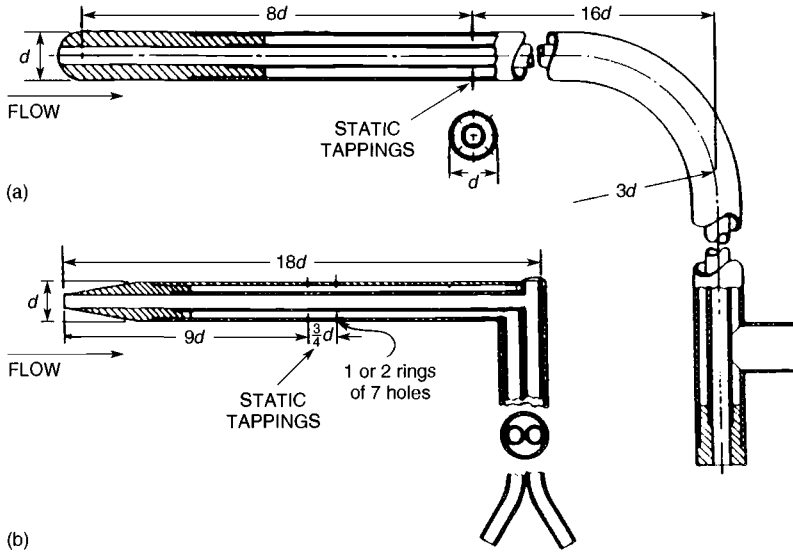


Figure 18.2. Two designs of pitot-static tube commended by Bean, both similar to NPL designs (reproduced from Bean 1971 with permission of ASME): **(a)** Round-nosed pitot-static tube; **(b)** Conical-nosed pitot-static tube.

diagram of a tube recommended by Bean (1971). A National Physical Laboratory (UK) design appears to differ in that it has an ellipsoidal nose, which is claimed to be less sensitive to lack of smoothness between the tip and the cylindrical section. Figure 18.2(b) shows an alternative design. Note in Figure 18.2 the coaxial arrangement of the tubes so that the total pressure is carried in the center of the tube and the static pressure is carried in the outer annulus. Note also the sensing holes for the static pressure. Bean (1971) recommends that d should be between about 5 and 8 mm ($\frac{3}{16}$ to $\frac{5}{16}$ in.).

At low Reynolds number based on the tube diameter (<100) when the effect of viscosity becomes important, k diverges from unity. Goldstein (1965) gave k values for conical entry and round-nosed designs in the range 1.020–1.055 at about 0.6 m/s ($Re \approx 300$), a minimum value of $k = 0.991$ at about 3–3.5 m/s (Re 1,600–1,900), and a value within about 0.1% of unity for 6–27 m/s (Re 3,000–10,000). Bean (1971) also reckoned that, for correct alignment (less than 12°), the reading should be within 0.3%. For further information, the reader is referred to ISO 3966 or Ower and Pankhurst (1966).

At high velocities in a gas, there are compressibility effects. The simple Equation (18.1) has to be replaced by one that allows for these effects. Using Equation (2.16)

$$\frac{p_0}{p} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\gamma/(\gamma-1)} \quad (2.16)$$

where $M^2 = \rho V^2/(\gamma p)$ for a perfect gas, we obtain a modified equation for V

$$V = \sqrt{\frac{2\gamma}{\gamma-1} \frac{p}{\rho} \left\{ \left(\frac{p_0}{p}\right)^{(\gamma-1)/\gamma} - 1 \right\}} \quad (18.2)$$

Bean (1971) suggested that, below about 15 m/s for air at sea level, there is negligible difference between this value and that from Equation (18.1) (with $k = 1$).

We can see this if we rewrite Equation (18.2) as

$$V^2 = \frac{2\gamma}{\gamma-1} \frac{p}{\rho} \left\{ \left(1 + \frac{\Delta p}{p} \right)^{(\gamma-1)/\gamma} - 1 \right\} \quad (18.3)$$

If we then use the binomial theorem to expand the inner bracket, we obtain

$$V^2 = \frac{2\gamma}{\gamma-1} \frac{p}{\rho} \left\{ 1 + \frac{\gamma-1}{\gamma} \frac{\Delta p}{p} - 1 \right\} \quad (18.4)$$

assuming that we can neglect terms of order $(\Delta p/p)^2$ and greater. This reduces to

$$V^2 = \frac{2\Delta p}{\rho} \quad (18.5)$$

In neglecting terms of order $(\Delta p/p)^2$ or greater powers, we have neglected terms of order $(\frac{1}{2}\rho V^2/p)^2$ or greater. If $\rho = 1.3$, $V = 15$ m/s, and $p = 10^5$ Pa (1 bar), then the error resulting from ignoring $\frac{1}{2}\rho V^2/p$ is of order 0.15%, confirming Bean's value.

One major problem with the pitot tube is the effect of unsteady flow on its reading, and because turbulence is very common in normal industrial flows, the pitot tube will tend to read high (cf. Goldstein 1936). MacMillan (1954) discussed the effects of low Reynolds number and noted (MacMillan 1957) that in a shear flow there was an apparent shift in the effective center of the pitot tube of $0.15d$ in the direction of high velocity, where d is the outside diameter of the pitot tube.

One manufacturer offers a device that is not strictly a pitot tube (Figure 18.3) but consists of a cylindrical probe with sensing at the end. The differential pressure resulting from the flow acts on a spring-loaded diaphragm, the movement of which is sensed by an inductive displacement transducer. Bean (1971) refers to designs with upstream and downstream pointing tubes. (cf. Cutler 1982)

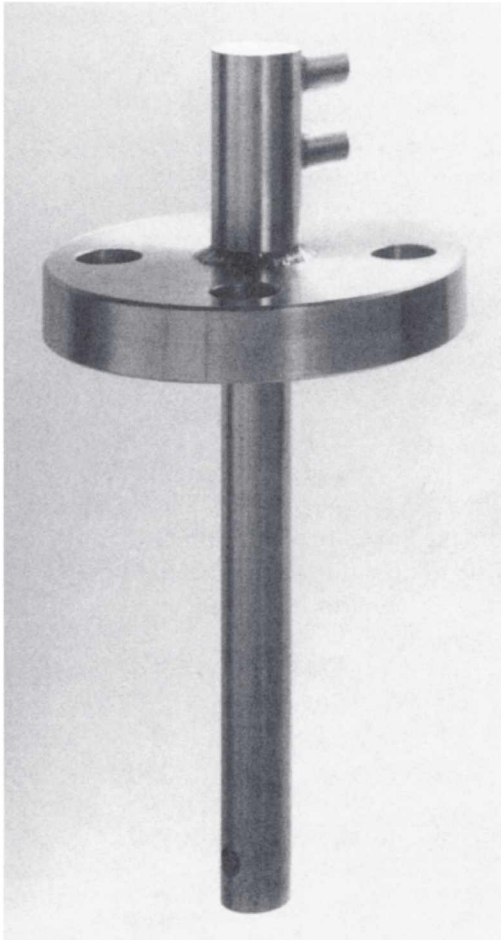


Figure 18.3. Probe-type differential pressure sensor [reproduced with permission of Laaser (UK) Ltd.].

18.3 DIFFERENTIAL PRESSURE PROBES – PITOT-VENTURI PROBES

In order to increase the pressure difference in the pitot tube in low flow rates, the pitot-venturi has been used. In this device, the total pressure is compared with a depressed pressure created by the flow at the

throat of the venturi. This is shown in Figure 18.4(a) (Bean 1971). The pressure difference can be increased even further by using a double venturi [Figure 18.4(b)]. The outer venturi causes a reduced pressure at the outlet of the inner venturi, thus increasing the flow through the inner one and reducing the throat pressure even more.

Another device depending on differential pressure and having some similarities to a venturi is the Elliot-Nathan flow tube illustrated by Hayward (1977c), which appears to have a sharply converging and diverging venturi, in the diverging portion of which is a hollow cone in which the downstream pressure is measured. Hayward commented that it is claimed to have an outstanding performance. However, strictly this is a full-bore meter, since its installation would have to be in a spool piece.

Cairney (1991) described the use of a bidirectional pitot-venturi inserted into the bled-steam line of a generating plant in which there was a possibility of a flow reversal due to a major transient in the system. The device consisted of two very small venturi meters with axes parallel but facing opposite directions in which the differential pressure was measured between the throats. The device was sensitive to Reynolds number and turbulence level.

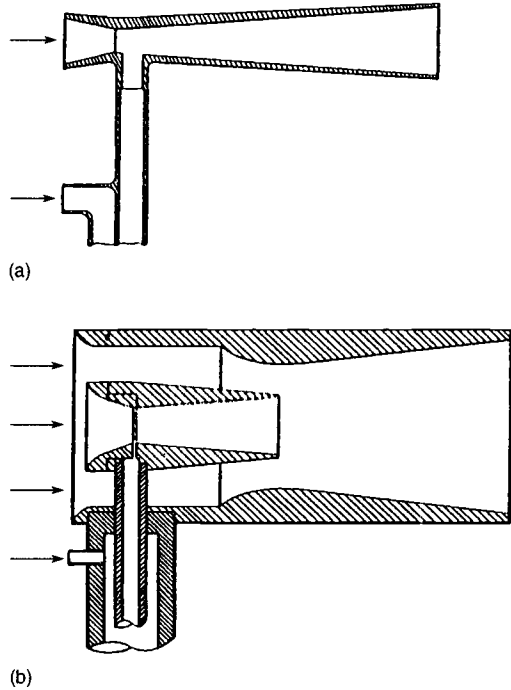


Figure 18.4. Pitot-venturis (Bean 1971; reproduced with permission of ASME): **(a)** Single venturi; **(b)** Double venturi.

18.4 INSERTION TARGET METER

The flow acts on the disk (Figure 18.5), which is supported by a lever. The small movement of the lever is sensed by an inductive displacement, counterweight, or magnetic coupling to a pneumatic sensor. Liquid flows can be measured in a range of about 0.05–4 m/s and gas flows from about 0.6 m/s up to about 5 m/s. It may be applicable to temperatures up to 500°C. Measurement uncertainty for gas flows is claimed as $\pm 1.5\%$ measured value.

18.5 INSERTION TURBINE METER

18.5.1 GENERAL DESCRIPTION OF INDUSTRIAL DESIGN

The insertion turbine flowmeter is a small diameter turbine and pickup on the end of a relatively small diameter probe, usually inserted into the pipe via an isolation valve. The propeller may be able to cope with velocities for liquids of 0.1–12 m/s or possibly higher, and for gases of 1 m/s or less to 50 m/s or possibly more, with a range

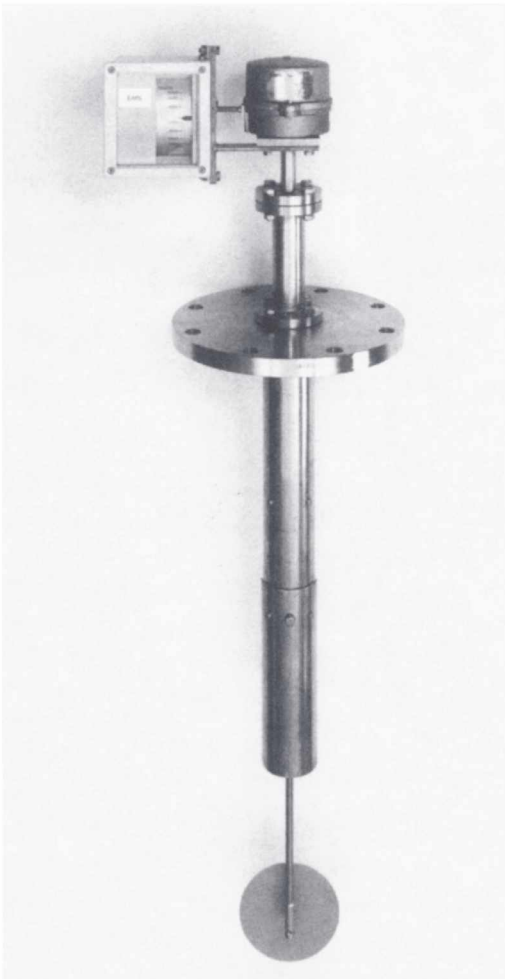


Figure 18.5. Insertion target disk flow probe [reproduced with permission of Laaser (UK) Ltd.].

for one probe of about 10:1. An operating range of as much as 100:1 may be quoted in some designs. The turbine probe, shown in Figure 18.6 retracted into the threaded pipe fitting, may have ball bearings, jeweled pivot, tungsten carbide journal, or other. Ideally this will result in a rotational speed for a limited range proportional to the flow rate past the head. There is likely to be a minimum velocity for which the propeller will turn and another for which it can be expected to give results within the normal precision. For very low drag, the pickup may be radiofrequency or other low drag type. A typical manufacturer's specification follows:

Diameter of rotor	15–30 mm
Linearity	$\pm 0.5\text{--}1\%$ of full scale
Repeatability	0.05–0.25%
Turndown	10:1
Construction	Stainless steel
Temperature range	-200 to 150°C

The turbine is inserted through an isolation valve of through-bore type with a clear concentric opening of diameter depending on the meter. The complete device consists, typically, of a small stainless steel turbine on the end of the stainless steel insertion tube (Figure 18.6). External handles may indicate the alignment of the probe, and this should be axial to avoid errors.

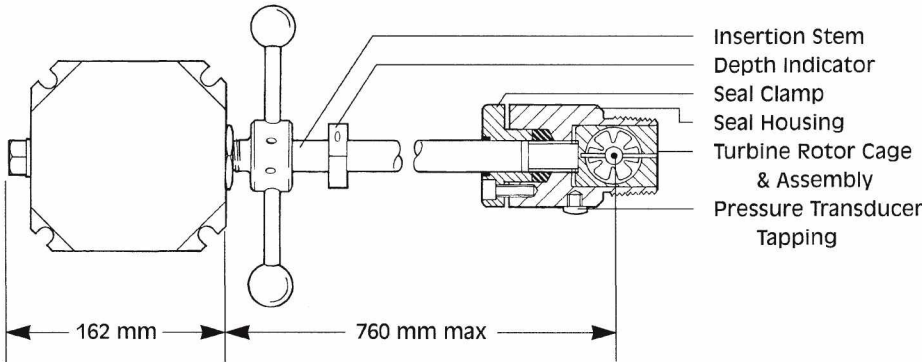


Figure 18.6. Insertion turbine probe showing the turbine cage retracted into the screwed pipe fitting (reproduced with permission from Flow Automation (UK) Ltd.). Note: Figure is drawn for clarity, not correct alignment.

The insertion tube passes through an outer tube, which seals the line pressure. When this pressure tube is inserted into the valve, the valve can be opened, and the turbine can be inserted through the valve into the line. Fluid seal mechanisms need to avoid leakage but allow the probe to be inserted. In high pressure applications, inappropriate selection can result in the probe's being expelled from the process line, or at least moved from its set position.

These probes are suitable for water and hydrocarbons, particularly the latter because of their lubricating properties. They are not likely to be suitable for high velocities or hot water.

If such a probe is to be inserted and used as a single reading device for pipe flow monitoring, then the position of the probe in the pipe will need to be carefully considered as discussed earlier.

To obtain a full flow profile, a series of measurements can be made, and the results can be integrated to obtain the flow rate. There is usually a limit to the proximity to the pipe wall at which measurements can be made. However, the nonslip condition at the wall should allow a curve to be fitted for this region. The profile shape thus determined allows the computation of volumetric flow. In this mode, they are also used for in situ calibration, although the overall uncertainty is likely to be in the range $\pm 2\text{--}5\%$.

Various other designs of wall-mounted probe are available for hot-tapping or fitting into a T-piece in the wall. An example is shown in Figure 18.7. These paddle-wheel-type probes essentially measure flow near the wall of the pipe, and the deduction of mean flow in the pipe will be obtained by reference to the manufacturer's literature.

18.5.2 FLOW-INDUCED OSCILLATION AND PULSATING FLOW

Small diameter probes at the center of a pipeline of, say, 2 m diameter or greater, with high velocity flow, are prone to oscillate. This may appear as a pulsation but can lead to failure of the support tube in extreme cases.

Ower (1937) developed a theory for a vane anemometer, showed that it overestimated average air speed, and obtained an expression for the error. Experiments confirmed the theoretical conclusions.

Jepson (1967) investigated the effect of pulsating flow on current meters and concluded that fluctuations of the velocity vector parallel to the mean flow always cause overestimation, whereas fluctuations perpendicular generally cause under-registration. He suggested that errors could be reduced by

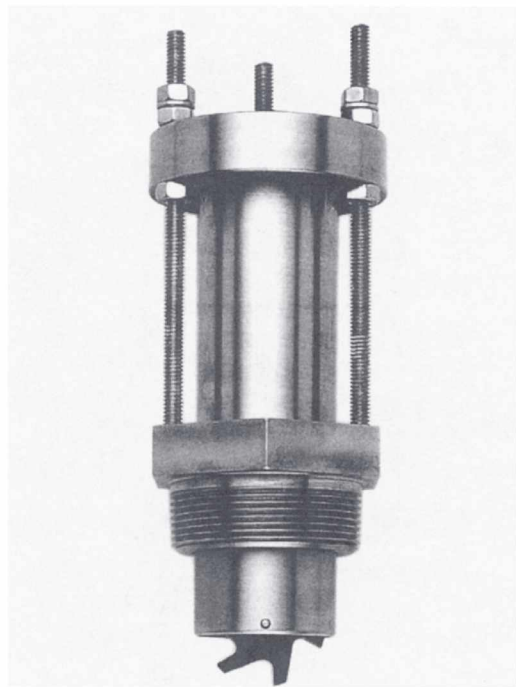


Figure 18.7. Paddle-wheel-type flow sensor (reproduced with permission of Peek Measurement).

- a. using blades with a large aspect ratio,
- b. making the tip-to-tip diameter small to keep the blade thickness as small as possible,
- c. keeping the blade angle to a minimum, and
- d. using low density material for the rotor.

Jepson's paper is a useful source of earlier references on current meters.

18.5.3 APPLICATIONS

Application of turbine probes to large diameter pipe flow measurement should be undertaken with care, recognizing the need for an integration routine, the blockage effect caused by the probe, the effect of wall on probe calibration, the errors in measurement of pipe diameter, and the total uncertainty of final value of flow rate.

Turbine probes have been used in applications such as compressor efficiency and surge control, pipeline leak detection, odorizers, samplers, and checking throughput. Some have claimed that this device can be used for custody transfer to measure with high accuracy and reduce cost by avoiding the purchase, installation, and

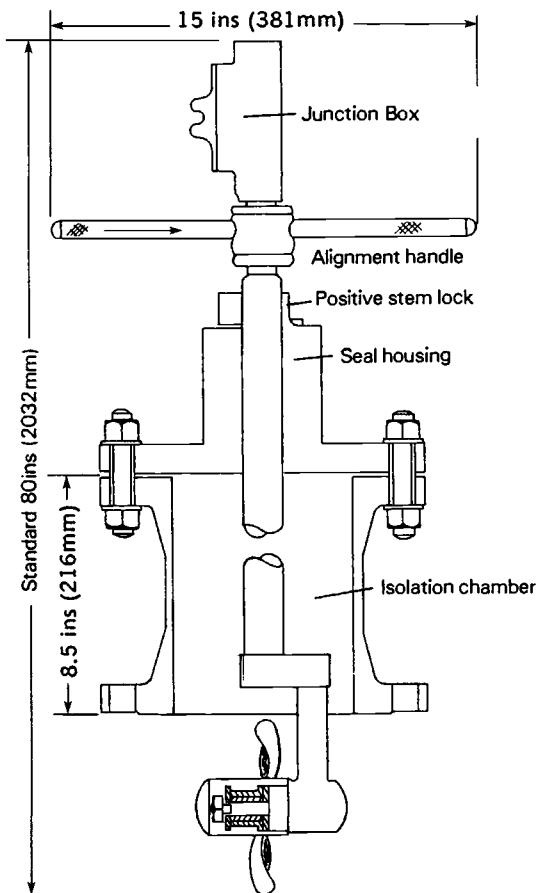


Figure 18.8. Mauflo crude oil insertion flowmeter (reproduced with permission from F.M.A. Ltd.).

maintenance of a large bore flowmeter. In my opinion this is rarely likely to be achievable. However, if calibrated *in situ*, the probe may provide a long term check on performance, although short term repeatability may not be so good.

Another device has been developed for crude oil flow measurement (Figure 18.8) and consists of a rotor of up to 140 mm diameter that is capable of coping with wax and fibrous materials in the flow. For a flow range of 0.15–12 m/s, the uncertainty is claimed to be about $\pm 0.3\%$ at the top of the range and $\pm 5\%$ at the bottom with a repeatability of $\pm 0.15\%$.

Raustein and Fosse (1991) used insertion turbine meters for fuel gas flow measurements and estimated measurement uncertainty as ± 2 – 5% . They are also used for flare gas metering where it appears that uncertainty is likely to be in the range ± 5 – 10% .

18.6 INSERTION VORTEX PROBES

A commercial vortex-shedding insertion probe, which has been available in the past, had a rectangular slot in the end of the insertion bar (about 32 mm diameter), and the bluff body spanned this perpendicular to the axis of the bar. The insertion bar was retractable and of length up to about 900 mm. A turndown of 20:1 was claimed with repeatability of 1–1.5% of reading.

18.7 INSERTION ELECTROMAGNETIC PROBES

Insertion electromagnetic probes have been the subject of considerable development, and their performance has been analyzed by Hemp (1995b) and Zhang and Hemp (1994, 1995) to obtain the sensitivity of its signal to proximity to the wall. Industrially, it has also advanced in terms of accuracy and of low power usage.

The probe shown in Figure 18.9(a) is claimed by the manufacturer to have a range up to about 5 m/s, the actual maximum depending on the insertion length with maximum flow allowed for insertions of 300 mm or less. The accuracy is given as $\pm 2\%$ of rate or ± 2 mm/s, whichever is the greater. Insertion is up to 1 m with typical materials for wetted parts of stainless steel and PVC. The temperature range for water is given as 0–60°C. It is designed for local measurement of water velocity. Figure 18.9(b) shows the installation arrangement (cf. Baird 1993 who described an insertion probe for pipes of 25–300 mm).

These devices in various forms have been tried out for current flow measurement, for ships' logs, and for pipe flows for many years. For more references, the reader is referred to an earlier review by Baker (1983).

One manufacturer offers probes with different sizes of disc-shaped head, in one case with a spherical head with electrodes projecting radially, and in another with a ring with electrodes projecting radially inward. The heads are directionally sensitive and operate at 128 Hz. The measurement uncertainty is claimed as 1% of reading $+ 5$ mm/s, and the direction is claimed to be better than 5° (cf. Bowden and Fairbairn 1956).

Another insertion probe, but for axial insertion, was developed for blood flow in arteries (Mills 1966, cf. Baker 1968).

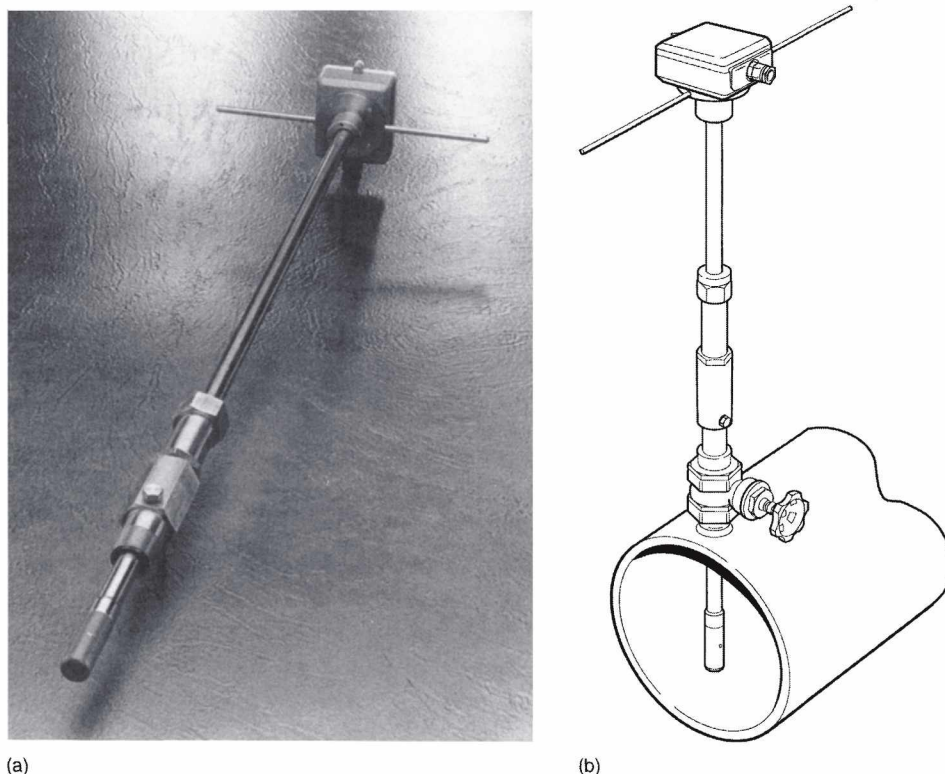


Figure 18.9. Electromagnetic-type insertion probe (reproduced with permission from ABB Ltd.): **(a)** Pipe insertion probe; **(b)** Insertion arrangement.

18.8 INSERTION ULTRASONIC PROBES

The doppler ultrasonic principle lends itself to applications as a flow probe, provided there are sufficient scatterers in the flow; most of the reservations have already been set out in Chapter 13.

A very much more precise instrument, depending on the transit-time principle, has been developed by Rawes and Sanderson (1997). The device consists of a bar, with ultrasonic transducers on each end, that is inserted through a fitting in the pipe wall. Once in, the bar is turned into the flow so that it lies parallel with the pipe axis. It is then possible to send ultrasonic pulses from one transducer via reflections on the pipe wall to the other, both up- and downstream. This may lead to quite wide flow coverage and flow averaging. Its performance, even downstream of disturbing fittings, appears to be better than $\pm 2.5\%$ for higher flow rates. Figure 18.10 shows the essentials of this meter.

Olsen (1991) reported on the use of transit-time techniques to measure air velocity. The frequencies used were 3.1 Hz (sonic) and 40 kHz (ultrasonic). He claimed that the precision was better than a pitot-static tube and that the small size, simplicity, and low price compared well with other methods.

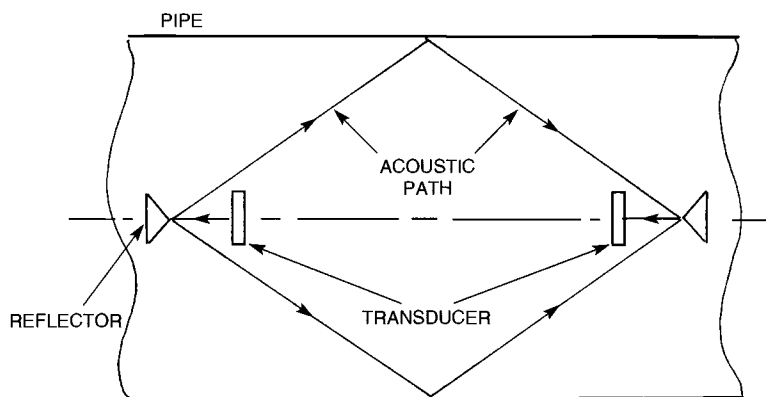


Figure 18.10. Ultrasonic-type insertion probe (after Rawes and Sanderson 1997).

18.9 THERMAL PROBES

Thermal probes are used so frequently as part of a full-bore flowmeter spool piece that they have been included in Chapter 15.

18.10 CHAPTER CONCLUSIONS

The turbine insertion probe has been an essential workhorse of the industry and is likely to continue to be, particularly where nonconducting liquids and gases are involved.

The latest developments in electromagnetic probes are very promising, and further designs and developments may well appear.

The novel ultrasonic meter of Rawes and Sanderson (1997) may well set a new standard for precision in the measurement of bulk flows using insertion methods. It is likely to be the subject of considerable development, in order to perfect both the mechanical insertion system and the ultrasonic sensing and averaging procedure.

No doubt a further area of development will be the secondary equipment and its ability to obtain the best averaging of flow traverse data or of the ultrasonic response.

One of the weakest parts of in situ calibration using insertion probes tends to be the measurement of internal pipe dimensions. The ultrasonic techniques should provide a precise means of achieving the cross-sectional area.

The problems of blockage and the effect of wall proximity on the flow around the probe may lend themselves to computational modeling. However, the ultrasonic techniques may, again, offer a means to obtain the profile modifications due to the probe.

Modern Control Systems

19.1 INTRODUCTION

This chapter does not attempt more than a cursory review of modern control systems, as it is outside my main area of expertise. Those knowledgeable in electronics and control will pass quickly over this chapter! Others who wish to delve more deeply may, for instance, turn to Sanderson (1988), Brignell and White (1994), or other experts, or for a good, recent introduction to the control of machinery Foster (1998) is recommended. I am pleased to acknowledge valuable insights obtained from Cranfield lectures given by Bill Black, David Clitherow, Doug de Sa, and others.

This book is about flowmeters that depend, in the main, on mechanical or thermo/fluid mechanical effects. In only one case of mainstream meters, the electromagnetic meter is the signal electrical throughout. The aim of the book has been to bring together information on the performance of these meters based on physical, experimental, and industrial experience. I have made the assumption that the electrical signal interpretation, which is based in the meter, is capable of doing the job set by the designer of interpreting the meter output into a standard electrical form. This assumption is not always adequate (e.g., where electrical signals from the same instrument do not appear to agree with each other). Nevertheless, the developments in design of electrical systems are assumed to be such that the reader will not benefit from a detailed description or that the circuit will be changing so fast that any statement will be out of date.

Once the signal is available, we shall need to feed it into the overall control system. It is clear that if it does not match the rest of the control system's protocol, conditioning may be necessary, but this is an expert job, and detailed signal conditioning will not be undertaken lightly. In many cases, interface modules that do this conversion will be available.

From the time when process operators adjusted valves to control flows on the basis of visual flow readings, electrical control has steadily replaced the operators' functions. Like the operators, the sensitivity of the electrical systems could result in undercorrecting or overcorrecting and could cause unstable operation by amplifying the natural fluctuation of the process. The operator then needed to keep an eye on the automatic control and intervene if necessary. These control systems were analogue. They used an electrical signal and amplified it, integrated it, and differentiated it in a way analogous to mathematical actions. However, the operator was always ultimately responsible for setting the values and able to override if necessary.

The advent of digital computers allowed the acquisition of much more data and more detailed assessment of the performance of the plant.

However, as digital computers superseded (largely) analogue computers, so digital control has superseded analogue control in many cases so that the computer can respond to signals and adjust the plant in a far more detailed and sophisticated way.

19.1.1 ANALOGUE VERSUS DIGITAL

Signal conditioning is a major subject in its own right, and the reader is referred to other sources such as Sanderson (1988). Sanderson classifies transducers by

- ❑ type of output – voltage, current, or charge;
- ❑ type of signal – unidirectional, bidirectional, alternating, or transient;
- ❑ information content of signal – by level or by amplitude, frequency, or phase modulation of a carrier wave;
- ❑ relationship between physical variable and electrical output – linear or nonlinear; and
- ❑ output impedance.

Analogue signal conditioning is often to

- ❑ convert output to voltage;
- ❑ change impedance as required; and
- ❑ amplify, filter, demodulate, and linearize.

From classical control theory with its sophisticated mathematics and relative inflexibility, the modern control systems use the power of the computer to oversee the process, to question the sensors, to identify problems and rectify them, and to provide the operator with a vast amount of information and control, but they are packaged in a way that is easily accessible.

Distributed control systems (DCS), which are analogue, need to be fast with higher bandwidth and built-in redundancy (e.g., for the petrochemical industry).

In much primarily mechanical plant, the control was originally exercised by electromechanical relays. These relays would control the electrical power to the controlling elements in the system (e.g., valves). When these were superseded by programmable logic controllers (PLC), the logic used in the relays was carried over as ladder logic.

One most important feature of digital systems is that they are less susceptible to interference in that they can carry their own checking protocol within the message.

19.1.2 PRESENT AND FUTURE INNOVATIONS

A major step in the move toward modern control was to remove the analogue systems that were much less flexible, and could not easily be reprogrammed by a computer command, and to introduce the microprocessor. This then allowed a further sophistication – the individual instruments became intelligent and were able to be instructed to do simple tasks and, in turn, could provide much more information to the central computer. The computer could store past information and compare past performance with current operation, thus providing further checks on plant performance and safety.

This led to the idea that the communication system could be a signal highway (or bus), to which any instrument with the right protocol and any computer could be connected. The signal highway could be of copper, but equally could be of optical fiber, electromagnetic waves, or superimposed signals carried by the electricity main or telephone lines. Transmitters and repeaters could then include satellites, and the system could be, essentially, unbounded.

At the other end of this communication link is the computer or computers that control the process and that will be able to work in tandem if necessary, and the plant operators will work to visual display unit (VDU) screens with the latest windows presentations and graphical portrayal of each part and aspect of the plant and its operation.

All this requires a generally accepted common communication language or protocol to which instruments and systems conform and that will ensure each message is correctly sent and received, without interference.

