ON-SITE PROVING OF GAS TURBINE METERS

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INTRODUCTION

With the present day volatility in the price of natural gas, there is continued emphasis by gas producers, transporters, and distributors on accurate gas measurement at custody transfer points. Accurate metering ensures that both the buyer and seller are treated fairly and equitably in each transaction. Some of the other benefits of accurate measurement include better line balance and minimization of lost and unaccounted for gas volumes, more secure deliverability, fewer custody transfer disputes, and optimization of metering equipment maintenance.

Natural gas flow rate measurement errors at a field meter station can have many causes. For example, assume a field meter station utilizes a gas turbine meter to measure throughput. The turbine meter underwent a complete check and flow calibration at the factory prior to installation at the field meter station. However, many things can adversely affect the meter performance after it has been installed on site. Causes for meter error may include rotor bearing wear, dirt accumulation on the rotor blades, or mechanical damage to one or more blades. Operational parameters stored in the flow computer may have been entered incorrectly in introduce errors in the flow rate calculations.

The best method for identifying and eliminating these or other sources of meter error is with in-situ (on-site) calibration of the meter. That is, the measurement accuracy of the field meter station should be verified under actual operating conditions by comparing to a 'master' meter or field meter 'prover.' Although the following discussion focuses on ways to field prove the flow performance of gas turbine meters, the basic concepts and test methods may be applied to most types of gas flow meters.

FIELD PROVER CLASSIFICATIONS

A field meter prover can be classified as either a 'primary' flow standard or a 'secondary' flow standard. A primary flow standard is any measurement device that determines gas flow rate from the fundamental physical measurements of mass (M), length (L) or volume (V), temperature (T), and time (t). Since there are national and international standards for these fundamental physical measurements, field meter provers that determine flow rate from such measurements are considered primary flow standards. Primary flow standards usually provide the most accurate flow rate measurements. Examples of primary flow standards include weigh tank systems, swept-volume piston provers, and bell provers.

Measurement devices based on other techniques or methods are categorized as secondary flow standards. For highest accuracy, a secondary flow standard (sometimes also called a 'transfer' standard) is typically calibrated using a primary flow standard at operating conditions. Examples of secondary flow standards include critical flow Venturis (or sonic nozzles), turbine meters, and orifice meters.

Two comprehensive reports on the subject of field meter proving have been produced by Park, et al.[1] and Gallagher.[2] In addition, a paper by Rudroff[3] provides guidance on the mechanics of planning and conducting a field proof of a gas turbine meter. Much of the following information is referred to in detail in these reports and technical papers.

FUNDAMENTAL PRINCIPLES

Figure 1 illustrates the control volume associated with two flow meters plumbed in series. Assume that the upstream meter (Meter No. 1) represents the field meter and that the downstream meter (Meter No. 2) represents the prover being used to check the field meter.

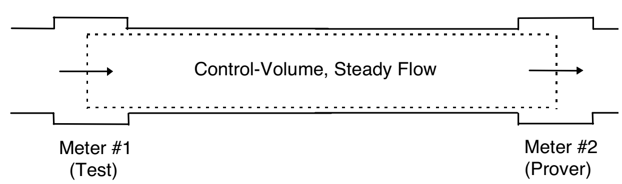

FIGURE 1. Control Volume for Two Flow Meters in Series

Now also assume that there is steady flow through the two meters. In that case, the mass flow rate through each meter will be the same (i.e., the law of conservation of mass applies). *That is, for steady-state flow conditions, at any given instant, the mass flow rate through Meter No. 1 will be equal to the mass flow rate through Meter No. 2.* This conservation of mass can be expressed mathematically by the following equation:

$$-\iint_{CS_1} (\rho V_n)\, dA = \iint_{CS_2} (\rho V_n)\, dA \text{ at time } t$$

where:

ρ = gas density
V_n = the component of gas velocity normal to the control surface
A = cross sectional area of prover
$_{cs}$ = a surface of the control volume where fluid crosses

Thus, the performance of the two meters should be compared on a mass flow rate basis. For most turbine flow meters, the measured flow rate is expressed in terms of volumetric flow rate. The line pressure (and, probably, the gas temperature) will be different at the two meter locations, so the actual volumetric flow rate measured by the two meters will not be equal. As an alternative to comparing the field turbine meter and the prover on a mass flow rate basis, the volumetric flow rates recorded by the two meters can be adjusted to 'standard' conditions and then compared. Standard volumetric flow rate is mass flow rate that has been referenced to arbitrary temperature and pressure conditions (e.g., a predetermined pressure and temperature, such as 14.73 psia and 60°F, respectively) for the flowing gas composition. Standard volumetric flow rate is proportional to mass flow rate through the application of standard gas density and is, therefore, conserved from location 1 to location 2.

