North Sea
FLOW
Measurement Workshop

Paper 6.1

Thermal Gradient Effects on Ultrasonic Flowmeters in the Laminar Flow Regime

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1 INTRODUCTION

With the rising oil price and depletion of 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.

Ultrasonic meters can be used for measurement of high viscosity oils. In order to do so there are two challenges that are commonly acknowledged: (1) the need to overcome increased signal attenuation; and (2) the requirement to operate accurately through the transition region where velocity profiles vary dramatically. It is often assumed that accurate measurement is the laminar regime is less difficult than in transitional flow. However, relatively little information on the performance of ultrasonic meters in laminar flow has been published.

This paper focuses on experimental test results obtained in laminar conditions where the oil temperature and ambient temperature are different. Tests have been performed in a variety of situations, with different installation conditions, meter types and with insulated meter runs. An explanation of the mechanism by which thermal gradients affect the performance of ultrasonic flow meters is provided, supported by diagnostic data. Results from tests on a novel flow conditioner, designed to improve performance of ultrasonic meters in the laminar regime, are also presented.

2 BACKGROUND

In the majority of industrial applications of ultrasonic flow meters, the flow is turbulent. Many text books and technical review papers that deal with ultrasonic meters barely mention laminar flow, and fail to speculate about performance issues in that regime. Numerical calculations of the 'hydraulic correction factor' or 'flow coefficient' for fully developed laminar flow show it as single valued, being 0.75 for a diameter path [1] and close to 1 for Gauss-Jacobi integration with four paths [2], suggesting that (in the absence of upstream bends etc), the calibration of an ultrasonic meter does not vary in laminar flow.

With increasing focus on heavy oils, a need has arisen to achieve custody transfer levels of uncertainty with products of high viscosity flowing at relatively low Reynolds numbers. In order to achieve the performance required there are two obstacles that are generally acknowledged: (1) the need to overcome increased signal attenuation; and (2) the requirement to operate accurately through the transition region where velocity profiles vary dramatically.

By optimising meter body design, transducers, electronics and signal processing, the issues of high attenuation and the effects of transitional velocity profiles can be managed. By combining the right elements, this means that meters can operate through the turbulent and transition regions with viscosities in excess of 1000 cSt [2]. This might tend to suggest that measurement to custody transfer levels of uncertainty can be easily achieved laminar flow. However, laminar flow has complexities of its own.

First let us deal with the issues of flow velocity profile. It is known that laminar flow can potentially require a long length of straight pipe before it becomes fully developed. Data in Miller's Internal Flow Systems [3] shows laminar flow in a pipe (following a smooth inlet from a tank) reaching fully developed conditions in approximately 3 diameters at a Reynolds number of 100 but more than 50 diameters at a Reynolds number of 2,000. So for Reynolds numbers below say 300, the upstream straight length requirements should not be particularly demanding. For Reynolds numbers approaching transition, profile distortion may persist for much longer. However, we know that velocity profile distortions can be dealt with by use of multiple paths. Furthermore, conventional flow conditioners and/or reducing nozzles can also be used to condition the profile. So it can be reasoned that flow profile distortions are not particularly problematic in laminar flows when an appropriate multipath meter design is used.

Now let us consider what other issues may arise in laminar flow. In Miller's book [3] the following statement is made:

"In internal flow, turbulence is a phenomenon which is considered desirable in some situations and undesirable in others; for instance, it is responsible for the majority of pressure losses but it also makes many heat transfer, mass transfer and combustion processes economically possible."

In the book Boundary Layer Theory [4], in the chapter on *Thermal boundary layers in laminar flow*, Schlichting gives the following example:

"*If we imagine a solid body which is placed in a fluid stream and which is heated so that its temperature is maintained above that of the surroundings then it is clear that the temperature of the stream will increase only over a thin layer in the immediate neighbourhood of the body and over a narrow wake behind it."*

It is clear therefore that the absence of turbulent mixing in laminar flows can result in thermal boundary layers where the temperature in a thin layer next to the pipe wall varies between the pipe wall temperature and the 'core' temperature.

