27th International North Sea Flow Measurement Workshop
20 – 23 October 2009

A MULTIPATH ULTRASONIC METER WITH REDUCING NOZZLE FOR IMPROVED PERFORMANCE IN THE LAMINAR/TURBULENT TRANSITION REGION

Gregor J Brown, Cameron, UK
Terry Cousins, Cameron, USA
Donald R Augenstein, Cameron, USA
Herbert Estrada, Cameron, USA

---

1 INTRODUCTION

With rising worldwide energy demands and depletion of existing conventional oil reserves the production of heavy oil is becoming increasingly common. The high viscosity of heavy oils presents measurement challenges for most types of flow meter. For example it limits the maximum flow of PD meters, reduces the turndown of turbine meters and can result in measurement errors in Coriolis meters.

Ultrasonic meters can be used for measurement of high viscosity oils. However, in order to do so with high accuracy they have to cope with increased signal attenuation and changing velocity profiles through the transition from turbulent to laminar flow.

This paper explains the technical challenges faced when using ultrasonic meters for high viscosity/low Reynolds number flows and shows how these conditions can adversely affect the performance of some designs of ultrasonic meter. Modelling using velocity profile data and analysis of meter diagnostic data is presented in order to illustrate the physical processes that are at work. Test data is presented to demonstrate the performance of conventional and improved ultrasonic meter designs.

The improved ultrasonic meter design incorporates a reducing nozzle to flatten and stabilise the velocity profile in the transition region. The impact of this design feature on permanent pressure loss is also evaluated.

2 BACKGROUND

In 2001 US Department of Energy figures classified only 30% of the estimated 9 -13 trillion barrels of total world oil reserves as conventional oil, with heavy oil and extra heavy oil/bitumen accounting for 15% and 55% respectively. Production of heavy oils is more technically demanding than conventional production, requiring new and improved recovery methods and increased crude sales values to ensure its economic viability. The depletion of existing conventional reserves, rising worldwide energy demand, and new technology developments are therefore leading to increased activity in heavy oil production.

In the terminology of the American Petroleum Institute (API), heavy oil is classified as having API gravity of less than 22.3 (equivalent to a density of greater than around 920 kg/m^3 at 60 ºF). In terms of flow measurement, the relatively high density of the liquid is not problematic in itself. Consider for example that water has a density of around 1000 kg/m^3 and can be measured relatively easily by a wide variety of techniques. The main challenge comes from the fact that the higher the density of a crude oil is, the more viscous it is likely to be. This is illustrated in the viscosity versus temperature data shown in Figure 1 below for crude oils with densities between 830 and 930 kg/m^3 at 20 ºC.

Another important property of viscous oils is evident in Figure 1; the rate of change in viscosity with temperature increases with increasing density/viscosity, e.g. for a reduction in temperature from 30 to 20 ºC, the increase in viscosity for the three oils in Figure 1 is as follows:

- 832 kg/m^3 oil, viscosity changes from 4.6 to 5.6 cSt, an increase by a factor of 1.22
- 890 kg/m^3 oil, viscosity changes from 17.2 to 25.8 cSt, an increase by a factor of 1.5
- 930 kg/m^3 oil, viscosity changes from 124 to 240 cSt, an increase by a factor of 1.9

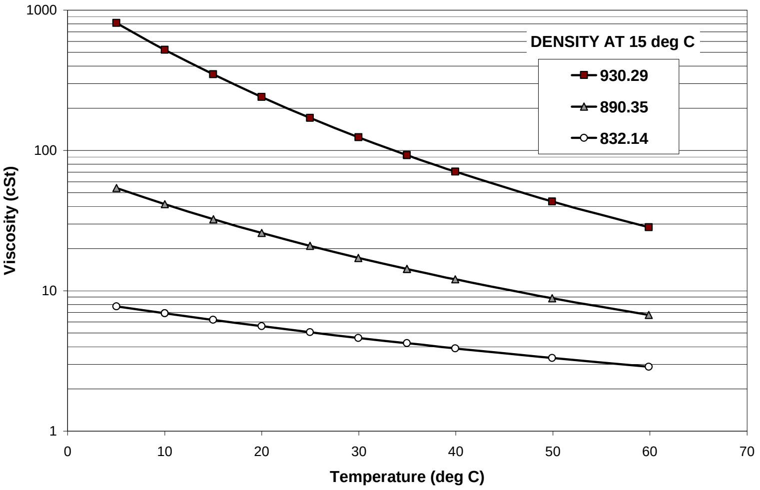


Figure 1 Viscosity versus temperature for crude oils of different densities

In addition to having high viscosity, heavy oils also tend to have higher than normal levels of contaminants such as wax, asphaltenes, sand, water, heavy metals and sulphur. Combined with potentially high process temperatures, these issues combine to produce a challenging environment for high accuracy flow measurement.

3 METERING OF VISCOUS OILS USING MECHANICAL AND MASS FLOW METERS

Some of the problems associated with measurement of viscous oils using conventional mechanical meters have been known for a long time. In the case of turbine meters, the increased drag associated with higher viscosity oils is known to make the meters more non-linear and restrict their operating range. Design modifications such as use of helical blades can reduce the sensitivity of turbine meters to viscosity changes, but even these modifications have a limited effect in terms of the increase in range of viscosity that turbine meters can easily handle. When it is taken into consideration that they also have moving parts and that for custody transfer they normally require regular in-situ proving, it is clear that turbine meters are not ideally suited for high accuracy measurement of highly viscous oils.

The relatively limited applicability of turbine meters in terms of viscosity range is illustrated in Figure 2 below, from the API Manual of Petroleum Measurement Standards [1]. The superimposed red line in Figure 2 shows the max viscosity limits taken from the FMC Smith Meter MV Series helical turbine meter data sheet [13].

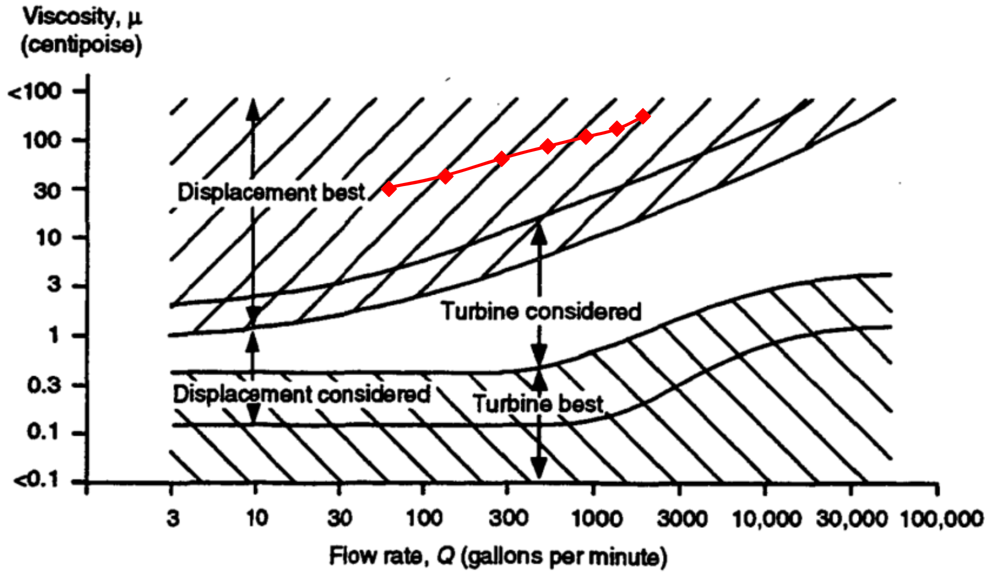


Figure 2 API selection guide for displacement and turbine meters

Positive displacement (PD) meters are recognised as being more robust and much less sensitive to viscosity changes. However, even these meters have limitations in high viscosity applications. They are also relatively expensive and have a lower flowrate capacity than turbine or ultrasonic meters of the same nominal pipe diameter, as illustrated in Figure 3 below. This means an application that could be served by one turbine or ultrasonic meter might require two or more PD meters in parallel.

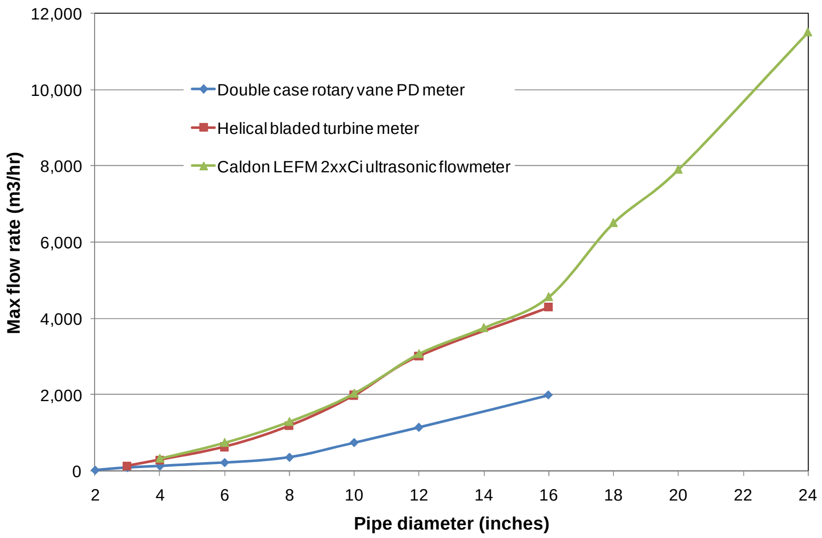


Figure 3 Flowrate capacity of PD, turbine and ultrasonic flowmeters

The response of PD meters to changes in viscosity is dependent on the amount of 'slippage' that occurs; that is the amount of fluid that leaks past the clearances in the internals of the meter. It is generally recognised that as the viscosity increases the amount of slippage reduces; the meter becomes more linear and the meter factor tends towards a constant value. However, when the viscosity gets very high it creates another problem, in that the increased shearing stresses may make it necessary to use larger clearances (thus reintroducing slippage) or reduce the maximum flowrate capacity of the meter. For example, a 16-inch PD meter might require increased clearances or reduced capacity for operation at viscosities above 200 cP. Increasing the clearances means that the same meter would be less accurate when measuring lower viscosity products of say 10 cSt.