Modern telemetry systems provide the means of controlling plant and measuring its performance from a central station. The three aspects of the process are

- the measurement technology, which has been the prime subject of this book. The microcomputer has revolutionized this technology with smart and intelligent instrumentation. One aspect that may need attention is signal conditioning. The signal from a sensor may not be suitable for a system and may need to be adapted: from analogue to digital, from digital to analogue, or other change.
- the method of transmission, which, having been by means of copper wires for much of its history to the present, is now the subject of continual and highly sophisticated alternative developments.
- the controls, both instrument outputs and process adjustment means. From direct, but remote, control of instruments, through analogue computers, this has now firmly arrived at digital computer techniques.

19.1.3 INDUSTRIAL IMPLICATIONS

The range of industries is now very wide and includes water and sewage treatment and distribution, oil and gas production and pipelines, process industries of all sorts, and manufacturing including the production of flowmeters. The techniques are, of course, not limited to process flow but can also be used in the electricity generation and distribution industry, the electrical power transmission industries (e.g., rail networks), building services, and lighting control.

This means that the typical company supplying such equipment and systems is increasingly a software-based company, using proprietary equipment and supplying the software for the control station and the outstations, and using whichever transmission system is most appropriate to the task and the customer's needs.

19.1.4 CHAPTER OUTLINE

This chapter provides an overview of how the signals from these meters are used in the control of plant. In referring to meter or flowmeter, the reader should understand that broadly the same will be true of other instruments within the system. I shall start from the assumption that, despite increasing coordination between signal specifications (protocols), there is still sufficient variation to make it impossible to use

an unconditioned signal from any instrument. The modern control systems will, therefore, in some form, require the following components (Figure 19.1):

- a. the instrument;
- b. an interface box that translates electrical signal to and from the instrument into a signal as required by the remainder of the system;
- c. a communication language or protocol;
- d. a communication medium;
- e. an interface that receives the signals and translates them to the computer protocol;
- f. a computer that receives signals, checks their validity and appropriateness against limits, models, and past experience; transmits control information to both meters and valves, etc.; and displays its actions with the maximum user accessibility.

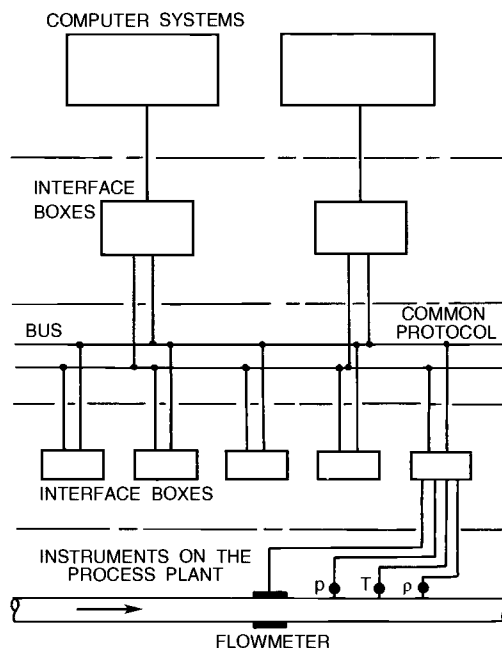


Figure 19.1. Main components required for a modern control system.

19.2 INSTRUMENT

19.2.1 TYPES OF SIGNAL

The rest of this book is about the details of the flow sensor and how it senses a flow of fluid and converts this into an electrical signal. The signal will usually be one or more of the following:

- ❑ an analogue signal of 0–20 or 4–20 mA or a voltage or a resistance;
- ❑ a frequency signal – this may be of high or low frequency or both;
- ❑ a digital signal in which the flow quantity is transmitted in a numerical form, more likely to be binary than decimal;
- ❑ an intelligent communication that allows the sensor to provide more than one piece of information, and the user, through a computer, to instruct the sensor (and other devices) in various ways.

Twenty-five years ago we were concerned with the cleanness of the signal, which came from a meter and provided just one parameter, the flow. The advent of computing power, and particularly the microprocessor, has allowed more information from the signal to be interrogated and multiple outputs to be handled. Thus interest is now being shown in the details of the noise in the signal, which previously we smoothed out and discarded.

19.2.2 SIGNAL CONTENT

Amadi-Echendu and Hurren (1990) (cf. Amadi-Echendu and Higham 1990, Amadi-Echendu and Zhu 1992) discussed the use of noise in flowmeter signals to obtain information about the condition of process plant and condition monitoring in general. For instance, the variation of pulse spacing in a turbine meter may be obtained, and the nature of this fluctuation may be interpreted as due to the signature of the plant made up of pulsatile flow, etc. Change in the signature may indicate a change in conditions in the plant. To date, there does not seem to be much theoretical analysis to help our understanding of the relationship between secondary signal and the physical fluid behavior. However, some recent work by Krafft et al. (1996) has started to do just this by developing the theoretical prediction of the effect of air bubbles in a magnetic flowmeter and comparing this to the signal obtained.

The microprocessor has allowed a further, and major, step. It is now possible to build into the instrument means for checking the validity of the signal, for adjusting range, and for selecting the form of signal. The meter may, therefore, be capable both of transmitting and receiving information.

For such meters, the terms *smart instrument* and *intelligent instrument* were coined and associated with various degrees of on-board computational power. The definition of these terms can be found in Higham and Johnston (1992), but the continual developments resulting from microprocessor power mean that the distinction is likely to become less relevant or used. They suggested that *smart* be applied to sensors that automatically compensate for nonlinearity, temperature effects, etc., and can be configured before installation to provide a predetermined range and span. However, they would normally need to be taken out of service to be reconfigured.

The term *intelligent* includes the smart features, but also implies two-way communication with the control system either by a separate communication port or by a superimposed digital signal on the 4- to 20-mA signal. This greatly increases the versatility of the sensor, allowing self-diagnostics and other routines and commands to be sent from the control system to the instrument.

Brignell and White (1994) list the principal subsystems within an intelligent sensor as

- a primary sensing element,
- excitation control,
- amplification and possibly variable gain,
- analogue filtering,
- data conversion,
- compensation,
- digital information processing, and
- digital communications processing.

A most important attribute of the intelligent sensor is addressability, which is a necessary part of the bus communication, but it also will put time constraints on the signal transmission each way. Thus the instrument's microprocessor

- processes signals from one or more sensors and replaces complex analogue electronics;

- ❑ is capable of re-ranging, diagnosing problems, etc.;
- ❑ oversees the communication of the sensor with the wider control system; and
- ❑ manages display functions if present.

The microprocessor then allows enhanced accuracy by putting in the calibration curve, which provides

- ❑ high turndown ratios leading to increased rangeability – but beware of hidden errors,
- ❑ increased manufacturing flexibility, and
- ❑ increased reliability and with a quicker identification of any problems, an improved maintenance philosophy.

There may be some disadvantages such as a change in working practices, the possibility of changing ranges without considering safety implications, and the situation of being harder to repair by in-house people and so more likely to lead to a throw away/replacement philosophy. However, many of these problems have the solution implicit in the new technology, by using passwords, more sophisticated algorithms, etc.

19.3 INTERFACE BOX BETWEEN THE INSTRUMENT AND THE SYSTEM

The system designer and builder is unlikely to be the manufacturer of all the instruments in the system. There was a trend this way about 25 years ago, but the greater flexibility of the microprocessor has encouraged control system design to be capable of coping with a variety of instruments from different manufacturers and for various industries. Thus, although some instruments may incorporate the interface in the sensor, the simplest arrangement is to have an interface box that ensures a common signal protocol. This reflects the comparative cheapness of the microprocessor-based box, in comparison with the cost of expert time to adapt an instrument to a particular protocol. These devices are sometimes called controllers and may be capable of handling from 1 to 30 or so sensors with individual loops.

The output of this box (to the highway/bus) will be one of the emerging signal protocols. This will probably be a digital signal that has a code to describe the nature of the particular data set that follows the code. The codes will then define particular instruments (or control devices) to which the signals are directed or from which they come and, in addition, will define the nature of the data, whether flow rate etc., or range, or some other quality. Assuming that these code and data strings will be vying for space on the same pair of wires, or the same optical fiber, the instrument will need to set up a “handshake” arrangement with the rest of the system to allow time for the signal to be transmitted. This in turn will have repercussions for data transmission rates.

This type of outstation may also provide local control and may have main and standby units.

19.4 COMMUNICATION PROTOCOL

19.4.1 BUS CONFIGURATION

In the past, connection of sensors to the central control has been in the form of a star where each instrument communicates via a, usually, wire pair to the center (Brignell and White 1994). There is clearly much cabling required, and it becomes most dense at the center. The modern alternative, the bus, allows all devices to share a common pair of wires. Disadvantages are that

- ❑ devices need to be addressable and so to have a level of intelligence,
- ❑ breakage in the bus can cause a loss of communication, and
- ❑ pressure on the bus may limit time availability for signal transmission.

To overcome the particular danger of bus breakage, a complete loop bus can be used, or even two bus loops separated physically to avoid the common destruction of both.

The system components will use one of the standard protocols, for instance:

- ❑ 4 to 20 mA signal for analogue instruments;
- ❑ digital instruments with digital communication;
- ❑ flow computer output with serial interface that provides an encoded binary transmission; and
- ❑ a modern bus with high level, computationally controlled protocols (e.g., HART, Profibus, and Fieldbus), which interface to intelligent instruments.

It should be remembered that digital information in computers is carried as a series of 1s or 0s and so, for instance, the logic 1 could be portrayed as a burst of higher frequency, whereas the logic 0 would be portrayed as a burst of lower frequency. The message would consist of a series of 1s and 0s, with indications of start and finish of words, and any instrument attached to the cables would respond when its call sign was sensed on the bus/highway.

Brignell and White (1994) discussed the ISO's (see ISO 7498-1:1995) seven-layered communications protocol for an Open Systems Interconnect (OSI) strategy.

Physical layer	Relates to connections such as physical connectors, and electrical procedures for handshaking, etc.
Data link layer	Ensures reliable and error-free transmission.
Network layer	Appears to ensure standard signals and appropriate communication paths regardless of where the signal originates.
Transport layer	Provides a reliable data transfer exchange service between the instruments, etc., and the user.
Session layer	Manages the interaction between different applications on different systems connected to the network.

Presentation layer	Ensures a common data format.
Applications layer	Provides services that can be accessed and used by the computer systems.

These cover, broadly, the levels (a) to (f) in Section 19.1.4.

19.4.2 BUS PROTOCOLS

Manufacturing Applications Protocol (MAP) is based on the seven-layer ISO/OSI model. General Motors was largely responsible for its development (Brignell and White 1994). Statistical tasks for optimizing production, order processing, planning, and different logistical services may be available from such protocols (Lindner 1990). However, MAP is not widely used as a general-purpose instrumentation standard due to the severe requirements for the hardware/software processing resulting from the full seven-layer protocol.

Wood (1994) described HART and Profibus as examples of proven specifications that provided background and influence for the modern fieldbus proposals.

Highway Addressable Remote Transducer (HART) digital communications protocol developed by Rosemount (Brignell and White 1994) uses digital techniques to allow an intelligent instrument to adjust range and span, to undertake flow calculations, to apply self diagnostics, and to communicate (Howarth 1994). HART uses point-to-point or all-digital modes. In the former, it superimposes digital data on a 4 to 20 mA signal by two defined frequencies of ± 0.5 mA amplitude: 1,200 Hz for binary 1 and 2,200 for binary 0. Message rates are typically three per second. It uses a master/slave protocol (Brignell and White 1994) in which the master is the only device that may initiate a message transaction. Up to two masters can communicate with connected field instruments, typically a hand-held communicator and a control system or PC-based workstation. At the applications layer, HART commands are of three types: universal commands to read an identifier, variable, or current; common practice commands (e.g. calibrate or perform self test); and transmitter-specific commands (e.g., start/stop totalizer, read density calibration factor).

Profibus was set up by a group including AEG, ASEA Brown Boveri, Honeywell, Klockner-Moeller, Lands & Gyr, Robert Bosch and Siemens, with the objective of creating a digital fieldbus standard consistent with the ISO/OSI model in ISO 7498 (Squirrell 1994).

Interoperable Systems Project (ISP) was formed in 1992 (Allen 1994) with the objective of achieving a single, open, interoperable, international fieldbus.

World FIP (Desjardins 1994) was to promote the acceleration of a viable international fieldbus standard of manufacturing and process control and resulted from a memorandum of intent in 1993.

The ISP and World FIP (North America) combined in 1996 to form Fieldbus Foundation, which produced in 1998 an interoperable replacement for 4 to 20 mA control, that is at least as reliable and replaceable. Members represent instrument suppliers and end users.

Fieldbus is a communication system specially designed for networking of transducers (Brignell and White 1994). The protocol model has physical layer, data link layer, applications layer, and also a user layer.

In conclusion it should be noted that this brief introduction to fieldbuses was believed to be correct at the time of writing, but it should also be remembered that there are other fieldbuses available in addition to those mentioned in this section, and the reader should obtain expert advice on the one most appropriate to a particular application.

19.5 COMMUNICATION MEDIUM

19.5.1 EXISTING METHODS OF TRANSMISSION

Pneumatic transmission has been used widely in the past and had the great virtue of being intrinsically safe. As with analogue electrical signals, standard signal ranges ensure that instruments are compatible and that pressure supply failure will not produce an undetected error. Thus 20–100 kPa and 20–180 kPa are two such (Sanderson 1988). It is likely that with optical transmission offering an alternative and, in most cases, intrinsically safe communication system, and with the inherent compatibility of optical with digital electrical systems, pneumatic systems will become less common.

Analogue electrical transmission depends on a small current, voltage or resistance to provide the signal from a sensor. Much instrumentation still in service uses these signals, and so analogue communication is likely to be around for some time. Again it is likely to decrease, and, indeed, sensors that exploit digital and other modern technologies are likely to be designed. Many analogue sensors included a frequency output, which allowed pulses to be counted at the end of the communication line and a total flow or other parameter to be obtained.

It is now possible to send complete information on parameter values by digital signals, which do not need to be converted from a milliampere value to a flow rate but which provide the flow rate in the required units. In addition, the sensor can be addressed and will respond to questions or commands. The communication has thus become two-way.

19.5.2 PRESENT AND FUTURE TRENDS

The means by which the components in a modern control system are connected are now many and various and exhibit some of the ingenuity that stems from the great power of modern signal processing. Thus communication for a conventional plant may well be via copper wires or, where there are dangers due to explosive environments, via optical cables. The transmission system may be by the public switch telephone network (PSTN). A frequency signal is essentially the signal used in telephone communications and, therefore, in most of the world's communication by all forms of transmission and in repeater stations such as satellites. For oil and gas, transmission may be by microwave, which requires line of sight between the control station and outstation, or by radio waves bounced off the troposphere or linked by satellite. A utility may develop means for sending the signals down the power lines that link every house, whereas an international oil or gas company may be able to afford radio links via their own satellite or rented satellite space.

Combinations of these and other methods will be selected depending on the most appropriate route for the particular application. For added security, dual methods may be used providing a fall-back in case one method fails. The options are likely to increase.

The bus/highway will increasingly link computers with control and communication capability and smaller modems with hand-held devices for field checking. The development of such data highways is likely, eventually, to lead to the elimination of all but digital information. Standardization will address the frequency levels of the bus, which will affect the speed of response of the instruments.

The sensor will, therefore, be capable of linking with computer control systems anywhere in the world or beyond and the wire connections may become an antenna. Even the problems of power supply are overcome with local power generation from solar, wind, or nuclear sources.

19.5.3 OPTIONS

Imrie (1994) commented that the type of measurement being made will determine to a large extent the type of communications required:

- ☐ Is the variable to be used in a control or a time-critical application?
- ☐ What is the likely rate of change of the variable?
- ☐ Is continuous trend monitoring required, or is it event-driven?
- ☐ Does the sensor lend itself to power saving?

The location will determine security, protection, safety, interference, availability of services (e.g., power and signal networks), and the access requirements of ownership of land.

The consequence of data loss, the rate of data transfer, the response time, one-way or two-way transfer, and the size of data package all will affect choice.

The nature of the application will be important: control or monitoring, regulatory and insurance, safety, effect on product or services.

Imrie then went over the options: cable direct connected, service radio, satellite communications, mains borne, meteor burst, licensed radio, and delicensed low power radio. In addition, careful consideration of networks and protocols and of power management is necessary. Imrie suggested the following applications:

For	Use
Real-time measurement and/or control	Cable or radio
Time critical	Cable or radio
Event-driven with a fast response	Cable or radio with appropriate protocol
Event-driven with a slow response	A link establishment network over telephone or service radio networks, or low use power radio

Long-term trend measurement	Link establishment network over telephone or service radio networks with overnight historical file transfer
Multinode system	Suitable network, not multiple point-to-point links

19.6 INTERFACE BETWEEN COMMUNICATION MEDIUM AND THE COMPUTER

For a particular company designing a control system, this interface will be essentially a modem that converts the electrical signal transmitted to the size and form needed by the computer. The transmitted signal will be specified by the protocol or by the communication network management, and the other side of the interface should be under the control of the system designer and meet the requirements of the computer interface.

19.7 THE COMPUTER

By comparing the operation of the process with models programmed into the computer, the system could set the control points, removing operator error and reducing operator time.

There is wide use of modeling and simulation techniques for areas as diverse as network analysis for water distribution systems, chemical mixing, and pump control using software suites (Fowles 1994).

The control station, which will represent the latest in user-friendly presentation, will probably include a computer model of the network, based on predictions such as those of Miller (1990). Thus, the computer will enable the user to identify on a map where the control is taking place, will provide the user with a virtual control panel, will allow the user to obtain trends and to compare them with expectation, and will provide alarm controls, etc.

There are over 100 suppliers of SCADA/telemetry packages (Fowles 1994). Supervisory control and data acquisition (SCADA) systems send data from the field to the model, and the model assesses the data to distinguish good from bad. Supervisory control will include emergency shutdown, with a block valve in an automatic control loop. Thus in the water industry, there may be local automatic control shutdown of the pumping system, which operates against, say, the depth of water in the tanks. In the oil and gas industry, this may not be the case.

Microprocessor-based protection systems (MBPS) are usually concerned with an undesired end event (Churchley 1994/5).

19.8 CONTROL ROOM AND WORK STATION

Little needs to be added to comments already made. It will be obvious that this area will increasingly represent the most modern office with well-laid-out computer

equipment readily available to the operator, encouraging able people with a high level of technical training and interest to operate the plant with a suitable level of responsibility.

19.9 HAND-HELD INTERROGATION DEVICE

The development of hand-held devices is particularly important in the following two areas:

- a. As a device for the process plant engineer enabling him or her to interrogate the system while at various points on the site. Lindner (1990) suggested that these devices will usually be battery powered and will be used for identification, configuration, and diagnostics and used the graphical title of digital screwdrivers! These instruments will need to be capable of operating in adverse environments.
- b. Providing the utility industry with instruments that can interrogate the utility meter via a remote reading pad using inductive or other pickup, so that, without entering the property, the meter reader can obtain meter identification and logged information. In due course, these may give way to signals transmitted, say, by telephone or radio links.

19.10 AN INDUSTRIAL APPLICATION

Boettcher and Hickling (1993) described the systems for Sizewell B power station in the United Kingdom. The man-machine interface was designed to ensure that all information was available to the operators with high reliability. The user interacts via graphical screen displays of windows, icons, menus, and pointers, a so-called WIMP environment. This results in an extensive interactive capability of the computer system. Alarm systems are available in various formats. Safety and reliability are fundamental to such a system. A distributed computer architecture was used to allow flexibility and extendability.

Communication was over local networks. The data processing and control systems were divided into two levels. Level 1 comprised the set of programmable logic controller type devices and interacted directly between devices and the control room panels, reading sensor and status of devices and issuing control signals. These were high integrity control systems (HICS). Information was transmitted to level 2 distributed control system (DCS) for display on the VDUs. Separation, duplication, and redundancy were built into HICS (cf. Cluley 1994/5, who discussed reliability, the use of redundancy, environmental effects, self-calibration, display devices, software reliability, component selection, environmental factors, and measures of reliability).

19.11 FUTURE IMPLICATIONS OF INFORMATION TECHNOLOGY

From a mechanical engineer's viewpoint, there appear to be a number of fairly obvious ways in which the power of information technology will continue to influence

the instrumentation and control field. These are likely to be driven by the rapidly advancing computer technologies.

Control Station

Leaving aside the possibilities that humans may communicate with the computer by means other than hands, feet, and voice, the visual graphics, the virtual reality developments, and the windows environment of modern computers will demand of the process engineering designer increasingly detailed simulations of hardware, software, physical behavior, and safety scenarios with chaos theory implications. In many cases, the expert system will provide the operator with an instant assessment of the most likely causes of failure or of unusual operations.

Behind such information will be increasingly detailed computer graphics but, more importantly, increasingly detailed understanding of the complex nature of flow phenomena. The computer experts, with engineers and scientists, will attempt to model various theories of catastrophic failure to enable the computer system to recognize possible symptoms at an early stage.

Communications

The move to digital has released most constraints from the transfer of information and opened up the vast array of current communications used in telephone, space, etc. This is likely to yield new and unforeseen additional possibilities.

However, it is possible that, in some cases, the work on neural communication, and the methods used there, may introduce faster links in the communications chain or new statistical insights into information transfer.

Sensors

Most sensors in use today stem from the mechanical engineer's need to measure quantities in the most efficient ways prior to information technology. If one starts from the power of the microprocessor, and asks how one would design a sensor, the answer may be far more complicated computationally, but extremely simple and basic from a mechanical viewpoint. Already we are seeing a move to microsensors that can be built into chips. These depend on thermal and vibrational techniques. Other chemical or tactile effects may be exploited in the near future.

An information technology specialist may offer many other possibilities, but the instrument engineer, be he or she inventor, designer, manufacturer, entrepreneur, or chief executive, should not wait for such suggestions but rather be prepared to think the unthinkable. In the short term, this may appear to be a lessening of the importance of mechanical engineering in instrumentation, but in the longer term it is likely to lead to an enhancement of the instrument engineer's ability to measure the real world.

Some Reflections on Flowmeter Manufacture, Production, and Markets

20.1 INTRODUCTION

I do not claim an expertise in manufacture, but I want to make three points briefly in this chapter and then to enlarge on the last of them.

Market information is usually inaccessible, and there may be a case for research centers to work together to provide a better source of data for industry. It is clear that there is not always an entirely satisfactory information flow, even within companies, between the marketing department and the technical and manufacturing operations.

My first point is, therefore, to suggest the need for better market information both within and outside companies.

My second point is to indicate the need for the flow metering industry, on the one hand, and the science base as it relates to instrumentation, on the other hand, to create an effective means for technology transfer, which in the process will raise the profile of the sector and ensure that government is aware of its importance. Past experience of encouraging collaboration suggests that industry and the science base do not always appreciate the value of working together. Collaboration between industry and the science base should be mutually beneficial in providing an antenna for new information on developing technologies.

The third point is that the production of instrumentation is a special case in which the effect of the production process on the final accuracy of the instrument may, in some cases, be predicted and used to specify the production requirements. Although a special case, it may be important for other products; the ability to measure dimensional precision and variation at each stage of the production may lead to higher quality in other products.

20.2 INSTRUMENTATION MARKETS

The flowmeter has been described as the cash register of the process industry. The following figures give some indication of the turnover. Information on meter markets tends to be so commercially valuable that it is treated as highly confidential. Individual firms are unwilling or, for commercial reasons, unable to reveal their own knowledge. This section, therefore, is gleaned from information in the public domain.

Halsey (1986) obtained a 20% response to a questionnaire sent to 330 companies thought to be using flowmeters. He analyzed the resulting data in various ways. He

obtained the percentage of meters used by industry type.

Fluid	Industry Type	Share of Flowmeters (%)
Liquids	Water	18
	Hydrocarbons	28
	Other chemicals	14
	Food	3
	Other	Negligible
	Total	63
Gases	Air	5
	Steam	12
	Hydrocarbons	12
	Other chemicals	2
	Other	1
	Total	32
Mixtures	Total	5

As shown here, the meters used were predominantly orifice plates:

Orifice plates	56%
Other differential pressure	3%
Variable area	19%
Positive displacement	9%
Turbine	7%
Vortex	1%
Electromagnetic	4%
Ultrasonic	Negligible
Mass	1%

Of these, 66% were owned by large users (defined as having over 600 meters), and only 4% were with small users (with 60 meters or fewer). Of the meters, 83% were used for continuous flow, whereas 8% were used for batch processing; 5%, for custody transfer; and the remaining 4%, for research and development (R&D) or as flow alarms. He concluded that positive displacement meters were in decline, whereas electromagnetic meters were increasing market share. Vortex and mass were increasing, and smaller users were more likely to have a greater proportion of newer types. He made no estimate of the total market but noted that about 70 companies used over 17,000 flowmeters.

No doubt the situation has changed since 1986. More recently, Lawton Smith (1994) has given some overall figures. These are approximately as follows:

Flow measurement	
industry worldwide	\$700–800 million p.a.
Value of oil and gas	
monitored by flowmeters	
• in the United Kingdom	>\$30 billion p.a.
Total of all industrial sectors that depend	
on flow metering	
• in the United Kingdom	\$250 billion p.a.
• worldwide	\$10,000 billion p.a.

These figures should probably be increased by about 5% p.a., suggesting a world-wide flow measurement industry of US \$1 billion or more by the year 2000. Some may suggest that this is a low estimate, but it will depend on the secondary instrumentation, etc., which is included in any estimate.

She used gas meters as an example of utility markets in the three countries that she studied:

Market	Number Produced p.a.
United Kingdom	2 million
France	400,000
Belgium	60,000–80,000

Lawton Smith estimated the following comparative figures for numbers of manufacturers and suppliers:

United Kingdom	200
France	30
Belgium	20

A DTI report (1996) suggested that, of the total instrumentation and control industry in the United Kingdom, about 40% is in process monitoring and control. Of the six major producing countries, breakdown of the total instrumentation and control industry production between them was as follows:

United States	36%
Japan	24%
Germany	8.5%
United Kingdom	7%
Italy	2.5%
France	1%

These figures allow an approximate estimate to be made of the likely market and the competition worldwide. Within the flowmeter market, at the time of writing, Rosemount was probably the dominant producer, with a range of competitors who, typically, were international companies with operations around the world, often each national branch having a particular flowmeter expertise and flowmeter type or application.

20.3 MAKING USE OF THE SCIENCE BASE

Lawton Smith (1994) identified various issues relating to technology transfer in the United Kingdom and other countries. The first was the gap between the products of the research base and the needs of the prospective manufacturer. There is increasing recognition of the need for funding the developments in this gap, and the public purse is often unwilling to do so. This gap could also be reduced by facilitating exchanges of people between industry and the science base. Some may contend that industrial research in the science base may inhibit the generation of new ideas. However, in its place, it is an essential part of ensuring that engineering

and applied science research is focused on the real problems of industry, and this cross-fertilization will probably generate new ideas.

20.4 IMPLICATIONS FOR INSTRUMENT MANUFACTURE

I claim an interest but not an expertise in the manufacture and production of flowmeters and other instrumentation. The comments that follow are derived mainly from observations I have made during visits to companies and from reading and thinking around the subject. Consequently, they are perhaps rather idiosyncratic. However, I have benefited from the books by Hayes et al. (1988) and Slack et al. (1995).

It is widely recognized that the design and implementation of a new product must take account of, and be done in close collaboration with, the production, manufacturing, and marketing functions. Hayes et al. (1988) made this point when they suggested that it is preferable for the upstream product development and the downstream process development to take place linked together by continuous information exchange. Each must be confident enough and prepared to risk changes as they develop a product together. This requires skills, appropriate attitudes, and mutual trust.

Muhlemann et al. (1992) stated, in the context of TQM (Total Quality Management), that "For an organization to be truly effective, every single part of it, each department, each activity, and each person and each level, must work properly together, because every person and every activity affects and in turn is affected by others."

Slack et al. (1995) defined "Quality [as] consistent conformance to customers' expectations." This requires, for the instrument manufacturer, that the design process (possibly based on a computer model of the meter) and the production process lead to the precision, accuracy, and price expected by the user.

20.5 THE SPECIAL FEATURES OF THE INSTRUMENTATION INDUSTRY

Modern instrumentation frequently is defined by a precise analytical expression or algorithm. The output signal should, therefore, be related to the measured parameter with a high level of confidence. This confidence comes not only from the design but also from the quality of manufacture. It is this combination that creates a high quality instrument, in other words, an accurate instrument with a small value of signal uncertainty.

It is, therefore, reasonable to assume that we can deduce from the manufacturing process the effect it has on the parameters that control precision and should be able to identify a method of production of the instrument, which will ensure that the finished product is of specified quality. We should be able to use an error equation for changes in each parameter.

This could be used forward or backward. We might establish a value of precision in measurement of time difference necessary to achieve an accuracy for the flowmeter and design the time measurement system accordingly. Or we may

establish the likely value of time error for a given design and construction and apply this to achieve the uncertainty in flow rate.

20.6 MANUFACTURING CONSIDERATIONS

We review briefly various aspects of flowmeter manufacture relating to these points or emphasised in the literature in relation to instrumentation.

It is essential to think through the whole product development in detail before you start so that problems and changes will be kept to a minimum. (see Finkelstein and Finkelstein 1994 on requirements engineering). Preliminary work must look at the scope for innovation as well as the constraints: commercial, contractual, procedural, or legal.

There must be a clear understanding of product development, together with valid reasons for undertaking it, in order to define:

- ❑ the specification of the system;
- ❑ the management of the development;
- ❑ the documentation of the development process;
- ❑ rigorous project cost management, scheduling, and decisions relating to changes; and
- ❑ parameters by which progress will be measured against predicted time and costs.

Those with a stake in the process need to be identified and involved from the start. The successful development will require excellent communications between the team and those with whom they need to liaise.

This can be achieved by setting up multidisciplinary teams in order to remove the need to transfer information from one function to another (cf. Constable 1994 on concurrent engineering). Thus the sequential system of concept to R&D to development to production planning, etc., is broken down with great savings in terms of time to market. The team given the task of developing a product should be full-time. This may cause staffing problems when first introduced but, in the longer term, should be less intensive in staff time. The team should have a leader to whom the team answers, who also ensures a clear plan and design brief. The result of this is likely to be much increased responsibility pushed down the organization, with beneficial effects on staff motivation.

20.6.1 PRODUCTION LINE OR CELL?

The observations in this chapter are based on flowmeter production methods. It is apparent that much flowmeter production is custom-made, responding to the particular requirements of each customer. Clearly a few flowmeter ranges are produced in large quantities on a production line (e.g., domestic gas meters).

The conclusion of this is that, for many production operations, a cell system will offer a useful way to respond to the custom-made nature of the business but will also provide a more satisfactory working arrangement and a delegation of responsibility for quality, method, and delivery to the leader and members of the cell.

The nature of much instrument production is based on designs of advanced theoretical and experimental refinement. As a result, the considerations of production layout, materials purchase, storage and use, and a fully computerized production control may take second place to design. However, with modern computer control, the opportunity should be taken to optimize the production system and to install a tracking system that will document the details of the production and calibration on every instrument. The assessment of the needs of a particular manufacturer is a task that some universities (e.g., Cambridge) find it useful to give to their able final year students, to the mutual benefit of the student, the university, and the company.

20.6.2 MEASURES OF PRODUCTION

Those involved in the design of flowmeters will appreciate the need to measure in other areas, and the need to do so in the production system should come as no surprise. In many cases, this requires an intelligent appraisal of the system and a clear understanding of what we are seeking to achieve. However, some useful tools such as benchmarking the company's performance against companies in the same business and measuring productivity and other calculable aspects of the production process are available. A brief discussion of these tools follows.

- a. Productivity. Hayes et al. (1988) described an approach that measures improvements in conversion of resources into products. It uses a measure – the single factor productivity – for the ratio of product output to resource input. The sum of these factors, weighted according to the amount of resource used in the product, provided total factor productivity, and its improvement could then be measured with time.
- b. Benchmarking. Hayes et al. (1988) encouraged benchmarking by identifying competitors and noncompetitors who have a superior performance/product and by using trade journals, annual reports, visits, etc., to focus on the reasons for performance differences.
- c. Control charts. Control charts will plot key measures of production with time. They may measure process quality; defective items; process production times, etc.; product precision; and mean time to failure of products.

In the next section, we consider one such measure, particularly appropriate to instrumentation, where a key factor is final accuracy and where the link between production process and accuracy can sometimes be analyzed.

20.7 THE EFFECT OF INSTRUMENT ACCURACY ON PRODUCTION PROCESS

Variation in manufacture, in many cases, is due to wear or operator variation. Although this will also be true for instrumentation, I would like to suggest an additional reason: small variations in production precision and materials. I shall first indicate this by descriptive examples and then offer some simple mathematical examples.

20.7.1 GENERAL EXAMPLES OF THE EFFECT OF PRECISION OF CONSTRUCTION ON INSTRUMENT QUALITY

Let us consider examples from various flowmeters.

