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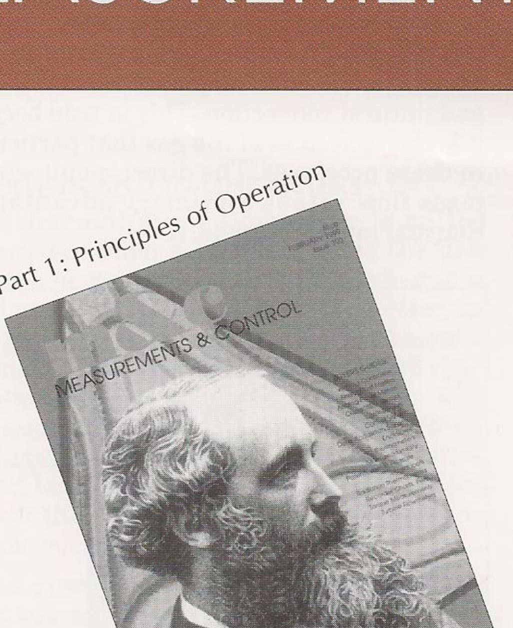
Part 1: Principles of Operation



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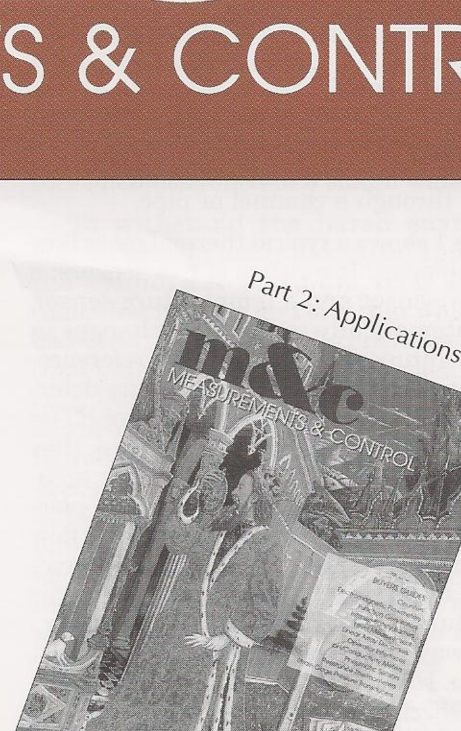
Part 2: Applications



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INDUSTRIAL THERMAL MASS FLOWMETERS

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INDUSTRIAL THERMAL MASS FLOWMETERS

Part 1: Principles of Operation

Improved performance, enhanced user-interfaces and reduced cost-of-ownership have helped thermal mass flowmeters gain acceptance in the industrial process control market for gas flow measurements. Part I provides a comprehensive explanation of the thermal principle of operation. It also compares the performance and cost-reduction attributes of the industrial thermal mass flowmeter to those of other flow monitoring instruments. In the April issue of M&C, Part II will include a description of the various sensor, flow body and electronics configurations available today, and a list of typical applications.

JOHN G. OLIN, Ph.D.

An industrial thermal mass flowmeter measures either the mass velocity at a point in a flowing gas or the total mass flow rate through a channel or pipe.

Figure 1 shows a typical thermal mass flow sensing element. It consists of two sensors: a mass flow sensor and a temperature sensor, which automatically corrects for changes in gas temperature. Both sensors are reference-grade platinum resistance temperature detectors (RTDs). The electrical resistance of RTDs increases as temperature increases; thus, they are the most commonly used sensors for accurate temperature measurements. The electronics pass current through the mass flow sensor, thereby heating it to a constant temperature differential ($T_v - T_a$) above the gas temperature, T_a, and measure the heat, q_c, carried away by the cooler gas as it flows past the sensor. Hence, it is called a "constant temperature thermal anemometer."

Since the heat is carried away by the gas molecules, the heated sensor directly measures gas mass velocity (mass flow rate per unit area), ρU. The mass velocity is typically expressed as U_s in engineering units of normal meters per second, or nm/s, referenced to normal conditions of 0° or 20°C in temperature and at 1 atm. If the gas temperature and pressure are constant, then the instrument's measurement can be expressed as actual meters per second, or m/s. When the mass velocity is multiplied by the cross-sectional area of a flow channel, the mass flow rate through the channel is obtained.

Mass flow rate, rather than volumetric flow rate, is the direct quantity of interest in almost all practical and industrial applications, such as any chemical reaction, combustion, heating, cooling, drying, mixing, fluid power, human respiration, meteorology, and natural convection. This is true because it is the *molecules* of the gas that participate in these processes. The direct monitoring of mass flow rate is a primary advantage of thermal mass flowmeters.

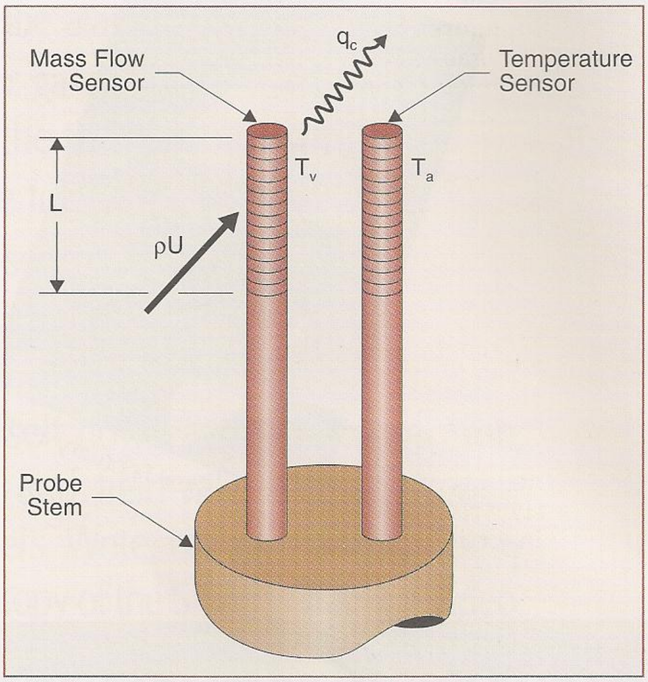

FIGURE 1. The industrial thermal mass flow sensing element consists of two sensors — a mass flow sensor and a temperature sensor. T_v is the temperature of the heated mass flow sensor. T_a is the gas temperature measured by the temperature sensor. ρ is the gas mass density. U is the gas velocity. q_c is the heat convected away by the flowing gas stream. L is the length of the heated end of the sensor.

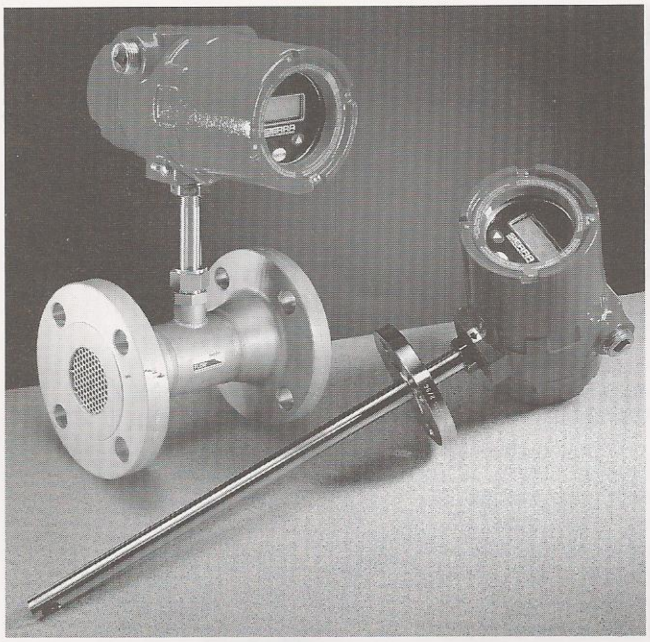

FIGURE 2. Industrial thermal mass flowmeters are available in both insertion and in-line configurations.

