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ASRDI OXYGEN TECHNOLOGY SURVEY

Volume VI:

Flow Measurement Instrumentation

MANN

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

PREFACE

This publication is part of an oxygen safety review in progress by the NASA Aerospace Safety Research and Data Institute (ASRDI). The objectives of the review include:

1. Recommendations to improve NASA oxygen handling practices by comparing NASA and contractor oxygen systems including the design, inspection, operation, maintenance and emergency procedures.

2. Assessment of the vulnerability to failure of oxygen equipment from a variety of sources so that hazards may be defined and remedial measures formulated.

3. Contributions to safe oxygen handling techniques through research.

4. Formulation of criteria and standards on all aspects of oxygen handling, storage, and disposal.

This special publication provides a summary of information available on liquid and gaseous oxygen flowmetering including an evaluation of commercial meters. The instrument types, physical principles of measurement and performance characteristics are presented. Problems concerning flow measurements of less than ±2% uncertainties are also reviewed. Recommendations concerning work on flow reference systems, the use of surrogate fluids and standard tests for oxygen flow measurements are presented.

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FOREWORD

A survey of the literature combined with the results of a joint government-industry cooperative program on cryogenic flowmetering is presented. The objective was to establish the state of the technology and art of oxygen flowmetering in liquid and gaseous states. Only those meters with demonstrated performance were considered. These were classified as quantity, head, momentum and velocity types. A comparison of the performance of these devices and a discussion of future requirements for flow reference systems and metering are given.

I would like to thank many of my colleagues for valuable discussions during the preparation of this volume. In particular, I appreciate the contributions of J. A. Brennan of the NBS Cryogenics Division and the support of the Compressed Gas Association, a cosponsor of much of the experimental program on cryogenic flowmetering at NBS. I am indebted to ASRDI Project Manager Paul Ordin of the NASA-Lewis Research Center for his support and many helpful suggestions during the course of this work.

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KEY WORDS

Argon; calibration; cryogenics; flowmeter; measurement; nitrogen; oxygen.

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1. INTRODUCTION

The purpose of this report is to provide a summary of information available on liquid and gaseous oxygen flowmetering, including an evaluation analysis of commercial meters Efforts have been made to include information on oxygen flow measurement from documents of government agencies including NASA centers such as Marshall Space Flight Center, Johnson Space Center, Kennedy Space Center, as well as the U. S. Air Force, U. S. Navy and the U. S. Atomic Energy Commission. In addition, industrial trade organizations whose member firms deal with oxygen as a commodity have been asked to contribute. These include the Compressed Gas Association, the American Gas Association and the International Oxygen Manufacturer's Association. With this scope of input it is believed that an assessment can be made of the state of the art of oxygen flowmetering.

The information presented is based on oxygen flow measurement experiences; however, when considered applicable, flow measurement techniques and performance for other cryogenic fluids are related to possible oxygen performance.

The diversity of use of oxygen as an aerospace rocket oxidizer, in steel making, welding, non-ferrous metal refining, treatment of sewage, and breathing atmospheres to mention only a few applications does introduce limitations in providing summary information from which selections could be made to satisfy a particular requirement.

The method of presentation for oxygen flow instrumentation is as follows:

a. A description of the instrumentation type.
b. A schematic description of the physical principle of measurement.
c. A detailed description of the operation of typical meters including basic materials of construction.
d. Performance characteristics including but not limited to precision, accuracy and sensitivity of measurements.
e. A ranking or ordering of the particular types of flowmeters as to flow range, precision and accuracy.
f. Recommendations for application to oxygen service.

From this structure it is possible to show that the cryogenic applications of oxygen flow measurement can be reasonably identified. Liquid oxygen is only one of five or six commercially important cryogenic fluids, and the flowmetering description and methodology are easily identified. High purity is taken for granted because of the reactivity of oxygen to materials of construction and accumulated impurities [Bankaitis, Schueller, 1972]. Problems concerning safety of liquid oxygen flowmetering systems are discussed and include possible effects of materials, cryogenic impurities, etc. With respect to gaseous oxygen (GOX), as far as the literature is concerned it is considered as another gas in that it is not specifically identified separately from ambient temperature gas measurement problems. The report reviews briefly oxygen gas flow measurement systems, methods, and instruments used, performance parameters and because of the reactivity with materials, safety considerations.

Most information of flowmetering performance is provided during the developmental stages of the meter. Because of the reactivity of oxygen both in the cryogenic and gaseous form, surrogate fluids are used in these developmental phases with performance on oxygen occurring only during final proving

A discussion of the appropriateness of surrogate fluids is included in the following section, but it should be realized that strict adherence to citations concerned only with oxygen would severely limit this report and unjustly restrict information available from the literature and other sources cited.

2. FLOW AS A MEASUREMENT PROCESS

"Measurement is the assignment of numbers to material things to represent the relations existing among them with respect to particular properties. The number assigned to some particular property serves to represent the relative amount of this property associated with the object concerned." [Eisenhart, 1963]

Within the scope of this report, flow is the particular "thing" for which numbers will be assigned These numbers will relate the flow of a fluid to time, weight, volume, temperature, density, viscosity, and other thermophysical or transport properties, either known or unknown, which concern the flow measurement process.

For the particular case in point, oxygen flow (either liquid or gaseous) is the time dependent movement of a quantity of oxygen (weight or volume) past a certain datum point in a closed pipe or conduit. Flow may be either totalized or rate, depending on the particular measurement instrument involved.

The magnitude of the numbers assigned to flow measurement is found by comparison with a standard. Provision, promulgation, and maintenance of standards for measurement processes hold different meanings based on need or use. It is sufficient to say that there exists a hierarchy of standards ranging from the primary standards of mass, length, and time maintained by the National Bureau of Standards (NBS), to laboratory standards of a company or research agency, to the working standards, each related by comparison to the next higher level.

In the case of flow measurement, NBS does not provide or maintain primary standards for flow although calibration facilities are provided for certain fluids [Wildhack, et al., 1965 and Mason, 1968]. These flow calibrations are based on a combination of one or more primary standards such as mass, length (volume) and time and therefore provide a relationship between flow and primary standards. Flow prover test stands capable of providing calibration of the highest quality require continuous and exacting attention to detail in establishing and maintaining the measurement capability.

2.1. Precision, Accuracy, and Total Uncertainty

It is possible to conceive that once a flowmeter has been developed, it is capable of continuously producing numbers which represent the actual flow of fluid. These numbers are generated each time the meter is put into service and exist irrespective of observation: i.e., the meter is producing data all the time it is in service and does not depend on someone or something to record or interpret the number output. The meter is therefore generating a large population of numbered values representing primarily the flow of the fluid. When a reading of the flow dependent number is taken either by human, electrical or mechanical means, this reading is in essence a sampling of the larger population of number values. Evaluation of meter performance during the sampling period can then be established as well as a prediction of the meter performance sometime in the future.

This concept leads naturally to the consideration of the flow measurement process as a statistical problem and when considered in this light allows the many and varied tools of statistical analysis to be applied to the measurement problem. This concept is not new or original but has had a rather slow and tedious record of acceptance.

The "one number" approach to a description of meter performance is very appealing to both user and meter manufacturer alike. More exacting technological requirements and detailed analysis of flowmetering performance over the past years shows that flowmeters can and do measure other properties in addition to flow. These include but are not limited to such things as fluid temperature, density, viscosity, hydrodynamic forces, etc. The selection of the term "flowmeter" merely states that under a certain set of well defined conditions the number values generated by the device are predominately related to flow. Where, in the past, the flow measurement could allow errors of 5 percent or more, this propensity of the meter to indicate conditions of the process unrelated to flow has not been a problem.

Within the past ten years the situation has changed dramatically. The reasons are both economic (custody transfer) and technological (process control). Requirements now demand errors of less than 1 percent and in some cases to a tenth of 1 percent.

The "one number" approach cannot survive this new context. Statistical methods allow a description of meter performance consistent with the new requirements but also require an understanding of the entire process of flow measurement and the terms used in this description.

The demands for less uncertainty in flow measurement require an increase in the quantity and quality of information concerning the measurement process. Relationships or, in their absence, limits must be established for dependent functions including the flow parameter. In addition, experimental error, present in any measurement, must be defined and quantified with increasing certainty as total flow uncertainty is lowered.

Measurement technology is not a static field and is improving and changing to meet the demands placed upon it. There are wide usage of terms which express the quality of measurement and only fair agreement to term definition. This report will use terms which have been defined and accepted within NBS and are recommended for general usage.

With these general thoughts in mind, it is possible to proceed to define these terms and quantities as they will be used in this report.

Consider first the concept of experimental error.

"From a statistical viewpoint, the measurement may be considered as a process operating on a physical system. The outcome of the measuring process is a number or an ordered set of numbers The process is influenced by environmental conditions, the variations of which constitute the major cause for the uncertainty known as experimental error." [Mandel, 1967]

"The desire to obtain simple measures of experimental error has led to two concepts: precision and accuracy. When properly defined these concepts are extremely useful for the expression of the uncertainty of measurement." [Mandel, 1967]

The precision of a measurement process, in this case flow measurement, is the degree of mutual agreement characterized by independent measurements of a single quantity, total flow or flow rate, found from repeated applications of the measurement process under specified conditions. Accuracy of the flow measurement process on the other hand is the degree of conformity of such measurements to the true value. Therefore accuracy has to do with closeness to the truth while precision only closeness together [Eisenhart, 1962]. These definitions will not change the common practice of using the terms interchangeably but should be of use in understanding of the data and evaluations contained in this report.

Three additional terms must also be defined and related to the measurement process. When the limiting mean value of a measurement of flow accomplished by a particular measurement process does not agree with the true value, the measurement process is said to have a systematic error or bias.

The final term definition results from the abuse of previously defined terms for precision and accuracy and is generally used when considering reference values or standards.

When calibrating or evaluating the performance of a flowmeter in comparison to a reference or standard, the investigator needs an estimate of the total uncertainty of the reference or standard. This total uncertainty is given as an estimated value of the sum of the allowance for known sources of systematic error and an allowance for random error (precision). The allowance for systematic errors is in reality the limits to a bias correction while the allowance for random error must include a confidence level.

An example of an uncertainty statement is taken from Dean, et al., (1971) and describes the cryogenic nitrogen flow facility at NBS, Boulder, Colorado: "... the uncertainty of the measurement of totalized mass flow is estimated to be ±0.18 percent. This figure includes an uncertainty of ±0.12 percent for known sources of systematic errors plus an uncertainty of ±0.06 percent for random error. The estimated uncertainty due to the random error is three times the standard deviation calculated from 23 applications of the calibrated masses over a period of three months."

2.2. Flowmeter Calibration

The ability of any flow measurement device to measure flow rate (or total flow) is generally defined in reference to the base units of length (volume), mass and time. A calibration would then compare the performance of the metering device to that of the reference system or standard over a certain defined set of operating conditions. Flow reference standards can be of several different types or combinations such as volume-time, mass-time, static, dynamic, continuous or intermittent in operation. In general the flow standard for gases is volumetric while for liquids it would be on a mass basis.

The Fluid Meters Section of NBS maintains a number of flow systems for both liquids and gases and provides calibration and test services limited generally to water, liquid hydrocarbons and air as the calibration fluid [Mason, 1968]. The total uncertainties of these calibrations are also available [Wildhack, Powell, and Mason, 1965]. A more recent description of the NBS Fluid Meters calibration capabilities is given by Ruegg and Shafer (1970). Figure 2.1 is taken from this publication and describes the range and estimated systematic error.

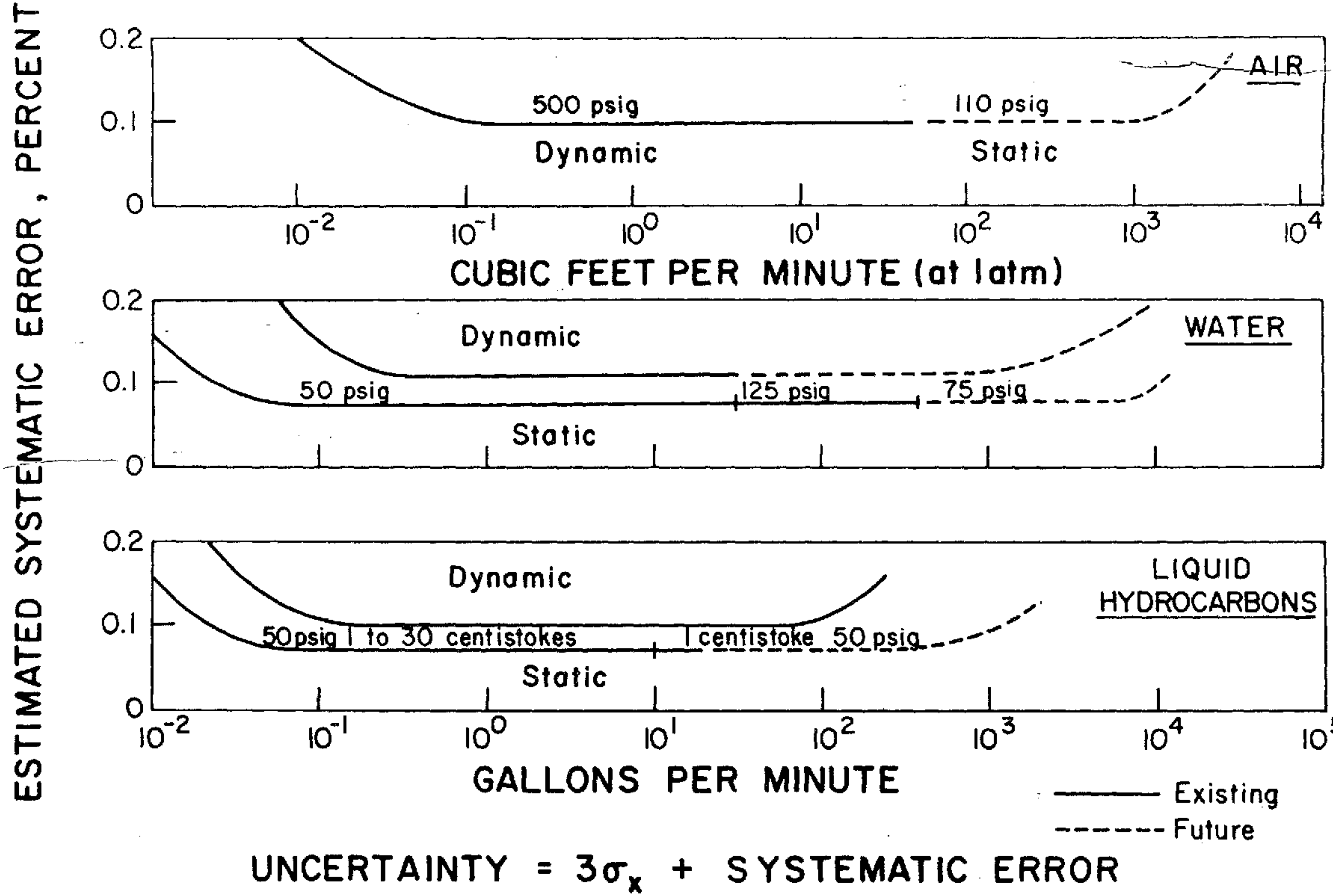


Figure 2.1. Calibration Facility Capabilities - Ambient Temperature - Fluid Meters Section, NBS. [Ruegg and Shafer, 1970]

These facilities are not always useable if a calibration is required for a specific fluid other than water, air or hydrocarbons. In the case of GOX this is not a particular problem as the characteristics of metering GOX at ambient temperatures are similar to air.

LO_2 is a different story. Very few if any flow reference systems use LO_2 as the process fluid primarily because of the hazards involved. Certain dedicated systems such as rocket engine test stands or other similar systems were modified to perform as a flow reference standard when it was necessary to calibrate one type of flowmeter.

To describe each of the reference systems used or referred to in the literature is not within the scope of this report and may, in any event, be impossible as some of the installations no longer exist and were inadequately documented at the time of use. As a result, meter performance information is suspect in all but a few cases as the meter performance is directly dependent on the comparison with the reference or standard. In the following sections which describe specific meters every attempt is made to define the methods and basis of calibration, but as pointed out earlier the description of the reference system is generally inadequate.

The establishment and maintenance of a flow measurement standard or reference must involve continuing use and analysis of the device or facility as a measurement system. This means repeated measurements of the flow and comparison with the basic standards upon which the flow reference system is based. These repeated observations will of course indicate consistency and provide confidence both for the operator of the facility and the ultimate user of the calibration. Of equal or greater importance, these repeated measurements will provide data upon which decisions can be made mathematically to determine if the observations are truly a random sample of the total population of observations.

The NBS Cryogenic Flow Research Facility described in the appendix indicates one method of establishing and maintaining a flow standard or reference system. The joint NBS-Compressed Gas Association program provides and documents in the open literature most of those elements necessary for a well developed measurement system. These elements can be outlined as follows:

1. Broad based requirement or need for such a facility and program.[ISA, 1967]
2. Fiscal and technical support from the ultimate beneficiaries. [Mann, 1971]
3. Documentation of facility design after full study of the area of impact and recognition of previous or related state of the art. [Dean, et al., 1968]
4. Full disclosure of construction and fabrication techniques including all new or innovative sub systems designed to improve or facilitate measurement technology. [Mann, et al., 1970]
5. Proving of the flow system and full documentation of all calculations, experiments, operating procedures, sources and values of experimental error. [Dean, et al., 1971]
6. Evaluation of generic classes of flowmetering devices over a broad range of expected operation conditions. [Brennan, et al., 1971; Brennan, et al., 1972; Brennan, et al., 1973]
7. Establishment of specifications, tolerances and recommended practices for flow devices used in commerce. [NBS, 1971]

8. Interlaboratory comparison of the facility capability to make and maintain flow measurements. [Dean, et al., 1971]
9. Promulgation of transfer standards traceable to the NBS for the field certification of new meters to put in service or recertification of in-service meters [in progress].

The restrictions to such a program are also apparent from the documentation. Only moderate flow rates are available (20-200 gpm), only one fluid is used as a process fluid (liquid nitrogen) and the program structure was designed primarily to evaluate existing meters and systems rather than for development of new measurement devices based on the most recent scientific discoveries.

It is interesting to note that a potentially equal or more effective program for cryogenic flowmetering was suggested but not acted upon in the early 1960's. A study and proposal [NBS, 1963] was made by NBS in response to a request by NASA. General specifications for this facility included provision for liquid hydrogen, liquid nitrogen and water at flow rates of up to 600 lbs/s of liquid hydrogen with a flow range of 100 to 1. The system was a "blow down" system where liquid was stored in one cryogenic container and forced under pressure through the meter into a second catch tank. The anticipated total uncertainty of flow rate was initially 0.2 percent with a design goal of 0.1 percent or better. The reference system was gravimetric and would have provided wide flexibility in the evaluation of high flow rates. Liquid oxygen was not to be included in the initial configuration but provision was made for adding this capability at a later date.

The consequences of not providing such a general type cryogenic reference system are already apparent and will become more so in the near future. Recent NASA requirements have resulted in studies of the availability of cryogenic (liquid or cryogenic gaseous oxygen and hydrogen) flowmeters having mass flow measurement uncertainties of one percent or less [Hayakawa, et al., 1972]. Of sixteen candidate systems two were chosen as "promising" with only one of these two having "some data at cryogenic gas temperature." A continuing program such as that proposed could have provided NASA with a catalog of flow instrumentation with established performance at cryogenic temperatures.

The proposed cryogenic flowmeter system would also have provided assistance in the current nation wide energy "crisis." As a short term (10-20 year) program to minimize energy shortages, natural gas is being liquefied and imported to populous East and West coast cities. Liquefied natural gas (LNG) is a cryogenic fluid (111 to 140 K) and the purchase from foreign governments involves economic as well as political decisions. Quantity measurements are necessary to assure fair value received and present methods of using the ship as a reference volume are inadequate [Mann, et al., 1973].

The proposed facility would have provided invaluable data, expertise, and experimental facilities for the selection of measurement systems to be used in purchasing of possibly up to 20 percent of our natural gas requirements in 1980.

2.3. Gaseous Oxygen at Ambient Temperatures

The characteristics, performance, precision and accuracy of flow measurement devices for gaseous oxygen at ambient temperatures and high pressures have not been identified by the present literature search. This situation is believed to be caused by the fact that oxygen gas at ambient temperatures is not considered unique enough to warrant specific attention. Except for material compatibility problems (which are not unique to flow instrumentation), the state of the art of measurement of flow is described in a number of publications. One of the most comprehensive is the report of the American Society of Mechanical Engineers Research Committee on Fluid Meters [ASME, 1971]. This report includes most if not all general usage types of meters, and details the theory of operation and recommended practice. Chapter I-5 (Appendix A), on differential pressure meters, is most applicable to ambient temperature oxygen flow.

2.4. Surrogate Fluids

In addition to a lack of documentation of the calibration process we are forced also to consider the problem of surrogate fluids. Many of the flow facilities set up to establish the performance of flow measuring devices for oxygen have been constructed for the expressed purpose of defining the ability to make the measurement on water and to transfer the calibration to liquid oxygen [Bucknell, 1962; Deppe and Dow, 1962]. Their findings indicate that this is not a feasible procedure if accuracy and precision is required at a level better than ±2 percent, although Tantam, et al. (1960) believe water calibrations are valid for positive displacement meters. This, of course, does not answer the question: if water cannot be used to provide accurate calibrations of meters designed for service with liquid oxygen, what fluids can be used? From the publications so far reviewed (see bibliography), there is not one case of a direct meter comparison of liquid oxygen to any other similar cryogenic fluid. Liquid nitrogen has been used as a surrogate fluid for liquid oxygen meters primarily because the process may not result in errors appreciably greater than the error of the measurement process itself.

To establish this comparison, selected properties of cryogenic fluids and water are presented in table 2.1. It is immediately evident from this table that water may not be a good candidate for a surrogate fluid. The table also indicates that the fluids oxygen, argon, and nitrogen should be, on the basis of the properties compared, interchangeable fluids in the calibration process. For lack of evidence to the contrary, performance cited in this report is based on this assumption.

The objective of providing a "state of the art" review of oxygen flowmeter instrumentation has not been completely met. This may well be caused by the fact that precise flow measurement is more "art" than technology and technology is generally the content of most publications. In the absence of a clear technology the author has included extensive quotations from the literature cited in an attempt to provide the "art" as well as the science of oxygen flow measurement.

3. QUANTITY METERS

Quantity meters involve the concept of a summing up of a number of individual measurements to a total fixed quantity. These types of meters are also known as totalizing meters and are favored in commerce over "rate" meters. It is believed that the quantity meter will provide a more accurate measure of an amount of product bought or sold [ASME, 1971].

At the present time most commerce in cryogenic liquids is performed using quantity type positive displacement volumetric meters or total truck weighing. The latter, placing the truck on a commercial or test scale before and after delivery, is also a quantity type of metering but based on weight rather than volume. Truck weight is found to be less satisfactory because of the large quantities necessary to maintain accuracy and precision and the growing requirement for multiple small deliveries. The literature has no detailed assessment of the measurement process as related to truck weight deliveries but indicates significant errors caused by wind, snow, ice and other environmental conditions [Fox, 1967].

Table 2.1. Selected Properties of Cryogenic Fluids* and Water

	Water	Oxygen	Argon	Nitrogen	Hydrogen
Temperature (K)	300	90.18*	87.28*	77.37*	20.27*
Density (g/cm^3)	1.0	1.1407*	1.394*	0.807*	0.071*
Liquid Dynamic Viscosity (Poise)	0.01002	0.0019	0.0026	0.0015	0.00013
Liquid Kinematic Viscosity (cm^2/s)	0.01002	0.00166	0.00187	0.00186	0.00183
Reynolds Number** (2.54 cm. Diam. - 100 gpm)	0.316×10^6	1.91×10^6	1.73×10^6	1.70×10^6	1.75×10^6
Thermal Contraction*** $\frac{L(293\ K) - L(T)}{L(293\ K)} \times 10^5$ [304 SS]	0.0	270	272	280	296

* Normal Boiling Point [Roder, 1973].

** Dimensionless parameter - $\frac{\text{(density)(fluid velocity)(pipe diameter)}}{\text{(dynamic viscosity)}}$

*** Where L is length.

Table 2.2. Selected Properties of Oxygen and other Atmospheric Gases*

	Air	Oxygen	Argon	Nitrogen	Hydrogen
Temperature (K)**	300	300	300	300	300
Density (gm/ℓ)*	1.177	1.301	1.624	1.138	0.082
Dynamic Viscosity (poise $\times 10^{-6}$)	184.6	206.3	229.4	178.6	89.59
Kinematic Viscosity (cm^2/s)	0.157	0.159	0.141	0.157	1.093
Reynolds Number*** (2.54 cm diam. - 100 scfm)	1.51×10^5	1.48×10^5	1.68×10^5	1.51×10^5	0.216×10^5

* Hilsenrath, J., et al., 1955.

** Pressure @ 1 atmosphere ($1.01325 \times 10^5\ N/m^2$)

*** Dimensionless parameter - $\frac{\text{(density)(fluid velocity)(pipe diameter)}}{\text{(dynamic viscosity)}}$.

The following sections describe quantity type positive displacement volumetric meters currently used in commerce.

3.1. Screw Impeller Volumetric Flowmeter

This meter, in several configurations, is widely used in commerce of liquid oxygen, liquid nitrogen, and liquid argon. Five references from the literature cite performance, design, or usage information. Crawford (1963) discusses materials of construction and design as does Close (1968) and Close (1969). Fox (1967) refers to the screw impeller meter as a method of cryogenic flow measurement for custody transfer but gives no operational performance or design data. The most extensive evaluation was performed by Brennan, Dean, Mann and Kneebone (1971) and provides performance information for several configurations on liquid nitrogen.

Physical Principle of Measurement. Operation of this meter can be described by reference to figure 3.1. This shows the electric counter configuration [Williams, 1963]. There also is a meter fitted with a mechanical counter as shown in figure 3.2, but the operating principle of the primary elements are the same. The meter consists of a central screw type impeller that meshes with two deeply grooved sealing screws, all selectively matched for close tolerances and fitted closely into a surrounding metal housing. The special contour of the screws provides a line seal where the rotors mesh. See figure 3.1. Liquid entering the meter causes rotation of the impellers, and the displacement obtained is transmitted through a gear system either to an electronic counter or to a totalizing register. Meters are provided with a cooldown method composed of a priming line which allows circulation of liquid through the meter without causing operation of the impellers.

Design. The meters described by Crawford (1963) and Close (1968), (1969) are the same in design and the following description is provided by Crawford (1963):

"The rotors are made of hard, graphitic carbon. For liquid nitrogen and argon service, the rotors are impregnated with wax, because, although liquid oxygen provides adequate lubrication in contact with carbon, liquid nitrogen and argon do not provide the same film lubrication properties. Bearings of carbon and polytetrafluoroethylene are used. Gears are brass; although, in some meters nylon-brass gear pairs have given superior service. Materials for liquid oxygen meters are tested for combustibility prior to use, and care is exercised in manufacturing and assembly to assure cleanliness and freedom from combustible contaminants. No lubricant other than the wax treatment of the rotors is used in any of these meters. A screen is provided on the inlet to the measuring chamber to prevent entrance of foreign material."

The meter body or case generally is made of brass or bronze.

Two methods of counting the revolutions of the primary element are used. In the first, illustrated in figure 3.1, the rotating motion is geared to an electrical pulse transmitter. Ten magnets are evenly spaced around and attached to a rotating shaft. A stationary reed switch attached to the meter housing is activated as the magnets move past the reed. The switch closure transmits an impulse to a magnetic impulse counter.

The mechanical register is illustrated in figure 3.2. The counter operates warm and is separated from the liquid cryogenic temperature components by a stainless steel tube. The drive shaft

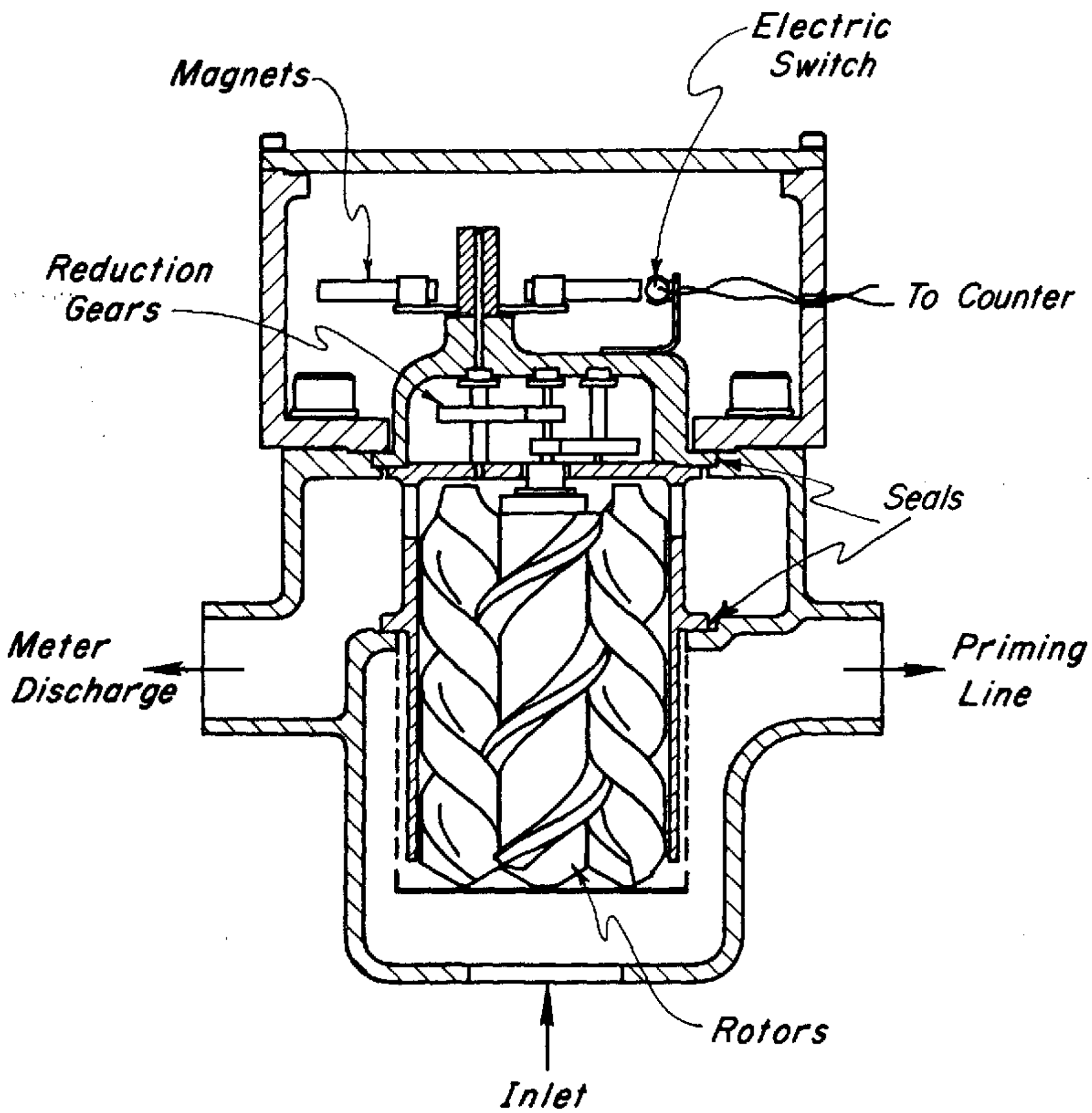


Figure 3. 1 Screw Impeller Meter with an Electric Counter.

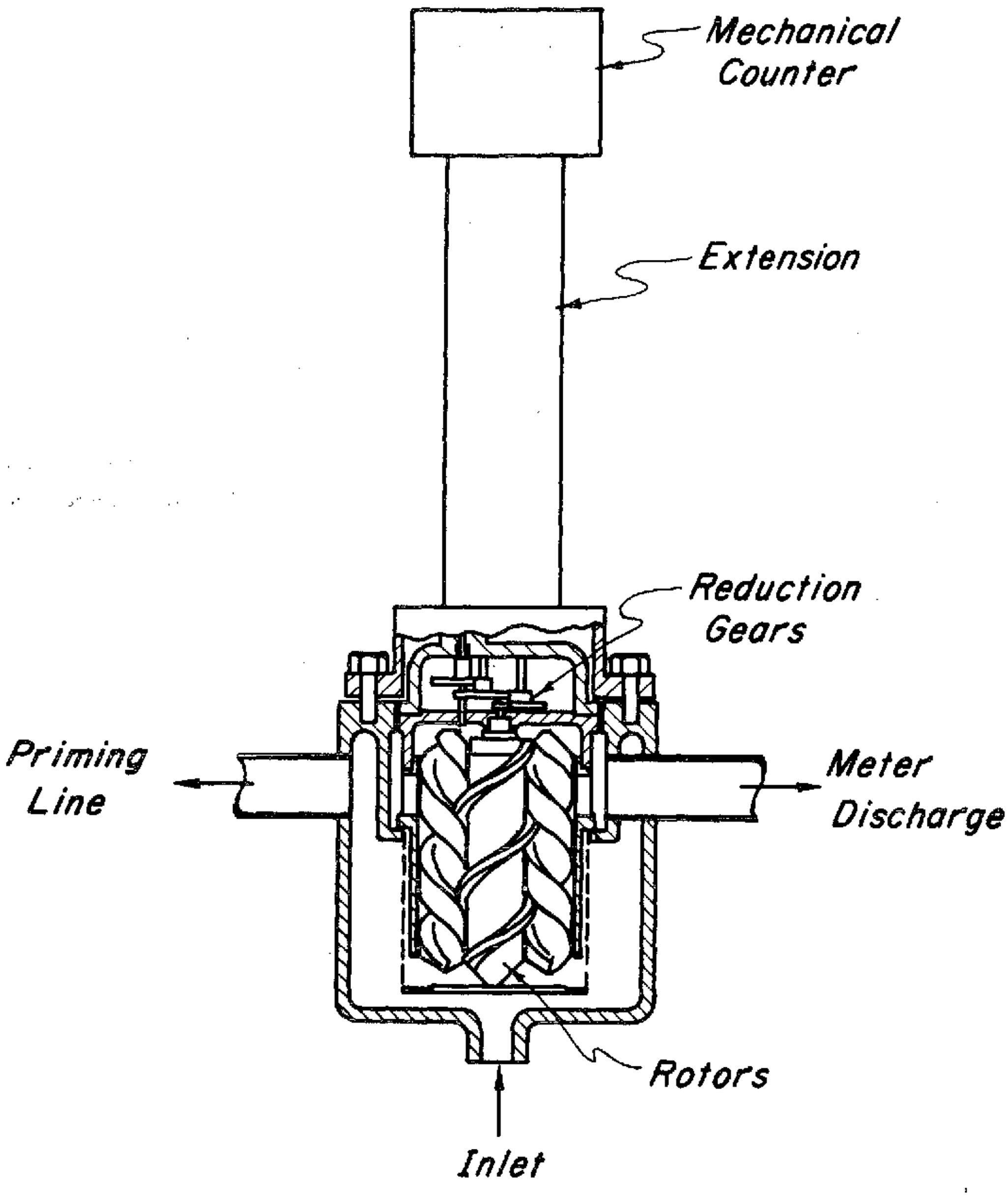


Figure 3. 2 Screw Impeller Meter with a Mechanical Counter.

is sealed at the warm end with an "O" ring seal. The counter is sealed and a vent provided to permit exit of any shaft leakage while also preventing entry of moisture. Counter covers are color coded according to the intended cryogenic service. This is a safety measure, because oxygen meters require special materials and cleaning procedures [Bankaitis, et al., 1972]. The differentiation is also needed because the gearing in each meter differs to provide readings in terms of cubic feet of equivalent gas at NTP*. According to Crawford (1963), the one-inch meter has a capacity of 3.5 to 50 gallons per minute at pressures up to 350 psig.

Performance Characteristics. There is no detailed, published information on the performance of this meter other than that of Brennan, et al. (1971). Crawford (1963) indicated that the meters in commercial service are calibrated in a closed liquid oxygen weighing system accurate to within ±0.1 percent. He further indicates that these meters are calibrated routinely on an annual basis. No further information is available to establish the creditability of the liquid oxygen flow proving system. Crawford does indicate "some meters in continuous service will pass more than 15 million gallons without having the accuracy fall outside of the ±1 percent tolerance."

Close (1969) describes the liquid nitrogen cryogenic flowmetering calibration system of the Linde Division of Union Carbide located at Tonawanda, NY. Close indicates that "total system uncertainty is less than 0.25 percent for volumetric meters and less than ±0.1 percent for mass flowmeters. Mass flowmeter calibration error is nominally the scale uncertainty." With this prover as a basis, the following table is extracted from Close (1969).

Table 3.1. Screw Impeller Meter Performance [Close, 1969]

	Screw Impeller Mechanical Register
Error with Calibration, % of Point	± 1
Repeatability, % of Point	± 0.5
Range	10:1 (50 gpm - LO_2)
Typical Flow Range ΔP, psi	4.5 - LO_2
Meter Back Pressure*	4:1
Over Range Protection	Recommended
Applicable Flow Conditions	Variable, Pulsating
Straight Pipe Requirement	None

$$^{*}\text{Meter back pressure} = \frac{\text{Inlet Pressure - Vapor Pressure}}{\Delta P} : 1$$

The performance of this type of meter as reported by Brennan, et al. (1971) was assessed using the flow prover located at NBS-Boulder, Colorado. The description of this flow prover and test procedures is provided in Appendix B and a provisional accuracy statement for this facility is given in Dean, Brennan, Mann, and Kneebone (1971).

*Normal temperature and pressure (70°F and 1 atmosphere).

A total of five screw impeller meters were evaluated. Meters evaluated included those from two different manufacturers and in two configurations, one with electric register and the other with mechanical register. Table 3.2 is extracted from Brennan, et al. (1971) and is a performance summary.

Table 3.2. Screw Impeller Meter Performance

Meter Type	Screw Impeller Electrical Register	Screw Impeller Mechanical Register
Precision (3σ) at Start, %	$\pm$ 0.66	$\pm$ 0.33
Precision (3σ) at End, %	$\pm$ 0.51	$\pm$ 0.48
Bias at Start, %	+ 1.17	+ 0.28
Bias at End, %	+ 0.55	+ 0.50
Volume Metered, m^3 (gal)	1249 (330,000)	870 (230,000)
Maximum Flow Rate, m^3/s (gpm)	0.0025 (40)	0.0032 (50)
Minimum Subcooling, K	5	5*

*Cavitation occurred at 4 K.

The data included in the table are for one meter of each type. The pressure drop for the meter with the electric counter is given in figure 3.3, while the pressure drop for the meter with the mechanical register is given in figure 3.4.

<u>Summary and Conclusions</u>. A screw impeller type volumetric flowmeter has been used extensively in the metering of liquid oxygen, liquid nitrogen, and liquid argon. The flow rates are relatively low (up to 50 gallons per minute) and the meter is primarily used for trailer truck dispensing of fluids. According to the manufacturer the meter materials are carefully selected for the service aniticpated and meters are recalibrated on an annual basis.

The following quotation from Brennan, et al. (1971) has a bearing on the methodology of use of this meter in service. "Some meters are installed on tank trucks with permanent piping into the meters. Often the meter is silver soldered or welded in place. Thus, the entire meter is not conveniently removable as a unit. When meter service is required, the register and positive displacement metering elements are unbolted and removed from the bowl or shell that remains attached to the piping on the truck (see figures 3.1 and 3.2).

The positive displacement element is repaired and may be recalibrated on a test stand using another bowl. After recalibration, the meter is placed in stock for eventual use with still another bowl. This approach is convenient from the viewpoint of servicing meters but does not allow optimum meter accuracy.

The above type of meter operation assumes the interchangeability of meter positive displacement elements and meter bowls. Our experience is that operation of the same positive displacement element in different bowls can yield a different average value or bias. This is caused by a variability of seal performance between different combinations of positive displacement elements in bowls. Tightening the bolts on the top of the bowl makes both the top seal (see figure 3.1), that prevents liquid from leaking around the counter support, and the bottom seal, that is meant to prevent liquid from bypassing the meter element. There is no way to check directly the bottom seal for leakage. We have

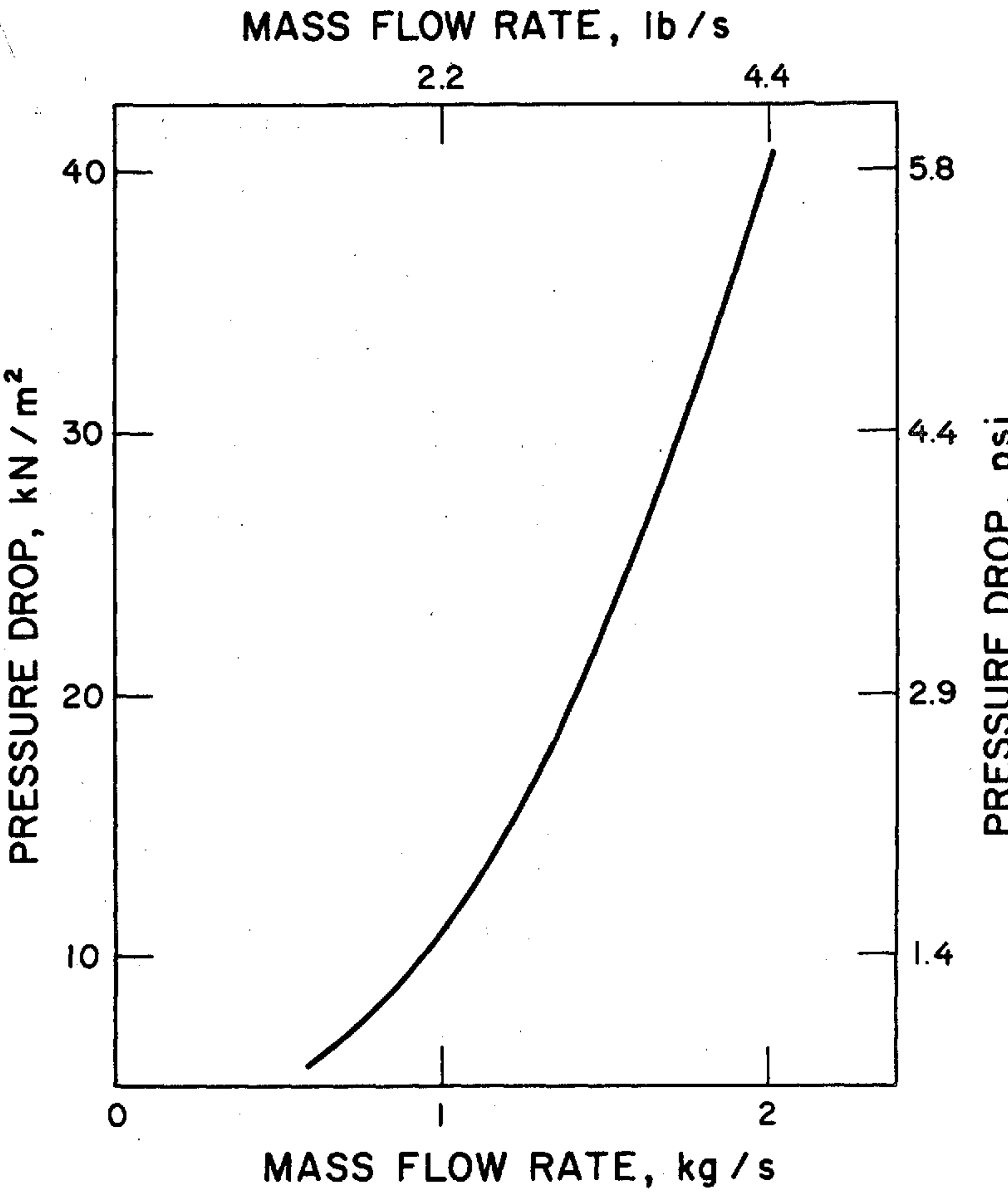


Figure 3.3 Pressure Drop - Screw Impeller Meter with Electric Counter - LN_2.