This issue is of particular relevance when considering the production and transportation of heavy crude oils. Take for example transportation of Venezuelan Merey crude. Using the data from the EI database [5], the viscosity this crude oil at 20 °C is around 2,091 cSt, whereas at 40 deg °C it is around 418 cSt. For a flowrate of 2000 m3/hr in a 16-inch pipe, the resulting pressure loss is about 1.62 bar per 100 m of pipe at 20 °C, and only 0.32 bar per 100 m at 40 °C. This illustrates that for ease of transportation, it makes sense to maintain heavy crudes at temperatures above ambient.

The paper by Hogendoorn et al [6] presented at the 2009 North Sea Flow Measurement Workshop explored the issue of thermal boundary layers in laminar flow by means of computational fluid dynamics (CFD) simulations. Two-dimensional simulations were performed to reduce computational effort. The CFD simulations were used to evaluate the effects of the thermal gradients on the velocity profiles upstream of the meter and in the measurement section. These velocity profiles in the measurement section were in turn integrated according to a model of the ultrasonic meter, and the effects evaluated. It was concluded that the effects were generally small, with the extreme case of a 20 °C difference in temperature producing an error of less than 0.3 % in the meter reading. Hogendoorn's paper also included a test performed with a reasonably high viscosity oil, and a protruding gasket, the intention being that the gasket would simulate a velocity profile disturbance similar to that caused by thermal gradients. This test showed no significant impact on the meter calibration, confirming that the changes in velocity profile caused boundary layer disturbances (including thermal gradients) have little impact on some models of multipath ultrasonic meter.

However, as will be shown in this paper, changes in velocity profile are only part of the story, and as it turns out, not the most important part. Cameron's evaluation of thermal gradient effects began several years ago with a programme of laboratory testing at low Reynolds numbers, and was later supplemented this by CFD analysis. By adopting this approach it was discovered that velocity profile changes in laminar flow are of little concern, but that another physical mechanism can potentially result in significant errors due to the presence of thermal gradients.

3 PRELIMINARY INVESTATIONS INTO LAMINAR FLOW

Most available calibration facilities are not particularly suitable for testing industrial ultrasonic meters in laminar flow, particularly if a variation in fluid temperature is desirable. Take for example a 6-inch meter that we wish to calibrate over a 5:1 turndown in the laminar regime. At a nominal velocity of 5 m/s the oil would have to have a viscosity of close to 400 cSt to keep the Reynolds number below 2,000. If we wanted to achieve the same at temperature of 40 °C, the oil would have to have a viscosity of around 1,900 cSt at 20 °C. This general requirement combining availability of viscous product plus control of fluid temperature is rare.

In early 2005, NEL purchased a quantity of Primol 352 for use in their oil flow calibration facilities. This fluid initially had a viscosity of around 215 cSt at 20 °C, allowing laminar flow to be achieved at velocities below roughly 2.8 m/s at 20 °C and below 0.9 m/s at 40 °C. Although not ideal for laminar flow alone (as higher velocities are desirable), this fluid provided a good opportunity for testing through the laminar, transitional and turbulent regimes, complete with variation in temperature. Later a second product, Paraflex, was added to the range of stock oils at NEL, this oil having a viscosity of approximately 330 cSt at 20 °C, thus extending the range of tests conditions that can be obtained.

The facilities at NEL use a weighbridge as a primary standard, with PD meters used as secondary standards for the high viscosity oils. The use of two 8-inch PD meters in parallel allows maximum flowrate in the region of 500 m^3/hr when using the most viscous product, dependent of course on the test line configuration.

The majority of tests described in this paper were conducted at NEL. As the test facilities are indoor, the ambient temperature was generally in the range of 18 to 22 °C.

Figure 1 below shows the results of a calibration performed at 20 °C at NEL using the Primol product. The meter used was a standard 6-inch Caldon LEFM 240C (full bore design) with four paths. It was installed with approximately 50 diameters of straight pipe upstream and no flow conditioning. The range of velocity covered by these test results is from 0.5 to 5 m/s.