INSERTION AND IN-LINE METERS

A thermal mass flowmeter is called *immersible* when it is immersed in the flow stream, or a *capillary-tube* type when it is configured as an in-line mass flowmeter for low gas flows. In use, the thermal mass flowmeter is inserted through a sealed compression fitting or flanged stub in the wall of a duct, pipe, stack, or other flow passage. In this case, it is usually called an *insertion* meter. Insertion meters measure gas velocity over the range of 0.5 to 150nm/s for typical gases.

In another common configuration, the dual-sensor probe is permanently fitted into a pipe or tube (typically 8 to 300mm in diameter) with either threaded or flanged gas connections. This configuration is called an *in-line* thermal mass flowmeter. In-line meters are directly calibrated for the total gas mass flow rate flowing through the pipe. Typical insertion and in-line meters are shown in Figure 2.

THERMAL MASS FLOWMETER APPLICATIONS

Thermal mass flowmeters are seldom used to monitor liquid flows because avoidance of cavitation problems limits the temperature of the mass flow sensor, T_v, to only 10° to 20°C above the liquid temperature. This results in reduced velocity sensitivity and increased dependence on small changes in liquid temperature. Additionally, industrial liquid flows can cause sensor contamination and fouling.

Common applications include: combustion air; preheated air; fuel gas; stack gas; natural gas distribution; food processing; semiconductor manufacturing gas distribution; heating, ventilation, and air conditioning; multi-point flow monitoring in large ducts and stacks; drying; aeration and digester gas; occupational safety and health monitoring; environmental, natural convection, and solar studies; fermentors; and human inhalation monitoring.

SENSOR DESIGN

Figure 3 shows typical sensors used in thermal mass flowmeters. The most common mass flow sensor is a reference-grade platinum wire (approximately 25μm in diameter and 20Ω in resistance) wound around a cylindrical ceramic mandrel, such as alumina. Alternatively, the sensors are (1) thin platinum films deposited on a glass or ceramic substrate, or (2) RTD sensors micromachined in a silicon wafer.

To withstand the harsh environment encountered in many industrial applications, the cylindrical platinum RTD is tightly cemented into the end of a thin walled stainless-steel, Hastelloy, or Inconel tube, which is sometimes referred to as the "sheath." Typically the mass flow sensor is 3mm in outside diameter and 2 to 6cm long. Because the gas temperature usually varies in industrial applications, thermal probes almost always have a separate, but integrally mounted, unheated platinum RTD sensor for measuring the local gas temperature, T_a. When operated in the constant-temperature mass flow mode, the temperature difference (T_v–T_a) is usually in the 30° to 100°C range.

The temperature sensor is constructed just like the mass flow sensor, but has a resistance in the 300 to 1000Ω range. As shown in Figure 1, the dual-sensor probe has the mass flow and temperature sensors mounted side-by-side on a cylindrical probe stem (usually 6 to 25mm in diameter and 0.1 to 3m long). A shield is usually provided to prevent breakage of the sensing head. The spatial resolution of this shielded mass flow sensor is 1 to 2 cm. The meter electronics are mounted either directly on the probe stem or remotely.

RESEARCH APPLICATIONS

Industrial thermal mass flowmeters evolved from hot-wire anemometers which are used to measure the point velocity and/or turbulence of clean gases and liquids in research, product development, and labo-

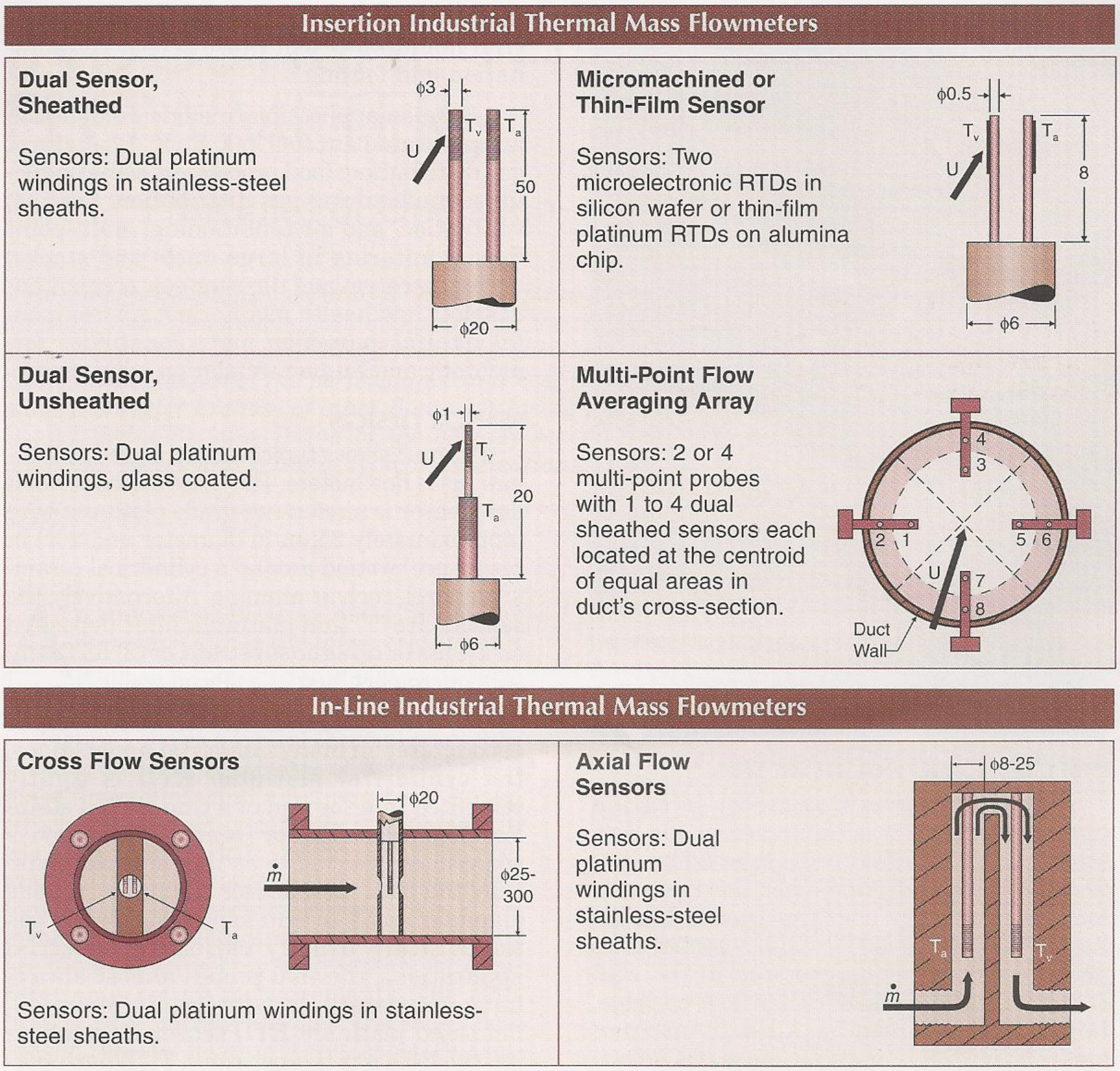

FIGURE 3. Typical sensor configurations of industrial thermal mass flowmeters. All dimensions are in millimeters. T_v indicates the heated velocity sensor. T_a indicates the temperature sensor. $\dot{m}$ is the total mass flow rate.

ratory applications. Because of their fragile nature, anemometers are not used for industrial applications. Typically the gas is ambient air. Constant-temperature, filtered, degasified water is the primary liquid application. The research thermal anemometer's velocity sensor is either a fine tungsten hot wire with a diameter of 4 to 5µm or a thin 0.1µm thick platinum film plated on a cylindrical or wedge-shaped quartz sensor. Hot-film sensors trade-off lower frequency response for increased ruggedness and are used in gas and liquid flows.