FIELD PROVER TYPES

Several different measurement technologies have been employed for field proving natural gas turbine meters. These are discussed, in general terms, in the following sections.

PISTON PROVERS

Since primary flow standards are typically the most precise, they are the preferred choice for field proving gas turbine meters. Unfortunately weigh tanks and bell provers are typically too large and bulky for most natural gas pipeline applications. At present, the only viable primary flow standard available for field meter proving gas turbine meters is the gas piston prover. Figure 2[4] shows one gas piston prover design that has been used successfully in high pressure natural gas applications.

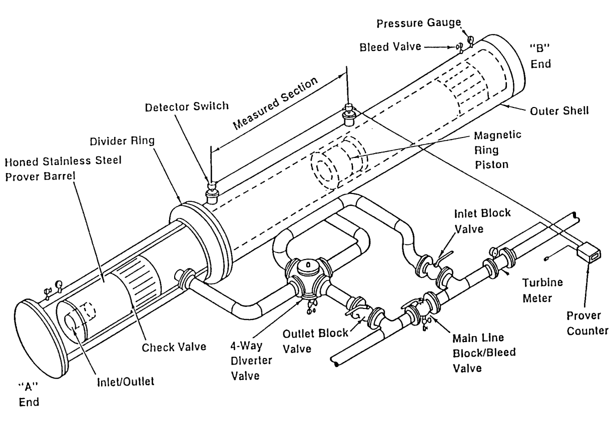

FIGURE 2. Three-Dimensional Drawing of Double-Wall Bi-Directional Gas Piston Prover[1]

The prover configuration shown in Figure 2 was developed by Amoco in the early 1990s. It includes a four-way diverter valve that makes it possible for the piston to travel in either direction through the prover.

A gas piston prover typically consists of a section of pipe of known inside diameter and a movable piston that travels inside the pipe. The piston is accelerated from rest by the flowing gas until it achieves a constant speed. While the piston travels through the pipe at a constant velocity, proximity sensors and electronic timing circuits measure the interval of time it takes the piston to move a predetermined distance. By knowing the amount of time it takes for the piston to travel a fixed distance and by knowing the inside dimensions of the pipe, it is possible to accurately calculate the volumetric flow rate of the gas at the operating conditions.

At present, there are no industry standards for piston-type gas provers. However, several variations of field piston provers have been devised. Park, et al.,[1] has summarized the performance limits of several gas piston provers (see Table 1).

Advantages of gas piston provers include (1) their compactness as portable primary flow measurement standards and (2) their ability to function at high pressure with any gas composition (thus, allowing meter calibrations to be performed under actual operating conditions).

The primary disadvantages of gas piston provers are (1) that they must be manufactured to high precision and, thus, are relatively expensive and (2) they must be large to directly calibrate at high flow rates. Currently, the largest turbine meter that can be calibrated with a portable piston prover is an 8-inch (203 mm) diameter meter.

SONIC NOZZLES

Sonic nozzles or critical flow Venturis are flow restrictions that accelerate flow to the speed of sound in the narrowest portion or throat of the nozzle, at which time the maximum flow rate is attained. If the thermodynamic state of the gas at the sonic nozzle throat is known, then the speed of gas at the throat may be calculated from state equations, since the throat speed is equal to the speed of sound at those conditions. For a given nozzle geometry, under sonic or 'choked' flow conditions, the flow rate through the nozzle is a function only of the upstream (i.e., plenum) temperature, pressure, and gas composition. Figure 3 shows a typical sonic nozzle installation configuration.