The results in Figure 1 are presented with no linearization applied to the meter. It can be observed that there is meter factor variation of the order of 0.7 % in the transition region between 2,000 and 4,000 Re, but that for Reynolds numbers below 2,000 the meter factor is constant within +/- 0.1 %. The implication of the constant meter factor below 2,000 Re is that the velocity profile is not varying significantly in the laminar regime <u>and</u> that the path velocity measurement accuracy does not vary significantly over the range of conditions in this test.

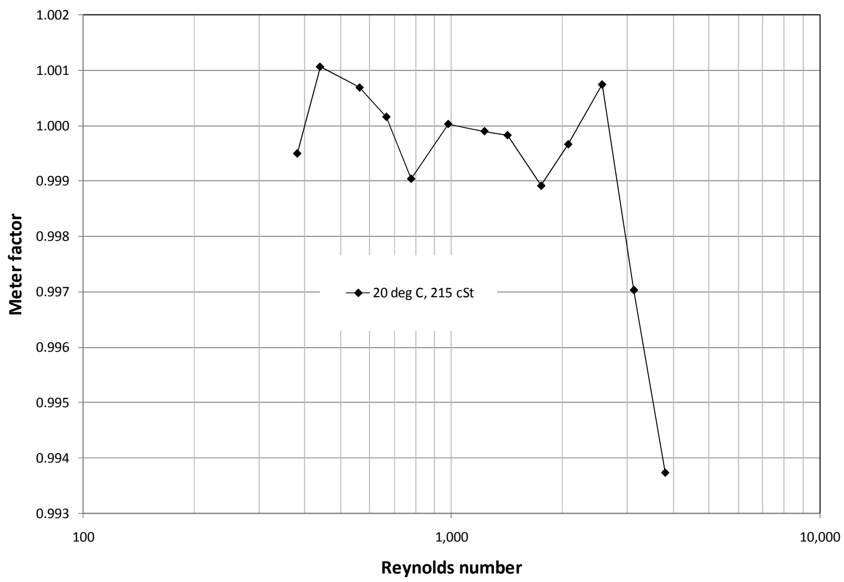


Figure 1 Calibration results for transitional and laminar flow with an oil temperature of 20 °C

Figure 2 below shows the data from Figure 1 with the addition of results obtained using the Primol product at two elevated temperatures of 30 and 45 °C. Also shown on the graph are results from a lower viscosity oil, of 22 cSt at 30 °C, used to obtain further data in the turbulent regime. From this graph it can be observed that there are differences in meter factor of around 0.5 % in the transition region between approximately 3,000 and 5,000 Re, and that in turbulent flow, above 5,000 Re, there is good agreement between the different test conditions. However, what is remarkable is the lack of agreement at Reynolds numbers below 3,000. Taking the data close to 2,000 Re, the meter factor has changed by almost 1.4 % at 30 °C and around 3.5 % at 45 °C.