Typical applications for hot-wire and hot-film research thermal anemometers include: one-, two-, and three-dimensional flow and turbulence studies; validation of computational fluid dynamics codes; environmental and micrometeorological measurements; turbomachinery; and boundary-layer measurements. Bruun[1] is an excellent reference source for the theory and applications of hot-wire and hot-film anemometers. Another comprehensive source is Fingerson and Freymuth[2]. Freymuth[3] describes the eighty-year history of research thermal anemometers. All of these references are applicable to industrial thermal mass flowmeters. Today, hot-wire and hot-film anemometers are the most widely used instruments for fluid mechanics studies.

COST OF OWNERSHIP

If mass flow rate is the quantity of direct interest, then the thermal meter has major advantages over conventional alternative technologies such as differential pressure producing devices (orifice plates or multipoint pitot tubes) and linear volumetric flow devices (such as turbine, vortex, or ultrasonic flowmeters). To monitor gas mass flow rate, all these devices require additional transducers for the measurement of temperature and absolute pressure, as well as a flow computer to calculate gas mass flow rate from the three measured quantities.

This results in a higher initial cost when compared with the thermal meter, as well as higher installation and maintenance costs, because three separate transmitters plus a flow computer must be purchased, installed, and maintained. Total *cost of ownership* is the sum of the present values of all these cost elements over an instrument's lifetime, and is generally recognized as the best parameter for comparing the true cost of instrumentation.

Thermal mass flowmeters have less fugitive emissions because only two gas connections are required, whereas differential pressure producing devices require five connections, and linear devices four connections. Because the individual errors of all three sensors and the flow computer must be included, the overall accuracy of alternative technologies is typically less than thermal meters. The square root of the sum of the squares of the four individual standard deviations is the preferred method for estimating the overall standard deviation of compensated mass flow measuring devices[4].

Multi-variable transmitters are now available which simultaneously measure the differential pressure across an orifice plate, the absolute pressure, and the temperature, and integrally perform the mass flow rate computation. Although they improve on older approaches, these devices still have both a higher initial cost and cost of ownership than thermal meters and require five gas connections versus two.

PRINCIPLE OF OPERATION

Figure 4 shows the first law of thermodynamics applied to an industrial mass flow sensor, such as shown in Figures 1 to 3. Application of the first law to thermal sensors provides the basis for determining the desired quantities — point velocity, point mass velocity, or total mass flow rate.

Applied to Figure 4, the first law states that the energy into the control volume equals the energy out plus the energy stored. Making the practical simplifying assumptions of steady-state operation (i.e., no energy stored) and no heat transfer via radiation, we get:

$$w = q_c + q_L \tag{1}$$

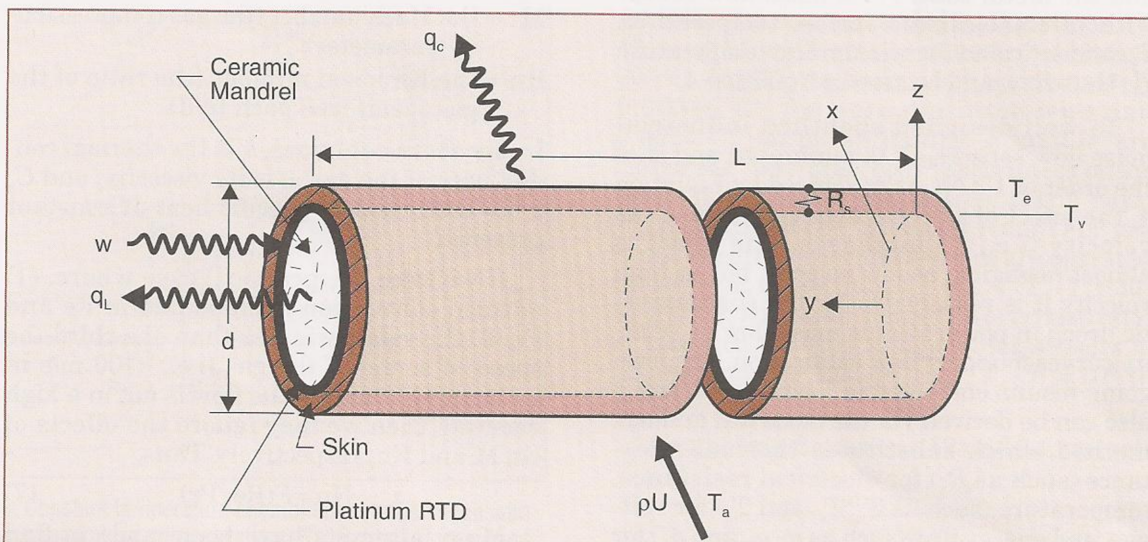

FIGURE 4. First law of thermodynamics applied to a sheathed industrial thermal mass flow sensor: w is the electrical power (watts) supplied to the sensor, q_L is the conductive heat lost, T_e is the average surface temperature of the sensor over its length L, d is the sensor's outside diameter, and R_s is the thermal resistance of the skin (metal tube plus cement layer).

The heat transfer q_c due to natural and forced convection is normally expressed in terms of the heat transfer coefficient, h, as:

$$q_c = hA_v\,(T_e - T_a) \qquad (2)$$

where $A_v = \pi dL$ and is the external surface area of the mass flow sensor. The electrical power, w, is usually expressed as:

$$w = E_v^2 / R_v \qquad (3)$$

where E_v is the voltage across the mass flow sensor, and R_v is its electrical resistance.

For the sensor shown in Figure 4, q_L is the heat conducted from the end of the heated mass flow sensor of length L to the remainder of the sensor's length. Most of this heat is convected away by the flowing gas, and a small fraction is conducted to the probe stem. In well designed velocity sensors, q_L is at most 10% to 15% of w, a fraction which decreases as velocity increases.

Figure 4 shows a mass flow sensor with a sheath, or "skin". In this case, the surface temperature, T_e, is slightly less than the temperature, T_v, of the platinum winding or film because a temperature drop is required to pass the heat, q_c, through the intervening skin — the cement layer and the metal tube. This is expressed as:

$$T_e = T_v - q_c R_s \qquad (4)$$

where R_s is the thermal resistance of the skin in units of K/W (degrees Kelvin per watt). R_s is a constant for a given sensor and is the sum of the thermal resistances of the cement layer and the metal tube. For a mass flow sensor without a sheath, the surface temperature, T_e, is identical to the wire or film temperature T_v, thus R_s would be zero in Equation 4.