Coriolis mass flow meters are now used regularly for custody transfer of crude oil. They are also used extensively in process applications where extremely high viscosities can be encountered, for example in the measurement of food products. Early experimental investigations with relatively low viscosity hydrocarbon products concluded that there was no obvious influence of liquid viscosity on Coriolis meters [2]. However, a recent study carried out by NEL has shown that Coriolis meter performance is affected by changing fluid viscosity. The NEL tests were carried out using two commercial Coriolis meters and showed errors in excess of 0.4 to 1% when the viscosity was increased to around 200 cSt [3].

4 METERING OF VISCOUS OILS USING ULTRASONIC FLOW METERS

Ultrasonic meters are now used regularly for custody transfer measurement of crude and refined oils. Significant potential benefits of ultrasonic meters are related to the fact that they have no moving parts and can be both non-intrusive and non-invasive.

It may appear at first glance that ultrasonic meters are ideally suited to measurement of viscous oils. However, when the principles of operation and the characteristics of viscous flows are examined in more detail it becomes apparent that measurement of viscous oils presents two issues that must be addressed:

- The response of the meter to extreme changes in fluid velocity profile
- Attenuation of the ultrasonic signals transmitted through the fluid

4.1 Hydraulic Effects in Low Reynolds Number Flows

Since the early studies of Osborne Reynolds in 1883 it has been known that the character of pipe flow changes from laminar conditions to turbulent conditions at Reynolds numbers greater than ~2,000 [4].

Laminar flow is characterised by the fluid moving in a direction parallel to the pipe axis and the velocity profile being parabolic in shape. Turbulent flow is chaotic with the flow at any point in the pipe having a mean velocity with superimposed random velocity components in three dimensions. Turbulent flow has a flatter profile shape that varies as a function of Reynolds number.

In the transition region between laminar and turbulent flows the profile continuously switches back and forth between states similar to laminar and turbulent flows, being similar to turbulent flow for a larger fraction of the time at higher Reynolds numbers in the transition region and being similar to laminar flow for a larger fraction of the time at lower Reynolds numbers in the transition region.

As multipath ultrasonic meters measure velocity on discrete paths and then combine these to obtain an estimate the average axial velocity in the pipe it is well known that these meters can exhibit some sensitivity to changing profiles. The sensitivity of different path configurations to velocity profile changes in turbulent flows (including distorted flows) has been the subject of numerous studies, e.g. [5]. A number of papers have also been published that include some data on the response of ultrasonic meters to changing velocity profiles in the transition region e.g. [6, 7]. However, there has been very little systematic study in this area.

The following subsections explore the issues of the influence of velocity profile in the transition region in more detail.

4.1.1 Reynolds number and viscosity limits for transitional flow

Before reviewing the performance of ultrasonic meters in the transition region and the hydraulic features of transitional flow, it is useful to consider the Reynolds number range in which transitional flow occurs, and the restrictions that this might place on standard ultrasonic meters in terms of liquid viscosity.

Numerous text books and papers suggest that transitional flow occurs in a range of Reynolds numbers between 2,000 and 5,000. Schlichting [8] states that numerous experiments show a lower bound for transitional flow of around 2,000 but that some researchers succeeded in maintaining laminar flows up to Reynolds numbers of 20,000 and 40,000 by minimising the disturbance to the flow entering the pipe.

For industrial flow metering applications it would seem consistent that transition is most likely to occur in the lower region between 2,000 and 5,000 Re, as inlet conditions are generally subject to significant disturbance. However, a number of other factors also affect the transition to turbulence, including wall roughness, vibration and heat transfer.

For application of the established range of full-bore Caldon ultrasonic meters, we at Cameron recommend a lower Reynolds number limit of 10,000. This is based on both field and laboratory experience with multipath ultrasonic flowmeters. Independent of the flow measurement performance of the device, Caldon meters are capable of providing diagnostic information that can be used to determine if the flowmeter is operating in laminar, transitional or turbulent flow. The diagnostic parameter most useful in this respect is termed the flatness ratio, FR, which is simply the sum of the outside path velocity measurements divided by the sum of the inside path velocity measurements, or with reference to Figure 4,

$$FR = \frac{V_1 + V_4}{V_2 + V_3}$$

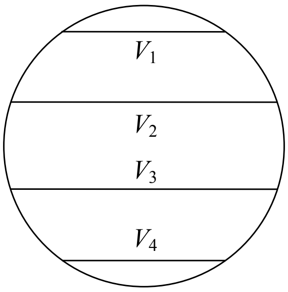


Figure 4 Schematic of a 4-path Caldon meter

Analysis and experiment show that in the case of 4-chord Gaussian path arrangements for turbulent flow the flatness ratio is greater than 0.75 and for laminar flow is less than 0.45.

Figure 5 below is a graph of experimental flatness ratio data showing the transition from laminar to turbulent flow as a function of Reynolds number. The flatness data was taken directly from the ultrasonic meters and the Reynolds number was calculated independently based on measurements of flowrate and viscosity performed by the test laboratories. One meter was a 6-inch Caldon meter tested at NEL in the UK and the other was a 12-inch Caldon meter tested at SPSE in France. It can be observed that in the case of the 6-inch meter the transition occurred at Reynolds numbers between 3,000 and 5,000 but in the case of the 12-inch meter the transition occurred between 6,000 and 9,000 Re. Although pipe diameter may be one influencing factor in this case, it does not seem likely that pipe diameter alone is responsible for the difference in the critical Reynolds number (as this would be contrary to the whole notion of Reynolds number similarity). What is clear however, is that the critical Reynolds number is subject to various influences, hence this is why we at Cameron apply a conservative lower limit of 10,000 Re for our standard Caldon ultrasonic meters.

It can also be seen from the data in Figure 5 that the transition occurs over a relatively narrow range of Reynolds number (and hence flowrate), meaning that the flow profile change is quite abrupt.

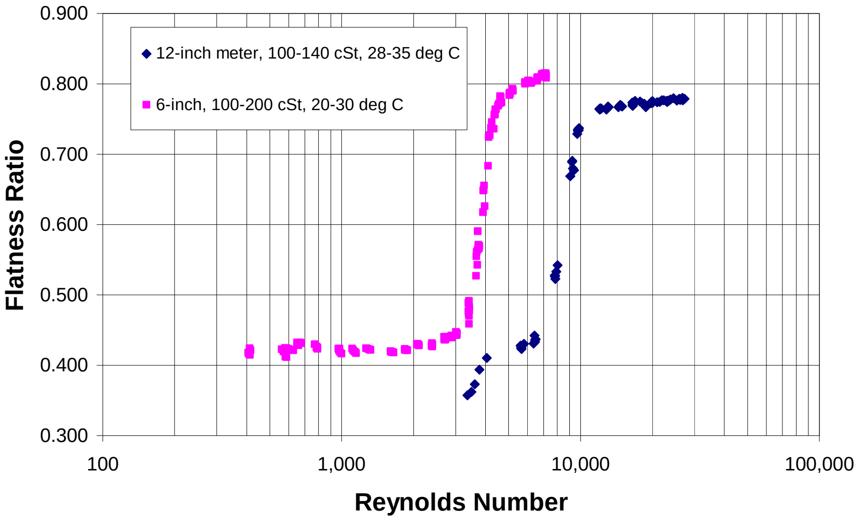


Figure 5 Flatness ratio data showing the transition from laminar to turbulent flow

We can examine the impact of imposing a low Reynolds number limit on the maximum viscosity for an ultrasonic flow meter very simply. Reynolds number is given by

$$Re = \frac{UD}{\nu}$$

where U is the mean velocity, D is diameter and ν is kinematic viscosity. Rearranging we can write

$$\nu_{max} = \frac{U_{min}D}{Re_{min}}$$

Figure 6 below shows the viscosity limits calculated for minimum Reynolds numbers of 5,000 and 10,000, assuming a minimum flow velocity of 1 m/s. It can be seen that the maximum viscosity in centistokes at 10,000 Re works out to approximately 2.5 times the pipe diameter in inches, and 5 times the pipe diameter for 5,000 Re. Clearly these Reynolds number based limits are very restrictive if we wish to measure heavy oils.

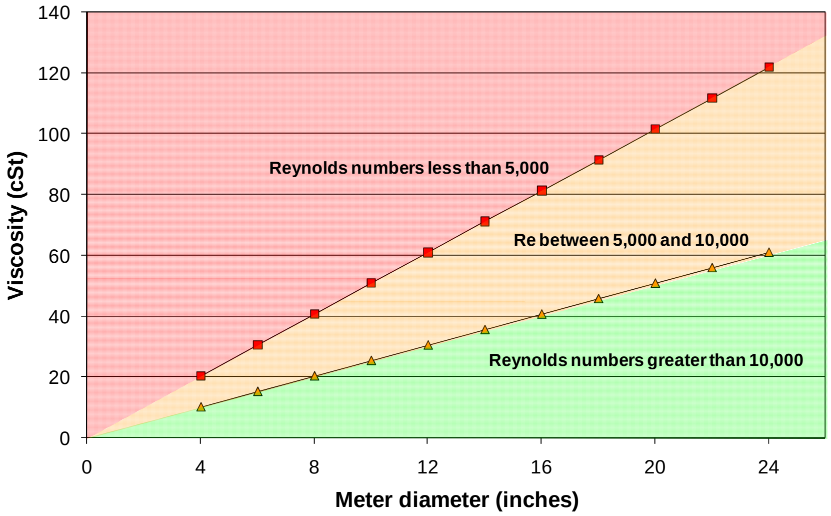


Figure 6 Viscosity limits for 1 m/s min. velocity as a function of diameter and Reynolds no.