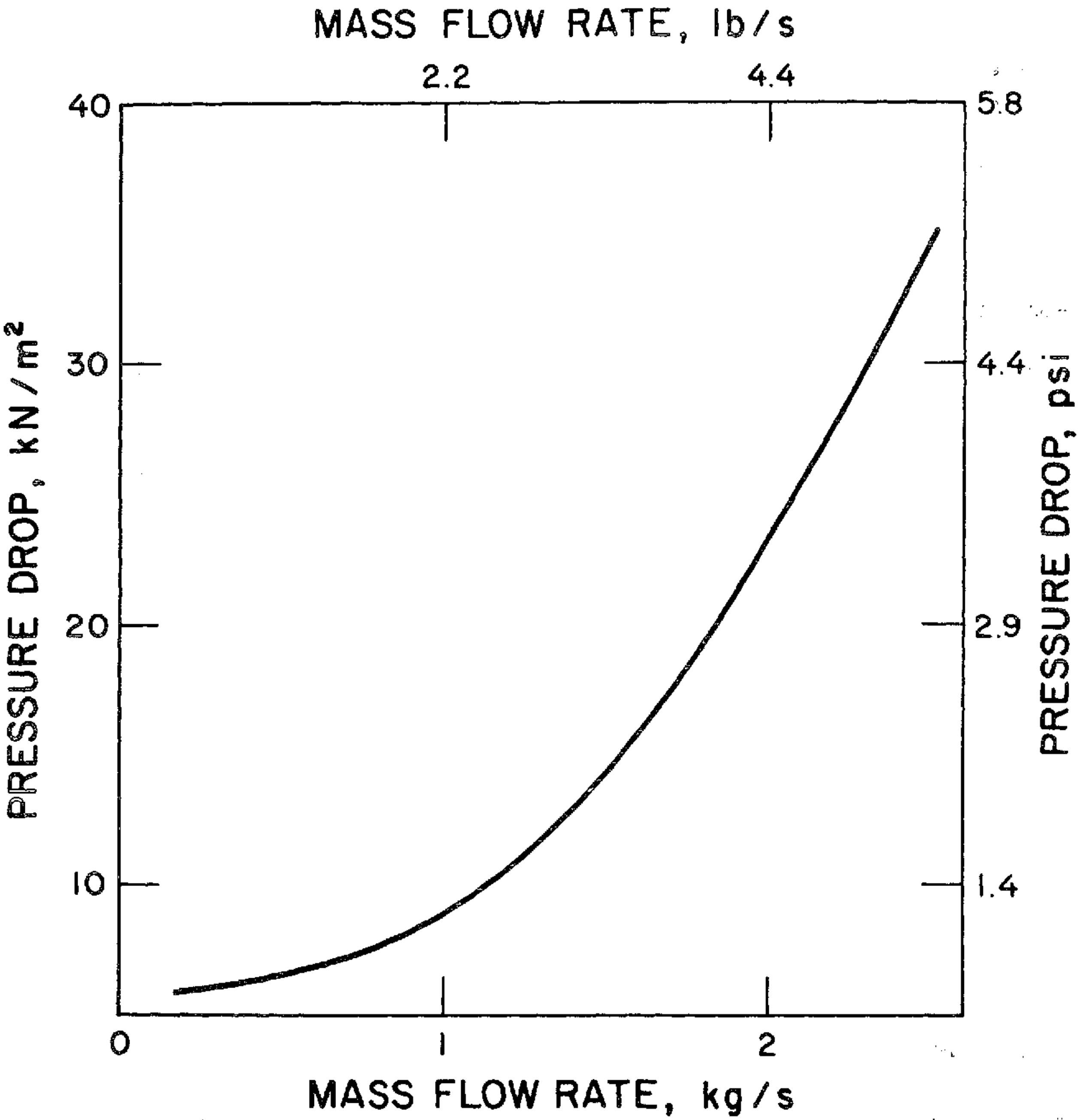


Figure 3.4 Pressure Drop - Screw Impeller Meter with Mechanical Counter - LN_2.

observed differences in biases of 1/2-percent when operating the same positive displacement element in different bowls. Leakage through this seal will result in underregistration."

3.2 Rotating Vane Volumetric Flowmeter

Only one reference could be found for the operation and performance of this meter at cryogenic temperatures operating on liquid oxygen, nitrogen, or argon. Brennan, Dean, Mann and Kneebone (1971) included this meter in their evaluation of positive displacement flowmeters currently used in commerce. The inclusion in the joint NBS-Compressed Gas Association program was based on demonstrated, although unpublished, use of the meter for custody transfer of liquid oxygen, liquid nitrogen, and liquid argon.

Physical Principle of Measurement. Operation of the meter can be shown by reference to figure 3.5. With the inlet outlet configuration as noted, the dumbbell-shaped element rotates in a clockwise mode, while the two vanes rotate in a counter clockwise mode. The result is a measured quantity of fluid entrapped between dumbbell element and vanes and propelled through the meter by means of the energy associated with the flowing fluid. The meters of this type evaluated by Brennan, et al. (1971) consisted of a basic metering element (cross-hatched section of figure 3.5) and various configurations of a mechanical counter and extension. The extensions and mechanical counters were made interchangeable to accommodate the three fluids: liquid oxygen, liquid nitrogen, and liquid argon. Each fluid, of course, required a different gearing between primary element and the mechanical counter, based on density of the fluid being measured.

Design. According to the meter manufacturer, the rotors, bearing plates, and housing are all made of 356-T6 aluminum alloy. The meter suppliers specifications are:

fluid - liquid oxygen
maximum flow rate - 0.0063 m^3/s (100 gal/min)
minimum flow rate - 0.0013 m^3/s (20 gal min)
maximum pressure - 2.068 MN/m^2 (300 psia)
material - aluminum
register - mechanical or combined mechanical and electrical.

This type of meter is available with a vapor eliminator and an automatic temperature compensator. Neither of these features were evaluated nor is performance information available from the manufacturer.

Some difficulty was encountered during the evaluation of this meter by Brennan, et al. (1971) caused by malfunction of the register gearing which is normally attached to the primary element and driven through a shaft and gearing arrangement. Binding of the register drive shaft caused excessive load on the primary element gearing. This high loading caused one of the meters to fail, with the rotor coming into contact with the meter case after 113.5 m^3 (30,000 gal) of liquid were metered. A simplified mechanical register was attached to a new meter body, and all testing was performed in this configuration.

Performance Characteristics. The performance of this type of meter as reported by Brennan, et al. (1971) was assessed using the flow prover located at NBS, Boulder, Colorado. The description of this flow prover and test procedures are given in Appendix B.

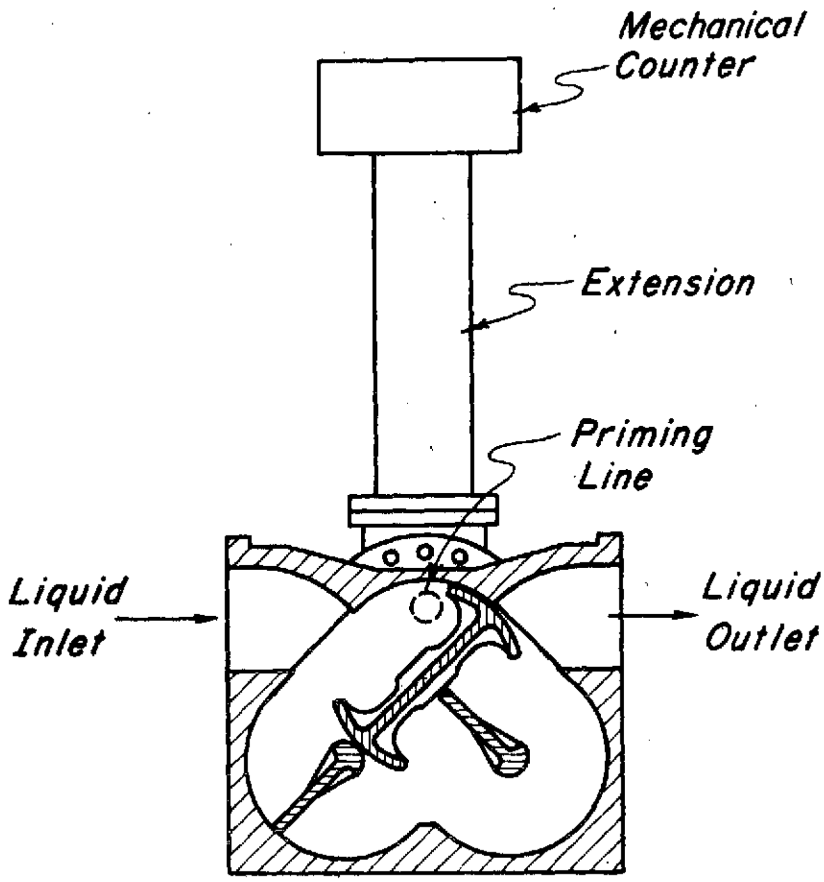


Figure 3.5. Rotating Vane Meter.

Two primary measuring elements were evaluated under this program. The first meter failed during tests as indicated above and all data were accumulated on the second test meter. The following table extracted from Brennan, et al. (1971) indicates the meter performance.

Table 3.3. Rotating Vane Meter Performance

	Rotating Vane
Precision (3σ) at Start, %	$\pm$ 0.69
Precision (3σ) at End, %	$\pm$ 2.1
Bias at Start, %	+ 1.04
Bias at End, %	- 4.8
Volume Metered, m^3 (gal)	1333 (352,219)
Maximum Flow Rate, m^3/s (gpm)	0.0063 (100)
Minimum Subcooling, K	3

The pressure drop for the meter is given in figure 3.6.

Summary and Conclusions. This meter was only recently adapted to cryogenic service. The manufacturer indicated that previous to this adaptation it had been used extensively in flow measurement of ambient temperature commercial fluids. The performance of the meter suffered from this lack of experience in cryogenic service. The large shift in bias as shown in the table indicates that the meter was subject to high wear. It was believed that this wear was primarily caused by the high loading of the mechanical register. This excessive wear is also reflected in the decrease of precision at the end of the testing. Additional design modification based on continued cryogenic service should result in a meter adequate for liquid oxygen, liquid nitrogen, and argon service.

3.3. Oscillating Piston Volumetric Flowmeter

This meter is a true quantity meter under the criteria established for classification of meters. It is also referred to as a ring piston meter or a rotary piston meter. There are two references in the literature which give actual operating performance of this meter in cryogenic service. The work of Tantam (1960) describes the operation of an oscillating piston meter in commercial service for the volumetric flow measurement of liquid oxygen. The work of Brennan, et al. (1971) provides detailed performance information on the oscillating piston meter operating on liquid nitrogen. A third reference, Angerhofer (1965), gives some additional information which is not significantly different from the first two references.

Physical Principle of Measurement. The physical principle of operation is shown in figure 3.7. Liquid is admitted to the inlet and is first taken out the priming line to cool the meter to operating temperature. After priming, liquid flows through the piston assembly. Liquid enters the piston assembly through the inlet port and displaces the piston horizontally around the vertical axis. The piston is kept from turning by a slot that rides on a plate. The resulting motion is an oscillation that is geared to a mechanical register. Liquid passes out the discharge port to the meter outlet. The sealing of the measuring chamber depends on sliding contact between the cylinder base (in the plain of the paper) and the lower open end of the piston, and on a combination of rolling and sliding line contact between the cylinder and piston side walls.

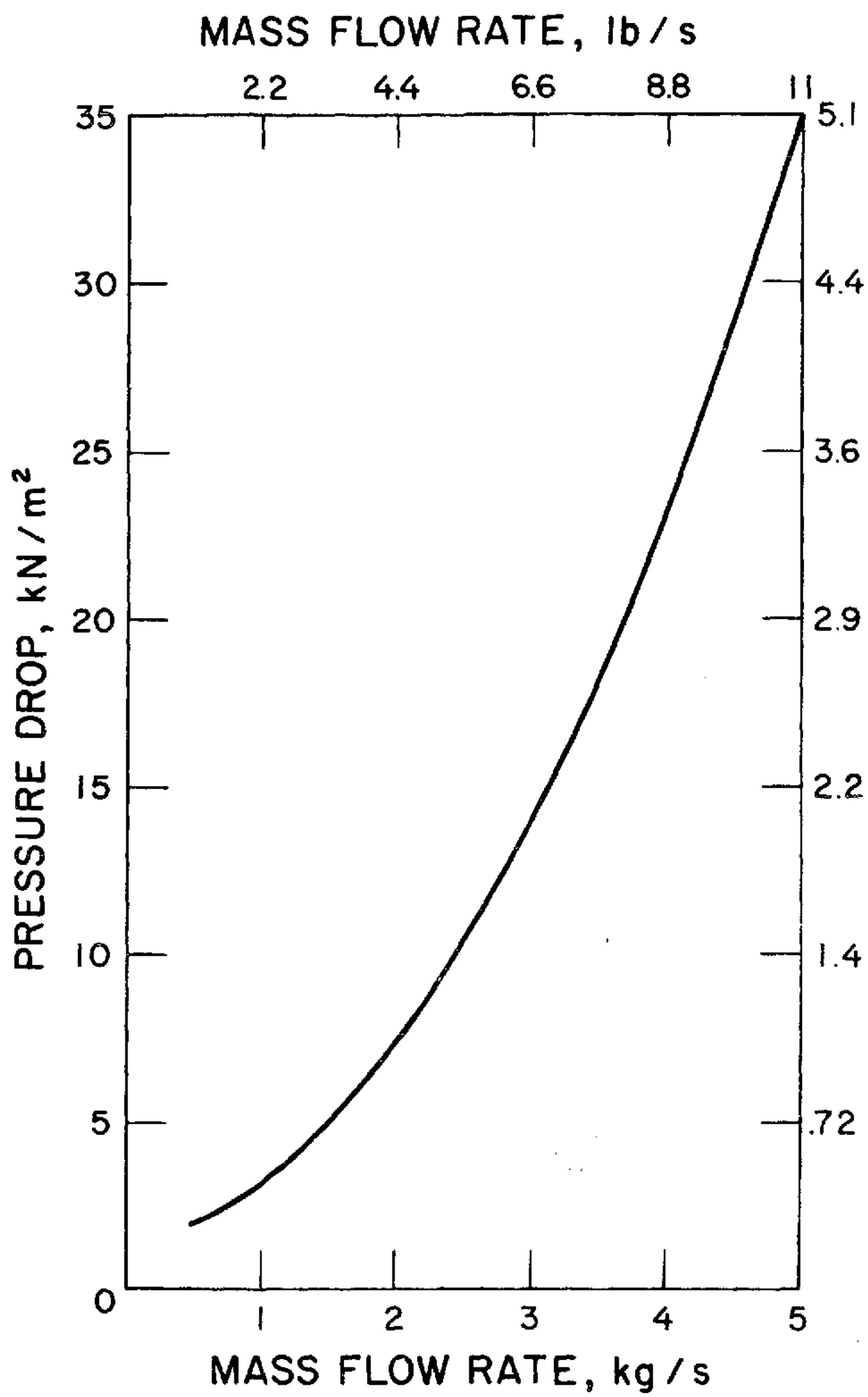


Figure 3.6 Pressure Drop - Rotating Vane Meter.

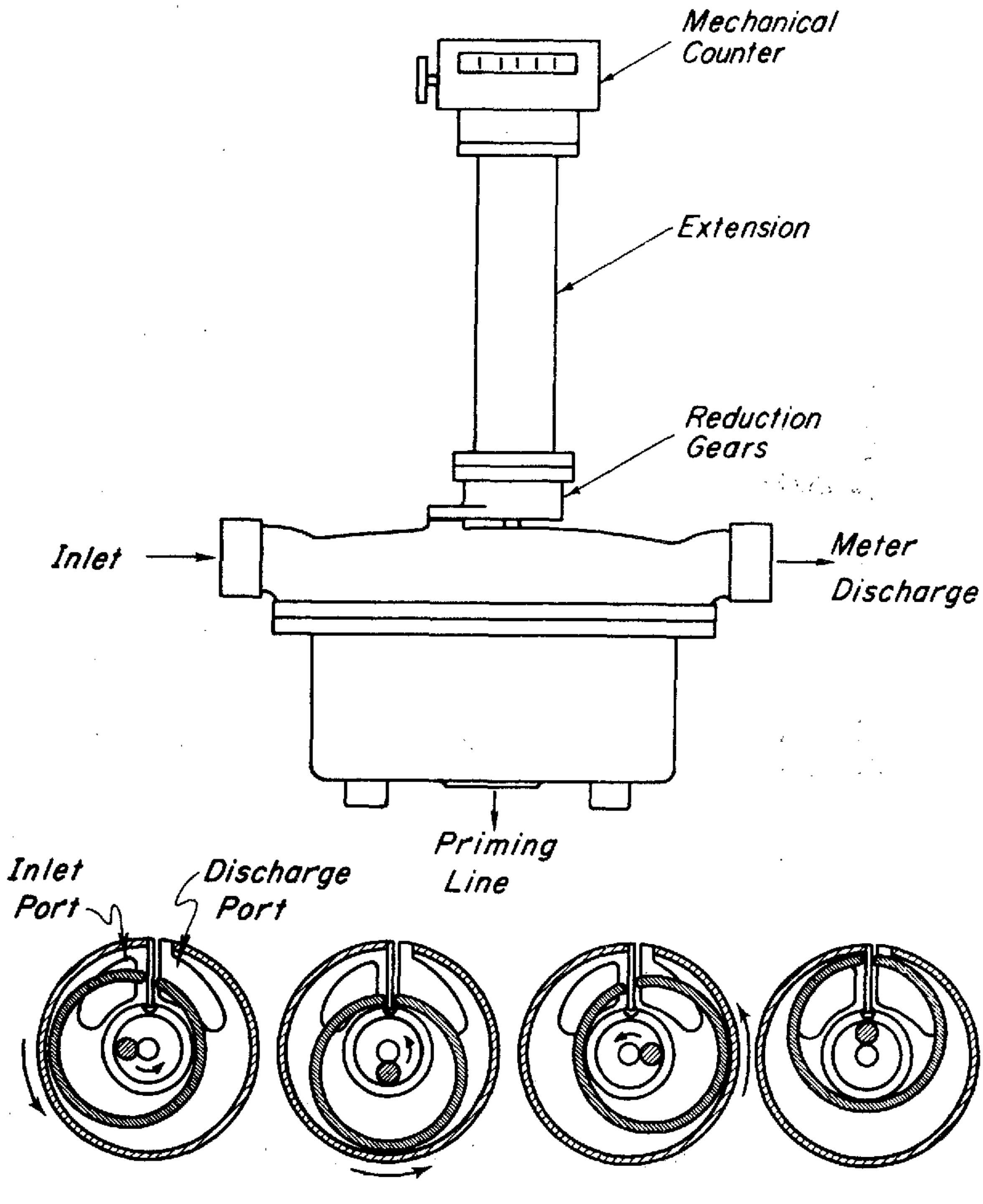


Figure 3. 7 Oscillating Piston Meter.

Design. The meter of Tantam (1960) describes the piston, the center guide ring which bears against the piston spindle and the division plate as constructed from polytetrafluoroethylene (PTFE) impregnated bronze. The density of this porous bronze material was 6 grams per cubic centimeter, allowing an impregnation with PTFE to 25 percent of the total volume. The pore size of the bronze is 6-15 microns. A thin surface coating of 0.0005 inches of PTFE was applied to the pistons to assist in initial seating of the piston. No material specification was given for the meter body.

Tantam indicates two possible methods of monitoring the output of the driven shaft. The first is a direct drive to mechanical counter with a 13 inch extension member to allow the counter to operate at ambient conditions. The second method is an electronic pickup, which was designed to reduce to a minimum the mechanical work required by the metering body, to eliminate the mechanical counter structure which introduces heat leak through the pipe line insulation, and to have the convenience of a remote counting unit.

The pickup probe housing, which contains a permanent magnet and coil, has been inserted through the lid of the meter body and its physical dimensions have been adjusted to make the unit acceptable for all meter sizes. The permanent magnet was fixed to the piston center spindle and revolves under the pickup coil to give the single pulse per revolution required. This small electrical pulse was obtained every time the piston passed beneath the stationary pickup probe, signifying that a discrete volume had been displaced. This pulse was then directed to an electronic counter.

The development program described by Tantam covered four metering sizes: a 1-1/2 inch size for 2500 gallons per hour maximum flow, a 1-inch size for 100 gallons per hour maximum flow (both rated for working pressures up to 300 psig), a 3/4-inch meter for 500 gallons per hour, and a 3/8-inch meter for 300 gallons per hour.

Three meters of this type were also evaluated by the joint NBS-CGA Program. Performance evaluation was reported by Brennan, Dean, Mann, and Kneebone (1971). One of these meters was designated as a 1-inch nominal size meter while the other two were designated as 1-1/2 inch nominal size meters. No specifications were provided with the 1-inch meter; however, it was determined by talking with the meter supplier that the maximum flow rate should be limited to 0.003155 m^3/s (50 gal/min). The name plate specifications on the 1-1/2 inch meter are:

size - 0.0381 meters (1-1/2 inch)
maximum flow rate - 0.0044 m^3/s (70 gal/min)
minimum flow rate - 0.00088 m^3/s (14 gal/min)
maximum pressure - 2.413 MN/m^2 (300 psia)
material - aluminum piston, brass case.

These three meters were fitted with a mechanical counter and extension shaft to allow direct registration of the piston revolutions.

Performance Characteristics. As reported by Tantam, calibrations of his oscillating piston volumetric flowmeter were conducted on liquid oxygen. "The test apparatus used to establish calibrations on liquid gas consisted of a holding tank, a pump, the pipeline containing the meters to be calibrated, a throttling valve, and a receiving tank mounted on scales. The vent gases from the second tank were metered by an orifice and a manometer. Temperatures were measured before each meter and in the gas vent.

Operation involved the pumping of liquid against the pressure preset at the throttling valve to avoid boiling in the meter lines. The vent gas flow was recorded, and the temperatures required for density corrections noted. Calibration runs were carried out with liquid flowing at a constant rate at the beginning and end of runs to insure full pipelines. The flow rate was obtained by timing the runs On the smallest test set up, 175 gallons could be transferred per run with an estimated error of 3/4 percent maximum. A larger apparatus, capable of flow rates up to 3000 gallons per hour of liquid is available, making transfers of 600 gallons possible." [Tantam, 1960]

No other data were provided on the flow prover used by Tantam. The meters themselves were apparently calibrated on water and the following table extracted from Tantum indicates the percent deviation from a standard on water and on liquid oxygen.

Table 3.4. Oscillating Piston Meter Performance

Flow Rate Gallon/h	LO_2 Deviation	Water Deviation	Deviation LO_2 from Water
300	-0.4	-0.53	+0.13
450	-0.55	-0.47	-0.08
800	+0.8	-0.5	+1.3
1700	+0.35	-0.7	+1.05
2100	+0.22	-1.05	+1.27

From this table, the mean deviation of LO_2 from water calibration is equal to 0.73 percent (deviation from standard 14.67 counts per gallon). According to Tantam it is expected that water can be used for the routine testing of meters of this type and a factor applied to obtain the correct liquid oxygen calibration.

Tantam describes a field test of a one-inch diameter built into a pump unit delivering liquid oxygen to a customer plant. In this case the whole vehicle was weighed before and after delivery of the liquid for comparison. He indicates the general result was within the expected ±2 percent. Examination of the one inch meter after the passage of 750,000 gallons of liquid oxygen has shown no significant wear. He further indicates that no electrical fault or mechanical failure has yet been reported from the meters used in the field.

No pressure loss data are given by Tantam other than indicating that such loss with liquid oxygen is similar to that of water.

The performance of this type of meter as reported by Brennan, et al. (1971) was evaluated using the flow prover located at NBS-Boulder, Colorado. A description of this flow prover is provided in Appendix B and the provisional accuracy statement for this facility is given in Dean, Brennan, Mann, and Kneebone (1971).

A summary of the performance of two of the three oscillating piston meters is shown in table 3.5.

Table 3.5. Oscillating Piston Performance

	1-Inch Oscillating Piston	1-1/2 Inch Oscillating Piston
Precision (3σ) at Start, %	± 1.7	± 0.63
Precision (3σ) at End, %	± 1.5	± 0.54
Bias at Start, %	+ 2.2	+ 1.33
Bias at End, %	+ 1.6	+ 0.61
Volume Metered, m^3 (gal)	908 (240,000)	1225 (323,800)
Maximum Flow Rate, m^3/s (gpm)	0.0032 (50)	0.0044 (70)
Minimum Subcooling, K	5	5

Data on only one of the 1-1/2 inch nominal size meters are presented as the second meter underwent a reduction in registration by about 1 percent during these tests and was replaced. The change in bias is significant and indicates wear.

Pressure loss data for these two meters are given in figures 3.8 and 3.9.

Summary and Conclusions. An oscillating piston type volumetric flowmeter has been adapted and modified for service at cryogenic temperatures. The meter has a demonstrated capability of measurement of liquid oxygen and liquid nitrogen. Flow rates are relatively low (up to 70 gallons per minute) and the meter is primarily used for trailer truck dispensing of fluids. Meters evaluated on liquid nitrogen indicated significant wear by a change in bias after the passage of approximately 300,000 gallons of liquid nitrogen, although service on oxygen did not indicate physical wear of the meter parts. The meters themselves are massive and require significant amounts of liquid for cooldown. With adequate procedures, the meters should be capable of making cryogenic flow measurements on oxygen and nitrogen with an uncertainty in the range of 0.5 go 2 percent.

4. HEAD METERS

This type of meter is probably the oldest method of measuring flowing fluids. The following definition of a head meter is drawn from the ASME (1971). The distinctive feature of meters classified as head meters is that means are employed to cause a marked change in the static press[illegible] e fluid while it is passing through the primary element; this pressure change is measured as the difference between the static head and the total head at one section of the channel.

There have been various applications to cryogenics of these head type meters in [illegible] f orifice plates, venturi and flow nozzles. The appeal for the use of these meters in cryo[illegible]enics is more than simplicity and stems from the possibility of eliminating the necessity for calibrati[illegible] oper design and application theory and practice are followed. Design methods for square edge orifi[illegible] ozzles, and venturi tubes are provided in ASME (1971). Recommended practice for flange mounted sharp edge orifice plates can be found in ISA (1970); DIN (1952); and ISO (1967). The application in [illegible] s been to follow these recommendations developed on water, correcting only for thermal contr[illegible] temperature change when operation is desired at cryogenic temperatures. Reynolds num[illegible] eases caused by the low viscosity of cryogens and other differences between water and cryogens have not been considered extensively.

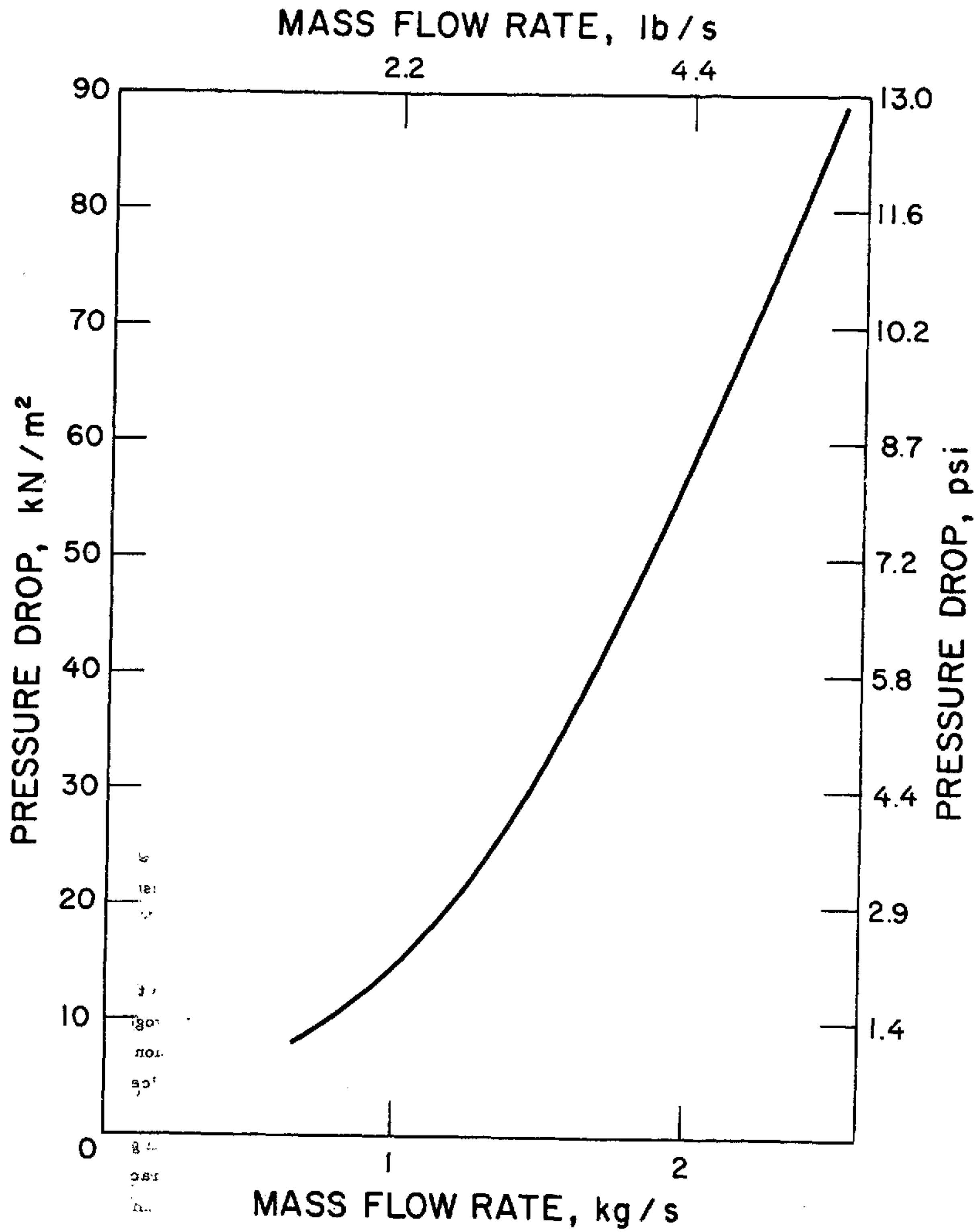


Figure 3.8 Pressure Drop - Oscillating Piston Meter - 1 inch - LN_2.

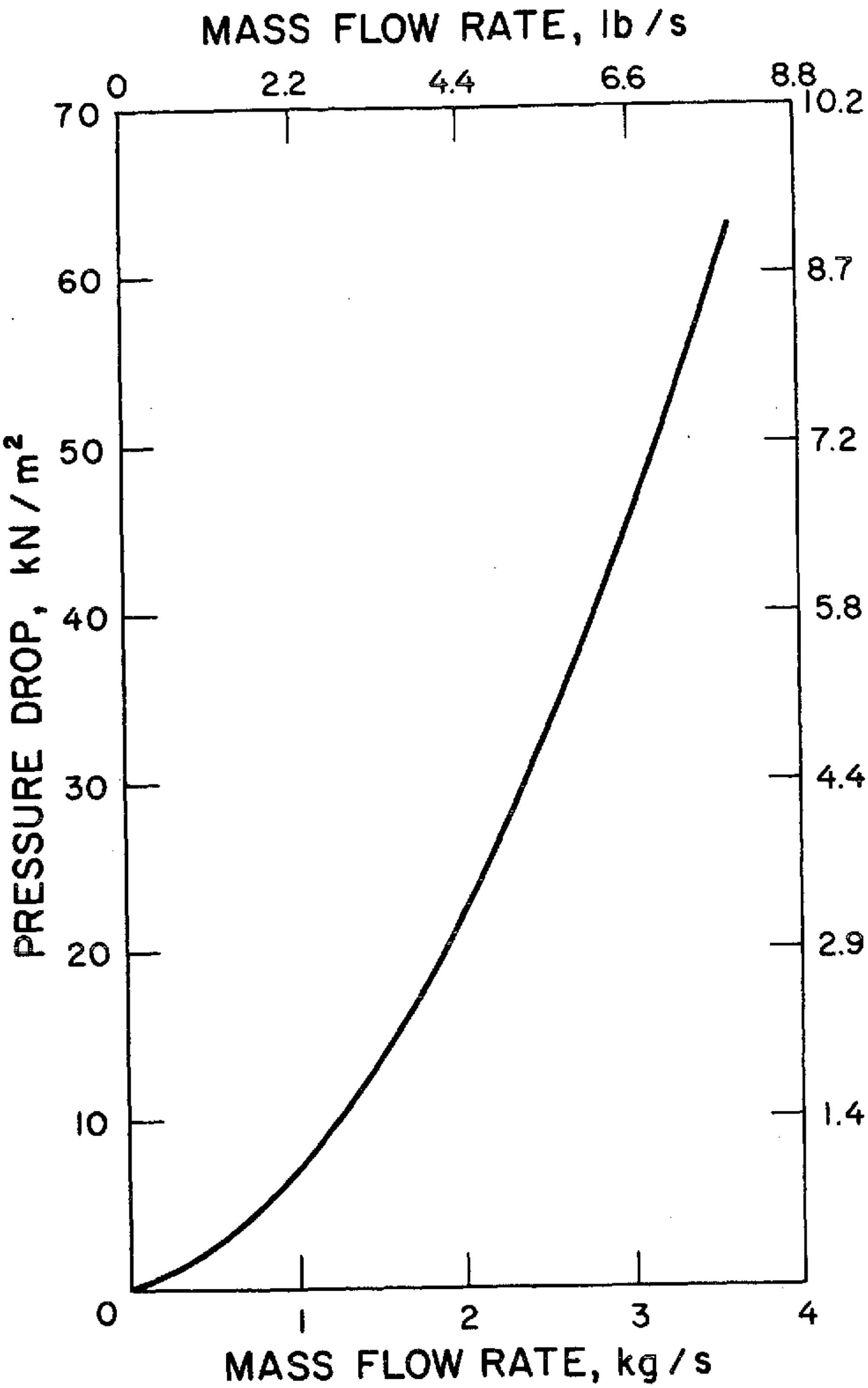


Figure 3.9 Pressure Drop - Oscillating Piston Meter - 1-1/2 inch-LN_2.

4.1. Orifice Meters

An orifice plate metering installation is quite simple (see figure 4.1). It generally consists of two flanges with an orifice plate of proper design bolted between the flanges. Various types of orifice taps are used such as edge taps or vena contracta taps depending on the specification to be used.

Physical Principle of Measurement. Theoretical development of the flow equations for head type meters may be found in ASME (1971) and in Appendix A. For the restricted case assumed for cryogenic flow, that of an incompressible measurement in closed pipelines in which the orifice plate has been placed, performance is based upon transforming a small portion of static pressure into kinetic during the passage of the fluid through the narrow opening of the orifice. If d is the diameter of the opening, and ρ the density of the liquid fluid, the mass flow rate q_m of this fluid depends only on the static pressure difference ΔP upstream and downstream of the orifice. The term g is the gravitational constant.

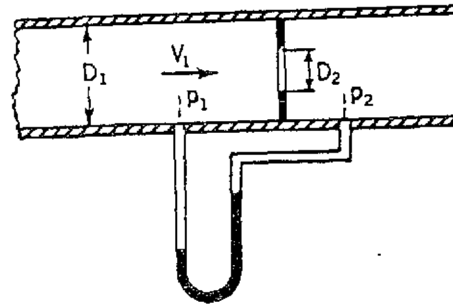
$$q_m = \alpha \frac{d^2\pi}{4} \sqrt{2g\rho\Delta P}$$

α, the so called flow coefficient, can be taken from tables in national or international standards such as DIN (1952) or ISO recommendation R-541 (1967). The standardized flow coefficients, obtained by direct calibration of geometrically similar orifice flowmeters, are valid in any practical application which fulfills the condition of geometrical similarity according to the detailed prescriptions of the standards.

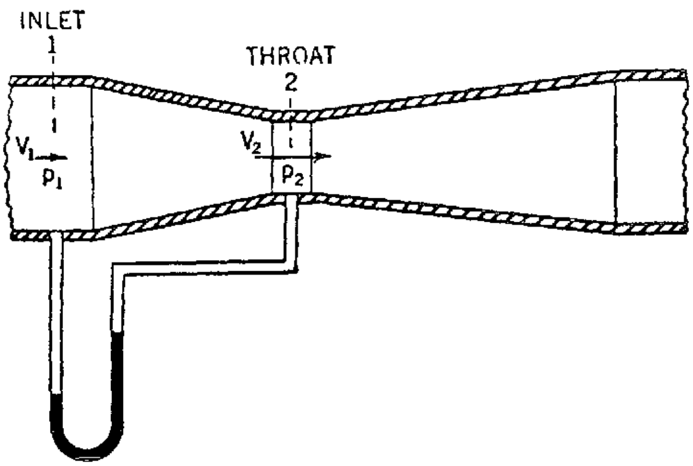
Design. Much of the literature on orifice meters as applied to cryogenic service is concerned with the application of the basic design to cryogenics. A good discussion of this is taken from Crawford (1963). "Most cryogenic liquids and gases have relatively low viscosity at low temperatures. This tends to yield relatively high Reynold's numbers and reasonably stable flow coefficients for a given flow measurement range. We (Linde Division, Union Carbide) have used standard ASME flow coefficients appropriate for the physical design and flowing conditions; and based on a wide variety of gas and liquid applications, there are no indications that the standard coefficients are inappropriate. Investigations of liquid flow with orifices [Richards, Jacobs and Pestalozzi, 1960] and venturi [Purcell, Schmidt, and Jacobs, 1960] under specific design and operating conditions indicate reasonable conformance with theory."

Crawford continues for the specific case of liquid flow, "For cryogenic measurements, the specific weight or density data for most fluids is available from several sources. Important factors in establishing accurate specific weight or density data are accurate measurements of fluid temperature and pressure. Depending on the fluid and the temperature range, satisfactory temperature measurement can be made using vapor pressure thermometers, thermocouples, or resistance thermometers. Care must be exercised to ensure that heat leak to the thermal sensing element via the lead connections does not become large enough to cause significant errors.

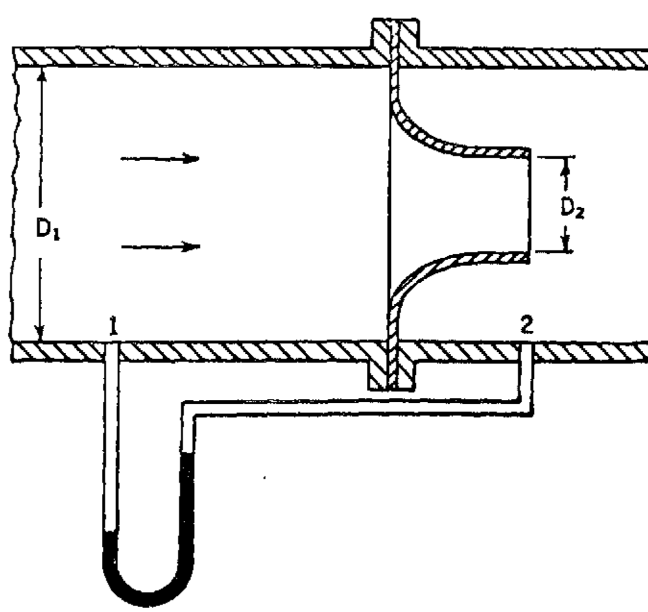
Differential pressure readings are made with conventional transmitters or other standard instruments. For liquid measurements differentials are selected as high as practical while still minimizing possibilities of flashing or vaporization in the primary element. The liquid should preferably be measured on the discharge of a pump, or with pressure and temperature conditions that insure against the occurrence of two phase flow.



(a) Orifice



(b) Venturi



(c) Nozzle

Figure 4. 1 Head-Type Flow Meters.

Stainless steel is a commonly used material for orifice plates; piping will usually be copper, copper alloy, aluminum or stainless steel. Possible differential contraction must be determined with respect to the orifice plate, flanges, gaskets, and bolting materials. Gaskets of polytetrafluoroethylene (PTFE), and filled PTFE are commonly used depending on the application. For oxygen service, the gasket must be of an inert material and must be cleaned thoroughly before installation, as must be the primary element, the connecting lines, and the parts of the secondary element in contact with the oxygen."

Crawford also cautions about the installation of tap lines. "Because it is difficult to make accurate orifice measurements for cryogenic liquids, the installation usually requires design features that differ from designs for ordinary liquids. The pressure tap lines connect the cold point at the element to a warm point at the transmitter or secondary element. Heat leak will cause liquid in the tap lines to vaporize and the lines will fill with gas at the line pressure. If the lines leave the top of the orifice flange, liquid will tend to rise in the line, and an erratic boiling condition can exist in either or both the lines as the differentials vary. Heat leak into the line causes boiling, which can cause a fluctuating differential pressure if bubbles form and varying liquid heights exist in the tap lines."

Crawford recommends that horizontal tap lines be used to a support point, which also would serve as a substantial heat source. He indicates that, beyond this support point, the direction of the lines is usually not of concern.

The location of the orifice taps in application of head type meters to cryogenic service becomes a problem apparently caused by fluctuations of the ΔP measurement. Wenzel (1971) approaches the problem of stabilizing the pressures in a slightly different manner. He provides the following suggested reasons for faulty differential pressure measurements. "The pressure connection lines are led from the pressure taps, at a low temperature level, through the insulation material, until they penetrate the outer shell of the insulation casing, where approximately ambient temperature is reached. Because of this temperature rise, a change of phase in the lines inevitably must occur. A column of the sub-cooled liquid enters the line up to the point where saturation temperature is present. The mass of the liquid column and the gas-saturated vapor-cushion which is locked in the warmer portion of the tube forms a mass/spring-system capable of oscillation. Pressure fluctuations to initiate oscillations are believed to be always present. The saturated vapor in the gas cushion may condense abruptly during the portion of the fluctuation when the pressure has risen above the main pressure. This causes violent movements of the liquid column involving inertia pressures in either direction and variations of the temperature distribution along the lines. By the motions of phase change areas, variations of gravitation energy are also involved if the lines are not installed strictly horizontally. From all this it is concluded that the instability in the location of the interface between phases in the lines from the pressure taps from a cryogen orifice meter is the main source of error and unreliability hitherto observed in practical use of such meters."

Wenzel (1971) continues his development by defining what he considers conditions for stability. These conditions include a stable temperature distribution along the lines from the pressure taps, virtually no entrace of the liquid into these lines, virtually zero flow into the pressure taps, minimum quantity of saturated vapor adjacent to the phase change interface, and the necessity that phase change interfaces must be held at the same horizontal level.

Wenzel's solution is as follows: "It has been found that these requirements can be fulfilled by a heated connecting line consisting of two sections of extremely different thermal conductance. A small amount of heat is taken, for example, from the ambient air outside the insulation box, by an extended fin surface. This heat is conducted through two sections of the line, in turn, to the meter. The first section is a copper tube, with large metallic cross-section, occupying 98-99 percent of the total length of the two sections. The second section is made of a thin wall stainless steel tube that also serves as the pressure tap immediately at the meter. On account of the very different heat conducting ability of the two sections the temperature drop along the copper section is less than that along the very short pressure tap."

Operational Characteristics. One of the few references providing experimental data is that of Richards, Jacobs, and Pestalozzi (1960). The flow prover was a weigh-time device consisting of a supply dewar and weighing device, a vacuum insulated transfer line which contains the test orifice, a receiver, pressure instrumentation, pressurizing gas supply and vacuum pumps. Pipe diameter was 0.25 inches, orifice diameter ratios were 0.250, 0.440, 0.600 and 0.832, pressure taps were indicated vertical in orientation and were spaced about one pipe diameter upstream of the orifice and about a half pipe diameter downstream. Fluid temperatures were measured by vapor pressure thermometers.

The estimate of uncertainty of mass flow on water and liquid nitrogen was less than 3-1/2 percent. This estimated error in the prover system must be considered in assessing the performance of the orifice meter. With this uncertainty of the prover in mind Richards, et al. (1960) conclude that the "normal calibration curves (in which the pressure drop is expressed in pressure units) for nitrogen and hydrogen can be obtained from those for cold water by means of a simple density correction." Most cryogenic fluids have viscosities of from 5 to 10 times less than that of water, and the authors did note that a decrease in the value of the discharge coefficient occurred with increasing Reynold's number.

