Main Index

North Sea
FLOW
Measurement Workshop
1995

Paper 24:

TURBINE METERS FOR VERY HIGH PRESSURE -A NEW LOOK AT AN OLD CONCEPT

5·6

Authors:
H. H. Dijstelbergen, Instromet Intern., Belgium
J.T.M.Bergervoet,Instromet, The Netherlands
H.Bellinga, N.V. Nederlandse Gasunie

Organiser:
Norwegian Society of Chartered Engineers
Norwegian Society for Oil and Gas Measurement
Co-organiser:
National Engineering Laboratory, UK

TURBINE METERS FOR VERY HIGH PRESSURE - A NEW LOOK AT AN OLD CONCEPT

Harry H. Dijstelbergen
Vice-president Engineering and Manufacturing
Instromet International
Essen, Belgium

Jos T.M. Bergervoet
Project Engineer
Instromet B.V.
Silvolde, Netherlands

Henk Bellinga
Senior Research Engineer
N.V. Nederlandse Gasunie
(retired)

SUMMARY
Turbine meter design was originally limited by the requirement for calibration with atmospheric air. These limitations on the design are shown to be in conflict with performance at high pressure operating conditions. Calibration facilities operating at high pressure are now available and eliminate the need to design a meter that also passes low pressure calibration tests. The improvements by designing for high pressure only are highlighted and illustrated with test results.

The recent developments to eliminate installation effects are explained and illustrated with data on the results obtained.

Several systems to monitor performance of metering installations have been tried, and fast powerful diagnostics revealing failure or degradation of components are available.

AN OLD CONCEPT.

Turbine meters have been used for the measurement of fuel gas for nearly a century. They were based on earlier anemometers used for measuring wind strength and air currents in mines (Ref. /1/). The main difference is that whereas anemometers measure velocity at a point, turbine meters cover the entire duct.

Bonner and Lee (Ref. /2/) mention a patent applied for in 1901 by one Thomas Thorpe of Whilefield near Manchester. From following patents it is quite clear that the fuel gas industry was the target market. Gas was an expensive commodity at that time, mostly used for lighting. It was manufactured from coal, and the nature of that process was such that the gas was at low pressure.

Up to the sixties turbine meters were already manufactured in all sizes of which the largest reported by Bonner and Lee is one, measuring 1 m in diameter, used to measure coke oven gas at Dutch State Mines.

Most applications were at low pressure and one of the challenges for the meter designer was to lower the minimum flow rate that could be measured as much as possible by using very light rotors and bearings.

For turbine meters individual calibration was a necessity for any degree of acceptance. Meters capable of accurately determining gas quantities were available even in the early days. Very large "wet" positive displacement meters had accuracies that would even today be very acceptable.

The calibration was of course at low pressure, the application being at low pressure, and also because no alternative calibration methods were available. Thus the calibration with atmospheric air found its way into codes and regulations, and still figures today in OIML recommendations and most National standards and regulations.

According to these standards the meter has to show an error between certain limits when tested with air of atmospheric pressure. This requirement has to be met regardless of whether they will ever serve under those conditions.

In the 60's Lee, Evans and Karlby at Rockwell (Ref. /3/, Ref. /4/) did fundamental research on turbine meters for liquid and gas. This work resulted in the present gas turbine meter, capable of stable, reliable and accurate measurement over long periods of time as it is presently used. It was the availability of a high pressure test installation that made it possible to attain the superior performance of these meters.

Though of U.S pedigree, the application of turbine meters for high pressure gas metering occurred on a much larger scale in Europe than in the U.S. All the gas from the Slochteren field sold in the Netherlands to date has at least once been measured by a turbine meter.

HIGH PRESSURE CALIBRATION.

As mentioned above, it was really the availability of the Rockwell high pressure test loop at Dubois that made it possible to check the theories of Evans, Lee and Karlby and to design meters on this basis.

Of course, field tests had been made at elevated pressures both in the U.S. and in Europe, often with orifice plates as the reference. Repeatability of the tests and traceability of the reference are of a limited accuracy under those conditions and not suitable to improve performance significantly. Only with a dedicated installation equipped with traceable accurate reference standards, can major improvements be made.

In Europe, Gaz de France was probably the first to systematically investigate the performance of turbine meters at high pressure in their dedicated research facility in Alfortville (Ref. /5/). Their tests showed a significant influence of the pressure on the error of some of the meters they tested.

Gasunie, to reduce the uncertainty in their measurements, built three test facilities for high pressure calibration, and were the first to test each meter they had in use individually at operating conditions (Ref. /6/).

Other research and testing facilities for high pressure gas were built in Europe and intercomparisons were made to assess the accuracy of their results (Ref. /7/, Ref. /8/).

Calibration of meters can now be carried out at operating conditions for most applications, and the reasons for a calibration at low pressure have disappeared.

Tradition, however, dictates that in many countries it is still required that any meter shall be approved with atmospheric air.