4.1.2 Performance of conventional ultrasonic flowmeters in transitional flow

There are a number of papers and test reports that provide examples of degraded ultrasonic meter performance in the transition region. In a similar fashion to the improvements that multipath designs offer in turbulent flows [5], we find that multipath meters with four or more paths perform better than single-path and two-path meters. For example, Figure 7 below shows data from a test carried out by the author while working at NEL in 1997 on a 6-inch two-path (Krohne) ultrasonic meter [7]. The viscosity of the oil was approximately 24 cSt. It can be observed that the meter shows average errors of more than 2% at a Reynolds number of 3,000 but less than 0.5% at Reynolds numbers both above and below this value.

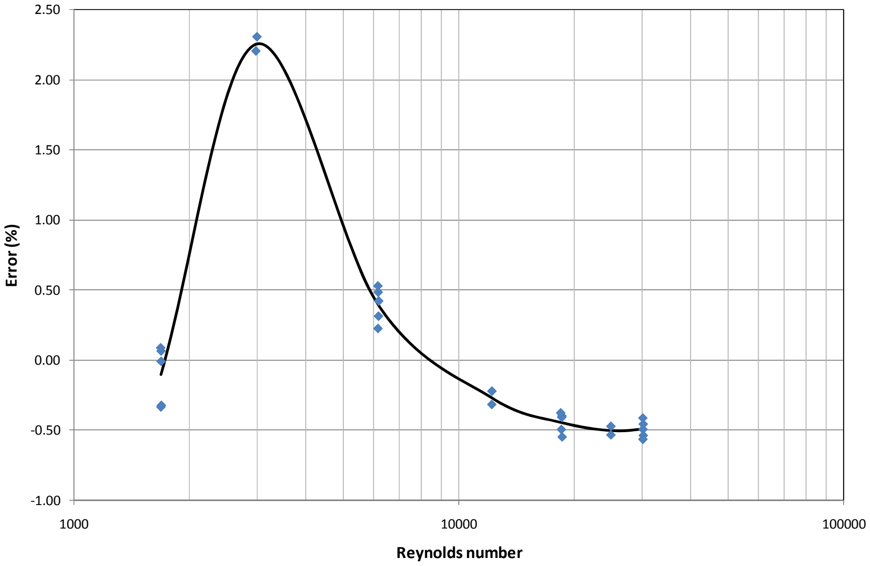


Figure 7 Two-path meter performance through the transition region

Figure 8 shows data from a 6-inch 4-path Caldon 240C full-bore flowmeter tested through the transition region at NEL in 2006. It shows that linear performance within +/- 0.15% can be achieved above and below the transition region, but that in transition the average errors can increase to around +/- 0.5%. Recent tests obtained at NEL using a multipath meter from another manufacturer show broadly similar results [3]. The NEL data was corrected offline by as a function of Reynolds number whereas Caldon meters perform any required linearization correction internally.

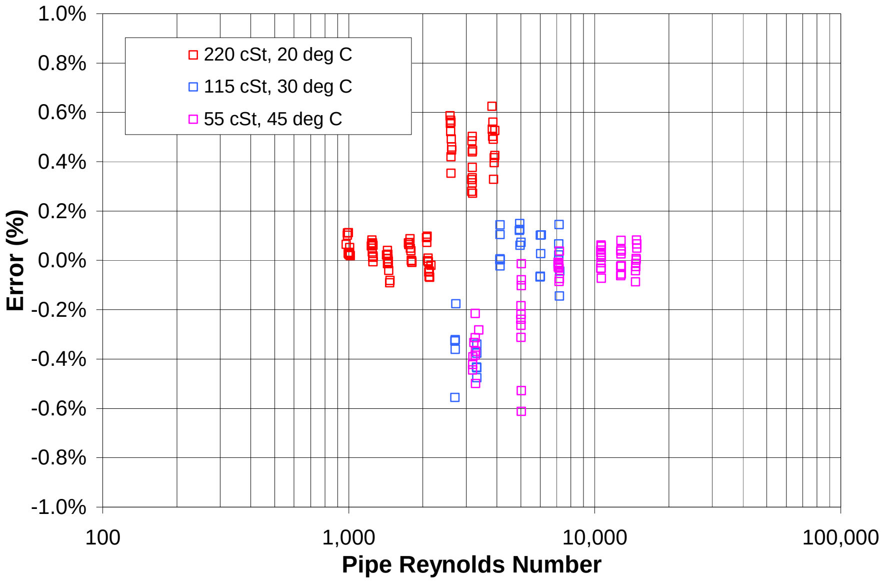


Figure 8 Four-path meter performance through the transition region

It can be observed in Figure 8 that the data 'scatter' appears to be greater in the transition region between 2,000 and 5,000 Re. It has been observed in transition, over the same range as the rapid variation in the shape of the mean velocity profile, that the short term repeatability of a standard meter is poorer than normal. This is illustrated in Figure 9 below, which plots the standard deviation of the path velocity measurements versus Reynolds number. This data is from the same four-path meter that was used to produce the data in Figure 8. It can be seen that the standard deviations peak in the transition region at a Reynolds number of around 3,000, with paths 1 and 4, which are closer to the pipe wall, being affected the most. Above the transition, in turbulent flow, the standard deviations decrease, and as should be expected they are lower still in the laminar region below 2,000 Re.

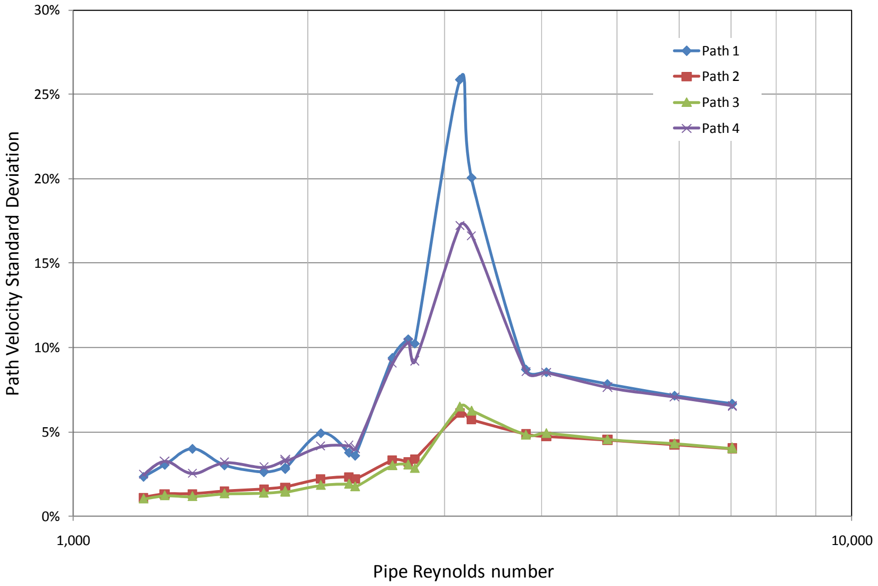


Figure 9 Path velocity standard deviations through the transition region

From the test data we can conclude that multipath designs can perform better than more simple designs in the transition region by virtue of better flow integration and the availability of additional information for correction purposes. However, it is clear that their performance is still degraded relative to how they behave in turbulent flow. Given that good performance can be achieved in turbulent flow conditions, and in laminar flow conditions, it remains to be explained exactly why performance is poor and the meter less repeatable in the transition region, where the flow profile is changing between its laminar and turbulent forms.

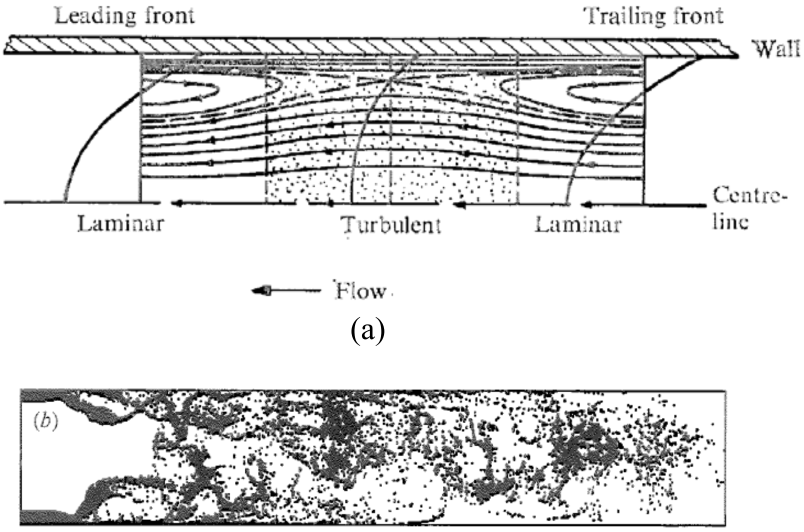
4.1.3 The behaviour of transitional flow

Experimental studies of laminar to turbulent transition have been carried out by a number of researchers [e.g. ref 9] and have included detailed measurement of velocity profiles using hot-wire anemometry. These studies have been complimented more recently with computational simulations of transitional flows using large eddy simulation methods [10]. Both the experimental and computational studies reveal some of the important characteristics of transitional flow.