Richards, et al. (1960) also addressed themselves to a problem frequently noted in projected use of head type meters in cryogenic service. This is the operation of the head type meter in respect to the saturation conditions of the fluid being metered. "The authors made many attempts to obtain points off the calibration curve with the upstream static pressure lower than the vapor pressure. In spite of the fact that the downstream static pressure was as much as 10 inches of mercury below the vapor pressure, all of the points fell on the calibration curve within the accuracy of the experiments. It is therefore probable that the time required for the nucleation and growth of bubbles in both nitrogen and hydrogen is so long that errors in the measurement of flow with sharp edge orifices, arising from downstream pressures, probably will not occur."

The experimental error in determining the discharge coefficients is less than 8 percent maximum error "which is significantly greater than conventional calibration accuracy claimed for orifice plates (0.5 percent)."

With all the above uncertainties in measurement capabilities the authors do make the following conclusions.

"1. Sharp edge orifices may be used for the measurement of the flow of liquid nitrogen and liquid hydrogen with the same confidence as with cold water, if the same care is taken with the liquefied gases as with water and if it is assured that single phase flow exists upstream of the orifice.

2. A vapor pressure bulb installed upstream of, and adjacent to, the orifice will reliably indicate that single phase flow exists at the entrance to the orifice.

3. As long as single phase flow exists upstream of the orifice, the level of the static pressure downstream of the orifice can be as much as 10 inches of mercury below the vapor pressure without influencing the flow measurement with liquid nitrogen and liquid hydrogen.

4. A straight line calibration curve obtained with cold water (and plotted as log volume flow rate versus log head drop across orifice) may be used with liquid nitrogen and liquid hydrogen, the errors depending on the Reynold's number effect and the accuracy of the instrumentation used. The deviation of the liquid nitrogen and liquid hydrogen data from the calibration curve is in general no greater than that of the cold water data.

5. It should be stressed that, because the accuracy of the measurements in this investigation was poor compared with that used for calibrating orifices with cold water, the orifice coefficients reported here should not be used for design purposes. However, the results indicate a definite possibility that orifice coefficients obtained with water may be used with liquid nitrogen and liquid hydrogen.

6. Isentropic orifice theory is just as applicable to liquid nitrogen and liquid hydrogen, for the same range of parameters used in these experiments, as to cold water.

7. It should be noted that the Reynold's numbers in these experiments are below the range usually used with flow orifices, and that the pipe diameter is smaller than is customarily used There is no reason to believe that the circumstances detract from the fulfillment of the purposes of this investigation."

The work of Wenzel (1971), incorporating the modified tap design described above, does not provide extended data (four data points between 750 and 1750 kg/h) on the performance of this modification. Wenzel does indicate that an orifice flowmeter was designed, manufactured, and installed in accordance with the German Standard (DIN 1952]. "The direct calibration by a weigh tank method confirmed the flow coefficient given by the standard by +0.7 percent." Wenzel further indicates that no single measurement in liquid nitrogen service is outside the predetermined tolerances as specified. His conclusion, "This proves that, by improving the differential pressure measurement as described above, cryogenic liquid flow measurement by means of standardized orifice plates is possible and in complete accordance with the standards."

Brennan, Stokes, Mann, and Kneebone (1973) report on the performance evaluation of an orifice meter installation of Wenzel's (1971) design. This meter is illustrated in figure 4.2. Liquid flow in the metering section is indicated by the arrows in the figure. Pressure drop measurements are made with four corner taps which communicate with an annular chamber on each side of the orifice. Pressure tap lines are constructed in a special way in an attempt to eliminate pressure oscillations in the line. Unique features are indicated on the figure. The degasification lines are used to control the liquid-vapor interface in the annular space. The design goal was to maintain the interface at the entrance to the pressure line taps which would keep the amount of saturated liquid to a minimum. When the meter is used in a pressurized transfer configuration, the degasification lines are independently connected to the ullage space of the upstream vessel. Since the tests reported were conducted with a

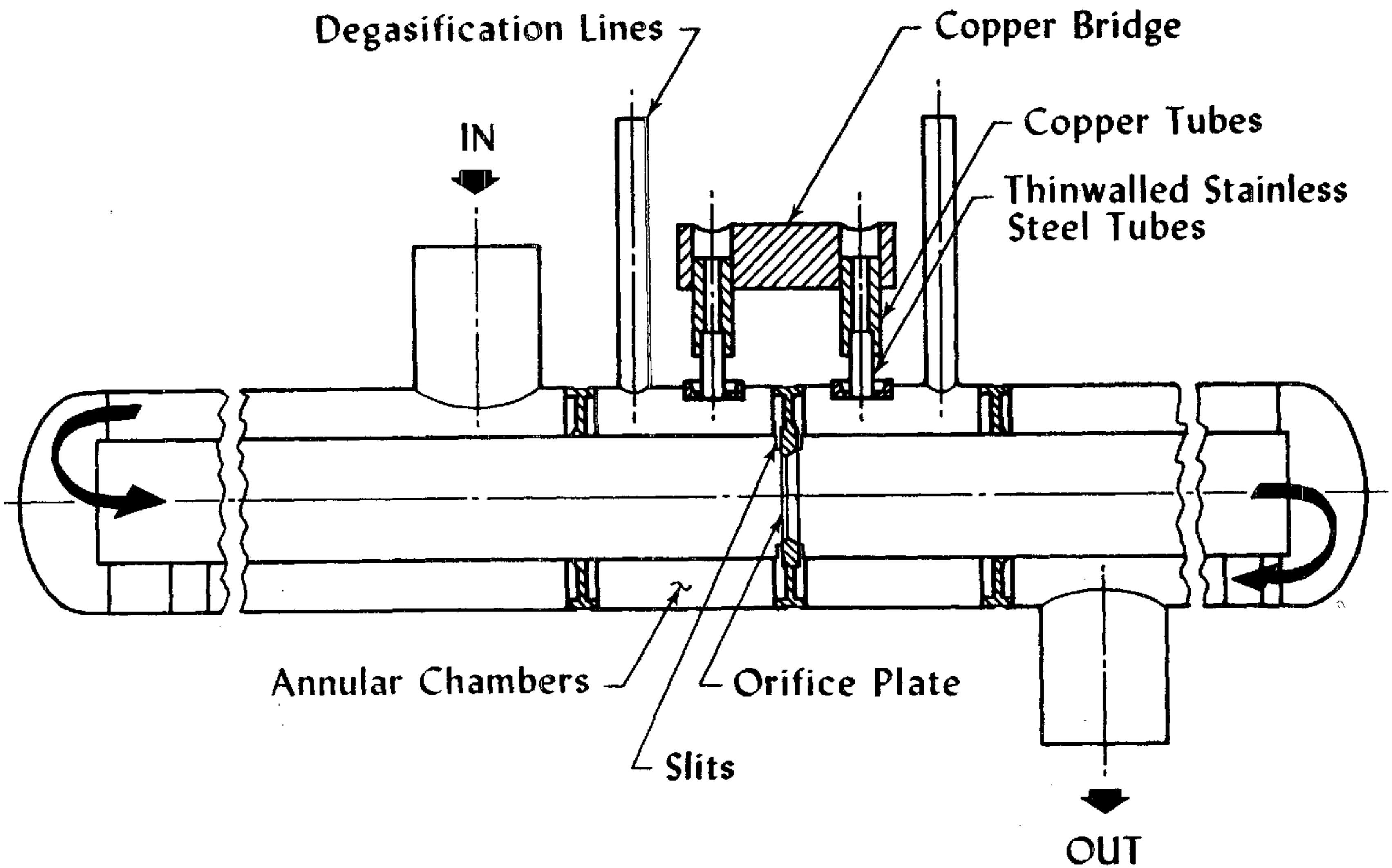


Figure 4.2. Cryogenic Orifice Flowmeter.

pump flow system, it was not possible to connect the degasification lines to an appropriate ullage space. Therefore a consistent test procedure was adopted whereby test conditions were established, and then gas from the degasification line was bled momentarily. The lines were then valved off and the test started after a two minute wait. The purpose of this procedure was not to try to duplicate any particular set of installation conditions but rather to develop an internally consistent set of data.

The meter supplier specifications are:

fluid - liquid nitrogen
maximum flow rate - 7.66 lb/s (3.48 kg/s)
maximum pressure drop - 25 inches of H_2O (6.2 kN/m^2)
orifice diameter (calculated) - 1.84 inches (46.65 m) at 293 K
pipe inside diameter - 3.27 inches (83 m)
standardized flow coefficient - 0.6395.

Only one of these meters was tested. Since there are no moving parts to this meter, no stability or second rangeability tests were run. Results from the boundary test were inclusive and not reported (see test procedure Appendix B).

Results of the evaluation of Brennan, et al. (1973) are shown in figure 4.3. The deviation from predicted discharge coefficient may be caused by the coefficient calculation being based on a specified orifice diameter rather than the actual measured value. The scatter of the data at the low flow rate significantly effects the performance over the full flow range of 3 to 7-1/2 lb/s This scatter may well be caused by the inability to make precise pressure differential measurements in the region of 3 inches of water ΔP. For the three groups of data between 4 and 7-1/2 lb/s, it can be seen that the 3σ precision could well be within ±1 percent. All data are on liquid nitrogen.

4.2. Venturi Meter

A ccording to ASME (1971) the venturi tube combines into a single unit a short constricted portion between two tapered sections and is usually inserted between two flanges in a pipe. Its purpose is to accelerate the fluid and temporarily lower its static pressure (see figure 4.1). Suitable pressure connections are provided for observing the difference in pressures between the inlet and the constricted portion or throat. The application of the venturi tube to cryogenics would be for the purpose of reducing the total pressure loss through the meter and at the same time allowing a design that would provide a high pressure differential versus flow. The proportions of venturi tubes used for metering liquids are usually substantially the same as those originally adopted in 1887 by its inventor Clemens Herschel. Standard dimensions have been adopted and may be found in ASME (1971) and elsewhere. When applied to cryogenic service for nitrogen, oxygen and argon, only one reference in the literature cited provides detailed information [Purcell, Schmidt and Jacobs, 1960]. Other citations [Close, 1968, 1969; Angerhofer, 1965] provide performance information, but, in general, refer to the original work of Purcell, et al. (1960).

Design. Operation of a venturi meter flowing incompressible liquid can be shown to be

$$q = KA_2 \sqrt{2g\rho\Delta P}.$$

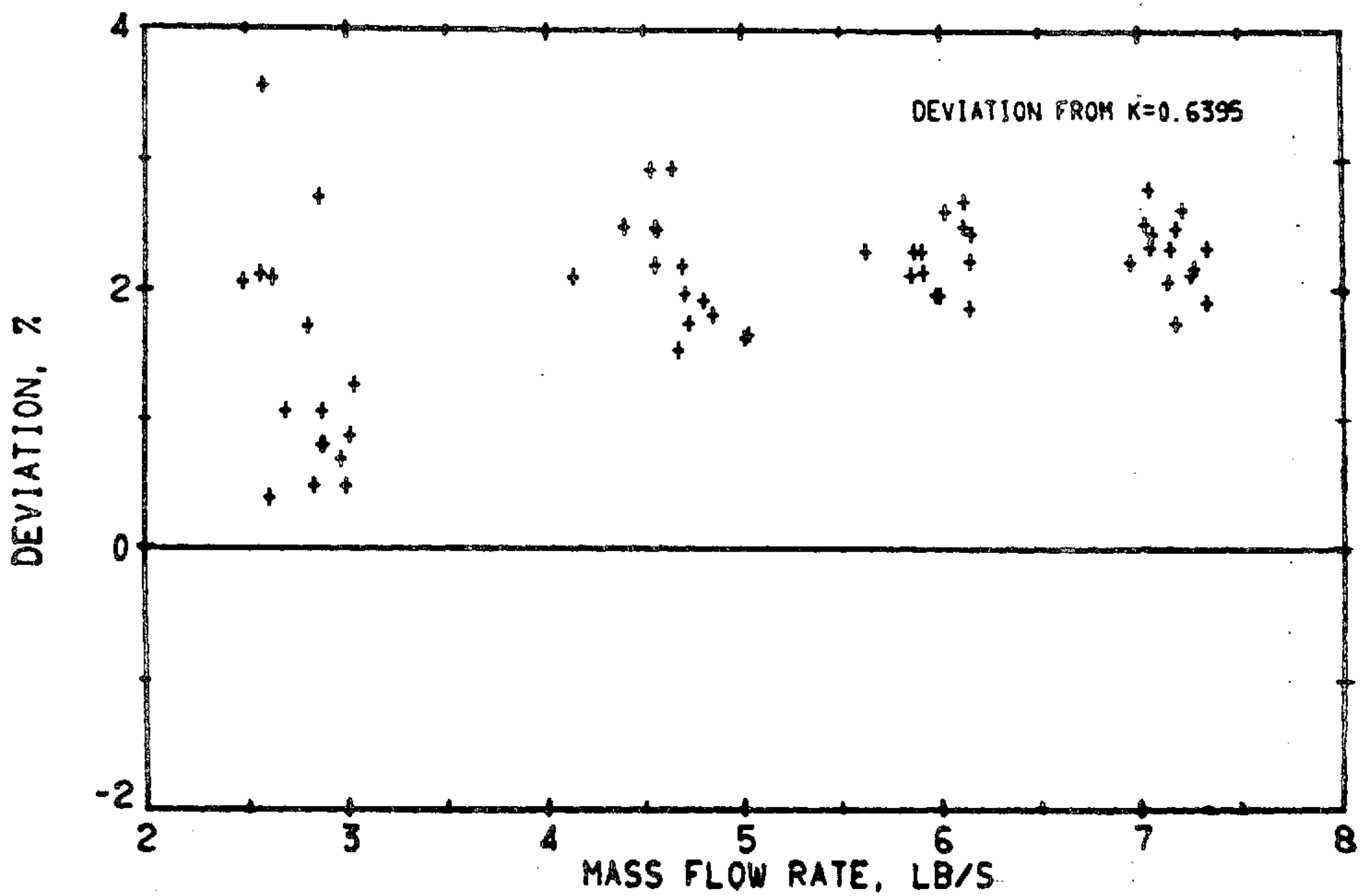


Figure 4.3 Cryogenic Orifice Meter Performance.

The numerical value of the dimensionless flow coefficient K is not necessarily an indication of accuracy. The coefficient K may be unity, less than unity, or greater than unity. Dimensional analysis and dynamic similarity show that K is the function of Reynold's number and pipe-to-throat diameter ratio. Since the venturi meter is basically a head meter, most comments made for orifice plate meters apply to venturi meters in cryogenic service. The advantage of the venturi meter over the orifice meter is the definite location of the vena contracta. It is therefore simpler to define the minimum static pressure to be realized in the metering process and avoid incipient or fully developed boiling of the fluid due to low vapor pressure.

In the work of Purcell, et al. (1960), the venturi meter was designed for operation on liquid hydrogen and all data are confined to this liquid hydrogen flow. No other detailed information is available from the literature on the design of venturi meters for cryogenic service .

Operational Characteristics. Because the work of Purcell, et al. (1960) was confined to liquid hydrogen, an explanation of the operating characteristics will not be given here. No other detailed report on the operational characteristics of a venturi meter applied to liquid oxygen has been found in the literature. Angerhofer (1965) cites a ±1 percent of range repeatability for rangeability of three to one. Close (1968) consolidates cryogenic orifice and venturi performance into total error of ±3 percent of range, repeatability ±1 percent of range, rangeability of three to one with a typical full range ΔP of 4 to 20 psi. Close cites practice to maintain the orifice or venturi throat pressure well above the vapor pressure to avoid cavitation.

4.3. Flow Nozzle Meters

The flow nozzle is illustrated in figure 4.1. The purpose of the curved entrance is to lead the fluid smoothly to the throat or measuring section so that it shall issue from throat without contraction as a straight cylindrical jet having the same diameter as the throat. The flow nozzle therefore performs the same function as the entrance column and throat of a venturi. It decreases the cross section of the stream in the pipe in a known ratio and also produces a difference of pressure between the entrance and the throat by accelerating the fluid. In effect, the flow nozzle is a venturi tube that has been simplified and shortened by omitting the long diffuser on the outlet side. Only two citations in the literature, both by Bucknell (1964, 1966), provide information on cryogenic performance.

Design. Referring to figure 4.1, the equation for a flow nozzle is similar in all respects to that of other head type meters and is

$$q = KA_2 \sqrt{2g\rho\Delta P}\,.$$

General forms of the nozzle, relative diameters and velocity of approach coefficients (K) are as specified in ASME (1971).

Operational Characteristics. Bucknell (1964) provides information on the use of differential head flowmeters in liquid oxygen service based on the Pratt & Whitney flow prover. "Differential head flowmeters appear to be limited to steady state and narrow flow range applications where about ±1 percent of full scale repeatability and ±3 percent of full scale accuracy are acceptable. However, they do have advantages in reliability for certain corrosive fluids or violent operating requirements. It is difficult to eliminate pressure oscillations caused by boiling in pressure lines. Additionally, the magnitude of pressure oscillation caused by real flow oscillations will be affected by the presence of gas in the pressure lines."

Bucknell goes on to state that larger than standard static pressure tap hole sizes are often used to reduce lag and over shoot during transients. Bucknell indicates that one method of treating the pressure oscillations in pressure lines may be to minimize them by horizontal installation of the lines beyond the point where all liquid has vaporized as has been suggested by Crawford (1963). Bucknell (1964) reports data on liquid oxygen for two long radius standard ASME (1971) nozzles. The liquid oxygen prover for testing of these nozzles is assumed to be as described by Bucknell (1962) and is a weigh time system having a "calibration error estimate of ±0.30 percent on liquid oxygen." The following data are provided by Bucknell, Lowler and Street (1964).

"Two nozzles were calibrated in our liquid oxygen system by gravimetric techniques. These limited liquid oxygen calibration data are presented to show that reasonable data can be expected using standard ASME nozzles and coefficients without calibration. Following are liquid oxygen calibration results for two ASME long radius nozzles with pressure taps at 1 D and 1/2 D; terminology is as cited in ASME (1971).

Table 4.1. Flow Nozzle Liquid Oxygen - Water Performance

D (inches)	β	R_D	$C(LO_2)$	C(ASME)
3.260	0.504	1.1×10^6	0.987	0.993
3.260	0.504	1.6×10^6	0.992	0.994
2.157	0.510	1.4×10^6	1.012	0.994
2.157	0.510	1.9×10^6	1.000	0.994
2.157	0.510	2.2×10^6	1.003	0 995

Area contraction of 0.5 percent has been included in computing liquid oxygen discharge coefficients. Each value shown above represents an average of two to three tests; scatter of individual points was about ±0.5 percent for both nozzles tested. Little significance is attached to individual trends from point to point. It is only noted that the average liquid oxygen discharge coefficient of 0.999 compares reasonably with 0.994 expected above the Reynold's number of 10^6."

Summary for Head Type Meters. The estimated uncertainty in cryogenic flow measurement using head type meters is in a range of ±1 to ±3 percent. This is composed of the uncertainty in bias shift caused by thermal contraction of the material, uncertainty in the effect of increased Reynolds number, and a large imprecision traceable to the methods of pressure measurement and pressure tap design.

Use with liquid oxygen seems only limited by material compatibility.

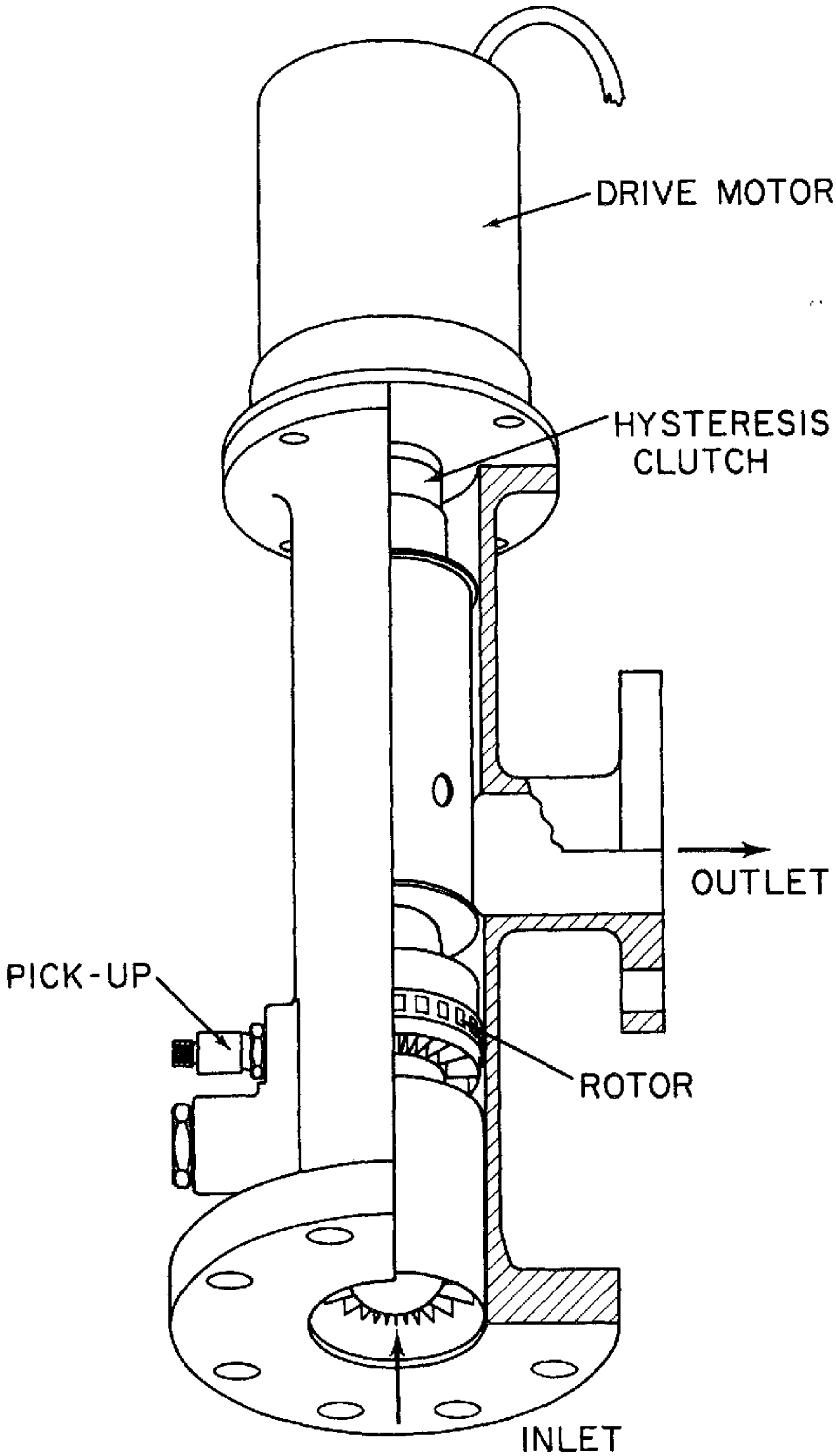


Figure 5. 1 Cryogenic Angular Momentum Mass Flowmeter.

5. RATE METERS - FORCE - MOMENTUM

These measuring devices are characterized by a mechanism which removes or redirects a portion of the momentum of the flowing fluid in such a way as to give an indication of the mass flow rates. ASME (1971) provides a few examples of this type of meter in the hydrometric pendulum, the vane meter, and the transverse momentum meters. Meters of the latter class, the transverse momentum meters are the ones currently being converted to cryogenic service. Much effort over the past 15 to 20 years has been directed to developing a flowmeter for the measurement of mass flow. This is particularly true in the case of liquid hydrogen [Wyle, 1965]. For the case of cryogenic fluids -- liquid oxygen, liquid nitrogen or liquid argon -- it seemed sufficient to measure volumetric flow and correct for density changes through temperature measurement and thermodynamic property relations. This was true until quite recently when Close, Hayford, Carroll, and Rubin (1971) described an angular momentum mass flowmeter developed for commercial cryogenic service for the liquids oxygen, nitrogen and argon.

The following subsections will discuss those meters reported in the literature which provide performance data on liquid oxygen or could be inferred by operation on liquid nitrogen or liquid argon.

The work of the Wyle Laboratories (1965) describes the performance of a flow prover operated on liquid nitrogen and liquid hydrogen for tests of different mass flowmeters for cryogenic service. Since the purpose and the reported data of the Wyle work is primarily concerned with liquid hydrogen, and only one liquid nitrogen test (of poor quality) is reported, we will only indicate the types of meters that were included under this program. These included a transverse momentum flowmeter consisting of a loop through which the fluid flows. Means of driving the flow piping in a constant manner are provided as well as a means of measuring the mass flow reaction. The second type, based on angular momentum, consisted of a turbine driven at constant speed imparting angular momentum to the fluid which is removed in the second turbine. Resultant torque on the second turbine is proportional to the mass flow rate. The third type, also angular momentum, consisted of a single turbine driven with a constant torque motor, and the turbine speed was inversely proportional to mass flow. The fourth type of meter was composed of twin in-line turbines with a linear spring coupling [Taylor and Pearson, 1966]. The phase angle between the turbines was proportional to flow momentum with the time delay of the signal from each of the rotating turbines proportional to mass flow. The fifth type of mass flowmeter was simply a turbine volume flowmeter with density compensation.

The work of Close, Hayford, Carroll, and Rubin (1971) describes a new angular momentum type mass flowmeter developed for cryogenic service. This paper presents the theory and practical application to commercial delivery of liquid oxygen and other liquefied industrial gases.

Physical Principle of Operation. Close, et al. (1971) provides the following description of the principle of operation (see figure 5.1), "It operates by imparting angular momentum to the through flow by means of a straight bladed rotor driven at constant torque. The incoming fluid is conditioned by the stator to have zero net angular momentum. The flow entering the rotor is coupled in such a way it exits with a rotational speed equal to that of the rotor. Under these conditions, rotor speed is inversely proportional to mass flow rate." The basic assumptions used by Close, et al., in the design of the meter indicates that the fluid exiting from the rotor has no change in angular momentum from that which is established by the rotor rotation, there is zero net angular momentum entering the rotor, mechanical, viscous, and pumping losses are negligible, and the velocity profile is flat. Close, et al. further provide a detailed explanation of the design of the transmitter, converter and readout.

Design. The meter of Close, et al. (1971) was designed to operate between 4 and 32 lb/s. Its linearity range above 8 lb/s is well within the design for truck mounting. Physically the meter has the appearance of a T section with the fluid entering and turning through 90 degrees. Drive motor and impeller are installed on the axis of the flow entrance. No details are given on the specific design requirements for liquid oxygen service.

Operational Characteristics. Performance characteristics of the meter described by Close, et al. (1971) are provided in the paper as well as in Brennan, et al. (1973). As indicated by Close, the meter was designed for mass flow rates between 4 and 32 lb/s. Close finds that most meters are linear within 0.5 percent. Below 8 lb/s the meter tends to underregister with approximately 2 percent underregistration at 4 lb/s. Repeatability of the meter is typically within 0.2 percent after cooldown and within 1 percent until cooldown is complete. Full range pressure drop for this meter is 6.5 psi in liquid nitrogen service.

The prover used by Close is described in Close (1969) where he states "Total system uncertainty is less than 0.25 percent for volumetric meters, and less than ±0.1 percent for mass flowmeters." Close comments on the calibration of the mass flowmeter "The mass flowmeter has proven to be linear in water or in cryogenic service but usually not with the same trim adjustments. A meter which is linear in water calibration typically is not linear in liquid nitrogen up to 10 percent. There usually is a zero offset towards underregistration from 3 to 7 percent." Close summarizes his work on the cryogenic mass flowmeter "The cryogenic mass flowmeter has been proven to be accurate well within the ±1 percent design goal. Calibration is stable even following repairs and a years service. Deliveries through this meter now can be certified accurate within 1 percent provided the meter is calibrated on a liquid nitrogen test stand where the scale of calibration is certified within 0.1 percent and traceable to the National Bureau of Standards. Correlation between water calibration and cryogenic fluids has not been successful to date."

Brennan, et al. (1973) describe the performance of the flowmeter of Close (1971) as submitted under the joint NBS-CGA flowmeter evaluation program. Tests of this meter were conducted at the NBS Cryogenics Division, Boulder, Colorado. The meter suppliers' specifications were:

fluid - liquid oxygen, nitrogen or argon (appropriate electronic adjustments required for each fluid)
size - 3 inch (7.62 centimeters)
maximum flow rate - 32 lb/s (14.5 kg/s)
minimum flow rate - 4 lb/s (1.8 kg/s)
maximum pressure - 350 psi (2.4 MN/m^2)
accuracy - ±1 percent 8 to 32 lb/s (3.6 to 14.5 kg/s) and
±2 percent 4 to 8 lb/s (1.8 to 3.6 kg.s)
register - electromechanical, smallest division - 10 lbs (4.5 kg).

Flow capacity of the NBS facility is approximately 23 lb/s (10.4 kg/s); therefore, it was not possible to test these meters throughout the manufacturer's stated flow range.

Two meters were tested under the NBS-CGA program. The performance of one of the meters (both meters gave similar performance data) was precision (3σ) at start of testing, ±1.43 percent; at the end of testing, ±1.05 percent. The bias at the start of the testing was +0.41 percent, at the end of testing -0.47 percent. Quantity of liquid metered during the test was 4,773,700 lb (2,165,314 kg). The maximum flow test rate was 23 lb/s (10.4 kg/s); minimum subcooling required was 8 K. Pressure drop for this meter is shown in figure 5.2.

6. RATE METERS - VELOCITY

In general, the term velocity meter is applied to those meters having a rotating primary element which is kept in motion by the direct movement or velocity of the fluid stream. Velocity meters may be designed for use in either open or closed channel flow, but in the case of cryogenics, closed channel flow or pipe flow is the exclusive application.

6.1 Turbine Meter

There are approximately 25 papers in the literature within the scope of this study which deal in one way or another with turbine flowmeters in cryogenic service. Of these references, only four could be considered primary data on design, performance, or application. The work of Grey (1959) provides the most detailed discussion of design and analysis of major errors in the application of turbine meters to cryogenics. The work of Deppe and Dow (1962) and Deppe (1966) provide application and performance information on cryogenic turbine meters in oxygen service in sizes from 6 to 18 inches in diameter. Bucknell (1962) provides performance and operational data on a 2-inch diameter liquid oxygen turbine flowmeter. Brennan, Stokes, Mann, and Kneebone (1972) provide extensive evaluation and liquid nitrogen performance of eight turbine flowmeters (1-1/2 to 2-inch diameter) representing five different manufacturers. The remaining references are summaries of these primary references or isolated instances of the application of turbine meters to cryogenic liquid flow.

Physical Principle of Measurement. Turbine flowmeters may be of three general types: radial, in which the fluid is caused to flow normal to the axis of the piping (a classic turbine meter); axial, with the flow generally in line with the axis of the piping; or a combination of these two. Cryogenic turbine flowmeters based on the cited literature are exclusively axial turbine flowmeters, as illustrated in figure 6.1. A basic description may be drawn from Grey (1959) "The primary element of the turbine meter is a freely spinning rotor having N blades, each inclined at an angle α to the axis of flow. The rotor is supported in guides or bearings mounted in a housing which forms a section of the pipeline. The angular velocity (rpm) of the rotor may be detected by one of a number of methods; e.g., a permanent magnet encased in a rotor body will induce an alternating voltage in a pickup coil mounted on the housing, constructed of a magnetic material so that the change in magnetic circuit reluctance, as each rotor blade passes the coil core, causes an alternating current to be induced in the coil. Capacitive and photoelectric methods of observing rotor rpm have also been proposed. The primary requirement, however, is that the angular velocity of the rotor be directly proportional to volumetric flowrate or, more correctly, to some average velocity of the fluid in the pipe."

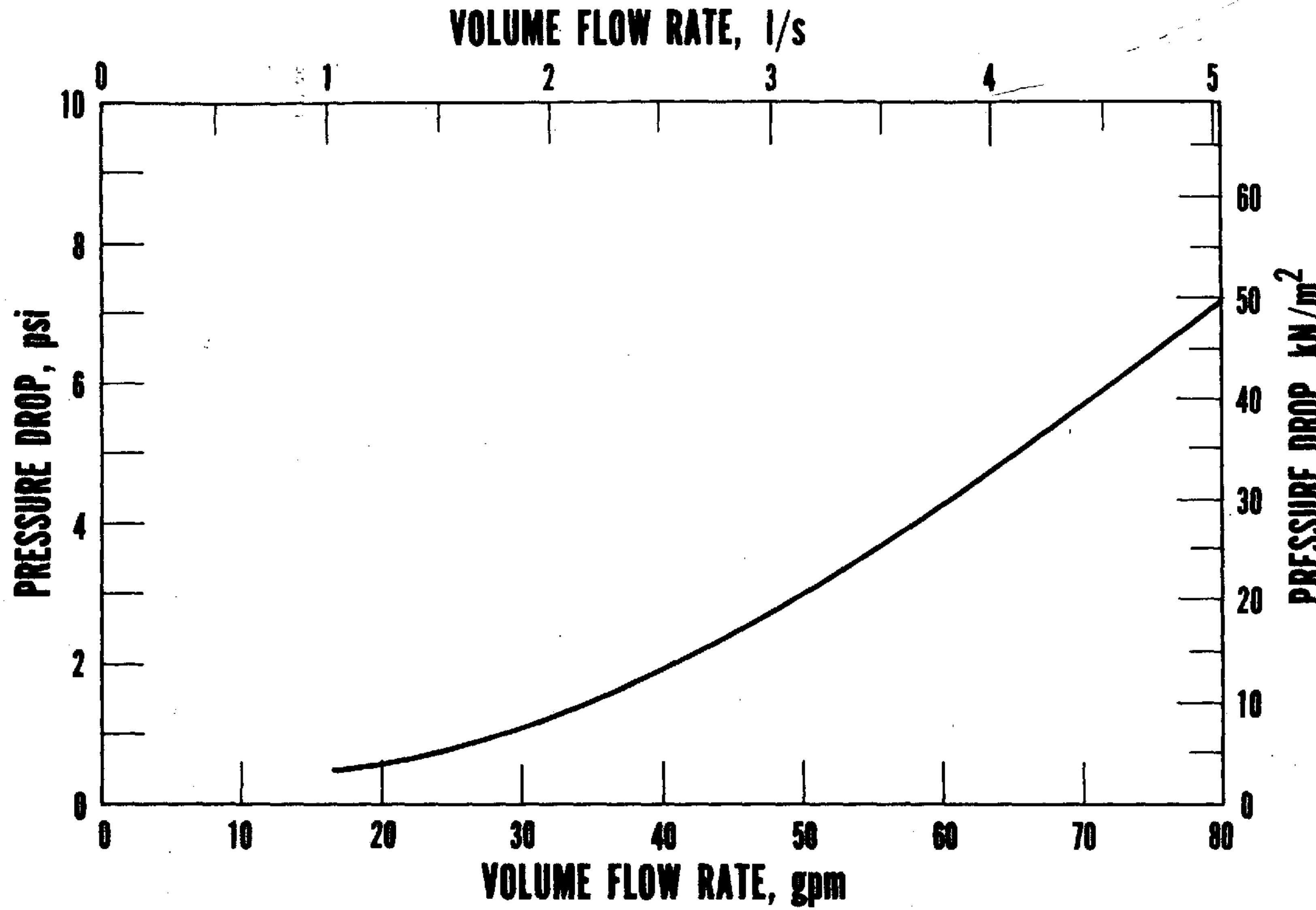


Figure 5.2. Pressure Drop - Angular Momentum Cryogenic Mass Flowmeter.

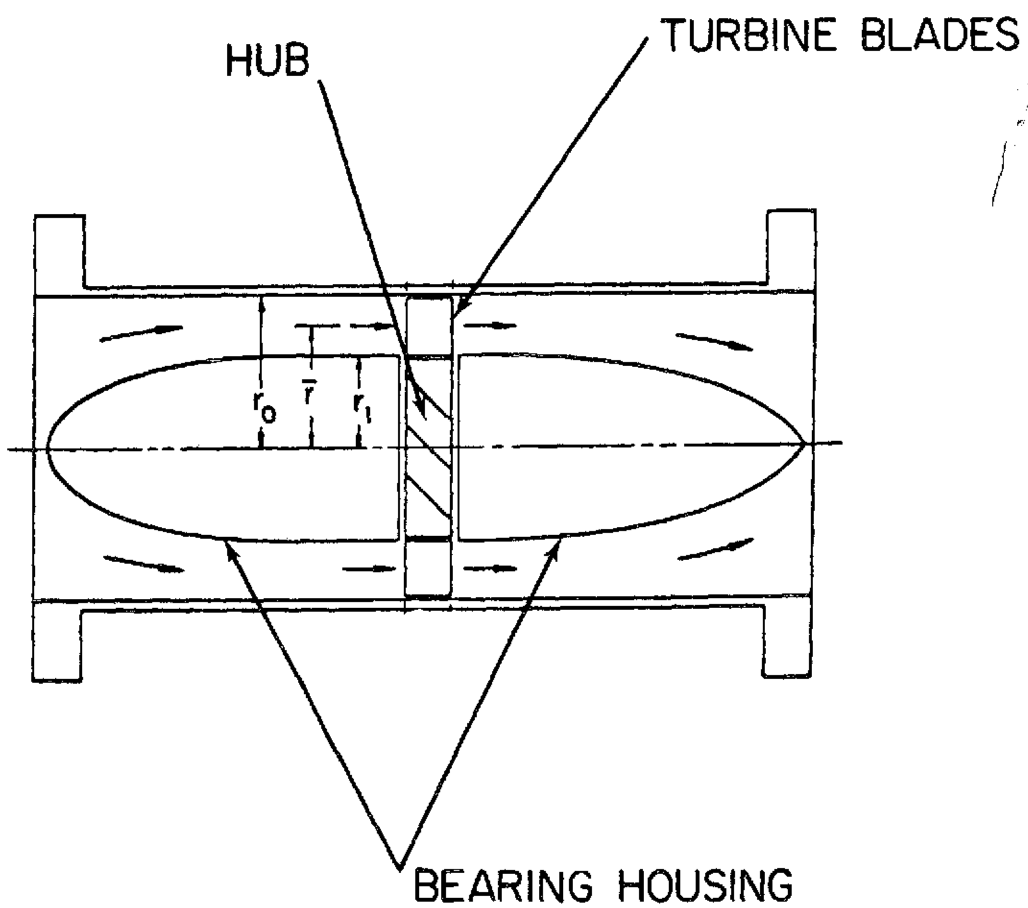


Figure 6. 1 Turbine Flowmeter.

Design. Grey (1959) and Thompson and Grey (1967) provide the most extensive design and analysis information for application of turbine flowmeters to cryogenic service. Included in these reports is a discussion and development of the relationship between the blade angle, rotor diameter, rotor angular velocity, and fluid velocity. Grey (1959) then goes on to consider possible errors which might cause deviation from this basic relationship. These errors are grouped into retarding torque on the rotor, changes in average velocity as result of the insertion of the meter, and swirl in the incoming fluid. He believes that all sources of error in a well designed turbine meter can be eliminated by proper engineering design, except to those caused by "fluid effects." These effects tend to produce changes in average pipe velocity rather than mechanical features of the meter. Of the four error sources of this type, only the effects of compressibility in the pure liquid state have been found to be negligible, and Grey (1959) then proceeds to treat the remaining three, cavitation, dimensional changes caused by temperature, and viscosity, in detail.

Very little information is available from the literature on materials used in turbine meters. Grey (1959) indicates that rotor materials of stainless steel and alloys of nickel, copper, silver, aluminum, and magnesium have been used. Roller bearings are generally of martensitic stainless steel (400 series or equivalent) while some journal bearings are graphitic. Meter body materials are generally of stainless steel to inhibit corrosion or are matched to rotor materials in an effort to reduce the relative thermal contraction effects at low temperatures.

Cunningham and Anderson (1965) do provide some data on selection, modification and performance of ball bearings in liquid oxygen service. They indicate special glass fiber in polytetrafluoroethylene retainers gave comparable friction coefficients in LO_2 as have been found for oil-mist-lubricated ball bearings tested over the same radial load range.

Operational Characteristics. Bucknell (1962) and Deppe and Dow (1962) present turbine meter performance data on liquid oxygen relative to a particular flow prover. Brennan, et al (1971) provide performance and evaluation of turbine type flowmeters operating on liquid nitrogen and reference to a well-defined prover.

The liquid oxygen flow prover described by Bucknell (1962) is a weight-time system having a "calibration error estimate of ±0.30% on liquid oxygen." Using this flow prover, Bucknell reports the performance of turbine type flow transducers on liquid cryogens. "In general, flow calibration systems do not have adequate repeatability to define turbine-type flowmeter repeatability. The only device with adequate repeatability is another meter. Common gating signals were used for total pulse counts from two meters in series for a five-minute flow duration to statistically estimate meter repeatability. More than one hundred runs were included for a sample of several meters of the same design for each fluid. Individual ratios were statistically combined after expressing each ratio as percent deviation from the mean for each specific pair of meters. Three sigma repeatability of these individual meters, in percent of flow, was estimated for these data to be ±0.63 percent for a 2-inch liquid oxygen meter, flow range of 100 to 200 gallons per minute; and ±0.45 percent for a 3-inch liquid hydrogen flowmeter, range of 450 to 750 gpm."

Although Bucknell did not provide pressure loss data, over the flow range for liquid oxygen, he did indicate that "at a 5 psi pressure loss with a back pressure of 20 psi above vapor pressure, cavitation appeared to begin in the meter. Cavitation error was indicated by a higher apparent cycle per gallon factor when the pressure was decreased." Bucknell also found that "Bearings are the most critical components of a cryogenic turbine flowmeter. Ball-bearings have been found to be the most satisfactory for our applications. Their ability to tolerate overspeed and gas operation is better suited to frequent runs of a few minutes duration with gas purging for extended periods before and after each run. It is expected that ball-bearings with tetrafluoroethylene retainers will improve meter performance Journal bearings have been very successful when operation on gas was infrequent and at low flow rates. Because of a different configuration, it may not be valid to attribute performance characteristic to the bearings alone; however, the meters tested with journal bearings were less linear but more repeatable in comparison to meters with the original ball bearings. Destructive failure of meters with journal bearings resulted at gas flow rates where bearings became unstable. Failures have also been encountered in liquid operation when wear increased journal bearing clearance beyond a critical value. Cleanliness precautions during repair, when cleaning for cryogenic service, for installation, and for operation have required much tighter control for ball-bearing meters than for journal-bearing meters."

The work of Deppe and Dow (1962) referenced turbine flowmeter liquid oxygen performance to a volume time calibration prover. The volume was calibrated with water and then corrected for temperature and pressure.

The following is a summary of the work of Deppe and Dow (1962) referenced turbine flowmeter "The calibration facility can flow water or liquid oxygen at rates up to 1000 lb/s. The tank volume is known to within 0.034 percent for water calibrations. Accuracies of meter calibrations are ±0.1 percent in water and ±0.18 percent in liquid oxygen. The repeatability of the system is better than ±0.5 percent. The shift of calibration factors from water to liquid oxygen for turbine meters, 6 and 8 inch diameter, varies from 0.2 percent to 2.0 percent and each meter is unique. No negative shifts were observed."

The data on large turbine flowmeters (greater than 8 inch diameter) for liquid oxygen service is quite meager. Deppe (1966) points out the lack of calibration provers for cryogenic service and presents data on a liquid oxygen calibration for a 14-inch diameter turbine meter. The flow data were referenced to a NASA, Marshall Space Flight Center pump test stand and indicates a water to liquid oxygen k-factor shift (cycles per gallon) of approximately 1 to 1-1/2 percent. Deppe points out that large size meters (anything greater than 8 inches in diameter) present significant difficulties in establishing flow measurement capability due to the lack of large scale cryogenic flow provers.

The performance of turbine type flowmeters is reported by Brennan, Stokes, Mann and Kneebone (1972). The evaluation was conducted using the liquid nitrogen flow prover located at NBS, Boulder, Colorado. A description of this flow prover and a provisional accuracy statement of this facility is given in Dean, Brennan, Mann, and Kneebone (1971) and in Appendix B.

The criteria for evaluating meter performance are the precision and bias of the meter and the existence of flow rate, temperature, subcooling, pressure and time order (an indication of meter wear) dependencies. The bias is defined as the mean percent deviation from the measured NBS mass for repeated measurements at a specified set of flow conditions. The precision is a measure of the ability of the meter to reproduce the same bias for repeated measurements at the same flow conditions.

A detailed description of the meters and their performance is given in Brennan, et al. (1972). A summary of the performance of three of the meters included in this program is given in table 6.1.