PHYSICS OF THE TURBINE METER

The ideal turbine meter would have no retarding forces, infinitely thin rotor blades, total driving force concentrated at the mean blade radius and a uniform fluid velocity distribution entering the blades in an axial direction.

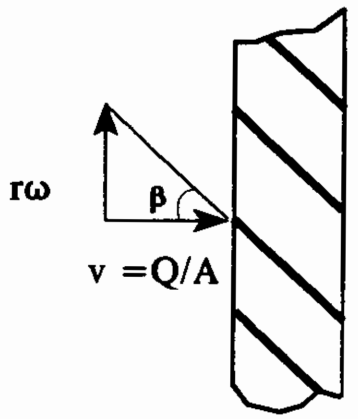

Figure 1. Velocity diagram for rotor radius (r) ideal case

From the blading diagram (Figure 1) it can be seen that for the ideal turbine meter, the rotational speed of the rotor would be:

1) $\omega_i = \tan \beta . Q / r . A$, where:

r	=	the mean radius of the rotor
A	=	the annular flow are
β	=	the blade angle
Q	=	the volume flow rate
ω_i	=	the (ideal) rotational speed of the rotor

and

V	=	Q/A

V	=	the velocity of the gas

for a given meter.

Equation 1) simply states that the rotor speed is directly proportional to the flow rate.

Thus by counting the number of revolutions of the rotor and scaling them for their apparent volume, the volume that has passed through the meter can be totalized. This characteristic is similar to a positive displacement meter.

In the real turbine meter there is a drag due to the mechanical friction of the bearings and gearing, as well as fluid drag on the blades and the hub.

The ratio ω/ω_i would indicate the per cent registration of the actual meter to the ideal meter. This per cent registration can be equated as the ratio of the driving forces to the retarding forces (Ref. /3/).

2) $\omega/\omega_i = PR = 1 - (M_R / M_d)$, where:

PR	=	Per cent registration
M_R	=	Total retarding torque
M_d	=	Available driving torque

The percentage registration is related to the error as

$$E = (\omega - \omega_i)/\omega_i = PR - 1$$

The available driving torque M_d is proportional to the kinetic energy of the fluid or:

3) $M_d = k.\rho.Q^2$

M_d = driving torque
ρ = fluid density
Q = volume flow rate
k = constant
and the retarding torque M_R :

4) $M_R = M_f + M_n$, where

M_f = retarding torque due to fluid forces
M_n = retarding torque due to mechanical forces.

Substituting in equation (2) yields:

5) $PR = 1 - [\, K.(M_f + M_n) / \rho.Q^2 \,]$, with

K = $1/k$

If the retarding torques can be regarded as constants then this equation simply states that for the meter to achieve its required accuracy, the retarding forces should represent only a fraction of the available driving torque.

Since the driving torque is directly proportional to the fluid density, and the density of gas is very small at low pressure, the retarding torques in the meter must be kept as small as possible for good low pressure performance. Mechanical friction forces should also be kept small in relation to the driving forces because they are less stable.

It should be noted that the energy extracted from the fluid is the amount required to overcome the retarding torque, and at higher densities, the available energy is far greater than required. Further, the retarding torque must be small, or proportional to the driving force, to have linear relationship for the accuracy of the meter.

It is usual to equate hydrodynamic forces in terms of dimensionless friction factors or drag coefficients:

6) $M_f = C.\rho.Q^2/2$

Lee and Karlby (Ref. /3/) separated the fluid drag in two parts, one (C_f), that is dependent on Reynolds' number and one that is not (C_s).

7) $C = C_f + C_s$

The Reynolds' independent part results in a constant percentage error.

8) $PR = 1 - K.C_s/2 - K.(C_f/2) - K.M_n / (\rho.Q^2) = A - f(Re) - K.M_n / (\rho.Q^2)$ with

A a constant.

The Reynolds' number dependent part behaves as the classical friction factor (Figure 2) and the form is reflected in the curve of the percentage registration PR as a function of Reynolds' number (Figure 3).