Of particular relevance to our work with ultrasonic meters, these studies of transitional flow show that in the process of switching back and forth between laminar and turbulent forms, two things happen that result in adverse effects on the performance of ultrasonic meters:

1) In the process of changing from laminar to turbulent and back again the axial velocity profile takes on forms that are like neither turbulent nor laminar flow
2) During the process of rapidly changing from one axial profile form to the other, the fluid must move in toward or out from the axis of the pipe and this generates non-axial flow components

Figure 10 below reveals the structure of transitional flow. Figure 10(a) shows a schematic illustrating the profile changing from a laminar (parabolic) form to a flatter profile and back again, and the generation of non-axial flow components at the leading and trailing edges of the turbulent zone that accompany this process [9]. Figure 10(b) shows a particle tracing simulation illustrating the initial laminar condition, the leading edge of a turbulent 'slug' with large vortices near the wall, and the turbulent zone that follows [10]. It is this behaviour that is responsible for the increased variability of the velocity measurements in the transition region as illustrated in Figure 9.



(a)

(b)

Figure 10 The structure of transitional flow

Figures 11 and 12 show the detailed form of the axial velocity profile at various stages during the velocity profile switching from its laminar-like state to the turbulent zone and back again. The data is plotted as the velocity, *u*, normalised by the velocity in the centre of the pipe, u_c, versus the normalised radial position, *r*/*R*. These figures represent the two different forms of turbulent feature that have been observed in the transition region, one termed a slug and the other a puff. The velocity profiles in each case are shown for five locations relative to the turbulent feature, x* = 0 being the velocity before the turbulent zone, x* = 2 being in the centre of the turbulent zone, and x* = 4 being downstream of the turbulent zone. Hence the profiles at x* = 1 and x* = 3 are located at points where the velocity profile is rapidly changing. Observe that although the two graphs are not identical they do bear a close similarity to one another.

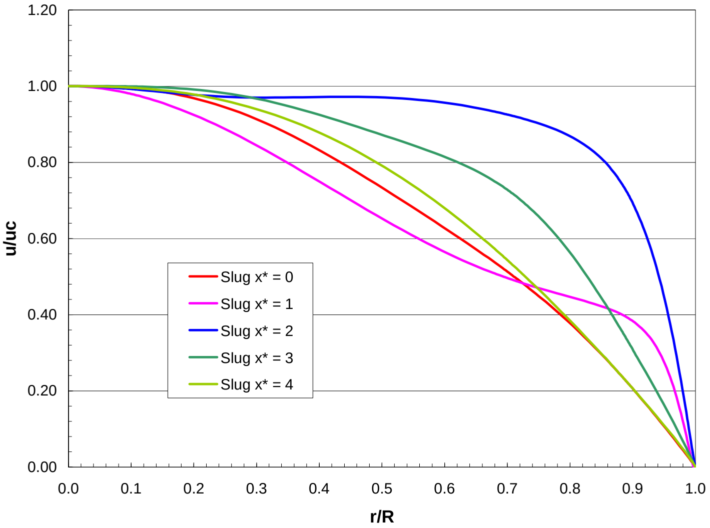


Figure 11 Average axial velocity profiles for a turbulent slug

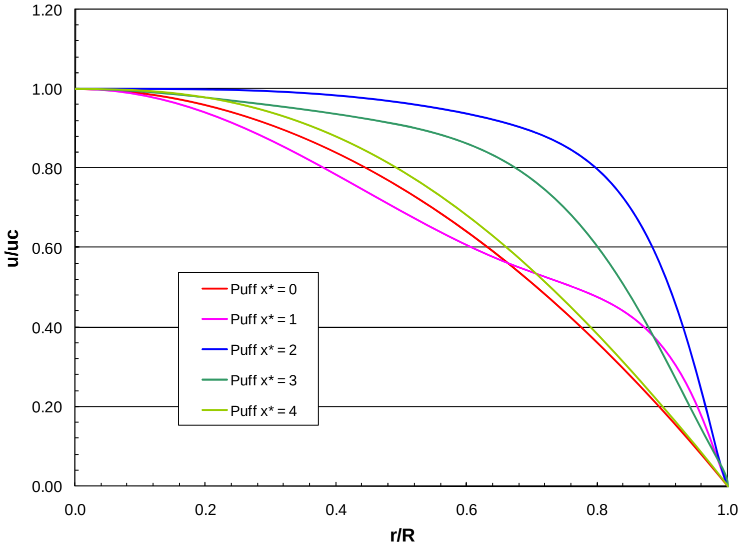


Figure 12 Average axial velocity profiles for a turbulent puff

The availability of such detailed velocity profile data allows an assessment of the impact of these velocity profile shapes on the accuracy of average velocity measurements made by multipath ultrasonic meters.

4.1.4 Velocity profile analysis

Previous analytical evaluations of the ability of the 4-chord Gaussian integration techniques used in Caldon meters show that path positions and weightings used in these meters are highly accurate for fully developed flow profiles in both turbulent and laminar conditions [e.g. ref 5]. These results are reproduced in Figure 13 below. However, it is obvious from Figures 11 and 12 above, that some of the velocity profiles observed in transitional flow are very different to either laminar or turbulent profiles, particularly those at x* = 1.

The velocity profiles in Figures 11 and 12 were modelled by fitting discrete data [11] with a function in the form:

$$\frac{u}{u_c} = r'\left(1 - r'^{j}\right)^{\frac{1}{n}} + (1 - r')\left(1 - r'^{s}\right)^{\frac{1}{t}} + mr'^{g}(1 - r')^{\frac{1}{k}}$$

where *r'* is the normalised radius *r/R*, and *j*, *n*, *s*, *t*, *g* and *k* are constants that change for each profile. This function can then be integrated over the whole cross-section to yield the true average velocity and along paths representing the ultrasonic flowmeter to yield the flowmeter's estimate of the average velocity.

The results of this exercise for a 4-path Gauss-Jacobi integration are shown in Figure 13 alongside the results for fully developed laminar and turbulent profiles as represented by parabolic and powerlaw equations. The data is plotted in the form of the velocity profile factor k_h (which is the true average velocity divided by the measured velocity) versus the velocity profile flatness ratio. It can be seen that in both turbulent and laminar conditions the velocity profile factor is very close to 1; the errors are predicted to be less than +/- 0.12%. In the transition region however, the results for the instantaneous transitional velocity profiles span from 0.994 to 1.005, i.e. +/- 0.6%. Clearly this offers an explanation for the magnitude of errors shown in Figure 8.

The data of Figure 13 suggests that a solution might be found by correlating the meter factor with the flatness ratio in a similar fashion to the methodology used by Caldon for very high Reynolds numbers [12]. However, practical experience has show that doing this alone is of limited use owing to factors that affect the reproducibility of flow profiles in the transition region (and hence the relationship between flatness ratio and meter factor).

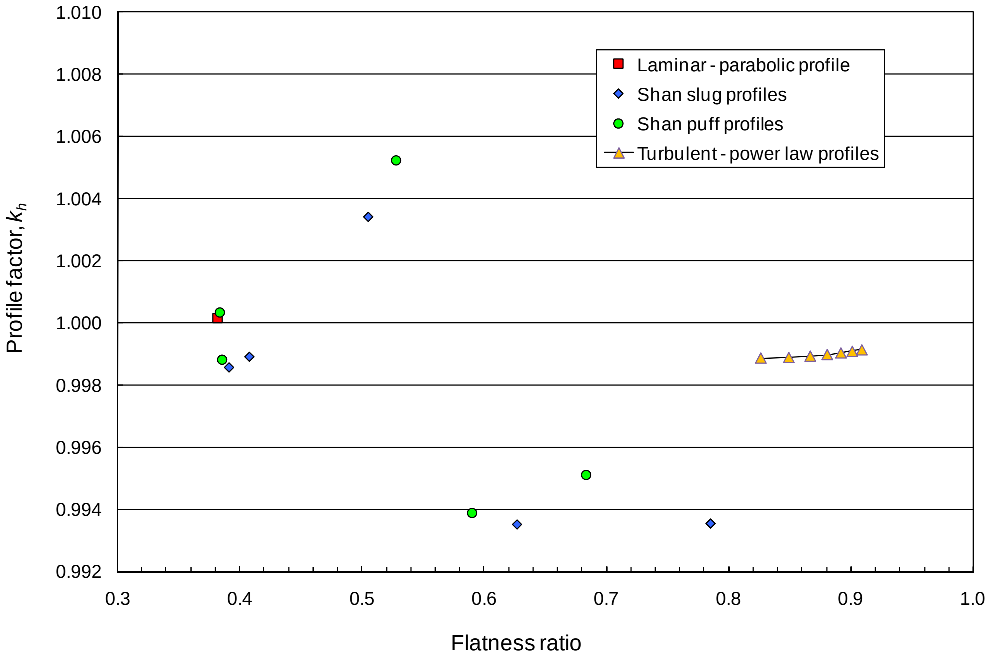


Figure 13 Profile factor versus flatness ratio for various forms of velocity profile

4.1.5 A solution to the problem of transitional flow

The preceding sections of this paper demonstrate that the poorer performance of ultrasonic meter in the transition region is related to the forms of velocity profile that occur when the flow is switching back and forth between laminar and turbulent states.

Avoiding operation in the transition region is difficult as often in the flow processes that we wish to measure there are severe constraints placed on the ability to adjust flowrate or viscosity. Experiments show that the bounds of transitional flow can be moved by altering flow conditions, but only to a limited extent, such that in practice any application range that crosses a Reynolds number range of 2,000 – 10,000 must experience transitional flow.

It has now been clearly demonstrated that it is the unusual velocity distributions that occur in transitional flow that cause problems with ultrasonic meters in this region, and this suggests that a solution may be found by artificially altering the velocity profiles upstream of the point of measurement.