- a. Orifice plate. Variation in the sharpness of the leading edge will probably be due to operator variation but may also be due to slight and random variation in the quality of the material.
- b. Variable area flowmeter. Even though the glass tubes are made on a mandrel and should have a well-controlled tolerance, the positioning of the scale will determine the accuracy, and as the device is essentially nonlinear, it will determine the shape of the characteristic. This variation in positioning may be an unavoidable consequence of the process and, being random, may produce a higher quality instrument from time to time.
- c. Positive displacement flowmeter. Variation of the clearance between the rotors and the stator will cause a change in performance. This probably has more to do with wear of the manufacturing process than a random effect.
- d. Electromagnetic flowmeter. The construction of the coil and its positioning relative to the flow tube and electrodes will affect the performance and, to some extent, may be random.
- e. Ultrasonic flowmeter. The placing of the transducer cavities and the resulting position of the transducers will determine the accuracy of the meter for varying flows.
- f. Coriolis flowmeter. The positioning of the sensors on the sensor tube, together with other masses, may cause harmonics resulting in dither and twist, and the consequent imprecision in the measurement of time difference between the two sensing points on the oscillating tube.

In developing a design and production program for a particular flowmeter, the company should be clear about the relationship between the primary construction variables and the resulting performance of the flowmeter. It should be possible to develop a curve for the effect of the precision of positioning on the accuracy of the flowmeter. Assuming a normal distribution for the occurrence of mispositioning and a curve for the relation between mispositioning and the randomness of the signal, the variation of the randomness will result.

20.7.2 THEORETICAL RELATIONSHIP BETWEEN UNCERTAINTY IN MANUFACTURE AND INSTRUMENT SIGNAL QUALITY

We can express the equation of a flowmeter so as to relate the ratio of mass flow to flow signal, to the parameters that result from the theory of its design. Thus, as a simple example,

$$q_m/s = p_1 \times p_2 \times p_3 \times p_4 \quad (20.1)$$

where q_m is the mass flow rate, s is the flow signal from the meter, and $p_1, p_2, \dots$ are dimensional and other factors. A high quality flowmeter will perform to an uncertainty of better than 1%. (We are here ignoring the more complicated relationship that would be modeled computationally.) The values of the parameters, p_1 , etc., will

be affected by flow, installation, temperature, etc., and by manufacture. In general, we may, therefore, write

$$q_m/s = f(p_1, p_2, p_3, p_4) \quad (20.2)$$

or

$$q_m/s = f(\text{design, flow, installation, operating parameters, materials, component precision, manufacturing accuracy, quality of electronics}) \quad (20.3)$$

Assuming a constant design, flow, and installation and a design that minimizes the effect of operating parameters, this simplifies to

$$q_m/s = f(\text{materials consistency, component precision, manufacturing accuracy, electronics quality}) \quad (20.4)$$

Small variations in manufacture can be related to small variation in performance by

$$\delta(q_m/s) = \frac{\partial(q_m/s)}{\partial mc} \delta mc + \frac{\partial(q_m/s)}{\partial cp} \delta cp + \frac{\partial(q_m/s)}{\partial ma} \delta ma + \frac{\partial(q_m/s)}{\partial eq} \delta eq \quad (20.5)$$

where *mc* is used as the symbol for material consistency, *cp* for component precision, *ma* for manufacturing accuracy, and *eq* for electronics quality. Thus, if we can obtain the partial derivative of mass flow/flow signal to, say, materials consistency, we can decide for a certain quality of raw material whether the variation δmc will be acceptable or not in its effect on $\delta(q_m/s)$. It is unlikely that we could work with such general categories as materials consistency, and we will probably need to go to detailed descriptions of the components such as tube dimensions, elastic modulus, etc.

Similar equations will determine cost and life:

$$C = \frac{\text{Cost}}{\text{Unit}} = f(\text{materials consistency, component precision, manufacturing accuracy, electronics quality}) \quad (20.6)$$

This cost equation has similarities to the total factor productivity. In this, we are relating overall cost per unit at output to the input cost of individual components per unit. In the total factor productivity, the object appears to be to relate product output to material and process input. Again we can obtain the sensitivity of the total cost to variation in the cost of individual components by obtaining the differential form

$$\delta C = \frac{\partial C}{\partial mc} \delta mc + \frac{\partial C}{\partial cp} \delta cp + \frac{\partial C}{\partial ma} \delta ma + \frac{\partial C}{\partial eq} \delta eq \quad (20.7)$$

We can now weigh up the cost benefits of reducing material consistency from Equation (20.7) against the loss of performance from Equation (20.5).

We may also be able to write an equation for reliability of the finished product including time to failure.

So we could use a perturbation analysis or a parametric study using random behavior of variables. Slack et al. (1995) suggested that there are various places in which the use of graphical plots is recommended, both to track the change of a variable with time and also to see if there is a link between two variables. If we can measure the variation of the partial derivatives, we should have a prediction of possible performance change in the future.

20.7.3 EXAMPLES OF UNCERTAINTY IN MANUFACTURE LEADING TO INSTRUMENT SIGNAL RANDOMNESS

Case 1: Ultrasonic Flowmeter

The simple equation for volumetric flow rate in the transit time flowmeter is

$$q_v = \frac{\pi D^3 \Delta t}{4 t_m^2 \sin 2\theta} \quad (20.8)$$

where D is the diameter of the pipe, Δt is the time difference for upstream-going and downstream-going pulses, t_m is the mean time of transit used to obtain sound speed, and θ is the angle of the beam to the pipe axis. We can identify the fractional change to the signal due to changes in the parameters by

$$\frac{\delta q_v}{q_v} = \frac{3\delta D}{D} + \frac{\delta \Delta t}{\Delta t} - \frac{2\delta t_m}{t_m} - 2 \cot 2\theta \delta \theta \quad (20.9)$$

Each term indicates the extent to which manufacturing variation will affect the signal size at a particular volumetric flow rate. If we take $D = 100$ mm (noting that errors are likely to be greatest with smaller sizes) and $\theta = 30^\circ$, then the manufacturing error will be as follows:

Term	Assumption	Error
1st	Pipe diameter tolerance < 0.5 mm	$\pm 3 \times 0.5/100$ $= \pm 1.5\%$
2nd	1-ns uncertainty in timing circuits (Table 13.1)	$\pm 1\%$
3rd	Negligible uncertainty in t_m	0%
4th	Angular precision approx. $1.2 \times$ error in radians (assume within 0.01 radians or 1 mm in 100 mm)	$\pm 1.2\%$

Thus variation in manufacture could be up to 3.7% or, taking the root-mean-square, 2.2%. However, this neglects other variation caused by the precise positioning of the transducer mounts in the pipe wall. This will cause a variation in the size of t_m from which sound speed is calculated.

It is unlikely that any of these manufacturing variations will be problematic if kept to within these values, provided that the design of the electrical adjustments allows for such variation.

It is likely that other manufacturing details will be more problematic:

- The orientation of the transducers will, if misaligned, cause a significant reduction in signal.

- b. The matching of the transducers for frequency response will be important.
- c. The mounting of the transducers to ensure constant forward transmission and minimal backward transmission will be essential.

Even though the dimensional changes will alter the calibration, they should not affect the final accuracy. However, (a) might result in the beam being swept out of range of the receiving transducer, and (c) could result in secondary interfering signals reaching the receiving transducer. In either case, the precision of manufacture will have an effect on the ultimate value to which the ultrasonic flowmeter can be calibrated.

Case 2: Coriolis Flowmeter

If we start with the simple equation of the flowmeter (Chapter 17)

$$q_m = \frac{K_s \tau (1 - \omega^2 / \omega_s^2)}{2Kd^2} \quad (20.10)$$

we can identify the possible dependence of the parameters.

K_s (spring constant in twisting) depends on:

- material (quality control),
- wall thickness and consistency (quality control),
- precise dimensions (manufacture),
- positioning of components on tube (assembly and welding);

τ (transit time difference) depends on:

- positioning of sensors (manufacture),
- sensitivity of signal (design),
- electronic circuitry (design and construction);

ω (driving frequency) depends on:

- trueness of sinusoidal excitation (electronic design and construction),
- precise measurement of frequency (electronic design and construction);

ω_s (natural frequency of U-tube in twisting oscillation) depends on details similar to those for K_s ;

d (width of U-tube) may be affected by temperature or other causes of dimensional change in manufacture or operating parameters.

(Damping, etc., if included, would also introduce further dependence.) To illustrate the same point for one important manufacturing feature we refer to Lange et al. (1994), who showed that the calibration constant is sensitive to the positioning of the detector masses. They showed that the dependence of the calibration on the fluid density can be rather strong if the positions of the detector masses are not chosen very carefully. Thus to retain the calibration constant within, say, 0.1% of the target value, the positioning of the detector masses for a pipe of half-length 200 mm may need to be within 0.02 mm. It seems likely that, to achieve the highest quality of Coriolis meter, attention to such detail and precise production techniques will be essential.

Thus we find that manufacturing precision in certain flowmeters can result in two quite distinct effects:

- a. Variation in the calibration constant but no loss in ultimate performance;
- b. Reduced quality of signal due to strength, secondary signals, or harmonics, which limits the ultimate accuracy to which the meter can be calibrated.

20.8 CALIBRATION OF THE FINISHED FLOWMETERS

In most cases where the meters are for use on liquid, a gravimetric system as described in Chapter 4 may be used. In other cases, it may be more convenient to use transfer standards or pipe provers. The accuracy of the calibration facility is an essential part of ensuring the quality and consistency of the product and should be built into the logging system for the documentation of each unit produced.

One such calibration system using a critical nozzle test stand as part of the production line for thermal mass flowmeters was described by Caron (1995). A second rig tested the sensitivity of the meter to temperature. A third obtained the transfer function of the flowmeters.

20.9 ACTIONS FOR A TYPICAL FLOWMETER COMPANY

Management should address some basic considerations:

- a. For current products
 - Are they preeminent technologically?
 - Is the production system as efficient and flexible as possible?
 - Has the market been mapped adequately, and do the products and the production system respond adequately?
 - How does the company's products compare with those of competitors?

Assuming that the questions in (a) have been answered to give an optimum market sensitivity, the gap in market coverage should be clear.

- b. For new products
 - What product should be added to the range that most complements existing production?
 - What other or competing manufacturers' products or designs are entering the market?
 - What is the critical time to market?

The existing industry competitors should be identified, and the weaknesses of their products, which will be met by strengths in any new designs, should be assessed. In some cases, the instrument type may be a substitute for a previous one. For instance, the orifice plate is likely to give way to other devices in many industries, and the Coriolis will become more used in certain applications.

The manufacturer must also address the following details of the production process:

- ☐ Style of manufacture,
- ☐ Computer-based information and communication to control and expedite product documentation, from material selection to final calibration and shipping,
- ☐ Buy in or make on site, and
- ☐ Quality control.

Future Developments

21.1 MARKET DEVELOPMENTS

Lawton Smith (1994, cf. Kinghorn 1988) gave some overall figures (cf. Chapter 20) for the flow measurement industry worldwide, which, with likely growth, suggest that by the year 2000 it could be in excess of US \$1 billion.

With this growing market in mind, we look at the new challenges that face the flowmeter engineer, the current and future devices that may provide solutions, the implications of information technology, and new production methods. The chapter concludes with some suggestions for the way ahead, after a brief review of comments that I made on future developments in an earlier book (Baker 1988/9).

21.2 EXISTING AND NEW FLOW MEASUREMENT CHALLENGES

Oil Exploration and Processing

The oil industry, dealing as it does with high value products, is likely to continue to ask for more accurate meters capable of operating in adverse conditions, in multi-phase flow, and in subsea installations.

Corneliussen (1991) of the Amoco Norway Oil Company described field experience with Hod metering, which was the first small unmanned production platform in the Norwegian sector of the North Sea and started production in September 1990. Gas and liquid were transported in a three-phase pipeline 13 km to another platform from which Hod was operated. He described problems that appeared to have stemmed from trapped air and gas in the pipeline and from a highly acidic well that eventually caused the destruction of some of the meters. This is one example of the increasingly difficult environment in which North Sea instrumentation must work. Subsea metering of multicomponent flows is likely to become more common.

At the other end of the process, the measurement of fuel and flare gas for CO₂-tax is discussed by Raustein and Fosse (1991) of the Norwegian Petroleum Directorate. Existing measurement uncertainty for fuel gas by either orifice plates or insertion turbine meters is estimated to be ± 2 –5%. By using insertion turbine meters, flow sampling and thermal techniques, and ultrasonic flowmeters, overall accuracies for flare gas metering are estimated to be ± 5 –10%. In the future, Raustein and Fosse expected the quantities to be subject to taxation.

Between these two extremes of well extraction and gas discharge, there is the detailed audit, monitoring, and control of each stage of the production process, and

the consequent requirement for accurate metering to cope with many and varied fluids and installations.

Utilities

The last 10 years have seen major developments in the measurement of water and gas for domestic purposes. The fluidic flowmeter described by Sanderson (1994) is an elegant solution to the need for a wide-ranging no-moving-part domestic water meter.

The ultrasonic gas meter is another example of the developments in utility flow measurement in the last 10 years.

Kinghorn (1988), in the context of domestic water metering, mentioned remote reading using mains-borne radio and telephone networks. Much development of alternative communication systems has taken place in the past 10 years.

Mass

Although the Coriolis meter is proving to be a very accurate device for high value liquids, the availability of a mass meter for gases is still a problem because all existing meters, apart from the Coriolis, tend to be dependent on gas type. Ultrasonic and thermal meters appear to lead the field at present.

Multiphase

Multiphase continues to be a challenge for the industry, which has not been solved to an adequate extent. Of particular importance is the need to measure the flow of moist gases and of steam in all its states.

Proving and Calibration

The compact prover and packages of sonic nozzles offer possibilities for site calibration.

Kinghorn (1996) discussed the measurement of fluid properties essential to the flow measurement operation and the problems involved and suggested the use of meters capable of compensating for changes in fluid properties. For example, gas composition can cause a 0.28% discrepancy (Kinghorn 1988).

Local Velocity Measurement

A whole area outside the scope of this book is local measurement of flow to aid the design of fluid-handling equipment, both single and multiphase, ranging from ducts and valves in single-phase flow systems to mixers and reactors in biotechnology plant.

Research, Development, and Data Acquisition

A continuing theme for at least 30 years has been installation effects, and it is likely to remain so to some extent, where flowmeters, which are sensitive to flow profile, are in use. Computational methods of predicting the performance of flowmeters is likely to become more accurate.

Kinghorn (1996) saw the possibility of reducing capital investment resulting from flowmeter installation by developing meters less susceptible to installation:

- ❑ less affected by pulsation,
- ❑ less affected by upstream conditions,

- ❑ capable of multiphase flow measurement, and
- ❑ unaffected by condensate

so that the flow conditioning becomes much less costly.

21.3 NEW DEVICES AND METHODS

21.3.1 DEVICES PROPOSED BUT NOT EXPLOITED

In this section, devices that, to my knowledge, have not been exploited are mentioned.

The many patents in this area demonstrate the activity in coming up with new ideas. Digests of these are available, for instance in the *Journal of Flow Measurement and Instrumentation*. No attempt has been made in this book to review and assess their relative merits. I know of the devices that follow from my previous experience or from recently published articles.

Brain et al. (1975) reported work on a gas ionization flowmeter in which the ionization was due to a recessed and extended circumferential foil source of strontium 90/yttrium 90 emitting beta radiation. The collection system removed ions in a short axial length to create a striation in the ionized gas that was sensed by a drop in current, and, hence, the transit time was obtained. The measurements were for 3–30 m/s, and $\pm 1\%$ was claimed for most of the range. It was considered unsuitable for gases with water vapor, oil mist, or solid particles, and, possibly, attitudes to nuclear material make it unattractive.

Al-Rabeh and Hemp (1981) described a flowmeter that used the electrostatic charges in insulating fluids. By setting a series of electrodes in the wall of the pipe, each forming a ring, and by connecting these alternately, the passing of charge created a sinusoidal signal, the frequency of which was proportional to the flow rate. Nakano and Tanaka (1990) appear to have used a similar concept for oil hydraulic pipelines.

Ellis et al. (1993) did further work on this type of meter to assess its value for two-phase flows and claimed that it was possible to obtain the individual flow rates of oil and gas in their rig. The meter consisted of two parts, one similar to that of Al-Rabeh and Hemp (1981) and the second of two helical electrodes that obtained the capacitance of the mixture and hence the void fraction. The void fraction measurements appeared to be subject to an error of as much as 30%, although the total flow rate error was within about $\pm 7\%$ rate (cf. Xingbin et al. 1996 and Yiming and Haibo 1996).

Kolmakov and Korolev (1990) described an Alfvén-wave flowmeter for those interested in novel forms of flowmeter suitable for liquid metal flow measurement!

Flowmeters depending on nuclear magnetic resonance (NMR) have been suggested over the last 20 years or so. To my knowledge, there is none on the market at present. Gol'dammer et al. (1990) showed that the NMR signal amplitude of a gas-liquid mixture in diamagnetic fluids depended on the liquid content regardless of the physicochemical properties of the medium and conditions of flow. This linearity is destroyed if paramagnetic centers are added to the fluid. Nuclear magnetic resonance has also been used in liquid-gas flows to measure the mean velocity of the liquid averaged over the liquid volume and also the average of the liquid fraction

(Kruger et al. 1996). The combination provided the mass flow of the liquid. In tests using water as the liquid, they claimed to determine the mass flow to within $\pm 5\%$.

Morgan and Aliyu (1993) described an ionic flowmeter for measuring small rates of gas flow from $0.30 \mu\text{g/s}$ to 0.20 mg/s . They claimed that the meter was best for dry nonexplosive gases at a pressure of $4 \times 10^2 \text{ Pa}$. The tube had an ion discharge at the center. Ions were collected by ionization chambers up- and downstream of the discharge, and under flowing conditions the currents were unequal. The internal shape of the duct consisted of two chambers larger than the pipe joined by a small duct within which the discharge occurred. The plot of mass flow against differential current looked linear and promising. The meter did not directly measure mass flow and required calibration. Each gas required separate calibration, and humidity could also cause changes.

Joshi (1991) described a surface-acoustic-wave (SAW) flow sensor. The oscillation frequency is temperature-dependent, and the element is heated above ambient, and placed in the flowing gas so that convective cooling will make the frequency flow-rate-dependent.

O'Rourke (1996) discussed the MTI gas meter, which uses an undulating wave-shaped membrane that essentially forms traveling compartments of a positive displacement sort and appears to give good performance. It is an interesting new concept that may well have wider application (cf. Chapter 9).

A rather different aspect of domestic gas metering is the frequency of sampling necessary to achieve required accuracy (Braathen 1996). Braathen claimed that sampling every 16 seconds (a conservative figure as 32 seconds may be enough) or less gave total gas passed to better than 0.1% and that, by using random intervals, fraud could be avoided. The advantage is gain in battery life.

Herzog and Roth (1990–1) described the principle of a differential pressure sensor using optical fibers in a Mach-Zehnder interferometer configuration. They used it to measure the pressure difference between two tubes across the pipe flow, the upstream one with a pressure tapping facing upstream and the downstream one with a tapping facing downstream – a sort of averaging pitot tube with the upstream and downstream tappings in two different tubes.

Strunck et al. (1996) described an intriguing use of four diode lasers – two beams focused and intersecting in the boundary layer, and two intersecting on the axis of the bell-mouth entry. The lasers heterodyned at a controlled frequency, and the signals were picked up by four pin diodes on the other side, which gave velocity at two points and, hence, the profile to $\pm 0.15\%$ for $500\text{--}5,000 \text{ m}^3/\text{h}$.

Baker and Higham (1992) suggested that by using multiple bypasses in a flowline, each of which contained a flowmeter, the flow in the main could be measured. The interesting outcome of this type of approach is that the multiple flowmeters can be mass produced, possibly using microtechniques, and if one fails, it can be replaced without either ceasing to monitor the line flow or closing down the line. A computer can monitor all flows and set an alarm when one needs to be removed for replacement. Large lines could be monitored by small meters, with a simple and robust contraction in the line to create the pressure drop to drive the flow through the bypass meters.

Another variant of this would be to divide the flow in a large main into individual paths through individual identical meters, each of which could be replaced without closing down the flow. This could lead to a calibration facility in which a large flow

is measured with many small identical flowmeters, each capable of calibration on a small, but highly accurate, flow facility.

A new method that benefits from the great power of computing uses pattern recognition to obtain superficial gas-liquid flow rates (Beg and Toral 1993). They appear to have suggested that the technique could be extended to oil-water-gas three-phase flows by adding pressure sensors and an orifice plate with differential pressure, capacitance, and gamma-ray sensing; introducing a new hydrodynamic system for scaling pipe diameter and fluid properties; and using neural network pattern recognition techniques.

Bignell (1996b) discussed a new gas flow standard consisting of two hemispheres with two inner perforated hemispheres. The hemispheres are separated by a flexible diaphragm. Gas enters one hemisphere until the diaphragm rests on the perforated hemisphere. The gas flow is then reversed through a valving system.

21.3.2 NEW APPLICATIONS FOR EXISTING DEVICES

Kinghorn (1996) commented on the new lease of life for the venturi because it can cope with wet gas flows better than the orifice, and the need to extend the flow range data for venturis.

Krafft et al. (1996) have investigated further the use of the electromagnetic flowmeter to measure both the velocity of the continuous phase and the bubble velocity.

Guilbert and Sanderson (1996a) described an ultrasonic flowmeter that obtained both flow and density from ultrasonic means to obtain mass flow of liquids.

Guilbert et al. (1996) described a clamp-on flowmeter for the measurement of low liquid flows in small tubes. The method depends on the interaction of ultrasound with temperature changes.

The idea of using electromagnetic sensing for a fluidic flowmeter has now been applied by Sanderson (1994). Some years ago, I considered the idea of combining an electromagnetic flowmeter with a vortex-shedding meter. Such a combination would offer interesting possibilities in condition monitoring.

The use of cross-correlation techniques was mentioned in Chapter 13 with regard to ultrasonics, but its use with other sensors has also been mentioned. Roach et al. (1995) used two gamma-ray transmission gauges to obtain oil, water, and gas flow rates, presumably by some form of cross correlation. Their tests were in the field in a line from test manifold to test separator with water cut from 25% to about 95%. The device operated on vertical flows, both up and down, with uncertainty reckoned at 4% for liquids, 7.5% for oil, 4.5% for water, and 7.9% for gases with water cut to 3.6% relative, including separator and flowmeter errors. Development of an intelligent cross correlator has been reported recently by Yang and Beck (1997, cf. Yang et al. 1996a).

Yang et al. (1996b, cf. Yang 1996) proposed capacitance and charge sensing for flow measurement in oil pipelines. They also appear to be considering capacitance and ultrasonic tomography to give three-component flow information.

21.3.3 MICROENGINEERING DEVICES

Nguyen (1997) has provided a very useful review of micromachined flow sensors, which he claimed would amount to 19% of the MEMS (microelectromechanical

systems) market by 2000, over US \$2.5 billion. The first sensor based on silicon technology appeared in the 1970s. The result of this technology is new applications such as the measurement of acoustical flow in a thermal microphone. Nguyen divides the sensors into two categories: nonthermal and thermal. The nonthermal sensors use drag force on a cantilever, pressure drop measured with capacitive or piezoresistive sensors, and Coriolis force. Nguyen also suggested that a microturbine and a bistable fluidic element are also possible. The thermal sensors use hot-wire/hot-films, calorimetric sensors similar to the thermal flowmeter or time-of-flight techniques. The reader is referred to the article, which gives useful diagrams and characteristics of the meters. Nguyen suggested that an integrated analogue-to-digital converter on the chip would be useful. The combination of these devices with bypass methods would seem to be very powerful.

Legtenberg et al. (1991) described a resonating microbridge mass flow sensor, which was suspended inside a micro flow channel. Thin film technologies and micromachining were used for the fabrication of the sensor wafer and a cap wafer with opposing V-grooves. A low-temperature glass-bonding technique was used to assemble the wafers, allowing for feed-through of the electrical connections. Measurements showed sensitivities of the resonance frequency of several kHz/cm³ for nitrogen gas flows at average temperature elevations of the microbridge in the range 20–100°C. Figure 21.1 gives a clear view of the device. Flow of gas past the heated structure will change the resonant frequency. Thermal excitation and piezoresistive detection are used to induce and measure the vibrations.

Jitschin et al. (1995) described a very small orifice with a nominal diameter of 100 μm and operating in the molecular flow regime for applications in ultra-high-vacuum technology. van der Wiel et al. (1993) developed a silicon sensor and made it using industrial bipolar process and micromachining. It depends on convective heat transfer resulting from the liquid flow. Because it is designed for use in biomedical flows up to 2 m/s, the temperature differences

are 1–10°C, and the sensitivity is 10 mV/(m/s). The overall dimensions of the chip are 2.0 mm $\times$ 1.6 mm $\times$ 0.4 mm, which includes the heater and the diodes for temperature measurement.

Another low cost silicon sensor (Nguyen and Kiehnscherf 1995) produced through microtechnology for mass flow measurement of liquids and gases is shown in Figure 21.2. Its operation is based on heat transfer to the flow from heated resistors in the flowing fluid and the consequent temperature difference of the sensors in the flow. The range is 0–500 ml/min for gases and 0–10 or 0–500 ml/min for liquids.

van Oudheusen (1991, 1992; a very useful and thorough treatment of this developing area to which the reader is referred) developed a direction-sensitive

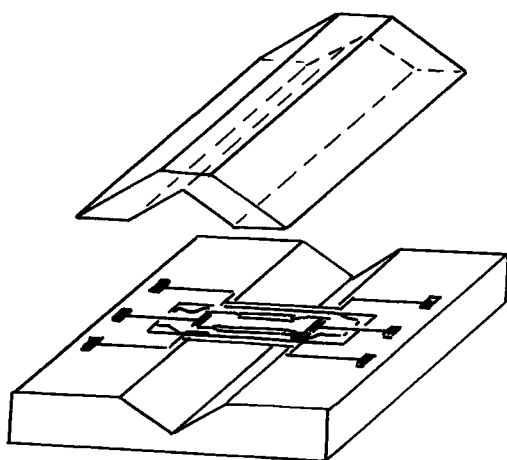


Figure 21.1. Resonating microbridge mass flow sensor. The channel height is 420 μm (reproduced from Legtenberg et al. 1991 with permission of Elsevier Science Ltd.).

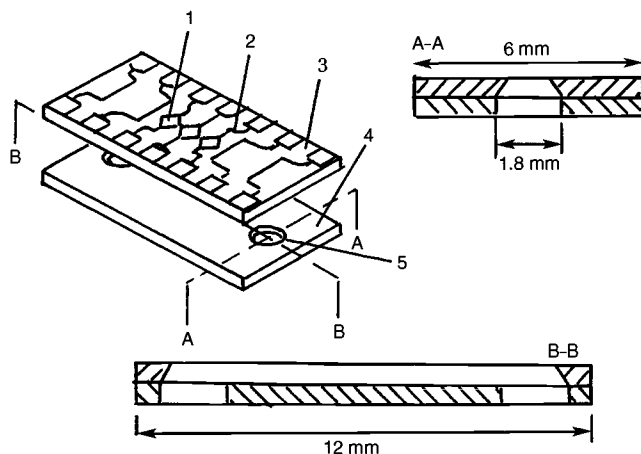


Figure 21.2. Thermal flow sensor: 1, polysilicon resistances; 2, aluminum conductor; 3, silicon chip; 4, Pyrex plate; and 5, flow inlet and outlet (reproduced from Nguyen and Kiehnscherf 1995 with permission of Elsevier Science Ltd.).

microelectronic thermal sensor. The advantage of silicon thermal flow transducers is that complete integration in standard silicon technology is possible. There is the possibility of including electronics for signal conversion on the sensor device to achieve a smart sensor. The sensor could be shielded by a thermally conducting cover layer so that direct contact between the sensor and the fluid was avoided. The operation was based either on heat transfer from the sensor or flow-induced temperature gradients.

21.4 NEW GENERATION OF EXISTING DEVICES

Ten years ago in an introductory guide to flow measurement (Baker 1988/9), I suggested that the future developments would lie in the areas of

- a. nonintrusive, noninvasive, clamp-on, particularly ultrasonic, techniques;
- b. optical-sensing methods with existing devices;
- c. intelligent, smart, self-monitoring and monitoring of the plant performance; and
- d. multiphase flow measurement.

Of these, it is probably true to say that the most significant advance has been in the area of (c), where the electronic engineers have continuously exploited the possibilities of the extremely powerful microprocessor technology.

I suggested the possibility of an ultrasonic master meter that clamped on to a pipe and, through doppler, transit-time, and correlation signals, together with nondestructive testing of the wall condition and material, could provide a highly precise measurement. This remains a line of development that might provide an in situ calibration for a much simpler meter depending on quite different effects. For instance, the pipe vibration, or thermal effects, might be used as an indicator of flow and calibrated with such an all-purpose master flowmeter.

Optical methods have clearly moved on substantially in 10 years, although their direct use in process sensors is an area for further development. With Morris (1983), I reviewed the suitability of flowmeters for optical sensing, and more recently Dickinson (1988) considered design of optical flowmeter sensors. His paper has examples of variable area, turbine, thermal, and cross correlation. Dickinson suggested that a noninvasive pickup to sense rotation in turbine and positive displacement without any moment/movement would be useful, as would the development of a vortex meter using optical methods for steam.

I have already commented on the needs for more development in the area of multiphase, particularly subsea, flows.

The new areas that I would add to the preceding list are

- Mass meters for gases. I have already referred to this and suspect that it lies in the improvement of thermal and ultrasonic methods, possibly linked together.
- Microsensors. Here the possibilities appear to be very exciting and, if used as bypass flowmeters, very extensive.

21.5 IMPLICATIONS OF INFORMATION TECHNOLOGY

21.5.1 SIGNAL ANALYSIS

The idea that the signal, far from being smoothed, contains data that can reveal information about the meter and the process (Amadi-Echendu and Higham 1990, Amadi-Echendu and Hurren 1990) is an important insight. It suggests that the noise may carry a fingerprint of the meter and may be interpreted by powerful computational methods to indicate change, possibly correction factors and indeed many other factors. However, if this is to be fully realized, it is essential that models linking the flowmeter/flow system behavior with the noise characteristics be developed, to give a firm theoretical basis. Cheesewright et al. (1998) have provided an appropriate caution.

21.5.2 REDESIGN ASSUMING MICROPROCESSOR TECHNOLOGY

An intriguing question to ask is: what would have been the design of flowmeters, which are now dominated by mechanical engineering, if the power of the microprocessor had been available? For instance, the "fingerprint" of the vibration of a pipe might be stored in the microprocessor for various flow rates and be used as a look-up table, but the actual sensor might be a simple vibration sensor.

21.5.3 CONTROL

There seems little doubt that the extraordinary developments in control, and the increasing flexibility will continue in the next 10 years or more.

For instance, by collecting data on correct plant operation, together with models of the plant flows, and by checking current flows against the data and model, any plant changes (e.g., occurrence of leaks) can be identified (Valenti 1995).

21.5.4 RECORDS, MAINTENANCE, AND CALIBRATION

Yeung et al. (1994) reported on a detailed survey of the use of insertion meters in collaboration with four water utilities and six consultancy companies in the United Kingdom. There were deficiencies in records, equipment maintenance and calibration, work instructions, and training. There was also a difference in perception between management and the technicians, and an ignorance about the ownership and maintenance of records.

There is clearly a need for training in essential information such as standard documents and their requirements. This is an area where modern information and communication methods, particularly linked with modern control, should ensure that much documentation is automatically produced. Gallagher (1992) also raised the issue of certification of technicians.

21.6 CHANGING APPROACHES TO MANUFACTURING AND PRODUCTION

This book has not pretended to give guidance in the area of manufacturing and production. However, a key change in recent thinking has been the use of cell manufacture and the consequent increase of ownership and responsibility by the team in the cell. In the area of instrumentation, I have suggested that there is room for careful understanding of the links between production methods and instrument accuracy. It is an area where precise measurement of the production process, and of the product as it moves through the process, is needed and where new methods are likely to be developed.

21.7 THE WAY AHEAD

21.7.1 FOR THE USER

The competition in the market place is fierce, and the products vary in performance and quality. Time spent, and advice sought, on the best device for a particular application may save much time and money in the next 10 years.

21.7.2 FOR THE MANUFACTURER

The ability to tailor the product to the user's needs, the opportunity for the user to send an order by modern communication (e-mail, etc.), and the ability to deliver this special order in days or even hours, is possible, and some manufacturers are achieving remarkable order-to-delivery times. Staying in business requires that steps be taken to achieve this. In the process, it is likely to lead to the rethinking of many aspects of production to the benefit of the company, and if all the work force is involved, to their benefit also.

21.7.3 FOR THE INCUBATOR COMPANY

New start-up companies run by an entrepreneur with a brilliant new concept are to be encouraged, but in this competitive market, they will need support. Such support exists and is likely to be more widely available as time passes. At its best,

it may provide space on a relatively low rent and for a manageable forward time commitment, advice on legal and other matters, advice on financial needs, and possibly training in the manufacturing, management, finance, and marketing areas with which a scientist/inventor may be less familiar.

21.7.4 FOR THE R&D DEPARTMENT

This book has brought much of the literature together, but I would not pretend that it is remotely exhaustive. The R&D department will need to work with the whole company to coordinate the new product needs. However, from the other side, the availability of exhaustive literature searches to find new ideas, should raise many possibilities. Developments in the science base should also be explored, and joint projects should be initiated. Collaboration with the science base can be mutually beneficial, providing the industrial partner stays the course and is not put off by the very different culture, particularly of the university research interface.