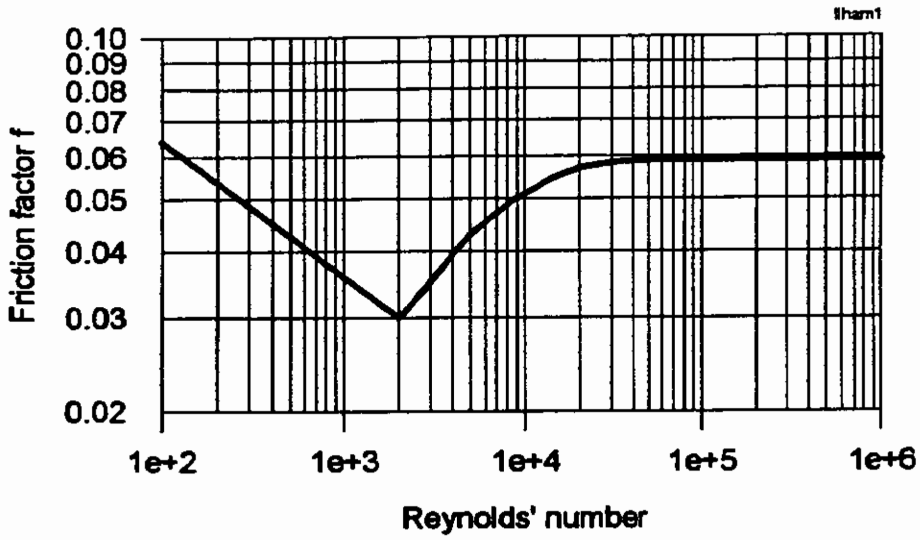

Figure 2. Friction factor f as a function of Re

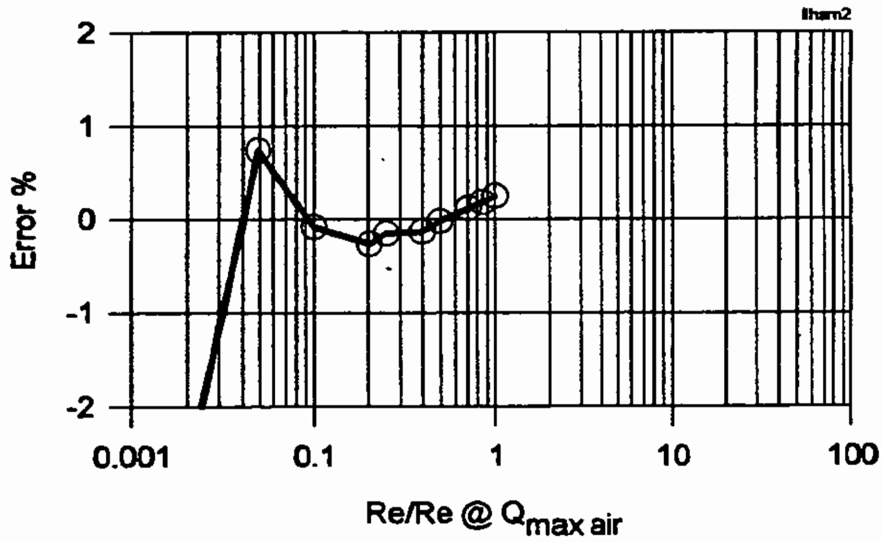

Figure 3. Calibration curve for original meter for atmospheric air

The turbine meter *minimum flow rate* is determined by the mechanical friction M_n. A series of tests with air at low pressure usually establishes the flow rate at which the meter achieves acceptable registration. The minimum flow rate for any other set of conditions can then be determined by equating the new conditions to the minimum conditions determined by test. Since the driving torque must be equal for both conditions, it can be expressed as follows:

9) $$M_n = [\,\rho_{air\ base}\, Q^2_{min\ air}] = [\rho_m\, Q^2_{min\ operating}]$$

where $\rho_{air\ base}$ is the density of air at base conditions at which the minimum flow test was carried out, and ρ_m the density of the measured gas at operating conditions.

From this we find for the minimum flow rate under operating conditions:

10) $$Q_{min\ operating} = Q_{min\ air} \sqrt{\left(\frac{\rho_{air\ base}}{\rho_m}\right)}$$

From figure 3 it is clear that for low Reynolds' numbers the meter linearity suffers. In order to keep the calibration curve within the legal tolerances, it may for some meters be necessary to lower the Reynolds' number at which the flow becomes turbulent. Several ways exist to achieve this. Eujen (Ref. /9/) gives a description of a meter where the gas expands from a ring shaped nozzle onto the turbine wheel. In this design, the driving force now also becomes dependent on the Reynolds' number, and the form of the equations governing the meter behaviour also becomes more complicated. In general one can say that means to force the transition from linear to turbulent to lower Reynolds' number, complicate the design and also the behaviour of the meter. It introduces more parameters that tend to make meter behaviour less predictable.

Axial forces

Though the design of the meter is such that axial forces should in first approximation be compensated, it is clear that any remaining axial force will increase, proportional to the velocity head. These forces will therefore increase linearly with pressure. Axial forces on the turbine wheel may result in slight displacements or deformations that affect accuracy. These effects may not be noticeable at low pressure.

The above shows that the requirement for a calibration curve that is inside the legal limits for the low Reynolds' numbers of atmospheric air, determines the design of a turbine meter to a large extent.

The behaviour of present generation turbine meters is excellent as is shown in a paper by van der Kam (Ref. /6/). Operating experience at Gasunie showed that the few cases when problems arose, could be attributed to one or more of the following causes:

Bearing problems

Contamination problems

Bearing problems may result from contamination or from overloading or just wear. The result is an uncertainty of the position of the rotor which may affect the accuracy. For high pressure use, an increase in bearing friction would be insignificant in most cases, as sufficient driving power is available.

Contamination may take the form of deposits in the flow channel or on the blades. Solid particles may be jammed between rotor and flow channel, damaging the rotor and/or its bearings.