Location	Volume ft^3 (m^3)	Pressure psig (MPa)	Flow Rate acfh (m^3/h)	Uncertainty %	Reference
Amoco Houston, TX USA	2.8073 (0.0795)	1440 (9.9)	18,000 (510)	--	Beaty (1991)[4]
Brooks Instrument	--	870 (6.0)	14,000 (396)	±0.20	Reid & Pursley (1986)[5]
Gasunie Groningen, NL	5.2036 (0.14735)	870 (6.0)	8,800 (250)	±0.13	Bellinga, et al. (1985)[6]
Gasunie Groningen, NL	49.3982 (1.3988)	960 (6.6)	71,000 (2,000)	±0.15	Bellinga & Delhez (1993)[7]
OGASCO Houston, TX USA	4.811 (0.1362)	1440 (9.9)	18,000 (510)	±0.35	Ting & Halpine (1991)[8]
PTB Braunschweig, FRG	5.2337 (0.14820)	1300 (9.0)	16,000 (450)	±0.13	Schmitz & Aschenbrenner (1990)[9]

TABLE 1. Performance Summary of Gas Piston Provers[1]

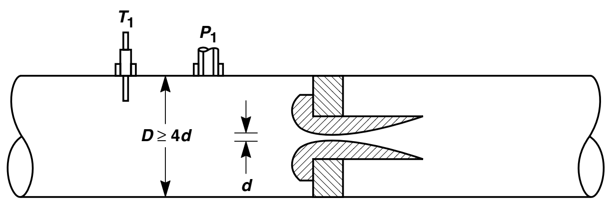

FIGURE 3. Example Sonic Nozzle Installation Configuration

where:

D = inside diameter of the meter tube plenum

d = inside diameter of the nozzle throat

T_1 = nozzle plenum gas temperature

P_1 = nozzle plenum gas pressure

A sonic nozzle has no moving parts. The contour geometry may vary. At one extreme is the critical flow orifice. Cunningham[10] showed that a standard, thin, square-edged orifice would not achieve choked flow. Ward-Smith[11] showed, however, that an orifice would choke if the plate were made thicker than a standard, thin, square-edged orifice. For plate thickness to bore diameters between 1 and 6, the Ward-Smith data suggest that the sharp-edged orifice discharge coefficient is 0.839, with an estimated accuracy of ±1.6% of reading. Higher discharge coefficients may be achieved by contouring the upstream edge of the bore. Higher pressure recovery may be achieved by contouring the downstream edge of the bore.

The American Society of Mechanical Engineers (ASME) has published a standard for toroidal throat Venturi nozzles.[12] The International Standards Organization (ISO) has also published a standard for critical flow Venturi nozzles.[13] The ASME standard states that the discharge coefficient for toroidal throat Venturi nozzles approaches 0.9935 at high throat Reynolds numbers (which is a ratio of inertial forces to viscous forces). The ASME nozzle standard provides detailed construction and installation specifications. The ASME discharge coefficient correlation is estimated to have a bias limit of ±0.5% (95% confidence).[14] Better nozzle discharge coefficient data may be obtained by flow calibration.

Sonic nozzles are currently being applied as a measurement transfer standard in many flow laboratories, and they have been successful in field prover applications. Chevron has operated several sonic nozzle banks as field provers at custody transfer sites.[15] Their largest facility has eight sonic nozzles with a flow capacity of 302,000 acfh (8,500 m3/h) at 1,000 psig (6.9 MPa). The system also includes a turbine meter as a 'master' meter.

One unique sonic nozzle prover developed as a portable field prover for calibration of turbine meters is shown in Figure 4 and described in detail by Beeson.[16] The device, called a Digicell®, contains 11 binary-weighted sonic nozzles with a resolution of 58.7 acfh (1.66 m3/h) and a maximum flow rate of 138,000 acfh (3,900 m3/h) at a maximum operating pressure of 1,440 psig (9.9 MPa).

The maximum flow capacity of this prover is equivalent to a 12-inch (300 mm) turbine meter. Reliant Energy Corporation uses the Digicell® mounted on a truck for field meter proving applications. The measurement accuracy of the Reliant system is quoted as ±0.5% of reading. A system such as the Reliant truck-mounted system must be flow calibrated as a complete system.