21.7.5 FOR THE INVENTOR/RESEARCHER

The power of modern microprocessing and micromechanics suggests that a fresh look should be taken at every flow measurement application to see if extremely simple sensors can provide signals capable of interpretation.

21.8 CLOSING REMARKS

Flow measurement is a large and varied subject. It uses almost every area of classical physics in the many and varied sensors. It attempts to measure mean flows in pipes downstream of complex pipework. It attempts to cope with highly adverse conditions both of fluid and of environment. From long experience, I have found that most of the questions posed in industrial workshops are unanswerable, and that the solutions to the questions would merit a major research program.

It is also an increasingly competitive field for the manufacturer, in which any sort of collaboration is viewed with suspicion. It is, therefore, refreshing to find Letton (1994) making a plea for cooperation in product development in this high technology industry.

There is, however, a role for the science base, in terms of the development of new devices, the application of more sophisticated signal processing, and the assembly of even more powerful control techniques.

There is another area where the leading university groups can bring a fresh insight and an on-going collaboration. Those who develop their expertise in the operation of industry can provide an independent and critical view on the modernization of the whole production function: to reduce waste, to improve quality control, to place more responsibility and initiative on the production cell, to improve the automatic acquisition of documentation on the product flow, and to improve responsiveness and speed of delivery to the customer.

I hope that this book will provide a significant contribution, firstly to the fulfilment of those people whose work is with flowmeters, from invention to application; secondly to the quality of the product; and thirdly to the financial success of the enterprises, so that the industrial flows of fluids of all sorts will, as a result, be measured more precisely and reliably.

Bibliography

The list of books on flow measurement is lengthy, and I am sure that this one is not complete, but it represents those books that I have come across. That they are included here is neither a recommendation nor otherwise. Each author provides a particular emphasis: practical, theoretical, fluid, bulk measurement, probes, anemometry.

- Baker, R. C. (1988/9) *An Introductory Guide to Flow Measurement*. London: Mechanical Engineering Publications.
- (1996) *An Introductory Guide to Industrial Flow*. London: Mechanical Engineering Publications Ltd.
- Benard, C. J. (1988) *Handbook of Fluid Flowmetering*. Morden, England: The Trade & Technical Press.
- BS 7405 (1991) *Guide to the Selection and Application of Flowmeters*. London: British Standards Institution.
- Cascetta, F., and Vigo, P. (1988) *Flowmeters – A Comprehensive Survey and Guide to Selection*. Research Triangle Park, NC: ISA Publishing.
- Cheremisinoff, N. P., and Cheremisinoff, P. N. (1988) *Flow Measurement for Engineers and Scientists*. New York: Marcel Dekker.
- Danen, G. W. A. (Ed.) (1985) *Shell Flowmeter Engineering Handbook*, 2nd ed. New York: McGraw-Hill.
- DeCarlo, J. P. (1984) *Fundamentals of Flow Measurement*. Instrument Society of America.
- Endress and Hauser (1989) *Flow Handbook*. Switzerland: Flowtec AG (English edition).
- Fowles, G. (1993) *Flow Level and Pressure Measurement in the Water Industry*. Butterworth Heinemann. (Series editor B. E. Noltink in association with the Institute of Measurement and Control).
- Furness, R. A. (1989) *Fluid Flow Measurement*. Harlow U. K. Longman. (In association with The Institute of Measurement and Control.)
- Furness, R. A., and Heritage, J. E. (1989) *The Redwood Flowmeter Directory*. London: IBC Technical Services.
- Hayward, A. T. J. (1977) *Repeatability and Accuracy*. London: Mechanical Engineering Publications.
- (1979) *Flowmeters: A Basic Guide and Source Book for Users*. London: Macmillan.
- Hogrefe, W., Kirchof, U., Mannherz, E., Marchewka, W., Mecke, U., Otto, F., Rakebrandt, K.-H., Thone, A., and Wegener, H.-J. (1995) *Guide to Flowmeasurements*. Gottingen: Bailey-Fischer & Porter GmbH.
- Jones, F. E. (1995) *Techniques and Topics in Flow Measurement*. New York: CRC Press.
- Linford, A. (1961) *Flow Measurement & Meters*, 2nd ed. London: E. & F. N. Spon.
- Miller, R. W. (1996) *Flow Measurement Engineering Handbook*, 3rd ed. New York: McGraw-Hill.
- Noltink, B. E., Ed. (1988) *Instrumentation Reference Book*. London: Butterworths.
- Ower, E., and Pankhurst, R. C. (1966) *The Measurement of Air Flow*, 4th ed. London: Pergamon Press.
- Scott, R. W. W., Ed. (1982) *Developments in Flow Measurement – 1*. London: Applied Science Publishers.
- Shercliff, J. A. (1962) *The Theory of Electromagnetic Flow-Measurement*. Cambridge: Cambridge University Press.

- Spink, L. K. (1978) *Principles and Practice of Flow Meter Engineering*, 9th ed. Foxboro MA : The Foxboro Co.
- Spitzer, D. W. (1990) *Industrial Flow Measurement*. Instrument Society of America.
- Spitzer, D. W., Ed. (1991) *Flow Measurement*. Instrument Society of America.
- Upp, E. L. (1993) *Fluid Flow Measurement*. Houston: Gulf Publishing.

A Selection of International Standards

In most cases I have omitted full details and directed the reader to the document name or number. This is partly for conciseness and partly because continual update results in new dates for the standards.

Organizations involved in the development of standards include: International Standards Organization (ISO), International Electrotechnical Commission (IEC), Organisation Internationale de Métrologie Légale (OIML), American Gas Association (AGA), American National Standards Institute (ANSI), American Petroleum Institute (API), American Society of Mechanical Engineers (ASME), British Standards Institution (BSI), CEN/CENELEC European Standards, Institute of Petroleum (IP), and Instrument Society of America (ISA).

General

ISO 4006:1991 *Measurement of fluid flow in closed conduits – Vocabulary and symbols*, Bilingual edition (BS 5875).

See also ASME MFC-1M-(R1986)

Uncertainty

ISO 5168:1978 *Measurement of fluid flow – Estimation of uncertainty of a flow-rate measurement* (BS 5844).

See also ISO/DIS 11631 and ASME MFC-2M-(R1988)

Selection

BS 7405:1991 *Guide to selection and application of flowmeters for the measurement of fluid flow in closed conduits*. AMD 8879 (1995).

Calibration

ISO 4185:1980 *Measurement of liquid flow in closed conduits – Weighing method* (BS 6199-1).

ISO 7278 *Liquid hydrocarbons – dynamic measurements. Proving systems for volumetric meters* (BS 6866).

ISO 8316 *Measurement of liquid flow in closed conduits using weighing and volumetric methods. Method for measurement by collection of the liquid in a volumetric tank* (BS 6199).

ISO 7066 *Assessment of uncertainty in calibration and use of flow measurement devices*.

ISO 8222 *Petroleum measurement systems – Calibration – Temperature corrections for use with volumetric reference measuring systems*.

ISO 8316 *Measurement of liquid flow in closed conduits – Method by collection of the liquid in a volumetric tank* (BS 6199-2).

ISO 9368 *Measurement of liquid flow in closed conduits by the weighing method*.

In situ Verification

ISO 2975 *Measurement of water flow in closed conduits – Tracer methods* (BS 5857).

ISO 3354 *Measurement of clean water flow in closed conduits – Velocity-area method using current-meters in full conduits and under regular flow conditions*.

ISO 3966 *Measurement of fluid flow in closed conduits – Velocity area method using Pitot tubes* (BS 1042-2.1).

ISO 4053 *Measurement of gas flow in conduits – Tracer methods* (BS 5857-2.1).

ISO 7194 *Measurement of fluid flow in closed conduits – Velocity-area methods of flow measurement in swirling or asymmetric flow conditions in circular ducts by means of current-meter or Pitot static tubes* (BS 1042-2.3).

See also IP publications on proving

Orifice, Venturi, and Nozzles

ISO 5167-1:1997 *Specification for square-edged orifice plates, nozzles and venturi tubes inserted in circular cross-section conduits running full*. (BS 1042 Section 1.1).

ISO 2186 *Fluid flow in closed conduits – Connections for pressure signal transmissions between primary and secondary elements*.

ISO/TR 3313 *Measurement of pulsating flow in a pipe by means of orifice plates, nozzles or venturi tubes* (in particular in the case of sinusoidal or square wave intermittent periodic-type fluctuations).

See also BS 1042, *API Manual of Petroleum Measurement Standards* Chapter 14.3, ASME MFC-3M-1989, etc.

Critical Flow Venturi Nozzles

9300:1990 *Measurement of gas flow by means of critical flow venturi nozzles*.

ANSI/ASME MFC-6M-1987 *Measurement of gas flow by means of critical flow venturi nozzles*.

Variable Area

BS ISO 11605:1995 *Paper and board. Calibration of variable-area flowmeters*.

Positive Displacement Meters

ISO 2714 *Liquid hydrocarbons – Volumetric measurement by displacement meter systems other than dispensing pumps*.

See BS for information on diaphragm meters and rotary displacement meters for gas. See also IOML.

Turbine Meters

ISO 2715 *Liquid hydrocarbons – Volumetric measurement by turbine meter systems.*

ISO 9951 *Measurement of gas flow in closed conduits – Turbine meters.*

ANSI/ASME MFC-4M-1986 *Measurement of gas flow by turbine meters.*

AGA *Measurement of fuel gas by turbine meters*, Transmission Measurement Committee Report, No 7.

ANSI/ISA RP31.1 *Recommended practice specification, installation and calibration of turbine flowmeters.*

ANSI/API 2534 *Measurement of liquid hydrocarbons by turbine meter systems.*

Vortex Meters

ISO/TR 12764:1997 *Measurement of fluid flow in closed conduits – Flowrate measurement by means of vortex shedding flowmeters inserted in circular cross-section conduits running full.*

Electromagnetic Flowmeters

ISO 6817:1997 *Measurement of conductive liquid flow in closed conduits. Method using electromagnetic flowmeters.*

ISO 9104:1991 *Measurement of fluid flow in closed conduits – Methods of evaluating the performance of electromagnetic flowmeters for liquids.*

Ultrasonic Flowmeters

ISO/TR 12765:1998 *Methods using ultrasonic transit-time flowmeters.*

AGA Transmission Measurement Committee Report No 9, *Measurement of gas by multipath ultrasonic meters*, June 1998.

ASME MFC-5M-1985 *Measurement of liquid flow in closed conduits using transit-time ultrasonic flowmeters.*

See also BS 4331 and VDE/VDI

Thermal Mass Flowmeters

ISO/CD-14511:1998 *Measurement of fluid flow in closed conduits – Thermal mass flowmeters.*

Coriolis Flowmeters

ISO 10790:1998(E) *Measurement of fluid flow in closed conduits – Coriolis mass flowmeters.*

Probes

ISO 3966 *Method using pitot-static tubes.*

ISO 7145 *Determination of flowrate of fluids in closed conduits of circular cross-section – Method of velocity measurement at one point of the cross-section (BS 1042-2.2).*

Modern Control Systems

ISO 7498-1:1995 *Information technology open systems interconnection basic reference model.*

Some Other Industry-Specific Standards*Petroleum*

API Manual of Petroleum Measurement Standards and other API documents.

Water

ISO 4064 *Measurement of flow in closed conduits.*

BS 5728 *Flow of cold potable water in closed conduits.*

ISO 7858 *Measurement of water flow in closed conduits.*

ANSI/AWWA (American Water Works Association) documents.

Gas

ANSI/CGA (Compressed Gas Association) documents.

ANSI/UL (Underwriters Laboratories) documents.

Road Vehicle Testing

ISO 7440 *Road vehicles – Fuel injection equipment testing.*

(Sources used to compile this list are: ISO and BS indexes, Institute of Measurement and Control Yearbook 1998, Upp 1993).

Conferences

This list does not include some of the major professional institution conferences of which flow measurement is a part, and that are run by ASCE, ASME, ISA, AGA, etc. Nor does it include all the conferences that impinge on flow measurement but that cover much wider areas and for which it is not necessarily central (e.g., Gas Research Conferences).

Proceedings of a Symposium on Flow Measurement in Closed Conduits, National Engineering Laboratory, Scotland, 1962.

Institute of Measurement & Control Symposium on the Measurement of Pulsating Flow: 1970.

Proceedings of the International Conference on Modern Developments in Flow Measurement, England: Harwell (Published 1972 by Peter Peregrinus Ltd.), 1971.

Fluid Dynamics Measurements (from Fluids Engineering Conference, Pittsburgh, PA, and Winter Annual Meeting, Washington, DC), ASME Fluids Division, 1971.

Conference on Fluid Flow Measurement In The Mid 1970's, Scotland: National Engineering Laboratory, 1975.

FLO-CON 77 Proceedings of the Symposium – The Application of Flow Measuring Techniques, Brighton, UK: 1977.

FLOMEKO 1978 – Flow Measurement of Fluids, North Holland, Amsterdam: 1978.

Proceedings Transducer 77 Conference Flow Measurement Session, Wembley, London, 1977.

Measurement in Polyphase Flows, Winter Annual Meeting, ASME Fluids Engineering Division, San Francisco, CA, 1978.

NEL Fluid Mechanics Silver Jubilee Conference, East Kilbride, Glasgow, Scotland: 1979.

Flow, Its Measurement and Control in Science and Industry, Flow 81 Conference, St Louis, ISA, 1981. (Three volumes)

International Conference on Advances in Flow Measurement Techniques, Coventry, England, 1981.

BHRA Conference on Physical Modelling of Multi Phase Flow, Coventry, 1983.

Conference on Flow Metering and Proving Techniques in the Offshore Oil Industry, Aberdeen, 1983.

International Gas Research Conference, London, 1983.

International Conference on the Metering of Natural Gas and Liquefied Hydrocarbon Gases, London, 1984.

64th Annual GPA Convention, Houston, Texas, 1985.

International Conference on the Metering of Petroleum and its Products, OYEZ, London, 1985.

Conference on Fluid Control and Measurement, Tokyo, Japan, 1985.

Flow Measurement for Water Supply, London, 1985.

Second International Conference on Multi-phase Flow, London: BHRA, 1985.

International Conference on Flow Measurement in the Mid 80's, 9–12 June. Glasgow: NEL, 1986.

International Symposium on Fluid Flow Measurement, Washington, DC, 1986.

Trends in Instrumentation – Signal processing for flow measurement PC-based instrumentation systems, IOP Short Series No. 5 in collaboration with Trident International Exhibitions Limited, 25th February 1987.

Second International Conference on Flow Measurement, BHRA, London, 1988.

FLUCOME '88, H S Stephens & Associates, Sheffield, September 1988.

North Sea Metering Workshop, National Engineering Laboratory, East Kilbride, Scotland, 1988. (This series started in 1983 in Stavanger, Norway.)

FLOMEKO '89 – Fifth International Conference on Flow Measurement, Dusseldorf, Germany, 1989.

Multiphase Flow – Proceedings of the Fourth International Conference, BHRA, Cranfield, 1989.

North Sea Flow Measurement Workshop, Haugesund, Norway, 1989.

Second International Symposium on Fluid Flow Measurement, Calgary, Alberta, Canada, June, 1990.

International Conference on Flow Measurement of Commercially Important Fluids, London, England, 1990.

North Sea Flow Measurement Workshop, National Engineering Laboratory, Scotland, 1990.

Advanced Techniques for Integrated Circuit Processing, International Society Optical Engineers, Proceedings of SPIE, 1990/91.

FLUCOME '91, Proceedings of the 3rd Triennial International Symposium on Fluid Control, ASME, Measurement and Visualization, San Francisco, Cal, USA, 1991.

North Sea Flow Measurement Workshop, Norwegian Society of Chartered Engineers, 1991.

International Conference on Flow Measurement in Industry and Science, London, 1991.

North Sea Flow Measurement Workshop, National Engineering Laboratory, Scotland, 1992.

Annual Symposium on Instrumentation for the Process Industries, Texas, 1992.

FLOMEKO '93 – Sixth International Conference on Flow Measurement, Korea, 1993.

North Sea Flow Measurement Workshop, Bergen, Norway, 1993.

Flow Measurement for the Utilities, Amsterdam, The Netherlands, 4–5 Nov. 1993.
Sponsored by *J. Flow Meas. Instrum.*

FLOMEKO '94 – International Conference on Flow Measurement in the Mid 90's,
Glasgow, Scotland, June 1994.

FLUCOME '94, Toulouse, France, 1994.

North Sea Flow Measurement Workshop, Peebles, Scotland, 1994.

ISA Advances in Instrumentation and Control: International Conference and Exhibition, 1995.

North Sea Flow Measurement Workshop, National Engineering Laboratory, Scotland, 1995.

Advances in Instrumentation and Control: International Conference and Exhibition (ISA), 1996.

FLOMEKO '96 – Eighth International Conference on Flow Measurement, Beijing, China, 1996.

North Sea Flow Measurement Workshop, Peebles, Scotland, 1996.

Symposium on Flow in Open Channels and Closed Conduits, Gaithersburg, MD, 1997.

North Sea Flow Measurement Workshop, Kristiansand, Norway, 1997.

FLOMEKO '98 – Ninth International Conference on Flow Measurement, Lund, Sweden, 1998.

North Sea Flow Measurement Workshop, Gleneagles, Scotland, 1998.

References

- Abe, J., and Yoshinaga, A. (1991) Flowmeasurement by using fluidic resistor. *FLUCOME '91 3rd Triennial Int. Symp. on Fluid Control, Measurement and Visualization*. San Francisco: 559–564.
- Abernathy, F. H., and Kronauer, R. E. (1962) The formation of vortex streets. *J. Fluid Mechanics*, 13:1–20.
- Adiletta, G., Cascetta, F., della Valle, S., Pagano, S., and Vigo, P. (1993) Twin rigid straight pipe Coriolis mass flowmeter. *Measurement*, 11:289–308.
- Advisory Committee on Aeronautics (1916–17) Reports and Memoranda, No. 295:675.
- AGA-8 (1992) *Compressibility and supercompressibility for natural gas and other hydrocarbon gases*, Transmission Measurement Committee Report No. 8, AGA Catalog No. XQ 1285, Arlington, VA.
- Agarwal, A. K., and Turgeon, M. (1984) New MicroMotion mass flowmeters. *Energy Progr.* 4:221–224.
- Agricola, J. B. (1997) Gas well flowline measurement by ultrasonic flow meter. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 16.
- Ahmad, K., Baker, R. C., and Goulas, A. (1986) Computation and experimental results of wear in a slurry pump impeller. *Proc. I. Mech. E, Pt. C*, 200:439–445.
- Al-Asmi, K., and Castro, I. P. (1992) Vortex shedding in oscillatory flow: Geometrical effects. *J. Flow Meas. Instrum.*, 3:187–202.
- Al-Khazraji, Y. A., Al-Rabeh, R. H., Baker, R. C., and Hemp, J. (1978) Comparison of the effect of a distorted profile on electromagnetic, ultrasonic and differential pressure flowmeters, *FLOMEKO 1978 – Proc. Conf. on Flow Measurement of Fluids*. Groningen, The Netherlands (Amsterdam: North-Holland Publishing Co.): 215–22.
- Al-Khazraji, Y. A., and Baker, R. C. (1979) Analysis of the performance of three large-electrode electromagnetic flowmeters. *J. Phys. D*, 12:1423–34.
- Al-Khazraji, Y. A., and Hemp, J. (1980) *Electromagnetic flowmeters and methods of measuring flow*. U.S. Patent 4,434,666, March 6, 1984 (filed April 4, 1980).
- Allen, C. (1994) The Interoperable Systems Project (ISP). *Meas. Control*, 27(2):38–41.
- Al-Rabeh, R. H. (1981) *The theory and performance of electromagnetic flowmeters*. PhD Thesis, Imperial College, University of London, England.
- Al-Rabeh, R. H., and Baker, R. C. (1979) Optimisation of conventional electromagnetic flowmeters. *Fluid Mechanics Silver Jubilee Conference*, National Engineering Laboratory, Glasgow, Scotland, Paper 6.1.
- (1986) On the ability to dry calibrate an electromagnetic flowmeter. *J. Phys. E: Sci. Instrum.*, 19:203–206.
- Al-Rabeh, R. H., and Hemp, J. (1981) A new method for measuring the flow rate of insulating fluids. *Int. Conf. on Advances in Flow Measurement Techniques*, BHRA, Paper K3:267–278.
- Al-Rabeh, R. H., Baker, R. C., and Hemp, J. (1978) Induced flow measurement theory for poorly conducting fluids. *Proc. Roy. Soc. A*, 361:93–107.
- Aly, F. A., and Lee, L. L. (1981) Self consistent equations for calculating the ideal gas heat capacity, enthalpy and entropy. *Fluid Phase Equilibria*, 6:169–179.
- Amadi-Echendu, J. E., and Higham, E. H. (1990) Additional information from flowmeters via signal analysis. *IEEE Trans. Instrum. Meas.*, 39(6):998–1003.
- Amadi-Echendu, J. E., and Hurren, P. J. (1990) Identification of process plant signatures using flow measurement signals. *IEEE Trans. Instrum. Meas.*, 39(2):416–20.

- Amadi-Echendu, J. E., and Zhu, H. (1992) Signal analysis applied to vortex flowmeters. *IEEE Trans. Instrum. Meas.*, 41(6):1001–4.
- Amadi-Echendu, J. E., Zhu, H., and Higham, E. H. (1993) Analysis of signals from vortex flowmeters. *J. Flow Meas. Instrum.*, 4:225–32.
- Amini, A., and Owen, I. (1995) The use of critical flow venturi nozzles with saturated wet steam. *J. Flow Meas. Instrum.*, 6:41–7.
- Anon. (1966a) Measuring befeater gin. *Control*, 10(96):292.
- (1966b) Positive displacement flowmeter for low rates. *Control*, 10(98):437.
- (1988) NEL launches offshore multiphase flowmeter project. *Process Eng.*, October: 33.
- (1993) Non-intrusive multiphase flow meter. *Offshore Research Focus*, No. 98:9.
- (1994) Multiphase flow meter. *Offshore Research Focus*, No. 103:2.
- ANSI/API 2530:1985 Orifice metering of natural gas and other related hydrocarbon fuels.
- Aoki, T., Nukui, K., Okamura, S., and Kimura, Y. (1996) Development of fluidic gas meter (improvement of sensitivity at lower flow rate using U-shaped target sensor), *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 138–43.
- API (1992) American standard method for measurement of liquid hydrocarbons by displacement meter systems, Chapter 5.2, *Manual of Petroleum Measurements*. Washington DC: American Petroleum Institute.
- Arasi, J. A. (1989) Coriolis mass flowmeter passes NGL line field trials. *Oil & Gas J.*, January: 59–61.
- Arnberg, B. T., Britton, C. L., and Seidl, W. F. (1973) Discharge coefficient correlations for circular-arc venturi flowmeters at critical (sonic) flow. *ASME*, Paper No 73-WA/FM-8.
- Arnold, R. M., and Pitts, R. W. (1981) *Fluid flow meters for mixed liquid and gas*. U.S. Patent 4,272,982, June 16.
- Aschenbrenner, A. (1989) Calibration of the new test rig for large gas meters of the Physikalisch-Technische Bundesanstalt. *FLOMEKO 5th Int. Conf. on Flow Measurement*: 11–22.
- Asher, R. C. (1983) Ultrasonic transducers for chemical and process plant. *Phys. Technol.*, 14:19–23.
- (1997) *Ultrasonic Sensors: For Chemical and Process Plant*. Bristol, UK: Institute of Physics.
- ASME/ANSI MFC-7M (1987) Measurement of gas flow by means of critical flow venturi nozzles. New York: ASME.
- Aston, N. A. J., and Evans, G. V. (1975) The integrated pulse velocity method applied to the calibration of flowmeters in high pressure natural gas systems. *Conf. on Fluid Flow Measurement in the Mid 1970's*, Paper D-2, East Kilbride, Scotland: National Engineering Laboratory.
- Athane, B. (1994) Implications of legal metrology in flow measurement for the utilities. *J. Flow Meas. Instrum.*, 5(2):67–9.
- Atkinson, K. N. (1992) A software tool to calculate the over-registration error of a turbine meter in pulsating flow. *J. Flow Meas. Instrum.*, 3:167–172.
- Awberry, J. H., and Griffiths, E. (1926–7) Further experiments with the Ewing ball-and-tube flowmeter. *Proc. Roy. Soc., Edinburgh*, 47:1.
- Baird (1993) Innovations in magnetic flowmetering. *Advances in Instrumentation and Control: Int. Conf. Exhib. (ISA)*, Chicago, 48:879–84.
- Baker, P. D. (1983) *Positive displacement liquid meters*. International School of Hydrocarbon Measurement (ISHM), University of Oklahoma, April 12–14.
- Baker, R. C. (1968) Solutions of the electromagnetic flowmeter equation for cylindrical geometries. *Brit. J. Appl. Phys. (J. Phys. D)*, 1(2):895–9.
- (1969) Flow measurement with motion induced magnetic field at low magnetic Reynolds number. *Magnetohydrodynamics*, 3:69–73 (in Russian).
- (1970a) Effects of non-uniform conductivity fluids in electromagnetic flowmeters. *J. Phys. D*, 3:637–9.
- (1970b) Linearity of motion-induced-magnetic-field flowmeter. *Proc. I.E.E.*, 117:629–33.
- (1973) Numerical analysis of the electromagnetic flowmeter. *Proc. I.E.E.*, 120:1039–43.
- (1976) Some recent developments in ultrasonic and electromagnetic pipe flow measurement at Imperial College. *ACTA IMEKO VII*: 339–48.
- (1977) Liquid metal electromagnetic flowmeters in fast reactor technology. *Prog. Nucl. Energy*, 1:41–61.
- (1982) Electromagnetic flowmeters, pp. 209–51. *Developments in Flow Measurement – 1* (R. W. Scott, Ed.), London: Applied Science Publishers Ltd.

- (1983) A review of recent developments in electromagnetic flowmeasurement. *Third Beer-Sheva Seminar on MHD – Flows and Turbulence*, Ben-Gurion University of the Negev, Beer-Sheva, Israel, 23–27 March 1981. Also in *Prog. Astro. Aero.*, 84:225–59.
- (1985) Principles and Practice of electromagnetic flowmeasurement. *Technisches Messen*, 52(1):4–12 (in German).
- (1988) Measuring multiphase flow. *Chem. Engineer*, October: 39–45.
- (1988/9) *An Introductory Guide to Flow Measurement*. London: Mechanical Engineering Publications.
- (1989) Multi-phase flow moves on. *Control Instrum.*, February: 35, 37.
- (1991a) Response of bulk flowmeters to multiphase flow. *Proc. I. Mech. E. Part C: J. Mech. Eng. Sci.*, 205:217–29.
- (1991b) Turbine and related flowmeters: Part I – Industrial practice. *J. Flow Meas. Instrum.* 2:147–62.
- (1993) Turbine flowmeters: Part II – Theoretical and experimental published information. *J. Flow Meas. Instrum.*, 4:123–44.
- (1996) *An Introductory Guide to Industrial Flow*. London: Mechanical Engineering Publications.
- (1998) Flowmeter, pp. 245–7. *Instruments of Science, An Historical Encyclopedia* (R. Bud and D. J. Warner, Eds.), New York and London: Garland Publishing Inc.
- Baker, R. C., and Deacon, J. E. (1983) Tests on turbine, vortex and electromagnetic flowmeters in 2-phase air-water upward flow. *Int. Conf. on Physical Modelling of Multi-Phase Flow*, Coventry, England, BHRA Fluid Engineering, Paper H1: 337–52.
- Baker, R. C., and Hayes, E. R. (1985) Multiphase measurement problems and techniques for crude oil production systems. *Petrol. Rev.*, November: 18–22.
- Baker, R. C., and Higham, E. H. (1992) *Flowmeters*. U.K. Patent 2 231 669 B, December 23.
- Baker, R. C., and Morris, M. V. (1983) The suitability of flowmeters for optical techniques. *Proc. Int. Conf. on Optical Techniques in Process Control*. The Hague, Netherlands (Cranfield, England: BHRA) (post-deadline paper).
- (1985) Positive-displacement meters for liquids. *Trans. Inst. M.C.*, 7:209–20.
- Baker, R. C., and Smith, D. J. M. (1990) *Flowmeter Specification Guidelines*, Report No. 14. Cranfield, England: FLOMIC.
- Baker, R. C., and Tarabad, M. (1978) The performance of electromagnetic flowmeters with magnetic slurries. *J. Phys. D*, 11:167.
- Baker, R. C., and Thompson, E. J. (1975) A two beam ultrasonic phase-shift flowmeter. *Conf. on Fluid Flow Measurement in the Mid 1970s*, East Kilbride Glasgow, Scotland: National Engineering Laboratory.
- (1978) *Measurement of fluid flow*, U.S. Patent 4,078,428, March 14.
- Baker, R. C., Deacon, J. E., Lenn, C. P., and Smart, M. D. (1985) The effect on electromagnetic flowmeters of a change in fluid. *Meas. Control*, 8:5–10.
- Ball, J. M. (1977) Viscosity effects on the turbine flowmeter. *Proc. Symp. on Flow Measurement in Open Channels and Closed Conduits*. NBS, Gaithersburg, MD: 847–69.
- Balls, B. W., and Brown, K. J. (1959) The magnetic flowmeter. *Symp. on Flow Measurement: Trans. Soc. Instrument Technology*, June: 123–30.
- Barnes, R. G. (1982) Positive displacement liquid meters. *Adv. Instrum.*, 37(3):1197–204. *Proc. ISA. Int. Conf. Exhib.*, (Philadelphia, PA, USA, 18–21 October 1982), Research Triangle Park, USA, Instrum Soc Am.
- Barry, J. J., Sheikholeslami, M. Z., and Patel, B. R. (1992) *Numerical simulation of flow through orifice meters*, Gas Research Institute, GRI-92/0060.1.
- Batchelor, G. K. (1967) *An Introduction to Fluid Dynamics*. Cambridge: Cambridge University Press.
- Bates, C. J. (1981) Laser doppler anemometry measurements of the flow through an orifice plate. *Flow Its Measurement and Control in Science and Industry*, 2:59–68.
- Bates, I. P. (1991) Field use of K-lab flow conditioner. *Proc. North Sea Flow Measurement Workshop 1991* (two volumes), Norwegian Society of Chartered Engineers, October 22–24, 1991 (9th in the series).
- Battye, J. S. (1993) The correlation flowmeter – A detailed investigation of an attempt to improve its performance. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 492–499.
- Baucom, W. E. (1979) Evaluation of a coriolis mass flow meter for pulverized coal flows. Prepared by The Energy Conservation Division, The University of Tennessee Space Institute,

- Tullahoma, Tennessee 37388, for the U.S. Dept. Energy (Contract No. DE-AC02 - 79ET10815, December).
- Baumhoel, J. (1994) Use of clamp-on transit-time ultrasonic flowmeters in Aircraft mass fuel flow, hydraulic fluid leak detection, & ground support applications. *Instrumentation in the Aerospace Industry: Proc. Int. Symp. ISA*, Research Triangle Park, NC: 243–60.
- (1996) Pipeline management using networked clamp-on transit-time flowmeters. *Proc. Int. Pipeline Conf.*, Calgary, Canada, New York: ASME, Vol. 2:1123–8.
- Bean, H. S. (1971) *Fluid Meters, Their Theory and Application*, 6th ed. New York: ASME.
- Beck, M. S., and Plaskowski, A. (1987) *Cross Correlation Flowmeters – Their Design and Application*. Bristol, UK: Adam Hilger.
- Beeson, J. (1995) Ultrasonics meters improve NorAm's custody transfer. *Pipeline & Gas J.*, 222(7):21–4.
- Beg, N., and Toral H. (1993) Off-site calibration of a two-phase pattern recognition flowmeter. *Int. J. Mult. Flow*, 19:999–1012.
- Bellinga, H., and Delhez, F. J. (1993) Experience with a high-capacity piston prover as a primary standard for high-pressure gas flow measurement. *J. Flow Meas. Instrum.*, 4:85–90.
- Bellinga, H., and Stronk, H. B. (1974) The practical application of flow straighteners with turbine flowmeters for gas. *IMEKO VII Conf.*, London, Paper BFL/242:1–11.
- Bellinga, H. Hoek, C. P., Kooi, A., Van Laak, F. A. L., and Orbous, P. J. (1981) Using a piston prover as a primary standard in high-pressure gas metering. *Flow, Its Measurement and Control in Science and Industry, Flow '81 Conference*, St Louis, ISA.
- Benabdelkarim, M., and Galiana, C. (1991) Nonradioactive densitometer for continuous monitoring of cement mixing process. *Proc. First Int. Conf. on Health, Safety and Environment in Oil And Gas Exploration and Production*, Society of Petroleum Engineers of AIME, Texas: 539–45.
- Benard, C. J. (1988) *Handbook of Fluid Flowmetering*. Morden, England: The Trade & Technical Press.
- Bentley, J. P., and Benson, R. A. (1993) Design conditions for optimal dual bluff body vortex flowmeters. *J. Flow Meas. Instrum.*, 4:205–14.
- Bentley, J. P., and Nichols, A. R. (1990) The mapping of vortex fields around single and dual bluff bodies. *J. Flow Meas. Instrum.*, 1:278–86.
- Bentley, J. P., Benson, R. A., and Shanks, A. J. (1996) The development of dual bluff body flowmeters. *J. Flow Meas. Instrum.*, 7:85–90.
- Bernier, R. N., and Brennen, C. E. (1983) Use of the electromagnetic flowmeter in a two-phase flow. *Int. J. Multiphase Flow*, 9:251–7.
- Betts, K. S. (1990) Mass flow sensors: Measuring up to new applications. *Mechanical Eng.*, 112: 72–5.
- Bevir, M. K. (1970) The theory of induced voltage electromagnetic flowmeters. *J. Fluid Mech.*, 43:577.
- (1972) The effect of conducting pipe connections and surrounding liquid on the sensitivity of electromagnetic flowmeters. *J. Phys. D*, 5:717.
- Bevir, M. K., O'Sullivan, V. T., and Wyatt, D. G. (1981) Computation of electromagnetic flowmeter characteristics from magnetic field data. *J. Phys. D*, 14:376.
- Bignell, N. (1994) A secondary standard ultrasonic gas flowmeter. *FLOMEKO '94 Flow Measurement in the Mid-90s*, Glasgow, Scotland: NEL, Paper 7.2.
- (1996a) Comparison techniques for small sonic nozzles. *J. Flow Meas. Instrum.*, 7:109–14.
- (1996b) Positive displacement gas flow standard. *Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 544–9.
- Bignell, N., Collings, A. F., Taylor, K. J. H., Martin, B. J., Braathen, C. W., Peterson, M., and Welsh, C. (1993a) An ultrasonic domestic gas meter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 403–9.
- (1993b) Calibration of ultrasonic domestic gas meter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 410–5.
- BIPM/IEC/IFCC/ISO/IUPAC/IUPAP/OIML (1993) *Guide to the Expression of Uncertainty in Measurement*, 1st ed. Geneva, Switzerland: ISO.
- Birch, J. R., and Lemon, D. D. (1995) Non-intrusive flow measurement techniques for hydroelectric applications. *ASCE Waterpower – Proc. Int. Conf. on Hydropower*, 3:2049–58.
- Birker, B. (1989) Theory, design and performance of the straight tube mass flowmeter. *Mass Flow Measurement Direct and Indirect*, London: IBC Technical Services Ltd.