Table 6.1. Turbine Meter Performance Summary

Liquid Nitrogen

	1-1/4 inch	1-1/2 inch	1-1/2 inch
Precision (3σ) at Start, %	± 0.69	± 0.54	± 0.57
Precision (3σ) at End, %	± 0.66	± 0.48	± 0.87
Bias at Start, %	+ 3.82	+ 0.10	+ 1.08
Bias at End, %	+ 3.85	+ 0.09	+ 1.23
Liquid Volume Metered during Stability Test	384,000 gal (1,453,600 ℓ)	810,000 gal (3,066,200 ℓ)	624,000 gal (2,362,100 ℓ)
Maximum Test Flow	80 gpm (5ℓ/s)	180 gpm (11.3 ℓ/s)	130 gpm (8.2 ℓ/s)
Minimum Subcooling, K	6	14	16

Summary and Conclusions. Turbine meters are used in cryogenic service for a number of reasons. The sensing element is relatively small, minimizing the mass to be cooled, meters maintain good precision (significantly less than ±1%), and they demonstrate good rangeability. The output signal requirements are moderate for a straight run upstream of the element. Turbine meters can be damaged by excessive rotor speed from gassing or two-phase flow and as with other volumetric meters with mechanical elements, protection must be provided through the design, cooldown procedures, and proper piping. The pressure drop through the turbine meter is fairly high for use in some cryogenic systems. A typical pressure drop for a 1-1/4 inch meter from Brennan, et al. (1972) is shown in figure 6.2.

To achieve the full potential of the turbine meter as a flow measurement element in cryogenics, the meter must be calibrated at the cryogenic temperature.

6.2. Vortex Shedding Meter

The application of this device to measurement of flow of liquids and gases is fairly recent [Rodely, 1969]. It is neither mentioned nor classified by ASME (1971). The phenomena upon which it is based is the Karman vortex trail and is mentioned in several texts. The description by Binder (1949) is typical. "Certain phenomena associated with the flow around cylindrical cylinders, elliptical cylinders, and flat plates are explained by reference to the so called Karman vortex trail. Consider the flow around a circular cylinder at Reynold's numbers above about 20, eddys break off alternately on either side in a periodic fashion. Behind the cylinder is a staggered, stable arrangement or trail of vortices. The alternate shedding produces a periodic force acting on the cylinder normal to the undisturbed flow. The force acts first in one direction and then in the opposite direction. Let f represent the frequency of this vibration in cycles per unit time, d the diamter (of the cylinder) and v the undisturbed velocity. Experiments have shown values of the dimensionless ration fd/v between 0.18 and 0.27.

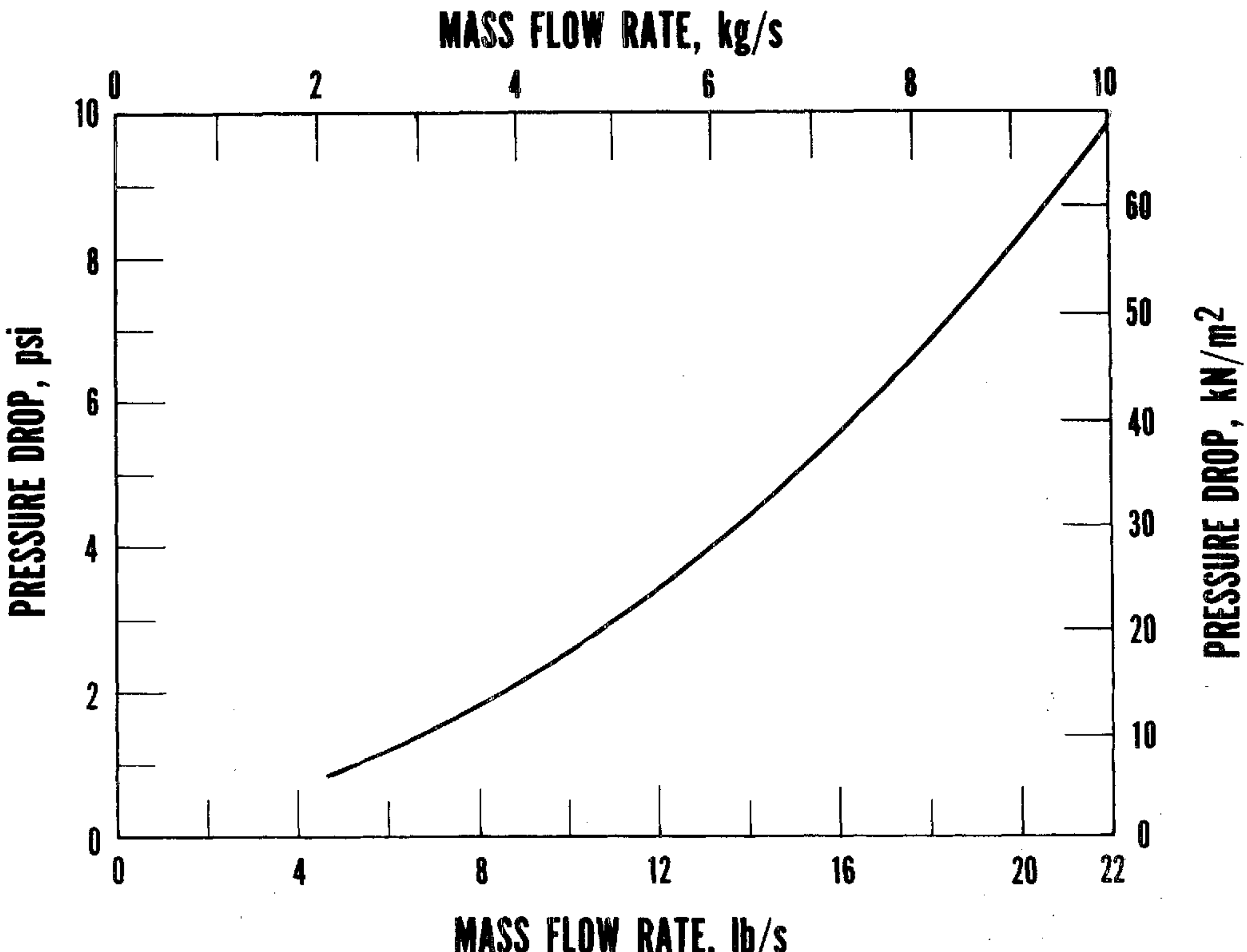


Figure 6.2. Pressure Drop - Turbine Meter - 1-1/4 inch - LN_2.

If the frequency of the vortex peeling approaches or equals the natural frequency of the elastic system consisting of the cylinder and its supports, the cylinder may have a small alternating displacement normal to the stream flow. The vibration of some smoke stacks, the vibration of some transmission lines, and the fatigue failure or progressive fracture of some transmission lines have been attributed to this resonance phenomenon."

A device based on this principle but adapted for closed conduit flow was submitted to the joint NBS-CGA flowmeter evaluation program [Dean, et al. (1968); Brennan, et al. (1971), (1972)]. Evaluation of this meter on liquid nitrogen by Brennan, et al (1973) is the only citation in the literature.

Design. The meter evaluated on liquid nitrogen by Brennan, et al. (1973) consisted of a bluff body located normal to the flow stream pipe. The flowmeter basically measures the average velocity of the flow passing through the meter. The frequency f of the vortex shedding is given approximately by

$$f = \frac{0.87V}{D} \text{ Hz}$$

where V is the velocity in feet per second and D is the nominal meter diameter in feet. The bluff body is in the shape of a modified delta with its base facing upstream.

The vortex sensors consist of electronically self heated resistance elements whose temperatures and therefore resistances vary as a result of the velocity variations adjacent to the body. These velocity variations reflect directly the action of the vortices as they peel off from the downstream edge of the bluff body.

Operational Characteristics. The meter reported by Brennan, et al. (1973) is illustrated in figure 6.3. The sensing element is in a stationary bluff body located in the flow stream. The vortices are generated by the bluff body at a rate that is dependent on the volumetric flow rate. The sensor detects the vortices and generates the signal which is treated electronically to yield the pulse output directly proportional to the volumetric flow.

The specifications supplied by the meter supplier are:

size - 1-1/2 inches (3.81 cm)
maximum flow rate - 120 gpm (7.57 dm^3/s)
minimum flow rate - 12 gpm (0.757 dm^2/s)
working pressure, 150 psi (1.03 MN/m^2)
pressure loss - 2.4 velocity heads
calibration accuracy - ±0.25 %
repeatability - better than ±0.1 %
linearity - ±0.5 %.

There are two outputs from the electronics supplied with this meter. One output gave one pulse per gallon (264.17 $pulse/m^3$) and was connected to an electric motor driven mechanical register. The second output gave 135 pulses per gallon (35,633 k $pulses/m^3$).

Two of the vortex shedding meters were tested. Tests were conducted at the liquid nitrogen flow prover located at NBS, Boulder, Colorado.

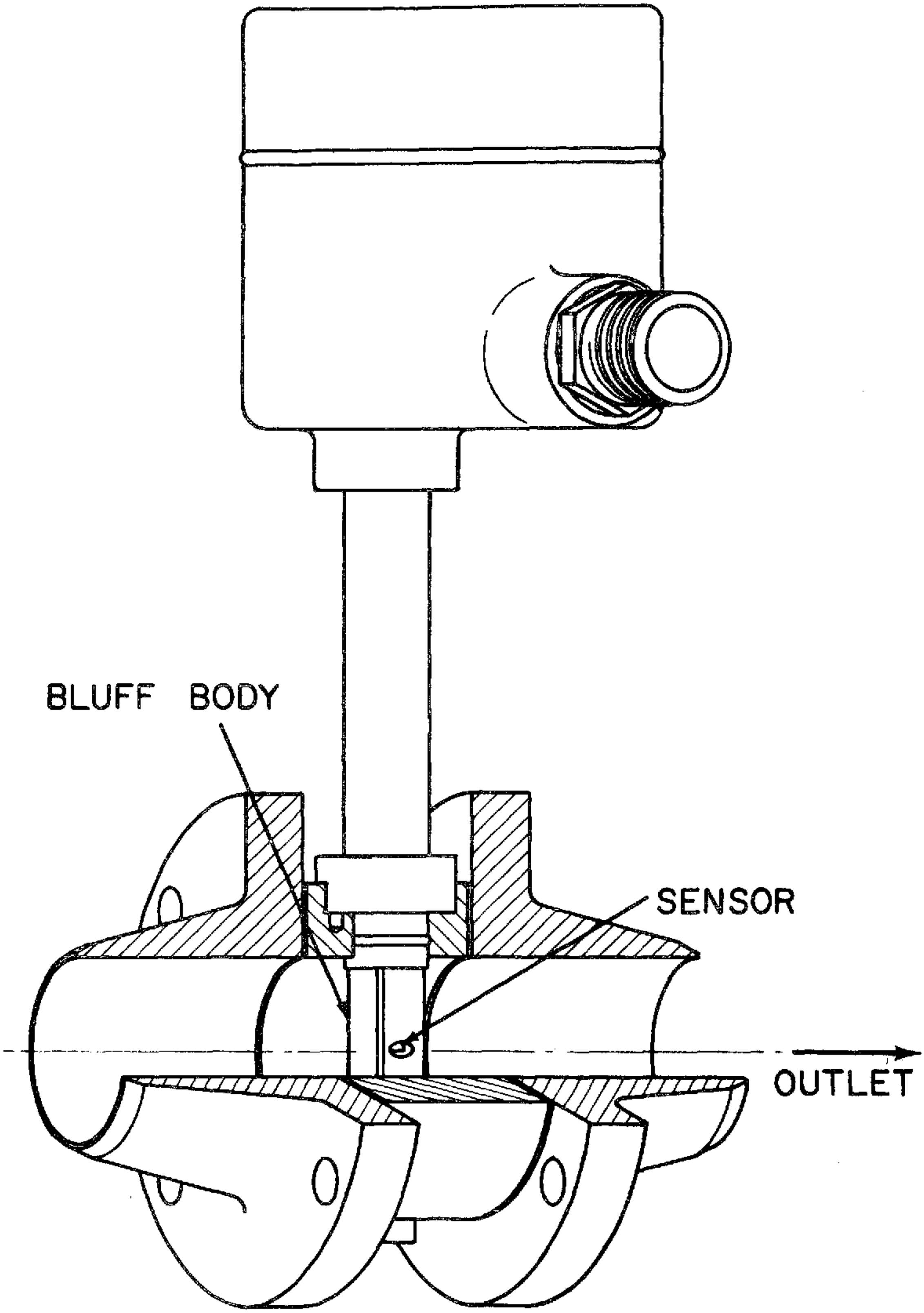


Figure 6. 3 Vortex Shedding Flowmeter.

An assessment of the performance of one of these vortex shedding meters is a precision (3σ) of ±0.45 percent, a bias of -0.82 percent and a minimum subcooling requirement of 6 K.

The range of linearity of the meter was found to be significantly less than that specified by the meter supplier. Rather than 10 to 1 over a range of 12 to 120 gallons per minute, the linear flow range for liquid nitrogen was 20 to 100 gallons per minute or a rangeability of 5 to 1. Pressure drop data on liquid nitrogen are shown in figure 6.4.

7. CONCLUSIONS

The state of the art of oxygen flowmetering instrumentation as described in the open literature is at best of marginal use to investigators charged with responsibility for measurements of oxygen flow with total uncertainties of less than one percent. At ambient temperatures and moderate pressures, standard flow measurement devices can be used with calibrations traceable to NBS. The most critical problem in the use of these measurement devices deals with material compatibility in pure oxygen service, a problem not unique to flow measurement.

At cryogenic temperatures where oxygen is in liquid form, specific applications have been made and described. These applications are in most cases undocumented in total uncertainty of flow measurement and therefore of only moderate interest.

A more serious problem is the use of surrogate fluids to describe the performance of a liquid oxygen flowmeter. Several attempts have been made to calibrate meters on water flow and transfer this calibration to liquid oxygen. These attempts have not met with success and in addition the errors of one to two percent are not systematic and suggest an almost total lack of understanding of why such a surrogate system fails with cryogenic fluids when it can be utilized quite well at ambient temperatures.

This problem of surrogate fluids is responsible for two other serious deficiencies in establishing the state of the art of oxygen flow instrumentation.

When water calibrations were found to be inadequate other cryogenic fluids were used, particularly liquid nitrogen, as a means of establishing the performance of the meter at cryogenic temperature but without the hazards of working with liquid oxygen. It is possible to show that no great error should be incurred between the meter performance on liquid nitrogen and liquid oxygen. This conclusion was drawn with little direct evidence based on meter performance -- at least in the open literature. A particular volumetric flowmeter design could be used on one or more cryogenic fluids with only a change in readout to reflect the density differences between fluids. No experimental verification of this practice is available in the literature. Even if this information were present it might only apply to a particular meter type and size.

The second problem caused by the surrogate water calibration failure was the inadequate documentation of the cryogenic reference standard developed for cryogenic meter calibrations. Only a few investigators describe their reference system in adequate detail to provide a base of confidence in the resulting meter performance. Only in one or two instances were meters tested on one reference system and then retested on a second system, as a method of determining the extent of systematic error between the two reference systems.

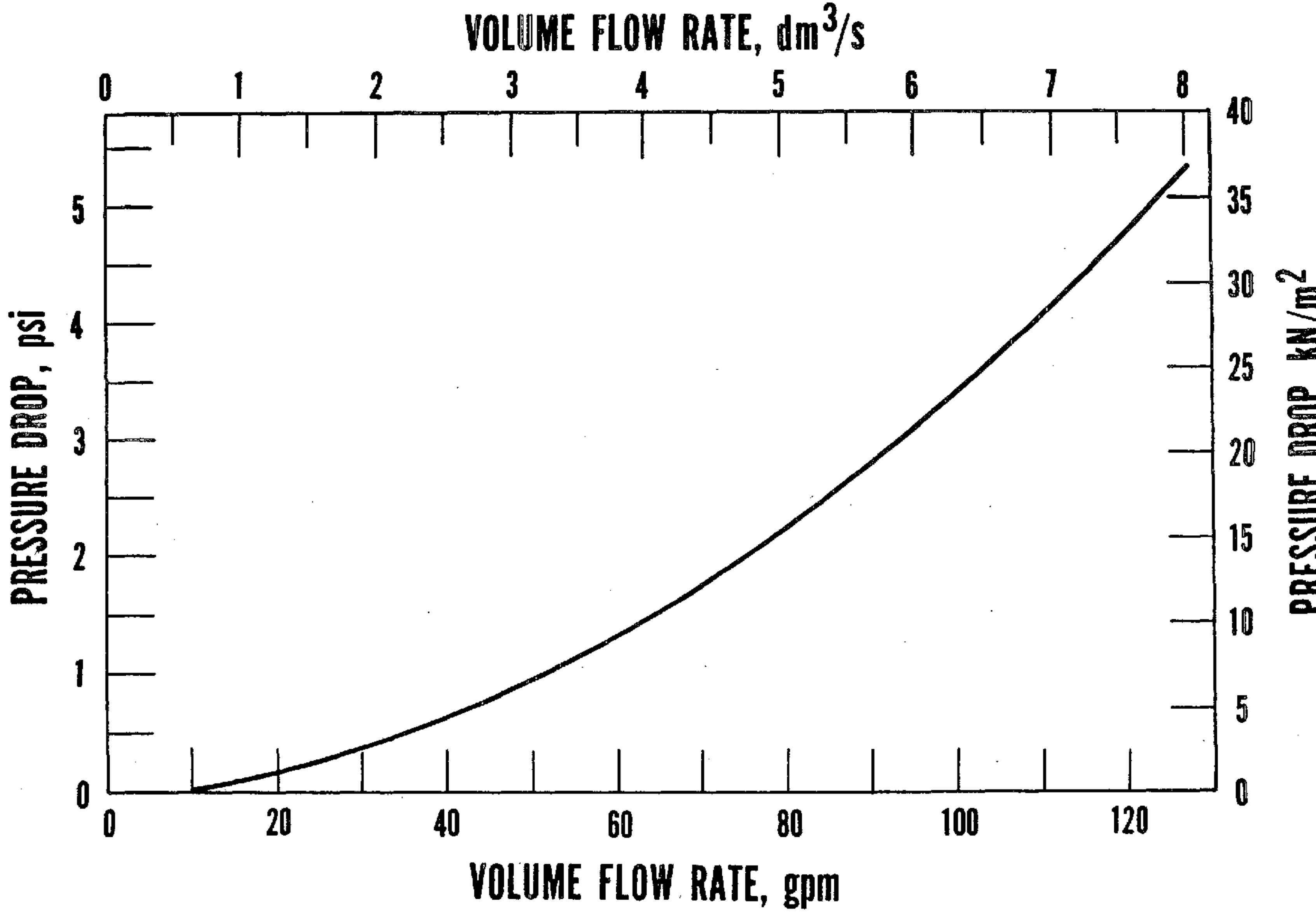


Figure 6.4 Pressure Drop - Vortex Shedding Meter - LN_2.

With these limitations in mind a summary of documented performance of cryogenic flowmeters is shown in figures 7.1 and 7.2. All data are based on cryogenic flow -- principally liquid nitrogen.

The following conclusions may be drawn from the two figures for documented cryogenic flow of liquid oxygen or a reasonable surrogate.

Turbine meters have the widest application from very small flow to intermediate flows of near 5000 gallons per minute. There seems to be no practical limit in size. Expected precision is good -- from ± 0.5 to ± 1.00 percent (3σ). Cooldown is a problem in that great care must be taken not to overspeed the turbine with gas flow. This can be accomplished by bypassing the meter during cooldown. Rangeability of individual meters is typically 10:1.

The moving parts are a potential hazard in oxygen service. Even with great attention to material selection and cleaning, failure of the turbine bearings could cause local heating -- possibly to the ignition point of the materials of construction or the ball bearing separators selected for cryogenic service.

The greatest problem in the application of turbine meters to any type of flow measurement is the rather fine balance between the hydrodynamic forces which drive the turbine wheel and the retarding forces of friction and fluid viscosity. The meter factor k (pulses/gallon) is a sensitive measure of the balance of these forces and only slight changes in the bearing operation or slight damage to the blading caused by impurities can change the meter factor slightly. A meter factor shift of this type will change the accuracy but not necessarily change the precision and the operator is unaware of the shift until recalibration. One method of guarding against such a set of circumstances is to place two meters in series during flow measurement period or periodically as a test.

The momentum meter cited in this report is a rather new entry into cryogenic flow measurement. Its main advantage is a direct reading of mass flow. Indicated precision (3σ) is good and may even be improved relative to volumetric meters when the density measurement uncertainty is added to the volumetric flow to give inferred mass flow.

Current designs are apparently limited to the indicated flow range (for liquid nitrogen). Rather extensive redesign would be necessary to increase or decrease the flow range.

The safety problems with this meter may be somewhat more favorable than the turbine after proper selection and cleaning of the materials of construction. The fluid driven (or retarded) blading is not subject to overspeeding during cooldown and therefore the support bearing is less liable to damage or over heating. Gross shifts in the meter factor k (pulses/gal) may be detected at zero flow by comparing the speed of the driven member with that of the driving motor, an advantage over the turbine meter. The meter is in commercial service in liquid oxygen.

Vortex shedding as a metering method is even more recent than the above cited momentum meter [Rodely, 1969]. It has the advantage of having no moving parts at the low temperatures and therefore requires minimum maintenance. The meter factor (pulses/gallon) depends only on the inside diameter of the pipe and the width across the bluff body face [Rodely, 1969]. Cryogenic evaluation is limited but performance on liquid nitrogen and oxygen show some deviations from ambient service. The linear flow range is decreased from 10 to 1 (specified by the manufacturer) to about 5 to 1 with liquid nitrogen.

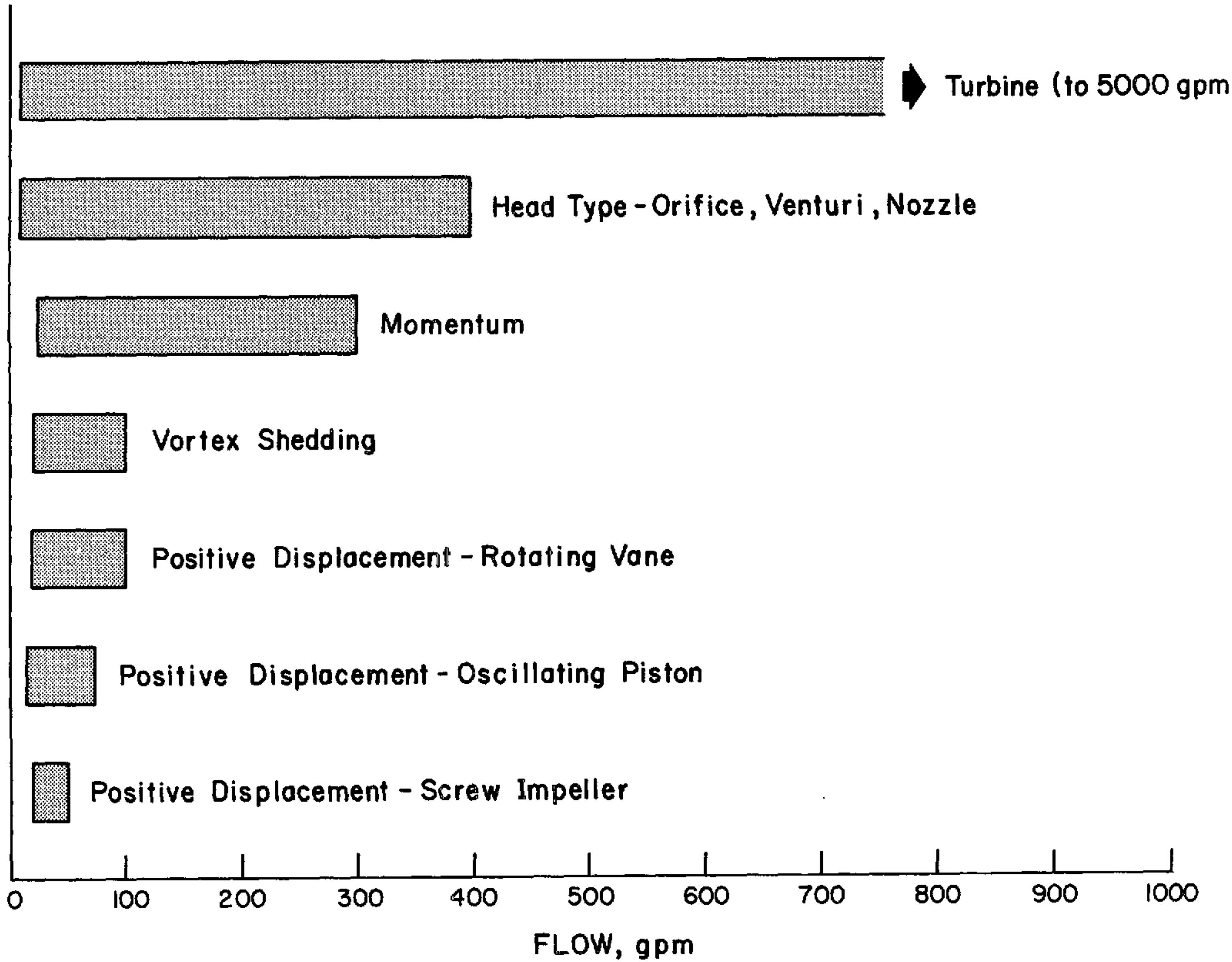


Figure 7. 1 Flow Ranges of Cryogenic Meters - Experimental Data from the Literature.

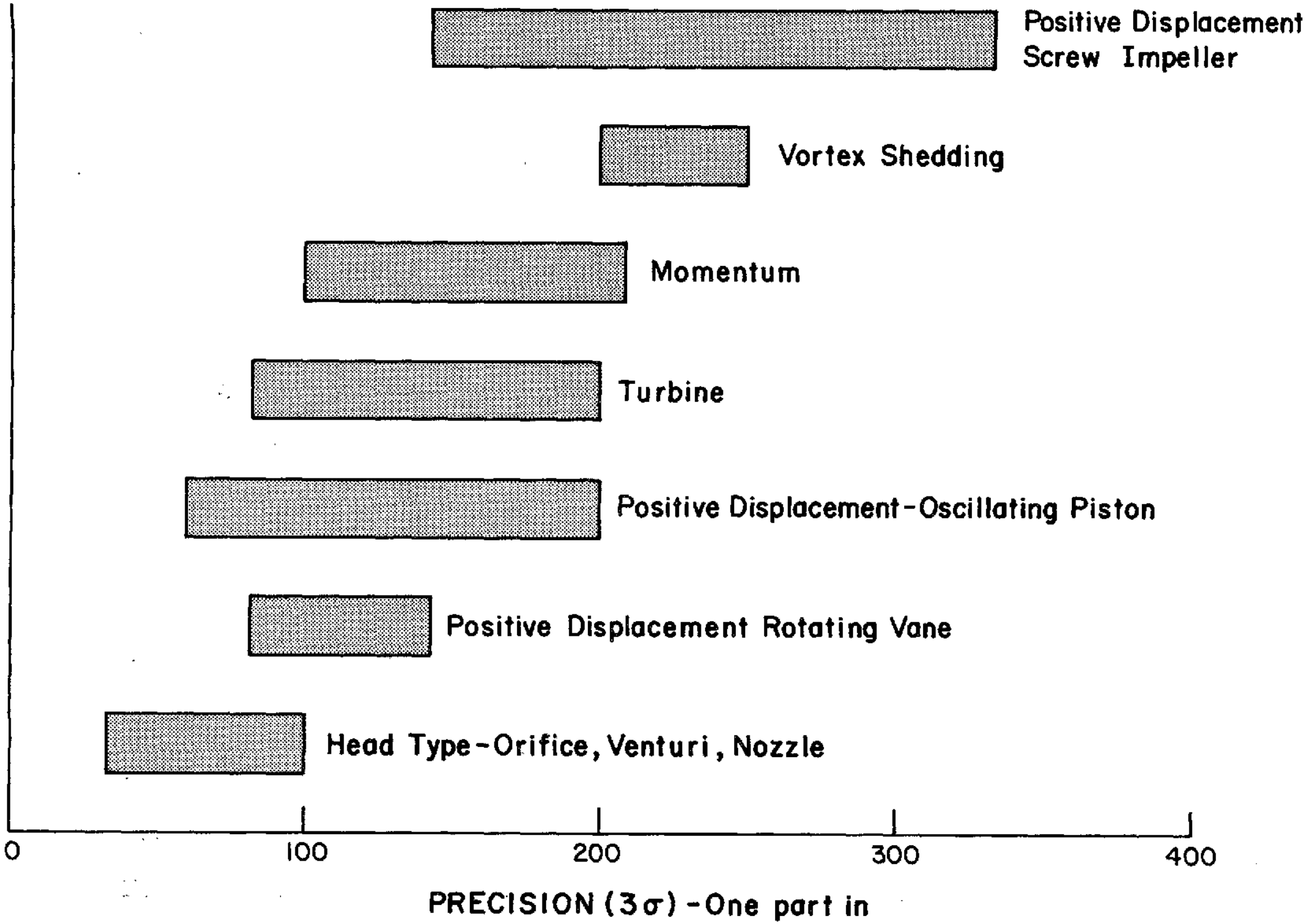


Figure 7.2 Precision of Cryogenic Flowmeters - Experimental Data from the Literature.

This reduction in rangeability may be caused by changes in relative positions of the sensor and location of maximum vortex effect caused by thermal contraction and in the signal treatment in the secondary or output electronics. These and other possibilities are presently under investigation.

Precision of this device is quite good, ±0.4 to ±0.5 percent, and should be considered in future work with liquid or gaseous oxygen. Operation is dependent apparently only on geometry and therefore material compatibility problems should be simplified. No moving parts at operating temperature should eliminate cool down problems with liquid service. Primary interest should be centered on vortex detection methods as to long term stability and internal location of the sensor. Meters for large diameter pipe flow seem not to be a problem as models to 36 inches in diameter are available.

Head-type flowmeters have been used extensively for gas measurement but have not shown great promise for cryogenic service. Precision of reported cryogenic meters is relatively poor (±1 - ±3%) when compared with other methods and even with ambient temperature gas service.

The advantages of this meter concept for cryogenic service (no moving parts, standard design, simplicity, etc) should continue to be considered when selecting measurement methods. Evidence is available that suggests the lack of precision may be caused by the methods of pressure detection and measurement. Conventional pressure taps and lines may not be a suitable method of output signal detection. Direct pressure measurement by electronic or mechanical transducers located at or within the meter should be investigated in future work.

The general class of meters designated as positive displacement types are restricted in size and flow capacity merely by the bulk and inertia of the primary measurement elements. Precision is good (±0.3 to ±0.6%), units are generally rugged and dependable in service. It is believed that any future improvements will be minor in nature as the presently available models satisfy most requirements of the intended service. Some work could be done on temperature compensation (density measure) and vapor elmination during cooldown for operation with cryogenic fluids.

8. RECOMMENDATIONS

In order to take full advantage of the current state of the art of oxygen flow measurement it is necessary first to determine the actual total uncertainty required of the measurement system. From the literature it is apparent that if the total uncertainty in flow need not be less than ±2 percent then standard methods of calibration will be sufficiently accurate and precise. These standard methods would include study and analysis of materials compatibility with oxygen, design and construction for the intended medium (ambient, high pressure, cryogenic), cleaning for oxygen service and identification of the calibration method as to flow range or other pertinent operational conditions. Any decision to seek a total uncertainty of less than 2 percent should be based on a realistic value of the use of the information desired from the flow measurement system. If this realistic assessment is not made then a great deal of time, effort and resources can be wasted in attempting to provide a precision and accuracy unjustified by the ultimate use of the measurement device.

If on the other hand analysis has shown that measurements must be made to a total uncertainty of less than ±2 percent, the existing state of the art for oxygen service leaves a great deal to be desired. The general field of flow measurement of oxygen as a cryogenic fluid has been studied extensively [ISA, 1967] and numerous proposals have been made for study of this special measurement problem [NBS, 1963]. Most efforts at measurement systems capable of providing total uncertainty of less than ±2 percent have been project oriented, designed to solve a specific measurement problem using specific types of measurement devices. When the need for such a measurement system has passed, the facilities, techniques, and experience developed under these programs has in general been lost. This may well be because the requirements for oxygen measurement flow systems are intermittent. A measurement system capable of total uncertainties of less than ±2 percent cannot be provided without a long term commitment in resources and programs.

Current problem areas in the measurement of cryogenic oxygen flow can be summarized by the following recommendations.

8.1. Flow Reference Systems

A study should be initiated to identify those systems that are currently available, documentation of the flow range, pressure and temperature capabilities, precision and accuracy relative to basic standards and multiple test fluid capability. After establishing location and capability, a system of interlaboratory comparisons should be initiated by exchange of test meters and the monitoring of the performance of these test meters on each reference system. This interlaboratory comparison using reference test meters would identify systematic errors existing with respect to individual facilities relative to the entire reference measurement system.

8.2. Surrogate Fluids

A concerted program should be initiated designed to determine the appropriateness of the use of surrogate fluids for oxygen flowmetering calibrations. This would include a program designed to determine the reasons for water calibration errors experienced at cryogenic temperatures as well as experimental evidence that other fluids such as liquid nitrogen or liquid argon may be used as a test fluid in place of liquid oxygen.

8.3 Large Flow

General purpose flow reference systems are at the present time restricted to less than 200-300 gallons per minute. The trend for oxygen flow measurement is to higher and higher flow rates in excess of these values. The uniqueness of environment, the severity of constraints placed on design and operation and the requirements both economical and technological for mass flow measurements to total uncertainties of less than one percent, suggest that a fresh look be taken to the establishment of a general purpose cryogenic flow research center. Such a center would include as process fluids: water, liquid oxygen, liquid methane, LNG, liquid nitrogen, liquid argon, and liquid hydrogen. Flow range should extend to 60,000 gpm with provision for pressure and temperature controls.

A cryogenic flow research center of this size and scope would impact on present oxygen and hydrogen aerospace requirements, short term (10-20 year) energy importation programs (LNG), and long term requirements of a hydrogen-based energy system. The latter has tremendous potential in providing and distributing energy in an environmentally attractive form using many existing fossil fuel type systems, but will require high density liquid or slush storage systems to make it a workable concept in certain cases.

8.4. Uniform Performance Method

Standard tests or codes should be developed and adopted where possible for oxygen flow measurement. Number of experimental points, treatment of experimental data and reporting of error should be standardized to provide common and effective criteria for meter evaluation.

9. BIBLIOGRAPHY

9.1. General References

Allingham, W. D., Zero-Gravity Expulsion of Cryogens with Metal Bellows, Low Gravity Propellant Orientation and Expulsion (Proc. of Joint Symp. of Low Gravity Propulsion Orientation and Expulsion, AIAA and Aerospace Corp., Los Angeles, CA, May 21-23, 1968) 6, Western Periodicals, North Hollywood, CA (1968).

Alspach, W. J., C. E. Miller, and T. M. Flynn, Mass Flowmeters in Cryogenic Service, Cryogenics Div., National Bureau of Standards, Boulder, CO; Flow Measurement Symposium, ASME Flow Measurement Conf., Pittsburgh, PA (Sept. 1966).

ASME, Fluid Meters, Their Theory and Applications, 6th Ed. (ASME, New York, NY, 1971).

Bankaitis, H., and C. F. Schueller, ASRDI Oxygen Technology Survey. Vol. II - Cleaning Requirements, Procedures, and Verification Techniques, NASA Spec. Publ. 3072, Aerospace Safety Research and Data Institute, Lewis Research Center, Cleveland, OH, 76 pages (1972).

Binder, R. C., Fluid Mechanics, 2nd Ed., 361 pages (Prentice-Hall, Inc., New York, NY, 1949).

Bowers, K., R. L. Galley, and P. S. Vincelett, Flow Measurement in Rocketry, Instr. Control Syst. 34, 638-649 (April 1961).

Brennan, J. A., R. W. Stokes, D. B. Mann, and C. H. Kneebone (to be published 1973).

Close, D. L., Flow Measurement of Cryogenic Fluids, Instr. Control Syst. 41, No. 2, 109-114 (Feb. 1968).

Close, D. L., Cryogenic Flowmeters - Today, Cryogenics and Industrial Gases (Aug. 1969).

Crawford, R. B., A Broad Look at Cryogenic Flow Measurement, Instr. Soc. of Amer. J. 10, No. 6, 65-72 (June 1963).

Cunningham, R. E., and W. J. Anderson, Evaluation of 40-Millimeter-Bore Ball Bearings Operating in Liquid Oxygen at DN Values to 1.2 Million, NASA Tech. Note TN D-2637 (Jan. 1965).

Eisenhart, C., Realistic Evaluation of the Precision and Accuracy of Instrument Calibration Systems, J. Res. Nat. Bur. Stand. 67C, No. 2 (April-June 1962).

Galley, R. L. The State of the Art in Cryogenic Flowmetering, 17th Ann. Instrument Society of America Conf. (Oct. 1962).

Galley, R. L., Aerospace Flow Metrology, Douglas Aircraft Co., Instr. Control Syst. 39, No. 12, 113-117 (Dec. 1966).

Hilsenrath, J., Thermal Properties of Gases, Nat. Bur. Stand. Circ. 564 (1955).

Hayakawa, K. K., D. R. Udell, M. M. Iwata, C. F. Lytle, R. M. Chrisco, C. S. Grennough, and J. A. Walling, Cryogenic Flow Measurement and Cryogenic Liquid Detection Measurement Technology, Report SD 72-SA-0156-2, Space Division, North American Rockwell Corp., Downey, CA (Oct. 13, 1972).

ISA, Ad Hoc Committee Report on Flow Measurement, Instrument Society of America, Pittsburgh, PA (1967).

ISA, Standard Practices for Instrumentation, Instrument Society of America, Pittsburgh, PA (1970).

Kunkle, J. S., S. D. Wilson, and R. A. Cota (Eds.), Compressed Gas Handbook, SP-3045, National Aeronautics and Space Administration, Washington, DC (1969).

Laub, J. H., Read Mass Flow Directly with Thermal Flowmeters, Control Eng. 13, No. 4, 69-72 (April 1966).

Mandel, J., The Statistical Analysis of Experimental Data, Interscience Publishers, John Wiley and Sons, New York, NY (1967).

Mann, D. B., Cryogenic Flow-Metering Research at NBS (National Bureau of Standards, Boulder, CO, Institute for Basic Standards), Cryogenics 11, No. 3, 179-185 (June 1971).

Mann, D. B., D. E. Diller, N. A. Olien, and M. J. Hiza, Measurements of Liquefied Natural Gas in Commerce, American Gas Association Distribution Conf., Washington, DC (May 1973).

Mason, H. L. (Ed.), Calibration and Test Services of the National Bureau of Standards, Nat. Bur. Stand. Spec. Pub. 250 (1968).

Reynolds, R. W., Sundstrand Model 876 APU. Part I. Cryogenic Fluids, Rept. SA/AER-474, AFAPL-TR-67-49, Contr. AF33(615)-2474, Sundstrand Aviation, Rockford, IL (March 1967).

Rodely, A. E., Vortex Shedding Flowmeter, Measurements and Data, No. 18 (Nov.-Dec. 1969).

Roder, H. M., Liquid Densities of Oxygen, Nitrogen, Argon and Para Hydrogen, Nat. Bur. Stand. Tech. Note 361, revised (1973).

Ruegg, F. W., and M. R. Shafer, Flow Measurement: Procedures and Facilities at National Bureau of Standards, Flow Meas. Symp., San Francisco, CA; Amer. Soc. Heat. and Air-Cond. Eng., Inc., New York, NY (Jan. 1970).

Schuler, A. E., L. Thompson, C. Gibbs, et al., Liquid Hydrogen Mass Flowmeter Evaluation, Final Report NASA CR-60990, Contr. No. NAS8-1526, Wyle Labs, El Segundo, CA (Jan. 1965).

Tenenbaum, D. M., and D. H. Clark, Activation and Initial Test Operations of Large Rocket Engine-Thurst Chamber Test Facilities (Aerojet-General Corp., Sacramento, CA), 3rd AIAA Propulsion Joint Specialist Conf., Paper No. AIAA 67-455, Washington, DC (July 17-21, 1967).

Wildhack, W. A., R. C. Powell, and H. L. Mason, Accuracy in Measurements and Calibrations, 1965 Nat. Bur. Stand. Tech. Note 262 (1965).

Wyle Laboratory, Liquid Hydrogen Mass Flowmeter Evaluation, Final Report NASA CR-60990, Contr. No. NAS8-1562 (1965).

Bucknell, R. L., Calibration Systems and Turbine-Type Flow Transducers for Cryogenic Flow Measurements, Advances in Cryogenic Engineering 8, 360-370 (Proc. of the 8th Cryogenic Eng. Conf., Los Angeles, CA, Aug 14-16, 1962), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1963).

Bucknell, R., Cryogenic Flow Measurement. Flow Meters, Cryogenic Eng. News 1, No. 6, 28-30 (April 1966).

Bucknell, R., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement. Part I - LH_2, Cryogenic Eng. News 1, No. 4, 26-28 (Feb. 1966).

Bucknell, R., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement. Part II - LOX Systems, Pratt and Whitney Aircraft, Cryogenic Eng. News 1, No. 5, 30-34 (March 1966).

Bucknell, R. L., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement, Proc. 19th Ann. Instrument Society of America Conf. 19, Instrument Society of America, New York, NY (1964).

Cryogenic Flow Measurements, Tech. Memo No. 149 LRP, Contr. No. AF 04(645)8, Aerojet-General Corp., Sacramento, CA (Jan. 1962).

Dean, J. W., J. A. Brennan, and D. B. Mann, Cryogenic Flow Research Facility of the National Bureau of Standards, Advances in Cryogenic Engineering 14, (Proc. of the 14th Cryogenic Eng. Conf., Cleveland, OH, Aug 19-21, 1968), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1969).

Dean, J. W., J. A. Brennan, D. B. Mann, and C. H. Kneebone, Cryogenic Flow Research Facility Provisional Accuracy Statement, Nat. Bur. Stand. Tech. Note 606 (1971).

Deppe, G. R., Large Size Cryogenic Turbine Type Flowmeter Technology, Rept. No. NASA CR-54810, AGC 8800-60, Contr. No. NAS3-2555, Aerojet-General Corp., Sacramento, CA (June 1966).

Deppe, G. R., and R. H. Dow, The Design, Construction, and Operation of a Cryogenic Flow Calibration Facility, Advances in Cryogenic Engineering 8, 371-377 (Proc. of the 8th Cryogenic Eng. Conf., Los Angeles, CA, Aug. 14-16, 1962), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1963).

Groeper, L. H., In-Place Calibration of Cryogenic Flowmeters for Rocket Engine Testing, Advances in Instrumentation 24, Pt. 2 (Proc. of the 24th Ann. Instrument Society of America Conf., Houston, TX, Oct. 27-30, 1969), Instrument Society of America, Pittsburgh, PA (1969).

Mann, D. B., J. W. Dean, J. A. Brennan, and C. H. Kneebone, Cryogenic Flow Research Facility (unpublished NBS Report, Jan. 1970).

Mann, D. B., Cryogenic Flow-Metering Research at NBS (National Bureau of Standards, Boulder, CO, Institute for Basic Standards), Cryogenics 11, No. 3, 179-185 (June 1971).

Mortenson, L. N., The Evaluation of a Cryogenic Test Facility, Cryogenic Eng. News 2, No. 7, 16-21 (July 1967).

NBS Probes Cryogenic Flowmetering, Cryogenics Industrial Gases 5, No. 9, 24-27 (Nov.-Dec. 1970).

National Bureau of Standards, Recommendations to National Aeronautics and Space Administration Concerning a Flow Meter Calibration Facility (unpublished NBS Report, 1963).

National Bureau of Standards, Handbook 44, 3rd Ed. (1971), as amended.

Revelino, J. B., Techniques and Methods for the Calibration and Standardization of Transducers for Rocket Testing, Rept. No. Idep 347.70.00.00-A6-03, Aerojet-General Corp., Sacramento, CA (June 1962).

Shamp, F. F., New Device Proves Accuracy of Cryogenic Liquid Measurement, Pipe Line Ind. 32 No. 1, 41-44 (Jan. 1970).