Over recent years, Cameron has performed extensive experiments to evaluate the effectiveness of various forms of flow conditioning in transitional flows. Some forms of conventional flow conditioner that are normally intended to reshape the axial velocity profile and remove axial velocity components in turbulent flow can be shown to be of some use in limiting the effects of transitional flow. This is contrary to the implied conclusion of the NEL paper which stated that “the inclusion of an artificial flow conditioner provided no obvious improvement to the USM response under the elevated test viscosity conditions applied here” [3]. The apparent lack of improvement in the NEL experiments is a result of the type and location of flow conditioner, which was a tube bundle, presumably placed between 7 and 10 diameters upstream of the meter. This specific result does not disagree with our findings; tube bundles do relatively little to reshape the axial flow profile; but we found several other forms of flow conditioning to have quantifiable benefits in terms of limiting the effects of transitional flow.

One of the methods of altering the flow profile that we have thoroughly tested, and since incorporated into the Caldon product line, is the use of a reducing nozzle immediately upstream of the ultrasonic measuring section. This is illustrated in Figure 14 below in the form of an integrated nozzle with 8-path metering section and downstream expansion cone. The diameter ratio, β (throat diameter over pipe diameter) for this meter design is ~0.63, and was selected as a practical compromise between meter performance and pressure loss. Values of beta greater than 0.64 are not used for this purpose, as the flattening effect on the profile becomes less effective with increasing beta. A photograph taken looking into the throat of a 6-inch 4-path meter with integrated reducing nozzle is shown in Figure 15.

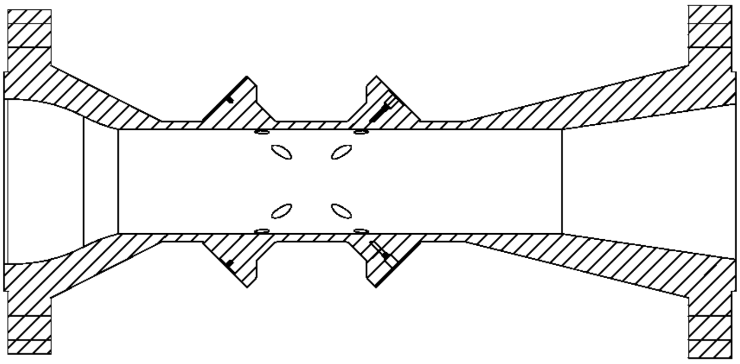


Figure 14 A schematic of an 8-path Caldon meter with reducing nozzle inlet and conical outlet

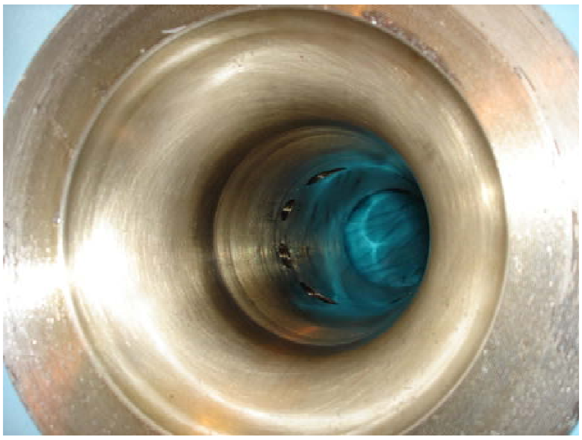
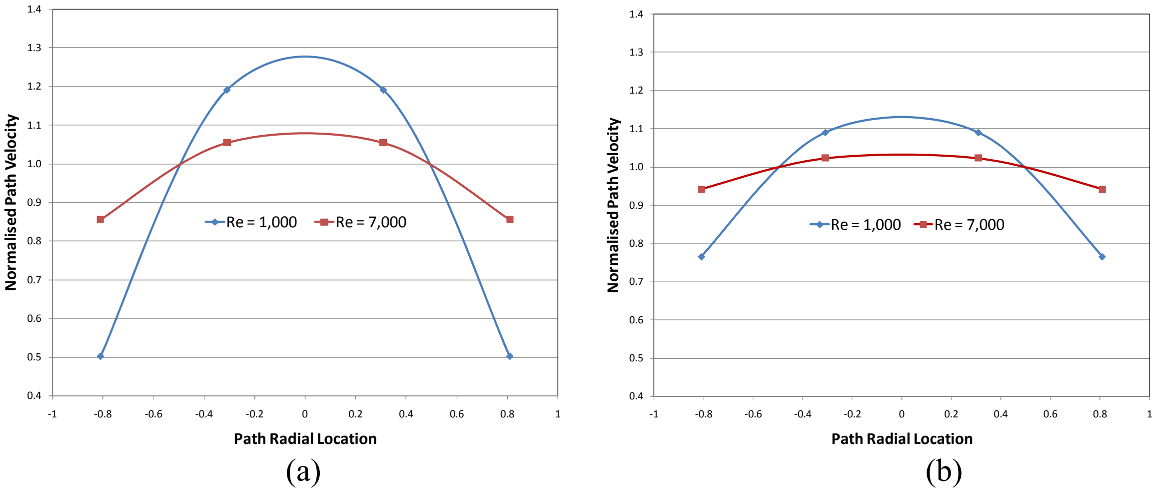


Figure 15 The a 6-inch test meter with reducing nozzle inlet

The contour of the nozzle has been designed so as to avoid negative pressure gradients along the nozzle surface. Such gradients could induce boundary layer separation and hence turbulence.

The reducing nozzle is beneficial in the transition region owing to the way it acts upon the fluid entering the measurement section. In effect what the nozzle does is to flatten and stabilise the velocity profile owing to the motion toward the centre of the pipe that is imposed on the fluid. Importantly this means that not only are non-axial flow components suppressed, but the shape of the axial velocity profile in both laminar and turbulent conditions become more alike. This is shown in Figure 16 below, where the normalised path velocities are shown for (a) a normal 6-inch full-bore meter and (b) a 6-inch meter with a reducing nozzle at Reynolds numbers of 1,000 and 7,000 (i.e. either side of the transition region).



(a) (b)

Figure 16 Velocity profiles for (a) a standard 4-path meter; and (b) a 4-path meter including a reducing nozzle with a diameter ratio of 0.63

Not only does the reducing nozzle bring the laminar and turbulent profiles closer together, but it also causes the profile to change more gradually as Reynolds number is reduced. This is shown in Figure 17 below, where the profile flatness measured by the meter is plotted versus Reynolds number. It is clear that in the case of the meter with integrated reducing nozzle the transition is much more gradual when compared with the abrupt profile transition that is seen by the full-bore meter.

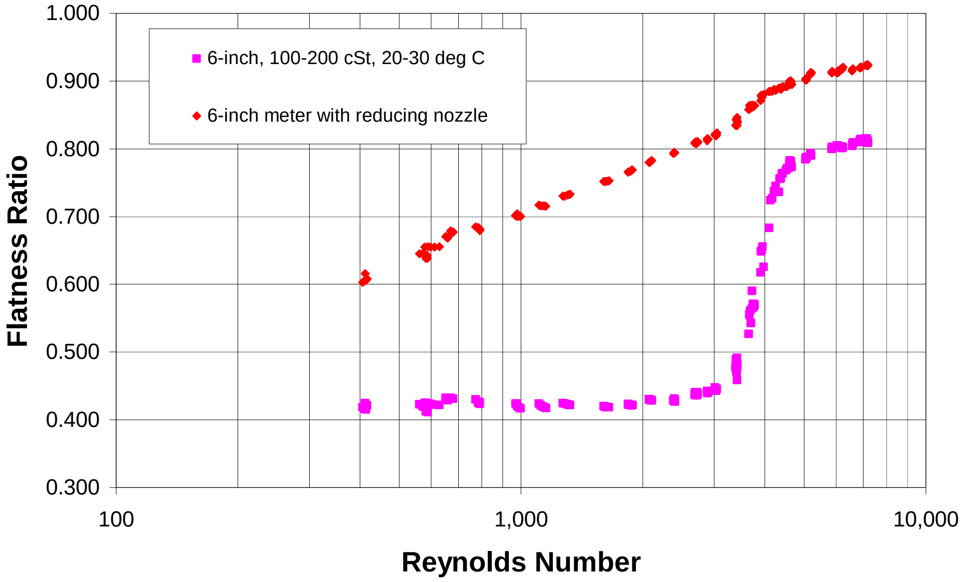


Figure 17 Flatness ratio versus Reynolds number for a conventional full-bore meter and a meter employing a reducing nozzle with a diameter ratio of 0.63

As suggested above, the ‘smoothing’ of the velocity profile accomplished by use of the reducing nozzle is accompanied by suppression of the very strong non-axial flow components normally found in transitional flows. The end result is that meter repeatability is also improved in the transition region relative to a standard full-bore meter design. This is illustrated in Figure 18 below, which plots the standard deviation of the path velocity measurements versus Re for the 6-inch meter with reducing nozzle of β = 0.63. By comparison with Figure 10 it can be observed that the effect of the reducing nozzle is to reduce the standard deviations in the transition region dramatically; from a peak value of than 25% in the case of the full-bore meter to less than 6% when the reducing nozzle is employed. It can also be observed that the reducing nozzle has a beneficial effect in turbulent flow.