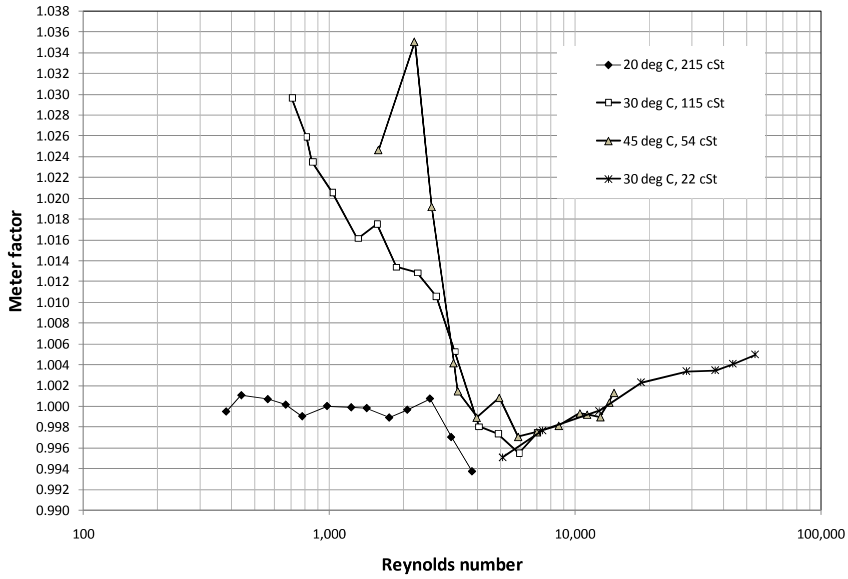


Figure 2 Calibration results showing the effects of oil temperature in the laminar regime

When the results shown in Figure 2 were first obtained, diagnostic data from the meter was scrutinised in an effort to understand what was happening. Parameters related to signal quality, such as gain and signal to noise ratio showed little change over the range of the tests. However, analysis of individual path velocity and sound velocity data produced much more informative results.

Figure 3 below shows the flatness ratio reported by the meter plotted versus Reynolds number for the Primol product at three temperatures. The flatness ratio is calculated by dividing the velocity measured on the two inside path of the 4-path meter (paths 2 and 3) by the velocity on the two outside paths (paths 1 and 4).

This diagnostic analysis clearly shows the transition from laminar to turbulent flow occurring at a Reynolds number of around 3,000. At 20 °C it can be observed that in laminar flow the flatness ratio levels out to a constant value of approximately 0.4 for Re < 1,000. However, as the temperature is increased, it can be observed that the flatness reduces, implying that either the velocity profile is changing or that somehow the path velocity measurements are being affected by the change in oil temperature.

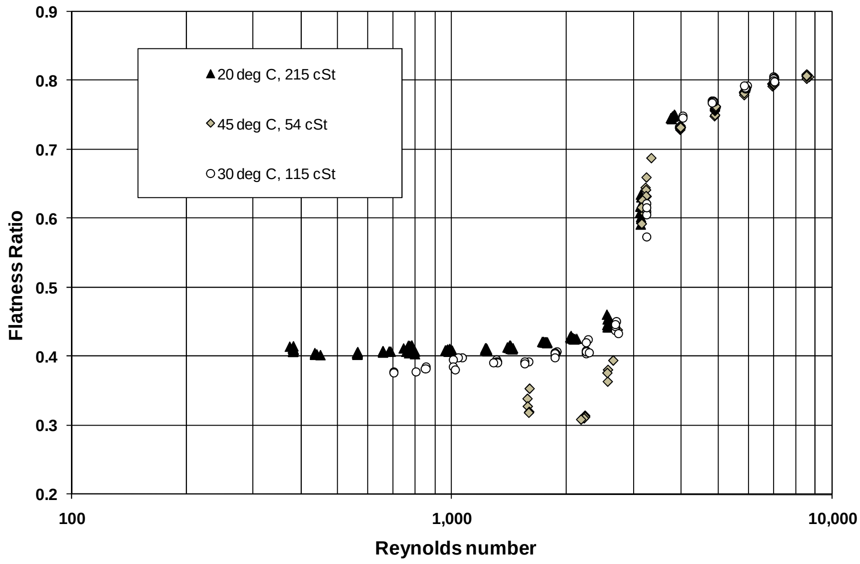


Figure 3 Flatness ratio verus Re at three temperatures

Figure 4 shows the difference between the individual values of sound velocity on each path and the weighted mean average of all four paths. From this graph it is clear that there is good agreement between the sound velocity values above 3,000 Reynolds number at all temperatures and also below 3,000 Reynolds numbers when the oil temperature was 20 °C. However, at 30 and 45 °C the outside paths (paths 1 and 4) register higher sound velocity values than the inside paths, the deviations being most significant on the bottom path, path 4.