Shamp, F. F., Bi-Directional Piston Prover Tests Turbines for Cryogenic Liquid Measurement, Pipeline Gas J. 197, No. 10, 64-71 (Aug. 1970).

9.3. Positive Displacement

Angerhofer, A. W., Cryogenic Instrumentation. II. Sensing Flow and Composition, Control Eng. 12, No. 11, 77-84 (Nov. 1965).

Brennan, J. A., J. W. Dean, D. B. Mann, and C. H. Kneebone, An Evaluation of Positive Displacement Cryogenic Volumetric Flowmeters, Nat. Bur. Stand. Tech. Note 605 (July 1971).

Fox, A., Cryogenic Flow Measurement for Custody Transfer, National Cylinder Gas Div., Chemetron Corp., Chicago, IL, ASME Winter Annual Meeting, Pittsburgh, PA (Nov. 12-17, 1967).

Mann, D. B., J. W. Dean, J. A. Brennan, and C.H. Kneebone, Cryogenic Flow Measurement-Positive Displacement Cryogenic Flowmeters (NBS Cryogenics Div., Institute for Basic Standards, Boulder, CO), Symposium on Flow - Its Measurement and Control in Science and Industry, Pittsburgh, PA (May 9-14, 1971).

Tantam, D. H. and A. Allen, A Volumetric Flowmeter for Liquid Oxygen, Advances in Cryogenic Engineering 5, 299-306 (Proc. of the 5th Cryogenic Eng. Conf., Sept. 2-4, 1959), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1960).

Thayer, R. M., Process of and Apparatus for Metering and Liquefied Gas, U.S. Patent 2,610,471, Union Carbide and Carbon Corp., New York, NY (Sept. 16, 1952).

Williams, D. A., Systems for Metering Liquefied Gas, U.S. Patent 3,093,973, Chemetron Corporation, Chicago, IL (June 18, 1963).

9.4. Head Type

Angerhofer, A. W., Cryogenic Instrumentation. II. Sensing Flow and Composition, Control Eng. 12, No. 11, 77-84 (Nov. 1965).

Bonnett, F. W., Critical Two Phase Flow of Nitrogen and Oxygen Through Orifices, Advances in Cryogenic Engineering 12, 427-437 (Proc. of the 12th Cryogenic Eng. Conf., June 13-15, 1966), K. D. Timmerhaus (Ed.), Plenum Press, Inc., New York, NY (1967).

Bucknell, R., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement. Part I - LH_2, Cryogenic Eng. News 1, No. 4, 26-28 (Feb. 1966).

Bucknell, R., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement. Part II. LOX Systems, Pratt and Whitney Aircraft, Cryogenic Eng. News 1, No. 5, 30-34 (March 1966).

Bucknell, R. L., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement (Proc. of the 19th Ann. Instrument Society of America Conf.) 19, Instrument Society of America, New York, NY (1964).

Burchflubmessung mit genormlen Düsen, Blenden und Venlurediisen, DIN 1952, Mai 1969, Beuth-Verlag, D-1 Berlin 30.

ISO Recommendation R-541, Measurement of fluid flow by means of orifice plates and nozzles (Jan. 1967).

Johnson, R. C., Real Gas Effects in Flow Metering, Lewis Research Center Tech. Memo X-52965, National Aeronautics and Space Administration, Cleveland, OH (Presented at the Symposium on Flow - Its Measurements and Control in Science and Industry, Pittsburgh, PA, May 10-14, 1971)(1971).

Kunkle, J. S., S. D. Wilson, and R. A. Cota (Eds.), Compressed Gas Handbook, SP-3045, National Aeronautics and Space Administration, Washington, DC (1969).

Morris, T. F., Flowmeter for Space, George C. Marshall Space Flight Center Tech. Note No. D-5517, National Aeronautics and Space Administration, Huntsville, AL (Dec. 1969).

Purcell, J. R., A. F. Schmidt, and R. B. Jacobs, The Venturi Tube as a Liquefied-Gas Flow Measuring Device, Advances in Cryogenic Engineering 5, 282-288 (Proc. of the 5th Cryogenic Eng. Conf., Sept. 2-4, 1959), K. D. Timmerhaus (Ed.), Plenum Press, Inc., New York, NY (1960).

Richards, R. J., R. B. Jacobs, and W. J. Pestalozzi, Measurement of the Flow of Liquefied Gases with Sharp-edged Orifices, Advances in Cryogenic Engineering 4, 272-285 (Proc. of the 4th Cryogenic Eng. Conf., Aug. 19-21, 1958), K. D. Timmerhaus (Ed.), Plenum Press, Inc., New York, NY (1960).

Wenzel, H., Improvement of Flow Measurement of Cryogens, IIR, Washington, DC (1971).

9.5. Momentum Mass Flow Meter

Close, D. L., R. M. Hayford, A. L. Carroll, Jr., and M. Rubin, Cryogenic Mass Flow Meter, Symposium on Flow - Its Measurement and Control in Science and Industry, Pittsburgh, PA (1971).

Pearson, D.B., W.H. Boyland, C. F. Taylor, and H. R. Wheelock, Cryogenic Mass Fuel Flow Transmitters, Tech. Rept. Contr. No. AF 33-657-9786 (Dec. 1962), Gen. Elec. Co., Lynn, MA.

Taylor, C. F., and D. B. Pearson, A True-Mass Cryogenic Flowmeter, ISA Trans. 5, No. 3, 203-210 (July 1966).

9.6. Turbine

Alspach, W. J., and T. M. Flynn, Considerations when using Turbine Type Flowmeters in Cryogenic Service, Advances in Cryogenic Engineering 10, 246-252 (Proc. of the 10th Cryogenic Eng. Conf., Philadelphia, PA, Aug. 18-21, 1964), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1965).

Bell, J. H., Jr., Measuring Flow of Cryogenic Fluids in a Liquid Rocket System, Cryogenic Technol. 1, No. 3, 115-116 (Chrysler Corp., Space Div.) (Mar.-April 1965).

Brennan, J. A., R. W. Stokes, D. B. Mann, and C. H. Kneebone, An Evaluation of Several Cryogenic Turbine Flowmeters, Nat. Bur. Stand. Tech. Note 624 (1972).

Bucknell, R. L., Calibration Systems and Turbine-Type Flow Transducers for Cryogenic Flow Measurements, Advances in Cryogenic Engineering 8, 360-370 (Proc. of the 8th Cryogenic Eng. Conf., Los Angeles, CA, Aug. 14-16, 1962), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1963).

Bucknell, R., Cryogenic Flow Measurement. Flow Meters, Cryogenic Eng. News 1, No. 6, 28-30, 43 (April 1966).

Bucknell, R., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement. Part I - LH_2, Cryogenic Eng. News 1, No. 4, 26-28 (Feb. 1966).

Bucknell, R., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement. Part II - LOX Systems, Cryogenic Eng. News 1, No. 5, 30-34 (March 1966).

Bucknell, R. L., T. D. Lowler, and R. L. Street, Cryogenic Flow Measurement, Proc. of the 19th Ann. Instrument Society of America Conf. 19, Instrument Society of America, New York, NY (1964).

Cryogenic Flow Measurements, Tech. Memo No. 149 LRF, Contr. No. AF 04(645)8, Aerojet-General Corp., Sacramento, CA (Jan. 1962).

Cryogenic Mass Flow System, Quantum Dynamics, Tarzana, CA, Instr. Control Syst. 38, No. 7, (July 1965).

Deppe, G. R., Large Size Cryogenic Turbine Type Flowmeter Technology, Rept. No. NASA CR-54810, AGC 8800-60, Contr. No. NAS 3-2555, Aerojet-General Corp., Sacramento, CA (June 1966).

Deppe, G. R., and R. H. Dow, The Design, Construction, and Operation of a Cryogenic Flow Calibration Facility, Advances in Cryogenic Engineering 8, 371-377 (Proc. of the 8th Cryogenic Eng. Conf., Los Angeles, CA, Aug. 14-16, 1962), K. D. Timmerhaus (Ed.), Plenum Press, New York, NY (1963).

Fox, A., Cryogenic Flow Measurement for Custody Transfer, National Cylinder Gas Div., Chemetron Corp., Chicago, IL, ASME Winter Ann. Meet., Pittsburgh, PA (Nov. 12-17, 1967).

Grey, J., The Turbine Flowmeter for Cryogenic Liquids, 14th Ann. Instrument Society of America Conf., Chicago, IL (Sept. 21-25, 1959).

Groeper, L. H., In-Place Calibration of Cryogenic Flowmeters for Rocket Engine Testing, Advances in Instrumentation 24, Pt. 2 (Proc. of the Ann. Instrument Society of America Conference, Houston, TX, Oct. 27-30, 1969), Instrument Society of America, Pittsburgh, PA (1969).

Thompson, R. C., and J. Grey, Turbine Flowmeter Performance Model, Final Rept. AMC-3, U.S. Army Missile Comm., Redstone Arsenal, Alabama, Greyrod Corp., Princeton, NJ (1967).

Turbine-Type Flowmeters for Cryogenic Service, Nat. Bur. Stand. Tech. News Bull. 49, No. 7, 103 (July 1965).

Chapter I-5*

Differential Pressure Meters: Theory of Fluid Flow in Terms of Differential Pressures and Equations for Differential Pressure Meters

I-5-1 Principal primary elements. In the differential pressure group these are the Venturi tube, the flow nozzle, and the thin-plate square-edged orifice. Other primary elements in this group, which are discussed in this chapter, are the nozzle-Venturi and other modifications of the Venturi tube, the quadrant-edged orifice, and eccentric and segmental orifices. In addition, there are the centrifugal (elbows), linear resistance (capillary tube and porous plug), and frictional resistance (pipe sections). The distinctive feature of this group of meters is that there is a marked pressure difference or pressure drop associated with the flow of a fluid through the primary element and that this pressure difference can be measured and related to the mass or volume rate of flow. Hence, the designation *differential pressure* meters. The theoretical considerations used as a basis for computing the rate of flow from the pressure measurements are the same for all meters in the group except for the last three. While the characteristics of fluid flow through all of the primary elements have been studied to some extent, such studies and tests with the Venturi tube (Fig. I-5-1), the flow nozzle (Fig. I-5-2), and square-edged orifice (Fig. I-5-3), have been very extensive. From such extensive studies the discharge coefficients and expansion factors for these three primary elements have been so well established that these meters are used extensively, even for important measurements, without calibration. Instructions on the construction, installation and operation of these primary elements are given in Part II.

I-5-2 Theory of the Flow of Fluids in Terms of Pressure Differences. In the following discussion and development of equations, a primary assumption is: the mass flow rate is constant with respect to a considerable period of time (e.g., 5 to 10 minutes or more), and the flow is steady. In the past, when the Bourdon gage or a liquid manometer were the principal pressure-indicating instruments, the adjective, "steady," implied that there were no noticeable periodic or cyclic pressure variations. Any momentary movements of the gages were entirely random and transitory, and thus the readings of the gages could be taken as sensibly steady and represented correctly the movement of the fluid. Today, with the high-speed pressure transducers and recorders that lack the inertia-dampening characteristics of the older gages, the sense of steadiness may be obscured. For this reason, a better statement of the requirement is that the flow is not subject to *pulsations* as that term is defined in Chapter I-3, Par. I-3-45.

I-5-3 The following letter symbols will be used throughout this chapter:

A	= Area of first or upstream section	sq ft or sq in.
a	= Area of second or downstream section	sq ft or sq in.
C	= Coefficient of discharge	ratio
D	= Diameter of pipe at upstream section	ft or in.

* ASME, 1972

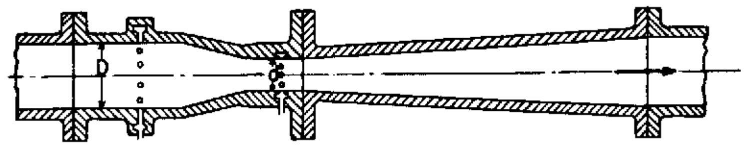


FIG. I-5-1 VENTURI TUBE—HERSCHEL TYPE

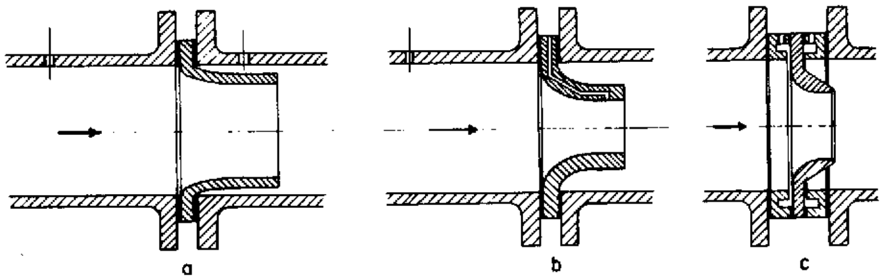


FIG. I-5-2 THREE FLOW-NOZZLE SHAPES AND LOCATIONS OF PRESSURE TAPS

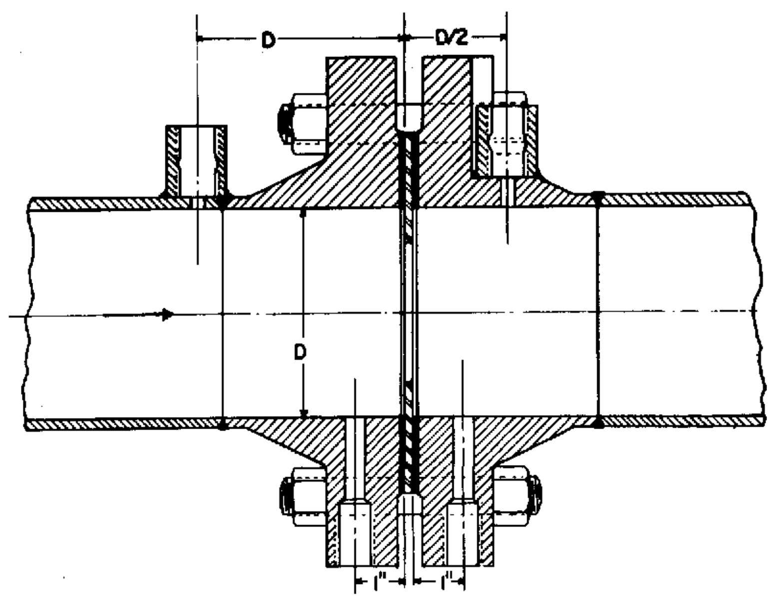


FIG. I-5-3 THIN-PLATE SQUARE-EDGED ORIFICE MOUNTED BETWEEN FLANGES. (TWO PAIRS OF PRESSURE TAP LOCATIONS ARE SHOWN; EITHER PAIR MAY BE USED, AND THE OTHER PAIR OMITTED OR PLUGGED. ALSO, THERE ARE SEVERAL DESIGNS OF SPECIAL FITTINGS IN WHICH ORIFICE PLATES ARE MOUNTED.)

d	= Diameter of primary element throat	ft or in.
E	= Velocity of approach factor = $1/\sqrt{1-\beta^4}$	ratio
F_a	= Thermal expansion factor (of metals)	ratio
G	= Specific gravity	ratio
g	= Local acceleration of gravity	ft/sec^2
g_c	= Proportionality constant = 32.174	
H	= Enthalpy = $p/\rho + u_i$	ft lb_f/lb_m
h	= Effective differential pressure	ft of fluid
J	= Mechanical equivalent of heat	778.16 ft lb_f/Btu
K	= Flow coefficient = CE	ratio
M	= Mach number	ratio
MW	= Molecular weight	
m	= Mass rate of flow	lb_m/sec
p	= Pressure, absolute pressure unless stated otherwise	psia or psfa
q	= Volume rate of flow	cfs
q_H	= Heat transferred to or from the fluid	ft lb_f/lb_m
R	= Universal gas constant (see Note)	1545.32 ft.lb_f /°R mole lb_m
R_g	= Individual gas constant = R/MW	ft lb_f/lb_m °R
R_D	= Reynolds number based on D	ratio
R_d	= Reynolds number based on d	ratio
r	= Ratio of pressures, p_2/p_1	ratio
T	= Absolute temperature	°R
u_i	= Internal energy of the fluid	ft lb_f/lb_m
u_k	= Kinetic energy of the fluid	ft lb_f/lb_m
V	= Velocity	ft/sec
V_s	= Velocity of sound in a fluid	ft/sec
v	= Specific volume	ft^3/lb_m
x	= Ratio of differential pressure to inlet pressure = $1 - r$	ratio
Y	= Expansion factor	ratio
Z	= Compressibility factor of a gas	ratio
β	= Ratio of diameters, d/D	ratio
γ	= Ratio of specific heats of an ideal gas, c_p/c_v	ratio
Γ	= Isentropic exponent of a real gas	
Δp	= Differential pressure, $p_1 - p_2$	psi
λ	= $1000/\sqrt{R_D} = 1000/\sqrt{\beta R_d}$	ratio
Λ	= Height of a section above a datum	ft
μ	= Absolute viscosity	lb_m/ft sec
ν	= Kinematic viscosity	ft^2/sec
ρ	= Density of fluid	lb_m/ft^3
σ	= Standard deviation	per cent
ϕ	= Flow function	ratio

Subscript 1 refers to the first or upstream section.

Subscript 2 refers to the second or throat section.

Subscript i refers to an ideal gas or a property thereof.

Subscript T refers to the theoretical rate or condition.

Subscript t refers to the total or stagnation pressure or temperature and may follow another subscript; thus, p_{1t} = total pressure at first section.

* refers to that section or to a fluid property or flow function, where or when the fluid speed is equal to the speed of sound.

Note: R = 10.7314 lb_fft^3/in.^{2} °R mole lb_m [21] Table 2-a and 10.7314 × 144 = 1545.32 lb_f ft/°R mole lb_m, so that for air R_g = 1545.32/28.9644 = 53.3525 lb_f ft/lb_m°R. For other gases R_g = 1545.32/MW.

Concerning the units for A, a, D, d and p, refer to Notes 1 and 7 following Table I-2-1.

I-5-4 Let the values of p_1, u_{i1}, u_{k1}, V_1, v_1, and ρ_1 be arithmetical mean values obtained by averaging over the whole section, A, and, if the fluid motion is not strictly steady (laminar) but turbulent, a time average over the section. Likewise, let p_2, ... ρ_2 be the corresponding values over a second section, a. Let the area of a be considerably less than A. Then, since the flow is constant and steady, the same number of molecules must pass through section a as through A; but they must travel faster through section a than section A. In order to produce this change in velocity, which gives an increase in kinetic energy, there must be a decrease in other kinds of energy, particularly potential energy as evidenced by a decrease of static pressure. Also, in general, there will be a decrease in the internal energy of the fluid represented by a decrease in temperature. The decrease in pressure and temperature between sections A and a results in a change in the fluid density. With liquids this change in density is generally negligible, whereas with gases it must be taken into consideration.

I-5-5 The energy exchange represented by the decrease in static pressure between sections A and a, and which can be measured, is used in evaluating the difference in the velocities at the two sections and thence the rate of flow. So far as the analytical considerations are concerned, it is immaterial whether the area change from A to a takes place gradually or abruptly. However, the manner in which the area changes has a significant effect upon the magnitude of the static pressure difference and the positions and manner of measuring the static pressures. If the area change is gradual, or relatively so, so that the stream cross section is more or less well guided in changing from A to a, the minimum cross section of the stream appears to coincide with a. However, if the area change is abrupt and there is no guidance to the stream and if the axial length of a is very short compared to the diameter of the pipe, the cross section of the stream continues to decrease for a short distance downstream of a. That section at which this cross-sectional area is a minimum is called the "vena contracta." The distance of the vena contracta from a and its area depend upon the relation of a to A and the characteristics of the flow.

If downstream of section a the channel area returns to the same cross-sectional area as at section A, either gradually or abruptly, the static pressure, temperature and velocity of the fluid tend to return to nearly the same values that occurred at section A.

I-5-6 The basic physical concepts or equations from which are developed the equations for the flow of fluids through differential pressure meters are the equations of continuity and energy. The equation of continuity for steady flow is a special case of the general physical law of the conservation of matter. According to this equation the mass of fluid passing any section, A, per unit time is not only constant but is equal to that passing a second section, a, per unit time; thus,

$$AV_1\rho_1 = aV_2\rho_2 \qquad \text{(I-5-1)}$$

I-5-7 The Energy Equation. The general energy equation is simply an energy accounting, and a statement of it is: as each pound of fluid passes from A to a, the increase of its total energy, kinetic plus internal, is equal to the work done on it plus the heat added to it. The work done upon the fluid due to the pressure change is $(p_1v_1 - p_2v_2)$ and that due to gravity or the change in gravitational potential (i.e., elevation) is $(\Lambda_1 - \Lambda_2)$. Thus, the general energy equation becomes

$$(u_{k2} + u_{i_2}) - (u_{k_1} + u_{i_1}) = (p_1v_1 - p_2v_2) + (\Lambda_1 - \Lambda_2) + q_H \qquad \text{(I-5-2)}$$

Particular attention should be paid to the fact that no assumption has been made about resistance to flow. Work done against resistance at the walls may be dissipated into thermal energy which stays in the fluid and increases its internal energy. In such a case, kinetic energy is lost, but the sum of the kinetic and internal energy is unaffected.

Since nothing has been said about the nature of the fluid, equation (I-5-2) is equally valid for liquids and gases; but the further developments are simpler for liquids because the compressibility of gases is an additional complication which requires the use of thermodynamics. Liquids will, therefore, be considered first.

I-5-8 For the development of the theoretical equations of flow it is both conventional and convenient to make the following limitations and assumptions concerning the section of pipe and the fluid flow through it.

1. The pipe section is horizontal so that the effect of gravity is the same at all sections, and $\Lambda_1 = \Lambda_2$.
2. In flowing from section A to section a, the fluid performs no external work.
3. The flow is steady and axial, and the velocity profile at each section is relatively flat and normal to the pipe axis.
4. There is no transfer of heat between the fluid and the pipe, i.e., q_H is zero.

5. With liquids *only*, there is no change of temperature between sections A and a, which implies there is no change in the internal energy of the fluid.

I-5-9 Theoretical Equations for Liquids. Since with an incompressible fluid, i.e., a liquid, the temperature does not change, the density is constant, so that $\rho_1 = \rho_2 = \rho$. Thus, the continuity equation, (I-5-1), becomes

$$AV_1 = aV_2 \qquad \text{(I-5-3a)}$$

or

$$V_1 = \frac{a}{A}V_2 \qquad \text{(I-5-3b)}$$

Under the conditions imposed by the preceding assumptions, the energy equation reduces to

$$u_{k1} + p_1 v_1 = u_{k2} + p_2 v_2 \qquad \text{(I-5-4)}$$

Because a *flat* velocity profile is assumed, the kinetic energy per pound (mass) is

$$u_k = \frac{V^2}{2g_c} \qquad \text{(I-5-5)}$$

and the general energy equation may be written

$$\frac{p_1}{\rho} + \frac{V_1^2}{2g_c} = \frac{p_2}{\rho} + \frac{V_2^2}{2g_c} \qquad \text{(I-5-6)}$$

Using the value of V_1 from equation (I-5-3) in equation (I-5-6) and rearranging gives

$$V_2^2 = 2g_c\left(\frac{p_1 - p_2}{\rho}\right)\left[\frac{1}{1 - \left(\frac{a}{A}\right)^2}\right] \qquad \text{(I-5-7)}$$

Attention is directed to two of the factors in equation (I-5-7). First, the quantity, $(p_1 - p_2)/\rho$, is equal to the difference between the static pressures at A and a if measured by a column of the flowing liquid h ft in height. This column is referred to generally as a "head" of the liquid and gives us, hence, the term, "differential head meter," which has been used in fluid-metering literature. The second is $1/[1 - (a/A)^2]$, the square root of which is called the "velocity of approach" factor, since it resulted from substituting the expression for V_1 into equation (I-5-6). If the areas are circular, which is by far the most common condition, the diameters are the dimensions measured and known; hence, $(a/A)^2$ may be and usually is replaced with $(d/D)^4 = \beta^4$.

Note 1: A column of liquid when used as a measurement of pressure has the dimensions of force per unit area and not simply length.

Note 2: A table of values of the velocity of approach factor, $\sqrt{1/1 - \beta^4}$, is given in Part II.

In general, the user of a differential pressure meter is interested in knowing the rate of flow in terms of either mass (weight) or volume for some unit of time, such as second, minute, hour, or day. The theoretical equation for the mass rate of flow is

$$m_T = \rho a V_2$$

$$= a\sqrt{2g_c\rho(p_1 - p_2)}\sqrt{\frac{1}{1 - \beta^4}} \text{ lb}_m \text{ per sec} \qquad \text{(I-5-8)}$$

with the use of equation (I-5-7). The theoretical equation for volume rate of flow is

$$q_T = a V_2$$

$$= a\sqrt{\frac{2g_c(p_1 - p_2)}{\rho}}\sqrt{\frac{1}{1 - \beta^4}} \text{ cfs} \qquad \text{(I-5-9)}$$

$$= a\sqrt{2g_c h}\sqrt{\frac{1}{1 - \beta^4}} \text{ cfs} \qquad \text{(I-5-10)}$$

where

$$h = \frac{(p_1 - p_2)}{\rho} \text{ ft of flowing fluid} \qquad \text{(I-5-11)}$$

The subscript, T, indicates a theoretical value in contrast to actual rates of flow. Frequently equations (I-5-8), (I-5-9) and (I-5-10) are called the theoretical "hydraulic" equation for the flow of a fluid through orifices, flow nozzles and Venturi tubes.

I-5-10 Theoretical Equations for Compressible Fluids, Gases. The assumption of no transfer of heat between the fluid and the pipe, which implies no friction, permits assuming that any change of state between sections A and a is a reversible isentropic (adiabatic) change for which

$$p_1 v_1^\gamma = p_2 v_2^\gamma = p v^\gamma = \text{a constant, } c' \qquad \text{(I-5-12)}$$

If it is assumed that it is an ideal gas for which the general equation of state is

$$p/\rho = R/T \qquad \text{(I-5-13)}$$

then the energy equation, (I-5-2), may be written in the form

$$p_1/\rho_1 + \frac{V_1^2}{2g_c} + u_{i1} = p_2/\rho_2 + \frac{V_2^2}{2g_c} + u_{i2} \tag{I-5-14}$$

or, applying the definition for enthalpy,

$$\frac{V_2^2}{2g_c} - \frac{V_1^2}{2g_c} = H_1 - H_2 \tag{I-5-15}$$

For an ideal gas

$$H_1 - H_2 = \int_{p_1}^{p_2} v\,(dp) \tag{I-5-16}$$

Using equation (I-5-12) in the form, $v = c'/p^{1/\gamma}$,

$$H_1 - H_2 = c' \int_{p_1}^{p_2} p^{1/\gamma}\,(dp) \tag{I-5-17}$$

the integrand of which is

$$H_1 - H_2 = c' p_1^{\left(\frac{\gamma-1}{\gamma}\right)} \frac{\gamma}{\gamma-1}\left(1 - r^{\frac{\gamma-1}{\gamma}}\right) \tag{I-5-18}$$

Since $c' = v_1 p_1^{1/\gamma}$ and with equation (I-5-15),

$$\frac{V_2^2}{2g_c} - \frac{V_1^2}{2g_c} = p_1 v_1 \left(\frac{\gamma}{\gamma-1}\right)\left(1 - r^{\frac{\gamma-1}{\gamma}}\right) \tag{I-5-19}$$

Since the mass rate of flow is the same at sections A and a,

$$m = AV_1\rho_1 = aV_2\rho_2 \tag{I-5-20}$$

Using equation (I-5-12) in the form $\rho_2/\rho_1 = r^{1/\gamma}$

$$V_1 = V_2\left(\frac{a}{A}\right) r^{1/\gamma} \tag{I-5-21}$$

Substituting the relation from equation (I-5-21) in (I-5-19) and solving for V_2 gives

$$V_2 = \left[\frac{\frac{2g_c p_1}{\rho_1}\left(\frac{\gamma}{\gamma-1}\right)\left(1 - r^{\frac{r-1}{\gamma}}\right)}{r - \left(\frac{a}{A}\right)^2 r^{2/\gamma}}\right]^{1/2} \tag{I-5-22}$$

Using this value of V_2 in equation (I-5-20) and noting that $\rho_2 = \rho_1 r^{1/\gamma}$ gives

$$m_T = a\left[\frac{2g_c p_1 \rho_1 r^{2/\gamma}\left(\frac{\gamma}{\gamma-1}\right)\left(1 - r^{\frac{\gamma-1}{\gamma}}\right)}{1 - \left(\frac{a}{A}\right)^2 r^{2/\gamma}}\right]^{1/2} \tag{I-5-23}$$

This equation may be modified by using $p_1 = (p_1 - p_2)/(1 - r)$ and $\beta^2 = (a/A)$ so that it can be written

$$m_T = a\left[\frac{2g_c (p_1 - p_2)\rho_1 r^{2/\gamma}\left(\frac{\gamma}{\gamma-1}\right)\left(\frac{1 - r^{\frac{\gamma-1}{\gamma}}}{1-r}\right)}{1 - \beta^4 r^{2/\gamma}}\right]^{1/2} \tag{I-5-24}$$

In contrast to equation (I-5-8), equation (I-5-24) is called the "theoretical adiabatic" equation for the mass rate of flow of an ideal compressible fluid across section a in terms of the initial pressure or the pressure difference and the density, ρ_1. Equation (I-5-24) may be written in the form

$$m_T = a\sqrt{\frac{2g_c\rho_1(p_1 - p_2)}{1 - \beta^4}}$$

$$\left[r^{2/\gamma}\left(\frac{\gamma}{\gamma-1}\right)\left(\frac{1 - r^{\frac{\gamma-1}{\gamma}}}{1-r}\right)\right]^{1/2}\left(\frac{1-\beta^4}{1-\beta^4 r^{2/\gamma}}\right)^{1/2} \tag{I-5-25}$$

This amounts to using the hydraulic equation, (I-5-8), modified by the expansion factor,

$$Y = \left[r^{2/\gamma}\left(\frac{\gamma}{\gamma-1}\right)\left(\frac{1 - r^{\frac{\gamma-1}{\gamma}}}{1-r}\right)\left(\frac{1-\beta^4}{1-\beta^4 r^{2/\gamma}}\right)\right]^{1/2} \tag{I-5-26}$$

The value of Y depends upon the diameter ratio, β, the pressure ratio, r, and the ratio of specific heats, γ. For routine computations it will be found convenient to prepare curves or tables from which the values may be read or to use the curves or tables in Part II.

The values of ρ_1 in equations (I-5-24) and (I-5-25) should be computed with the general equation of state for an actual gas which, for this purpose, may be written

$$\rho_1 = \frac{p_1}{Z_1 R_g T_1} \qquad \text{(I-5-27)}$$

in which Z_1 is the compressibility factor for the particular gas being metered, corresponding to the conditions defined by p_1 and T_1.

I-5-11 Since most materials expand or contract as their temperature increases or decreases, a factor, F_a, must be introduced to take account of any change of the area of section a of the primary element when the operating temperature differs appreciably (e.g., more than 50 F) from the ambient temperature at which the device was manufactured and measured. If the meter is to be used under temperature conditions within the range of ordinary atmospheric temperatures, any difference between the thermal expansion of the pipe and the primary element may be ignored; and the diameter ratio, β, may be considered to be unaffected by temperature. However, if the meter is to be used at temperatures outside the ordinary range, then the material for the throat liner of a Venturi tube, a flow nozzle or an orifice plate should have a coefficient of thermal expansion as close as possible to that of the pipe. (An exception to this last statement could be where an orifice plate is mounted in a special fitting such that its outer rim is not clamped rigidly between flanges.) Values of F_a are given in Part II, Fig. II-1-2.

I-5-12 Definition of Discharge Coefficient. The *actual* rate of flow through a differential pressure meter is very seldom, if ever, exactly equal to the theoretical rate of flow indicated by the particular theoretical equation used. In general, the actual rate of flow is *less* than the indicated theoretical rate. Hence, to obtain the actual flow from the theoretical equation, an additional factor, called the "discharge coefficient," must be introduced. This coefficient is represented by C and defined by the equation:

$$C = \frac{\text{actual rate of flow}}{\text{theoretical rate of flow}} \qquad \text{(I-5-28)}$$

The rate of flow may be in terms of mass (or weight) or volume per unit of time. When volume is used, it is necessary that both the actual and the theoretical volumes be at or be referred to the same conditions of pressure and temperature. Thus, the actual rate of flow through a Venturi tube, flow nozzle or orifice, when the general hydraulic equation is used, will be

$$m = \left(\frac{\pi d^2 F_a}{4}\right)\left(\frac{C}{\sqrt{1-\beta^4}}\right)\sqrt{2g_c \rho_1 (p_1 - p_2)} \qquad \text{(I-5-29)}$$

or

$$q = \left(\frac{\pi d^2 F_a}{4}\right)\left(\frac{C}{\sqrt{1-\beta^4}}\right)\sqrt{2g_c h} \qquad \text{(I-5-30)}$$

Using the adiabatic equation, (I-5-25), would require multiplying the right-hand side of equations (I-5-29) or (I-5-30) by the expansion term, equation (I-5-26).

The factor, $C/\sqrt{1-\beta^4}$, may be and frequently is replaced by the flow coefficient K, that is

$$K = C/\sqrt{1-\beta^4} \qquad \text{(I-5-31)}$$

Note: By using equation (I-5-31) and the relation given by equation (I-2-7) in equation (I-5-30) gives

$$q = aF_a K\sqrt{2gh} \qquad \text{(I-5-32)}$$

The values of C and K will be different for each different type of primary element, Venturi tube, flow nozzle and orifice. Also, with flow nozzles and orifices, the values will depend upon the locations of the pressure taps; and, with the orifice, the values will differ with the type of inlet edge, whether square and thin or rounded, i.e., the so-called quadrant-edge orifice. Since both C and K are ratios, their numerical values are independent of the system of units in which the various quantities are measured. Values of C for these several types and modifications of primary elements are given in Part II of this report in connection with the application and use of these meters.

I-5-13 Equations for Computing Actual Rates of Flow: Foot-Pound-Second-Fahrenheit Units. In most of the metering of fluids with differential pressure meters as used in this country, diameters are measured in *inches* (instead of feet); static pressures and differential pressures, in pounds (force) per sq in., inches of mercury or inches of water; densities, in pounds (mass) per cu ft; and temperatures, in degrees F. The rate of flow may be given in pounds per second (pps), cubic feet per second (cfs), or gallons per second (gps). Of course, these rates of flow may be given in terms of other time intervals such as a minute, hour or day. When in cubic feet or gallons, the temperature of the fluid should be obtained; and, with gases, the static pressures and, possibly, also the relative humidity will be needed in addition. For any combination of units of measure that may be selected, the necessary conversion factors can be combined with some of the other terms into a single numerical factor. Some of the more common combinations of units used in commercial work with the corresponding numerical factors are given below.

With p_1 and p_2 in psia, T_1 in deg R, and ρ_1 in lb_m per cu ft at the conditions p_1 and T_1, and using Y computed from equation (I-5-26),

$$m\ (lb_m \text{ per sec}) = \frac{d^2CY}{576}\left(\frac{F_a}{\sqrt{1-\beta^4}}\right)\sqrt{2 \times 144 \times 32.174\,\rho_1\,(p_1 - p_2)} \quad \text{(I-5-33)}$$

$$= 0.52502\left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right)\sqrt{\rho_1\,(p_1 - p_2)}$$

If the differential pressure is measured in inches of water at 68 F, then

$$(p_1 - p_2) = h_w\,\frac{62.3164}{1728} \quad \text{(I-5-34)}$$

Applying this relation to equation (I-5-33) gives

$$m\ (lb_m \text{ per sec}) = 0.099702\left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right)\sqrt{\rho_1 h_w} \quad \text{(I-5-35)}$$

or

$$q_1\ (\text{cfs at } p_1,\ T_1) = 0.099702\left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right)\sqrt{\frac{h_w}{\rho_1}} \quad \text{(I-5-36)}$$

Also,

$$m\ (lb_m \text{ per hr}) = 358.93\left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right)\sqrt{\rho_1\,h_w} \quad \text{(I-5-37)}$$

and

$$q_1\ (\text{cfh at } p_1,\ T_1) = 358.93\left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right)\sqrt{\frac{h_w}{\rho_1}} \quad \text{(I-5-38)}$$

I-5-14 As stated in Par. I-2-11 of Chapter I-2, the reference condition for fuel-gas volumes is 14.73 psia, 60 F and dry and is indicated by the subscript, b. Thus, for fuel gas the volume rate at the reference state is

$$q_b\ (\text{scfh}) = q_1\,\frac{519.69\ \ p_1 Z_b}{14.73\ \ T_1 Z_1} \quad \text{(I-5-39)}$$

For many years it has been the practice in the fuel-gas industry to evaluate the density of the gas at the metering conditions in terms of its specific gravity referred to air. Usually this specific gravity has been based on a determination with a specific gravity balance or the indication of a recording gravitometer, rather than the ratio of molecular weights (see Par. I-3-29). Applying equation (I-3-22), the density of the gas at p_1, T_1 will be

$$\rho_1 = 2.6991\,\frac{p_1 G}{T_1 Z_1} \quad \text{(I-5-40)}$$

Combining equations (I-5-31), (I-5-39), and (I-5-40) with (I-5-38) gives

$$q_b\ (\text{scfh}) = 7708\,KYd^2F_aZ_b\sqrt{\frac{h_w p_1}{GT_1Z_1}} \quad \text{(I-5-41)}$$

The use of these equations, or equivalents thereto, generally provides acceptable results as long as the metering conditions as represented by p_1, T_1 are not too far from normal ambient conditions (e.g., 0 psig $< p_1 <$ 200 psig and -20 F $< T_1 <$ 140 F). As the metering conditions depart further and further from normal ambient values, the use of equation (I-5-40) instead of (I-5-27) may give to rise to significant errors. This is because the behavior of *air* and *fuel gases*, as represented by the compressibility factor, Z, differ more and more as the metering conditions are extended further and further in either direction from the normal ambient air conditions. For this reason it is suggested that the use of equation (I-5-27) in conjunction with equation (I-5-33) and equations derived rigorously from these are to be preferred.

I-5-15 Strictly, equation (I-5-40) is applicable to a dry gas only. If the gas is wet, i.e., if there is water vapor mixed with it, then equation (I-3-39) should be used, which may be written in the form

$$\rho_1\ (\text{of wet gas}) = 2.6991\,\frac{G}{T_1 Z_1}\left[p_1 - p_w\left(1 - \frac{0.622}{G}\right)\right] \quad \text{(I-5-42)}$$

Hence, when a higher degree of accuracy is desired in metering a wet gas, the effect of moisture upon the density can be accounted for by replacing p_1 in equation (I-5-41) with $[p_1 - p_w (1 - 0.622/G)]$.

Also, average room temperatures are usually close enough to 68 F that the observed differential pressure in inches of water at room temperature may be assumed to be the same as if measured at 68 F without introducing any appreciable error as far as the solutions of many practical problems are concerned. However, there will be cases in which the requirements of the problem will justify taking account of any difference between the room and reference temperatures, and the method of doing this is illustrated in one of the typical problems in Part II.

I-5-16 Equations for Actual Rates of Flow: Kilogram-Meter-Second-Celsius Units. Let the several dimensions and quantities be expressed as follows:

- D = Centimeters (cm)
- d = Centimeters (cm)
- g_c = 980.652
- h_w = Centimeters of water at 20 C
- p = Kilogram$_f$ per square centimeter, (kg$_f$/sq cm), or gram$_f$ per sq cm, (gm$_f$/sq cm)
- Δp = Gram$_f$ per square centimeter, (gm$_f$/sq cm) or (gm$_m$/cm-sec^2)
- m = Gram$_m$ per second, (gm$_m$/sec), or kilogram$_m$ per second, (kg$_m$/sec)
- q = Cubic decimeter per second, (cu dm/sec), or cubic meter per second, m^3/sec)
- ρ = Gram$_m$ per cubic centimeter, (gm$_m$/cc)

The basic equation for differential pressure meters, as represented by equation (I-5-29), may be expressed in the manner

$$m\ (\mathrm{gm}_m/\mathrm{sec}) = \frac{\pi}{4}\,\mathrm{cm}^2 \left\{ 2\left[g_c\left(\frac{\mathrm{gm}_m}{\mathrm{cm-sec}^2}\right)\right]\left(\frac{\mathrm{gm}_m}{\mathrm{cm}^3}\right)\right\}^{1/2} \qquad \text{(I-5-43)}$$

Using $K = C/\sqrt{1-\beta^4}$, the numerical value of g_c, and the appropriate letter symbols,

$$m\ (\mathrm{gm}_m/\mathrm{sec}) = 34.783\,KYd^2F_a\sqrt{\Delta p\rho_1} \qquad \text{(I-5-44)}$$

or

$$m\ (\mathrm{kg}_m/\mathrm{sec}) = 0.034783\,KYd^2F_a\sqrt{\Delta p\rho_1} \qquad \text{(I-5-45)}$$

At 20 C (68 F), the density of water is 0.9982336 gm$_m$/cc; thus,

$$\Delta p = 0.99823\ h_w$$

and

$$m\ (\mathrm{kg}_m/\mathrm{sec}) = 0.034752\,KYd^2F_a\sqrt{h_w\rho_1} \qquad \text{(I-5-46)}$$

Also,

$$q_1\ (\mathrm{m}^3/\mathrm{sec\ at}\ p_1,\ T_1) = 0.000034752\,KYd^2F_a\sqrt{\frac{h_w}{\rho_1}} \qquad \text{(I-5-47)}$$

A committee on units of the International Gas Union recommends using as the reference or "standard" conditions for the evaluation of fuel gas volumes a pressure of 1013 millibars, a temperature of 15 C, and dry (i.e., free of water vapor).

Note 1: These conditions are equivalent to 14.696 psia, 59 F and dry. Thus a cu ft of gas at the AGA reference conditions (Par. I-2-11, Chapter I-2) would be 1.000385 cu ft at the IGU conditions.

Note 2: 1 atmosphere = 1013 millibars
= 760 mm Hg at 0 C
= 1.033226 kg/cm^2

The density of dry air at 1 atm and 0 C (273.16 K) is 0.00129304 gm$_m$/cc; thus

$$\rho_1\ (\mathrm{gm}_m/\mathrm{cc}) = 0.34185\,\frac{p_1G}{T_1Z_1} \qquad \text{(I-5-48)}$$

Combining equation (I-5-48) with (I-5-45) gives

$$m\ (\mathrm{kg}_m/\mathrm{sec}) = 0.020339\,KYd^2F_a\sqrt{\frac{p_1G}{T_1Z_1}\,\Delta p} \qquad \text{(I-5-49)}$$

and, with equation (I-5-47),

$$q_1\ (\mathrm{m}^3/\mathrm{sec}) = 0.000059431\,KYd^2F_a\sqrt{h_w\,\frac{T_1Z_1}{p_1G}} \qquad \text{(I-5-50)}$$

Applying the reference conditions given above,

$$q_b = q_1\,\frac{288.16}{1.033226}\,\frac{p_1}{T_1Z_1} \qquad \text{(I-5-51)}$$

and, combined with equation (I-5-50),

$$q_b \text{ (m}^3\text{/sec)} = 0.016575KYd^2F_a\sqrt{\frac{h_w\rho_1}{GT_1Z_1}} \quad \text{(I-5-52)}$$

The rates of flow per minute and per hour can be obtained, of course, by multiplying the above equations for m or q by 60 and 3600, respectively.

I-5-17 Determination of Discharge Coefficients. The basic procedure in determining discharge coefficients of differential pressure meters is to discharge the flow from the meter into a weighing tank, volumetric tank or holder. By noting the increase in the scale reading or the change in the content of the tank or holder for a measured interval of time, the *actual* mass rate of flow (corrected for air buoyancy) or volume rate of flow is determined. Simultaneously, the necessary indications of the meter under test are observed; and by substitution of these values in one of the equations given above, letting $C = 1$, the "theoretical" rate of flow is computed. The ratio of the actual rate of flow to the calculated theoretical rate is the discharge coefficient. Making such comparisons over even a limited range of conditions requires the use of much special equipment and involves considerable tedious computation if done manually. Fortunately, individual calibration is not now necessary in many cases since recognized coefficient values are available for use with certain definite and reproducible forms of differential pressure meters, which will be specified in later sections.