With this experience a research project was started in co-operation with Gasunie to design a meter that was less vulnerable to contamination, and with a stronger bearing arrangement, in short, the design of a High Stability Turbine meter.

Project design

The project was designed to investigate the following parameters:

The effect of removing the mechanical counter.

The influence of bearing size on minimum flow rate.

The effect of removing any rims or recesses that are presently used to improve low pressure calibration.

The clearance between rotor and housing and its influence on the calibration curve.

Experiments were carried out at Instromet B.V. in Silvolde with air of atmospheric pressure, at Instromet's 8 bar test facility in Utrecht, at the Gasunie test station in Groningen at pressures of 8 and 20 bar and at Gasunie's Bernouilli Laboratory in Westerbork at 60 bar.

Two sizes were used for the experiment. Initially a 12" meter was used to determine optimum design values. This was checked later with a 6" meter to determine whether the scaling rules would apply.

Both meters were first designed to be able to pass low pressure calibration. The calibration curves obtained with this design served as reference.

The effect of the mechanical counter

The mechanical counter is driven by the turbine wheel through a reduction gearing. A magnetic coupling provides a leak-tight feed from the pressurised part to the outside. In its simplicity, it provides a reliable back-up for the sophisticated data collection and conversion that is used in modern gas metering. Its insensitivity to electromagnetic interference and even lightning, and its independence of electricity supply or batteries are valued specifically by customers in remote and hostile environments. It does, however, exert a drag.

Electronic pick-ups need negligible amounts of energy and it was sensible to check whether it would be advantageous to replace the mechanical counter by an electronic one. The small drag the mechanical counter causes, would perhaps better be spent in heavier bearings.

Eliminating the mechanical counter, however, did not result in any measurable change in the calibration curve, even with air of atmospheric pressure.

Bearing size

Heavier bearings will have the effect to increase the retarding torque and therefore increase the minimum flow rate. The magnitude of the effect was experimentally determined.

With atmospheric air the calibration curve dropped about 1 % at 10% of maximum flow rate. However, for gas at 8 bar the curve dropped only 0.5% at 5% of Q_{max}.

This proved that the envisaged bearings would be suitable for use at transmission pressures.

Eliminating recesses in the flow channel

As was explained earlier, turbulence is deliberately introduced to move the transition point from laminar to turbulent to lower Reynolds' numbers. If this is not done, the meter will overregister at these low Reynolds' numbers. For small size meters the effect of the laminar flow is wholly or partly compensated by the drag from bearings. For larger size meters the transition point to turbulent flow has to be lowered to have a sufficient linear range.

Figure 4 gives the calibration curves at different pressures for the meter of figure 3, but with heavy bearings. In figure 3, the effect of the laminar flow regime is clearly visible. Heavier bearings, as in figure 4a, clearly compensate the effect. Eliminating the recesses that induce early turbulence gives indeed a dramatic change in the calibration curve (Fig. 4b).

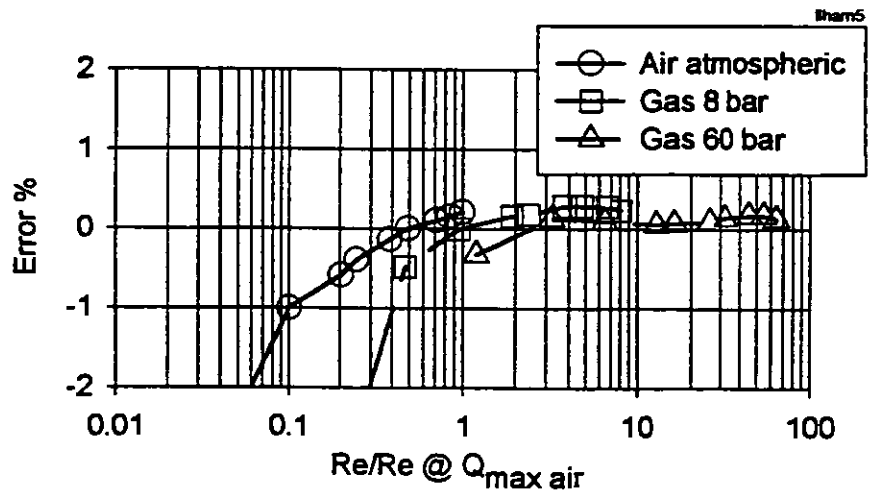

Figure 4a. Calibration curve for the same meter as of figure 3 with heavier bearings