Since a sonic nozzle has no moving parts, the accuracy of the meter is a function of: (1) the biases associated with the predictive thermodynamic model and discharge coefficient correlation, (2) the degree of thermodynamic equilibrium at the nozzle plenum and throat, and (3) the accuracy of the temperature, pressure, and gas composition measurements. A sonic nozzle requires only two sensors to measure the stagnation (often approximated by the static) temperature and pressure

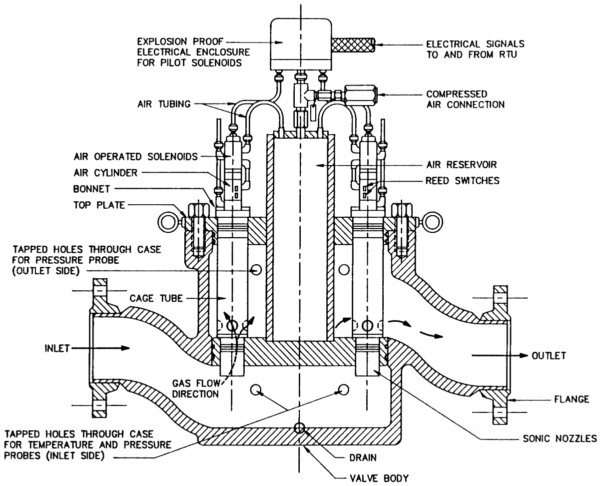

FIGURE 4. Digicell® Sonic Nozzle Prover[16]
(This device is used in a field meter prover system currently owned and operated by Reliant Energy Corporation.)

in the low velocity (relative to the nozzle throat) plenum region upstream.

There are no mechanical contact surfaces requiring lubrication in a sonic nozzle. No inertia occurs to delay transient response. At the choked condition, sonic nozzle flow rate is a function only of the upstream thermodynamic state. The thermodynamic conditions downstream do not affect flow rate through the nozzle unless the nozzle unchokes (due to the back-pressure rising above the choking value, for example). The thermodynamic state in the upstream nozzle plenum may be determined by measuring the temperature, pressure, and composition of the flowing gas. Although the one-dimensional isentropic process models and state equations required to calculate sonic nozzle flow rate are well defined, the calculations can be computationally intense under non-ideal conditions, depending on the level of reality employed by the model.

No pressure drop measurement across the nozzle is required to determine flow rate, as is the case with subsonic flow resistive meters, but one may be taken to ensure that the nozzle is choked. Sonic nozzles are often installed with an upstream stagnation plenum. Therefore, they are relatively insensitive to upstream piping configurations and may be installed in relatively short meter runs.

Due to the severe flow contraction and acceleration within a sonic nozzle, severe pressure and temperature drops may occur from the plenum to the throat. The resulting thermodynamic changes in the gas may affect composition in the nozzle throat region. For some natural gas mixtures, especially rich grades near production sites, retrograde condensation of heavy hydrocarbons may occur. In addition, natural gases that contain hydrogen sulfide (sour gas) can, under some conditions, undergo an oxidation reduction process to produce elemental sulfur in the gas phase. There have been reports of solid sulfur deposition in sonic nozzles, usually resulting from a retro-sublimation process (plating out the elemental gas-phase sulfur) in the throat of sonic nozzles.[17] The sulfur deposits can affect measurement accuracy by changing the nozzle throat geometry, reducing the throat diameter.

TURBINE METERS

A turbine meter may be used to prove the performance of another turbine meter, although care must be taken in doing so. A turbine meter measures fluid flow by the rotation of helical blades located around a circular hub. Essentially, a turbine meter is a volumetric meter, and its calibration constant is in pulses per unit volume. In the United States and ISO turbine meter standards, the maximum permissible measurement error is $\pm 1\%$ of reading over the operating range of the meter. Better accuracy can be achieved through calibration of the meter at the actual operating conditions (i.e., the same pressure, temperature, and gas composition).

The primary guidelines for turbine meters used in the United States are contained in American Gas Association (AGA) Report No. 7[18] and ANSI/ASME MFC-4M-1986,[19] while the relevant international standard is ISO 9951.[20] Additional supporting information is contained in the AGA Gas Measurement Manual.[21] These documents contain information on the following: construction, installation, operation, performance, flow measurement, calibration, and field checks. Information in the standards on turbine meters as field provers is minimal.