A considerable number of tests have been made to determine the discharge coefficients of certain differential pressure meters. As would be expected, the conditions under which these tests were made and their relative self-consistency differ rather widely, so that it is difficult to form an altogether satisfactory estimate of the proper discharge coefficient to be used in any given case. However, it is possible to refer the results of some of these different groups of tests to a common basis for comparison, and, to the extent that this is possible, composite discharge coefficient values can be obtained which are reliable within the limits of the experimental observations. Before a discussion of the correlation of coefficients of discharge, some comments on the correlation tools or ratios may be helpful.

I-5-18 Kinematic Similarity of Fluid Flows. Much can be learned about the flow of a fluid through a particular (large) channel by careful study of the flow through a small model of the original or prototype, *provided* that there is complete kinematic similarity between the model and the prototype. This condition may be satisfied if two conditions are fulfilled:

1. The model channel must be geometrically similar to the prototype channel. For example, if the prototype channel is a Venturi tube of certain shape, then the model should be a geometrically similar Venturi tube.
2. The flow pattern or pattern of the streamlines in the model must be similar to that in the prototype. The flow pattern of the streamlines, in turn, is determined by all the forces acting.

Among the forces just referred to are those arising from conditions preceding and following the special section. In other words, the flow pattern in such a section as the Venturi tube has been influenced by what preceded the tube, and what follows the tube may have some slight influence also. Therefore, the second condition cannot be fulfilled unless under the first condition the upstream and downstream configurations of the model are geometrically similar to those of the prototype. Although in principle this similarity of configuration should extend indefinitely, both preceding and following the particular section, in practice an approximation to such complete geometric similarity may have to suffice. No rules are at hand to be a guide as to what may constitute "approximate" geometric similarity, but possibly the requirements given in Fig. II-II-1 of Part II could be considered as minimums. However, it must be recognized that results of tests made with a model will be of doubtful value when applied to the prototype unless there is both geometric and dynamic similarity between model and prototype. Moreover, the correlation of test data, as discussed below, depends on the same degree of complete kinematic similarity.

I-5-19 Types of Forces Acting. Various specific laws of similarity or similitude could be devised, depending upon the type of forces acting. The types of forces are inertia, viscous, pressure, and elastic or compressible.

By inertia force is meant the resistance of an inert mass to acceleration. The magnitude of the inertia force is proportional to the product of particle mass and particle acceleration.

It is customary to consider separately two cases, or combinations, each with only three forces; thus: (1) viscous, inertia, and pressure and (2) elastic, inertia, and pressure. The first combination is characterized by the Reynolds number and the second combination by the Mach number. In each case, specifying two of the forces automatically specifies the third force because the three forces are in equilibrium. Therefore, in case (1) a significant pair of forces can be taken as viscous and inertia, whereas in case (2) a significant pair of forces can be taken as inertia and elastic.

I-5-20 Reynolds Number. Assume that the fluid is incompressible and that the flow takes place within a completely enclosing channel or that bodies having motion with respect to the stream are fully immersed in the fluid so that free surfaces do not enter into consideration and gravity forces are balanced by buoyant forces. For such a flow, the inertia and viscous forces are the only ones which need to be taken into account. Mechanical similarity exists if, at points similarly located with respect to the bodies, the ratios of the inertia forces to the viscous forces are the same.

Since the product of mass multiplied by acceleration is proportional to volume × density × (velocity/time), the inertia force is proportional to

$$\frac{L^3\rho V}{T} = \frac{L^3\rho V}{L/V} = L^2\rho V^2 \qquad \text{(I-5-53)}$$

where V is some characteristic velocity (for example, the average velocity over a fixed cross section of pipe), L is some characteristic length (such as the internal diameter of a pipe), ρ is density, and T is time.

The magnitude of the viscous or laminar internal friction force is proportional to the viscous shear stress, S, times some length squared, or SL^2. Since, for laminar flow, the shear stress equals the dynamic viscosity, μ, times the velocity gradient, the shear stress, S, is proportional to $\mu V/L$. The shear force, SL^2, is thus proportional to μVL. Then the dimensionless ratio, (inertia force)/(viscous force), is proportional to

$$\frac{L^2\rho V^2}{\mu VL} = \frac{\rho VL}{\mu} \qquad \text{(I-5-54)}$$

The name, Reynolds number, has been given to the ratio, $\rho VL/\mu$.

I-5-21 In the product composing the Reynolds number, L denotes any linear dimension of the section of the channel; and, in its use with differential pressure meters, the custom is to replace L with either D or d. Also, for brevity and convenience R_D is used to represent $DV_1\rho/\mu$ and R_d to represent $dV_2\rho/\mu$.

If viscous, inertia, and pressure forces determine the flow of an incompressible fluid for a prototype, then mechanical similarity between model and prototype is realized when the Reynolds number of the model equals the Reynolds number of the prototype.

With incompressible fluids, i.e., liquids, any change in the temperature as the fluid moves from section 1 to section 2 is in general so slight as to be negligible. Thus, the general practice is to assume that the density, ρ, and viscosity, μ, are the same at the two sections, so that $R_d = dV_2\rho/\mu_1$. On the other hand, with compressible fluids, gases, there may be an appreciable decrease in temperature accompanying the decrease in pressure from p_1 to p_2. These changes affect both ρ and μ, so that the correct evaluation of R_d is

$$R_d = \frac{dV_2\rho_2}{\mu_2} \qquad \text{(I-5-55)}$$

However, as T_2 is not readily determined by direct measurement, it is convenient to note that $V_2 = 4m/\pi d^2\rho_2$ and, thus, $R_d = 4m/\pi d\mu_2$. Since, in the great majority of cases the effect on the viscosity of the temperature change from T_1 to T_2 is small, it is customary to assume $\mu_2 = \mu_1 = \mu$. This results in arbitrarily using for evaluating the Reynolds number the relations,

$$R_d = \frac{d\,V_2\,\rho_2}{\mu} = \frac{4m}{\pi d\mu} \qquad \text{(I-5-56)}$$

and

$$R_D = \frac{D\,V_1\,\rho_1}{\mu} = \frac{4m}{\pi D\mu} \qquad \text{(I-5-57)}$$

In the preceding equations for R_d and R_D, the normal unit in which d and D would be expressed is the foot. As stated in Par. I-5-13, it is the general practice to express these diameters in inches; and, to keep d and D in inches, it is necessary to use

$$R_d = \frac{d\,V_2\,\rho_2}{12\,\mu} = \frac{48\,m}{\pi\,d\,\mu} \qquad \text{(I-5-58)}$$

and

$$R_D = \frac{D\,V_1\,\rho_1}{12\,\mu} = \frac{48\,m}{\pi\,D\,\mu} \qquad \text{(I-5-59)}$$

Note 1: By dividing equation (I-5-59) by equation (I-5-58), it will be seen that $R_D = \beta R_d$.

Note 2: The normal metric units for the Reynolds number are: D and d, in centimeters; V_1 and V_2, in cm/sec; ρ_1 and ρ_2, in gm_m/cm^3; m, in gm_m/sec; and μ, in poise.

I-5-22 Mach Number. Consider a flow of a compressible fluid through two geometrically similar channels in which the only forces involved are inertia, pressure and elastic. A significant ratio is (inertia force)/(elastic force). The inertia force is proportional to $\rho L^2 V^2$. The elastic or compressible force is proportional to EL^2, where E is the bulk modulus of the fluid. The bulk modulus, E, in turn, equals ρV_s^2, where V_s is the acoustic velocity in the fluid. Then the ratio, (inertia force)/(elastic force), is proportional to

$$\frac{\rho L^2 V^2}{EL^2} = \frac{\rho L^2 V^2}{\rho V_s^2 L_2} = \frac{V^2}{V_s^2} \qquad \text{(I-5-60)}$$

If inertia, pressure and elastic forces determine the flow for a prototype, then mechanical similarity between model and prototype is realized when the ratio, V^2/V_s^2, for the model equals the corresponding ratio, V^2/V_s^2, for the prototype.

For purposes of flow similarity, the ratio, V/V_s, could be used as well as the ratio, V^2/V_s^2. The name Mach number has been given to the ratio, V/V_s.

I-5-23 Application of Similitude. The use of some parameter, as Reynolds number or Mach number, for correlating data depends upon the forces involved in the flow.

For example, consider the flow of a liquid through a Venturi meter. The Reynolds number could be a very useful parameter for correlating data because the forces in the number are those involved in the flow. For flow over a weir, however, other forces may exist, such as gravity, surface tension and capillarity. The Reynolds number does not involve these additional forces and, therefore, may not be adequate for a complete correlation of data.

As another example, consider the flow of a gas through a meter. At low velocities the gas may behave as an incompressible fluid, and the Reynolds number may provide a suitable parameter for correlating data. At high velocities, however, compressibility effects may be present, and it may be necessary to use the Mach number in order to correlate the data.

I-5-24 Fluid-Flow Characteristics: Jet Contraction with an Orifice. With both the Venturi and the flow nozzle, the section used in all computations is the one of minimum cross-sectional area of the tube or the nozzle; and the fluid stream completely fills this section, being guided by the walls of the tube or the nozzle. With an orifice, the fluid stream is not so guided; and, as shown in Fig. I-5-4, the cross section of the stream continues to decrease after passing through the orifice. In strict analogy to the Venturi tube, the area of the minimum jet section, known as the vena contracta, which corresponds to the throat of a Venturi tube, should be used in the flow equation. However, no satisfactory method of actually measuring this minimum jet area is known, whereas measuring the diameter of an orifice and thus determining its area is a relatively simple matter. Some

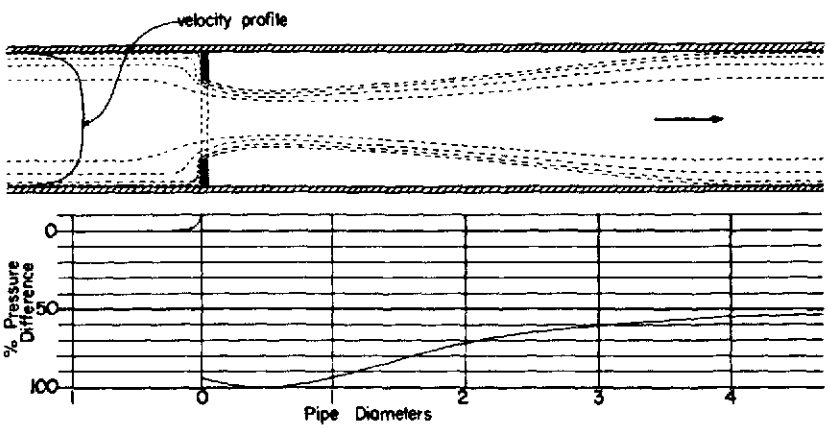


FIG. I-5-4 DIAGRAMMATIC REPRESENTATION OF FLUID FLOWING THROUGH THIN-PLATE SQUARE-EDGED ORIFICE AND RELATIVE PRESSURE CHANGES investigators have represented the ratio of the minimum jet area to the orifice area by a separate factor, which then is called a contraction coefficient. Doing this, however, is of no practical advantage, and including the effects of contraction in the orifice discharge coefficient is more convenient. By experiment, the amount of contraction has been found to depend primarily upon the diameter ratio, β, and the properties of the fluid; hence, the discharge coefficient will also vary with these factors when it includes the effects of jet contraction.

With any one fluid and the same differential pressure, the relative amount of jet contraction increases as the diameter ratio, β, decreases. That is, the ratio of the jet area, at the vena contracta, to the area of the orifice decreases as β decreases. This is exactly what would be expected, because, as the fluid particles near the wall of the channel converge toward the orifice, they attain greater radial velocity inward when β is small than when β is large. Thus, the discharge coefficients for orifices will increase as β increases [1, 2, 3].

At ordinary rates of flow, the fluid properties that have the most influence on jet contraction are compressibility or its reciprocal, expansibility, and viscosity. Between the plane of the orifice and the vena contracta, a radial pressure gradient outward is present; and, if the fluid is a gas, it expands transversely as well as longitudinally, whereas in the case of a liquid, no expansion occurs. The cross section of the vena contracta is, therefore, larger with a gas than with a liquid [1]. Hence, if the hydraulic equation is used, as is the general practice, the discharge coefficient corresponding to a given jet velocity, computed from the use of the inlet density, will be numerically lower for a gas than for a liquid like water.

As the viscosity of the fluid increases, jet contraction decreases. The reason is that, as the viscosity increases, the effects of friction against the surface of the channel extend farther toward the center and the radial velocity of the stream filaments from near the wall is small in proportion to the axial velocity of the central filaments, thus making the vena contracta increase. Hence, for the same jet velocity, the discharge coefficient for such a fluid as a medium oil will be higher than for water [4, 5].

I-5-25 Static-Pressure Changes Close to an Orifice. All studies of the static-pressure changes close to a thin-plate square-edged orifice have shown that on the inlet side, within the last 0.1 D, the static pressure increases to a maximum at the corner of the plate and pipe wall. On the outlet side, the static pressure continues to decrease until a minimum is reached somewhere between about 1/4 D and 1 D. For any one orifice, the distance from the plane of the inlet side of the orifice plate to that where the static pressure at the wall of the pipe is the minimum is nearly independent of the rate of flow if the fluid is a liquid. Regardless of the nature of the fluid, this distance is a function of the orifice-to-pipe diameter ratio, β, becoming less as the value of β becomes larger. Numerous experiments by different investigators over a wide range of pipe sizes have shown that the relation between the position of minimum static pressure and the diameter ratio, β, is independent of the actual pipe size for pipes 2 in. or more in diameter. The average maximum and minimum values of the distance of the plane of minimum static pressure from the plane of the orifice, as reported by different experiments, are shown in Fig. I-5-5.

With the flow of compressible fluids, that is, gases, through an orifice, the position of minimum static pressure has been found to depend upon the rate of flow as indicated by the pressure ratio, $p_2/p_1 = r$, as well as upon the diameter ratio, β. If the value of r is very close to unity, the position of the minimum pressure will be indistinguishable from that observed with liquids [6]. However, as r decreases, that is, as the rate of flow increases, the position of the minimum static pressure moves farther away from the orifice. The fact that the plane of minimum static pressure is not at the orifice indicates that the area of the jet at the plane of minimum pressure is less than the area of the orifice. The general assumption is that this plane of minimum pressure coincides with that of the vena contracta. A few observations

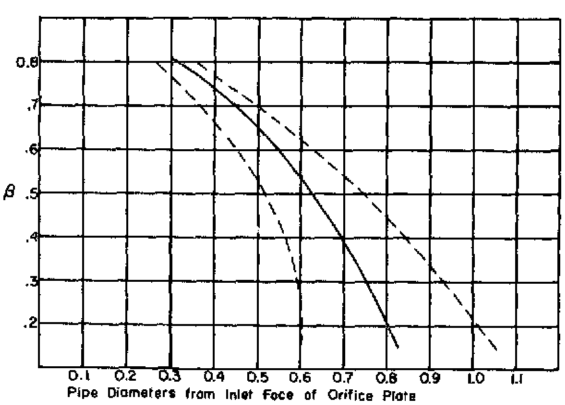


FIG. I-5-5 LOCATION OF VENA CONTRACTA OUTLET PRESSURE TAP WITH CONCENTRIC SQUARE-EDGED ORIFICES (BROKEN LINES SHOW MAXIMUM VARIATION LIMITS.)

on a stream of fluid passing at low rates of flow through an orifice held in a large glass tube indicated that, under those conditions, the assumption is correct. However, this assumption has not been positively established as valid for all conditions, although no experimental evidence to the contrary is known. Therefore, the coincidence of the plane of the vena contracta and that of minimum static pressure must be considered as an assumption even though it may be used as if it were a fact.

I-5-26 Location of the Pressure Taps. The locations of pressure taps used in test programs and commercial work will be named and defined as follows:

1. *Flange Taps.* The centers of the pressure holes are respectively 1 in. from the upstream, or inlet, and downstream, or outlet, faces of the orifice plate. Allowing for a 1/16-in. gasket, the centers of the holes will be 15/16 in. from the bearing faces of the flanges.

2. *Taps at One D and One-Half D.* The center of the inlet pressure tap is located one pipe diam preceding the inlet face of the orifice plate. The center of the outlet pressure tap is placed one-half pipe diam following the inlet face of the orifice plate, regardless of the value of the diameter ratio, β.

3. *Vena Contracta Taps.* The center of the inlet pressure tap is located between one half and two pipe diam from the upstream face of the orifice plate; usually a distance of one pipe diam is used. The center of the outlet pressure tap is placed at the position of minimum pressure (which is assumed to be the plane of the vena contracta) as given in Fig. I-5-5.

Note: Due to the flatness of the pressure gradient in the region of ½ D, the differential pressures observed with D and ½ D taps and with vena contracta taps are almost the same over the central part of the range of β ratios.

4. *Corner Taps.* The pressure holes open in the corner formed by the pipe wall and the orifice plate. The method of doing this and the widths of the openings, either single holes or ring slits, is shown in Fig. I-5-2 (c). The axial width of the slits or openings should be 0.02 D. The same corner-tap proportions are used with both nozzles and orifices.

I-5-27 Effects of Installation and Construction. In the development of equations such as (I-5-8) and (I-5-29), a uniform fluid velocity was assumed, thus neglecting any effects of normal stream turbulence. This normal turbulence can be greatly increased, by the configuration of the channel or obstructions in the channel, so that the distribution of velocity becomes very irregular or a pronounced spiral motion may be set up. For convenience in this discussion, this exaggerated turbulence will be termed "disturbed flow." Obviously, the character of this disturbed flow will depend in large measure upon the installation conditions, that is, upon the type of fittings and their relative distances, on the inlet side particularly, from the primary element. The effects of the disturbances produced by several kinds of fittings have been extensively studied in connection with orifice meters and, to a lesser extent, with Venturi tubes and flow nozzles. In general, when disturbed flow is produced by the presence of fittings, the value of the discharge coefficient is more likely to be higher than when determined under normal flow conditions. The magnitude of the effect will depend mostly upon the type of fittings, the distance from the primary element, and the diameter ratio, β. The minimum installation requirements that should be fulfilled to reduce the effects produced by such fittings to the minimum are given in Part II of this edition.

The condition of the channel surface, that is, its relative roughness, will also affect the flow of a fluid through a differential pressure meter. In general, a higher discharge coefficient appears to be obtained with a rough-surface pipe than with a smooth-surface pipe of the same size. To date, no index for relative roughness, especially with respect to the interior surface of a pipe, has received general recognition. Thus, at present there is no reference basis by which to correlate such data as are now available on this factor. On the other hand, if a given surface grain size, that is, a given degree of absolute roughness, is assumed, the relative roughness will be greater with a small pipe than with a large one. Therefore, a reasonable expectation would be that a higher discharge coefficient will be obtained with square-edged orifices with the smaller pipe, whereas, with Venturi tubes, the coefficient will be slightly lower. This is exactly what has been observed when different sizes of pipe of ordinary commercial smoothness have been used. Thus, the size of the pipe is one of the factors influencing the values of discharge coefficients, particularly of orifices. With nozzles, the direction of pipe size effect may vary, depending to some extent upon the nozzle-approach curvature.

Note: The evaluation of a surface in terms of micro-inches is becoming a common practice. However, this does not give a measure of the "waviness," either circumferential or longitudinal, of the interior surface of a pipe. Also, some surface instruments measure the peak to hollow height, automatically integrate the area on either side of a mean center line, and divide this by the length of travel to give a "center-line average" (CLA) value of the surface.

I-5-28 Turning to the influences of primary-element construction, the discharge coefficient of a Venturi tube or a nozzle having a smooth finish, of the inlet cone or curved entrance, has often been observed to be higher than that of a tube or nozzle of the same size but having a rough finish.

With orifices, the jet contraction and the value of the discharge coefficient are influenced in a marked degree by the condition of the upstream edge, or corner, of the orifice. Dulling or rounding of this edge decreases the jet contraction; and, if the rounding is continued, contraction will be entirely suppressed as in the case of a nozzle. Dulling or a very slight rounding of a sharp square corner produces relatively much more effect than a small change in the curvature of an orifice of well-rounded approach.

The width of the cylindrical face or edge of the orifice, as measured normal to the plane of the inlet face of the orifice plate, has also to be considered. If this edge width is relatively large (e.g., one-third of the orifice diameter), the amount of jet contraction is decreased, if not suppressed altogether. Therefore, to make possible the comparison of data and to facilitate reproducing given conditions, the orifice edge width must be kept small in proportion to the other dimensions of the orifice plate.

The position of the center of the orifice with respect to the axis of the pipe, that is, whether the orifice is placed concentrically or eccentrically in the pipe, will affect the pressure gradient, particularly on the outlet side. If the orifice is displaced away from the pressure taps, the differential pressure may be a few per cent different than when centered. On the other hand, as the orifice is displaced toward the pressure taps, the differential pressure will become increasingly erratic. Therefore, the orifice must be placed concentric with the pipe, particularly when the pressure taps are within one or two pipe diam of the orifice plate. This, of course, does not apply to eccentric and segmental orifices.

The variation of the static pressure or pressure gradient on the inlet and outlet sides of the orifices was mentioned in Par. I-5-25 and depicted in Fig. I-5-4. Evidently, the value of the differential pressure, $(p_1 - p_2)$ or h, which will be observed, will depend upon the positions of the pressure taps. Consequently, the discharge coefficient value determined or used will also be influenced by the position of the pressure taps. This applies to flow nozzles also, but to a lesser degree.

I-5-29 Correlation of Discharge Coefficients: Flow Nozzles and Orifices. In Pars. I-5-20 – I-5-22, it was indicated that the properties of the flowing fluid may influence the character of the flow through a differential pressure meter and, thereby, the coefficient. This fact may be represented by the relation

$$C = \psi\left(\frac{d\,V\rho}{\mu}, \frac{V}{V_s}, \beta\right)$$

$$C = \psi \quad (R_d, M, \beta) \qquad \text{(I-5-61)}$$

Or

$$K = \psi' \quad (R_d, M, \beta)$$

In the paragraphs above, most of the constructional features that will affect the character of the flow and, therefore, the coefficients are described briefly. All of them can be described in terms of a ratio of either the pipe diameter, D, or the throat diameter, d. For example, if the actual roughness, i.e., the peak to hollow height of the inner surface of a pipe, is ζ, then the relative roughness is ζ/D. Again, let υ be the radius of curvature of the inlet edge of an orifice, and the relative degree of curvature would be υ/d. But d may be specified in terms of $\beta = d/D$; hence, υ/d may be replaced by υ/D. For an absolutely square edge, $\upsilon = 0$.

Thus, the general relation between the discharge or flow coefficients and the fluid characteristics plus the constructional features of the primary element may be represented by

$$\begin{aligned} C &= \psi_2 \quad (R_d, M, \beta, D) \\ K &= \psi_2' \quad (R_d, M, \beta, D) \end{aligned} \qquad \text{(I-5-62)}$$

With incompressible fluids, the Mach number, M, has so little if any effect upon C or K that it can be and usually is omitted from the relation. On the other hand, the diameter ratio, β $(= d/D)$, must remain an independent variable, because limiting it to a single value, which would be implied if represented by D only, would destroy the flexibility of these meters, especially the orifice. The evaluation of the actual relations represented by equations (I-5-62) must be derived from the results of actual tests.

I-5-30 Application of Coefficients for Incompressible Fluids to Compressible Fluids: Expansion Factors. As stated previously, with compressible fluids there is a change of the fluid density accompanying the pressure change from p_1 to p_2. In the general relationship, represented by equation (I-5-62), this was represented by the Mach number, M. However, in considering the effects of fluid compressibility when both the initial and jet velocities are less than the acoustic velocity, V_s, it is usually convenient to replace M with the acoustic ratio, x/γ, as discussed in Par. I-3-42, resulting in equation (I-3-54). Thus the general relation may be written

$$C = \psi_3 (R_d, x/\gamma, \beta, D)$$
$$K = \psi' (R_d, x/\gamma, \beta, D) \tag{I-5-63}$$

In Venturi tubes and flow nozzles, the expansion which accompanies the change in pressure takes place in an axial direction *only*, due to the confining walls of these differential producers. The adiabatic expansion factor, Y, equation (I-5-26), compensates for this unidirectional expansion. With the thin-plate orifice, there are no confining walls, and the expansion takes place *both* radially and axially. To take account of this multidirectional expansion, an empirical equation for the expansion factor is derived from tests.

I-5-31 Expansion Factor, Y, for Square-Edged Orifices: Dependence upon the Acoustic Ratio. Assume that a particular orifice can be tested with air ($\gamma = 1.4$) over a fairly wide range of differential pressures so that $(p_1 - p_2)/p_1 = x$ will range from about 0.01 to 0.45 or over. Also, the static pressure is to be taken on the upstream or inlet side of the orifice, so that the inlet density, ρ_1, can be determined directly. For defining the flow coefficient, K, based on the inlet conditions, the hydraulic equation in the following form will be used

$$m = \frac{\pi d^2}{4} F_a (nK_1) \sqrt{2g_c \rho_1 (p_1 - p_2)} \tag{I-5-64}$$

The values of K_1 obtained from such a series of tests will be found to decrease as the ratio, x, increases. Moreover (except for large-diameter ratios), this relation is very nearly linear, so nearly so that the plotted values of K_1 to x can be represented by a straight line, such as line I in Fig. I-5-6.

Note: In equation (I-5-64) n is a numerical constant that takes account of the units in which the several factors are measured.

Now, suppose that the series of tests is repeated, using the same orifice but a different gas, for which $\gamma = 1.28$. As before, the plotted values of K_1 to x can be represented by a straight line. However, this line will be steeper than the first, as illustrated by line II

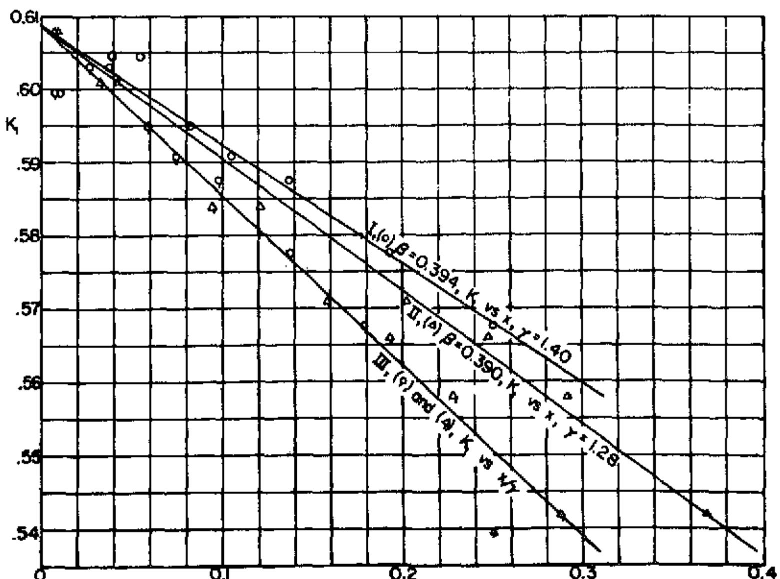


FIG. I-5-6 COEFFICIENTS FOR HYDRAULIC EQUATION OBTAINED FROM TESTS WITH GASES, PLOTTED AGAINST DIFFERENTIAL PRESSURE RATIO AND ACOUSTIC RATIO (K_1 IS THE FLOW COEFFICIENT, INCLUDING THE EXPANSION FACTOR, Y_1.)

in Fig. I-5-6. The slopes of these two lines are seen to be very nearly inversely proportional to the ratios of the specific heat. Hence, if values of x/γ instead of x are used as abscissas, the results from both tests could be expected to be represented by a single line, such as is shown to be the case by line III in Fig. I-5-6.

If K_0 denotes the value of K_1 when $x = 0$ (as obtained by extending line III to the 0-abscissa line) and ϵ (epsilon) is the slope of the K_1-to-x/γ line, then the value of K_1 corresponding to any particular value of x will be given by

$$K_1 = K_0 - \epsilon\frac{x}{\gamma}$$

$$= K_0\left(1 - \frac{\epsilon}{K_0}\frac{x}{\gamma}\right) \qquad \text{(I-5-65)}$$

$$= K_0 Y_1$$

from which

$$Y_1 = \left(1 - \frac{\epsilon}{K_0}\cdot\frac{x}{\gamma}\right) \qquad \text{(I-5-66)}$$

Y_1 is termed the "net expansion factor" since it is introduced to take account of the effects of expansion as an expansible fluid flows through an orifice, and the "hydraulic" equation is used for computing the rate of flow [1, 7]. As indicated, the value of Y_1 depends upon the acoustic ratio and the ratio of ϵ/K_0. From many experiments, the slope factor, ϵ, has been found to be practically independent of all orifice shape factors except β.

K_0 has been defined as the limiting value of K_1 as $x \to 0$; and, sometimes, it is referred to as the water flow coefficient, K_w. In Par. I-5-29, this value of K_0 or K_w was shown to be a function of D, R_d and β. Hence, the ratio, ϵ/K_0, apparently should also be a function of D, R_d and β. However, the effect of D and R_d upon the ratio, ϵ/K_0, is generally so small that it may be neglected and ϵ/K_0 considered as a function of β only.

I-5-32 For gases, the product of the flow coefficient and the expansion factor for a given orifice meter shape (i.e., "shape" = a constant) is seen to be dependent upon the Reynolds number and the acoustic ratio. The effects of these two ratios are doubtlessly interdependent to a slight extent but not enough to be of any practical significance. A possible reason for the inability to separate the effects of the two ratios is that the Reynolds numbers of most, if not all, of the tests from which the expansion factor was determined were fairly high. Thus, any changes of K due to the Reynolds-number effect would be small enough to be entirely masked by the much greater effect of expansion [7].

If $K = K_0 = K_w$, a relation which is used often without stating it, the complete hydraulic equation for use with orifice meters when metering gases is obtained. This equation is

$$m = 0.52502\, K\, Y_1 d^2 F_a \sqrt{\rho_1\, \Delta p}$$

$$m = 0.52502\, \frac{C}{\sqrt{1-\beta^4}}\, Y_1 d^2 F_a \sqrt{\rho_1\, \Delta p} \qquad \text{(I-5-67)}$$

Repeating what has been stated above, K is a function of D, R_d and β, while Y_1 is a function of x/γ and β. When the procedure as outlined, is followed and the inlet pressure and temperature are used to evaluate the density, ρ_1, (or ρ_1 is obtained directly with a densitometer), the corresponding expansion factor Y_1, is given by

$$Y_1 = 1 - (0.410 + 0.350\beta^4)x/\gamma \qquad \text{(I-5-68)}$$

This equation applies when the inlet pressure tap is between 0 and D, as measured from the inlet face of the orifice plate, and the outlet pressure tap is at the downstream corner or within the distance shown by the "mean" line of Fig. I-5-5, as measured from the inlet face of the orifice plate, also. In other words, it applies to corner, flange, vena contracta and D and 1/2 D taps. Furthermore, this equation applies *only* with jet velocities *below* the velocity of sound at the conditions in the jet, that is, where $V_2 < V_s$ [1].

In equation (I-5-65) and the development of equation (I-5-68), the coefficient of discharge, C, could have been used instead of the flow coefficient, K. However, if this had been done, the numerical factors in equation (I-5-68) would be different. As it is, expansion factors computed with equation (I-5-68) *must* be used with the flow coefficient, K, or the equivalent, $C/\sqrt{1-\beta^4}$.

Note: Another equation for the expansion factor is

$$Y = 1 - (0.3707 + 0.3184\beta^4)\,(1 - r^{1/\gamma})^{0.935} \qquad \text{(I-5-69)}$$

This equation is given in ISO Document No. R 541 (1966), "Measurement of Fluid Flow," and is intended, primarily, for use with corner taps [8].

I-5-33 In many cases, especially where flange taps are used, the static pressure is taken from the downstream or outlet pressure tap. If the fluid is a gas, this means that the density, ρ_2, is determined from p_2 and either T_1 or T_2, whichever is measured (usually there is very little difference between T_1 and T_2). The expansion factor, Y_2, which corresponds with the use of ρ_2 (i.e., p_2), can be evaluated from Y_1 in the following manner. Using the relation given by equation (I-5-27), equation (I-5-67) may be written

$$m = nKY_1\sqrt{\Delta p\, p_1} = nKY_2\sqrt{\Delta p\, p_2} \qquad \text{(I-5-70)}$$

Here, as before

$$\Delta p = p_1 - p_2$$

and also, by definition,

$$x = \Delta p/p_1 = 1 - r$$

and

$$r = p_2/p_1$$

From equation (I-5-70),

$$Y_2 = Y_1\sqrt{\frac{p_1}{p_2}} = Y_1\sqrt{\frac{1}{r}} = Y_1\sqrt{\frac{1}{1-x}} \qquad \text{(I-5-71)}$$

Let $x_2 = \Delta p/p_2$. Then, $\Delta p = x_2 p_2 = x p_1$, or

$$x = x_2\frac{p_2}{p_1} = x_2\frac{p_2}{p_2 + \Delta p} = \frac{x_2}{1 + x_2} \qquad \text{(I-5-72)}$$

Applying this to equation (I-5-71) gives

$$Y_2 = Y_1\sqrt{1 + x_2} \qquad \text{(I-5-73)}$$

Using the value of Y_1 from equation (I-5-68) and the value of x from equation (I-5-72) gives

$$Y_2 = \left(\sqrt{1 + x_2}\right) - (0.41 + 0.35\beta^4)\frac{x_2}{\gamma}\frac{1}{\sqrt{1 + x_2}} \qquad \text{(I-5-74)}$$

I-5-34 Mass Flow by Modifications to the Orifice Meter. It has been shown both analytically and experimentally that by using the principle of the "magnus effect" a pressure difference directly proportional to mass flow rate may be obtained. To do this, a cylinder is rotated in the flow conduit, and the circumferential velocity of the cylinder causes the fluid velocity in one gap between the cylinder and conduit wall to increase, while in the opposite gap the fluid velocity is decreased. The difference in the static pressures at the two gaps is directly proportional to the total mass rate of flow irrespective of fluid density [9].

Again, it has been shown both analytically and experimentally that by placing two matched orifices in series a pressure difference can be obtained that is directly proportional to the mass rate of flow. To do this, a constant volumetric flow is recirculated through the system in such a way as to be subtracted from the flow through one of the orifices and added to the flow through the other. The difference between the separate differential pressures is proportional directly to the net mass rate of flow. Several modifications of this procedure are possible [10].

It is to be noted that, with both of the procedures outlined above, auxiliary power is required. In the first procedure, power is required to rotate the cylinder; in the second, a power-driven pump is required to produce the recirculating flow. The procedures in both cases are proprietary.

I-5-35 Definite Relationships between C, R_d, β, and D: Venturi Tubes. The discharge coefficients for the Herschel type Venturi tube, Fig. I-5-1, recommended by the Technical Committee on Measurement of Fluid Flow in Closed Conduits, of the International Organization for Standardization [11], are:

1. For a Venturi tube with a rough-cast convergent inlet section and

$$4 \text{ in.} \le D \le 32 \text{ in.}$$
$$0.3 \le \beta \le 0.75$$
$$2.10^5 \le R_D \le 2.10^6$$
$$C = 0.984 \pm 0.70 \text{ per cent.}$$

2. For a Venturi tube with a machined convergent inlet section and

$$2 \text{ in.} \le D \le 10 \text{ in.}$$
$$0.4 \le \beta \le 0.75$$
$$2.10^5 \le R_D \le 1.10^6$$
$$C = 0.995 \pm 1.0 \text{ per cent.}$$

3. For a Venturi tube with a rough welded sheet-iron convergent inlet section and

$$8 \text{ in.} \le D \le 48 \text{ in.}$$
$$0.4 \le \beta \le 0.7$$
$$2.10^5 \le R_D \le 2.10^6$$
$$C = 0.985 \pm 1.50 \text{ per cent.}$$

Note: The tolerance values following the values of C are twice the standard deviation as given in the ISO Recommendation. This is in agreement with the method of evaluating tolerances used in this report.

I-5-36 Flow Nozzle Coefficients. From a review of several thousand tests of flow nozzles of the ASME long-radius type, in 2-in. and larger pipes, the following equation has been derived for use with pipe-wall pressure taps at 1 D and 1/2 D, and $0.30 \lesseqgtr \beta \lesseqgtr 0.825$ and $10^4 \lesseqgtr R_d \lesseqgtr 10^6$

$$C = 0.99622 + 0.00059\,D$$

$$- (6.36 + 0.13D - 0.24\,\beta^2)\frac{1}{\sqrt{R_d}} \qquad \text{(I-5-75)}$$

A table of flow nozzle coefficients computed from the above equation is given in Part II.

I-5-37 Coefficients for Thin-Plate Square-Edged Orifices. The equations given below, in Chapter II-III, and the values given in Tables II-III-2, II-III-3 and II-III-4 are based on determinations made at Ohio State University and reported by the Joint AGA-ASME Committee on Orifice Coefficients [12, 13]. Although there have been many determinations made in different laboratories since those at OSU, the tests at any one laboratory have covered only a limited set of conditions. Attempts to correlate these later determinations as a group or with the OSU values have not resulted satisfactorily. Therefore, any improvement to the equations and tabulated values given in this report must await new determinations covering a sufficiently wide and comprehensive range of conditions [14, 44].

Since it may aid in making interpolations as well as providing a basis for tables for pipe sizes other than those given in Chapter II-III, the equations are given in detail. These equations apply particularly to pipes 2 in. and larger, to values of β between 0.2 and 0.75 and to values of R_d above 10^4. The letter symbols are as defined in Par. I-5-3 above, with special values as follows:

- K = Flow coefficient corresponding to any specific set of values of D, β, and R_d (or R_D)
- K_0 = The limiting value of K for any specific values of D and β when R_d (or R_D) becomes infinitely large
- C = $K/E = K\sqrt{1-\beta^4}$ and $R_D = \beta R_d$

I-5-38 For flange pressure taps, K_e = the particular value of K for any specific values of D and β, when $R_d = (10^6 d)/15$. $K = K_0\,(1 + A/R_d)$, and $K_0 = K_e\,[(10^6 d)/(10^6 d + 15A)]$.

$$K_e = 0.5993 + \frac{0.007}{D} + \left(0.364 + \frac{0.076}{\sqrt{D}}\right)\beta^4$$

$$+ 0.4\left(1.6 - \frac{1}{D}\right)^5\left[\left(0.07 + \frac{0.5}{D}\right) - \beta\right]^{5/2}$$

$$- \left(0.009 + \frac{0.034}{D}\right)\left(0.5 - \beta\right)^{3/2}$$

$$+ \left(\frac{65}{D^3} + 3\right)\left(\beta - 0.7\right)^{5/2} \qquad \text{(I-5-76)}$$

and

$$A = d\left(830 - 5000\beta + 9000\beta^2 - 4200\beta^3 + \frac{530}{\sqrt{D}}\right) \qquad \text{(I-5-77)}$$

Note: In equation (I-5-76), each of the last three terms, for some value of β, reduces to the form $x\sqrt{-1}$, i.e., to an "imaginary" number. In such cases the term is to be dropped.

I-5-39 For 1 D and 1/2 D taps and vena contracta taps, $K = K_0 + b\lambda$, and $\lambda = 1000/\sqrt{R_D} = 1000/\sqrt{\beta R_d}$. For the 1 D and 1/2 D taps,

$$K_0 = (0.6014 - 0.01352D^{-1/4}) + (0.3760 + 0.07257D^{-1/4})\left(\frac{0.00025}{D^2\beta^2 + 0.0025D} + \beta^4 + 1.5\beta^{16}\right) \qquad \text{(I-5-78)}$$

and

$$b = \left(0.0002 + \frac{0.0011}{D}\right) + \left(0.0038 + \frac{0.0004}{D}\right)[\beta^2 + (16.5 + 5D)\,\beta^{16}] \qquad \text{(I-5-79)}$$

For vena contracta taps,

$$K_0 = 0.5922 + 0.4252\left(\frac{0.0006}{D^2\beta^2 + 0.01D} + \beta^4 + 1.25\beta^{16}\right) \qquad \text{(I-5-80)}$$

and

$$b = 0.00025 + 0.002325\,(\beta + 1.75\beta^4 + 10\beta^{12} + 2D\beta^{16}) \qquad \text{(I-5-81)}$$

I-5-40 Sonic-Flow Primary Elements. In 1839, from theoretical studies of Bernoulli's and Venturi's works, Saint Venant and Wantzel developed a general equation of the discharge of fluids from aperatures by which the existence of a sonic-flow limit could be inferred. The phenomenon that the mass rate of flow of a gas through a nozzle reaches a maximum that is directly proportional to the inlet pressure was observed by Weisbach in 1866 and again by Fleigner in 1874. In recent years, the sonic-flow nozzle has been used as a reference meter, as a transfer standard, as a control for regulating the flow of a gas, and as a propulsion engine.

I-5-41 Maximum Theoretical Flow Rate of an Ideal Gas. From the equation of continuity, the theoretical mass flow rate per unit area is

$$\frac{m_T}{a} = \rho_2 V_2 \tag{I-5-82}$$

Using the relation $V_2 = M_2 V_s$ and the equation for an ideal gas in the form,

$$\rho_2 = \frac{p_2 (MW)}{RT_2} \tag{I-5-83}$$

equation (I-5-82) becomes

$$\frac{m_T}{a} = \frac{p_2 (MW) M_2 V_s}{R\, T_2} \tag{I-5-84}$$

For an ideal gas the acoustic speed is

$$V_s = \sqrt{\gamma\, g_c\, R\, T_2/(MW)} \tag{I-5-85}$$

Inserting this value into equation (I-5-84) and multiplying all terms by $\sqrt{T_{1t}}\,/\,p_{1t}$ develops the flow function, θ_i, for an ideal gas

$$\theta_i = \frac{m_T \sqrt{T_{1t}}}{a\, p_{1t}} = \frac{p_2}{p_{1t}} \sqrt{\frac{T_{1t}}{T_2}}\, M_2 \sqrt{\frac{g_c (MW)\, \gamma}{R}} \tag{I-5-86}$$

In order to simplify equation (I-5-86), the ratios of p_2/p_{1t} and T_{1t}/T_2 are needed. For an ideal gas the relation between the specific heats and the gas constant is

$$c_p - c_v = R/J\ (MW) \tag{I-5-87}$$

or

$$c_p = [\gamma/(\gamma - 1)]\, [R/J\ (MW)] \tag{I-5-88}$$

The first law of thermodynamics may be written

$$c_p T_{1t} = c_p T_2 + V_2^2/2 g_c J \tag{I-5-89}$$

Combining equations (I-5-88) and (I-5-89) and since $V_2 = M_2 V_s$,

$$\frac{T_{1t}}{T_2} = 1 + \frac{V_2^2}{2 g_c c_p J T_2} = 1 + \frac{(\gamma - 1)\, V_2^2 (MW)}{2 g_c\, \gamma\, R\, T_2} \tag{I-5-90}$$

Inserting the acoustic speed from equation (I-5-85),

$$\frac{T_{1t}}{T_2} = 1 + \frac{\gamma - 1}{2} M_2^2 \tag{I-5-91}$$

According to the isentropic process equation,

$$\frac{T_{1t}}{T_2} = \left(\frac{p_{1t}}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} \tag{I-5-92}$$

from which the ratio of p_{1t}/p_2 is found to be

$$\frac{p_{1t}}{p_2} = \left(1 + \frac{\gamma - 1}{2} M_2^2\right)^{\frac{\gamma}{\gamma - 1}} \tag{I-5-93}$$

Inserting these relations into equation (I-5-86) defines the flow function for an ideal gas entirely in terms of the throat Mach number and the gas properties.

$$\theta_i = \frac{m_T \sqrt{T_{1t}}}{a\, p_{1t}}$$

$$= M_2 \sqrt{\left(1 + \frac{\gamma - 1}{2} M_2^2\right)^{-\frac{\gamma + 1}{\gamma - 1}}} \sqrt{\frac{\gamma\, g_c (MW)}{R}} \tag{I-5-94}$$

The maximum mass rate of flow can be found by setting the logarithmic derivative of the flow function, equation (I-5-94), to zero.

$$\frac{d\theta_i}{\phi_i} = 0 = \frac{dM_2}{M_2} - \frac{\frac{(\gamma + 1)}{2} M_2 dM_2}{1 + \left(\frac{\gamma - 1}{2}\right) M_2^2} \tag{I-5-95}$$

$$\frac{dM_2}{M_2}\left[1+\frac{\gamma-1}{2}M_2^2-\frac{(\gamma+1)}{2}M_2^2\right]=0 \qquad \text{(I-5-96)}$$

Setting the bracketed term to zero,

$$1-\frac{1}{2}M_2^2-\frac{1}{2}M_2^2=0 \qquad \text{(I-5-97)}$$

$$M_2^2=1 \quad \text{and} \quad M_2=1 \qquad \text{(I-5-98)}$$

Thus, it is seen that the theoretical maximum mass rate of flow per unit area exists at a throat Mach number of unity. This is called "choked" flow and, frequently, also "critical" flow.