- Blackett, P. M. S., and Henry, P. S. H. (1930) A flow method for comparing specific heats of gases: Part II – The theory of the method. *Proc. Roy. Soc. London Ser. A*, 126:333–54.
- Blows, L. G. (1981) Towards a better turbine flowmeter. *Int. Conf. on Advances in Flow Measurement Techniques*, Warwick, England (Cranfield, England: BHRA Fluid Engineering): 307–18.
- Blumenthal, I. (1984) Improving productivity through mass flow measurement and control. *Enhancing Productivity, Proc. Pacific Cascade Instrumentation '84 Exhib. Symp.* ISA: 163–8.
- (1985) Direct mass flow rate and density monitoring using a Coriolis/gyroscopic sensor base. *Tappi J.* (USA), 68(11):82–4.
- Boer, A. H., and Volmer, W. (1997) Test results Krohne 8" ultrasonic flowmeter. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 32.
- Boettcher, D. B., and Hickling, E. M. (1993) Instrumentation and control systems for Sizewell B. *Meas. Control*, 26(5):132–8.
- Bonfig, K. W., Hofman, F., Reinhold, I., and Feuerstein, M. (1975) A new method of magnetic inductive flow measurement. *Conf. on Flow Measurement in the Mid-1970s*, Paper E-3.
- Bonner, J. A. (1977) Pulsating effects in turbine meter. *Pipe Line Industry*, March: 57–62.
- (1993) A new international standard, ISO 9951: The measurement of gas flow in closed conduits – turbine meters. *J. Flow Meas. Instrum.*, 4:99–100.
- Bonner, J. A., and Lee, W. F. Z. (1992) The history of the gas turbine meter. *AGA Distribution and Transmission Conf.*, Kansas City, Mo.
- Bosio, J., Wilcox, P. L., Erdal, A., and Sinding, H. (1990) Gas flowmeters repeatability and accuracy might be impeded by elemental sulphur deposition. *North Sea Flow Measurement Workshop*, Scotland: National Engineering Laboratory: 23–5.
- Botros, K. K., Jungowski, W. M., and Petela, G. (1992) Gauge line effects and DP transmitter response to flow pulsation through orifice plate. *J. Flow Meas. Instrum.*, 3:130–44.
- Boucher, R. F. (1995) Minimum flow optimization of fluidic flowmeters. *Meas. Sci. Technol.*, 16:872–9.
- Boucher, R. F., Churchill, D., Mazharoglu, C. and Parkinson, G. J. (1991) A fluidic by-pass venturimeter. *FLUCOME '91 3rd Triennial Int. Symp. on Fluid Control, Measurement and Visualization*, San Francisco: 565–69.
- Boucher, R. F., and Mazharoglu, C. (1988) Low Reynolds number fluidic flowmetering. *J. Phys. E: Sci. Instrum.*, 21:977–89.
- Bowden, K. F., and Fairbairn, L. A. (1956) Measurement of turbulent fluctuations and Reynolds stresses in a tidal current. *Proc. Roy. Soc. London Ser. A*, 237:422.
- Braathén, C. (1996) Sampling interval in electronic domestic gas meters. *FLOMEKO'96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 324–9.
- Bragg, M. I., and Lynneworth, L. C. (1994) Internally-nonprotruding one-port ultrasonic flow sensors for air and some other gases. *Control '94, Conference Publication No. 389*, IEE:1241–7.
- Brain, T. J. S. (1978) The calibration of meters with gases. *Short Course Notes on The Principles and Practice of Flow Measurement*, Lecture No. 7, East Kilbride: National Engineering Laboratory.
- Brain, T. J. S., and Macdonald, L. M. (1975) Evaluation of the performance of small-scale critical flow venturis using the NEL gravimetric gas flow standard test facility. *Fluid Flow Measurement in the Mid '70s*, Edinburgh, Scotland: 103–25.
- Brain, T. J. S., and Reid, J. (1978) Primary calibrations of critical flow venturi nozzles in high-pressure gas. *FLOMEKO 1978 Flow Measurement of Fluids*, Groningen, The Netherlands: 55–64.
- (1980) *Primary calibrations of critical flow venturi nozzles in high-pressure gas*. NEL, Report No 666.
- Brain, T. J. S., Reid, J., and MacDonald, C. (1975) Further development of the NEL pulsed gas-ionization flowmeter. *Conf. on Fluid Flow Measurement in the Mid 1970's*, Paper D-4.
- Branch, J. C. (1995) The effect of an upstream short radius elbow and pressure tap location on orifice discharge coefficients. *J. Flow Meas. Instrum.*, 6:157–62.
- Brennan, J. A., McFaddin, S. E., Sindt, C. F., and Kothari, K. M. (1989) The influence of swirling flow on orifice and turbine flowmeter performance. *J. Flow Meas. Instrum.*, 1:5–8.
- Brennan, J. A., Sindt, C. F., Lewis, M. A., and Scott, J. L. (1991) Choosing flow conditioners and their location for orifice flow measurement. *J. Flow Meas. Instrum.*, 2:40–4.
- Brignell, J., and White, N. (1994) *Intelligent Sensor Systems*. Bristol and Philadelphia: Institute of Physics Publishing.
- Britton, C., and Mesnard, D. (1982) A performance survey of round- and diamond-shaped averaging pitot-type primaries. *Meas. Control*, 15:341–50.

- Brobeil, W., Frohlich, R., Schafer, R., and Schulz, K. H. (1993) Flow measurements in slurry applications with switched DC-fields magflowmeters. *FLOMEKO'93 Proc. 6th Int. Conf. on Flow Measurement*, Korea Research Institute of Standards and Science: 635–41.
- Brockhaus, H., van der Pol, R., Schoth, U., and Klein, J. W. (1996) Capacitive electro magnetic flowmeter (EMF) using microsystems technology. *FLOMEKO'96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 635–41.
- Brown, A. F., and Kronberger, H. (1947) A sensitive recording calorimetric mass flowmeter. *J. Sci. Instrum.*, 24:151–5.
- Brown, G. J. (1996) Oil flow performance of ultrasonic meters. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- (1997) Factors affecting the performance of ultrasonic flowmeters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 33.
- BS 3812:1964 *Estimating the dryness fraction of saturated steam*. British Standards Institution.
- BS 7405:1991 *Guide to the selection and application of flowmeters*. British Standards Institution.
- BS EN ISO 9300:1995 *Measurement of gas flow by means of critical flow venturi nozzles*.
- Buckle, U., Durst, F., Howe, B. and Melling, A. (1992) Investigation of a floating element flowmeter. *J. Flow Meas. Instrum.*, 3:215–25.
- Buckle, U., Durst, F., Kochner, H., and Melling, A. (1995) Further investigation of a floating element flowmeter. *J. Flow Measurement and Instrumentation*; 6:75–8.
- Bucknell, R. L. (1963) Calibration systems and turbine type flow transducers for cryogenic flow measurements. *Adv. Cryogenic Engineering*, 8:360–370.
- Butterworth, D., and Hewitt, G. F. (1977) *Two-Phase Flow and Heat Transfer*, Harwell Series. Oxford, UK: Oxford University Press.
- Buttle, R. S., and Kimpton, A. (1989) ESKOM's flow calibration facility. *FLOMEKO 5th Int. Conf. on Flow Measurement*: 1–10.
- Cairney, W. D. (1991) Typical flow measurement problems and their solution in the electricity supply industry. *J. Flow Meas. Instrum.*, 2:217–24.
- Campion, P. J., Burns, J. E., and Williams, A. (1973) *A Code of Practice for the Statement of Accuracy*. London: HMSO.
- Caron, R. W. (1995) Use of sonic nozzles in a manufacturing environment. *Proc. Int. Instrumentation Symp.*, ISA: 543–58.
- Carpenter, B. (1990) Choose the right material for mass flow meters. *Chem. Eng. Prog.*, 86 (10):55–60.
- Cascetta, F. (1994) Application of a portable clamp-on ultrasonic flowmeter in the water industry. *J. Flow Meas. Instrum.*, 5:191–4.
- Cascetta, F., and Vigo, P. (1988) *Flowmeters – A comprehensive survey and guide to selection*, Research Triangle Park, NC: ISA Publishing.
- Cascetta, F., Cignolo, G., Gorla, R., Martini, G., Rivetta, A., and Vigo, P. (1992) Experimental intercomparison of Coriolis mass flowmeters. *Trans. Inst. Meas. Control*, 14:99–107. (Also *Int. Conf. on Flow Measurement of Commercially Important Fluids*, London: IBC Technical Services Ltd, Feb/Mar 1990.)
- Cascetta, F., della Valle, S., Guido, A. R., and Vigo, P. (1989a) A new type of Coriolis acceleration mass flowmeter. *Proc. IMEKO XI*, Houston: 511–21.
- (1989b) A Coriolis mass flowmeter based on a new type of elastic suspension. *Measurement*, 7:182–91.
- Casperson, C. (1975) The vortex flowmeter. *Conf. Fluid Flow Measurement in the Mid 1970's*, National Engineering Laboratory: Glasgow, Scotland, Paper C-1.
- (1993) A new flowmeter for measurement of gas flow at very low rates. *FLOMEKO'93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 306–12.
- Chambers, J. (1994) The EMC test house – Before and after. *Engineering*, May: 39–42.
- Chanaud, R. C. (1965) Observations of oscillatory motion in certain swirling flows. *J. Fluid Mech.*, 21:111–27.
- Chapman, N. R., and Etheridge, D. W. (1993) A step change in domestic metering technology from leather diaphragms to ultrasonics. *Flow Measurement for the Utilities*, Amsterdam, The Netherlands.
- Chateau, F. (1991) Mass delivery meters using the Coriolis effect for batching hydrocarbon binders. *Bulletin de Liaison des Laboratoires des Ponts et Chaussées*, No. 175:13–19.

- Cheesewright, R., Atkinson, K. N., Clark, C., ter Horst, G. J. P., Mottram, R. C., and Viljeer, J. (1996) Field tests of correction procedures for turbine flowmeters in pulsatile flows. *J. Flow Meas. Instrum.*, 7:7–17.
- Cheesewright, R., Bisset, D., and Clark, C. (1998) Factors which influence the variability of turbine flowmeter signal characteristics. *J. Flow Meas. Instrum.*, 9:83–9.
- Chen, J., Karras, M., Tahkola, E., and Tombery, J. (1993) Flow measurement of medium consistency pulp suspension by cross-correlation flowmeter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 506–8.
- Chesnoy, A. B. (1993) Sonic nozzles meter natural gas at K-Lab. *J. Flow Meas. Instrum.*, 4:73–6.
- Chisholm, D. (1967) Flow of incompressible two-phase mixtures through sharp-edged orifices. *J. Mech. Eng. Sci.*, 9:72–8.
- (1977) Two-phase flow through sharp-edged orifices. *J. Mech. Eng. Sci.*, 19:128–30.
- Chisholm, D., and Leishman, J. M. (1969) Metering of wet steam. *Chem. Process Eng.*, 50:103–6.
- Chisholm, D., and Watson, G. C. (1966) *The flow of steam/water mixtures through sharp-edged orifices*. East Kilbride, Glasgow: NEL, Report No. 213.
- Churchley, A. (1994/95) Reliability of microprocessor based protection systems. *Meas. Control*, 27(10):315–22.
- Clark, C. (1992) The measurement of dynamic differential pressure with reference to the determination of pulsating flows using DP devices. *J. Flow Meas. Instrum.*, 3:145–50.
- Clayton, C. G., Ball, A. M., Clark, W. E., and Spencer, E. A. (1962a) The accurate measurement of turbulent flow in pipes – Using radioactive isotopes – The isotope dilution method. *Proc. Symp. on Flow Measurement in Closed Conduits*, HMSO, Vol. 2: Paper E-3.
- Clayton, C. G., Clark, W. E., and Ball, A. M. (1962b) The accurate measurement of turbulent flow in pipes – Using radioactive isotopes – Using the isotope velocity method and the effect of some restrictions on optimum operation. *Proc. Symp. Flow Measurement in Closed Conduits*, HMSO, Vol. 2: Paper E-4.
- Cluley, J. C. (1994/95) Designing for reliability. *Meas. Control*, 27(10):310–4.
- Cole, J. H. (1985) Drag turbine mass flowmeter development. *Flow, Its Measurement and Control in Science and Industry*, ISA Conf., St Louis, Mo, Vol. 2:441–51.
- Coleman, M. C. (1956) Variable area flow meters. *Trans. Instn. Chem. Eng.*, 34:339.
- Collings, A. F., Bignell, N., Hews Taylor, K. J., and Marting, B. J. (1993) Ultrasonic metering of gas flows. *Proc. Ultrasonic Int. Conf.*, Vienna, Austria, Butterworth Heinemann: 205–8.
- Collins, D. B., and Gacesa, M. (1970) Measurement of steam quality in two-phase up flow with venturi meters. ASME Paper No. 70-FE-6.
- Collis, D. C., and Williams, M. J. (1959) Two-dimensional convection from heated wires at low Reynolds numbers. *J. Fluid Mech.*, 6:357.
- Conrad, F., and Trostmann, E. (1981) A servo controlled volume rate flowmeter. *BHRA Int. Conf. on Advances in Flow Measurement Techniques*, Warwick, UK, Paper H3.
- Constable, G. (1994) Concurrent engineering – Its procedures and pitfalls. *Meas. Control*, 27(8):245–7.
- Cook, C. E., and Bernfeld (1967) *Radar Signals*. New York: Academic Press.
- Corneliusson, S. E. (1991) Field experience with Hod metering. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- Cornish, D. (1994/5) Instrument performance. *Meas. Control*, 27(10):323–8.
- Coulthard, J., and Yan, Y. (1993a) Vortex wake transit time measurements for flow metering. *J. Flow Meas. Instrum.*, 4:269–72.
- (1993b) Comparisons of different bluff bodies in vortex wake transit time measurements. *J. Flow Meas. Instrum.*, 4:273–6.
- (1993c) Ultrasonic cross-correlation flowmeters. *J. Inst. Meas. Control*, 26(6):164–7.
- Cousins, T. (1971) The performance of long bore orifices at low Reynolds numbers. *Modern Developments in Flow Measurement*, Paper 4.1:160–79.
- (1975) An experimental investigation into the design parameters of the Dall tube. *Conf. on Fluid Flow Measurement in the Mid-1970s*, Glasgow, Scotland: National Engineering Laboratory, Paper J2.
- (1977) Vortex meters. *Transducer 77 Conf.*
- Cousins, T., and Hayward, A. T. J. (1993) Development of the T-ring vortex meter. *J. Flow Meas. Instrum.*, 4:197–204.

- Cousins, T., et al. (1973) A linear and accurate flowmeter using vortex shedding. *Symp. Power Fluidics for Process Control*, University of Surrey, Guildford, England: 45–56.
- Cousins, T., Hayward, A. J. T., and Scott, R. (1989) Design and performance of a new vortex shedding flowmeter. *FLOMEKO '89 Proc. 5th Int. Conf. on Flow Measurement*, Dusseldorf, Germany: 151–67.
- Cox, T. J., and Wyatt, D. G. (1984) An electromagnetic flowmeter with insulated electrodes of large surface area. *J. Phys. E: Sci. Instrum.*, 17:488–503.
- Crawshaw, J., and Chambers, J. (1984) *A Concise Course in A-Level Statistics*. Cheltenham, UK: Stanley Thornes.
- Cunningham, W. J., and Astami, K. (1993) The effect of ultrasonic frequency on the accuracy of gas flowmeters. *Meas. Sci. Technol.*, 4:1476–8.
- Cutler, G. D. (1982) Averaging Pitot-type primaries. *Meas. Control*, 15:436–7.
- Dall, H. E. (1962) Flow tubes and non-standard devices for flow measurement with some coefficient considerations. *Proc. Symp. on Flow Measurement in Closed Conduits*, HMSO, Edinburgh: Paper D-1:385–94.
- Danen, G. W. A. (Ed.) (1985) *Shell Flowmeter Engineering Handbook*, 2nd ed. New York: McGraw Hill.
- Davis, R. W., and Mattingly, G. B. (1997) Numerical modelling of turbulent flow through thin orifice plates. *Symp. on Flow in Open Channels and Closed Conduits*, Gaithersburg, MD.
- Davis, R. W., Moore, E. F., and Purtell, L. P. (1984) A numerical-experimental study of confined flow around rectangular cylinders. *Phys. Fluids*, 27:46–59.
- Davis, T. C. E. (1990) Fiscal measurement and proving experience with coriolis meters. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Deacon, J. E. (1983) Electromagnetic flowmeter installation tests. *IMEKO*, Budapest, Paper E3:85–91.
- Dean, R. W. (1988a) Field experience using Coriolis mass meters II. *North Sea Metering Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- (1988b) Advantages of mass measurement for petroleum liquids. *Petrol. Rev.* 42:32–3.
- Dean, T. L., Dowty, E. L., and Jiskoot, M. A. (1990a) The design manufacture and testing of the Texaco subsea three phase metering system. *North Sea Flow Measurement Workshop*, National Engineering Laboratory, October 23–25.
- Dean, T. L., Dowty, E. L., and Jiskoot, R. J. J. (1990b) The development of a subsea three phase metering system, *European Oil and Gas Conf.*, Palermo, October.
- de Boer, G., and Lansing, J. (1997) Dry calibration of ultrasonic gas flow meters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 18.
- Decker, M. M. (1960) The gyroscopic mass flowmeter. *Engineers Dig.*, 21(7).
- de Jong, J. (1978) Comparison of some 500 mm diameter flowmeters. *FLOMEKO 1978 – Proc. Conf. on Flow Measurement of Fluids*, Groningen, The Netherlands: 565.
- de Jong, S., and van der Kam, P. M. A. (1993) High pressure recalibration of turbine meters. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*: 121–8.
- de Kraker, F. (1989) The performance of a coriolis meter when used to measure density. *Mass Flow Measurement Direct and Indirect*, London, England.
- de Leeuw, H. (1994) Wet gas flow measurement by means of a venturi meter and a tracer technique. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- (1997) Liquid correction of venturi meter readings in wet gas flow. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 21.
- Del Grosso, V. A., and Spurlock, E. M. (1957) *The feasibility of using wholly external ultrasonics to measure fluid flow within thick walled metal pipes*. NRL Report 4967.
- Delsing, J. (1991) The zero-flow performance of a sing-around ultrasonic flowmeter. *J. Flow Meas. Instrum.*, 2:205–8.
- Desjardins, M. (1994) *World FIP. Meas. Control*, 27(2):42–6.
- de Vries, H., Loogmann, L. L., van Dellen, K., and Broekgaarden, G. J. (1989) Ultrasonic gas flow measurements with reflection mode in underground pipelines. *FLOMEKO '89 Proc. 5th Int. Conf. on Flow Measurement*, Dusseldorf, Germany: 325–32.
- Dickinson, G. (1988) Design considerations for optical flowmeter sensors. *2nd Int. Conf. on Flow Measurement*, London UK (Cranfield, England: BHRA).
- Dijstelbergen, H. H. (1964) Rotameter dynamics. *Chem. Eng. Sci.*, 19:853.
- (1970) The performance of a swirl meter. *J. Phys. E*, 3:886–8.

- (1982) Gas meters, Chapter 5. *Developments in Flow Measurement – 1*, (R. W. W. Scott, Ed.), London: Applied Science Publishers.
- Dimaczek, G., Fassbinder, H.-G., Emmel, A., and Kupfer, R. (1994) High-precision Coriolis mass flowmeter for bulk material two-phase flows. *J. Flow Meas. Instrum.*, 5:295–302.
- Dixey, M. (1993) Putting reliability at the centre of maintenance. *Prof. Eng.* June:23–5.
- Dominick, J., Durst, F., Raszillier, H., and Zeisel, H. (1987) A method to measure mass and volume flow rates of two-phase flows. *Int. J. Multiphase Flow*, 13:685–98.
- Drentham, J. G., and Huijsmans, F. J. J. (1993) Gassonic-400 & P Sonic & Q Sonic ultrasonic gas flow meters. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 285–98.
- DTI (1993) *Product standards – Electromagnetic compatibility*. Department of Trade and Industry UK Regulations.
- (1996) *Competitive analysis of the UK instrumentation and control industry*. Department of Trade and Industry.
- Durst, F., and Raszillier, H. (1990) Flow in a rotating straight pipe, with a view on coriolis mass flow meters. *J. Fluids Eng., Trans. ASME*, 112:149–54.
- Dykesteen, E. (1992) Multiphase metering. *Chem. Eng. Res. Design*, 70(1):32–7.
- Dykesteen, E., Hallanger, A., Hammer, E., Samnoy, E., and Thorn, R. (1985) Non-intrusive three-component ratio measurement using an impedance sensor. *J. Phys. E: Sci. Instrum.*, 18:1985.
- Eccles, A., Green, N., and Porkess, R. (1993a) *MEI Structured Mathematics – Statistics 2*. Sevenoaks, UK: Hodder & Stoughton.
- (1993b) *MEI Structured Mathematics – Statistics 3*. Sevenoaks, UK: Hodder & Stoughton.
- Eide, J. M. (1991) Operational experience, compact prover as a portable calibration unit. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- Eide, J. M., and Gwaspari, S. C. (1996) Comparison test and calibration of Coriolis meters. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- Ekrann, S., Boe, A., Schmidt, F., Jacobsen, E., Time, R. W., and Vatne, H. (1988) *Means and method for measuring a multi-phase distribution within a flowing petroleum stream*. U.S. Patent No. 4,751,842, June 21.
- Ellis, S. M., Hemp, J., and Sanderson, M. L. (1993) A static charge flowmeter for measuring two-phase flows. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 654–60.
- El Wahed, A. K., and Sproston, J. L. (1991) The influence of shedder shape on the performance of the electrostatic vortex flowmeter. *J. Flow Meas. Instrum.*, 2:169–79.
- El Wahed, A. K., Johnson, M. W., and Sproston, J. L. (1993) Numerical study of vortex shedding from different shaped bluff bodies. *J. Flow Meas. Instrum.*, 4:233–40.
- Endress, U., et al. (1989) *Flow Handbook*. Reinach, Switzerland: Flowtec AG (English edition).
- Erdal, A., and Andersson, H. I. (1997) Numerical aspects of flow computation through orifices. *J. Flow Meas. Instrum.*, 8:27–37.
- Erdal, A., and Cabrol, J. F. (1991) Comparison of repeatability, reproducibility and linearity for turbine, coriolis and ultrasonic meters tested at 100 bars on natural gas. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers, October 22–24.
- Erdal, A., Lindholm, D., and Thomassen, D. (1994) Development of a flow conditioner. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- Eren, H. (1995) Particle concentration characteristics and density measurements of slurries using electromagnetic flowmeters. *IEEE Trans. Instrum. Meas.*, 44(3):783–86.
- Erickson, G. P., and Graber, J. C. (1983) Ultrasonic flowmeters for hydroelectric plants. *Mechanical Eng.*, November: 84–8.
- Ewing, J. A. (1924/5) A ball-and-tube flowmeter. *Proc. Roy. Soc. Edinburgh*, 45:308.
- Fakouhi, A. (1977) *The influence of viscosity on turbine flow meter calibration curves*. PhD thesis, University of Southampton.
- Falvey, H. T. (1983) Effect of gradients on acoustic velocity meter. *J. Hyd. Eng.*, 109:1441–53.
- Faraday, M. (1832) Experimental researches in electricity. *Phil. Trans.*, 15:175–7.
- Fenwick, J. S., and Jepson, P. (1975) The problems and needs in large volume gas measurement. *Transducer '75 Conference*.
- Ferreira, V. C. S. (1988) *Flow patterns inside a turbine type flowmeter*. PhD thesis, Cranfield Institute of Technology, England.
- Finkelstein, A., and Finkelstein, L. (1994) Requirements engineering for instrument systems. *Meas. Control*, 27(8):248–50.

- Finlayson, A. J. (1992) Industrial review: Selection of flowmeters for nuclear fuel processing. *J. Flow Meas. Instrum.*, 3(1):3–8.
- Finnof, C., Stainton, D. Saenz, C. P., and Smith, J. E. (1976) *Apparatus and method for measuring fluid mass flow*. British Patent 1,535,817, June.
- Fischbacker, R. E. (1959) The ultrasonic flowmeter. *Trans. Soc. Inst. Tech.*, 11:114.
- Fischer, C. (1994) Development of a metering system for total mass flow and compositional measurements of multiphase/multicomponent flows such as oil/water/air mixtures. *J. Flow Meas. Instrum.*, 5:31–42.
- Fischer, R. (1995) Calculation of the discharge characteristic of an orifice for gas-liquid annular-mist flow. *Int. J. Multiphase Flow*, 21(5):817–35.
- Fling, W. A., and Whetstone, J. R. (1985) Development of basic orifice discharge coefficients. *64th Annu. GPA Convention*, Houston, TX.
- Foster, K. (1998) *An introductory guide to the control of machinery*. London: Professional Engineering Publications.
- Fowles, G. (1994) Introduction. *Meas. Control*, 27(7):213.
- Frank, R., Mazars, J., and Rique, R. (1977) Determination of mass flowrate and quality using a venturi and turbine meter. *Proc. I. Mech. E, Pt. C*, 200:63–76.
- Frankvoort, W. (1989) Results of the evaluation of the performance of mass flow meters using a prover loop. *Mass Flow Measurement Direct and Indirect, Proc. Int. Conf. Mass Flow Measurement*, London: IBC Publications.
- Frantzen, K. H., and Dykesteen, E. (1990) Field experience with CMI multiphase fraction meter. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Fulton, J., Hammer, E. A., and Haugs, A. (1987) Deflection of orifice plates at high differential pressure. *North Sea Flow Metering Workshop*, Stavanger.
- Furness, R. A. (1982) Turbine flowmeters, pp. 171–207. *Developments in Flow Measurement – 1*, (R. W. W. Scott, Ed.), London: Applied Science Publishers.
- (1988) Mass flow measurement – A technological growth area. *Petrol. Rev.*, 42(502):35–8.
- (1989) The application, standardisation and future use of coriolis type mass meters in the oil and process industries. *Mass Flow Measurement Direct and Indirect, Proc. Int. Conf. Mass Flow Measurement*, London: IBC Publications.
- (1991) BS 7405: The principles of flowmeter selection. *J. Flow Meas. Instrum.*, 2(4):233–42.
- Furness, R. A., and Heritage, J. E. (1989) *The Redwood flowmeter directory*. London: IBC Technical Services.
- Furness, R. A., and Jelffs, P. A. M. (1991) Flowmeters – Their role in loss reduction in refining. *Petrol. Rev.*, 45:544–9.
- Gadshiev, E. M., Grigor'yants, S. E., Gusein-zade, K. P., and Smirnov, V. P. (1988) Metrological support to hot-water meters in use and during production. *Meas. Techniques*, 151–4.
- Gainsford, S. (1990) Tested performance of the Hitec/Multi-fluid water fraction meter. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Gajan, P., Mottram, R. C., Hebrard, P., Andriamihafy, H., and Platet, B. (1992) The influence of pulsating flows on orifice plate flowmeters. *J. Flow Meas. Instrum.*, 3:118–29.
- Gallagher, J. E. (1992) Flow measurement – The next ten years. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Gallagher, J. E., LaNasa, P. J., and Beaty, R. E. (1994) The Gallagher flow conditioner. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- Gerrard, D. (1979) Measure viscous flows over 150:1 turndown by PD meter techniques. *Control Instrum.*, 11(4):39–41.
- Gerrard, J. H. (1966) The mechanics of the formation region of vortices behind bluff bodies. *J. Fluid Mech.*, 25:401–13.
- Ginesi, D. (1990) Flow measurement solved with venturi-cone meter. *InTech*, February: 30–32.
- (1991) Choosing the best flowmeter. *Chem. Eng.*, NY, 98(4):88–100.
- Ginesi, D., and Annarummo, C. (1994) Application and installation guidelines for volumetric and mass flowmeters. *ISA Trans.*, 33(1):61–72.
- Gold, R. C., Miller, J. S. S., and Priddy, W. J. (1991) Measurement of multiphase well fluids by positive displacement meter. *Offshore Europe Conf.*, Aberdeen, Scotland: SPE, Paper 23065.
- Gol'dgammer, K. A., Terent'ev, A. T., and Zaaliev, M. I. (1990) Effect of magnetic properties of a medium on the metrological characteristics of an NMR meter for measurement of gas-liquid flows. *Meas. Techniques*, 33(7):676–9.

- Goldstein, S. (1936) A note on the measurement of total head and static pressure. *Proc. Roy. Soc. Ser. A*, 155:570–5.
- (1965) *Modern Developments in Fluid Dynamics*. New York: Dover.
- Goodrich, L. D. (1979) Design and performance of the drag disc turbine transducer. *Int. Colloq.*, Idaho Falls, ID, June.
- Goujon-Durand, S. (1995) Linearity of the vortex meter as a function of fluid viscosity. *J. Flow Meas. Instrum.*, 6:235–8.
- Grattan, E., Rooney, D. H., and Simpson, H. C. (1981) *Two-phase flow through gate valves and orifice plates*. East Kilbride, Scotland: NEL, Report No. 678.
- Gray, D. E., Benjamin, N. M. P., and Chapman, B. N. (1991) Effects of environmental and installation specific factors on process gas delivery via mass flow controller with an emphasis on real time behaviour. *Proc. SPIE, Int. Soc. Optical Eng.*, 1392, 402–10.
- Gray, J. O., and Sanderson, M. L. (1970) Electromagnetic differential flowmeter. *Electronic Lett.*, 6(7):194.
- Gregor, J., Norman, R. S., Bass, R. L., and Spark, C. R. (1993) Establishment of a new natural gas metering research facility for improving flow measurement accuracy under field conditions. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*: 27–41.
- Grendstad, J., Eide, J. and Salvesen, P. (1991) Testing of coriolis meters for metering of oil, condensate and gas. *Proc. North Sea Flow Measurement Workshop*. Norwegian Society of Chartered Engineers, Norway.
- Grenier, P. (1991) Effects of unsteady phenomena on flow metering. *J. Flow Meas. Instrum.*, 2:74–80.
- Grey, J. (1956) Transient response of the turbine flowmeter. *Jet Propulsion*, February: 98–100.
- Griffiths, A., and Newcombe, J. (1970) Large-Volume Gas Measurement 36th Autumn Res. Meet. of the Institution of Gas Engineers.
- Griffiths, C., and Silverwood, P. A. (1986) *Selection and application of flow measurement instrumentation*. HYDRIL Production Technology Division, Bulletin 5126-A.
- Grimley, T. A. (1996) Multipath ultrasonic flow meter performance. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- (1997) Performance testing of ultrasonic flow meters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 19.
- Grini, P. G., Maehlum, H. S., and Brendeng, E. (1994) In situ calibration of coriolis flowmeters for high-pressure gas flow calorimetry. *J. Flow Meas. Instrum.*, 5:285–8.
- Grob, G. R. (1984) Direct mass measurement of fluids – The ultimate answer to process control and loss auditing. *Process Measurement, Control and Applications, PROMECON Control and Instrumentation Exhib. Conf.*, London, England: 190–8.
- Grumski, J. T., and Bajura, R. A. (1984) Performance of a coriolis-type mass flowmeter in the measurement of two-phase (air-liquid) mixtures. *Mass Flow Measurements ASME Winter Annu. Meet.*, New Orleans.
- Guilbert, A. R., and Sanderson, M. L. (1996a) Novel ultrasonic mass flowmeter for liquids. *IEE Colloq. (Dig.)*, No. 092:8/1–8/4.
- (1996b) The development of curved reflective surfaces for ultrasonic beam redirection in high speed gas flow measurement. *Ultrasonics*, 34:441–5.
- Guilbert, A. R., Law, M., and Sanderson, M. L. (1996) A novel ultrasonic/thermal clamp-on flowmeter for low liquid flowrates in small diameter pipes. *Ultrasonics*, 34:435–9.
- Gwaspari, S. C. (1990) Multiple regression footprinting of meter factors. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- H. M. Customs and Excise (1995) *Mineral (hydrocarbon) oils: Duty and VAT: Warehousing and related procedures*, Notice 179.
- Hafner, P. (1985) New developments in magflowmeters. *Int. Conf. on Flow Measurement in the Mid-1980s*, Glasgow, Scotland: National Engineering Laboratory, Paper 9.1.
- Haga, J., Sotono, Y., and Hanai, J. (1995) Development of a fuel vapor flow meter. *JSAE Rev.*, 16(2):185–7.
- Hagenmeyer, H., Schulz, K.-H., Wenger, A., and Keita, M. (1994) Design of an advanced Coriolis mass flowmeter using hoop mode. *FLOMEKO '94 Conf. on Flowmeasurement in the Mid 90s*, Scotland: National Engineering Laboratory.
- Hahn, B. V. (1968) Theory of the sliding-vane meter. *Siemens Rev.*, XXXV (9):362–6.
- Hakansson, E., and Delsing, J. (1992) Effects of flow disturbance on an ultrasonic gas flowmeter. *J. Flow Meas. Instrum.*, 3:227–34.