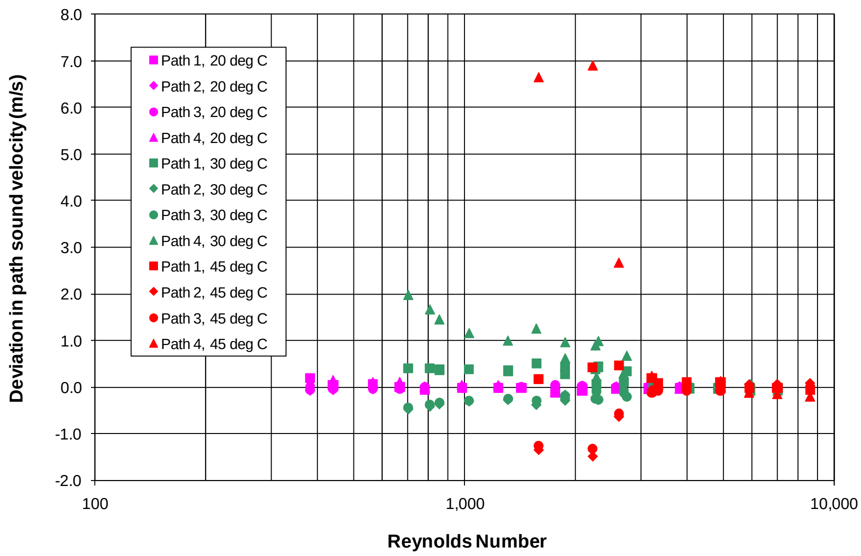


Figure 4 Relative variations in path velocity of sound readings

Taken collectively, the information in Figures 2 – 4 clearly suggest the presence of thermal gradients in laminar flow conditions.

As these results were obtained during the period when the meter design with the integrated reducing nozzle was being developed and tested, results were also obtained using a 6-inch reducing nozzle meter with a 4-inch throat. The results obtained at two temperatures, 20 and 30 °C, are shown in Figure 5 below. At 2,000 Reynolds number, the maximum difference in meter factor at the two conditions is approximately 1.4%.

These results illustrate that the magnitude of effect of the thermal gradients are very similar for the meter with the reducing nozzle and the one without. The reasons for this will be discussed later in the paper.

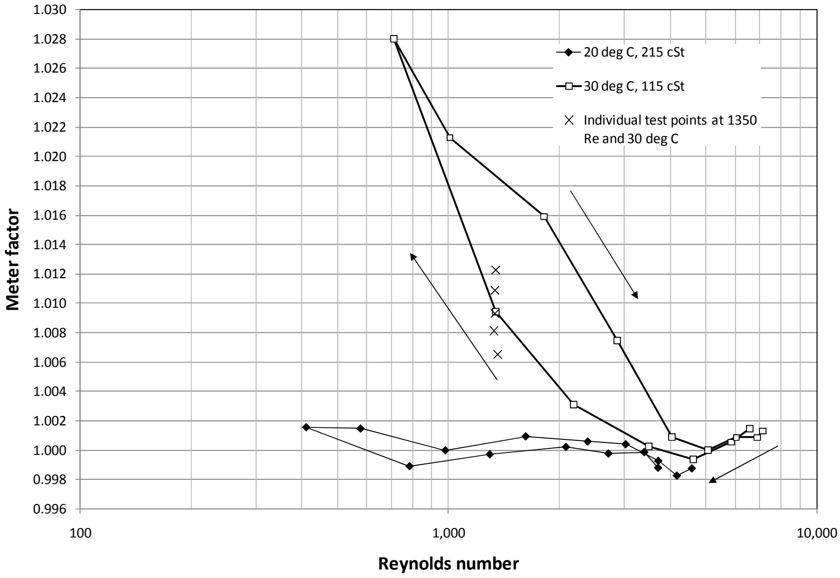


Figure 5 Thermal gradient effect on a reduced bore meter at 30 °C

A further interesting feature of the data in Figure 5, is that there is an apparent 'hysteresis' in the data. This is evidence of a 'thermal time delay' as the whole system reacts to the changing conditions. In Figure 5, the results were obtained in a calibration where the tests started at high Reynolds numbers, progressed down through the Reynolds number range, and then back up again (as illustrated by the arrows). Looking in detail at the test points obtained at 1,350 Reynolds number and 30 °C (shown in Figures 5 and 6) it can be observed that the 'spread' in meter factor at that condition is not in fact a lack of repeatability but a gradual increase in meter factor of more than 0.5% over a period of 15 minutes. This effect will also be discussed further on in the paper.