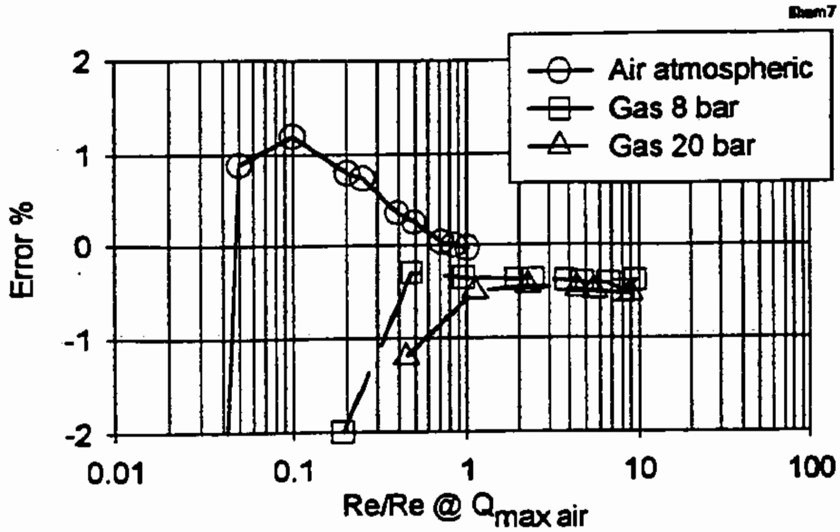

Figure 4b. The same meter as in figure 4a but now without any recesses in the flow duct

The laminar region now peaks for a value of the normalised Reynolds' number of 0.1 instead of 0.05 as in figure 3. The Reynolds' number is normalised with respect to the value of Re with air of atmospheric pressure at maximum flow rate. The clearance between rotor and flow channel for the meter of figure 4b is very small.

Plotting measured data as a function of Reynolds' number, a single continuous curve is found if the friction at the lower flow rates, where it becomes important, is compensated for. This is illustrated in figures 5 and 6, where first the raw data are plotted as a function of the Reynolds' number and then a correction is mathematically made to the data to compensate for a certain mechanical drag. If the right value for the mechanical drag is introduced, a single smooth curve is indeed obtained. Of course, any systematic difference in the different calibration laboratories would still remain.

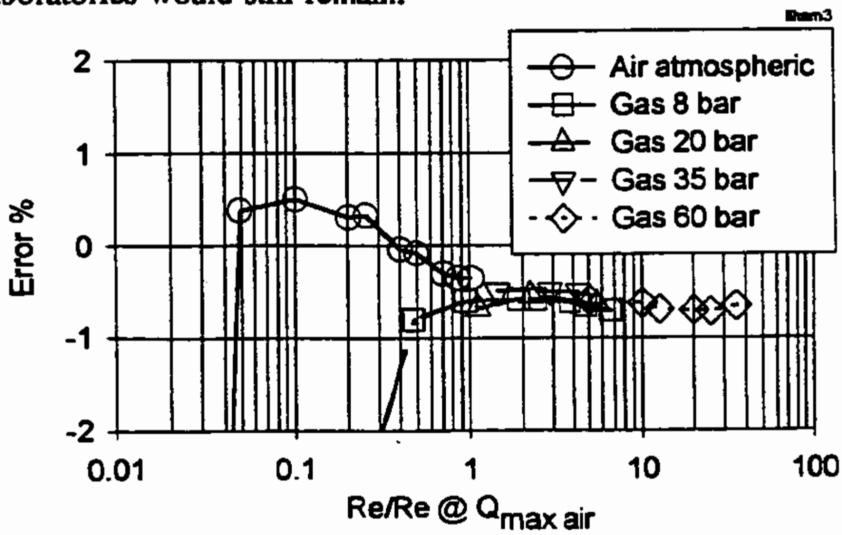

Figure 5. Calibration curve for meter with heavy bearings and increased clearance measured at different pressures

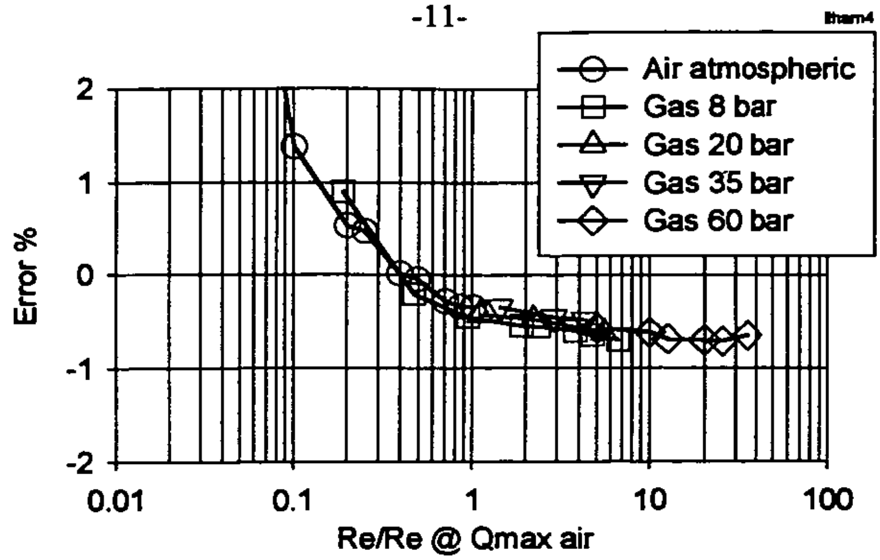

Figure 6. The same data as in figure 5, but now compensated for mechanical friction.