- (1994) Effects of pulsating flow on an ultrasonic gas flowmeter, *J. Flow Meas. Instrum.*, 5(2):93–101.
- Hall, A., and Shaw, C. (1988) Field experience of two phase flow measurement. *North Sea Flow Metering Workshop*, Paper 2.3.
- Hall, R. (1990) Measuring mass flow and density with coriolis meters. *InTech*, 37(4):45–6.
- Halsey, D. W. (1986) A survey of industrial usage of flowmeters. *Meas. Control*, 19(5):52–5.
- Halttunen, J. (1990) Installation effects on ultrasonic and electromagnetic flowmeters: a model-based approach. *J. Flow Meas. Instrum.*, 1:287–92.
- Hamblett, L. S. J. (1970) Flowmeters. Displacement and inferential types. *Glenfield Gazette*, No. 231:24–7.
- Hannisdal, N-E. (1991) Metering study to reduce topsides weight. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers: 22–24.
- Harrie, P. M. (1991) Mass flow and density. *Meas. Control*, 24.
- Harriger, J. W. (1966) High-pressure measurement. *32nd Autumn Res. Meet. of the Institution of Gas Engineers*.
- Harrison, P. (1978a) The calibration of flowmeters with liquids. *Short Course notes on the Principles and Practice of Flow Measurement*, Lecture No. 6, East Kilbride, Scotland: National Engineering Laboratory.
- (1978b) National standards, transfer standards, traceability and the BCS. *Short Course Notes on the Principles and Practice of Flow Measurement*, Lecture No. 8, East Kilbride, Scotland: National Engineering Laboratory.
- Harrison, P., and Williamson, J. (1985) Accuracy of flowmeters used in a survey of domestic water consumption in Scotland in October 1982. *Proc. Flow Measurement for Water Supply*, London.
- Hastings, C. R. (1968) LE Flowmeter – A new device for measuring liquid flow rates. *Westinghouse Engineer*, 28(6):183.
- (1970) The LE acoustic flowmeter – An application to discharge measurement. *New England Water Works Assoc.*, 84:127.
- Hayes, E. R. (1988) *The prediction of droplet motion and breakup using a vortex model for turbulent flows*. PhD thesis, Cranfield Institute of Technology.
- Hayes, R. H., Wheelwright, S. C., and Clark, K. B. (1988) *Dynamic Manufacturing*. New York: The Free Press, A division of Macmillan Inc.
- Hayward, A. T. J. (1977a) *Repeatability and Accuracy*. London: Mechanical Engineering Publications.
- (1977b) *Measuring the repeatability of flowmeters*. NEL Report No. 636, Dept. of Industry, UK.
- (1979) *Flowmeters: A Basic Guide and Sourcebook for Users*. London: The Macmillan Press.
- Hayward, A. T. J., and Furness, R. A. (1989) A portable gravimeter prover for the in-line proving of direct mass flowmeters. *Mass Flow Measurement Direct and Indirect, Proc. Int. Conf. on Mass Flow Measurement*, London: IBC Publications.
- Haywood, R. W. (1968) *Thermodynamic Tables in SI Units*. Cambridge: Cambridge University Press.
- Head, V. P. (1946–7) An extension of rotameter theory and its application in new practical fields. *Instrument Practice*. December 1946: 64–71; February 1947: 135–41.
- (1956) A practical pulsation threshold for flowmeters. *Trans. ASME*, 78:1471–9.
- Hebrard, P., Malard, L., and Strzelecki, A. (1992) Experimental study of a vortex flowmeter in pulsatile flow conditions. *J. Flow Meas. Instrum.*, 3:173–86.
- Heinrichs, K. (1991) Flow measurement by a new push-pull swirlmeter. *Sensors & Actuators, A: Phys*, 27:809–13.
- Hemp, J. (1975) Improved magnetic field for an electromagnetic flowmeter with point electrodes. *J. Phys. D: Appl. Phys.*, 8:983–1002.
- (1982) Theory of transit time ultrasonic flowmeters. *J. Sound Vibration*, 84(1):133–47.
- (1988) Flowmeters and reciprocity. *QJMAM*, 41(4):503–20.
- (1991) Theory of eddy currents in electromagnetic flowmeters. *J. Phys. D: Appl. Phys.*, 24:244–51.
- (1994a) Weight vector for thermal diffusion flowmeters, Part 1: General theory. *J. Flow Meas. Instrum.*, 5(3):217–22.
- (1994b) The weight vector theory of Coriolis mass flowmeters. *J. Flow Meas. Instrum.*, 5(4):247–53.

- (1994c) Error in potential measurements due to nonuniform contact impedance of electrodes. *Quart. J. Mech. Appl. Math.*, 47(1):175–82.
- (1995a) Weight vector for thermal diffusion flowmeters, Part 2: Application to a particular configuration. *J. Flow Meas. Instrum.*, 6:149–56.
- (1995b) Theory of a simple electromagnetic velocity probe with prediction of the effect on sensitivity of a nearby wall. *Meas. Sci. Technol.*, 6:376–82.
- (1996) A theoretical investigation into the feasibility of Coriolis mass flowmeters for low density fluids. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 265–70.
- (1997) Weight vector theory – Sound fields in moving fluids. *Ultrasonics in Flow Measurement*, Bedford, England: Cranfield University.
- (1998) A review of the weight vector theory of transit time ultrasonic flowmeters. *FLOMEKO '98 9th Int. Conf. on Flow Measurement*, Lund, Sweden.
- Hemp, J., and Hendry, L. A. (1995) The weight vector theory of Coriolis mass flowmeters – Part 2. Boundary source of secondary vibration. *J. Flow Meas. Instrum.*, 6:259–64.
- Hemp, J., and Sanderson, M. L. (1981) Electromagnetic flowmeters – A state of the art review. *BHRA Int. Conf. on Advances in Flow Measurement Techniques*, Coventry, England, Paper E1: 319–40.
- Hemp, J., and Sultan, G. (1989) On the theory and performance of coriolis mass flowmeters. *Mass Flow Measurement Direct and Indirect, Proc. Int. Conf. Mass Flow Measurement*, London: IBC Publications.
- Hemp, J., and Versteeg, H. K. (1986) Prediction of electromagnetic flowmeter characteristics. *J. Phys. D: Appl. Phys.*, 19:1459–76.
- Hemp, J., and Wyatt, D. G. (1981) A basis for comparing the sensitivities of different electromagnetic flowmeters to velocity distribution. *J. Fluid Mech.*, 112:189–201.
- Hendrix, A. R. (1982) Positive displacement flowmeters: High performance – With a little care. *InTech*: 47–9.
- Henke, R. W. (1955) Positive displacement meters. *Control Eng.*, 2(5):56–64.
- Heritage, J. E. (1989) The performance of transit time ultrasonic flowmeters under good and disturbed flow conditions. *J. Flow Meas. Instrum.*, 1:24–30.
- Hermant, C. (1962) Application of flow measurement by the comparative salt-dilution to the determination of turbine efficiency. *Proc. Symp. on Flow Measurement in Closed Conduits*, HMSO, Vol. 2: Paper E-2.
- Herschy, R. W. (1995) *Streamflow Measurement*, 2nd ed., E & F N Spon.
- Herzl, P. J. (1982) A vortex flowmeter with 2 active vortex generators. *Adv. Instrum.*, 37:1205–16.
- Herzog, J. P. (1992) An optical fibre vortex sensor for flow rate measurements. *Sensors & Actuators, A: Phys.*, 32:696–700.
- Herzog, J. P., and Roth, P. (1990–1) Optical fiber flowmeter with temperature correction. *Sensors & Actuators, A: Phys.*, 25(1–3):219–23.
- Herzog, M. W., Brobeil, W., Schafer, R., and Meyre, S. (1993) Breakthroughs in the design of the next generation of electromagnetic water meters. *Flow Measurement for the Utilities*, Amsterdam.
- Hetsroni, G. (1981) *Handbook of Multiphase Systems*. New York: McGraw Hill.
- Higham, E. H., and Johnston, J. S. (1992) A review of 'smart' and 'intelligent' flowmetering systems. FLOMIC, Report No. 18.
- Higham, E. H., Fell, R., and Ajaya, A. (1986) Signal analysis and intelligent flowmeters. *Meas. Control*, 19:47–50.
- Higson, D. J. (1964) The transient performance of turbine flowmeters in water. *J. Sci. Instrum.*, 42:337–42.
- Himpe, U., Gotte, B., and Schatz, M. (1994) Influence of upstream bends on the discharge coefficients of classical venturi tubes and orifice plates. *J. Flow Meas. Instrum.*, 5:209–16.
- Hisham Marshad, A., and Irvine Halliday, D. (1994) Intensity-modulated optical-fibre vortex-shedding flowmeter. *Canad. J. Electr. Comput. Eng.*, 119:75–9.
- Hobbs, J. M., and Humphreys, J. S. (1990) The effect of orifice plate geometry upon discharge coefficient. *J. Flow Meas. Instrum.*, 1:133–40.
- Hochreiter, H. M. (1958) Dimensionless correlation of coefficients of turbine type flowmeters. *Trans. ASME*, October: 1363–8.
- Hofman, F. (1993) Magnetic flowmeter with flowshaping flowtube. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea Research Institute of Standards and Science: 445–51.

- Hogrefe, W., Kirchhof, U., Mannherz, E., Marchewka, W., Mecke, U., Otto, F., Rakebrandt, K.-H., Thone, A., and Wegener, H.-J. (1995) *Guide to Flowmeasurements*, Gottingen: Bailey-Fischer & Porter GmbH.
- Holden, J. L., and Peters, R. J. W. (1990) Practical experiences using ultrasonic flowmeters on high pressure gas. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- (1991) Practical experiences using ultrasonic flowmeters on high pressure gas. *J. Flow Meas. Instrum.*, 2:69–73.
- Hooper, L. J. (1962) Discharge measurements by the Allen salt-velocity method. *Proc. Symp. on Flow Measurement in Closed Conduits*, HMSO, Vol. 2: Paper E-1.
- Hopkins, D., Savage, P. F., and Fox, E. (1995) Problems encountered during research into flow rate, pattern of water consumption and unaccounted-for water losses in urban areas. *J. Flow Meas. Instrum.*, 6:173–9.
- Horner, B., Mesch, F., and Trachtler, A. (1996) A multi-sensor induction flowmeter reducing errors due to non-axisymmetric flow profiles. *Meas. Sci. Technol.*, 7:354–60.
- House, R. K., and Johnson, R. T. (1986) Practical application of hydrostatic fluid bearing design principles to a turbine flow meter. *Instrumentation in the Aerospace Industry, Proc. 32nd Int. Instrumentation Symp.*, Seattle, Washington, USA. [Also in *ISA Trans.*, 26(3):59–63, 1987.]
- Howarth, M. (1994) HART – Standard for 4–20mA digital communications. *Meas. Control*, 27(1):5–8.
- Huijsing, J. H., van Dorp, A. L. C., and Loos, P. J. G. (1988) Thermal mass-flow meter. *J. Phys. E: Sci. Instrum.*, 21:994–7.
- Hulin, J.-P., Fierfort, C., and Condol, R. (1982) Experimental study of vortex emission behind bluff obstacles in a gas liquid vertical two-phase flow. *Int. J. Multiphase Flow*, 8:475–90.
- Hulin, J.-P., and Foussat, A. J. M. (1983) Vortex flowmeter behaviour in liquid-liquid two-phase flow. *Int. Conf. Physical Modelling of Multi-Phase Flow*, Coventry, England: BHRA Fluid Engineering, Paper H3:377–90.
- Hunter, J. J., and Green, W. L. (1975) Blockage and its effect on a drag plate flowmeter. *Conf. on Fluid Flow Measurement in the Mid 1970's*, East Kilbride, Scotland: National Engineering Laboratory, Paper C-2.
- Hussain, Y. A., and Baker, R. C. (1985) Optimised non-contact electromagnetic flowmeter. *J. Phys. E: Sci. Instrum.*, 18:210–13.
- Hussain, Y. A., and Farrant, D. (1994) Coriolis mass flow measurement using single straight tube. *Proc. Study Day on Massflow at RAI (Studiedag 'Massflow')*, Amsterdam: MRBT.
- Hussein, I. B., and Owen, I. (1991) Calibration of flowmeters in superheated and wet steam. *J. Flow Meas. Instrum.*, 2:209–16.
- Hussein, I. B., Owen, I., and Amin, A. M. (1992) Energy metering system for high quality saturated steam. *J. Flow Meas. Instrum.*, 3:235–40.
- Hutton, S. P. (1974) The effect of inlet flow conditions on the accuracy of flowmeters. *Int. Mech. E Conf. Publ.*, 4:1–8.
- (1986) The effects of fluid viscosity on turbine meter calibration. *Flow Measurement in the Mid 80s*, East Kilbride, Scotland: National Engineering Laboratory, Paper 1.1.
- Ifft, S. A. (1996) Partially closed valve effects on the V-cone flowmeter. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 49–54.
- Ifft, S. A., and Mikklesen, E. D. (1993) Pipe elbow effects on V-cone flowmeter FED. *Fluid Meas. Instrum. ASME, Fluid Engineering Division, Proceedings Fluids Engineering Conference*, Washington, DC, 161:37–41.
- Igarashi, T. (1986) Fluid flow around a bluff body used for a Karman vortex flowmeter. *Fluid Control Meas.*, 2:1003–8.
- IGE (1988) *Gas meter installations for pressures not exceeding 100 bar. Recommendations on Gas Measurement Practice*, The Institution of Gas Engineers, (IGE/GM/1) Communication 1388.
- IMC (1998) *Instrument Engineer's Year book*. London: Institute of Measurement and Control.
- Imrie, A. (1994) Communication options in the water industry. *Meas. Control*, 27(7):221–4.
- Inkley, F. A., Walden, D. C., and Scott, D. J. (1980) Flow characteristics of vortex shedding meters. *Meas. Control*, 13:166–70.
- IoP (1982) List of data required for a comprehensive evaluation of a liquid flowmeter. *Petroleum Measurement Paper*, No. 1, Institute of Petroleum, June 1982.

- ISA (1959) Terminology, dimensions and safety practices for indicating variable area meters (Rotameters): RP16.1 Glass tube, RP16.2 Metal tube, RP16.3 Extension type glass tube. Instrument Society of America, Report (recommended practice) RP 16.1.2.3.
- (1960) Nomenclature and terminology for extension type variable area meters (Rotameters). Instrument Society of America, Report (tentative recommended practice) RP 16.4.
- (1961a) Installation, operation, maintenance instructions for glass tube variable area meters (Rotameters). Instrument Society of America, Report (recommended practice) RP 16.5.
- (1961b) Methods and equipment for calibration of variable area meters (Rotameters). Instrument Society of America, Report (tentative recommended practice) RP 16.6.
- ISO 3354:1988 *Measurement of clean water flow in closed conduits – Velocity area method using current meters in full conduits under regular flow conditions*, Geneva: International Organisation for Standardisation.
- 3534:1977 *Statistics – Vocabulary and symbols*. Geneva: International Organisation for Standardisation.
- 3966:1977 *Measurement of clean water flow in closed conduits – Velocity area method using pitot-static tubes*. Geneva: International Organisation for Standardisation.
- 5167–1:1991 *Measurement of fluid flow by means of pressure differential devices – Part 1: Orifice plates, nozzles and Venturi tubes inserted in circular cross-section conduits running full*. Geneva: International Organization for Standardization.
- 5167–1: Amd 1:1998. Section 8.3.2.1 Discharge Coefficient, C. Geneva: International Organisation for Standardisation.
- 5168:1978 *Measurement of fluid flow – Estimation of uncertainty in a flowrate measurement*. Geneva: International Organisation for Standardisation.
- 7498–1:1995. *Information technology open systems interconnection basic reference model*. Geneva: International Organisation for Standardisation.
- ISO/CD 14511:1998 *Measurement of fluid flow in closed conduits – Thermal mass flow meters*. Geneva: International Organisation for Standardisation.
- Ito, H., Watanabe, Y., and Shoji, Y. (1985) A long-radius inlet nozzle for flow measurement. *J. Phys. E: Sci. Instrum.*, 18:88–91.
- Itoh, I., and Ohki, S. (1993) Mass flowmeter detecting fluctuations in lift generated by vortex shedding. *J. Flow Meas. Instrum.*, 4:215–24.
- Jackson, G. A., Gibson, J. R., and Holmes, R. (1989) Three-path ultrasonic flowmeter for small-diameter pipelines. *J. Phys. E: Sci. Instrum.*, 22(8):645–50.
- (1991) Three-path ultrasonic flow meter with fluid velocity profile identification. *Meas. Sci. Technol.*, 2(7):635–42.
- James, R. (1965–6) Metering of steam/water two-phase flow by sharp-edged orifices. *Proc. I Mech. E*, 180:549–72.
- Jamieson, A. W., Johnson, P. A., Spearman, E. P., and Sattary, J. A. (1996) Unpredicted behaviour of Venturi flowmeter in gas at high Reynolds numbers. *North Sea Metering Workshop*, Peebles, Scotland.
- Jepson, P. (1964) Transient response of a helical flowmeter. *J. Mech. Eng. Sci.*, 6:317–20.
- (1967) Currentmeter errors under pulsating flow conditions. *J. Mech. Eng. Sci.*, 9:45–54.
- Jepson, P., and Bean, P. G. (1969) Effect of upstream velocity profiles on turbine flowmeter registration. *J. Mech. Eng. Sci.*, 11:503–10.
- Jepson, P., and Chamberlain, D. (1977) Operating high pressure orifice metering installations. *FLO-CON 77 Proc. Symposium – The Application of Flow Measuring Techniques*, Brighton, England: 285–319.
- Jepson, P., and Chipchase, R. (1975) Effect of plate buckling on orifice meter accuracy. *J. Mech. Eng. Sci.*, 17.
- Jitschin, W., Weber, U., and Hartmann, H. K. (1995) Convenient primary gas flow meter. *Vacuum*, 46(8–10):821–4.
- Johannessen, A. A. (1993) Evaluation of ultrasonic liquid flowmeters. *North Sea Flow Measurement Workshop*, Bergen, Norway.
- John, H., Hain, K., Bruderie, F., Reimann, J., and Vollmer, T. (1982) Tests of an advanced true mass flow meter (TMFM) in gas-liquid flow. *Measurement in Polyphase Flows – 1982, AIAA/ASME Joint Fluids, Plasma, Thermophysics, and Heat Transfer Conference*, St Louis, MO: 55–60.

- Johnson, J. E., Harris, R. E., and McKee, R. J. (1989) Primary mass flow calibration for a recirculating gas flow loop. *FLOMEKO 5th Int. Conf. on Flow Measurement*: 23–30.
- Johnson, M. W. (1990) Computation of flow in a vortex-shedding flowmeter. *J. Flow Meas. Instrum.*, 1:201–8.
- Johnson, M. W., and Farroll, S. (1995) Development of a turbine meter for two-phase flow measurement in vertical pipes. *J. Flow Meas. Instrum.*, 6:279–82.
- Johnson, R. C. (1970) Calculation of the flow of natural gas through critical flow nozzles. *J. Basic Eng.*, 92:580–9.
- (1971) *A set of Fortran IV routines used to calculate the mass flow rate of natural gas through nozzles*. NASA TM X-2240
- Jones, F. E. (1992) Application of the equation to the treatment of laminar flowmeter calibration data. *Ind. Metrol.*, 2:91–6.
- Jongerus, P. F. M., van der Beek, M. P. and van der Grinten, J. G. M. (1993) Calibration facilities for industrial gas flow meters in The Netherlands. *J. Flow Meas. Instrum.*, 4:77–84.
- Joshi, S. G. (1991) Surface-acoustic-wave (SAW) flow sensor. *IEEE Trans. Ultrason. Ferroelectr. Freq. Control*, 38(2):148–54.
- Joslin, G. (1879) British Patent 2428.
- Kalotay, P. (1994) On-line viscosity measurement using Coriolis mass flowmeters. *J. Flow Meas. Instrum.*, 5:303–8.
- Karnik, U., Jungowski, W. M., and Botros, K. (1991) Effects of flow characteristics downstream of elbow/flow conditioner on orifice meter accuracy. *Proc. 9th North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- (1994) Effect of turbulence on orifice meter performance. *J. Offshore Mech. Arctic Eng.*, 116(2):77–85.
- Katys, G. P. (1964) *Continuous Measurement of Unsteady Flow*. Chapter 2, p. 37, London: Pergamon Press.
- Katz, L. (1971) Improved flowmeter accuracy with electromechanical feedback. *Proc. 1st Symp. on Flow – Its Measurement and Control in Science and Industry*, Pittsburgh, PA. (Published by ISA, 1(2):669–78, 1974).
- Kawano, T., Miyata, T., Shikuya, N., Takahashi, S., Handoh, M., Itoh, I. and Biles, B. (1992) Intelligent flowmeter. *Proc. Conf. on Advances in Instrumentation and Control*, ISA, Houston: 997–1009.
- Kay, J. M., and Nedderman, R. M. (1974) *An Introduction to Fluid Mechanics and Heat Transfer*. Cambridge: Cambridge University Press.
- Kaye, G. W. C., and Laby, T. H. (1966) *Tables of Physical and Chemical Constants*. Harlow UK: Longmans.
- Keech, R. P. (1982) The KPC multichannel correlation signal processor for velocity measurement. *Trans. Inst. M. C.*, 4.
- Keita, N. M. (1989a) The zero drift effect in coriolis mass flow meter. *Mass Flow Measurement Direct and Indirect, Proc. Int. Conf. on Mass Flow Measurement*, London: IBC Publications.
- (1989b) Contribution to the understanding of the zero shift effects in Coriolis mass flowmeters. *J. Flow Meas. Instrum.*, 1:39–43.
- (1990) Performance of coriolis mass flowmeters in the metering of light fluids. *Int. Conf. on Flow Measurement of Commercially Important Fluids*, London, England.
- (1994) Behaviour of straight pipe Coriolis mass flowmeters in the metering of gas: theoretical predictions with experimental verification. *J. Flow Meas. Instrum.*, 5:289–94.
- Keyser, D. R. (1973) The calibration correlation function for positive displacement liquid meters. *Trans. ASME, Series I*, 95(2):180–8.
- Kiehl, W. (1991) Difference measurement using Coriolis mass flowmeters. *J. Flow Meas. Instrum.*, 2:135–8.
- Kiehl, W., and Gartner, U. (1989) Two coriolis meters in one line. *Mass Flow Measurement Direct and Indirect, Proc. Int. Conf. Mass Flow Measurement*, London: IBC Publications.
- Kim, C. H., Lee, D. K., and Paik, J. S. (1993a) Mean velocity measurement of pipe flow by ultrasonic correlation. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 550–558.
- Kim, R. K., Swain, J. C., Kramer, G. S., Cooper, D. L., Schuluer, L. E., and Haubert, T. D. (1993b) Progress on development of a compact gas meter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea Research Institute of Standards and Science: 393–402.

- Kim, Y., and O'Neal, D. L. (1995) Comparison of critical flow models for estimating two-phase flow of HCFC22 and HFC134a through short tube orifices. *Int. J. Refrigeration*, 18(7):447–55.
- King, L. V. (1914) On the convection of heat from small cylinders in a stream of fluid. *Phil. Trans. Roy. Soc.*, A214:373.
- King, N. W. (1988) Multi-phase flow measurement at NEL. *Meas. Control*, 21(8):237–239.
- (1990) Subsea multi-phase flow metering a challenge for the offshore industry? *Subsea 90 Int. Conf.*, London.
- (1991) A new oil and multi-phase flow laboratory at NEL. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- (1992) Building a new National Standard for oil and multiphase flow measurement. *J. Flow Meas. Instrum.*, 3:59–64.
- King, N. W., Sidney, J. K., and Coulthard, J. (1988) Cross-correlation flow measurements in oil-air mixtures. *2nd Int. Conf. on Flow Measurement*, London: BHRA.
- Kinghorn, F. C. (1982) The analysis and assessment of data, pp. 307–26. *Developments in Flow Measurement – 1* (R. W. W. Scott, Ed.), London: Applied Science Publishers.
- (1986) The expansibility correction for orifice plates: EEC data. *Int. Conf. on Flow Measurement in the Mid 80's*, Paper 5.2.
- (1988) Challenging areas in flow measurement. *Meas. Control*, 21(8):229–35.
- (1996) Industrial needs for cost-effective flow measurement. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 741–50.
- Kinghorn, F. C., and McHugh, A. (1981) The performance of turbine meters in two-component gas/liquid flow. *FLOW 81 Its Measurement and Control in Science and Industry*, St Louis, MO, Vol 2:471–92.
- Kinghorn, F. C., McHugh, A., and Dyet, W. D. (1991) The use of etoile flow straighteners with orifice plates in swirling flow. *J. Flow Meas. Instrum.*, 2:162–8.
- Kivilis, S. S., and Reshetnikov, V. A. (1965) Effect of a stabilised flow profile on the error of ultrasonic flowmeters. *Meas. Techniques*, No. 3:276.
- Kjolberg, S. A., and Berentsen, H. (1997) The Porsgrunn 2 test programme of multiphase meters: general results and examples of different meter performance. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 2.
- Klemp, S., Berentsen, H., and Pedersen, B. L. (1997) Asgard and Gullfaks satellites field developments – Efficient integration of multiphase meters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 4.
- Kleppe, K., and Danielsen, H. B. (1993) Scaling problems in the oil metering system at the Veslefrikk Field. *North Sea Flow Measurement Workshop*, Bergen, Norway.
- Kochner, H., Melling, A., and Baumgartner, M. (1996) Optical flow field investigation for design improvements of an ultrasonic gas meter. *J. Flow Meas. Instrum.*, 7:133–40.
- Kolahi, K., Gast, Th., and Rock, H. (1994) Coriolis mass flowmeasurement of gas under normal conditions. *J. Flow Meas. Instrum.*, 5:275–83.
- Kolin, A. (1936) An electromagnetic flowmeter. The principle of the method and its application to blood flow measurement. *Proc. Soc. Exp. Biol.*, 35:53.
- (1941) An AC flowmeter for measurement of blood flow in intact blood vessels. *Proc. Soc. Exp. Biol.*, 46:235.
- Kolmakov, I. A., and Korolev, V. N. (1990) Alfven-wave flowmeters. *Meas. Techniques*, 33(5):482–5.
- Komiya, K., Higuchi, F., and Ohtani, K. (1988) Characteristic of a thermal gas flowmeter. *Rev. Sci. Instrum.*, 59:477–9.
- Koning, H., Van Essen, G. J., and Smid, J. (1989) Time behaviour of turbine meters – Statistical analysis of (re)calibration results of turbine meters. *FLOMEKO '89 Proc. 5th Int. Conf. on Flow Measurement*: 333–40.
- Koudal, O., Bitto, E., and Wenger, A. (1996) A solution to the problem of installation sensitivity of small Coriolis mass flowmeters. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 256–9.
- Krafft, R., Hemp, J., and Sanderson, M. L. (1996) Investigation into the use of the electromagnetic flowmeter for two-phase flow measurements. *Advances in Sensors for Fluid Flow Measurement, IEE Colloq. (Dig.)* No. 092: 5/1–5/4.
- Kratzer, W., and Kefer, V. (1988) Two phase flow instrumentation: a survey and operational experience with new and easy-to-handle devices. *Cranfield Short Course Lecture*.

- Kristensen, B. D., Lofsei, C., and Froysa, K-E. (1997) Testing of noise suppression system for multipath ultrasonic gas flow meters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 17.
- Kritz, J. (1955) An ultrasonic flowmeter for liquids. *Proc. ISA*, 10:1-55-15-3.
- Kruger, G. J., Birke, A., and Weiss, R. (1996) Nuclear magnetic resonance (NMR) two-phase mass flow measurements. *J. Flow Meas. Instrum.*, 7:25-37.
- Kulin, G. (1975) *A Guide to Methods and Standards for the Measurement of Water Flow*. NBS Special Publication 421.
- Kurz, J. L. (1992) Characteristics and applications of industrial thermal mass flow transmitters. *Proc. Annu. Symp. on Instrumentation for the Process Industries*, Texas: 107-13.
- Lai, M-C., Lee, T., Xu, J. S., and Kwak, S. (1991) Inlet flow characterization of thermal mass air flow meters. *SAE Trans.*, 100:813-19.
- Laneville, A., Strzelecki, A., Gajan, P., and Hebrard, P. (1993) Signal quality of a vortex flowmeter exposed to swirling flows. *J. Flow Meas. Instrum.*, 4:151-4.
- Lange, U., Levien, A., Pankratz, T., and Raschillier, H. (1994) Effect of detector masses on calibration of Coriolis flowmeters. *J. Flow Meas. Instrum.*, 5:255-62.
- Langsholt, M., and Thomassen, D. (1991) The computation of turbulent flow through pipe fittings and the decay of the disturbed flow in a downstream straight pipe. *J. Flow Meas. Instrum.*, 2:45-55.
- Lapszewicz, J. A. (1991) Device for measurement of volumetric flow rates of gas mixtures. *Meas. Sci. Technol.*, 2(8):815-17.
- Laws, E. M. (1990) Flow conditioning – A new development. *J. Flow Meas. Instrum.*, 1:165-70.
- (1991) A further study of flow through tube bundles. *FLUCOME '91, 3rd Triennial Intl. Symp. on Fluid Control*, ASME, Measurement and Visualization, San Francisco, CA: 635-41.
- Laws, E. M., and Harris, R. (1993) Evaluation of a swirl-vor-tab flow conditioner. *J. Flow Meas. Instrum.*, 4:101-8.
- Laws, E. M., and Ouazzane, A. (1992) Effect of plate depth on the performance of a Zanker flow straightener. *J. Flow Meas. Instrum.*, 3:257-69.
- Laws, E. M., and Ouazzane, A. K. (1995a) A further investigation into flow conditioner design yielding compact installations for orifice plate flow metering. *J. Flow Meas. Instrum.*, 6:187-99.
- (1995b) A further study into the effect of length on the Zanker flow conditioner. *J. Flow Meas. Instrum.*, 6:217-24.
- (1995c) A preliminary study into the effect of length on the performance of the Etoile flow straightener. *J. Flow Meas. Instrum.*, 6:225-33.
- Lawson, B. (1988) Field experience using Coriolis mass meters I. *North Sea Metering Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Lawton Smith, H. (1994) Innovation strategies in the flow measuring industry: the cases of the UK, France and Belgium. *J. Flow Meas. Instrum.*, 5(1):23-30.
- Leder, A. (1996) LDA-measurements in the near wake flow of floats for variable-area flowmeters. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 468-73.
- Lee, W. F. Z., and Evans, H. J. (1965) Density effect and Reynolds number effect on gas turbine flowmeters. *J. Basic Eng., Trans. ASME*, 1043-57.
- (1970) A field method of determining gas turbine meter performance. *J. Basic Eng., Trans. ASME*, 724-31.
- Lee, W. F. Z., Blakeslee, D. C., and White, R. V. (1982) A self-correcting and self-checking gas turbine meter. *J. Fluids Eng.*, 104:143-8.
- Lee, W. F. Z., Kirik, M. J., and Bonner, J. A. (1975) Gas turbine flowmeter measurement of pulsating flow. *J. Eng. Power, Trans. ASME*, 531-9.
- Legtenberg, R., Bouwstra, S., and Fluitman, J. H. J. (1991) Resonating microbridge mass flow sensor with low-temperature glass-bonded cap wafer. *Sensors & Actuators, A: Phys.*, 27 (1-3):723-7.
- Lemon, D. D. (1995) Measuring intake flows in hydroelectric plants with an acoustic scintillation flowmeter. *ASCE Waterpower – Proc. Int. Conf. on Hydropower*, 3:2039-48.
- Lenn, C. P., and Oddie, G. M. (1990) The use of ultrasonic methods for monitoring secondary components (solid, liquid and gas) entrained in bulk liquid flows. *Int. Conf. on Basic Principles and Industrial Applications of Multiphase Flow*, London: IBC Technical Services.
- Letton, W. (1994) A cooperative approach to new product development. *North Sea Flow Measurement Workshop*, Peebles, Scotland.