In well designed sheathed industrial mass flow sensors, R_s is minimized and is of the order of 1 K/W. As evidenced by Equation 4, the effect of skin resistance increases as velocity (i.e., q_c) increases. The effect is almost negligible at low velocity, but at high velocity it is responsible for the characteristic droop in power versus mass flow or velocity curves found by flow calibration. The foregoing results embodied in Equations 1 and 4 also can be derived via the electrical analogy method which substitutes thermal resistance (such as R_s) for electrical resistance, temperature (such as T_v, T_e, and T_a) for voltage, and energy flux (such as w, q_c, and q_L) for current.

The presence of end conduction means that the temperature of the mass flow sensor varies with the axial coordinate y in Figure 4. The temperature, T_v, actually sensed by the mass flow sensor is the average temperature over length L, or:

$$T_v = (1/L)\int_o^L T_v(y)\,dy \qquad (5)$$

Bruun[1] presents an analytical solution for $T_v(y)$ for finite-length hot-wire sensors with end conduction. The average temperature given by Equation 5 is the correct expression for T_v and is so defined hereafter. Similarly, T_e is the average surface temperature over length L. For constant-temperature thermal mass flowmeters, the ceramic mandrel has a negligible radial temperature variation and therefore has local temperature $T_v(y)$ and average temperature T_v.

Since the flow around cylinders in cross flow is confounded by boundary-layer separation and a turbulent wake, it has defied analytical solution. Therefore, the film coefficient, h, in Equation 2 is found using empirical correlations. Correlations for h are expressed in terms of the following non-dimensional parameters:

$$\mathrm{Nu} = F(\mathrm{Re, Pr, Gr, M, Kn}) \qquad (6)$$

where,

Nu = hd/k, the Nusselt number (the heat-transfer parameter).

Re = ρVd/μ, the Reynolds number (the ratio of dynamic to viscous forces).

Pr = $\mu C_p/k$, the Prandtl number (the gas properties parameter).

Gr = the Grashof number (natural convection parameter).

M = the Mach number (the gas compressibility parameter).

Kn = the Knudsen number (the ratio of the gas mean free path to d).

In the above equations, k is the thermal conductivity of the gas; μ is its viscosity; and C_p is its coefficient of specific heat at constant pressure.

If we take the practical case where, (1) natural convection is embodied in Re and Pr, (2) the velocity is less than one-third the speed of sound of the gas (i.e., <100 m/s in ambient air), and (3) the flow is not in a high vacuum, then we may ignore the effects of Gr, M and Kn, respectively. Thus,

$$\mathrm{Nu} = F(\mathrm{Re, Pr}) \qquad (7)$$

Many attempts have been made to find universal correlations for the heat transfer from cylinders in cross flow. For an isothermal fluid, King[5] expresses Equation 7 as:

$$\mathrm{Nu} = A + B\,\mathrm{Re}^{0.5} \qquad (8)$$

where A and B are empirical calibration constants which are different for each fluid and each temperature. If Equation 8 is applied to gases, it is strictly true only if the gas is at constant pressure. Kramers[6] suggests the following correlation:

$$\mathrm{Nu} = 0.42\mathrm{Pr}^{0.2} + 0.57\mathrm{Pr}^{0.33}\mathrm{Re}^{0.50} \quad (9)$$

This correlation accounts for the variation in gas properties (ρ, k, μ, and Pr) with temperature and pressure. Kramers[6] evaluates these properties at the so-called "film" temperature $(T_e+T_a)/2$, rather than at T_a itself. Another comprehensive correlation is given by Churchill and Bernstein[7]. Several other correlations are similar to Equation 9, but have exponents for the Reynolds number ranging from 0.4 to 0.6. Others have 0.36 and 0.38 for the exponent of the Prandtl number.

Equations 8 and 9 are strictly valid only for hot-wire sensors with very high L/d ratios, in which case q_L and R_s are zero. The following universal correlation is suggested for real-world velocity sensors with variable gas temperature and non-zero q_L and R_s:

$$\mathrm{Nu} = \mathrm{A} + \mathrm{B}\mathrm{Pr}^{0.33}\mathrm{Re}^{n} \quad (10)$$

where constants A, B, and n are determined via flow calibration. Equation 10 is applicable to most commercial thermal sensors.

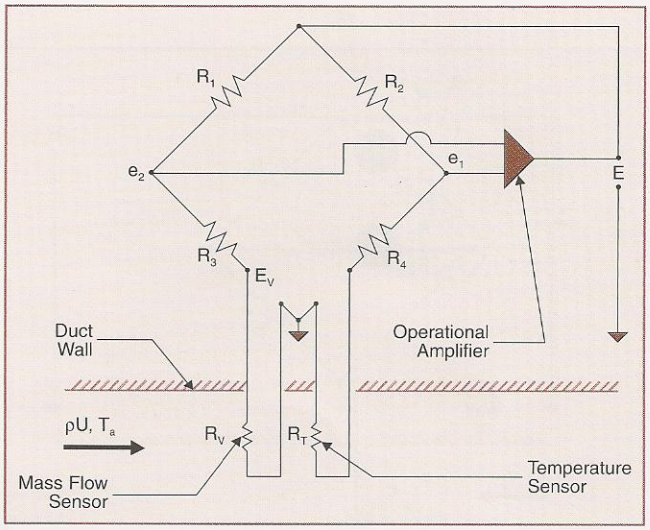

FIGURE 5. Constant temperature anemometer bridge circuit with automatic temperature compensation. R_1, R_2, and R_4 are fixed resistors selected to achieve temperature compensation. R_3 is the probe and cable resistance. R_v is the velocity sensor's resistance, and R_T is the temperature sensor's resistance. E is the bridge voltage output signal. Some temperature compensation circuits have an additional resistor in parallel with R_T.

Combining Equations 1, 2, 3, and 10 and recognizing that $h = kNu/d$, we arrive at:

$$E_v^2 / R_v = \left(\mathrm{A}k + \mathrm{B}k\mathrm{Pr}^{0.33}\mathrm{Re}^{n}\right)\left(T_v - T_a\right) \quad (11)$$

where A and B are new constants. A, B, and n are determined via flow calibration and account for all non-idealities including end conduction and skin resistance. Equation 11 is applicable to most commercial thermal sensors. Manufacturers of industrial thermal mass flowmeters may add other proprietary calibration constants or functions to Equation 11 to enhance correlation with flow-calibration data. For example, some manufacturers do not treat n as a constant, but allow it to decrease as Re increases.

For gas temperatures less than 200°C, the electrical resistance of the RTD mass flow and temperature sensors is usually expressed as:

$$R_v = R_{vo}\left[1 + \alpha_v\left(T_v - T_o\right)\right] \quad (12)$$

$$R_T = R_{To}\left[1 + \alpha_T\left(T_T - T_o\right)\right] \quad (13)$$

where R_{vo} and R_{To} are, respectively, the electrical resistance of the mass flow sensor and the temperature sensor at reference temperature, T_o (usually 0° or 20°C), and α_v and α_T are the temperature coefficients of resistivity at temperature T_o. When evaluated at the gas temperature T_a, the resistance R_a of the mass flow sensor is:

$$R_a = R_{vo}\left[1 + \alpha_v\left(T_a - T_o\right)\right] \quad (14)$$

Additional terms are added to Equations 12, 13, and 14 when gas temperatures exceed 200°C. At 20°C, α_v and α_T are approximately 0.00385°C^{-1} for pure platinum wire and 0.0024°C^{-1} for platinum film. R_v and R_a are called the "hot" and "cold" resistances of the mass flow sensor, respectively. The ratio R_v/R_a is called the "overheat ratio". For gas flows, sheathed thermal sensors are operated at overheat ratios from 1.1 to 1.4 ($T_v - T_a = 30°$ to 100°C). For water flows, the overheat ratio of research hot-film anemometer sensors is approximately 1.05 to 1.10 ($T_v - T_a = 10°$ to 20°C).