A problem that is difficult to prevent in practice is the presence of particles in the conduit. Welding beads may still be present in nominally clean piping, and start moving when flow rates are high. It was therefore decided to increase the space between rotor and housing to allow a passageway for these types of particles. For example in the configuration of figure 5 and 6 the clearance between rotor and flow channel has been somewhat increased with respect to figure 4b.

A number of different clearances were tried. It was found that the clearance could indeed be increased considerably without significantly affecting the performance of the meter (Figure 7). The form of the calibration curve remains the same but drops, as the blockage and therefore the velocity is reduced. For very large clearances the repeatability of the data becomes less.

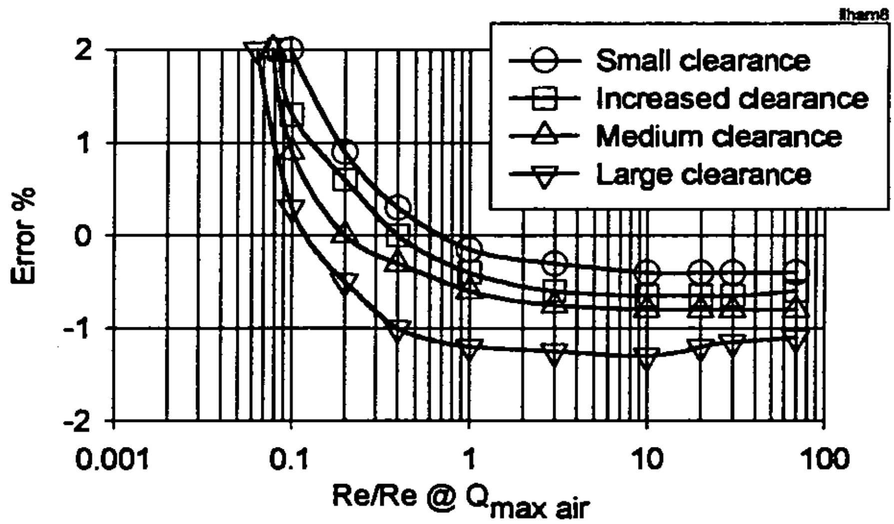

Figure 7. Calibration curves for different clearances between rotor and flow duct

In conventional meters, similarity of performance with Reynolds' number is limited to the fluid related variables: flow, density and viscosity. The geometry is too complicated to scale.

A 6" meter was also equipped with heavier bearings and recesses in the flow duct were removed. Similar experiments were then performed and it now proved possible to also scale the linear dimension (Figure 8). This illustrates that the performance for meters of this design can be better controlled, thus adding another tool for quality control. It is not sufficient now for these meters to have a calibration curve that falls within the limits set by OIML or legal metrology. The curves of these meters have to match a specific pattern as a function of Reynolds' number.

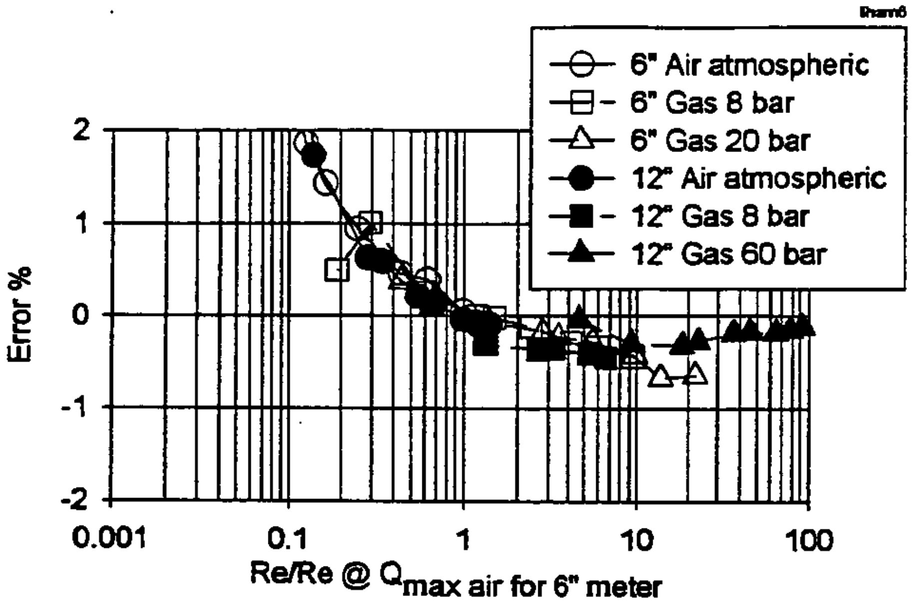

Figure 8. Calibration curves for 12" and 6" as function of Re (Data compensated for mechanical friction)

X4X STRAIGHTENER

Parallel to the High Stability project, research had been carried out to design a straightener that could be built into the same standard 3D length of the SM-RI meter and would satisfy the requirements that are set out in ISO 9951 (Ref. /10/). The standard requires the manufacturer to specify the installation conditions necessary for a specific set of flow perturbations to have not more than 0.33 % influence on the calibration curve.