- Lewis, D. C. G. (1975) Further development of a low-loss flowmetering device (Epiflo) based on the pressure difference principle. *Conf. Fluid Flow Measurement in the Mid-1970s*, Paper J3; Glasgow, Scotland: National Engineering Laboratory.
- Li, Y. T., and Lee, S. Y. (1953) A fast-response true-mass-rate flowmeter, *Trans. ASME*, 75:835–41.
- Lin, Z. H. (1982) Two-phase flow measurements with sharp-edged orifices. *Int. J. Multiphase Flow*, 8:683–93.
- Lindner, K-P. (1990) Fieldbus – A milestone in field instrumentation technology. *Meas. Control*, 23:272–7.
- Liu, C. Y., Lua, A. C., Chan, W. K., and Wong, Y. W. (1995) Theoretical and experimental investigations of capacitance variable area flowmeter. *Trans. Inst. Meas. Control*, 17(2): 84–9.
- Liu, K. T., Canfield, D. R., and Conley, J. T. (1986) Application of a mass flow meter for allocation measurement of crude oil production. *SPE Production Eng.*, 3(4):633–6.
- Liu, K. T., and Revus, D. E. (1988) Net-oil computer improves water-cut determination. *Oil & Gas J.*, Dec.
- Lowell, F. C., and Walsh, J. J. (1991) Performance analysis of multipath acoustic flowmeters under various hydraulic conditions. *Proc. Int. Conf. on Hydropower*, Part 3:2041–50.
- Lucas, G. P., and Turner, J. T. (1985) Influence of cylinder geometry on the quality of its vortex shedding signal. *FLOMEKO '85*, Paper C4:81–3.
- Luntta, E., and Hattlunen, J. (1989) Effect of velocity profile on electromagnetic flow measurement. *Sensors & Actuators*, 16(4):335–44.
- Lygre, A., Folkestad, T., Sakariassen, R., and Aldal, D. (1992) A new multi-path ultrasonic flow meter for gas. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland.
- Lynch, F., and Horciza, E. (1995) Flow measurement using low cost portable clamp-on ultrasonic flowmeters. *ASCE Waterpower – Proc. Int. Conf. on Hydropower*, 1:766–73.
- Lynnworth, L. C. (1988) Buffer rod designs for ultrasonic flowmeters at cryogenic and high temperature, plus and minus 200°C. *Proc. 34th Int. Instrumentation Symp.*, Albuquerque, NM, ISA: 697–702.
- (1989) *Ultrasonic Measurements for Process Control: Theory, Techniques, Applications*, Academic Press.
- (1990) Flexural wave externally-attached mass flowmeter for two-phase fluids in small-diameter tubing, 1-mm ID to 16-mm ID. *Proc. IEEE Ultrasonics Symp.*, Honolulu: 1557–62.
- (1994) Clamp-on transducers for measuring swirl, cross flow and axial flow. *Proc. IEEE Ultrasonics Symp.*, Cannes, France: 1317–21.
- Lynnworth, L. C., Hallewell, G. D., and Bragg, M. I. (1994) One-port profiler. *FLOMEKO '94 Flow Measurement in the Mid-90s*, Glasgow, Scotland: NEL, Paper 7.3.
- Macdonald, G. A. (1983) A vibrating vane mass flowmeter. *1st European Conf. on Sensors and Their Applications*, UMIST, Manchester: 58–9.
- MacMillan, F. A. (1954) Viscosity effects on Pitot tubes at low speeds. *J. Roy. Aero. Soc.*, 58: 570–2.
- (1957) Experiments on Pitot tubes in shear flow. *Aero. Res. Council, Tech. Rpt.*, R&M 3028, Vol. 58:570–72.
- Mainardi, H., Barriol, R., and Panday, P. K. (1977) Pulsating duct flow in the presence of an orifice plate. *Int. J. Mech. Sci.*, 19:533–46.
- Mair, W. A. (1965) The effect of a rear mounted disc on the drag of a blunt body of revolution. *Aeronautical Quart.*, 16:350–60.
- Majeed, G. H. A., and Aswad, Z. A. A. (1989) A new approach for estimating the orifice discharge coefficient required in Ashford-Pierce correlations. *Multiphase Flow – Proc. 4th Int. Conf.*, BHRA, Cranfield: 235–55.
- Majumdar, A. S., and Gulek, M. (1981) *Vortex shedding from single and compound prisms of various configurations*. ASME Paper No. 81-WA/FE-6.
- Malard, L., Wisnoe, W., Strzelecki, A., Gajan, P., and Hebrard, P. (1991) Air visualizations and flow measurements applied to the study of a vortex flowmeter: Influence of grid turbulence and acoustical effects. *FLUCOME '91, 3rd Triennial Int. Symp. on Fluid Control, Measurement and Visualization*, San Francisco, CA: 689–95.
- Mandrup-Jensen, L. (1990) Testing coriolis mass flowmeters for pattern approval. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Mankin, P. A. (1955) Measurement of liquid flow by positive displacement meters. *J. Southern California Meter Association, Instruments and Automation*, 453–7.

- Mark, P. A., Johnson, M. W., Sproston, J. L., and Millington, B. C. (1990b) The turbine meter applied to void fraction determination in two-phase flow. *J. Flow Meas. Instrum.*, 1:246–52.
- Mark, P. A., Sproston, J. L., and Johnson, M. W. (1990a) Theoretical and experimental studies of two-phase flows in turbine meters. *Int. Conf. Basic Principles and Industrial Applications of Multiphase Flow*, London: IBC Technical Services.
- Martin, J. J. (1949) Calibration of rotameters. *Chem. Eng. Prog.*, 45:338.
- Matsunaga, Y., Goto, S., Kuromori, K., and Ostling, H. (1988) New intelligent magnetic flowmeter with dual frequency excitation. *Proc. ISA/88 Int. Conf. Exhib.: Advances in Instrumentation*, ISA, 43(3):1259–67.
- Matsunaga, Y., Takahashi, S., and Kuromori, K. (1990) Numerical analysis of a vortex flowmeter and comparison with experiment. *J. Flow Meas. Instrum.*, 1:106–12.
- Mattar, L., Nicholson, M., Aziz, K., and Gregory, G. A. (1979) Orifice metering of two-phase flow. *J. Petrol. Technol.*, August: 955–61.
- Matthews, A. J., and Ayling, C. L. (1992) Compact large bore direct mass flow meters. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland.
- Mattingly, G. E. (1982) Primary calibrators, reference and transfer standards, pp. 31–71. *Developments in Flow Measurement – 1* (R. W. W. Scott, Ed.), London: Applied Science Publishers.
- (1990–1) Fluid flowrate metrology: Laboratory uncertainties and traceabilities. *Advanced Techniques for Integrated Circuit Processing*, *Int. Soc. Optical Eng.*, *Proc. SPIE*, 1392:386–401.
- Mattingly, G. E., and Yeh, T. T. (1991) Effects of pipe elbows and tube bundles on selected types of flowmeters. *J. Flow Meas. Instrum.*, 2:4–13.
- Mattingly, G. E., Pontius, P. E., Allion, H. H., and Moore, E. F. (1977) Laboratory study of turbine meter uncertainty. *Proc. Symp. on Flow Measurement in Open Channels and Closed Conduits*, NBS, Gaithersburg, MD: 33–54.
- Mattingly, G. E., Pursley, W. C., Paton, R., and Spencer, E. A. (1978) Steps towards an ideal transfer standard for flow measurement. *FLOMEKO Symp. on Flow*, Groningen, The Netherlands: 543–52.
- Mattingly, G. E., Yeh, T. T., Robertson, B., and Kothari, K. M. (1987) NBS research on ‘in situ’ flowmeter calibrations. *AGA Distribution Transmission Congress*, Las Vegas, NV.
- Mazzoni, A., Villa, M., and Bonuccelli, M. (1994) AGIP – Trecate multiphase test loop test facility description and specification. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- McCarthy, R. (1996) Five-path ultrasonic flowmeter completes one-year field trial. *Pipe Line & Gas Industry*, 79(4).
- McFaddin, S. E., Sindt, C. F., and Brennan, J. A. (1989) The effect of the location of an in-line tube bundle on orifice flowmeter performance. *J. Flow Meas. Instrum.*, 1:9–14.
- McKee, R. J. (1992) Pulsation effects on single- and two-rotor turbine meters, *J. Flow Meas. Instrum.*, 3:151–66.
- McKenzie, G. (1989) The performance of direct mass flow coriolis meters used for fiscal measurement of high value fluids, *Mass Flow Measurement Direct and Indirect*, *Proc. Int. Conf. Mass Flow Measurement*, London: IBC Publications.
- Medlock, R. S. (1976) The vortex flowmeter – Its development and characteristics. *Austral. J. Instrum. Control*, 24:24–32.
- (1986) The historical development of flow metering. *Meas. Control*, 19:11–22.
- Medlock, R. (1989) A review of the techniques of mass flow measurement. *Cranfield Short Course Lecture*. See also *Mass Flow Measurement Direct and Indirect*, *Proc. Int. Conf. on Flow Measurement*, London: IBC Publications.
- Medlock, R., and Furness, R. A. (1990) Mass flow measurement – A state of the art review. *Meas. Control*, 23:100–12.
- Mencke, D. (1989) Pattern approval of mass flowmeters. *Mass Flow Measurement Direct and Indirect*, *Proc. Int. Conf. on Mass Flow Measurement*, London: IBC Publications.
- (1996) Use of Coriolis mass flowmeters in custody transfer. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 232–37.
- Miau, J. J., and Hsu, M. T. (1992) Axisymmetric-type vortex shedders for vortex flowmeters. *J. Flow Meas. Instrum.*, 3:73–80.
- Miau, J. J., Yang, C. C., Chou, J. H., and Lee, K. R. (1993) A T-shaped vortex shedder for a vortex flowmeter. *J. Flow Meas. Instrum.*, 4:259–68.
- Miller, D. S. (1990) *Internal Flow Systems*, 2nd ed., London: Gulf Publishing Company.
- Miller, R. W. (1996) *Flow Measurement Engineering Handbook*, 3rd ed., New York: McGraw-Hill.

- Miller, R. W., DeCarlo, J. P., and Cullen, J. T. (1977) A vortex flowmeter – Calibration results and application experiences. *NBS Special Publication 484*, 2:549–70.
- Millington, B. C., Adams, C. W., and King, N. W. (1986) The effect of upstream installation conditions on the performance of small liquid turbine meters. *Int. Symp. on Fluid Flow Measurement*, AGA, Washington.
- Millington, B. C., and King, N. W. (1986) The performance of a turbine meter in gas/liquid flow with upstream flow conditioning. *Int. Conf. on Flow Measurement in the Mid '80s*, Glasgow, Scotland.
- (1988) Further developments of a jet mixer/turbine meter package for the measurement of gas-liquid mixtures. *FLUCOME '88 2nd Int. Symp. on Fluid-Control Measurement Mechanics and Flow Visualization*, Sheffield, England: 474–8.
- Millington, B. C., Frantzen, K., and Marshall, M. (1993) The performance of the Fluenta MPFM 900 Phase Fraction Meter. *North Sea Flow Measurement Workshop*, Bergen, Norway.
- Mills, C. J. (1966) A catheter tip electromagnetic velocity probe. *Phys. Med. Biol.*, 11:323–4.
- Minkin, H. L., Hobart, H. F., and Warshawsky, I. (1966) *Performance of turbine type flowmeter in liquid hydrocarbons*, NASA TN D-3770.
- Morgan, D. V., and Aliyu, Y. H. (1993) An ionic flowmeter for measuring small rates of gas flow. *Meas. Sci. Technol.*, 4:1479–83.
- Morrison, G. L., DeOtte, R. E., Moen, M., Hall, K. R., and Holste, J. C. (1990a) Beta ratio, swirl and Reynolds number dependence of wall pressure in orifice flowmeters. *J. Flow Meas. Instrum.*, 1:269–77.
- Morrison, G. L., DeOtte, R. E., Panak, D. L., and Nail, G. H. (1990b) Flow field inside an orifice flow meter. *Chem. Eng. Prog.*, 86(7):75–80.
- Morrison G. L., DeOtte, R. E., and Beam, E. J. (1992) Installation effects upon orifice flowmeters, *J. Flow Meas. Instrum.*, 3:89–94.
- Morrison, G. L., Hall, K. R., Holste, J. C., DeOtte Jr, R. E., Macek, M. L., and Ihfe, L. M. (1994a) Slotted orifice flowmeter. *AIChE J.*, 40(10):1757–60.
- Morrison, G. L., Hall, K., Holste, J. C., Macek, M., Ihfe, L., and DeOtte, R. E. (1994b) Comparison of orifice and slotted plate flow meters. *J. Flow Meas. Instrum.*, 5:71–7.
- Morrison, G. L., Hauglie, J., and DeOtte, R. E. (1995) Beta ratio, axisymmetric flow distortion and swirl effects upon orifice flow meters. *J. Flow Meas. Instrum.*, 6:207–16.
- Morrow, T. B., and Park, J. T. (1992–3) *Baseline conditions for orifice meter calibration*. Report GRI-92/0097, Gas Research Institute, Chicago, IL.
- Morrow, T. B., Park, J. T., and McKee, R. J. (1991) Determination of installation effects for a 100 mm orifice meter using a sliding vane technique. *J. Flow Meas. Instrum.*, 2:14–20.
- Morse, J. V. (1976) Field use of turbine meters in sand-laden oil. *J. Pet. Technol.*, 28:139–42.
- Morse, P. M., and Ingard, K. U. (1968) *Theoretical Acoustics*. New York: McGraw Hill.
- Mottram, R. C. (1981), Measuring pulsating flow with a differential pressure meter. *Proc. Conf. Flow 81 – Its Measurement and Control in Science and Industry*, ISA, St. Louis, MO, 2:347–61.
- (1989) Damping criteria for pulsating gas flow measurement. *J. Flow Meas. Instrum.*, 1:15–23.
- (1991) Vortex flowmeters – Installation effects. *J. Flow Meas. Instrum.*, 2:56–60.
- (1992) Introduction: An overview of pulsating flow measurement. *J. Flow Meas. Instrum.*, 3:114–17.
- Mottram, R. C., and Hutton, S. P. (1987) Installation effects turbine and vortex flowmeters. *FLOMIC Report No 3, Flow Measurement and Instrumentation Consortium*.
- Mottram, R. C., and Rawat, M. S. (1986) The swirl damping properties of pipe roughness and the implications for orifice meter installation. *Int. Conf. on Flow Measurement in the Mid 80's*, June 9–12. Glasgow, Scotland: NEL.
- (1988) Installation effects on vortex flowmeters. *J. Meas. Control*, 21:241–6.
- Mottram, R. C., and Ting, V. C. (1992) Presentation at 1992 AIChE Spring National Meeting, New Orleans, LA.
- Muhlemann, A., Oakland, J., and Lockyer, K. (1992) *Production and Operations Management* (6th ed.), London: Pitman.
- Murakami, M., Maruo, K., and Yoshiki, T. (1990) Development of an electromagnetic flowmeter for studying gas-liquid, two-phase flow. *Int. Chem. Eng.*, 30(4):699–702.
- Murdock, J. W. (1961) Two-phase flow measurement with orifice. *ASME Paper 61-GT-27*.
- Myhr, S. (1991) Field experience with coriolis mass meter on hydrocarbon liquid. *Proc. North Sea Flow Measurement Workshop*.

- Mylvaganam, K. S. (1989) High-rangeability ultrasonic gas flowmeter for monitoring flare gas. *IEEE Trans. Ultrason., Ferroelectr. Freq. Control*, 36(2):144–9.
- Nakano, K., and Tanaka, Y. (1990) Electrostatic flowsensor. *J. Flow Meas. Instrum.*, 1:191–200.
- Nakao, S-i., Yokoi, Y., and Takamoto, M. (1996) Development of a calibration facility for small mass flow rates of gas and the uncertainty of a sonic venturi transfer standard, *J. Flow Meas. Instrum.*, 7:77–83.
- NAMAS (1997) *The Expression of Uncertainty and Confidence in Measurement*, 1st ed. Teddington, UK: United Kingdom Accreditation Service.
- Nederlof, A. J. (1994) Product certification of the future. *J. Flow Meas. Instrum.*, 5(2):115–20.
- NEL (1997a) Installation effects on venturi tubes. *Flow Measurement Guidance Note*, East Kilbride, Scotland: National Engineering Laboratory, No. 2.
- (1997b) Ultrasonic meters for oil flow measurement. *Flow Measurement Guidance Note*, East Kilbride, Scotland: National Engineering Laboratory, No. 6.
- Newcombe, J., and Griffiths, A. (1973) High throughput flowmeters for gas sales and grid control. *12th World Gas Congr.*, Nice, France, Paper IGU/D: 12–73.
- Newcombe, J., Archbold, T., and Jepson, P. (1972) Errors in measuring gas flows at high pressure – Recent developments in correcting methods. *38th Autumn Res. Meet. Institution of Gas Engineers*.
- Nguyen, N. T. (1997) Micromachined flow sensors – A review. *J. Flow Meas. Instrum.*, 8:7–16.
- Nguyen, N. T., and Kiehnscherf, R. (1995) Low-cost silicon sensors for mass flow measurement of liquids and gases. *Sensors & Actuators, A: Phys.*, A49(1–2):17–20.
- Nicholson, S. (1994) Coriolis mass flow measurement. *FLOMEKO '94 Conf. on Flowmeasurement in the Mid 90s*, Scotland: NEL.
- Nishigaki, M., Ippommatsu, M., Ikeda, Y., and Nakajima, T. (1995) Measurement principle of the fluidic gas flowmeter. *Meas. Sci. Technol.*, 6(6):833–42.
- Nolan, M. E., Gaskell, M. C., and Cheung, W. S. (1985) Further developments of the British gas ultrasonic flowmeter. *Flowmeasurement in the Mid'80s*, Paper 11.2.
- Noltingk, B. E., Ed. (1988) *Instrumentation Reference Book*. London: Butterworths.
- Norman, R., Rawat, M. S., and Jepson, P. (1983) Buckling and eccentricity effects on orifice metering accuracy. *Int. Gas Research Conf.*, London.
- (1984) An experimental investigation into the effects of plate eccentricity effects and elastic deformation on orifice metering accuracy. *Int. Conf. on the Metering of Natural Gas and Liquefied Hydrocarbon Gases*, London.
- Oguri, Y. (1988) Wedge flowmeters for measuring bi-directional pipe flows. In *Mass Flow Measurement*, presented at ASME Winter Annual Meeting (Chicago, Illinois, 27 November–2 December 1988) (G. E. Mattingly and T. R. Hendrick, Eds.), FED Vol 73, pp 1–5 (ASME, New York).
- Ohlmer, E., and Schulze, W. (1985) Experience with CENG full- flow turbinometers for transient two-phase flow measurements under loss-of-coolant experiment conditions. *BHRA 2nd Int. Conf. on Multi-phase Flow*, London, England, Paper H1: 381–95.
- Okabayashi, M., and Yamasaki, H. (1991) Feasibility study of new fluidic gas meters. *FLUCOME '91, 3rd Triennial Int. Symp. on Fluid Control, Measurement and Visualization*, San Francisco, CA: 313–18.
- Okland, O., and Berentsen, H. (1994) Using the MFI multiphase meter for well testing at Gullfaks B. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- Olsen, E. (1991) An investigation of sonic and ultrasonic flowmeters with transducers in free stream. *J. Flow Meas. Instrum.*, 2:185–7.
- Olsen, L. F. (1974) *Introduction to liquid flowmetering and the calibration of liquid flowmeters*, National Bureau of Standards TN 831.
- Oppenheim, A. K., and Chilton, E. G. (1955) Pulsating flow measurement – A literature survey. *Trans. ASME*, 77:231–48.
- Orlando, V. A., and Jennings, F. B. (1954) The momentum principle measures true mass flow rate. *Trans. ASME*, 76:961–5.
- O'Rourke, E. L. (1993) The MTI compact electronic gas meter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea Research Institute of Standards and Science: 424–33.
- (1996) Results of the MTI compact electronic meter test program. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 313–18.
- Owen, I., and Hussein, I. B. (1991) Wet steam flowmeter correction factors. *J. Flow Meas. Instrum.*, 2:139–40.

- Owen, I., Hussein, I. B., and Amini, A. M. (1991) The impact of water slugs on wet steam flowmeters. *J. Flow Meas. Instrum.*, 2:98–104.
- Ower, E. (1937) On the response of a vane anemometer to an air-stream of pulsating speed. *Phil. Mag.*, Series 7, 23(157):992.
- Ower, E., and Pankhurst, R. C. (1966) *The Measurement of Air Flow*. Oxford: Pergamon Press.
- Paik, J. S., Lim, K. W., and Lee, K. B. (1990) Calibration of coriolis mass flowmeters using a dynamic weighing method. *J. Flow Meas. Instrum.*, 1:171–5.
- Paik, J. S., Mim, C. H., and Lee, D. K. (1994) Effect of variation of pipe velocity profile on the ultrasonic cross-correlation flowmeters. *FLOMEKO '94 Flow Measurement in the Mid-90s*, Glasgow, Scotland: NEL, Paper 7.1.
- Pal, R. (1993) Flow of oil-in-water emulsions through orifice and venturi meters. *Ind. Eng. Chem. Res.*, 32:1212–17.
- Pal, R., and Rhodes, E. (1985) Methods for metering oil and water production of wells. *BHRA 2nd Int. Conf. on Multi-phase Flow*, London, Paper H2: 397–411.
- Pannell, C. N., Evans, W. A. B., and Jackson, D. A. (1990) A new integration technique for flowmeters with chordal paths. *J. Flow Meas. Instrum.*, 1:216–24.
- Park, K-A. (1995) Effects of inlet shapes of critical venturi nozzles on discharge coefficients. *J. Flow Meas. Instrum.*, 6:15–19.
- Parker, M. (1990) Improving the performance of a glass tube variable area flowmeter. *Meas. Control*, 23:211–15.
- Parkinson, G. J. (1991) Fluidic flow sensors for industrial applications. *Meas. Control*, 24:4–10.
- Patel, B. R., and Sheikholeslami, Z. (1986) Numerical modelling of turbulent flow through orifice meters. *Int. Symp. on Fluid Flow Measurement*, Washington, DC.
- Paton, R. (1988) Calibration techniques for mass flowmeters. *Petrol. Rev.*, 42(502):40–2.
- Paulsen, F. (1991) Prover ball material problems. *Proc. of the North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- Pavlovic, V., Dimitrijevic, B., Stojcev, M., Golubovic, Lj., Zivkovic, M., and Stamenkovic, Lj. (1997) Realization of the ultrasonic liquid flowmeter based on the pulse-phase method. *Ultrasonics*, 35:87–102.
- Pawlas, G. E., and Pankratz, T. (1994) Fluid mechanics effects in Coriolis mass flowmeters, *FLOMEKO '94 Conf. on Flowmeasurement in the Mid 90s*, East Kilbride, Scotland: NEL.
- Pawlas, G., and Patten, T. (1995) Gas measurement using coriolis mass flowmeters. *ISA Advances in Instrumentation and Control: Int. Conf. Exhib.*, 50(3):781–90.
- Pereira, M. T., and Nunes M. (1993) Nozzle chamber to measure flow rates up to 5000 m³/h. *FOMEKO Proc. 6th Int. Conf. on Flow Measurement*, Korea: 380–386.
- Pereira, M. T., Pimenta, M. de M., and Taira, N. M. (1993) Flow metering with a modified sonic nozzle. *FLOMEKO '93 Proc. 6th Int. Conf. Flow Measurement*: 372–9.
- Peters, F., and Kuralt, T. (1995) A gas flowmeter of high linearity. *J. Flow Meas. Instrum.*, 6:29–32.
- Peters, J., and Schook, C. A. (1981) Electromagnetic sensing of slurry concentration. *Canad. J. Chem. Eng.*, 59:430–7.
- Pfrehm, R. H. (1981) Improved turbine-meter system measures ethylene accurately. *Oil & Gas J.*, 79(16):73–6.
- Place, J. D., and Maurer, R. (1986) Non-invasive fibre optic pick-up for a turbine flowmeter. *Conf. on Fibre Optics 86*, London.
- Plache, K. O. (1977) Coriolis/gyroscopic flow meters. *ASME 77-WA/FM-4*, 1977. Also *Austral. Process Eng.*, 6(9):47–51, 1978; and *Mechanical Eng.*, March 1979.
- (1980) Measuring mass flow using the coriolis principle. *Transducer Technol.* (GB), 2(3): 19–23.
- Plank, N. (1951) Slippage errors in positive displacement liquid meters. *Proc. 3rd World Petroleum Congress*, The Hague, Netherlands, Section IX, pp. 100–24. Leiden, The Netherlands: E. J. Brill.
- Potter, D. M. (1959) *Improvements in or relating to mass flow meters*. British Patent No. 860 657, June 12.
- (1961) British Patent Specification No. 986,831.
- Prahu, S. V., Mascomani, R., Balakrishnan, K., and Konnur, M. S. (1996) Effects of upstream pipe fittings on the performance of orifice and conical flowmeters. *J. Flow Meas. Instrum.*, 7:49–54.
- Priddy, W. J. (1994) Field trials of multiphase metering systems at Prudoe Bay, Alaska. *SPE 69th Annu. Tech. Conf. Exhib.*, New Orleans, LA: 531–43.

- Pursley, W. C. (1986) The calibration of flowmeters. *Meas. Control*, 19(5):37–45.
- Raszillier, H., Allenborn, N., and Durst, F. (1993) Mode mixing in coriolis flowmeters. *Arch. Appl. Mech.*, 63(4–5):219–27.
- (1994) Effect of a concentrated mass on coriolis flowmetering. *Arch. Appl. Mech.*, 64(6):373–82.
- Raszillier, H., and Durst, F. (1991) Coriolis-effect in mass flow metering. *Arch. Appl. Mech.*, 61:192–214.
- Raszillier, H., and Raszillier, V. (1991) Dimensional and symmetry analysis of coriolis mass flowmeters. *J. Flow Meas. Instrum.*, 2:180–84.
- Raustein, O., and Fosse, S. (1991) Measurement of fuel and flare as basis for the CO₂ – Tax. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- Rawes, W., and Sanderson, M. L. (1997) An ultrasonic insertion flowmeter for in-situ calibration. *Ultrasonics in Flow Measurement*, Cranfield University, Bedford, England.
- Reader-Harris, M. J. (1989) Computation of flow through orifice plates. *Numer. Meth. Laminar Turbulent Flow*, 6:1907–17.
- (1998) The equation for the expansibility factor for orifice plates. *FLOMEKO '98 9th Int. Conf. on Flow Measurement*, Lund, Sweden: 209–14.
- Reader-Harris, M. J., and Keegans, W. (1986) Comparison of computation and LDV measurement of flow through orifice and perforated plates, and computation of the effect of rough pipework on orifice plates. *Proc. Int. Symp. on Fluid Flow Measurement*, Washington, DC.
- Reader-Harris, M. J., and Sattary, J. A. (1990) The orifice plate discharge coefficient equation. *J. Flow Meas. Instrum.*, 1:67–76.
- (1996) *The orifice plate discharge coefficient equation – The equation for ISO 5167-1*. East Kilbride: National Engineering Laboratory, Flow Measurement Memo FL/462, September 1996 (Equation 11).
- Reader-Harris, M. J., Sattary, J. A., and Spearman, E. P. (1995) The orifice plate discharge coefficient equation – Further work. *J. Flow Meas. Instrum.*, 6:101–14.
- Reid, J., and Pursley, W. C. (1986). An on-line prover for the calibration of flowmeters in high pressure gas. *Int. Conf. on Flow Measurement in the Mid-80s*: Paper 8.3.
- Reimann, J., John, H., and Muller, U. (1982) Measurement of two-phase flowrate: A comparison of different techniques. *Int. J. Multiphase Flow*, 8:33–46.
- Reitz, W. C. (1979) Positive displacement meters maintenance. *Proc ISA Conf. Exhib., Advances in Instrumentation*. 34(2), (Chicago, 22–25 October 1979), Pittsburgh, PA: 259–61.
- Reynolds, L. (1994) Water industry specifications. *Meas. Control*, 27(7):214–15.
- Rezende, V. A., and Apple, C. (1997) Coriolis meter for LPG custody transfer at Petrobras. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 30.
- Rice, J. A. (1988) *Mathematical Statistics and Data Analysis*. Pacific Grove, CA: Wadsworth & Brooks/Cole Advanced Books & Software.
- Ricken, M. (1989) The swirlmeter – An universal flow measuring instrument. *FLOMEKO '89 Proc. 5th Int. Conf. on Flow Measurement*, Dusseldorf, Germany: 295–303.
- Rieder, A., and Drahm, W. (1996) A new type of single straight tube Coriolis mass flowmeter. *FLOMEKO '96 Proc. 8th Int. Conf. Flow Measurement*, Beijing, China: 250–4.
- Rivetti, A., and Martini, G. (1990) Helium flowrate transducers for space applications. *Cryogenics*, 30(10):835–9.
- Rivetti, A., Martini, G., Goria, R., and Lorefice, S. (1987) Turbine flowmeter for liquid helium with the rotor magnetically levitated. *Cryogenics*, 27:8–11.
- Rivetti, A., Martini, G., Goria, R., Cignolo, G., Capelli, A., and Alasia, F. (1989) Oil, kerosene, and water flowmeter calibration: The integrated IMGC gravimetric/volumetric primary facility. *FLOMEKO '89 Proc. 5th Int. Conf. Flow Measurement*, Dusseldorf, Germany.
- Rivetti, A., Martini, G., and Birello, G. (1994) LHe venturi flowmeters: Practical design criteria and calibration method. *Cryogenics*, 34 Suppl.:449–52.
- Rivkin, I. Y. (1978) *Method and apparatus for measuring mass flow rate of individual components of two-phase gas-liquid medium*. U.S. Patent 4,096,745, June.
- Roach, G. J., Watt, J. S., Zastowny, H. W., Hartley, P. E., and Ellis, W. K. (1995) Field trial of a gamma-ray multiphase flowmeter on Thevenard Island. *Nucl. Geophys.*, 9:1–17.
- Robinson, A. C., and Saffman, P. G. (1982) Three-dimensional stability of vortex arrays. *J. Fluid Mech.*, 125:411–27.

- Robinson, C. (1986) Obstructionless flowmeters: Smooth sailing for some, rough passage for others. *InTech*, 33(12):33–6.
- Robinson, D. I., Sinclair, C. S., Tallentire, A., and Turner, J. T. (1986) A critical appraisal of techniques for the determination of extremely low gas flow rates. *Int. Conf. on Flow Measurement in the Mid 80's*, East Kilbride, Scotland: National Engineering Laboratory: Paper 10.3.
- Rooney, D. H. (1973) Steam flow through orifices. *Report of a meeting at NEL on two-phase flow through orifices and nozzles*, Report No. 549:1–17.
- Rose, C., and Vass, G. (1995) New developments in flow measurement technology provide solutions to difficult process applications. *ISA Advances in Instrumentation and Control: Int. Conf. Exhib.*, 50(3):791–809.
- Roshko, A. (1954) *On the development of turbulent wakes from vortex streets*, NACA Report 1191.
- Rowland, J. E. (1989) Acceleration torque devices. *Cranfield Short Course Lecture*, Cranfield Institute of Technology, England.
- Saito, S., Hashimoto, M., and Wada, T. (1993) Development of Karman vortex flowmeter by use of laser diode: Measurement of flowrate in a pipe with small cross section. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 335–40.
- Sakai, K., Okabayashi, M., and Yasuda, K. (1989) The fluidic flowmeter – A gas flowmeter based on fluidic dynamic oscillation. *J. Flow Meas. Instrum.*, 1:44–50.
- Sakariassen, R. (1997) On-line quality control of ultrasonic gas flow meters. *North Sea Flow Measurement Workshop* Kristiansand, Norway, Paper 15.
- Salami, L. A. (1971) Errors in the velocity-area method of measuring asymmetric flows in circular pipes. *Proc. Int. Conf. on Modern Developments in Flow Measurement*, England: Harwell (published in 1972 by Peter Peregrinus Ltd.).
- (1984) Effect of upstream velocity profile and integral flow straighteners on turbine flowmeters. *Int. J. Heat Fluid Flow*, 5:155–65.
- (1985) Analysis of swirl, viscosity and temperature effects on turbine flowmeters. *Trans. Inst. Meas. Control*, 7:183–202.
- Sanderson, M. L. (1988) Signal processing, telemetry and display and recording. *Instrumentation Reference Book* (B. E. Noltingk, Ed.), Part 4, Chapters 4–6, London: Butterworths.
- (1994) Domestic water metering technology. *J. Flow Meas. Instrum.*, 5:107–13.
- Sanderson, M. L., and Hemp, J. (1981) Ultrasonic flowmeters – A review of the state of the art. *Int. Conf. on Advances in Flow Measurement Techniques*, Coventry, England, Paper G1: 157–78.
- Sanderson, M. L. and Torley, B. (1985) Error assessment for an intelligent clamp-on transit time ultrasonic flowmeter. *Int. Conf. Flow Measurement in the Mid 80's*, Glasgow, Scotland: NEL, Paper 11.3.
- Sato, S., Nukui, K., Ito, S. and Kimura, Y. (1996) Numerical analysis of fluidic oscillation applied to the fluidic gas meter. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 138–43.
- Sattary, J. A. (1991) EEC orifice plate programme – Installation effects. *J. Flow Meas. Instrum.*, 2:21–33.
- Sattary, J. A., and Reader-Harris, J. (1997) Computation of flow through venturi meters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 26.
- Scanes, E. P. (1974) A domestic oil flowmeter. *Kent Tech. Rev.*, 11:31–3.
- Schoenborn, E. M., Jr., and Colburn, A. P. (1939) The flow mechanism and performance of the rotameter. *Trans. Amer. Inst. Chem. Engr.*, 35:359–89.
- Scott, C. (1984) Sounding out ultrasonic flowmeters. *Control Instrum.*, August: 27, 29.
- Scott, R. W. W. (1975a) The use and maintenance of weighing machines in high accuracy liquid flow calibration systems. *Conf. on Fluid Flow Measurement in the Mid 1970's*, East Kilbride, Scotland: National Engineering Laboratory, Paper B-1.
- (1975b) A practical assessment of the performance of electromagnetic flowmeters. *Conf. on Fluid Flow Measurement in the Mid 1970's*, East Kilbride, Scotland: NEL, Paper E1.
- (1982) Liquid flow measurement – A general appraisal. *Developments in Flow Measurement – 1*, (R. W. W. Scott, Ed.), London: Applied Science Publishers, pp. 73–100.
- Selvikvag, O. (1997) The Norwegian regulations relating to fiscal measurements of oil and gas – 1997 update. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 8.
- Shafer, M. R. (1962) Performance characteristics of turbine flowmeters. *J. Basic Eng.*, December: 471–85.