Combining Equations 12 and 14, we get:

$$T_v - T_a = \frac{R_v - R_a}{\alpha_v R_{vo}} \quad (15)$$

Inserting this into Equation 11, we arrive at:

$$\frac{E_v^2}{R_v\left(R_v - R_a\right)} = \mathrm{A}k + \mathrm{B}k\mathrm{Pr}^{0.33}\mathrm{Re}^{n} \quad (16)$$

where new constants A and B have absorbed the constants α_v and R_{vo}.

Figures 5 and 6 show two electronic drives for thermal anemometer sensors. Figure 5 shows the commonly used constant-temperature anemometer Wheatstone bridge circuit described by Takagi[8]. In the constant-temperature mode, the hot resistance R_v, and hence the mass flow sensor's temperature, remains virtually constant, independent of changes in mass flow rate. With the addition of the temperature sensor shown in Figure 5, the bridge circuit also compensates for variations in gas temperature, T_a.

Another common analog sensor drive is the constant-current anemometer (Figure 6). In this mode, a constant current is passed through the mass flow sensor, and the sensor's temperature decreases as the mass flow increases. Since the entire mass of the mass flow sensor must participate in this temperature change, the sensor responds relatively slowly to changes in mass flow.

In laboratory operation, the resistance R_5 in Figure 6 is adjusted, and the corresponding current (I) through the sensor is measured with the ammeter (A). During calibration, the current is kept constant for each velocity setting, and the bridge is balanced by adjusting the resistance of R_5 and R_4 so that the galvanometer (G) reads zero. The velocity sensor's resistance, R_v, is determined from the bridge balance equation: $(R_v+R_3)/R_1=R_4/R_2$. Commercial constant-current thermal mass flowmeters automate the above process and deliver an output signal linearly proportional to mass flow rate.

Because the constant-temperature anemometer has a higher frequency response, excellent signal-to-noise ratio[2], and is easier to use, it is favored over constant-current anemometers by most industrial users. The constant-current anemometer with a very low overheat ratio is often used as a temperature sensor. Subsequently, references made herein to sensor electronics will be based on the constant-temperature anemometer.

In the constant-temperature anemometer drive shown in Figure 5, the resistances R_1 and R_2 are chosen to: (1) maximize the current on the mass flow sensor's side of the bridge so it becomes self-heated and (2) minimize the current on the temperature-sensor's side of the bridge so it is not self-heated and therefore is independent of velocity.

To further avoid self-heating, the temperature sensor must be sufficiently large in size. The ratio R_2/R_1 is called the "bridge ratio." A bridge ratio of 5:1 to 20:1 is normally used. In Figure 5 the operational amplifier, in a feedback control loop, senses the error voltage (e_2-e_1) and feeds the exact amount of current to the top of the bridge necessary to make the error voltage approach zero. In this condition, the bridge is balanced; i.e.,

$$\frac{R_1}{R_v+R_3}=\frac{R_2}{R_T+R_4} \tag{17}$$

or:

$$R_v=\frac{R_1}{R_2}(R_T+R_4)-R_3 \tag{18}$$

From Equation 18, we see that R_v is a linear function of R_T. This relationship forms the basis for analog temperature compensation.

Expressing the voltage E_v across the velocity sensor in terms of the bridge voltage E, we get:

$$E_v=\frac{ER_v}{R_1+R_3+R_v} \tag{19}$$

Inserting this into Equation 16, we arrive at the generalized expression for the first law of thermodynamics for the mass flow sensor:

$$E^2=G\left[\mathrm{A}k+\mathrm{B}k\left(\frac{\rho_s}{\mu}\right)^n \mathrm{Pr}^{0.33}U_s^n\right] \tag{20}$$

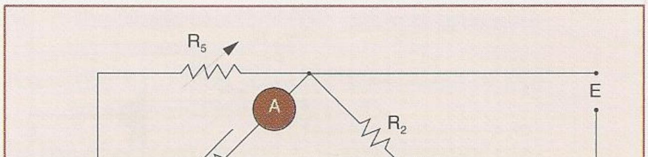

FIGURE 6. Constant current anemometer bridge circuit without temperature compensation. R_1 and R_2 are fixed resistors. R_3 is the probe and cable resistance. R_4 and R_5 are adjustable resistors. I is the constant current through the velocity sensor, and E is the bridge voltage output signal.

where $G=(R_1+R_3+R_v)^2(R_v-R_a)/R_v$, and A and B again are new constants. In Equation 20 we have recognized that conservation-of-mass considerations require that $\rho U=\rho_s U_s$, where ρ and U are referenced to the actual gas temperature and pressure, and ρ_s and U_s are referenced to normal conditions of 0° or 20°C temperature and 1 atmosphere pressure. To write Equation 20 in terms of U, we simply replace ρ_s by ρ and U_s by U.

TEMPERATURE COMPENSATION

The objective of temperature compensation is to make the bridge voltage E in Equation 20 independent of changes in the fluid temperature T_a. This is accomplished if (1) the term G in Equation 20 is independent of T_a and (2) compensation is made for the change in gas properties (k, μ, and Pr) with T_a. Since these gas properties have a weaker temperature dependence than G in Equation 20, for small temperature changes (less than ±10°C) in gas flows, only G requires compensation.

The two-temperature method is a typical procedure for compensating for both G and gas properties. In this method, fixed bridge resistors R_1, R_2, and R_4 in Figure 5 are selected so that E is identical at two different temperatures, but at the *same* mass flow rate. This procedure is accomplished during flow calibration and has variations amongst the manufacturers of industrial thermal mass flowmeters. □

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INDUSTRIAL THERMAL MASS FLOWMETERS

Part 2: Applications

In the first part of this two part article, a comprehensive explanation of the thermal principle of operation of thermal mass flowmeters was provided. Part 2 includes a description of the various sensor, flow body and electronics configurations available today. The author also provides information on typical applications.

JOHN G. OLIN, Ph.D.

The two-temperature method described at the end of Part 1 adequately compensates for temperature variations of less than approximately ±50°C. In higher temperature gas flow applications, such as the flow of preheated combustion air and stack gas, temperature variations can be higher.

The microprocessor-based digital sensor drive in Figure 1 provides temperature compensation for temperature variations ranging from ±50° to ±150°C. This sensor drive has no analog bridge. Instead, it has a virtual digital bridge that maintains $(T_v - T_a)$ constant within 0.1°C and algorithms that automatically compensate for temperature variations in k, μ, and Pr. For this digital sensor drive, the first law of thermodynamics can be written as:

$$w = \left[\mathrm{A}k + \mathrm{B}k \left(\frac{\rho_s}{\mu} \right)^n \mathrm{Pr}^{0.33} U_s^n \right] \Delta T \qquad (1)$$

where $\Delta T = (T_v - T_a)$ is now a known constant.

POINT MASS VELOCITY

Based on the first law of thermodynamics expressed above, we now can solve for either the actual point velocity U (m/s) or the point mass velocity U_s (nm/s) as measured by a single point insertion thermal mass flowmeter. Here we are assuming that the velocity vector is normal to the mass flow sensor. In the following, A, B, and n are constants, but are different for each case.