A schematic of a single-rotor turbine meter and its components is shown on Figure 5. Dual-rotor or 'self-correcting' or 'self-checking' turbine meter designs (called the Auto-Adjust® Turbine Meter described by Lee, et al.[22]) also exist. These are generally considered more robust than the single-rotor designs. They were originally conceived to overcome many of the operational shortcomings of the single-rotor design. Principal drawbacks of the turbine meter include (1) moving parts that are subject to wear and degradation in performance and (2) susceptibility of the meter to the effects of flow pulsation, swirl, and velocity profile distortion.

Integral straightening vanes (also called a flow conditioner) are usually installed in gas turbine meters to eliminate the influence of flow swirl on the rotor. This is particularly critical when a turbine meter is used as a prover placed downstream of a field turbine meter. The field turbine meter will generate swirl in the flow field downstream. This swirl must be eliminated before it reaches the turbine meter prover. Otherwise, the swirl may bias the prover measurement.

AGA Report No. 7 recommends installation of a straightening vane upstream of a turbine meter to eliminate any swirl in the flow stream. Usually, the

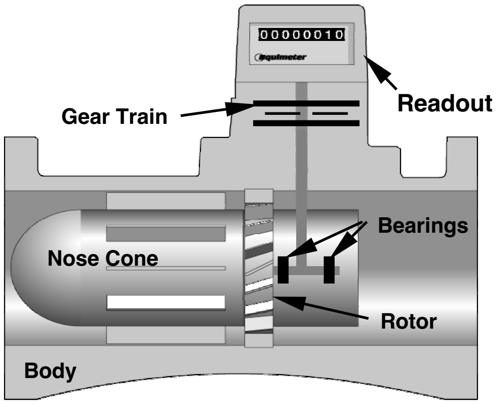

FIGURE 5.

straightening vane or flow conditioner consists of 19 tubes arranged in a concentric pattern. A typical turbine meter installation per the AGA Gas Measurement Manual is shown on Figure 6. This particular configuration provides for a proving meter (which could be a turbine meter) to be installed and operated in series with the field turbine meter.

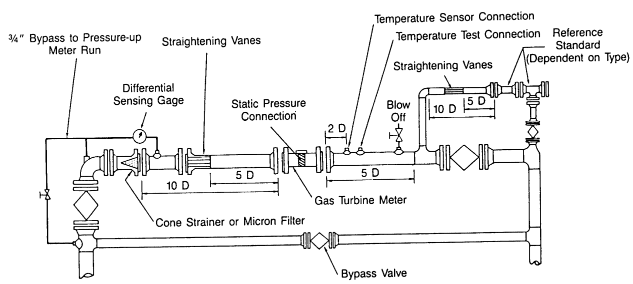

FIGURE 6. Turbine Meter Installation From AGA Gas Measurement Manual[21]

Chevron has had success using 12-inch (305 mm) diameter turbine meters to field prove several custody transfer meter stations.[15]

INSTRUMENTATION

Although the various provers described in this paper have different operating principles, they do have some common instrumentation. In particular, static pressure and temperature must be measured so that density and other physical properties of the gas can be computed from a gas model. Significant improvements in all provers can be achieved with the selection of state-of-the-art instruments for the measurement of temperature and pressure. The precise effect on uncertainty of the prover measurement will be dependent upon the model equation that computes the mass flow rate of the prover in relation to the meter being tested.

The uncertainty in prover measurements from the measurements associated with temperature and pressure is typically within ±0.05% of reading. Consequently, the largest uncertainty in a mass flow rate measurement for a field prover would be in the meter coefficient for a secondary standard, such as a turbine meter, or in the gas model for the density calculation. The uncertainties associated with the gas model and gas composition are discussed in the next section.

GAS COMPOSITION EFFECTS ON CALIBRATION ACCURACY

A complicating factor in field meter proving is the unavoidable reality that field meters are installed on natural gas pipelines and operate on natural gas. The uncertainty in the component concentrations of the gas mixture may significantly increase the composite uncertainty in the field meter calibration. A gas chromatograph (GC) is typically used to separate the gas mixture into its components and then measure the concentrations of the individual components. Guidance on the proper use of a GC is provided in Gas Processors Association (GPA) standards 2261[23] and 2286.[24] Uncertainties in gas composition affect calibration accuracy through the gas properties that are needed to compute flow rate.