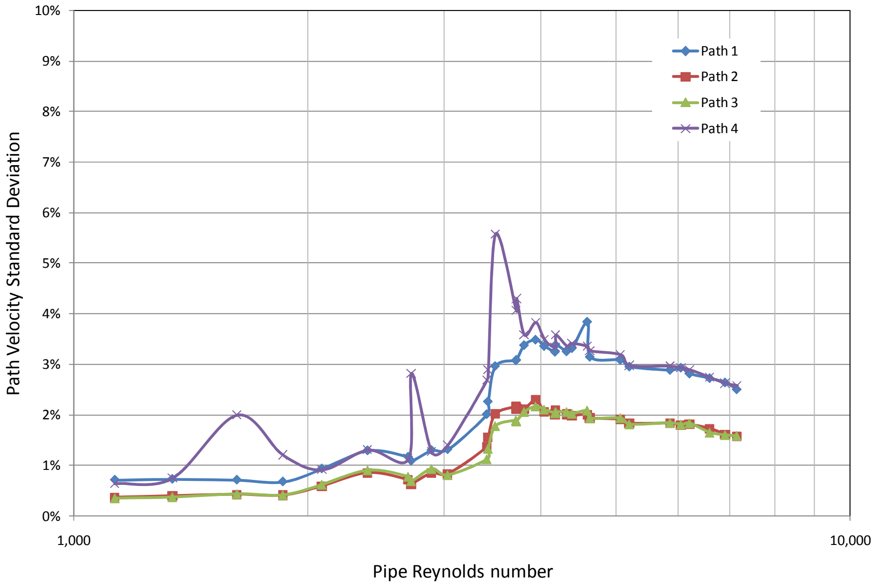


Figure 18 Path velocity standard deviations through the transition region for a meter employing a reducing nozzle of β = 0.63

Overall, the impact of the reducing nozzle on performance of the flow meter is to extend the performance that is normally achieved in turbulent flows right through the range of pipe Reynolds number for transitional flow. This is demonstrated in Figures 19 and 20, which show data for meters tested at NEL through the transition region. Figure 19 shows the performance of a 6-inch meter with reducing nozzle tested at a viscosity of ~270 cSt. This data straddles the range of Reynolds number where transition region effects would be seen in the case of a standard full-bore meter.

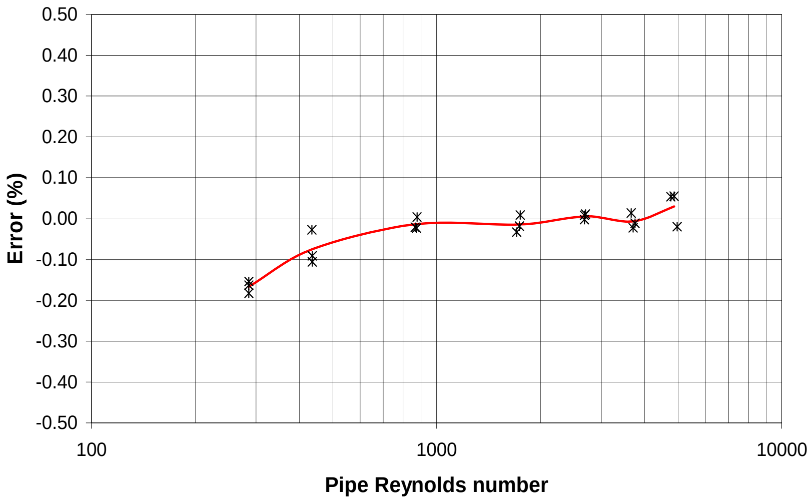


Figure 19 Error vs Re for a 6-inch meter with reducing nozzle at a viscosity of ~270 cSt

Figure 20 shows data from a 12-inch meter with reducing nozzle tested at three different temperatures resulting in viscosities of 50, 80 and 120 cSt. This result is important as it demonstrates stability of the meter factor within +/- 0.15% with changing temperature and viscosity through the transition region. The reason to stress the importance of this result is that when other ultrasonic meter types are tested and transitional flow does have an influence on the meter factor, a single product calibration over a limited temperature range may be insufficient to show the effects. It is variation in process conditions in the transition region that result in the lack of reproducibility shown for the full-bore meter in Figure 8 – the effects of transition could be calibrated out for any one condition, but not for all three.

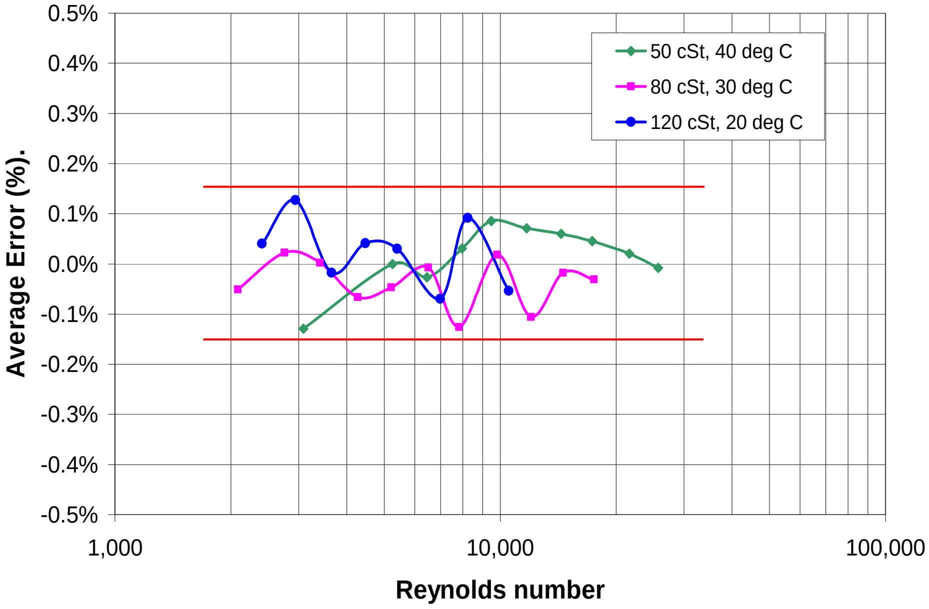


Figure 20 Error vs Re for a 12-inch meter with reducing nozzle

Figure 21 shows the data from Figure 19 presented as a function of flowrate rather than Reynolds number. This demonstrates that the meter design with reducing nozzle can achieve +/- 0.15% over a range of 10:1, and +/- 0.2% over a range of 15:1. It should also be noted that although this meter design is well suited to low Reynolds number applications, it can also be used at much higher Reynolds numbers and has been tested successfully up to Reynolds numbers of 900,000.

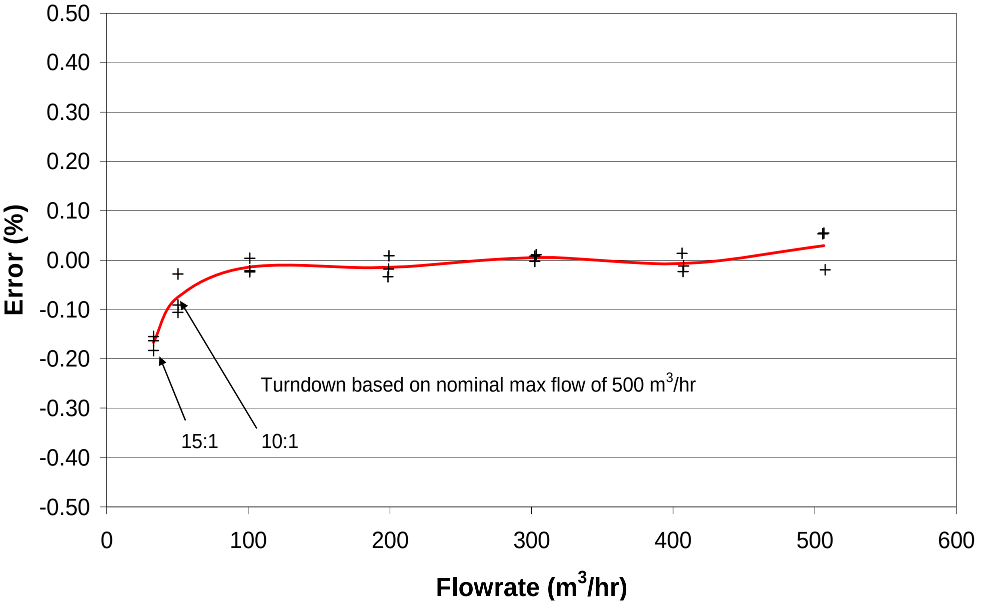


Figure 21 Error vs flowrate for a 6-inch meter with reducing nozzle at ~270 cSt

4.2 VISCOSITY AND ATTENUATION OF ULTRASOUND

Having established that the hydraulic problems in the transition region can be overcome, it is now required that we review the effects of attenuation in high viscosity applications.

The amplitude A of an ultrasonic signal that has been transmitted through a fluid is given by the exponential function:

$$A = A_0 \exp(-\alpha x)$$

Where A_0 is the amplitude of the signal in the absence of attenuation, x is the distance travelled and α is the attenuation coefficient.

For viscous absorption the attenuation coefficient is given by

$$\alpha_{vis} = \frac{8\pi^2}{3} \frac{K\nu}{c_f^{\,3}} f^2$$

where ν is the kinematic viscosity, c_f is the sound velocity of the fluid, f is the frequency of the ultrasound, and K is a constant that is dependent on the nature of the fluid.

The equations above show that the attenuation coefficient is proportional to viscosity and frequency squared. This allows us to examine the effect of viscosity on signal strength for various combinations of path length and transducer frequency typical of those used in ultrasonic flowmeters.

Figure 22 below shows the relative signal amplitude versus path length for three different combinations of viscosity and frequency. Taking first the case of a 5 cSt oil and a signal frequency of 1 MHz we can see that the signal is attenuated by less than 10% over a path length of 40 inches. If the oil viscosity is increased to 500 cSt, the 1 MHz signal would now attenuate much more rapidly, with the signal being reduced to less than 10% when the path length is just 10 inches long. Using a lower frequency of 0.5 MHz, the attenuation for a given viscosity is reduced, and now at 500 cSt with a 10-inch path length the attenuation is less than 50%.