The simplest case is isothermal flow with a hot-wire sensor having a very high length-to-diameter ratio (L/d). If the fluid is a gas, the pressure also must be constant (e.g., ambient pressure). In this case, using King's Law[1], the exponent n becomes 0.5. The applicable first-law and point-velocity expressions are:

$$E^2 = \mathrm{A} + \mathrm{B}U^{0.5} \qquad (2)$$

and,

$$U = \left[\frac{E^2 - \mathrm{A}}{\mathrm{B}} \right]^2 \qquad (3)$$

In the case of a real-world thermal sensor in an isothermal, constant-pressure flow having either end loss only, or both end loss and skin resistance, we have:

$$E^2 = \mathrm{A} + \mathrm{B}U^n \qquad (4)$$

and,

$$U = \left[\frac{E^2 - \mathrm{A}}{\mathrm{B}} \right]^{1/n} \qquad (5)$$

Often, Equation 5 is replaced with a polynomial of the form $U = F(E)$, where the function F is a fourth-order (or higher) polynomial whose coefficients are determined from flow-calibration data using least-squares curve-fitting software.

Since temperature varies in this case, we must use the point mass velocity U_s (nm/s) instead of the point velocity U (m/s). This is expressed as:

$$U_s = \frac{\mu}{\rho_s} \left[\frac{E^2/\mathrm{G} - \mathrm{A}k}{\mathrm{B}k\mathrm{Pr}^{0.33}} \right]^{1/n} \qquad (6)$$

For the digital sensor drive of Figure 1, the first law is given by Equation 1, and the point mass velocity by:

$$U_s = \frac{\mu}{\rho_s} \left[\frac{w/\Delta T - \mathrm{A}k}{\mathrm{B}k\mathrm{Pr}^{0.33}} \right]^{1/n} \qquad (7)$$

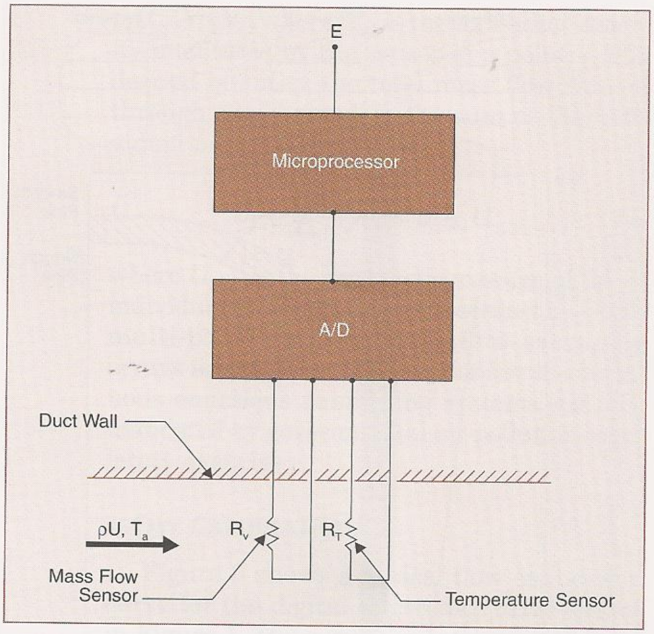

FIGURE 1. Microprocessor-based digital thermal anemometer. This system digitally maintains a constant temperature difference *(T_V-T_T)* and automatically corrects for the variation in gas properties with temperature. The probe-mounted electronics package delivers an analog output signal *E* and/or a digital RS-485 signal linearly proportional to gas mass velocity.

Current commercial industrial thermal mass flowmeters utilize temperature-compensation and "linearization" electronics which automatically calculate U_s as a linear function of E or w, based on the foregoing relationships.

CHANNEL FLOWS

If the flow in pipes, ducts, and stacks (hereafter called "channels") is perfectly uniform, then the total mass flow rate, $\dot{m}$, is measured by monitoring the point mass velocity U_s at any location in the channel's cross-section. In this idealized case, we could calculate $\dot{m}$ as:

$$\dot{m} = \rho_s A_c U_s \qquad (8)$$

where ρ_s, a constant, is the mass density of the gas at *normal* conditions of 0° or 20°C and atmospheric pressure, and A_c, another constant, is the channel's cross-sectional area. Unfortunately, the velocity profile is not uniform at most practical or accessible monitoring locations in channels.

In the case of pipes, most monitoring locations have upstream disturbances such as a single elbow, two elbows in the same plane, two elbows in separate planes, expansions, contractions, or valves. Such upstream disturbances create non-uniform, swirling flows that persist for many diameters downstream. *The single most common error in flow monitoring installations is lack of respect for the persistence of these non-uniform flows.* The old rule of thumb is that 10 diameters downstream of a disturbance is sufficient to eliminate flow non-uniformities. However, tests have shown that many non-uniformities, particularly swirling flows, persist for 20, 40, and even 80 diameters downstream.

In the case of ducts, the situation is even worse because length-to-diameter-ratios are reduced due to the large cross-sectional areas. Duct runs are typically tortuous and confounded by headers and louvers. Stacks suffer from the same maladies as ducts but are further complicated by multiple feeder ducts entering their base and often by severe temperature stratifications. Preheated air ducts have the same problem.

Industrial thermal mass flowmeters accommodate non-uniform channel flows via four configurations:

- **Single-point insertion meters** for fully developed flows or field-calibrated flows with minimal change in the shape of the velocity profile.
- **In-line meters** for fully developed flow in pipes.
- **Flow-conditioned in-line meters** for the flow in pipes with upstream disturbances within 20 diameters of the thermal mass flowmeter.
- **Multi-point flow-averaging arrays** for the flow in larger ducts and stacks (0.5m, or more, in diameter).

SINGLE-POINT INSERTION METERS

Single-point insertion meters monitor the point mass velocity at the channel's centerline. The total mass flow rate is calculated from this measurement as follows:

$$\dot{m} = \rho_s A_c \gamma U_{s,c} \qquad (9)$$

where $U_{s,c}$ is the point mass velocity component parallel to the channel's axis measured by the thermal meter at the channel's centerline and referenced to normal conditions of 0° or 20°C temperature and 1 atmosphere pressure, and γ is defined as $\gamma = U_{s,ave} / U_{s,c}$, where $U_{s,ave}$ is the average velocity over area A_c.

We derived Equation 9 by replacing U_s in Equation 8 with $U_s = U_{s,ave} = \gamma U_{s,c}$. Since velocity in channel flows is seldom uniform, γ is not unity. If a straight flow channel has a length-to-diameter ratio of 40 to 80, then its flow profile becomes unchanging and is called "fully developed." In fully developed flows, the viscosity of the gas has retarded the velocity near the walls, and hence γ is always less than unity. If the channel's Reynolds number is less than 2000, the flow is laminar; the fully developed profile is a perfect parabola; and γ is 0.5.

For fully developed flows, Figure 2 reveals that γ is a function of the channel's Reynolds number, $Re_D = \rho_s U_{s,ave} D / \mu$, where D is the internal diameter of a circular channel (e.g., a pipe). For rectangular and other non-circular channels, D is the *hydraulic diameter*, defined as four times the cross-sectional area of the channel divided by its perimeter. The curve for smooth walls in Figure 2 is based on a four-segment velocity profile presented in Reference 2 and supersedes the traditional results given in Reference 3. From Figure 2 we see that γ is a weak function of large Reynolds numbers, in which case, fortunately, it can be treated as a constant.