Gas density is, arguably, the most important composition dependent property in flow rate measurements. Under nearly all field meter proving scenarios, the gas density at both the field turbine meter and the proving meter must be known, and each will add uncertainty to the calibration. Some meters are sensitive to other thermodynamic properties, such as sound speed, and may require complex thermodynamic calculations involving more properties to accurately model the fluid dynamic behavior. Further, calibration constants of flow restricting meters, such as a sonic nozzle, are often correlated to Reynolds number, and become more sensitive to Reynolds number in the low range. Reynolds number effects also become more important at low flow in turbine meters. The introduction of Reynolds number requires that viscosity be known. Many meters are dependent on both thermodynamic and transport/diffusion properties for accurate flow rate measurements.

Natural gas mixture density calculations may be performed very accurately using the Detail Characterization Method of AGA Report No. 8.[25] This method combines features of a virial equation of state at low density, and exponential functions at high density. A nominal bias of ±0.1% on compressibility factor is specified for most pipeline flow meters. Important derived thermodynamic properties such as heat capacity, enthalpy, entropy, and sonic velocity may be obtained by extending the compressibility results of AGA Report No. 8[25] using thermodynamic principles.

Grouping C_{6+} components in the compositional analysis is very common in process natural gas chromatography. This can lead to gas property calculation errors of the same magnitude as the accuracy of a meter prover. For example, consider a 'light' gas with a molecular weight of 17 that actually contains 0.05% by volume of C_7, and 0.025% by volume of C_8. A mixture density calculation error of about 0.1% may be realized by assigning n-hexane properties to the C_{6+} grouping.

CONCLUSIONS

Currently, the most viable technologies as large volume field provers are the sonic nozzle, turbine meter, and gas piston prover. These devices have an established history of high accuracy measurements with high resolution. Since turbine flow meters are volumetric, the best accuracy is attained when the meter proving system is calibrated against a primary volumetric standard with the same gas at approximately the same temperature and pressure. In this case, the bias uncertainty from gas composition is reduced, and pressure, temperature, and the volumetric flow rate of the flow standard determine the accuracy.

A comparison of a gravimetric (mass) prover and piston (volume) prover in the calibration of a turbine (volume) meter has been shown to be somewhat better for the piston prover calibration on natural gas (assuming no mass storage between the meters), because density (hence, gas composition) must be used to relate mass flow rate to volume flow rate. However, density (hence, gas composition) must still be used to apply the calibration coefficient of a volumetric meter to determine mass flow rate in service.

Gas piston provers should be used in a manner that eliminates or reduces mass storage in the interconnecting pipe. The test gas should be passed through the interconnecting pipe and prover barrel for sufficient time to reach thermal equilibrium prior to initiation of the proving run(s). The temperature in the interconnecting pipe should be monitored during the calibration run to ensure that no mass storage occurs between the field meter and prover.

For repeatable results, rotational or cyclical flow meters, such as turbine meters, should be calibrated using a large number of cycles (~10,000) or an integral number of mechanical cycles. An example of a mechanical cycle would be a single rotation of the rotor in a turbine meter. Usually, a turbine meter will generate a large number of rotor revolutions in a short period. However, when a turbine meter is calibrated by a piston prover, the number of rotor revolutions is greatly reduced, and an integral number of revolutions should be counted for a pulse interpolation scheme.[6,7]

REFERENCES

1. Park, J. T., et al., "Metering Research Facility Program: Review of Field Meter Provers," Gas Research Institute Report No. GRI-95/0209, December 1995.

2. Gallagher, James E., "Flowmeter Calibration Technology for Natural Gas," Presented at the 1999 Committee on Petroleum Measurement Spring Meeting, American Petroleum Institute, New Orleans, LA, March 29-April 2, 1999.

3. Rudroff, Daniel J., "Onsite Proving of Gas Turbine Meters," International School of Hydrocarbon Measurement, Oklahoma City, Oklahoma, May 22-24, 2001.