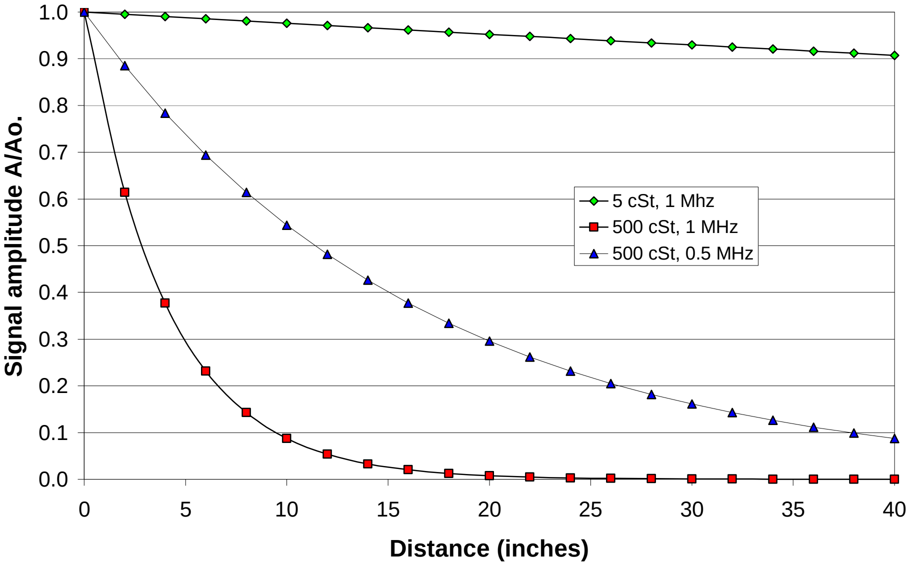


Figure 22 Signal amplitude vs distance for three combinations of frequency and viscosity

Given an understanding of how frequency and viscosity affect signal attenuation, it is important to consider how much attenuation can be allowed before the accuracy of the ultrasonic transit time measurement is affected. This is dependent on a number of factors, including the quality of the electronics, the method of transit time detection employed, the size of the meter and the minimum flowrate that we wish to measure.

In terms of the quality of the electronics, the ratio of the measured signal to the coherent (stationary or correlated) noise is important. If, for example, a signal-to-coherent noise ratio (SNR_c) of 200:1 is required for accurate transit time measurement, and the flowmeter has baseline SNR_c of 1000:1 , then we can estimate that the maximum allowable attenuation would be -14 dB (i.e. a five-fold reduction in signal strength). Then with knowledge of the meter geometry and signal frequency, it is possible to use the equations above to estimate the maximum viscosity corresponding to this amount of attenuation.

From the equations and the data presented in Figure 22, it is clear that viscous attenuation is more severe over long distances. However this potential limitation is offset by the fact that the transit times also increase in large pipe diameters, thus placing lower demands on the timing accuracy, and hence lower requirements in terms of the signal-to-noise ratio of the received signals. It can be shown that to achieve the same relative uncertainty in the transit time measurements at a given velocity, the required signal-to-coherent-noise ratio is inversely proportional to the path length (or pipe diameter). This is illustrated in Figure 23, which shows a simple example of the required value of SNR_c for a single diameter path at an angle of 45°, sound velocity equal to 1250 m/s, a minimum velocity of 1 m/s and the requirement for +/- 0.15% maximum uncertainty in the transit time measurements.

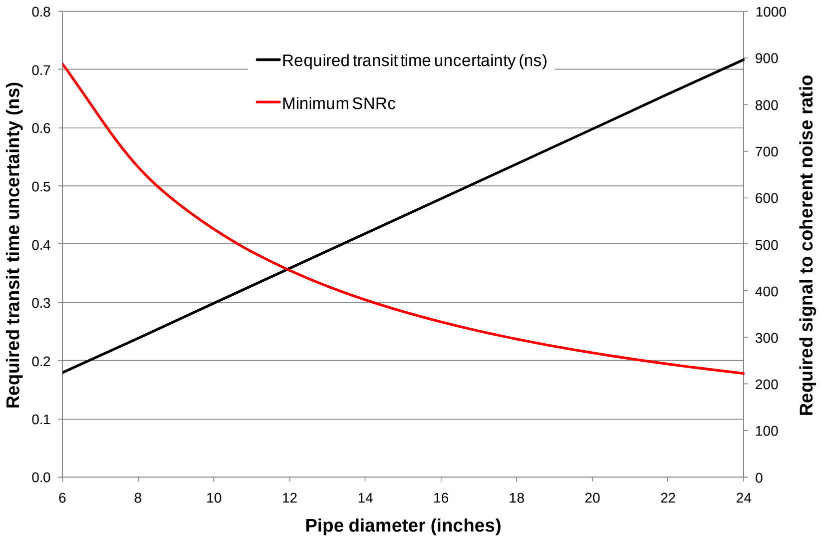


Figure 23 Example calculation of required SNR_c for a single diametric path

Reducing the signal frequency requires a higher signal-to-noise ratio to maintain the same timing uncertainty, but also reduces the viscous attenuation. To examine the trade off between these opposing requirements, we can use the baseline and minimum required SNR_c to calculate the allowable attenuation as a function of pipe diameter, and then translate this into a maximum viscosity.

Figure 24 below shows the result of performing this calculation for the single diametric path example that was also used to generate the data in Figure 23. A baseline SNRc of 1250 has been assumed for this example. It can be seen that the reducing demands on transit time measurement accuracy with increasing size do indeed act to offset the increased attenuation and the resulting maximum viscosity is fairly constant over a wide range of meter size. It can also be observed that the lower signal frequency of 0.5 MHz significantly increases the viscosity limit for meters of 12-inch diameter and above.

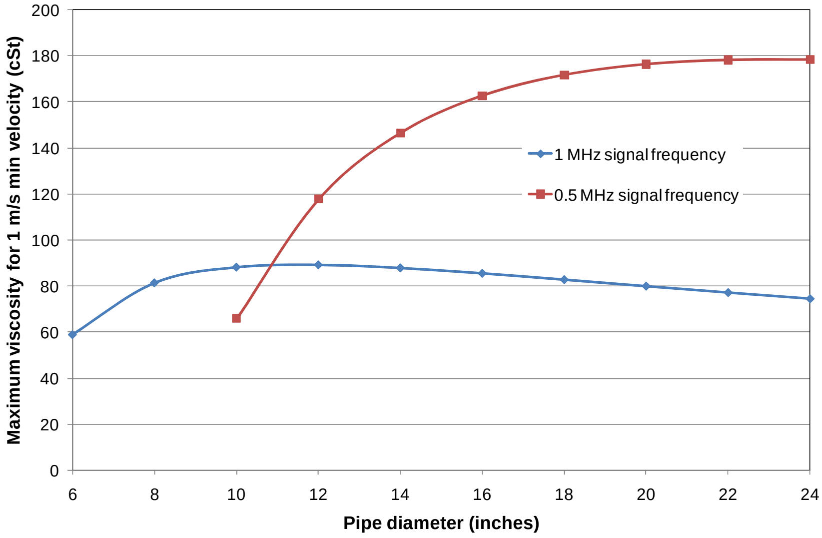


Figure 24 Example calculation of maximum viscosity for a single diametric path

When we now consider the multipath meter with reducing nozzle we find that there are further benefits of this design when considering high viscosity applications. Firstly the path lengths are obviously shorter than the paths in a full-bore meter of the same pipe diameter, so less attenuation is incurred. Secondly, although the overall transit times are also shorter, the velocity in the throat is much higher, and hence for the same velocity in the upstream pipe, the transit time difference is higher and the demands in terms of signal-to-noise ratio are lower. The result is that the meter with reducing nozzle can tolerate much higher viscosity than a full-bore meter of the same pipe size. This is illustrated in Figure 25 where the calculated maximum viscosity limit is compared for the single diametric path full-bore meter, and the multipath meter with reducing nozzle. As before the baseline SNRc and minimum pipe velocity are assumed to be 1250 and 1 m/s respectively, and the uncertainty limit is set at +/- 0.15%.

Figure 25 also shows the maximum viscosity calculated for the multipath meter with reducing nozzle when the velocity is increased to 5m/s. It can be observed that with the increase in the velocity, the maximum viscosity also increases. This means that in applications where a limited turndown is acceptable (such as batch offloading from floating production storage and offloading vessels where low velocities constitute only a very small part of the batch) even higher viscosities can be tolerated.