For example, γ is 0.85 ± 0.01 for Reynolds numbers exceeding 40,000. Wall roughness decreases γ by about 0.02 for typical welded stainless steel pipes.

So, in the case of true fully developed flows, Figure 2 can be used to determine γ and, based on Equation 9, a single-point insertion meter, located at the channel's center line, provides a relatively low-cost method for monitoring total mass flow rate.

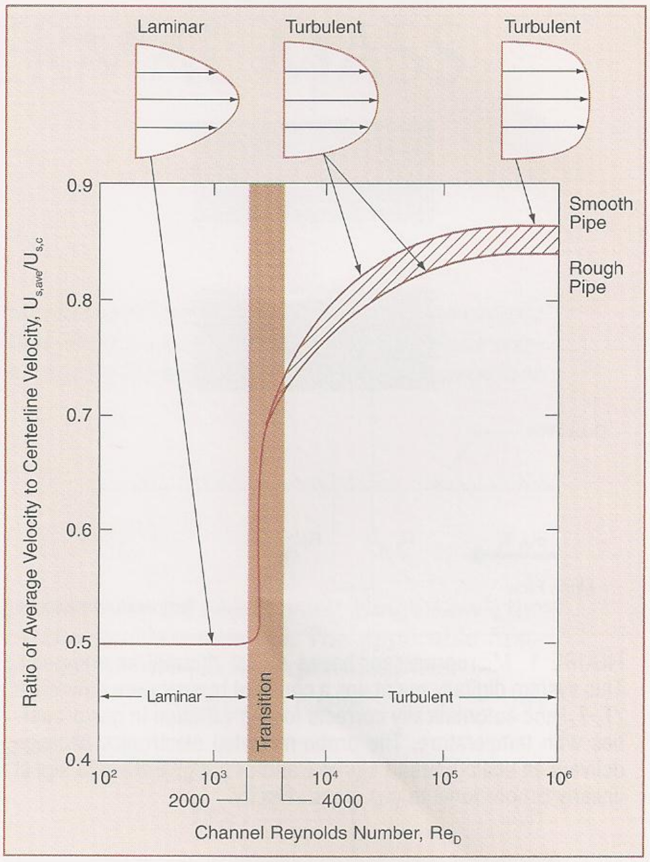

FIGURE 2. The ratio of average velocity to center-line velocity for fully developed flow in circular pipes[2].

IN-LINE FLOW CONDITIONED METERS

In-line flowmeters *of all types* substantially sacrifice their inherent accuracy if they are located too close to upstream disturbances. Thermal mass flowmeters and most other flowmeters require at least 20 diameters of straight pipe downstream of the disturbance to preserve their inherent accuracy. In cases where this exists, a non-flow-conditioned in-line meter does the job. In the more common case, where long straight pipe runs are either impractical or unavailable, a flow-conditioned in-line thermal mass flowmeter is required.

Some meters have built-in flow-conditioning modules located upstream of the sensing head. Table 1 shows the performance of a flow-conditioned meter for typical upstream disturbances, compared with orifice plates and flow nozzles[3].

In-line thermal mass flowmeters, with or without flow conditioning, are calibrated in terms of the total mass flow rate $\dot{m}$ passing through them, not in terms of the point velocity at the location of the sensing head. Thus, Equation 9 is not used to calculate $\dot{m}$, but is useful in describing the working principle of the in-line meter. For smart thermal meters with a digital sensor drive, the center-line velocity $U_{s,c}$ is given by Equation 7. Inserting this expression into Equation 9, we get:

$$\dot{m} = \mu \left[\frac{W / \Delta T \text{ - } \mathrm{A}k}{\mathrm{B}k\mathrm{Pr}^{0.33}} \right]^{1/n} \quad (10)$$

where we have absorbed the channel's cross-sectional area A_c into a new constant B and γ is absorbed into new constants B and n. The constants A, B, and n are determined from flow calibration of $\dot{m}$(kg/s) vs w. Flow-conditioned and non-flow-conditioned thermal mass flowmeters are calibrated with a perfect fully-developed flow upstream.

MULTI-POINT FLOW AVERAGING ARRAYS

Multi-point flow averaging arrays solve the problem of non-uniform flows in large ducts and stacks. As described by Olin[4], this method consists of a total of N (usually, N=4, 8, or 12) mass flow sensors located at the centroid of an equal area, A_c/N, in the channel's cross-sectional area A_c. The individual mass flow rate $\dot{m}_i$ monitored by each sensor is

$\rho_s U_{s,i}(A_c/N)$, where $U_{s,i}$ is the individual velocity monitored by the sensor at a point, i. The desired quantity, the total mass flow rate $\dot{m}$ through the channel, is the sum of the individual mass flow rates, or:

$$\dot{m} = \sum_{i=1}^{N} \dot{m}_i = \rho_s A_c U_{s,ave} \qquad (11)$$

where $U_{s,ave}$ is the arithmetic average of the N individual velocities $U_{s,i}$. As described by Olin[5], multi-point thermal mass flow-averaging arrays are used as the flow monitor in continuous emissions monitoring systems (CEMS) mandated by governmental air-pollution regulatory agencies.

FLOW CALIBRATION

Figure 3 shows a typical flow calibration curve for the digital electronics drive shown in Figure 1. The curve is nonlinear and logarithmic in nature. The nonlinearity is disadvantageous because it requires linearization circuitry, but is advantageous because it provides rangeabilities up to 1000:1 for a single mass flow sensor. Additionally, the high level output of several volts provides excellent repeatability and requires no amplification other than that for spanning. Since the critical dimensions of thermal mass flow sensors are small, current manufacturing technology is incapable of maintaining sufficiently small tolerances to insure sensor reproducibility.

Therefore, each thermal mass flowmeter must be calibrated over its entire velocity range, either at the exact gas temperature of its usage or over the range of temperatures it will encounter if it is to be temperature compensated. A 10 to 20 point velocity calibration is required to accurately determine the calibration constants A, B, and n. A least-squares curve fitting procedure usually is applied.

Proper gas flow calibration is based on two critical elements:

- A stable, reproducible, flow-generating facility.
- An accurate velocity transfer standard.

FLOW GENERATORS

Gas-flow generating facilities are of two types: open loop and closed loop. An open-loop facility consists of:

- A flow source such as a fan, pump, or compressed gas supply.
- A flow-quieting section, such as a plenum with flow straighteners, screens, or other means to reduce swirling, turbulence, or other flow non-uniformities.
- A nozzle to accelerate the flow and further flatten, or create uniformity, in the velocity profile.
- A test section or free jet into which the sensor probe is inserted.
- A means for mounting and sealing the probe and velocity transfer standard.

TABLE 1 — Straight pipe requirements to insure inherent flowmeter accuracy for thermal mass flowmeters, orifice plates, and flow nozzles. The orifice and flow nozzle data is from ISO Standard 5167. If a tubular flow conditioner is used, ISO 5167 requires 20 diameters between the upstream disturbances and the conditioner and 22 diameters between the conditioner and the orifice.