To show that the sonic flow occurs at the throat, the logarithmic differential of the continuity equation,

$$\frac{d\rho}{\rho}+\frac{dV}{V}-\frac{dA}{A}=0 \qquad \text{(I-5-99)}$$

is combined with the compressible-flow momentum equation

$$dp=-\frac{\rho}{g_c}\,VdV \qquad \text{(I-5-100)}$$

Then,

$$\frac{da}{a}=g_c\frac{dp}{\rho V^2}-\frac{d\rho}{\rho}=g_c\frac{dp}{\rho V^2}\left(1-\frac{V^2 d\rho}{g_c\,dp}\right) \qquad \text{(I-5-101)}$$

In isentropic flow the term, $g_c\,(dp/d\rho)$, is recognized as the square of the acoustic speed.

$$\frac{da}{a}=g_c\frac{dp}{\rho V^2}\left(1-M^2\right) \qquad \text{(I-5-102)}$$

A Mach number of unity occurs at $da = 0$, which means at the minimum conduit area. Additionally, this shows why sonic flow is not observed with a thin, square-edged orifice. With such an orifice, the function, da, is discontinuous at the "throat," and it does not smoothly attain the unique value of zero.

Since the maximum flow occurs in the throat at a Mach number of unity, the sonic flow function, θ_i^*, can be obtained from equation (I-5-94) by setting M_2 equal to unity. Thus,

$$\theta_i^*=\frac{m_T}{a^*}\,\frac{\sqrt{T_{1t}}}{p_{1t}}$$

$$=\sqrt{\gamma\left(\frac{1+\gamma}{2}\right)^{-\left(\frac{\gamma+1}{\gamma-1}\right)}}\sqrt{\frac{g_c\,(MW)}{R}} \qquad \text{(I-5-103)}$$

Let

$$F_i=\sqrt{\gamma\left(\frac{1+\gamma}{2}\right)^{-\left(\frac{\gamma+1}{\gamma-1}\right)}} \qquad \text{(I-5-104)}$$

F_i is called the ideal isentropic expansion function. Then the sonic flow function is

$$\theta_i^*=F_i\sqrt{g_c\frac{(MW)}{R}} \qquad \text{(I-5-105)}$$

I-5-42 The choking pressure ratio can be determined from equation (I-5-93) by setting M_2 equal to unity and inverting:

$$\frac{p_2}{p_{1t}}=\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \qquad \text{(I-5-106)}$$

When β approaches zero, the static pressure, p_1, approaches the total or stagnation pressure, p_{1t}, of the flowing fluid. For any value of the ratio, β, this total pressure can be measured with an impact or simple Pitot tube. But often, in the interest of minimizing upstream flow disturbances, the static pressure is measured with a pipe-wall pressure tap. The relation between p_1 and p_{1t} for any value of β is

$$\left(\frac{p_1}{p_{1t}}\right)^{2/\gamma}-\left(\frac{p_1}{p_{1t}}\right)^{\frac{\gamma+1}{\gamma}}$$

$$=\beta^4\,\frac{\gamma-1}{2}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}} \qquad \text{(I-5-107)}$$

For β less than 0.5 the following simpler equation is sufficiently accurate:

$$\frac{p_1}{p_{1t}} \approx 1 - \beta^4 \frac{\gamma}{2}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}} \qquad \text{(I-5-108)}$$

Note: The ratio between the outlet and inlet static pressures, $r_c = p_2/p_1$, at throat sonic velocity of an ideal gas, may be derived from equation (I-5-23) written in the form

$$m_T^2\left(\frac{1}{a^2}\right)\left(\frac{1}{2g_c\,p_1\,\rho_1}\right)\left(\frac{\gamma-1}{\gamma}\right)$$

$$= \left(r^{2/\gamma} - r^{\frac{\gamma+1}{\gamma}}\right)(1-\beta^4 r^{2/\gamma}) \qquad \text{(I-5-109)}$$

The derivative of equation (I-5-109), dm_T/dr, equated to zero reduces to

$$r_c^{\frac{1-\gamma}{\gamma}} + \left(\frac{\gamma-1}{2}\right)\beta^4 r_c^{2/\gamma} = \frac{\gamma+1}{2} \qquad \text{(I-5-110)}$$

Values of r_c for several values of β and γ are given in Table I-5-1.

I-5-43 As discussed in Par. I-3-17, the temperature indicated by a thermometer held in a moving stream is between the stagnation and static values. The use of thermometer wells designed to measure the stagnation temperature is recommended [15, 16, 17]. However, in isentropic flow, the stagnation temperature is conserved, even across the shock plane, so that a thermometer may be placed downstream and thus cannot affect the inlet flow.

I-5-44 Determination of the Maximum Ideal Flow Rate of a Real Gas. In actual use the gas discharged from a sonic flow primary element is a real gas. Real gases have two major deviations from the behavior of an ideal gas. First, they do not follow the ideal equation of state, pv/T = constant. Instead the real gas equation of state usually is written

$$\frac{p\,(MW)}{\rho R\;T} = Z\,(p,T) \neq 1 \qquad \text{(I-5-111)}$$

As indicated, Z is actually a function of pressure and temperature and in general varies between 0.2 and 4.2 [21]. Z is within 20 per cent of unity in the reduced pressure range from 6 to 9 and for reduced temperatures greater than unity. Also, Z is between 1.0 and 0.8 for reduced pressures between 0.0 and 0.5 and reduced temperatures greater then unity. The reduced pressure is the ratio of the actual pressure of the fluid to its critical pressure. Likewise, the reduced temperature is the ratio of the actual temperature of the fluid to its critical temperature. At the critical temperature of a fluid, the density of the gaseous and liquid phases are identical. Also, at temperatures above the critical no amount of pressure will cause a gas to condense. Second, the isentropic exponent of a real gas, Γ, is not constant and is not equal to the ratio of specific heats, c_p/c_v. Γ is also a function of stagnation pressure and temperature; but the effect of pressure is slight, and its value is given, usually, as a function of temperature alone.

Note: The effect of pressure on Γ is given by the relation

$$\frac{c_p/c_v}{\Gamma^*} = \frac{1-2cp^2}{Z} \cong \frac{1}{Z} \text{ when } p < 450 \text{ psia} \qquad \text{(I-5-112)}$$

where c is the second virial coefficient.

These differences must be included from the beginning in the derivation of a sonic flow function, ϕ^*, of a real gas. It is assumed that the gas flows isentropically and that sonic flow exists at the throat.

Table I-5-1 Values of r_c for Different Values of β and γ, as Determined by Equation (I-5-110)

β \ γ	1.10	1.15	1.20	1.25	1.30	1.35	1.40	1.45	1.667
0.00	0.5847	0.5744	0.5645	0.5550	0.5458	0.5369	0.5283	0.5200	0.4872
0.20	0.5849	0.5746	0.5647	0.5552	0.5460	0.5371	0.5285	0.5202	0.4874
0.40	0.5877	0.5775	0.5675	0.5580	0.5489	0.5401	0.5315	0.5232	0.4905
0.50	0.5923	0.5820	0.5722	0.5627	0.5537	0.5450	0.5364	0.5280	0.4956
0.60	0.6006	0.5905	0.5809	0.5715	0.5625	0.5538	0.5454	0.5372	0.5050
0.65	0.6072	0.5972	0.5877	0.5784	0.5694	0.5609	0.5526	0.5444	0.5124
0.70	0.6159	0.6061	0.5968	0.5875	0.5788	0.5703	0.5620	0.5541	0.5223
0.75	0.6280	0.6183	0.6092	0.6002	0.5914	0.5832	0.5750	0.5673	0.5358
0.80	0.6440	0.6347	0.6258	0.6171	0.6084	0.6005	0.5925	0.5849	0.5542
0.85	0.6671	0.6580	0.6496	0.6412	0.6332	0.6251	0.6177	0.6104	0.5809

Using * to denote the gas properties and conditions at sonic flow, then as for an ideal gas

$$\frac{p_1}{p_2} = \left(\frac{\rho_1}{\rho_2}\right)^{\Gamma} \qquad \text{(I-5-113)}$$

For pressure waves of small intensity, the acoustic speed is defined by

$$V_s = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} = \sqrt{\Gamma^* g_c \frac{Z\,R\,T}{(MW)}} = \sqrt{\Gamma^* g_c \frac{p}{\rho}} \qquad \text{(I-5-114)}$$

where Γ^* is the local value of the isentropic exponent at the speed of sound and not the mean value, Γ, required by equation (I-5-113). In order to proceed to the flow function, the choking pressure ratio must be found. Equation (I-5-113) is incorporated into the momentum equation, (I-5-100), and p is substituted for p_1 and is allowed to vary. This means that equation (I-5-113) must define the path of integration,

$$\int_0^V \frac{V\,dV}{g_c} = -\int_{p_{1t}}^{p_2} \frac{dp}{\rho} = \frac{p_2^{\,1/\Gamma}}{\rho_2} \int_{p_2}^{p_{1t}} p^{-1/\Gamma}\,dp \qquad \text{(I-5-115)}$$

where p_2 is taken as the fixed lower limit, and

$$\frac{V^2}{2g_c} = \frac{p_2^{\,1/\Gamma}}{\rho_2}\left(\frac{\Gamma}{\Gamma-1}\right)\left(p^{\frac{\Gamma-1}{\Gamma}}\right)\Bigg|_{p_2}^{p_{1t}} \qquad \text{(I-5-116)}$$

Inserting the limits

$$\frac{V^2}{2g_c} = \frac{p_2^{\,1/\Gamma}}{\rho_2}\left(\frac{\Gamma}{\Gamma-1}\right)\left(p_{1t}^{\frac{\Gamma-1}{\Gamma}} - p_2^{\frac{\Gamma-1}{\Gamma}}\right) \qquad \text{(I-5-117)}$$

Multiplying by $(p_2/p_2)^{\frac{\Gamma-1}{\Gamma}}$ and setting $V = V_s$ from equation (I-5-114),

$$\frac{p_2}{\rho_2}\left(\frac{\Gamma}{\Gamma-1}\right)\left[\left(\frac{p_{1t}}{p_2}\right)^{\frac{\Gamma-1}{\Gamma}} - 1\right] = \frac{\Gamma^* g_c p_2}{2 g_c \rho_2} \qquad \text{(I-5-118)}$$

from which the choking pressure ratio for the flow of a real gas is

$$\frac{p_2}{p_{1t}} = \left[\frac{\Gamma^*}{2}\left(\frac{\Gamma-1}{\Gamma}\right) + 1\right]^{-\frac{\Gamma}{\Gamma-1}} \qquad \text{(I-5-119)}$$

It is to be noted that this reduces to the pressure ratio for an ideal gas when $\Gamma^* = \Gamma = \gamma$.

As before, the maximum ideal rate of flow per unit area is determined by the continuity equation at sonic speed,

$$\frac{m_T}{a^*} = \rho_2 V_s = \sqrt{\Gamma^* g_c p_2 \rho_2} \qquad \text{(I-5-120)}$$

Inserting the stagnation pressure from equation (I-5-119) and the density from equation (I-5-113), which is

$$\rho_2 = \frac{p_{1t}(MW)}{Z\,R\,T_{1t}}\left(\frac{p_2}{p_{1t}}\right)^{1/\Gamma} \qquad \text{(I-5-121)}$$

equation (I-5-120) becomes

$$\frac{m_T}{a^*} = \sqrt{\frac{\Gamma^* g_c p_{1t}^2\,(MW)}{Z\,R\,T_{1t}}\left[1 + \frac{\Gamma^*}{2}\left(\frac{\Gamma-1}{\Gamma}\right)\right]^{-\frac{\Gamma+1}{\Gamma-1}}} \qquad \text{(I-5-122)}$$

Multiplying all terms by $\sqrt{T_{1t}}/p_{1t}$ completes the development of the sonic-flow function of a real gas [18, 19, 20],

$$\phi^* = \frac{m_T}{a^*}\frac{\sqrt{T_{1t}}}{p_{1t}} = \sqrt{\Gamma^*\left[1 + \frac{\Gamma^*}{2}\left(\frac{\Gamma-1}{\Gamma}\right)\right]^{-\frac{\Gamma+1}{\Gamma-1}}}\;\sqrt{\frac{g_c(MW)}{Z\,R}} \qquad \text{(I-5-123)}$$

Analogously to the ideal-gas derivation, the function, F, is called the real-gas isentropic expansion function, that is,

$$F = \sqrt{\Gamma^*\left[1 + \frac{\Gamma}{2}\left(\frac{\Gamma-1}{\Gamma}\right)\right]^{-\frac{\Gamma+1}{\Gamma-1}}} \qquad \text{(I-5-124)}$$

and the sonic-flow function may be written

$$\phi^* = F\sqrt{\frac{g_c(MW)}{Z\,R}} \qquad \text{(I-5-125)}$$

I-5-45 For certain gases, such as steam, ammonia and some of the refrigerants, the sonic-flow function varies much more than for other real gases. Also, the equations of state for most of these particular gases or "vapors" are tabulated in terms of pressure, temperature and specific volume. The existence of these tables makes it more convenient to calculate the mass rate of flow using the pressure and specific volume. Substituting the relation,

$$p_{1t}v_{1t} = \frac{Z R T_{1t}}{(MW)} \qquad \text{(I-5-126)}$$

into equation (I-5-22) and using equation (I-5-124),

$$\frac{m_T}{a^*} = F \sqrt{g_c} \sqrt{\frac{p_{1t}}{v_{1t}}} \qquad \text{(I-5-127)}$$

For the gases tabulated in Part II, the sonic flow functions and the isentropic expansion functions are presented as dimensionless ratios of the real gas function to the ideal-gas functions, (ϕ^*/ϕ_i^*) and (F/F_i). These tables show the departure of the real-gas sonic-flow from that of an ideal gas of like molecular structure and equal molecular weight, whose isentropic exponent is constant and equal to the theoretical ratio of specific heats, c_p/c_v. Applying the calibration coefficient relation,

$$Ca = a^* \qquad \text{(I-5-128)}$$

completes the development of the working sonic flow equations.

From equation (I-5-123)

$$m = a\,C\left(\frac{\phi^*}{\phi_i^*}\right)\phi_i^*\left(\frac{p_{1t}}{\sqrt{T_{1t}}}\right) \qquad \text{(I-5-129)}$$

The quantity, ϕ_i^*, is included in the tables of each gas and can be computed from equation (I-5-94).

From equation (I-5-127)

$$m = a\,C\,\sqrt{g_c}\left(\frac{F}{F_i}\right)F_i\sqrt{\frac{p_{1t}}{v_{1t}}} \qquad \text{(I-5-130)}$$

The quantity, F_i, is included in each table of F/F_i factors and can be computed from equation (I-5-104).

Note: In equation (I-5-129), if a is in sq. ft., p *must* be in psfa; and, if a is in sq in., p *must* be in psia. Likewise, in equation (I-5-130), as written a is ft^2 and p is psfa; but, if a is in.^{2} and p is psia, then the right side must be multiplied by 1/12.

I-5-46 The effect on ϕ^* of varying Γ^*, Γ and Z may be shown by the following procedure. Let

$$\Gamma^* = \Gamma + \Delta\Gamma = \Gamma\left(1 + \frac{\Delta\Gamma}{\Gamma}\right) = \Gamma(1+\epsilon) \qquad \text{(I-5-131)}$$

where ϵ represents $(\Gamma^* - \Gamma)/\Gamma$. Applying this to the right side of equation (I-5-116) gives approximately,

$$\sqrt{\frac{g_c(MW)\Gamma}{RZ\left(1+\frac{\Gamma-1}{2}\right)^{\frac{\Gamma+1}{\Gamma-1}}}\left(\frac{1+\epsilon}{1+\frac{\Gamma+1}{2}\epsilon}\right)}$$

$$\cong \sqrt{\frac{\Gamma}{\left(\frac{\Gamma+1}{2}\right)^{\frac{\Gamma+1}{\Gamma-1}}}}\left[1-\left(\frac{\Gamma-1}{4}\right)\epsilon\right]\sqrt{\frac{g_c(MW)}{ZR}} \qquad \text{(I-5-132)}$$

By a plot of the value of $\sqrt{\Gamma/[(\Gamma+1)/2]^{(\Gamma+1)/(\Gamma-1)}}$ versus Γ, the value of this radical between $\Gamma = 1.4$ and $\Gamma = 1.5$ is found to be very close to the straight line, $0.68\,[1 + 0.24(\Gamma - 1.4)]$, and the plotted points are only about 0.1 per cent below this line at $\Gamma =$ 1.3 and 1.6. Using this value for the radical gives

$$\phi^* = 0.68\,[1 + 0.24\,(\Gamma - 1.4)]\left[1 - \left(\frac{\Gamma-1}{4}\right)\right]\sqrt{\frac{g_c\,(MW)}{Z\;R}} \qquad \text{(I-5-133)}$$

Using $\sqrt{Z} = 1 + (1 - Z)/2$ as an approximation and neglecting the cross products of the second term in each bracket yields

$$1.46\sqrt{\frac{R}{g_c(MW)}}\,\phi^* = 1 + 0.50(1 - Z) + 0.24\,(\Gamma - 1.4) - \left(\frac{\Gamma-1}{4}\right)\epsilon \qquad \text{(I-5-134)}$$

Equal incremental changes in $(1 - Z)$, $(\Gamma - 1.4)$ and ϵ affect the value of ϕ^* in a decreasing order of importance of about 1/2, 1/4 and 1/8 times each change, respectively. Although the value of Z may be determined from tables for many real gases, the mean value of Γ may not be tabulated, so that its value would have to be determined. The local value of Γ can be determined from the acoustic speed data,

$$\Gamma^* = V_s^2/g_c\,Z\,R\,T = \frac{(c_p/c_v)\,Z}{1 - 2cp^2} \qquad \text{(I-5-135)}$$

where c is the second virial coefficient (about 10^{-6}/atm^2).

I-5-47 Determination of the Maximum Flow Rate of a Real Gas. The sonic-flow functions for a real gas such as steam, most refrigerants and many other commercial gases can be determined by the following theoretical method, which is sometimes called the "enthalpy" method. It is the most practical procedure to use for those gases of which the properties are tabulated. The downstream rate of flow is determined from the continuity equation and from an isentropic conservation of energy flow.

$$dw + J dq = dZ + \frac{V dV}{g_c} + J dH \qquad \text{(I-5-136)}$$

where w = work done by the fluid. For no work done, i.e., $dw = 0$, no heat added, and horizontal flow

$$\int_0^{V_2} \frac{V dV}{g_c} = -J \int_{1_t}^{2} dH \qquad \text{(I-5-137)}$$

or

$$\frac{V_2^2}{2g_c} = J(H_{1t} - H_2) \qquad \text{(I-5-138)}$$

The equation of continuity for use with tables is

$$\frac{m_T}{a} = \frac{V_2}{v_2} = \frac{1}{v_2} \sqrt{2g_c J(H_{1t} - H_2)} \quad \text{(I-5-139)}$$

The procedure consists of first fixing the inlet stagnation conditions and then using in turn several values of the downstream (i.e., throat) pressure and iteratively solving for the mass rate of flow until a maximum is indicated. Referring to Fig. I-5-7(a), the inlet stagnation conditions are known or assumed which fixes the inlet enthalpy. Then,

1. A value of the downstream pressure, p_2, is assumed and thereby a value of H_2, with which a value of the downstream velocity, V_2, is computed by equation (I-5-138).
2. Using equations (I-5-113) (if v_2 is not tabulated), (I-5-139), (I-5-127) and (I-5-125) in that order, values of ρ_2, m_T/a, F and ϕ^* are determined and the value of ϕ^* plotted as in Fig. I-5-7(b).
3. Steps 1 and 2 are repeated until the maximum value of ϕ^* is established.

For these computations it has been found advantageous to use the temperature the gas would have if it

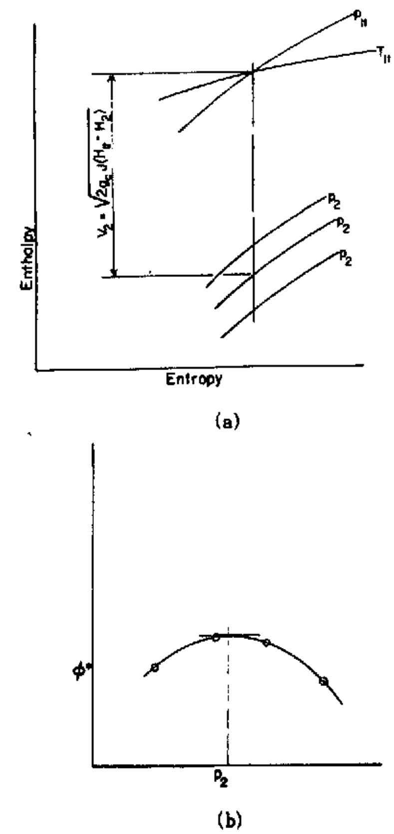


FIG. I-5-7 FLOW-MAXIMIZATION PROCEDURE

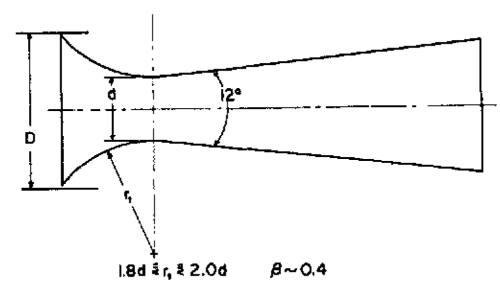


FIG. I-5-8 PROPORTIONS OF INTERIOR SURFACE OF CIRCULAR-ARC VENTURI

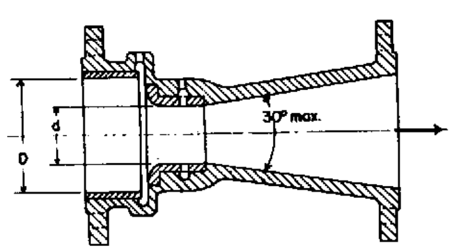


FIG. I-5-9 NOZZLE-VENTURI TUBE

were throttled isenthalpically, i.e., H = constant, to a very low pressure.

This method may be programmed for digital computer use where such equipment is used or available.

In Part II the functions are presented in the same format as in the preceding derivations for real gases. The use of the "working" equations, (I-5-127) and (I-5-129), is not affected.

I-5-48 Figure I-5-8 shows the proportions of the interior surface of a nozzle-Venturi, or circular-arc Venturi, used at sonic throat-velocity conditions in tests of jet-type propulsion engines. The overall pressure loss of these nozzle-Venturis may be as low as 15 percent of the pressure drop required to produce sonic velocity at the throat. The coefficient of discharge is between 0.990 and 0.995 [22, 23, 24, 25].

I-5-49 Modifications of the Three Basic Primary Elements: The Nozzle-Venturi. The nozzle-Venturi (Fig. I-5-9) was developed to provide a primary element of shorter overall length than the conventional Venturi tube but without a too greatly increased pressure loss. Although it has been used to a slight extent in this country, it has been used abroad enough to justify including it in the recommendation of the ISO Committee on Flow Measurement [11]. An abstract of these recommendations on the proportions and discharge coefficient is given in Part II.

I-5-50 Other Modifications of Venturi Tubes and Flow Nozzles. Several forms of flow tubes have been developed which combine features of the Venturi tube and the flow nozzle, as shown in Fig. I-5-10. The primary objective in the design of these special

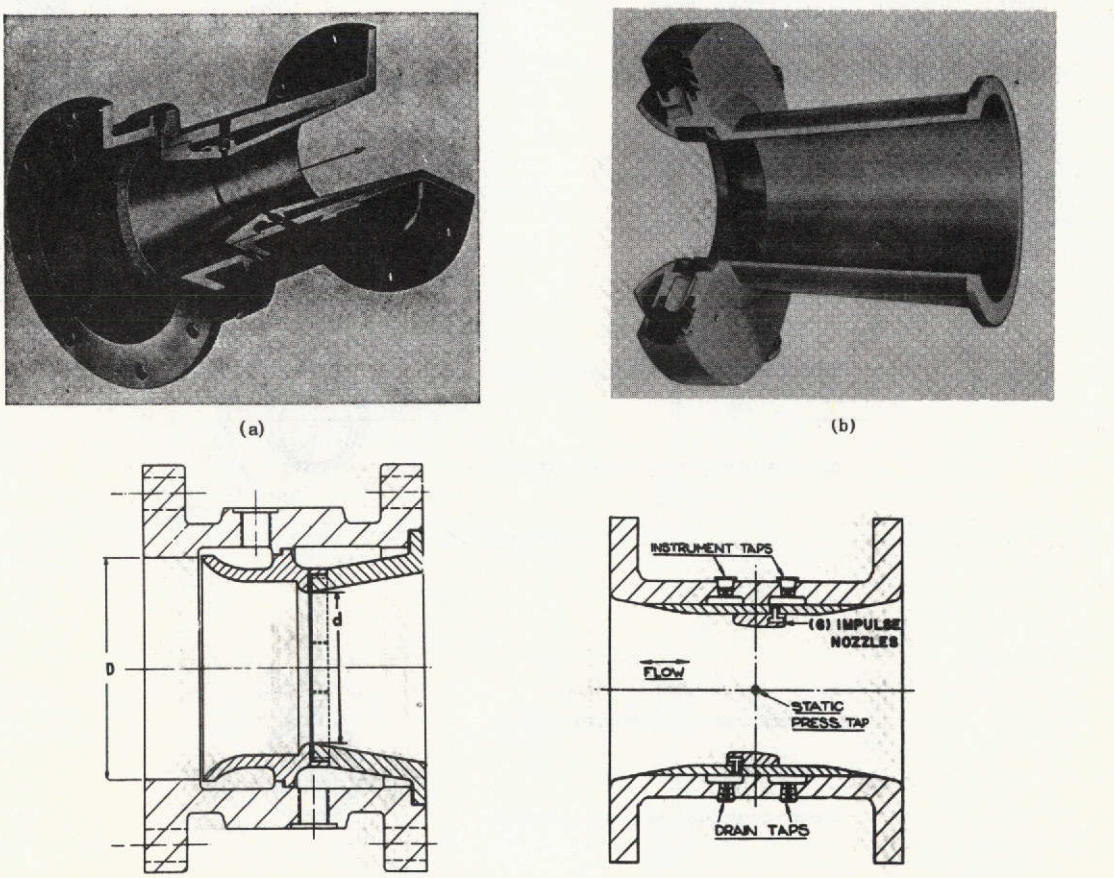


FIG. I-5-10 MODIFICATIONS OF FLOW NOZZLES AND VENTURI TUBES (PROPRIETARY) forms of flow tubes was to obtain a high differential pressure with as low an overall pressure loss as possible. To a notable extent these objectives were achieved, due in part to boundary layer effects. Because these special forms are proprietary, the committee presents no specific data on individual tubes [26–29].

I-5-51 Quadrant-Edge and Conical-Edge Orifices. The inlet edge of the quadrant-edge orifice is rounded, and usually the radius of the rounding is equal to the plate thickness (Fig. I-5-11). The angle of the entrance cone of the conical-edge orifice may range from 40 to 80 deg as measured from the face of the plate. These forms of orifices have the characteristic of having an almost flat, or constant, value of the discharge coefficient over a relatively low range of flow rates, as may be represented by values of R_d below about 50,000. For this reason these orifices have been the subject of test programs in several laboratories. Some of these programs have shown that the installation conditions and the velocity profile of the approaching fluid affect both the value and range of "flatness" of the coefficient. The ISO Technical Committee on Flow Measurement is asking that further studies be made on these orifices, especially the conical-inlet orifice [30, 31].

I-5-52 Eccentric and Segmental Orifices. A circular eccentric orifice is constructed and installed so that its center is not on the axis of the pipe. Generally, the eccentricity is such that one side of the

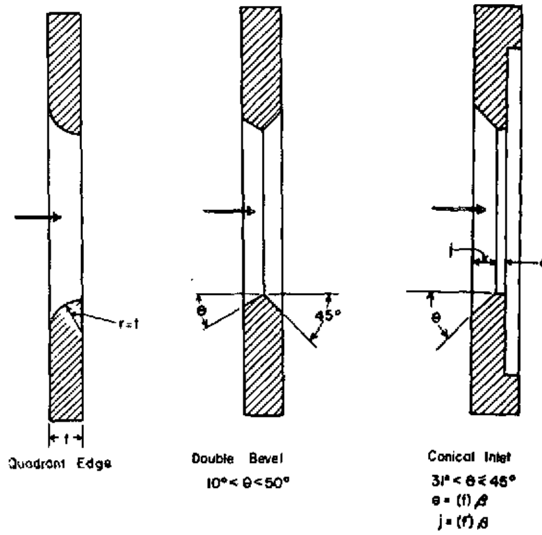


FIG. I-5-11 SPECIAL ORIFICE FORMS FOR MEASUREMENT OF LOW RATES OF FLOW

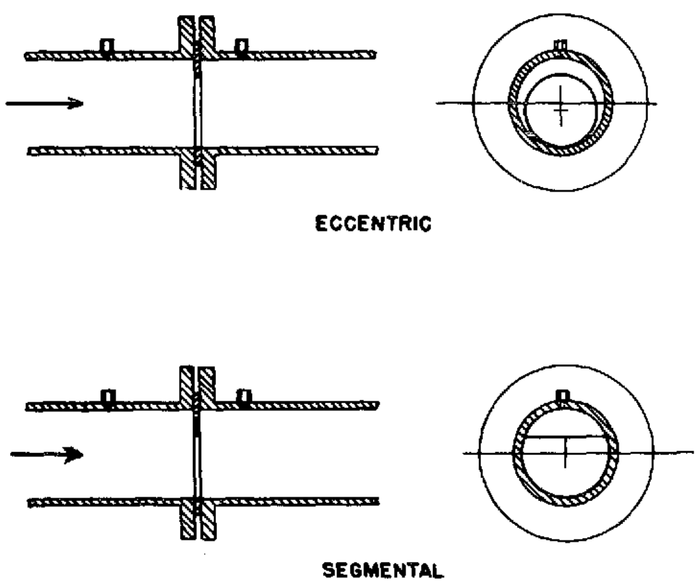


FIG. I-5-12 ECCENTRIC AND SEGMENTAL ORIFICES circular opening is substantially flush with the inside wall of the pipe. The point at which the eccentric opening is nearest to, or flush with, the wall of the pipe is diametrically opposite the pressure taps, thus placing the maximum depth of dam height on the side of the pressure taps, as shown in Fig. I-5-12.

Note: Sometimes with eccentric orifices, the pressure taps are located 90 deg around the pipe from the top when, because of the pipe location, the taps must be on the side of the pipe while the orifice hole must be at the bottom to provide maximum flushing.

With the segmental orifice, the shape of the restricting area is a segment of a circle of approximately the same diameter as that of the pipe. Thus, the orifice opening is irregular in shape. Sometimes, with very large pipes, only that part of the orifice plate forming the segment is used. The segmental orifice is installed with the segment on the same side of the pipe as the pressure taps.

Note: With segmental orifices the primary dimensional relation is the area ratio, a/A. The diameter ratio has the relative significance given by the relation, $\beta = \sqrt{a/A}$ However, it may be convenient, particularly for comparative purposes, to use this equivalent value of β as a parameter in the presentation of data on segmental orifices.

One reason for using an eccentric or segmental orifice is that the plane of the vena contracta is displaced farther downstream from where it would be with a concentric circular orifice of the same area ratio. Another reason for using them is that, by placing those portions of the openings, which are flush with the inside surface of the pipe, at the bottom (of horizontal lines), complete drainage of any extraneous matter will be obtained. This minimizes the danger of such matter affecting the accuracy of the meter.

Curves of the discharge coefficients and expansion factors for eccentric and segmental orifices are given in Part II [32, 33].

I-5-53 Overall Pressure Loss. The relative overall pressure losses associated with the several general types of differential pressure primary elements so far described and discussed are shown in Fig. I-5-13. It will be seen that the thin-plate square-edged orifice

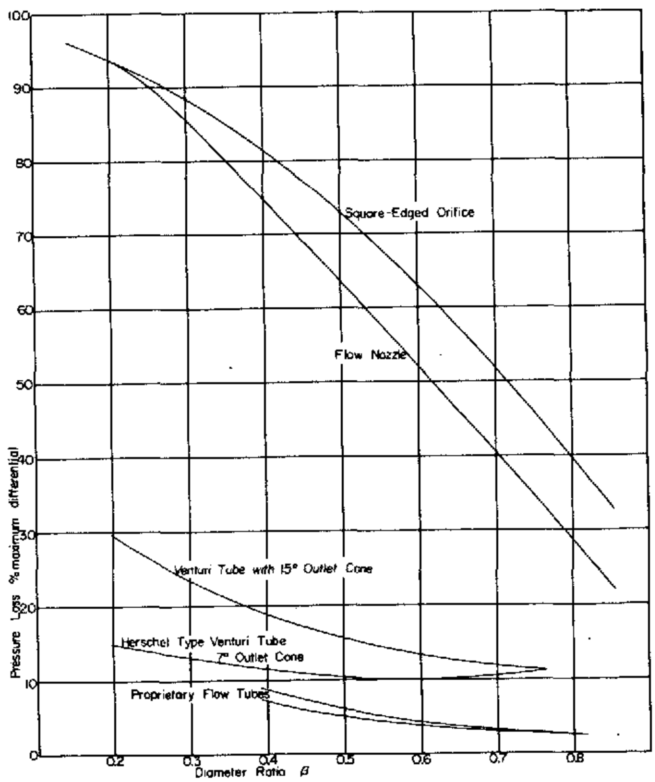


FIG. I-5-13 OVERALL PRESSURE LOSS THROUGH SEVERAL PRIMARY ELEMENTS has the highest loss, while some of the special forms of Fig. I-5-10 have the lowest. However, the extreme simplicity, reproducibility and adaptability of the orifice are largely responsible for its being the most widely used differential pressure element.

I-54 Centrifugal Meters: The Elbow Meter. When a fluid flows along a curved channel, it is subjected to angular acceleration, and the basic relation between acceleration, force and mass applies. The force in this case is evidenced by the difference between the pressures which are observed at the outside and inside of the curve, especially in closed channels. Thus, a very simple form of a centrifugal meter is a common pipe elbow, with pressure taps in the outer and inner surfaces in the plane determined by the curved center line of the elbow (Fig. I-5-14). Although for most of the tests of elbow meters that have been reported the pressure taps were located in a radial plane 45 deg from the elbow inlet, other locations have been used, notably 22 ½ deg from the inlet. An advantage of the 45-deg location is that flow in either direction can be measured. Special forms of elbows have been tried but gained little attention.

I-5-55 To develop a rational flow equation for an elbow meter, it is helpful to assume that the cross section of the elbow is rectangular. That the final equation can be applied to any cross-sectional shape is shown later. Other simplifying assumptions are the following:

1. The elbow is in a horizontal plane.
2. A uniform distribution of velocity exists over the cross section of the elbow.
3. There is no loss of either pressure or velocity from inlet face to outlet face.
4. There is no effect of fluid viscosity.
5. A uniform distribution of pressure exists over the inside and outside walls of the elbow.
6. There is a steady state of flow with respect to time.

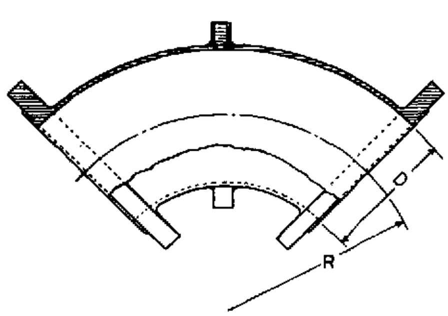


FIG. I-5-14 ELBOW METER

The symbols and units used in the development of an equation are:

A = Radial cross-sectional area of elbow	ft^2
D = Radial width of a rectangular elbow, also diameter of a circular cross section	ft
F = Force, due to fluid momentum	lb_f
p = Pressure	lb_f/ft^2
R = Radius of elbow center line	ft
S = Resultant sidewall force	lb_f
V = Fluid velocity	ft/sec
η (eta) = Height of duct section	ft
ρ (rho) = Fluid density	$slug/ft^3$

Subscript i = Inside

o = Outside

L = Pipe line (for pressure and velocity at inlet and outlet faces)

x = x-direction vector component

y = y-direction vector component

z = z-direction vector component perpendicular to the x-y plane (not shown in Fig. I-5-15).

The momentum flux equation as applied to an elbow states that the net force on the surface of the elbow equals the net flux of momentum through the surfaces of the elbow plus the rate of change of momentum inside the elbow, that is,

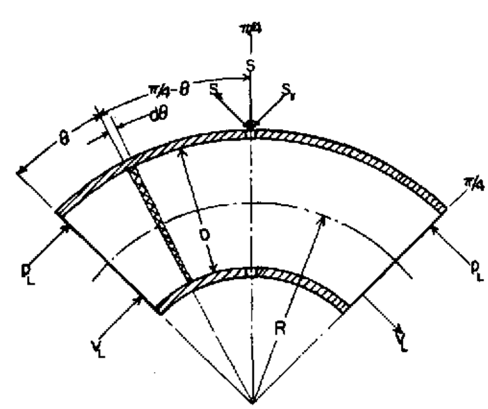


FIG. I-5-15 ELEMENTS OF FLUID FLOW THROUGH ELBOW METER

$$F + \int(-g\,\rho\,z)\,d(\text{vol}) = \int V(\rho\,V\cdot dA) + \frac{\partial}{\partial t}\int V\,\rho\cdot d(\text{vol}) \quad \text{(I-5-140)}$$

By the first assumption above, the second term disappears; and the last term disappears by the last assumption, so that equation (I-5-140) reduces to

$$F = \int V(\rho\,V\cdot dA) \quad \text{(I-5-141)}$$

Taking force components in the x- and y-direction

$$F_x = \int V_x(\rho\,V\cdot dA) \text{ and } F_y = \int V_y(\rho\,V\cdot dA) \quad \text{(I-5-142)}$$

The force, F, is composed of two parts, the pressure force on the flow areas of the elbow and the resultant force on the sidewalls of the elbow, S, for which the equation will be solved.

$$F_x = S_x - p_L A = \int V_{Lx}(\rho\,V_L\cdot dA) = \rho A\,V_L^2 \quad \text{(I-5-143)}$$

$$F_y = -S_y + p_L A = \int V_{Ly}(\rho\,V_L\cdot dA) = \rho A\,V_L^2 \quad \text{(I-5-144)}$$

$$S_x = \rho A V_L^2 + p_L A \text{ and } S_y = -\rho A V_L^2 - p_L A \quad \text{(I-5-145)}$$

$$S = \sqrt{S_x^2 + S_y^2} = \sqrt{2}\,\sqrt{(\rho A\,V_L^2 + p_L A)^2} \quad \text{(I-5-146)}$$

This is the net force on the sidewalls which is responsible for turning the flow. Equations (I-5-143) and (I-5-144) are easily generalized to include elbows of more or less than 90 deg. The number, S, in equation (I-5-146) is the value of a three-dimensional vector integral of the pressure over the sidewall surface of the elbow. To formulate this integral,

$$S = \int p\,dA \quad \text{(I-5-147)}$$

where dA is a vector quantity, and

$$dA_i = (R - D/2)\,\eta\,d\theta \text{ and } dA_o = (R + D/2)\,\eta\,d\theta \quad \text{(I-5-148)}$$

where $d\theta$ is the elemental angular movement of the fluid (Fig. I-5-15).

A uniform pressure, p_o, on the outside and p_i on the inside multiplying the differential area vector forms a differential force which has two components: one parallel to S, the other normal to it. These normal components taken about the axis of symmetry, $\pi/4$ (Fig. I-5-15), are symmetrically opposing. In addition, the pressure distribution on the top and bottom of the duct may be assumed to be symmetrically opposing. Thus, the total value of the force, S, over the sidewalls is

$$S = \int_o^{\pi/2} p_o\eta(R + D/2)\cos(\pi/4 - \theta)\,d\theta - \int_o^{\pi/2} p_i\eta(R - D/2)\cos(\pi/4 - \theta)\,d\theta \quad \text{(I-5-149)}$$

Now,

$$\cos(\pi/4 - \theta) = \cos(\pi/4)\cos\theta + \sin(\pi/4)\sin\theta$$

and

$$\cos(\pi/4) = \sin(\pi/4) = 1/\sqrt{2}$$

Also, $\displaystyle\int_o^{\pi/2}\cos\theta\,d\theta = \int_o^{\pi/2}\sin\theta\,d\theta = 1$

so that

$$S = p_o\eta(R + D/2)2/\sqrt{2} - p_i\eta(R - D/2)2/\sqrt{2} \quad \text{(I-5-150)}$$

or

$$S = (p_o - p_i)\,\eta R\sqrt{2} + \frac{(p_o + p_i)}{2}\,\eta D\sqrt{2} \quad \text{(I-5-151)}$$

Combining equations (I-5-151) and (I-5-146)

$$(p_o - p_i)\,\eta R\sqrt{2} + \frac{(p_o + p_i)}{2}\,\eta D\sqrt{2} = \sqrt{2}\,\sqrt{(\rho\eta D V_L^2 + p_L D)^2} \quad \text{(I-5-152)}$$

from which

$$(p_o - p_i) = \frac{\rho\eta D V_L^2}{\eta R} + \frac{p_L\eta D}{\eta R} - \frac{(p_o + p_i)\,\eta D}{2\eta R} \quad \text{(I-5-153)}$$

By equation (I-5-153) it appears that the height of a rectangular duct, η, is self-eliminating. On the basis of the assumption of no pressure loss between inlet and outlet, the approximate relation,

$(p_o + p_i)/2 \cong p_L$, may be applied to equation (I-5-153) to give

$$p_o - p_i = \Delta p = \frac{D}{R} \rho V_L^2 \qquad \text{(I-5-154)}$$

I-5-56 Equation (I-5-154) is an approximate theoretical equation for the measurement of fluid flow with an elbow and is applicable to any cross section that is symmetric with respect to the center-line plane. The theoretical rate of flow is

$$q_T = A V_L = A \sqrt{\frac{R}{D} \frac{\Delta p}{\rho}} \qquad \text{(I-5-155)}$$

Introducing a flow coefficient, K, determined by a calibration, and using the customary units of measurement (namely, A, in.^{2}; D and R, in.; p_L and Δp, psi; and ρ, lb$_m$/ft^3) gives as the equation for computing the rate of flow

$$q = 0.47268\ A\ K \sqrt{\frac{R}{D} \frac{\Delta p}{\rho}} \qquad \text{(I-5-156)}$$

cfs at the fluid density, ρ

or

$$q = 0.37125\ K\ D\ \sqrt{D\ R\ \Delta p/\rho} \quad \text{cfs} \qquad \text{(I-5-157)}$$

and

$$m = 0.37125\ K\ D\ \sqrt{D\ R\ \rho\ \Delta p} \quad \text{lb}_m/\text{sec} \qquad \text{(I-5-158)}$$

I-5-57 Unlike the three principal differential pressure meters, elbow meters have not been the object of tests and structual recommendations by a committee of the ASME or other technical societies. However, a review of published experimental data on elbow meters indicated that the relative roughness of the elbow surface had no significant effect on flow measurements. Also, for 90-deg elbows with pressure taps at 45 deg and tap hole diameters, δ, as recommended for orifice meters, the value of K may be computed by

$$K = 1 - \frac{6.5}{\sqrt{R_D}} \qquad \text{(I-5-159)}$$

when $10^4 \leq R_D \leq 10^6$, and $R/D \geq 1.25$. Flows computed with this value of K and uncalibrated elbows will be subject to a tolerance (uncertainty) of about ± 4 per cent. With a calibrated elbow, the tolerance should be comparable to that for other types of differential pressure meters. With either calibrated or uncalibrated elbows meters, a high degree of repeatability is attainable [34, 35].