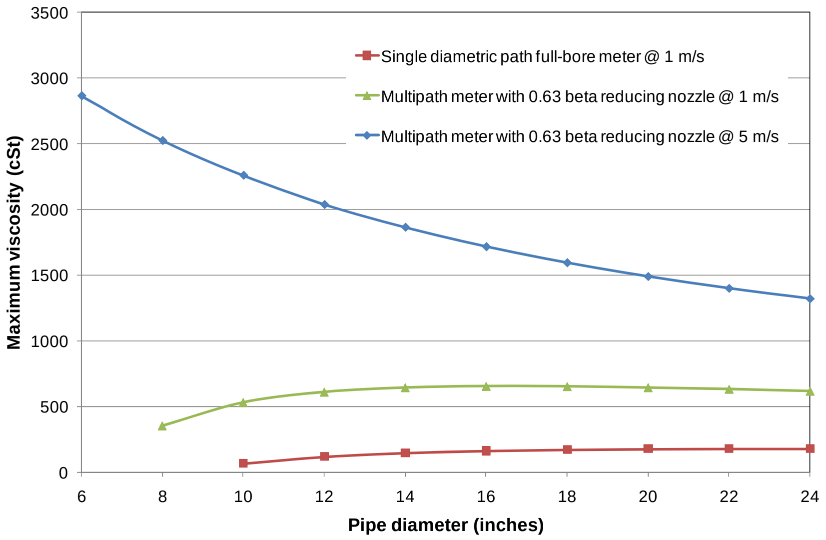


Figure 25 Example calculation for full-bore and reducing nozzle meters at 0.5 MHz

In reality the relationships between viscosity, meter diameter, received signal frequency and signal-to-noise ratio are more complex than we have described here. For example, the level of coherent noise can actually reduce with increasing viscous attenuation, with the result that the signal-to-noise ratio does not reduce as much as expected. Furthermore, it is possible to allow different signal frequencies for different sized meters and gain further improvements. For Caldon meters a database of results has been compiled by performing experiments and varying each of the relevant parameters, and this allows empirical evaluation of the maximum viscosity limits for each meter size. However the analytical results in this section do reflect the geneneral trends and serve to highlight the following facts:

- High viscosities can be tolerated without compromising measurement uncertainty
- The limit of viscosity increases with increasing flow velocity
- Use of appropriate signal frequencies can extend operation to higher viscosities
- The meter design with reducing nozzle can tolerate higher viscosities than a full-bore meter design

4.3 Pressure Loss Comparisons

One of the potential benefits of ultrasonic meters is that they can be non-intrusive and therefore create no more pressure loss than a standard piece of pipe of the same length. By adding a reducing nozzle to the meter the permanent pressure loss in increased, so it is of interest to compare this with other metering technologies to evaluate if the pressure loss advantage is retained.

Estimated pressure loss has been calculated for a nominal pipe diameter of 6-inches and viscosity of 100 cSt. The calculation for the turbine meter was taken from the Smith MV Series turbine meter data sheet [13]. The data for the tube bundle and one of the calcs for the Coriolis meter (calc 1) were obtained by assuming pressure drop proportional to velocity squared and interpolating the data published by NEL [3]. The PD meter data is taken from the data sheet for the Smith Meter 6" Steel Model G6 PD Rotary Vane Meter, and is actually for a low viscosity product of 2 cSt [13]. The second Coriolis meter calculation is taken from the E&H Promass F technical manual [14].

It can be seen from Figure 26 that the pressure loss increases in the following order:

- Full bore ultrasonic meter
- Ultrasonic meter with reducing nozzle and recovery cone
- Turbine meter
- Tube bundle
- PD meter
- Coriolis meters

It is clear therefore that the pressure loss advantage of the ultrasonic meter is retained, even when a diameter ratio of less than 0.64 is used to ensure an effective flattening of laminar velocity profiles.

The calculations for the tube bundle show a higher pressure loss than would normally be expected for this type of device. This is because the relatively high viscosity results in laminar or transitional conditions in the tubes themselves and increases the frictional losses relative to the results that would be obtained using the loss coefficient that is normally applied in turbulent flow.

The fact that the tube bundle alone creates a significant pressure loss is worthy of further discussion. This result emphasises the fact that an ultrasonic meter has the greatest pressure loss advantage when used without flow conditioning. Given the known sensitivity of typical multipath meter designs to swirl, dispensing with upstream flow conditioning requires the use of a meter design that is insensitive to swirl, such as the Caldon 280C 8-path meter design [5]. The Caldon meter with reducing nozzle can be made in both 4-path and 8-path formats, and therefore can be used in most circumstances without requiring additional upstream flow conditioning.

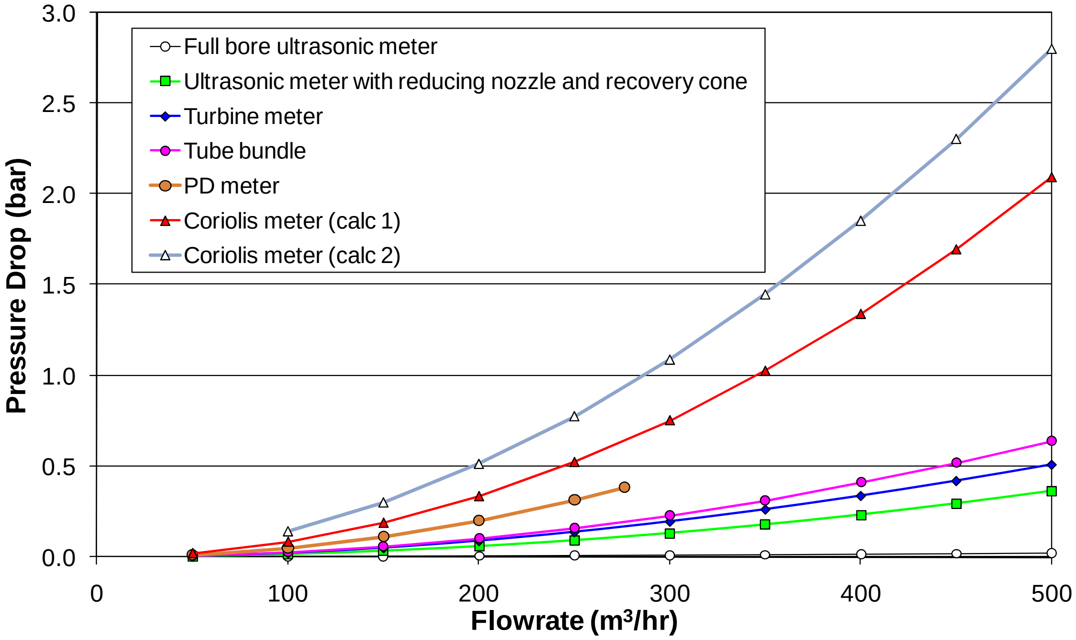


Figure 26 Permanent pressure loss comparison for meters in a 6-inch pipe

5 CONCLUSIONS

Metering of viscous oils presents challenges for many types of flowmeter. Positive displacement meters are well suited for high accuracy measurement of viscous oils but suffer from being fairly expense, having limited flow capacity and using moving parts that are susceptible to damage and wear. Turbine meters are sensitive to viscosity changes, are normally limited to lower viscosity products and also have moving parts. Coriolis meters appear promising for measurement of viscous oils but generate relatively high pressure losses and are reported to be susceptible to errors as viscosity increases.

Ultrasonic meters also have to overcome a number of issues in order to be used successfully for measurement of viscous oils at low Reynolds numbers. Firstly they must tolerate increased signal attenuation and still operate with sufficiently good signal-to-noise ratio to permit accurate transit time measurements to be made. Secondly, they must be able to accurately measure in the laminar/turbulent transition region where fluid mechanics dictate that rapid changes in velocity profile occur.

A method for improving ultrasonic meter performance in the transition region has been presented whereby a reducing nozzle with a diameter ratio of less than 0.64 is used to stabilise and flatten the velocity profile. It has been shown that this not only brings the laminar and turbulent profiles closer together but also results in a more gradual transition leading to improved meter performance in terms of both repeatability and reproducibility. Furthermore, in terms of attenuation this design extends the maximum viscosity that can be tolerated relative to a conventional full-bore meter of the same pipe size.

The resulting new meter design extends the turbulent flow performance of Caldon ultrasonic meters through the transition region whist still maintaining lower permanent pressure loss than turbine, PD or Coriolis meters.

REFERENCES

[1] API (1995), *Manual of Petroleum Measurement Standards – Chapter 5 – Metering, Section 1 – General Considerations for Measurement by Meters*

[2] Nicholson, S (1994) *Coriolis Mass Flow Measurement* Proceedings of Flomeko 93, Glasgow, Scotland, June 1994

[3] Miller, G J and Belshaw, R (2008) *Measurement of Flow in Viscous Fluids* 7th South East Asia Hydrocarbon Flow Measurement Workshop, Kuala Lumpur, Malaysia

[4] Reynolds, O (1883) *An experimental investigation of the circumstances which determine whether the motion of water in parallel channels shall be direct or sinuous and of the law of resistance in parallel channels.*

[5] Brown, G J, Augenstein, D R and Cousins, T (2006) *The relative merits of ultrasonic meters employing between two and eight paths* 5th South East Asia Hydrocarbon Flow Measurement Workshop, Kuala Lumpur, Malaysia.

[6] Fronek, V (1978) *Ultrasonic measurements of oil flow in a laminar flow-turbulent flow transition region* Proceedings of FLOMEKO '78, Groningen, The Netherlands, 11-15 September 1978, H H Dijstelbergen & E A Spencer (eds), North-Holland Publishing Co, Amsterdam, pp. 141-146.

[7] Brown, G J (1999) *Long Term Evaluation of Ultrasonic Meters, Project No: OR7 (OSDC57)*, National Engineering Laboratory Report No. 315/99, September 1999.

[8] Schlichting, H (1968) Boundary Layer Theory, McGraw-Hill, New York.

[9] Wygnanski, I J and Champagne, F H (1973) *On transition in a pipe, Part 1: The origin of puffs and slugs and flow in a turbulent slug* J Fluid Mechanics, Vol 59, Part 2, pp 281 - 335.

[10] Shan, H, Ma, B, Zhang, Z and Nieuwstadt, F T M (1999) *Direct numerical simulation of a puff and a slug in transitional cylindrical pipe flow* J Fluid Mechanics, Vol 387, pp 39 - 60.

[11] Shan, H (2006) Personal communication.

[12] Brown, G J, Estrada, H, Augenstein, D R and Cousins, T (2007) *Ultrasonic metering of liquefied natural gas for allocation and custody transfer*, 6th South East Asia Hydrocarbon Flow Measurement Workshop, Kuala Lumpur, Malaysia

[13] http://info.smithmeter.com/literature/online_index.html

[14] http://www.endress.com: *Technical Information Proline Promass 80F, 80M, 83F, 83M Coriolis Mass Flow Measuring System*