The low level perturbation as specified in ISO 9951 consists of two elbows in two perpendicular planes, a concentric expander to the next pipe size and a 2D straight pipe. The high level perturbation is similar , except that a half area plate is mounted in between the two elbows with the opening towards the outside radius of the first bend. The perturbators are drawn in figure 9. They can be arranged to give a clockwise or anti-clockwise swirl.

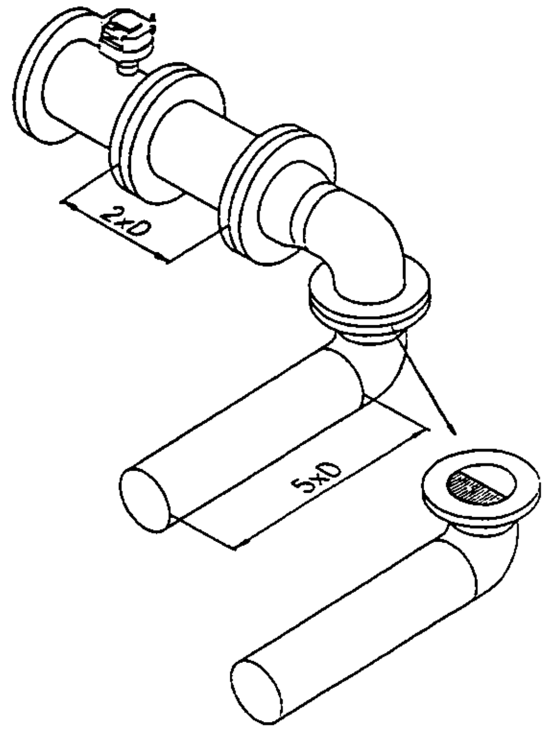

Figure 9. Flow perturbations according to ISO 9951

The research resulted in the development of the X4X straightener. This straightener, built into a normal 3D length turbine meter, satisfies the requirements of the standard without additional length or external straightening devices. Figure 10 shows the measured influence of the high level perturbations on a 6" meter.

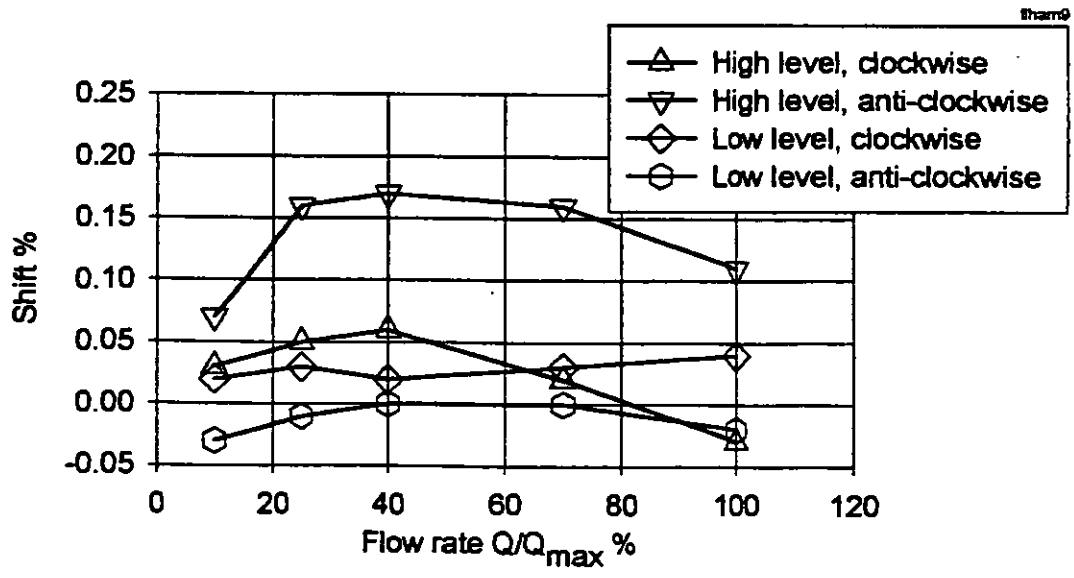

Figure 10. Shift in calibration curve for a 6" turbine meter equipped with an X4X straightener when tested with ISO 9951 perturbations using atmospheric air.

It was logical to incorporate this straightener in the new High Stability design as it makes it possible to greatly reduces the size of installations. It proved to be possible to incorporate this straightener without affecting the calibration curves. At the time of writing, a 16" meter with X4X straightener and High Stability design is undergoing an endurance test.