4. Beaty, R.E, "Field Operation of a Portable Piston Prover for Testing Gas Meters," 1991 Operating Section Proceedings, Paper 91-DT-96, AGA, Arlington, Virginia, pp. 673-678, 1991.

5. Reid, J. and W. C. Pursley, "An On-Line Prover for the Calibration of Flowmeters in High Pressure Gas," Proceedings of FLOWMEKO '86, Paper 8.3, National Engineering Laboratory, East Kilbride, Glasgow, Scotland, 1986.

6. Bellinga, H., et al., "Using a Piston Prover as a Primary Standard in High-Pressure Gas Metering," Proceedings of FLOWMEKO '85, Melbourne, Australia, 1985.

7. Bellinga, H. and F. J. Delhez, "Experience with a High-Capacity Piston Prover as a Primary Standard for High-Pressure Gas Flow Measurement," Flow Measurement and Instrumentation, Vol. 4, No. 2, pp. 85-89, 1993.

8. Ting, V. C. and J. C. Halpine, "Portable Piston Gas Prover for Field Calibration of Flowmeters," Society of Petroleum Engineers (SPE) Production Engineering, pp. 454-458, 1991.

9. Schmitz, G. and A. Aschenbrenner, "Experience with a Piston Prover as the New Primary Standard of the Federal Republic of Germany in High-Pressure Gas Metering," 18th World Gas Conference, Berlin, 1991.

10. Cunningham, R. G., "Orifice Meters with Supercritical Compressible Flow," Transactions of the ASME, Vol. 73, pp. 625-638, July 1951.

11. Ward-Smith, A. J., "Critical Flowmetering: The Characteristics of Cylindrical Nozzles with Sharp Upstream Edges," International Journal of Heat and Fluid Flow, Vol. 1, No. 3, pp. 123-132, September 1979.

12. *ASME/ANSI MFC-7M-1987*, "Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles," ASME, New York, N. Y., 1992.

13. *ISO 9300, 1990-08-15*, "Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles," ISO, CH-1211 Geneva 20, Switzerland, 1st Ed.

14. Hillbrath, H. S., "The Critical Flow Venturi-An Update," Flow: Its Measurement and Control in Science and Industry, Vol. 2, edited by W. W. Durgin, Instrument Society of America, Research Triangle Park, N. C., pp. 407-420, 1981.

15. Ting, V. C., et al., "On-Site Flow Calibration of Turbine Meters for Natural Gas Custody Transfer," SPE Production Engineering, pp. 161-165, May 1991.

16. Beeson, J., "Arkla Develops Mobile Turbine Flowmeter Prover," Pipeline Industry, pp. 43-49, April 1994.

17. Chesnoy, A. B., "Sonic Nozzles Meter Natural Gas at K-Lab," Flow Measurement and Instrumentation, Vol. 4, No. 2, pp. 73-76, 1993.

18. *AGA Transmission Measurement Committee Report No. 7*, "Measurement of Gas by Turbine Meters," AGA, Arlington, Virginia, 1996.

19. *ANSI/ASME MFC-4M-1986*, "Measurement of Gas Flow by Turbine Meters," ASME, New York.

20. *ISO 9951 : 1993 (E)*, "Measurement of Gas Flow in Closed Conduits—Turbine Meters," ISO, Geneva, Switzerland.

21. *AGA Gas Measurement Manual*, AGA, Arlington, Virginia, 1985.

22. Lee, W. F. Z., et al., "A Self-Correcting and Self-Checking Gas Turbine Meter," Journal of Fluids Engineering, Vol. 104, No. 2, pp. 143-149, June 1982.

23. *GPA Standard 2261-90*, "Analysis for Natural Gas and Similar Gaseous Mixtures by Gas Chromatography," GPA, 1990.

24. *GPA Standard 2286-86*, "Tentative Method of Extended Analysis for Natural Gas and Similar Gaseous Mixtures by Temperature Programmed Gas Chromatography," GPA, 1991.

25. *AGA Transmission Measurement Committee Report No. 8*, "Compressibility Factors of Natural Gas and Other Related Hydrocarbon Gases," 2nd Ed., AGA, Arlington, Virginia, Nov. 1994.

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