Flowmeter Type	Size (Inches) or Beta	Number of Diameters of Straight Pipe Between Flowmeter and Flow Disturbance					
		Upstream Disturbance					Downstream Disturbance of any Kind
		Single Elbow	Double Elbow, Same Plane	Double Elbow, Two Planes	Reducer (Ratio)	Globe Valve	
Orifice or Flow Nozzle	$\beta = 0.6$	18	26	48	9 (2:1)	26	7
	$\beta = 0.7$	28	36	62	14 (2:0)	32	7
Industrial Thermal Mass Flowmeter	0.25" to 0.75"	1	1	3	1 (4:1)	2	0
	1.0" to 4.0"	1	3	5	3 (4:1)	2	0
	6.0" to 8.0"	1	3	5	3 (4:1)	5	0

The test section or free jet must have a velocity profile which is uniform within approximately 0.5 to 1.0% in its central portion; a turbulence intensity less than about 0.5%; and, to avoid flow-blockage effects, an area large enough so that the projected area of the velocity probe is less than 5 to 10% of the cross-sectional area. For improved accuracy, corrections are made for flow-blockage effects. Manufacturers of small open-loop flow calibrators often determine the calibration flow velocity by measuring the pressure drop across the nozzle.

The closed-loop flow generating facility, or wind tunnel, has the same components, but the exit of the test section is connected via duct work to the inlet of the fan or pump so that the air mass inside the facility is conserved. Open-loop facilities are less expensive than closed-loop tunnels and are far more compact, making them suitable for flow calibrations in the field. But, a laboratory open-loop air-flow calibrator with a fan as the flow generator actually is closed loop with the loop closing within the laboratory.

For air velocities less than about 5m/s, open-loop calibrators can experience shifts due to changing pressure, temperature, or other conditions in the laboratory. Properly designed closed-loop wind tunnels generate precise, reproducible velocities from about 0.5 to 150m/s. When fitted with water chillers, they remove compression heating and provide a constant-temperature air flow within ±2°C. When fitted with an electric heater and proper thermal insulation, they provide air temperatures up to 300°C.

Pitot-tubes and laser doppler anemometers are the two most common velocity transfer standards used to calibrate thermal anemometers. The pitot tube usually has the classical "L" shape and an outside diameter of about 3mm. Its tip is located in the same plane in the test section as the thermal anemometer probe but is no closer than approximately 3cm. The focal volume of the laser doppler anemometer is similarly located. The pitot tube is far less expensive and easier to operate, but is difficult to use if air velocities are less than about 3m/s. A proper pitot-tube flow transfer standard should have its calibration recertified every six months by an accredited standards laboratory.

On the other hand, the laser doppler anemometer is a fundamental standard that accurately measures air velocity from 0.1 to 100m/s. Since it provides noncontact anemometry, it is usable at high temperatures. Its primary disadvantages are cost and complications associated with properly seeding the flow with particles.

COMMERCIAL METERS

Commercial thermal mass flowmeters have three elements — the sensor, the probe (or in-line flow body), and the electronics. Sensors and probes have been described in previous sections. The electronics of thermal mass flow systems are mounted in an explosion-proof or other industrial-grade housing either mounted directly on the probe or remotely (usually within 30m).

The electronics are powered with a 24 VDC source, or with 100, 115, or 230 VAC line voltage. The output signal typically is 0-5 VDC, 4-20 mA, RS-232, and/or RS-485 linearly proportional to gas mass velocity U_s (nm/s) over the range of 0.5 to 150nm/s for typical gases. In-line mass flowmeters have the same output signal options and are calibrated directly

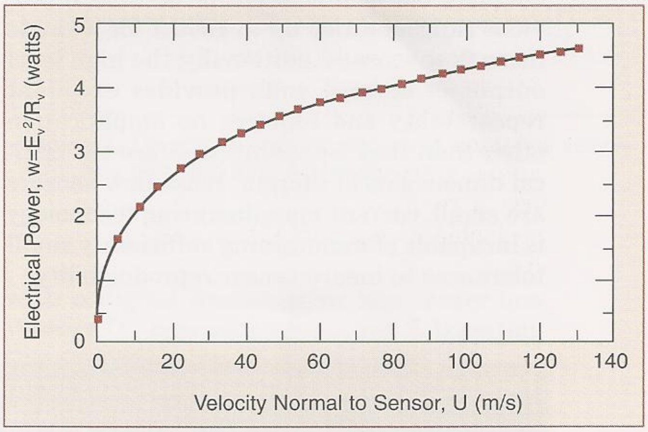

FIGURE 3. Typical flow calibration curve for an industrial thermal mass flowmeter. The electronics drive is that shown in Figure 1. The constant temperature differential (T_V-T_T) is 50.0°C. The cold resistances R_{Vo} and R_{To} of the velocity and temperature sensors at 20°C are approximately 20 and 200Ω, respectively.

TABLE 2 — Typical field-adjustment features of smart thermal mass flowmeters.

- Local reconfiguration access via finger touch through explosionproof window.
- Local alphanumeric display.
- Remote reconfiguration via RS-232 digital communication with a personal computer.
- Active validation of mass flow and temperature sensing RTD's and of system electronics.
- Field reconfiguration of range, zero, span, digital display's units, output signal alarm settings, totalizer, time response, and calibration-correction factors.
- Selection of one of several gases.

in mass flow rate $\dot{m}$ (kg/s) over the range of 0.05 mg/s to 5 kg/s for typical gases. In-line meters are now available with built-in flow conditioners which eliminate errors associated with upstream disturbances, such as elbows, valves, and pipe expansions.

Systems have either lower cost analog electronics or smart microprocessor-based electronics. Smart systems are now available with local field adjustments via finger-touch buttons activated through the explosionproof window. RS-485 digital communication facilitates the remote mounting of electronics up to 1.5 km from the probe. In the future, smart meters will feature fieldbus communications via hand-held communicators. Table 2 shows the field-adjustment features currently available with smart thermal mass flow systems.

The repeatability of thermal mass flowmeters systems is ±0.2% of full scale. The typical accuracy of a smart thermal meter is ±2% of reading over 10 to 100% of full scale and ±0.5% of full scale below 10% of full scale. Automatic temperature compensation facilitates temperature coefficients of ±0.04% of reading per °C within ±20°C of calibration temperature and ±0.08% of reading per °C within ±40°C. Smart systems accommodate high temperature applications and provide temperature compensation over a range of ±150°C. Pressure effects are negligible within ±300kPa of calibration pressure.

CONCLUSION

Because of their fragility, for many years hot-wire anemometers were considered inappropriate for industrial applications. In the 1960s and 1970s manufacturers introduced rugged, metal-sheathed thermal anemometer sensors to serve the growing need for mass flow measurement in industrial applications. As a result of twenty-five years of successful installations, industrial thermal mass flowmeters have now achieved the credibility formerly afforded only traditional approaches, such as orifice plates and linear volumetric flowmeters.

For the measurement of gas mass flow rate, industrial thermal mass flowmeters offer the lowest initial cost and total cost of ownership when compared to traditional approaches. Four thermal meter configurations — single-point insertion, in-line, flow-conditioned in-line, and multi-point flow averaging arrays — provide the solution to most industrial gas mass flow monitoring applications.

The flow-conditioned in-line configuration solves the biggest single installation problem — flow non-uniformities caused by upstream flow disturbances. Smart electronics provide a digital sensor drive, digital communication, direct verification of sensor operation, and complete field configurability. □

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THE AUTHOR

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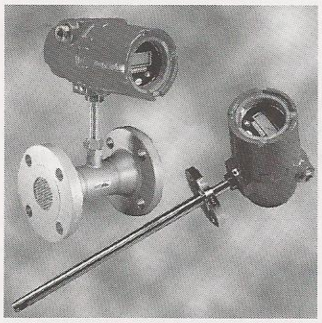


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