- Shakouchi, T. (1989) New fluidic oscillator, flowmeter, without control port and feedback loop. *J. Dynamic Systems, Trans. ASME*, 111(3):535–9.
- Shamp, F. F. (1971) Accuracy in cryogenic liquid measurements. *Trans. ISA*, 10:219–23.
- Sheikholeslami, M. Z., Patel, B. R., and Kothari, K. (1988) Numerical modelling of turbulent flow through orifice meters – A parametric study. *2nd Int. Conf. on Flow Measurement*, London.
- Sheppard, T. J. (1994) Solid state gas metering: The future. *J. Flow Meas. Instrum.*, 5:103–6.
- Shercliff, J. A. (1955) Experiments on the dependence of sensitivity on velocity profile in electromagnetic flowmeters. *J. Sci. Instrum.*, 32:441–2.
- (1962) *The Theory of Electromagnetic Flow-Measurement*. Cambridge: Cambridge University Press.
- Shufang, H., Yongtao, H., and Ligan, X. (1996) The practice on DN 1400 MM venturi tubes. *Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 37–42.
- Shuoping, Z., Zhijie, X., and Baofen, Z. (1996) Flow with differential pressure noise of orifice. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 605–11.
- Sidney, J. K., King, N. W., and Coulthard, J. (1988a) Cross-correlation flow measurements in oil-air mixtures. *2nd Int. Conf. on Flow Measurement*, BHRA, London.
- (1988b) The measurement of individual phase-flowrates using an ultrasonic cross-correlation flowmeter in air-kerosene mixtures. *Flucom '88*, H. S. Stephens & Associates, Sheffield, September.
- Siegwarth, J. D. (1989) Vortex shedding flowmeters for high velocity liquids. *Int. J. Heat Fluid Flow*, 10(3):232–44.
- Simpson, R. J. (1984) Flexible orifice plates. *Meas. Control*, 17:249–52.
- Sims, P. (1992) Mass flowmeter technology benefits blending. *Process Control Eng.*, 45(5):32–3.
- Sindt, Ch. F., Brennan, J. A., McFaddin, S. E. and Wilson, R. W. (1989) Effect of pipe surface finish on the orifice discharge coefficient. *FLOMEKO '89 Int. Conf. on Flow Measurement*, Dusseldorf, Germany: 49–56.
- Slack, N., Chambers, S., Harland, C., Harrison, A., and Johnston, R. (1995) *Operations Management*, London: Pitman Publishing.
- Sloet, G., and de Nobel, G. (1997) Experiences with ultrasonic meters at the Gasunie export stations. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 14.
- Smith, C. R., Greco, J. J., and Hopper, P. B. (1989) Low-loss conditioner for flow distortion/swirl using passive vortex generation devices. *FLOMEKO 5th Int. Flow Measurement Conf.*, Dusseldorf, Germany: 57–64.
- Smith, J. E. (1978) Gyroscopic/Coriolis mass flow meter. *Canad. Controls Instruments*, 117:29–31 (follows Willer's 1978 note).
- Smith, J. E., and Cage, D. R. (1985) *Parallel path coriolis mass flow rate meter*. U.S. Patent 4,491,025, January 1.
- Smith, M., and Morfey, C. (1997) The effect of developing flow on the accuracy of an ultrasonic gas meter. *Ultrasonics in Flow Measurement*, Cranfield University, Bedford, England.
- Smith, R. V., and Leang, J. T. (1975) Evaluations of correlations for two-phase flowmeters, three current – One new. *ASME Paper No 74-WA/FM-5*, 1974, and *ASME J. Eng. Power*, 1975.
- Smith, R. E., and Matz, R. J. (1962) A theoretical method of determining discharge coefficients for venturis operating at critical flow conditions. *J. Basic Eng.*, 84:434–46.
- Smorgrav, A. E. (1990) Multiphase flow meter KO 300 MFM. *North Sea Flow Measurement Workshop*. East Kilbride, Scotland: National Engineering Laboratory.
- Sparks, C. R., Durke, R. D., and McKee, R. J. (1989) Pulsation-induced errors in the primary and secondary systems of orifice meters. *FLOMEKO '89 Int. Conf. on Flow Measurement*, Dusseldorf, Germany: 31–8.
- Spearman, E. P., Sattary, J. A., and Reader-Harris, M. J. (1991) A study of flow through a perforated-plate/orifice-meter package in two different pipe configurations using laser Doppler velocimetry. *J. Flow Meas. Instrum.*, 2:83–8.
- Spencer, E. A. (1993) *Bibliography of the EEC orifice plate project*. Report EUR 14885 EN, Commission of the European Communities, Brussels, Belgium.
- Spencer, E. A., Heitor, M. V., and Castro, I. P. (1995) Intercomparison of measurements and computations of flow through a contraction and a diffuser. *J. Flow Meas. Instrum.*, 6:3–14.
- Spink, L. K. (1978) *Principles and Practice of Flow Meter Engineering*, 9th ed., Foxboro, MA: The Foxboro Company.

- Spitzer, D. W. (Ed.) (1991) *Flow Measurement*. Research Triangle Park, NC: Instrument Society of America.
- Spragg, W. T., and Seatonberry, B. W. (1975) A radioisotope dilution method for the precise absolute determination of the flowrate of gas under industrial conditions. *Conf. on Fluid Flow Measurement in the Mid 1970's*, East Kilbride Scotland: National Engineering Laboratory, Paper D-1.
- Sproston, J. L., El-Wahed, A., and Johnson, M. W. (1990) An electrostatic vortex-shedding meter. *J. Flow Meas. Instrum.*, 1:183–90.
- Sproston, J. L., Johnson, M. W., and Pursley, W. C. (1987) *Mass flow measurement*, FLOMIC Report.
- Squirrel, B. (1994) Profibus: A working standard fieldbus. *Meas. Control*, 27(1):9–14.
- Stansfeld, J., Atkinson, I., and Washington, G. (1988) A new mass flow meter and its application to crude oil metering. *North Sea Metering Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- Starling, K. E. (1994) Hydrocarbon mixture equations of state and their impact on gas flow measurement. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- Stobie, G. J. (1993) Metering in the real world. *North Sea Flow Measurement Workshop*, Bergen, Norway.
- Stoll, H. W. (1978) Current trends in flow measurement technology. *Proc. ISA Pacific Northwest Instrum. '78 Symp.*, Portland OR: 89–91.
- Stolz, J. (1978) A universal equation for the calculation of discharge coefficients of orifice plates. *Flow Measurement of Fluids* (H. H. Dijkstra and E. A. Spencer, Eds.), Amsterdam: North Holland, pp. 519–34.
- (1988) The first revision of ISO 5167. *North Sea Metering Workshop*, East Kilbride, Scotland: NEL, Paper 3.1.
- Stone, C. R., and Wright, S. D. (1994) Non-linear and unsteady flow analysis of flow in a viscous flowmeter. *Trans. Inst. MC*, 16:128–41.
- Strawn, C. (1991b) Mass meters for liquid measurement: Methods of proving coriolis mass flowmeters. *Proc. Int. School of Hydrocarbon Measurement*, University of Oklahoma, Continuing Engineering Education, Norman, OK: 148–50.
- Strunck, V., Muller, H. Krey, E.-A., and Dopheide, D. (1996) Gas flow metering with new laser doppler sensors monitoring centre line velocity and boundary layer. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 534–7.
- Sullivan, D. A. (1979) Historical review of real-fluid isentropic models. *ASME Winter Annu. Meet.*, New York: Paper 79-WA-FM-1.
- Sullivan, J. J. (no date) *A guide to the selection of MKS flow controllers and control valves for semiconductor processing*. MKS Instruments Inc., Burlington, MA 01803.
- Sullivan, J. J., Ewing, J. H., and Jacobs, R. P. (no date) *Calibration techniques for thermal-mass flowmeters*. (possibly published in *Solid State Technol.*) MKS Instruments Inc., Burlington, MA 01803.
- Sultan, G. (1992) Single straight tube coriolis mass flowmeter. *J. Flow Meas. Instrum.*, 3:241–6.
- Sultan, G., and Hemp, J. (1989) Modelling of the coriolis mass flowmeter. *J. Sound Vibration*, 132:473–89.
- Summers-Smith, J. D. (1994) *An Introductory Guide to Industrial Tribology*. London: Mechanical Engineering Press.
- Sun, Y., Xiong, H., Zhu, R., and Bi, L. (1996) Research and development on wear-resistant orifice plates. *Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 27–31.
- Suzuki, N., Nakabori, H., and Kitajima, A. (1975) New applications of ultrasonic flowmeters. *Flowmeasurement in the Mid '70s*, Glasgow, Scotland: NEL, Paper H-3.
- Suzuki, N., Nakabori, H., and Yamamoto, M. (1972) Ultrasonic method of flow measurement in large conduits and open channels. *Modern Developments in Flow Measurement*, Peter Peregrinus, pp. 115–38.
- Swanson, K. (1988) New developments in the measurement of slurries and emulsions using Coriolis effect mass flowmeters. *Proc. Pacific Cascade Instrumentation '88 Symp.*, ISA.
- Szebeszcyk, J. M. (1994) Application of clamp-on ultrasonic flowmeter for industrial flow measurements. *J. Flow Meas. Instrum.*, 5:127–31.
- Taha, S. M. R. (1994) Digital measurement of the mass-flow rate. *Sensors & Actuators, A: Phys.*, 45(2):139–43.

- Tai, S. W., Miao, J. J., Shaw, J. H., and Chen, Z. L. (1993) Signal-quality study of ring-type vortex flowmeters. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 320–6.
- Takahashi, S., and Itoh, I. (1993) Intelligent vortex flowmeter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 313–19.
- Takamoto, M. (1996) New flowmeter technology for the next century. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 7–12.
- Takamoto, M., and Komiya, K. (1981) Application of a ring to a bluff body of a vortex shedding flowmeter. *Trans. Soc. Instrum. Control Eng. Jap.*, 17:506–10 (in Japanese).
- Takamoto, M., and Terao, Y. (1994) Development of a standard vortex shedding flowmeter. *FLOMEKO '94 Flow Measurement in the Mid 90's*, Glasgow, Scotland: NEL.
- Takamoto, M., Ishiashi, M., Watanabe, N., Aschenbrenner, A., and Caldwell, S. (1993a) Intercomparison tests of gas flowrate standards. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 75–84.
- Takamoto, M., Utsumi, H., Watanabe, N., and Terao, Y. (1993b) Installation effects on vortex shedding flowmeters. *J. Flow Meas. Instrum.*, 4:277–85.
- Tan, P. A. K. (1973) *Theoretical and experimental studies of turbine flowmeter*. PhD Thesis, University of Southampton.
- (1976) Effect of upstream disturbances and velocity profiles on turbine meter performance. *I Mech E Conference Paper C77*.
- Tan, P. A. K., and Hutton, S. P. (1971) Experimental, analytical and tip clearance loss studies in turbine-type flowmeters. *Proc. Int. Conf. on Flow Measurements*, Harwell PPL Conference Publication 10:321–46.
- Tarabad, M., and Baker, R. C. (1982) Integrating electromagnetic flowmeter for high magnetic Reynolds numbers. *J. Phys. D: Appl. Phys.*, 15:739–45.
- Taylor, J. W., and Cassidy, H. P. (1994) Acoustic flowmeter comparison tests at BC Hydro, Canada. *FLOMEKO '94 Flow Measurement in the Mid-90s*, Glasgow, Scotland: NEL, Paper 7.4.
- Taylor, R. D. H., and Nuttall, R. C. H. (1993) On-line well monitoring and its application in a South Oman oil field. *Proc. Middle East Oil Show, AIME*, 2:229–34.
- Terao, Y., Choi, H. M., Edra, R. B., and Chen, Z. L. (1993) An experimental study on flow structure in vortex flowmeters. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 327–34.
- Teufel, M., Trimis, D., Lohmuller, A., Takeda, Y., and Durst, F. (1992) Determination of velocity profiles in oscillating pipe-flows by using laser Doppler velocimetry and ultrasonic measuring devices. *J. Flow Meas. Instrum.*, 3:95–102.
- Thomas, N. H., Auton, T. R., Sene, K., and Hunt, J. C. R. (1983) Entrapment and transport of bubbles by transient large eddies in multiphase turbulent shear flow. *Int. Conf. on Physical Modelling of Multi-Phase Flow*, Coventry, England, BHRA Fluid Engineering, Cranfield, Paper E1:169–84.
- Thompson, E. J. (1978) *Two beam ultrasonic flow measurement*. Ph. D. Thesis, University of London.
- Thompson, R. E., and Grey, J. (1970) Turbine flowmeter performance model. *J. Basic Eng., Trans. ASME*, 712–23.
- Ting, V. C., and Shen, J. J. S. (1989) Field calibration of orifice meters for natural gas flow. *J. Energy Resources Technol., Trans. ASME*, 111(1):22–33.
- Tobi, N. V. (1953) British Patent No. 726 271, May 27.
- Toral, H., Beg, N., and Archer, J. S. (1990) Multiphase flow metering by software. *Int. Conf. on Basic Principles and Industrial Applications of Multiphase Flow*, London: IBC Technical Services Ltd, April.
- Torkildsen, B. H., and Hanssen, B. V. (1996) Practical considerations related to multiphase metering of a well stream. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- Torkildsen, B. H., Helmers, P. B., and Kanstad, S. K. (1997) Topside and subsea experience with the Framo multiphase flow meters. *North Sea Flow Measurement Workshop*, Kristiansand, Norway: Paper 3.
- Trigas, A., and Hope, S. H. (1991) A comparison of the accuracy and process control capability of turbine and coriolis flowmeters. *Int. Conf. on Flow Measurement in Industry and Science*, London.
- Tritton, D. J. (1988) *Physical Fluid Dynamics*, 2nd ed. Oxford: Oxford University Press.

- Tsuchida, T., Terashima, Y., and Machiyama, T. (1982) The effects of flow velocity profile on the electromagnetic flowmeters. *Rept. of Researches, Nippon Institute of Technology*, 101–11.
- Tsuchiya, K. et al. (1970) Karman vortex flow meter. *Bull. Jap. Soc. Mech. Eng.*, 13:573–8.
- Tsukamoto, H., and Hutton, S. P. (1985) Theoretical prediction of meter factor for a helical turbine flowmeter. *Conf. Fluid Control and Measurement*, Tokyo, Japan.
- Tsutsui, H., and Yamikawa, Y. (1993) Coriolis force mass-flowmeter composed of a straight pipe and an additional resonance vibrator. *Jap. J. Appl. Phys. Part 1*, 32(58):2369–71.
- Tucker, H. G., and Hayes, W. F. (1982) Error analysis of a vibrating pendulum two phase flowmeter for oil well application. *Measurement in Polyphase Flows – 1982, AIAA/ASME Joint Fluids, Plasma, Thermophysics and Heat Transfer Conf.*, St Louis, MO: 45–53.
- Tullis, P., and Smith, J. (1979) Coriolis flowmeter. *NEL Fluid Mechanics Silver Jubilee Conf.*, East Kilbride, Glasgow: NEL, Paper 6.3.
- Turner, D. (1971) A differential pressure flowmeter with linear response. *Modern Developments in Flow Measurement*, Paper 4.3: 191–9.
- Turner, J., Wynne, R., and Hurren, P. (1989) *Computation and techniques in flow measurement and their applications to flowmeter diagnostics*. FLOMIC Report No. 6.
- Turner, J. T., Popiel, C. O., and Robinson, D. I. (1993) Evolution of an improved vortex generator. *J. Flow Meas. Instrum.*, 4:249–58.
- Tuss, B. (1996) Production evaluation and testing of a high viscosity and high gas volume fraction multiphase meter. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- Upp, E. L. (1993) *Fluid Flow Measurement*. Houston: Gulf Publishing.
- Urner, G. (1997) Pressure loss of orifice plates according to ISO 5167-1. *J. Flow Meas. Instrum.*, 8:39–41.
- Valenti, M. (1995) Detecting leaks to reduce energy costs. *Mech. Eng.*, 117(7):98–100.
- van Bloemendaal, K., and van der Kam, P. M. A. (1994) Installation effects on multi-path ultrasonic flow meters: The 'Ultraflow' project. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- van Dellen, K. (1991) Ultrasonic gas flow meters continue their rise. *Proc. North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers.
- van der Bent, H. (1993) Metering mass flow. *Process Eng.*, 43–4.
- van der Grinten, J. G. M. (1990) *Error curves of turbine gas meters*. Netherlands Metrology Institute.
- (1994) A comparison of the methods for uncertainty analysis based on ISO 5168 and the Guide prepared by ISO/TAG4/WG3. *FLOMEKO '94 Flow Measurement in the Mid 90's*, East Kilbride, Scotland: NEL.
- (1997) Recent developments in the uncertainty analysis of flow measurement processes. *North Sea Flow Measurement Workshop*, Kristiansand, Norway, Paper 11.
- van der Kam, P. M. A., and Dam, A. M. (1993) Large turbine meters for custody transfer measurement: the renovation of the Gasunie export stations. *J. Flow Meas. Instrum.*, 4:91–8.
- van der Kam, P. M. A., and de Jong, S. (1994) Gas turbine meters: standardization and operational experiences. *J. Flow Meas. Instrum.*, 5:121–6.
- van der Kam, P. M. A., and van Dellen, K. (1991) The effect of double bends out of plane on turbine meters. *J. Flow Meas. Instrum.*, 2:61–8.
- van der Kam, P. M. A., Dam, M. A., and van Dellen, K. (1990) Gasunie selects turbine meters for renovated export metering stations. *Oil & Gas J.*, 88:39–44.
- van der Wiel, J., Linder, C., and de Rooij, N. F. (1993) Liquid velocity sensor based on the hot-wire principle. *Sensors & Actuators A*, 37–8:693–7.
- van Dijk, G. J. A., and Huijsing, J. H. (1995) Bridge-output-to-frequency converter for smart thermal air-flow sensors. *IEEE Trans. Instrum. Meas.*, 44:881–6.
- van Oudheusen, B. W. (1991) The thermal modelling of a flow sensor based on differential convective heat transfer. *Sensors & Actuators*, 29:93–106.
- (1992) Silicon thermal flow sensors. *Sensors & Actuators A: Phys.*, 30(1–2):5–26.
- van Santen, H., Kolar, Z. I., and Scheers, A. M. (1995) Photon energy selection for dual energy γ - and/or X-ray absorption composition measurements in oil-water-gas mixtures. *Nucl. Geophys.*, 9(3):193–202.
- Vass, G. E. (1996) Users benefit from newer electronics in today's smarter magnetic flowmeters. *Advances in Instrumentation and Control: Int. Conf. Exhib.*, (ISA) 51(1):641–56.

- Vaterlaus, H.-P. (1995) A new intelligent ultrasonic flowmeter for closed conduits and open channels. *ASCE Waterpower – Proc. Int. Conf. on Hydropower*, 2:999–1008.
- VDI/VDE (1978) Variable area flowmeters – Accuracy. *VDI/VDE 3513*, p. 2 (in German).
- Vetter, G., and Notzon, S. (1994) Effect of pulsating flow on coriolis mass flowmeters. *J. Flow Meas. Instrum.*, 5:263–73.
- Vogtlin, B., and Tschabold, P. (no date) *Direct measurement of mass flow using the coriolis force*. E&H Flowtec publication.
- Vsesojuzny Nauchno-Issledovatelsky (1976) *Method of and apparatus for measuring the mass flowrate of individual components of a gas-liquid medium*. British Patent 1,528,232.
- Wagner, J. J. (1988) Effects of sensor design and application characteristics on Coriolis mass meter performance: An overview. *2nd Int. Conf. on Flow Measurement*, London.
- Walker, J. T. (1992) Advances in Coriolis technology for precision flow and density measurements of industrial fluids. *Proc. 47th Amer. Symp. on Instrumentation for the Process Industries*, Publ. Texas A&M Univ.: 69–73.
- Wallis, K. F. A. (1975) The long term repeatability of positive displacement liquid flowmeters. *Conf. on Fluid Flow Measurement in the Mid 1970s*, East Kilbride, Scotland: National Engineering Laboratory, Paper B-4.
- Wallis, K. F. A., and James, J. H. P. (1985) Performance of a fuel flow calibration system. *Int. Conf. on the Metering of Petroleum and Its Products*, OYEZ, London.
- Wallis, R. A. (1961) *Axial Flow Fans Design and Practice*. London: George Newnes.
- Wang, H., Priestman, G. H., Beck, S. B. M., and Boucher, R. F. (1996) Development of fluidic flowmeters for monitoring crude oil production. *J. Flow Meas. Instrum.*, 7:91–8.
- Washington, G. (1989) Measuring the flow of wet gas. *North Sea Flow Metering Workshop*, Hauge-sund, Norway.
- Watson, G. A., and Furness, R. A. (1977) Development and application of the turbine meter. *Proc. Transducer 77 Conf. Flow Measurement Session*, Wembley, London.
- Watson, G. G., Vaughan, V. E., and McFarlane, M. W. (1967) *Two-phase pressure drop with a sharp-edged orifice*. NEL Report No. 290, East Kilbride, Glasgow.
- Watt, J. S. (1993) Platform trial of a multiphase flow meter. *North Sea Flow Measurement Workshop*, Bergen, Norway.
- Watt, R. M. (1990) Computational modelling of coriolis mass flowmeters. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: National Engineering Laboratory.
- (1991) Modelling of coriolis mass flowmeters using ANSYS. *ANSYS Users Conf.*, Pittsburgh.
- Weager, B. (1993/4) NAMAS-approved flowmetering. *Meas. Control*, 26(10):298–301.
- Weberg, T. (1990) A new critical mass flow correlation for natural gas. *J. Flow Meas. Instrum.*, 1:209–15.
- Weigand, J. (1994) Gas flow measurement using laminar flow elements. *ASHRAE Trans.*, 100:973–9.
- Weinig, F. (1932) Stromung durch Profiligitter und einige Anwendungen auf die Stromung in Propellern. *Hydromechanische Probleme des Schiffsantriebs*: 171.
- Wemyss, W. A., and Wemyss, A. C. (1975) Development of the Hoverflo: A turbine flowmeter without bearings. *NEL Conf. Fluid Flow Measurement in the Mid 1970's*, Vol. 2, Paper H1.
- Wen Dong-Xu (1990) Smart fiber optics flowmeter. *Proc. SPIE, Int. Soc. Optical Eng.*, 1230:557–8.
- Wenran, W., and Yunxian, T. (1995) A new method of two-phase flow measurement by orifice plate differential pressure noise. *J. Flow Meas. Instrum.*, 6:265–70.
- Whitaker, T. S. (1993) A review of multiphase flowmeters and future development potential. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea: 628–34.
- (1996) Assessment of multiphase flowmeter performance. *North Sea Flow Measurement Workshop*, Peebles, Scotland: NEL.
- Whitaker, T. S., and Millington, B. C. (1993) Review of multiphase flowmeter projects. *North Sea Flow Measurement Workshop*, Norwegian Society of Chartered Engineers, Bergen, Norway.
- Whitaker, T., and Owen, I. (1990) Experience with two designs of differential pressure flowmeters in two-phase flow. *Int. Conf. on Basic Principles and Industrial Applications of Multiphase Flow*, April 1990 (IBC Technical Services, London).
- White, D. F., Rodely, A. E., and McMurtie, C. L. (1974) The vortex shedding flowmeter. *Flow, Its Measurement and Control in Science and Industry*, Pittsburgh, Pa, Instrument Society of America, 1(2):967–74.

- Widmer, A. E., Fehlmann, R., and Rehwald, W. (1982) A calibration system for calorimetric mass flow devices. *J. Phys. E: Sci. Instrum.*, 15:213–20.
- Willer, M. D. (1978) Gyroscopic principle key to mass flowmeter. *Canad. Controls Instrum.*, 117 (1):28.
- Williams, E. J. (1930) The induction of emfs in a moving fluid by a magnetic field and its application to an investigation of the flow of liquids. *Proc. Roy. Soc. London*, 42:466–78.
- Williams, T. J. (1970) Behaviour of the secondary devices in pulsating flow measurement. *Instrumentation Measurement, and Control Symp. the Measurement of Pulsating Flow*: 56–61.
- Wislicenus, G. F. (1947) *Fluid Mechanics of Turbomachinery*. New York: McGraw-Hill. (subsequent ed. New York: Dover).
- Withers, V. R., Inkley, F. A., and Chesters, D. A. (1971) Flow characteristics of turbine flowmeters. *Proc. Int. Conf. on Flow Measurements*, Harwell PPL Conference Publication 10:305–20.
- Withers, V. R., Strang, W., and Allnutt, G. (1996) Practical application of Coriolis meters for offshore tanker loading from the Harding Field. *North Sea Flow Measurement Workshop*, East Kilbride, Scotland: NEL.
- Witlin, W. G. (1979) Theory, design and application of vortex shedding flowmeters. *Proc. Symp. on Measurement Technology for the 80s: Analytical Instrumentation*, 17:120–5.
- Wojtkowiak, J., Kim, W. N., and Hyun, J. M. (1997) Computations of the flow characteristics of a rotating-piston-type flowmeter. *J. Flow Meas. Instrum.*, 8:17–25.
- Wong, H. A., Rhodes, E., and Scott, D. S. (1981) Flow metering in horizontal, adiabatic, two-phase flow. *Proc. 2nd Symp. on Flow: Its Measurement and Control in Science and Industry* (W. W. Durgin, Ed.), ISA, St Louis, Vol 2:505–16.
- Wood, G. (1994) Introduction. *Meas. Control*, 27(2):37.
- Wright, P. H. (1993) The application of sonic (critical flow) nozzles in the gas industry. *J. Flow Meas. Instrum.*, 4:67–72.
- Wu, G., and Yan, S. (1996) The calculation of the discharge coefficient of critical venturi nozzles using the finite element method. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 611–18.
- Wyatt, D. G. (1961) Problems in the measurement of blood flow by magnetic induction. *Phys. Med. Biol.*, 5:289–320.
- (1977) Theory, design and use of electromagnetic flowmeters. *Cardiovascular Flow Dynamics and Measurements* (N. H. C. Hwang and N. A. Normann, Eds.), University Park Press, Chapter 2, pp. 89–149.
- (1982) Blood flow and blood velocity measurement in vivo by electromagnetic induction. *Trans. Inst. M. C.*, 4:61–78.
- Xiaozhang, Z. (1995) New multi-meter system for flow measurement of water-oil-gas mixture. *ISA Advances in Instrumentation and Control: Int. Conf. Exhib.*, 50(1):113–20.
- Xingbin, L., Hetang, Q., Hongtao, L. and Zhihui, Y. (1996) Conductance cross-correlation flowmeter for measurement flowrate in an oil/water two phase flow. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 301–8.
- Xu, L. A., Yang, H. L., Zhang, T., Chen, W., Li, J., and Ran, Z. M. (1994) A clamp-on ultrasound cross-correlation flowmeter for liquid/solid two-phase flow measurement. *J. Flow Meas. Instrum.*, 5:203–8.
- Xu, Y. (1992a) Calculation of the flow around turbine flowmeter blades. *J. Flow Meas. Instrum.*, 3:25–35.
- (1992b) A model for the prediction of turbine flowmeter performance. *J. Flow Meas. Instrum.*, 3:37–43.
- Xu-bin, Q. (1993) The simple economic elbow meter for flow measurement. *Meas. Control*, 26:245–6.
- Yamasaki, H. (1993) Progress in hydrodynamic oscillator type flowmeters. *J. Flow Meas. Instrum.*, 4:241–8.
- Yamashita, Y. (1996) Development of coriolis mass flowmeter with a single straight tube as flow tube. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 265–70.
- Yang, W. Q. (1996) Key features of a newly-designed capacitance tomography system. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 480–5.
- Yang, W. Q., and Beck, M. S. (1997) An intelligent cross correlator for pipeline flow velocity measurement. *J. Flow Meas. Instrum.*, 8:77–84.

- Yang, W. Q., Beck, M. S., Lenn, C. P., and Hoyle, B. S. (1996a) A transputer-based cross correlator with intelligent functions. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 289–94.
- (1996b) A combined capacitance-electrodynamic transducer for oil pipeline flow velocity measurement. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 498–503.
- Yeh, T. T., and Mattingly, G. E. (1994) Pipeflow downstream of a reducer and its effects on flowmeters. *J. Flow Meas. Instrum.*, 5:181–7.
- Yeung, H. C., Herne-Smith, R. S., and Sanderson, M. L. (1994) Quality in the UK water industry - Insertion metering. *J. Flow Meas. Instrum.*, 5:133–40.
- Yiming, W., and Haibo, Y. (1996) Capacitance transducers and cross-correlation techniques for two phase flow measurement. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 307–12.
- Yokota, S., Son, W. C., and Kim, D. T. (1996) Unsteady flow measurement by using a drag-plate-type force flowmeter. *FLOMEKO '96 Proc. 8th Int. Conf. on Flow Measurement*, Beijing, China: 661–6.
- Yoo, S. Y., Lee, S. Y., Yoon, K. Y., Park, K. A., and Paik, J. S. (1993) Experimental study on the factors influencing discharge coefficients of sonic nozzles. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, 363–71.
- Yoshida, Y., Amata, Y., and Frugawa, M. (1993) Development of a partially-filled electromagnetic flowmeter. *FLOMEKO '93 Proc. 6th Int. Conf. on Flow Measurement*, Korea Research Institute of Standards and Science: 452–9.
- Youden, W. J. (1959) Graphical diagnosis of interlaboratory test results. *J. Industrial Qual.*, 15:11.
- Young, A. (1990) Coriolis flowmeters for accurate measurement of liquid properties. *Adv. Instrum., ISA Proc.*, 45 (4):1891–8.
- Zanker, K. J., and Cousins, T. (1975) The performance and design of vortex meters. *Conf. on Fluid Flow Measurement in the Mid 1970s*, East Kilbride, Glasgow, Scotland: National Engineering Laboratory, Paper C-3.
- Zanker, K. J., and Freund, W. R., Jr. (1994) Developments of multipath transit time ultrasonic gas flow meters. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- (1996) Practical experience with gas ultrasonic flow meters. *North Sea Flow Measurement Workshop*, Peebles, Scotland.
- Zedan, M. F., and Teyssandier, R. G. (1990) Effect of errors in pressure tap locations on the discharge coefficient of a flange-tapped orifice plate. *J. Flow Meas. Instrum.*, 1:141–8.
- Zhang, X. Z., and Hemp, J. (1994) Measurement of pipe flow by an electromagnetic probe. *ISA Trans.*, 33:181–4.
- (1995) Calculation of the virtual current around an electromagnetic velocity probe using the alternating method of Schwarz. *J. Flow Meas. Instrum.*, 5:146–9.

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FLOWMETER APPLICATION INDEX

In referring to this index the reader should note that:

- a) the applications recorded are those examples which I have encountered in reading the, mainly industrial, literature on meters **and are obviously not in any sense exhaustive**. In a few general cases I have, therefore, included references to other flowmeters;
- b) the reference to a particular flowmeter does not imply that other flowmeters are unsuitable, but rather that I was not aware of manufacturers **specifically** recommending other flowmeters for the application;
- c) even if a particular flowmeter is referred to, it does not necessarily mean that it will, actually, be suitable for the application;
- d) the reader should always check suitability of a flowmeter for a particular application with the manufacturer of the specific flowmeter, in terms of type of service, compatibility of materials, and the other factors set out in Chapter 3.

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