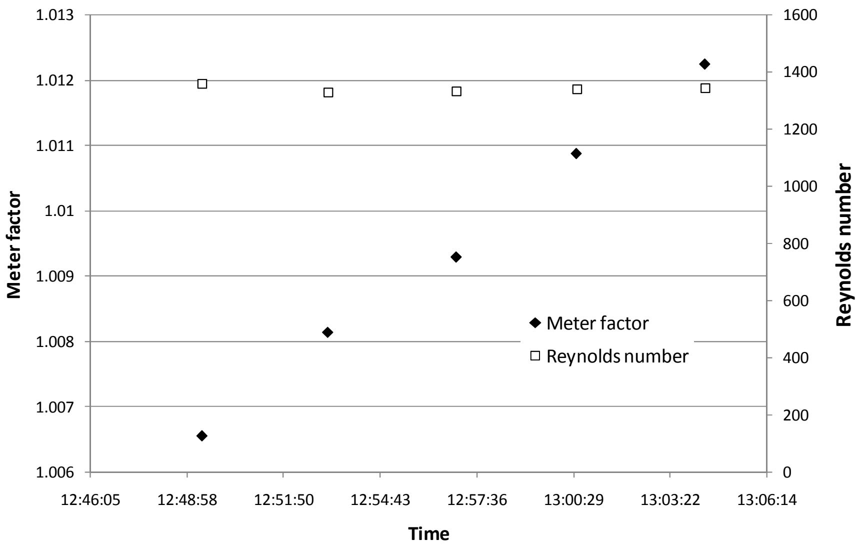


Figure 6 Variation in meter factor as a function of time at 30 °C and 1,350 Re

4 CFD ANALYSIS

After the results described in the previous section had been obtained, it was decided that computational fluid dynamics should be used to give some more insight into the problem.

The CFD modelling described here was carried out for Cameron by Neil Barton of NEL.

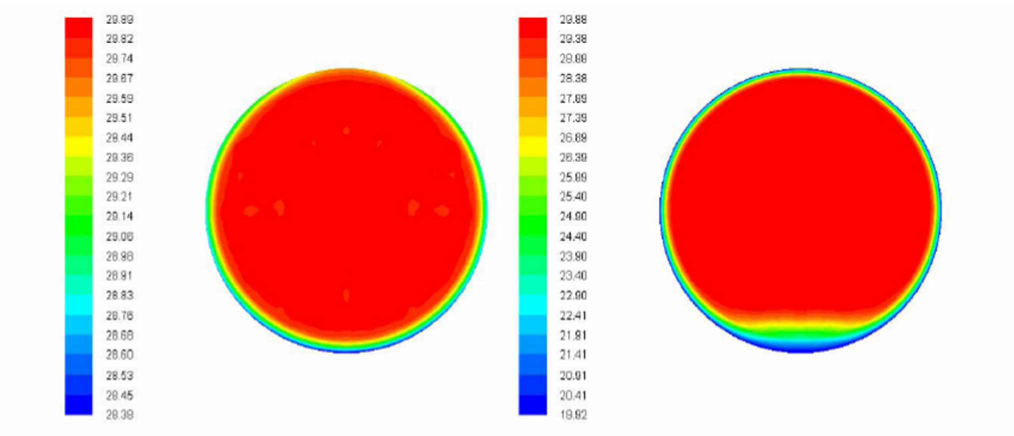
Flow was modelled in a straight pipe of 6-inch diameter, 200 diameters long. The oil temperature at the inlet to the pipe was set at 30 °C, and the density and viscosity properties of the oil versus temperature were matched to the Primol test oil (approx 860 kg/m^3 and 115 cSt at 30 °C). The simulation flowrate was set to approximately 23 kg/s, such that the resulting Reynolds number at the inlet would be close to 2,000. Further details are given in the NEL report [7].