I-5-58 The use of the scroll case of a turbine or pump and the guide-vane speed ring of a turbine may be classed as forms of centrifugal meters. In order to use either of these as a means of flow measurement, a calibration must be made using some other method of measurement, such as a tracer method. (See Chapter I-I-9 and [36].)

I-5-59 **Linear-Resistance Meters.** With these meters, the outstanding characteristic is the linear relationship between the pressure drop and the flow rate from zero flow up to some maximum rate. Because of this linear relationship, these meters have the same flow coefficient over this range; and, within this range, the determination of this coefficient at one rate is sufficient. Above this linear maximum rate, the pressure drop begins to increase very gradually at a faster rate than the flow. Hence, when calibrating one of these meters, enough different rates should be covered to determine the upper limit of linearity. If the meter is to be used above this point, the relation between rate of flow and pressure drop should be determined.

The two most common forms of primary elements are capillary tubes and porous plugs, as illustrated in Fig. I-5-16. With the capillary tube, the ratio of length to bore is the principal factor determining the relationship between rate of flow and pressure drop. Some writers recommend that the length should be 150 or more times the bore. A second factor is the character of the entrance and exit ends of the tube, i.e., whether these are square or smooth and well tapered [37]. Instead of a single tube, a bundle of capillary passages may be used to increase the flow capacity. Also, it is not necessary for the passage to be of circular cross section or even of a uniform shape [38].

The porous-plug form can be made by fastening a plug of suitable porous material within a section of pipe or tubing and providing connections for measuring the pressure drop across the plug. Some of the materials which may be satisfactory for the plug are steel wool, cotton waste, sintered alumina, glass wool, and layers of fine screening so placed that the wires of adjacent layers are not parallel. The first two are suited to use with dry gases; the sintered alumina could be used at elevated temperatures; and the last two could be used with wet gases or even liquids. With any given plug material, the principal factors influencing the rate-pressure drop relation are the tightness with which the material is packed and the length of the plug [39].

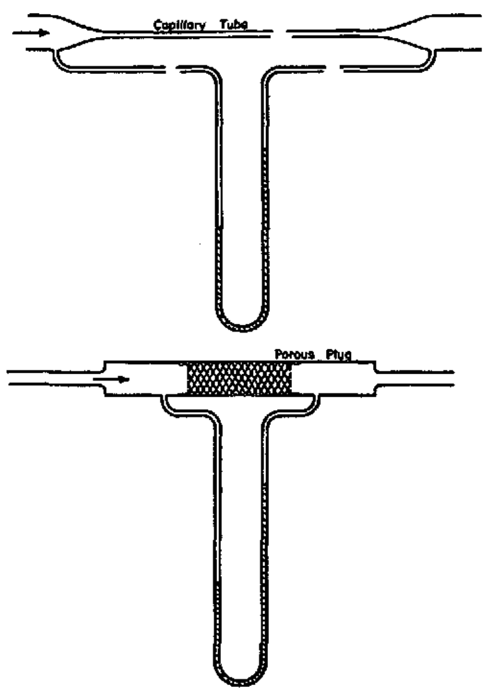


FIG. I-5-16 TWO FORMS OF LINEAR-RESISTANCE FLOW METERS

I-5-60 In order to have a linear relationship between the rate of flow and the pressure drop across the metering unit, it is necessary that the flow rate is low enough for the pressure drop to be a measure of the viscous drag. The transition from laminar flow (or viscous drag flow) to turbulent flow usually takes place at a Reynolds number between 2000 and 2500. Therefore, in order to meet the above requirement the Reynolds number applying to the metering passage should be less than 2000 (and below 1500 if possible). The basic relation for the flow through capillary tubes, commonly referred to as Poiseuille's equation, is [40]:

$$q = \frac{\pi}{8} C \frac{D^4}{16} \frac{1}{L\mu} (p_1 - p_2) \qquad \text{(I-5-160)}$$

If normal units were used, all linear dimensions would be in ft, ft² and ft³. However, the common practice in fluid metering is to have q in ft³/sec, while D, L and $(p_1 - p_2)$ are measured in in. and psi. To take account of this, we must write

$$q = \frac{\pi}{8} \frac{1}{16} C \frac{D^4}{(12)^4} \frac{12}{L} \frac{144\,(p_1 - p_2)}{\mu} \qquad \text{(I-5-161)}$$

$$= \frac{\pi C D^4}{1536\, L} \left(\frac{p_1 - p_2}{\mu}\right)$$

The coefficient, C, must be determined by a calibration. The other quantities are:

D = Diameter of the capillary tube, in.

L = Length of tube between points of pressure measurements, in.

$(p_1 - p_2)$ = pressure drop over the tube length, L, psi.

q = Volume rate of flow at the mean temperature and the mean pressure, cfs

μ = Viscosity of the fluid, lb_m/ft-sec

For use with a bundle of tubes, including the irregular areas between them, a multiplicity of thin, slit-like passages, or even a porous plug, the diameter term may be replaced with $D^4 = (16A)^2/\pi^2$, where A = the total cross-sectional area of the passages through which the fluid flows, in.². Thus, including the coefficient, C, the flow equation becomes

$$q = \frac{C\ A^2}{96\,\pi\, L} \left(\frac{p_1 - p_2}{\mu}\right) \qquad \text{(I-5-162)}$$

Since a direct determination of A may be difficult, or impossible, the product CA^2, or even CA^2/L, may be determined by the calibration [39].

In the general use of linear-resistance meters, especially capillary tubes, the effects of inlet and exit losses, and in the case of a coiled capillary, the effects of "slip" and curvilinear flow are usually included in the calibration coefficient, C. Also, when the flow rate increases to the point where laminar flow begins to break into turbulent flow, these equations will cease to hold [41, 42].

I-5-61 Frictional-Resistance Meters: Pipe Section. When the velocity of the fluid in a pipe line is very high, as in the steam and water lines of some modern power plants, the frictional pressure drop becomes significant. Hence, the difference in the static pressures between two sidewall pressure taps located an appreciable distance apart in a straight section of pipe may be measured readily with an appropriate differential pressure gage. Thus, when adequately calibrated, such a section of pipe with its pressure taps may be a satisfactory primary element for some purposes. Of course, each such section of pipe must be individually calibrated for the range of flows and the fluids to be metered.

If the object of the measurement is simply to monitor the relative steadiness of the flow, with little or no interest in the actual rate, it may suffice to use the pipe-line flow equation with an appropriate value of the friction factor. For this purpose, the equation for flow in a pipe may be written

$$m = 0.1515\, D^{5/2} \sqrt{\frac{\rho\, \Delta p}{f\, L}} \quad lb_m/sec \qquad (I\text{-}5\text{-}163)$$

in which D is in inches, Δp, in psi, and L, in ft, is the distance between the pressure taps. The friction factor, f, is a function of the relative roughness, ϵ/D and R_D. For steel pipe 4 in. i.d. and over and $R_D \geq 5(10^5)$, an average value of $f = 0.015$ may be assumed, and [43]

$$m = 1.237\, D^{5/2} \sqrt{\frac{\rho\, \Delta p}{L}} \qquad (I\text{-}5\text{-}164)$$

References

[1] "Notes on the Orifice Meter: The Expansion Factor for Gases," E. Buckingham; *Journal of Research*, National Bureau of Standards, vol. 9, July 1932, RP. 459, p. 61.

[2] "Einfluss der Expansion auf die Kontraktion hinter Stauranden (Influence of Expansion Upon the Contraction After an Orifice)," G. Ruppel; *Technische Mechanik und Thermodynamik*, vol. 1, Apri. 1930, p. 151 and Sept. 1930, p. 338.

[3] Discharge Coefficients of Square-Edged Orifices for Measuring the Flow of Air," H. S. Bean, E. Buckingham and P. S. Murphy; *Journal of Research*, National Bureau of Standards, vol. 2, Mar. 1929, RP. 49, p. 561.

[4] "Coefficients of Discharge of Sharp-Edged Concentric Orifices in Commercial 2-In., 3-In., and 4-In. Pipes at Low Reynolds Numbers Using Flange Taps," E. E. Ambrosius and L. K. Spink; *Trans. ASME*, vol. 69, Nov. 1947, p. 805.

[5] "Orifice Discharge Coefficients for Viscous Liquids," G. L. Tuve and R. E. Sprenkle; *Instruments*, vol. 6, Nov. 1933, p. 201.

[6] "Experiments on the Metering of Large Volumes of Air," H. S. Bean, M. E. Benesh and E. Buckingham; *Journal of Research*, National Bureau of Standard, vol. 7, July 1931, RP 335, p. 93.

[7] "Values of Discharge Coefficients of Square-Edged Orifices, Comparison of Results Obtained by Tests Using Gases with Those Obtained by Tests Using Water," H. S. Bean; *American Gas Association Monthly*, vol. 17, July 1935, p. 259.

[8] "Uber die Expansionszahl bei der Durchflussmessung mit Normblenden (About the Expansion Factor for the Measurement of Flow with Standard Orifices)," Guy Thibessard; *Sonderdruck aus Brennstoff-Warme-Kraft*, Bd. 12, 1960, Nr. 3 S. 97/101.

[9] "The Mass Flow Meter: A Method of Measuring Flow," D. Brand and L. A. Ginsel; *Instruments*, Mar. 1951, p. 331.

[10] "An Orifice Meter that Measures True Mass Flow," B. Fishman; 17th Annual Instrumentation-Automation Conference, Oct. 1962, paper 16.4.

[11] ISO Recommendation R 541, *Measurement of Fluid Flow by Means of Orifice Plates and Nozzles*, and ISO Recommendation 781, *Measurement of Fluid Flow by Means of Venturi Tubes*.

[12] "The Flow of Water Through Orifices," S. R. Beitler; Ohio State University Engineering Experiment Station Bulletin No. 89, May 3, 1935.

[13] "History of Orifice Meters and the Calibration, Construction, and Operation of Orifices for Metering;" Report of the Joint AGA–ASME Committee on Orifice Coefficients, 1935; Reprint by ASME, Dec. 1936.

[14] "A Statistical Approach to the Prediction of Discharge Coefficients of Concentric Orifice Plates," Rodger B. Dowdell and Yu-Lin Chen; ASME Paper, 1969.

[15] "Thermodynamics of Compressible Fluid Flow," A. H. Shapiro; The Roland Press, vol. 1, 1953, p. 79–83.

[16] "Measurement of Temperatures in High-Velocity Steam," J. W. Murdock and E. F. Fiock; *Trans. ASME*, vol. 72, Nov. 1950, p. 1155.

[17] "Measurement of Temperatures in High-Velocity Gas Streams," W. J. King; *Trans. ASME*, vol. 65, July 1943, p. 421.

[18] "Computation of the Critical Flow Function, Pressure Ratio and Temperature Ratio for Real Air," R. M. Reimer; *Trans. ASME, Journal of Basic Engineering*, vol. 86–2, June 1964, p. 169.

[19] "The Critical Flow Function for Superheated Steam," J. W. Murdock and J. M. Bauman; *Trans. ASME, Journal of Basic Engineering*, Sept. 1964, p. 507.

[20] "Review of Critical Flowmeters for Gas Flow Measurements," B. T. Arnberg, *Trans. ASME, Journal of Basic Engineering*, vol. 84–4, Dec. 1964, p. 447.

[21] "Tables of Thermal Properties of Gases," National Bureau of Standards, Circular 564, 1955.

[22] "A Theoretical Method of Determining Discharge Coefficients for Venturis Operating at Critical Flow Conditions," R. E. Smith and R. J. Matz; *Trans. ASME, Journal of Basic Engineering*, vol. 84–4, Dec. 1962, p. 434.

[23] "Accurate Measurement of Airflow Rate During Tests of Air-Breathing Propulsion Systems," R. E. Smith, Jr. and H. E. Wolff; Arnold Engineering Development Center, Tr.–68–22.

[24] "The Calculation of the Discharge Coefficient of Profiled Choked Nozzles and the Optimum Profile for Absolute Air Flow Measurements," B. S. Stratford; *Aeronautical Journal*, Royal Aeronautical Society (London), vol. 68, Apr. 1964, p. 237.

[25] "A Standard Choked Nozzle for Absolute Calibration of Air Flowmeters," D. W. Sparkes; *Aeronautical Journal*, Royal Aeronautical Society (London), vol. 72, Apr. 1968, p. 335.

[26] "The Dahl Flow Tube," I. O. Miner; *Trans. ASME*, vol. 78, Apr. 1956, p. 475.

[27] "Design and Calibration of the Lo-Loss Tube," L. J. Hooper; *Trans. ASME, Journal of Basic Engineering*, vol. 84–4, Dec. 1962, p. 461.

[28] "Calibrations of Six Beth-Flow Meters at Alden Hydraulic Laboratory, Worcester Polytechnic Institute," L. J. Hooper; *Trans. ASME*, vol. 72–8, Nov. 1950, p. 1099.

[29] "The Twin Throat Venturi: A New Fluid-Flow Measuring Device," A. A. Kalinske; *Trans. ASME, Journal of Basic Engineering*, vol. 82–3, Sept. 1960, p. 710.

[30] "The Quadrant-Edge Orifice: A Fluid Meter for Low Reynolds Numbers," M. Bogama and P. L. Monkmeyer; *Trans. ASME, Journal of Basic Engineering*, vol. 82–3, Sept. 1960, p. 729.

[31] "Quadrant-Edge Orifice Performance, Effect of Upstream Velocity Distribution," M. Bogama, B. Spring and M. V. Ramamoorthy; *Trans. ASME, Journal of Basic Engineering*, vol. 84–4, Dec. 1962, p. 415.

[32] "Calibration of Eccentric and Segmental Orifices in 4-in. and 6-in. Pipe Lines," S. R. Beitler and D. J. Masson; *Trans. ASME*, vol. 71, Oct. 1949, p. 751.

[33] "Experimental Study of the Effects of Orifice Plate Eccentricity on Flow Coefficients," R. W. Miller and O. Kneisel; *Trans. ASME, Journal of Basic Engineering*, vol. 91, Mar. 1, 1969, p. 121.

[34] Research Report RR 1790, "Pipe Bends as Flow Measuring Devices," J. A. Landstra and T. J. Vermeuten; Bataafse Internationale Petroleum Maatschappij, N. V.

[35] "Performance Characteristics of Elbow Flowmeters," J. W. Murdock, C. J. Foltz and C. Gregory; *Trans. ASME, Journal of Basic Engineering*, vol. 86–3, Sept. 1964, p. 498.

[36] "Improved Type of Flow Meter for Hydraulic Turbines," I. A. Winter; *Proceedings ASCE*, vol. 59, Apr. 1933, p. 565.

[37] "Gas Flow Meters for Measuring Small Rates of Flow," A. F. Benton; *Journal of Industrial and Engineering Chemistry*, July 1, 1919.

[38] "Liquid Metering with Capillary Flow Elements," C. E. Greeff; 19th Annual ISA Conference and Exhibit, Oct. 1964, Reprint No. 5.2-2.64.

[39] "Permeability of Glass Wall and Other Highly Porous Media," A. S. Iberal; *Journal of Research*, National Bureau of Standards, vol. 45, Nov. 1950, RP 2150, p. 398.

[40] "Recueil des Savants Etrangers, (Collection of Works of Foreign Scientists)," J. L. M. Poiseuille; pour 1842 Academie des Sciences.

[41] "Linear-Resistance Meters for Liquid Flow," R. C. Souers and R. C. Binder; *Trans. ASME*, vol. 74, July 1952, p. 837.

[42] "Study of Linear Risistance Flow Meters," F. W. Fleming and R. C. Binder; *Trans. ASME*, vol. 73, July 1951, p. 621.

[43] "Marks' Mechanical Engineers' Handbook," McGraw-Hill Book Co., New York, 6th ed., 1958, p. 3–73.

[44] "Formulation of Equations for Orifice Coefficients," H. S. Bean; ASME paper no. 70—WA/FM-2.

One very pressing cryogenic problem is that of cryogenic fluid flow measurement. An NBS programme, which focuses attention on the problem, has as its objectives to (1) establish present state-of-the-art by evaluating existing measurement methods, (2) establish methodology to maintain precision and accuracy of field-measurement devices, and (3) establish a comprehensive programme to develop new cryogenic fluid-measurement systems. The scope of this programme includes a precision measurement capability for measuring the flow of liquid nitrogen and liquid argon, a transfer of technology from the traditional cryogenic fluids to measuring the flow of liquefied natural gas and methane, and a concerted effort to develop new mass-flow measurements for cryogenic fluids such as slush or liquid hydrogen. Cryogenic flow-metering history is given as well as a description of three flow facilities that establish experimental confirmation of the cryogenic flow-measurement system under investigation.

Cryogenic flow-metering research at NBS

D. B. Mann

Precision flow measurement of cryogenic fluids encompass both commercial and technological requirements. Commercially distributing cryogenic fluids in less-than-truckload quantities makes it desirable to measure the total quantity delivered and billed to the consumer at the time of delivery; continual reweighing of the truck after each transaction – a common practice associated with large bulk shipments – is undesirable. Technologically, cryogenic fluids used as fuels and oxidizers in chemical reactions for rocket engines require very precise measurements based on fuel–oxidizer ratios. Flow-measurement uncertainty in this case results in penalties being assessed in the form of larger tanks, increased lift-off weight, and reduced payloads.

Problems associated with flow measurement of cryogenic fluids are neither new nor confined to one particular user group. They have been present in varying degree as long as cryogenic fluids have been available. This most probably dates back to the early 1900s when cryogenic fluids became commercially important. Cryogenic flow measurement was largely influenced by commercial exploitation of oxygen in the 20s, 30s, and 40s and more heavily influenced by the German work at Peenemunde prior to and during World War II. During this period and up to the present time, cryogenic flow measurement depended on experience gained and applying traditional flow-measuring methods. For example, applying water meters to cryogenic service has been extensive.

In the 1950s and early 1960s the national aerospace effort provided a large impetus to developing accurate and precise flow-measurement devices. The commercial field of cryogenics was expanding at a rapid rate and in combination with the impetus provided by the aerospace industry, flow-measurement problem areas continued to focus on requirements for more accurate flow measurements. A large effort for developing cryogenic flow-measurement devices resulted, and many excellent publications have resulted from this work.[1-15] However, these efforts were fragmentary; and it was becoming apparent that traditional metering was failing to give the precision and accuracy required. This does not infer that the traditional metering methods themselves were intrinsically limited, but was believed to result from the way measurement methods were applied to the measurement of cryogenic fluids.

The Cryogenics Division of the NBS Institute for Basic Standards has been involved in cryogenic fluid measurements since 1952.[2,3,5,12] During this time a number of attempts were made by the Division to co-ordinate and develop a programme on flow metering. Until recently, these efforts have not been sufficiently intensive.

During 1966–7 several events changed this situation. The remainder of this report summarizes the events leading to a flow-metering programme and describes our current efforts.

1. *The CGA proposed model code.* In June 1967 the Compressed Gas Association, a producer-oriented trade organization, proposed to the National Conference on Weights and Measures a model code for flow metering of cryogenic fluids. Although the Conference on Weights and Measures noted that cryogenic flow-measurement problems were, at that time, centralized only in a few states, the CGA proposal was a significant step in national recognition of cryogenic flow-measurement problems.

2. *The California code on cryogenic measuring devices.*[16] In the fall of 1967, the State of California Bureau of Weights and Measures began hearings on its own proposed code for cryogenic fluid-measurement devices used in California. The Cryogenics Division participated in

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these hearings. We provided technical information and at the request of the State of California and the Compressed Gas Association, provided standard data on the saturation densities of hydrogen, oxygen argon, and nitrogen gas for incorporation into the code. A separate document [17] references information on precision and accuracy expected in the code tabular data. After several hearings in the fall and winter of 1967–8, the code was adopted and made law in California on 1 June 1968. This code followed very closely the recommendations made in NBS Handbook 44.[18] Of the code specifications, one of the most significant was tolerance.

'The maintenance tolerance shall be four percent (4%) of the indicated delivery on underregistration and two percent (2%) of the indicated delivery on overregistration. The acceptance tolerance shall be one-half the maintenance tolerances.'

It was felt at the time, that, this honestly represented the capability of meters used in commerce.

3. *Instrument Society of America Ad-Hoc committee report on cryogenic flow measurement.* [19] An Ad-Hoc committee composed of industrial, government, and university people with a broad experience in cryogenics, returned a report to the ISA Conference in June 1967 that represented an extensive review and definition of the broad national needs for cryogenic measurements and standards. It also provided suggestions for the economic justification for meeting these national needs. The abstract of the final report by the Ad-Hoc committee recommended (a) a national standard and transfer standard, (b) an accepted methodology, (c) a national authority to develop the standards, transfer standards, and methodology, and (d) educating all personnel in the state-of-the-art of cryogenic fluid flow measurements. The report seemed to indicate that a massive programme was required.

In addition to the four recommendations, the Ad-Hoc committee report pointed out two separate and distinct applications of cryogenic flowmetering. The first is a moderate flow-rate totalized metering of commercial cryogenic fluids that might be encountered on trailer-truck type applications where a producer services a number of separate customers and bills them based on the meter reading. The other application is high flow-rate cryogenic metering of fuels and oxidizers at rates as high as hundreds of pounds per second as typified by the aerospace industry.

Since the Cryogenics Division was involved in all three of these events, we were asked for suggestions concerning what should be done. In the fall of 1967 we suggested that a flow research facility and programme be initiated having three aims.

1. To determine and evaluate traditional metering methods as applied to cryogenics by conducting a basic cryogenic flow-metering study. This portion of the programme would determine present status.

2. Provide the necessary methodology to preserve the accuracy and precision of field measuring methods.

3. Investigate new measuring methods.

Proposals for this programme were made in the fall 1967 when the Cryogenics Division was already involved in a NASA–Marshall Space Flight Centre sponsored programme (described later) to develop a mass flow meter that meters liquid hydrogen. It was believed that the major new effort should be aimed at accomplishing items (1) and (2), and that item (3) above could be pursued independently. The size and scope of the programme would be less ambitious than that indicated by the ISA Ad-Hoc committee report, but would concentrate on moderate flow rates and totalizing flow.

Liquid nitrogen flow research facility[20]

Problems of moderate flow-rate systems were approached jointly by NBS and the Compressed Gas Association (CGA) through a programme that would involve constructing a facility and developing a programme which would have maximum impact on their problems; these being ones of typical measurement situations. The Compressed Gas Association Incorporated is a non-profit membership corporation representing all segments of the compressed-gas industry in the US, Canada, and Mexico plus associates in other foreign countries. A joint co-ordinating committee was established involving staff members of NBS and participating members of the CGA. Meeting on an average of three to four times a year, this committee assessed progress and provided feedback on technical aspects of the programme and the meters selected for evaluation.

Initiating a programme and accomplishing the objectives required a facility for testing and evaluating meters. Such a facility must simulate the environment encountered in the field and provide precise and accurate measurement of flow over a broad range of pressures and temperatures.

Liquid nitrogen was chosen as the primary fluid of interest. As a cryogenic fluid, it is inert and convenient to handle, plus the fact that it was believed that most problems occurring in cryogenics would be encountered with liquid nitrogen flow. Liquid argon would also be used for confirmation. Table 1 lists the design capabilities of the facility. Figure 1 is a simplified flow schematic of the system, and Figure 2 is a photograph of the facility. Several features incorporated in this flow system are unique to cryogenics, but not necessarily to measurement science. Although it is a circulating system, a steady-state condition can be provided at both the meter test area and the gravimetric weigh system. Density and temperature of the liquid nitrogen process fluid may be adjusted through a broad range of conditions by adjusting the subcooler temperature and the flow through the bypass. Helium gas pressurization in the catch and weigh tank adjusts the total system pressure. Continuous operation over long time periods is assured by careful adjustment of the temperature conditioning in the subcooler and bypass where all heat leak from the system and pump energy is removed.

A commercially available sealed circulation pump, whose motor operates at ambient conditions, is enclosed in a pressurized case to eliminate the necessity for high pressure rotating seals. The pump is rated at 20–200 g p m

($0.00126–0.0126\ m^3\ s^{-1}$) at a head rise from 2·3 to 150 ft (0·69 to 45·7 m).

Figure 3 is a cross-section drawing of the catch and weigh tank – an arrangement that is unique to cryogenics. The weigh tank, while measuring rather large active weights of up to 400 kg, has a very low tare weight of the

Table 1. Flow research facility design objectives

Flow rate	$0.00126–0.0126\ m^3\ s^{-1}$
Pressure range	$0.0125–1.9\ MN\ m^{-2}$
Saturated liquid temperature range	64–114 K
Usable catch tank volume	$0.433\ m^3$
Usable weigh tank volume	$0.379\ m^3$
Load cell	4 448 N
Line size	0·076 m – liquid line 0·127 m – vacuum jacket

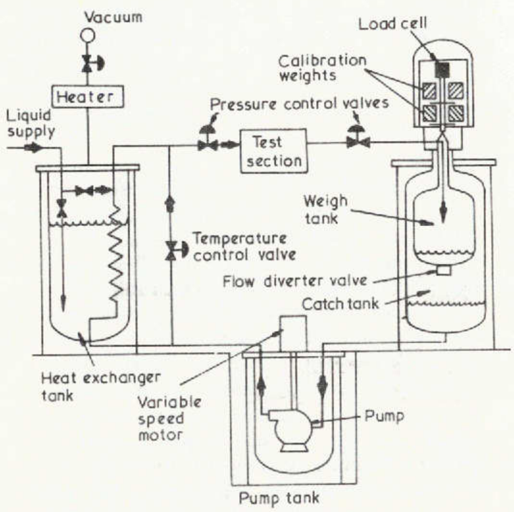


Figure 1. Schematic of flow loop

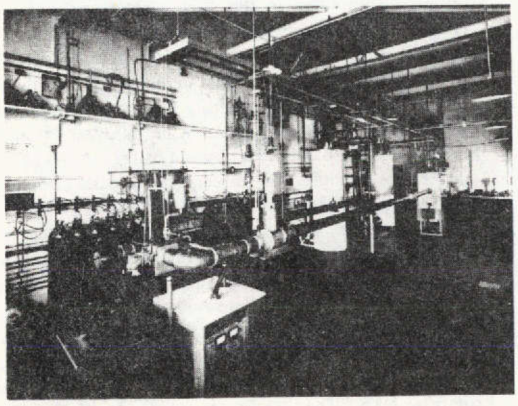
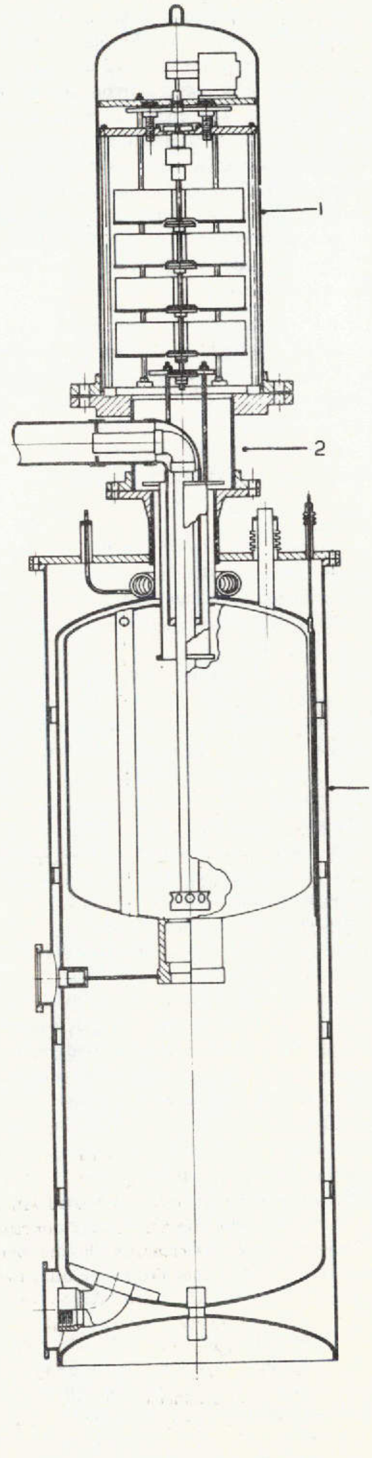


Figure 2. Photograph of flow facility



1 Weigh system 2 Liquid inlet
3 Weigh and catch tanks

Figure 3. Catch and weigh tank schematic

order of 70 kg. It hangs from the lower end of the load string and is supported by a stainless steel neck, flange, and three slender rods spaced at 120 degrees to allow the vacuum-insulated liquid inlet line to enter. The rods terminate at a plate forming part of the load-string and calibration system. Situated at the very top of the load string is the load cell. Four weights, each approximately 113 kg, are suspended on a mechanism which allows adding weights one at a time to the load string. Therefore, the original load-cell calibration can be checked under actual operating conditions.

Liquid accumulates in the weigh tank when the flow diverter valve located at the bottom of the tank is closed. Proper operation of this valve is critical to the measurement and must not leak liquid from the weigh tank during a weighing interval. To assure zero leakage, the valve design includes a double seal and a helium purge between the seals at a pressure that is always greater than the anticipated head accumulated in the weigh tank. Therefore, if one or both seals leak, pressurized helium passes from the seal to the weigh tank or catch tank and prevents any liquid from leaking between the weigh tank and the catch tank.

Table 2 contains a summary of the design objectives for flow-measurement accuracy and precision. These design objectives, it should be emphasized, are based on knowing the load-cell and readout-equipment performance, estimating corrections for buoyancy effects, and the uncertainties associated with the thermodynamic properties of nitrogen.

After completing the design of the facility, its performance had to be verified experimentally. Since no recognized dynamic-flow standard existed, verification was based first on the relationship between our weigh system (operating under static conditions) and the national standards; and second, on measurement methods (under dynamic situations) that are consistent with those methods recognized as valid within the field. To accomplish this we have accumulated over a period of a year, operational data on ten positive-displacement meters, involving four different types, and two turbine meters. Additionally, we related the operation of a single type of meter to two separate calibration systems.

A schematic block diagram, Figure 4, indicates the major components of the measurement system. Establishing a comparison between the standard masses for calibration and the national standards of mass is a rather straightforward problem. Also, the ability to provide and measure excitation voltage and load-cell output voltage is substantiated by relating them to the national standard for the volt. Substituting directly the weigh tank for the standard weights and calculating the errors associated with mass and voltage measurement provide a measure of our capability to weigh the liquid that accumulates in the weigh tank. Load-cell performance is monitored on a day-to-day basis by placing one or more standard weights on the load string and calculating the load-cell sensitivity. Making a pressure correction and keeping a control chart of the load-cell sensitivity will indicate any load-cell malfunction.

Operating the meters under test becomes the next step in evaluating system performance. Measuring meter performance involves counting the meter revolutions for a known quantity of liquid that passes through the system. Manipulating, in a variety of ways with the aid of a computer, the large quantity of data accumulated (Figure 5) provides the statistician with information about the accuracy and precision of this measurement system. The statistical treatment, devised jointly with the NBS Office of Measurement Services, employs a computer programme that identifies those

Table 2. Flow research facility design objectives – precision and accuracy [20]

Mass flow	%
3σ random error	0·08
Systematic error	0·1
Volume flow	
3σ random error	0·1
Systematic error	0·3

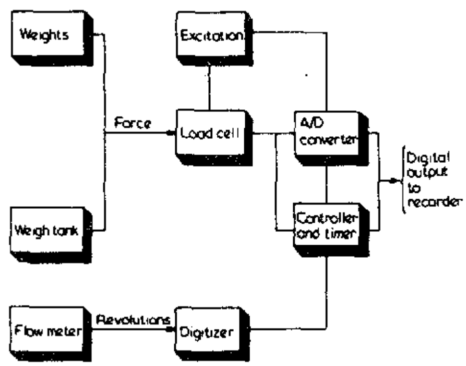


Figure 4. Block diagram of measurement system

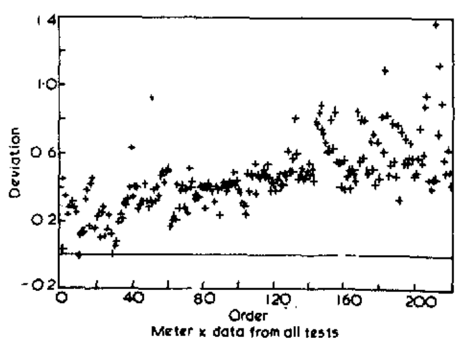


Figure 5. Typical meter performance data

important systematic data elements and reduces the data to its simplest form. If, after data reduction, the measurement system produces a random error of small proportions, we have secured the next step in proving the system capability.

The third step in proving the system requires that the performance relative to other calibration facilities be determined. This so-called 'round-robin' concept of proving the measurement capability involves sending a single meter or meters to several calibration facilities for evaluation. The capability of each calibration system, in making a measurement, then can be correlated.

NBS has begun such a programme. Two meters manufactured by a single producer have been calibrated on the producer's facility. They then sent them to NBS Boulder for evaluation. The two meters were then hand carried back to the producer facility by the two engineers operating the NBS flow-research facility. There the meters were recalibrated and the method of calibration observed. Data resulting from these series of tests provide information that rates the relative ability of the two calibration systems when making a mass-flow measurement. In the case cited, we found that the two calibration systems agreed to within 0·1%.

Proving a facility capability is a continuing operation that requires continued monitoring of the meter and the measurement-system interaction. Meters used were of a generic type (positive displacement) and ten meters representing four different principles were evaluated concurrently with the establishment of the facility capability. Rather than simple calibration, the programme as outlined required meter evaluation. Extensive tests, performed on at least two of each type of positive-displacement meter, included a rerun of the meter-supplier calibration; a performance test that adjusts temperature, pressure, and flow over a broad range; a stability test that runs the meter for 80 hours followed by rerunning the performance test and a period of overspeed operation. Evaluating meters based on this test programme is believed to be representative both of the positive-displacement type as well as the anticipated service conditions expected in the field.

Evaluating meters that operate on a positive-displacement principle is now complete and published results will appear in the near future. Our programme, planned for calendar year 1971, includes evaluating a number of other type flowmeters including turbine and head-type meters.

Liquefied natural gas (LNG) and methane flow-metering

Another programme, in which the Cryogenics Division also is heavily involved, concerns measurements of liquefied natural gas and methane. Sponsored by the American Gas Association, Incorporated and the Pipeline Research Committee of the American Gas Association, this programme is directed at accumulating data and techniques that are useful in this rapidly growing field. Although, it is believed that much of this information already has been established by those who work with cryogenic fluids, the LNG programme at NBS is directed to disseminate this information and to develop, when necessary, new information and techniques.

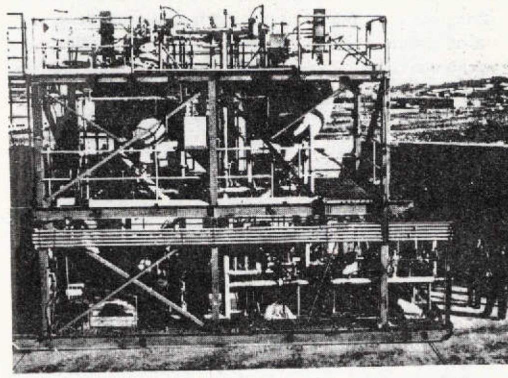


Figure 6. Photograph of LNG/methane test facility

Part of the LNG/methane effort concerns the metrology of flow-metering liquefied natural gas and methane. Under this programme we have secured, from the US Air Force, a flow facility suitable for liquefied natural gas. Located at an outdoor test area, the facility shown in Figure 6 is soon to be put into operation. All control and instrumentation are remotely located in a control room on the far side of the protective-barrier wall. The facility consists of two 300 gallon (1350 l) cryogenic vessels connected by vacuum-insulated piping. Dewar-to-dewar piping is 3 in (76 mm) diameter and the test loop, which includes a circulating pump, is 2 in (51 mm) diameter. Flow rates from 20 to 150 g p m or higher are possible with this system.

Direct density measurements, liquid-level experiments, and evaluation of flow-metering devices operating on LNG or methane are among the experiments proposed for this facility. The facility includes a gravimetric weigh system having less desirable features than the systems previously described for liquid nitrogen. Assessing the capability of this gravimetric weigh system to make a mass-flow measurement is achieved by interchanging meters with those at the liquid nitrogen flow-research facility.

The problems confronting the LNG industry are similar in many respects to those that represent the more traditional cryogenic field. Flow rates, both volumetric and mass measurement, are desirable if not essential at the time that the LNG commodity transfers ownership.

Many contracts involving international as well as intranational transfer of ownership involve total energy content of the LNG. Accurately measuring the total mass flow, mass flow rates, and heating value are basic to such contract obligations. The over-all measurement system requires a high degree of accuracy and precision.

Flow rates vary widely from 0·00126 to 0·0126 $m^3\ s^{-1}$ for a small commercial transfer operation to as high as

1·26 m^3 s^{-1} when unloading a ship. It is the objective of the flow portion of our programme to evaluate and prove flow-measurement methods at the low end of the requirement and at the same time provide useful information for metering higher flow rates.

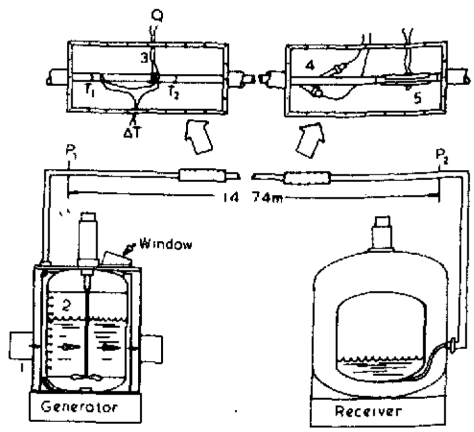
Direct density measurements also are important in computing mass flow from volume-flow measurements and are equally important in determining the total mass of a storage vessel. Measuring liquefied natural gas density constitutes probably the most important difference between LNG and the traditional cryogenic fluids. These traditional cryogenic fluids are generally of high purity. Over the years large quantities of physical property data have been accumulated for these pure fluids and the relationship between pressure, density, and temperature have been studied. Therefore, the density of traditional cryogenic fluids can be inferred to a degree from a measurement of pressure and temperature.

This is not the case, however, with liquefied natural gas which is a mixture of components that have various and different boiling-point temperatures. Presently no adequate method exists to predict the density of these fluid mixtures based on the pressure, temperature, and constituent fraction. Research that involves measuring the mixture properties as well as pure components should eventually lead to a satisfactory inferred density measurement. In the interim, much effort and interest is being expressed to find methods of measuring the density directly in the vessel or pipeline. Direct density measurement methods that have been successfully applied to traditional cryogenic fluids will be investigated for liquefied natural gas. These methods include capacitance, nuclear-radiation attenuation, hydrostatic head, etc. Some of these methods are now being used with LNG. The purpose associated with this portion of the LNG metrology programme is to evaluate these methods and to provide new measurement methods if possible.

Advanced technology

The advanced technology portion of the flow-metering programme is sponsored by NASA–Marshall Space Flight Center and is directed at developing and evaluating new measurement techniques and methods. Although existing types of cryogenic-measurement devices are commonly used, their performance for certain aerospace applications has not been adequate. Guidelines for developing and applying this metering instrumentation specify that the fluid is liquid or slush hydrogen and that it is desirable that flow measurements be made in direct mass-flow rate or by inferring mass-flow rate from measuring a volume flow and density. Additional guidelines specify that no active measurement element be located in the flow stream, that rangeability exceed ten to one, and that candidate systems have the capability to provide an over-all uncertainty of less than 1%. This last guideline is based on an estimate of state-of-the-art measurements of cryogenic fluids employing existing available concepts.

As our approach to the problem, we select those phenomena or techniques that have recently emerged out of research laboratories. They should be applicable to the general problems of cryogenic flow measurement and specifically to the programme as outlined by NASA. Next, a feasibility model is developed and evaluated using a third in-house cryogenic-flow facility. This facility originally was designed and constructed under a previous NASA programme on generation, storage, and flow characterization of slush hydrogen.[22] This slush/liquid hydrogen flow system is shown schematically in Figure 7. Liquid moves from one container to the other by pressure transfer through a 16 mm transfer line and test section. Volume-flow rate is measured by a liquid-level device in the generator. Fluid density is measured directly by nuclear-radiation attenuation or capacitance methods. Total-flow uncertainty is probably no better than 1–2%, but nonetheless is adequate for demonstrating the feasibility of new concepts.



1 Gamma-ray densitometer
2 Liquid-level indicator
3 Boundary layer thermal meter
4 Microwave coupler
5 Capacitance densitometer

Figure 7. Slush liquid hydrogen flow system

The current programme objective is to develop a feasibility model utilizing both our NBS in-house capability and the capability of those subcontractors who have demonstrated expertise in their particular specialty.

Investigations on a number of systems are being conducted simultaneously. These investigations include nuclear-magnetic resonance, microwave Doppler,[21] laser Doppler, and a system involving momentum change. Feasibility test results of these measurement systems will be studied and the two or three most promising will be developed and evaluated as prototype systems.

Summary

The flow-measurement's programme conducted by the Cryogenics Division of NBS–IBS can only have a limited effect on those problems directly associated with cryogenic flow measurement. The number of fluids involved, the pressure, temperature, flow rate, or totalizing flow all require specific solutions and experimental data. The fact that the programme focuses the interest of a large number of different groups on the general problems of cryogenic flow-metering is, we feel, of equal or greater importance. These groups include the major industrial producers of cryogens, the state regulatory agencies, meter manufacturers, and others having special measurement requirements. By establishing a precise and accurate flow-measurement system, we are beginning to demonstrate the intercomparison of measurement capability between this facility and other calibration facilities. We anticipate that this programme will extend into other areas of research, development, and evaluation of cryogenic flow measurement.

REFERENCES

1. CLOSE, D. L. *Cryogenics and Industrial Gases* 8, 19 (1969)
2. RICHARDS, R. J., and JACOBS, R. B. Advances in Cryogenic Engineering, Vol 4, 272 (Plenum, NY, 1960)
3. PURCELL, J. R., SCHMIDT, A. F., and JACOBS, R. B. Advances in Cryogenic Engineering, Vol 5, 282 (Plenum, NY, 1960)
4. TANTAM, D. H., and ALLEN, A. Advances in Cryogenic Engineering, Vol 5, 299 (Plenum, NY, 1960)
5. BURGESON, D. A. Advances in Cryogenic Engineering, Vol 5, 307 (Plenum, NY, 1960)
6. MINKIM, H. L., HOBART, H. F., and WARSHAWSKY, I. Advances in Cryogenic Engineering, Vol 7, 189 (Plenum, NY, 1962)
7. MORTENSON, L. N. Advances in Cryogenic Engineering, Vol 6, 379 (Plenum, NY, 1962)
8. BUCKNELL, R. L. Advances in Cryogenic Engineering, Vol 8, 360 (Plenum, NY, 1963)
9. DEPPE, G. R., and DOW, R. H. Advances in Cryogenic Engineering, Vol 8, 371 (Plenum, NY, 1963)
10. GALLEY, R. L. Inst Soc Amer, 17th Annual Conf, Paper 16.2.62 (1962)
11. GREY, J. Inst Soc Amer, 14th Annual Conf, Preprint 111-59 (1959)
12. ALSPACH, W. J., and FLYNN, T. M. Advances in Cryogenic Engineering, Vol 10, 246 (Plenum, NY, 1965)
13. FRESLOUR, J. W. *Instrument Control Systems* 38, 97 (1965)
14. ANDERSON, B. M. Instrument Process Ind Proc 19th Symp, 43 (1964)
15. BELL, J. H. *Cryogenic Technology* 1, 115 (1965)
16. ALFANO, J. Inst Soc Amer Conf No 69 586 (1969)
17. RODER, H. M., McCARTY, R. D., and JOHNSON, V. J. NBS Tech Note 361 (US Govern Print Office, 1968)
18. NBS Handbook 44, 3rd edn (US Govern Print Office, 1965)
19. Inst Soc of America, Ad Hoc Committee Report on Cryogenic Flow Measurement (1967)
20. DEAN, J. W., BRENNAN, J. A., and MANN, D. B. Advances in Cryogenic Engineering, Vol 14, 299 (Plenum, NY, 1969)
21. ELLERBRUCH, D. A. Cryogenic Engineering Conference, Boulder, Colo, Paper M-1 (1970)
22. SINDT, C. *Cryogenics* 10, 372 (1970)

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