DIAGNOSTICS

The increased importance that is attached to accurate and reliable metering has led to the construction of metering stations with two meters in series. In general, the indication of the two meters will not be exactly the same. The difference between the meters should however be small and constant. In case one meter fails, the difference between the two meter indications will change. The risk of both meters failing to the same extent at the same moment is very small. Automatic detection systems on this basis can not only detect any failure or drift at an early moment, they can also prevent unnecessary scheduled maintenance.

Different philosophies exist with regards to the choice of the meter types. Distrigaz (Ref. /11/) has considerable experience in systems with two turbine meters. This system has proved not only to be very accurate but also very fast. Any deviation in a meter of more than 0.15 % is detected within one hour. Practice has shown that even fouling, which one could expect to have the same effect on both meters, does not happen at the same time at the same rate, and will be detected.

In a metering station with several runs in parallel this leads to a system with a number of identical units that can all be intercompared and, if desired, interchanged.

In Germany, the requirement is for two meters of different operating principle. Both vortex meters and ultrasonic meters are used. Though the use of two different measuring principles gives even more certainty that the meters do not change at the same time, it may also create some problems. Range and repeatability need not be the same, and dynamic response of the meters is different. In general this would lead to more complicated decision strategies for error detection and/or a slower or less accurate diagnosis. Installations with a vortex meter as back-up become rather long, as the latter requires a considerable straight length. Combination of a turbine meter and the Instromet Q-Sonic would technically not require such long straight lengths, as the Q-Sonic is quite insensitive to perturbations in the velocity profile and can be installed in close proximity behind a turbine meter.

There are a number of reasons why a diagnostic system as above can be less attractive. Space may be a constraint, or the cost of a full back-up system considered too high for the station. In such cases a diagnostic system that in itself does not provide full metering capability can be chosen.

An early example of such a system, relying on turbine meter technology is the Equimeter Auto-Adjust (Ref. /12/). In this meter two turbine wheels are mounted, of which the second one has a very small blade angle and is subjected to the wake of the first. Under normal conditions it would turn very slowly and at a constant fraction of the speed of the first. Most malfunctions of the first turbine wheel will affect the swirl in its wake and show up in the ratio of the rotational speed of the two wheels. In fact, for a number of failure modes, the difference of the two rotor speeds is still an accurate measure of the flow rate.

Alternatively other measuring principles could be combined with a turbine meter and made into one instrument. Several possibilities are investigated by Instromet at this moment. It will be clear that one of these is the ultrasonic principle.

Such instrument would reliably detect any drift or malfunction but not necessarily have self-correcting capabilities.

ACKNOWLEDGEMENT
The co-operation of NV Nederlandse Gasunie and in particular of Dr. P.M.A. van der Kam and Mr. de Nobel is gratefully acknowledged.

REFERENCES

1. Ower E., Pankhurst R.C., The measurement of air flow, Pergamom Press, Oxford, 1977,ISBN 0-08-021282.

2. Bonner L.A., Lee W.F.Z., The history of the turbine meter, Paper presented at the 1992 A.G.A. Distribution/Transmission Conference, Kansas City Missouri, May 3-6, 1992.

3. Lee W.F.Z., Karlby H., A study of viscosity and its compensation on turbine flow meters, Trans. ASME, Journal of basic engineering, Vol.82, 1960, pp. 717-728.

4. Lee W.F.Z., Evans H.J., Density effect and Reynolds' number effect on gas turbine flowmeters, Trans. ASME, Journal of basic engineering, Vol.87, 1965 ,pp.1043-1057.

5. Castillon M.P., Calibration of gas meters with sonic nozzles, Symposium on flow, its measurement and control in science and industry, paper 1-3-22, Pittsburgh, U.S.A., May 10-14, 1971.

6. de Jong S., v.d. Kam P.M.A., High pressure recalibration of turbine meters, in Flow Measurement, Proceedings of Flomeko '93 Seoul, Korea, edited by S.D. Park and F.C. Kinghorn. ISBN 898-454-0180-6.

7. Spencer E.A., Eujen E., Dijstelbergen H.H., Peignelin G., Intercomparison campaign on high pressure gas flow test facilities, European communities EUR 6662, 1980. ISBN 92-825-1649-0

8. Diritti et al.,Intercomparison exercise of high pressure test facilities within GERG, GERG technical monograph TM6, Milan, 1993.

9. Eujen E., Untersuchungen über die Meßeigenschaften von Hochdruckgaszählern. II Teilbericht: Messungen an Schraubenradgaszählern, GWF, Vol. 105, Nr.4, Oct 23 1964.

10. Dijstelbergen H.H., Bergervoet J.T.M., Optimal straightening vanes for turbine meters, Fluid flow measurement 3rd international symposium, San Antonio, Texas, 1995, March 19-22, (Organised by GRI and SWRI).

11. Ballez G., Rombouts, P., Système de comptage de gaz naturel en unités d'énergie, Revue générale du gaz, Numéro 4, 1984, pages 5-17.

12. Lee W.F.Z., Blakesly D.C., White R.V., A self- correcting and self-checking gas turbine meter, Trans ASME,Vol. 104, 1982, pp.143-149.