Two different cases of heat loss were modelled. In the first case the pipe wall was cooled by convective heat loss, with the surrounding air temperature set to 20 °C. In the second case the temperature of the pipe wall was set to 20 °C. This second case represents a very extreme situation, which is highly exaggerated with respect to what is likely in practice (unless of course the differential between ambient and oil temperature is much much higher).

There are some differences in scope between the CFD analysis carried out by NEL and that described by Hogendoorn et al [6], as outlined in the table below. However, as will be shown these differences are not believed to have a significant impact on the general conclusions that can be drawn from the CFD analysis.

	NEL/Cameron	Hogendoorn et al
Temperature differentials	Case 1 – convective heat loss, fluid: 30 °C, ambient: 20°C Case 2 – fluid: 30°C, pipe wall: 20°C	Fluid: 35 °C Pipe wall: 15, 30, 35, 40 and 55 °C
Buoyancy effects included	Yes	No
Viscosity at 20 deg C	215 cSt	400 cSt
Reynolds numbers	2000	100, 500, 1500
Pipe diameter	6-inch	4-inch
Downstream location of meter	From 5 – 200 D, in 5 D steps	50 D
Meter design	4-path, full bore	5-path, reduced bore

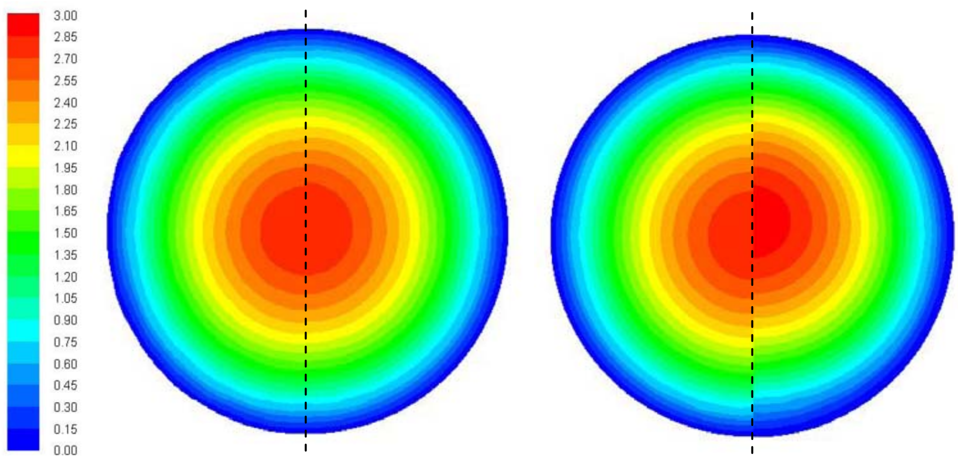
Figure 7 below shows colour plots of the temperature distribution inside the pipe at 200 diameters downstream. The effects of buoyancy are apparent in both plots. In the left hand figure, the convective heat loss creates a ‘hot spot’ at the top of the pipe where the heated air is rising from around the pipe. In the right hand plot, the higher density of the cooled liquid tends to lead to stratification at the bottom of the pipe. It should be noted that the colour scales are different on the two temperature plots, the span being 10 °C in the right hand plot and only 1.4 °C in the left hand plot.



(a) Convective heat loss (b) T_{wall} = 20 °C

Figure 7 Temperature contours at 200 diameters downstream

Figure 8 shows velocity profile colour contour plots at 200 D downstream. In each case the image is split in two, with the left hand half showing fully developed laminar flow and the right hand half showing the velocity profile for each thermal condition that has been modelled. From these images it is clear that there has been relatively little effect on the velocity profile in the convective heat loss case. In the case of the wall temperature set at 20 °C, the most obvious effect is the reduction in velocity towards the bottom of the pipe.



(a) Convective heat loss (b) T_{wall} = 20 °C

Figure 8 Velocity profile contours at 200 diameters downstream

Figure 9 shows the results of integrating the velocity profiles according to the design of a 4-path Gauss-Jacobi meter design, for each of the heat transfer cases, and comparing that with the results from fully developed laminar flow. It can be observed that in the case of convective heat loss, there is very little influence on the meter, with the shift reaching a maximum of about -0.025 % at 200 D. In the more extreme case where the wall temperature was set to 20 °C, the shift tends to a maximum of around -0.2 % at around 100 D. Note that the 'spikes' in the CFD results do not represent what is happening physically, but are a result of uncertainties in the modelling process.

Whilst we have to recognise that there are differences between this CFD work and that of Hoogendorn et al, the broad conclusion that can be drawn from both studies is that the changes in velocity profile due to thermal gradients should not affect the calibration of multipath ultrasonic meters by more than a few tenths of a percent. The question then is: what is the cause of the differences observed in the test results that are an order of magnitude greater?

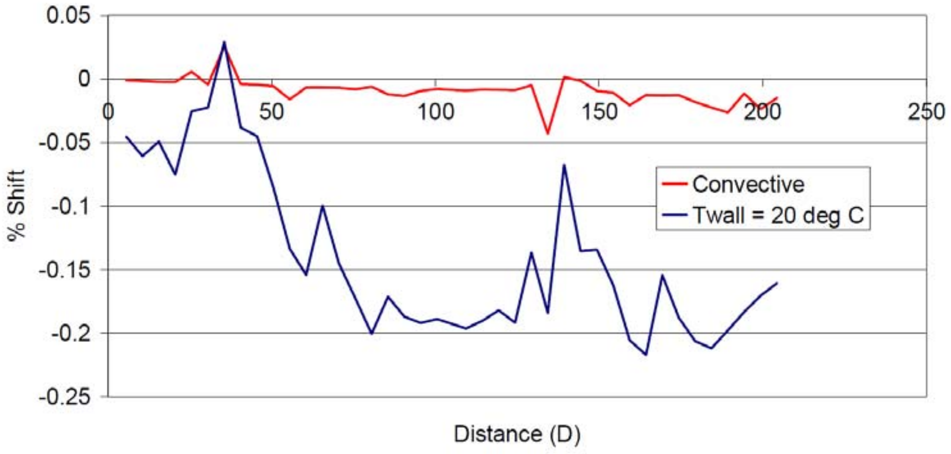


Figure 9 Simulation of the temperature related velocity profile effects on a 4-path meter

5 A MECHANISIM FOR THE EFFECTS OF THERMAL GRADIENTS

With the benefit of hindsight, the cause of the large differences in meter factor that occur when thermal gradients are present is rather obvious. The geometry assumed for the paths of an ultrasonic meter connect the transducer centres along a straight line, with a particular angle to the pipe axis. When thermal gradients are present, this is no longer true, as the ultrasound undergoes refraction as it passes through the sound velocity gradient that is also present as a consequence of the temperature gradient.

Consider what happens when the thermal gradient is present inside the meter body itself. In laminar conditions, the fluid in the recesses or cavities in front of the transducer housings is trapped and is either stagnant (at very low Re) or recirculated within the cavity. As such it will eventually take on the temperature of the oil closest to the pipe wall. The result (in the case of hot oil and a cooler ambient temperature) is that the ultrasound must pass from a region of higher sound velocity to a region of lower sound velocity.

With reference to Figure 10 below, Snell's law of refraction for acoustic waves can be written as

$$\frac{Cos\theta_i}{c_i} = constant = \frac{Cos\theta_1}{c_1} = \frac{Cos\theta_2}{c_2} = \frac{Cos\theta_3}{c_3} = \frac{Cos\theta_4}{c_4} \ldots$$

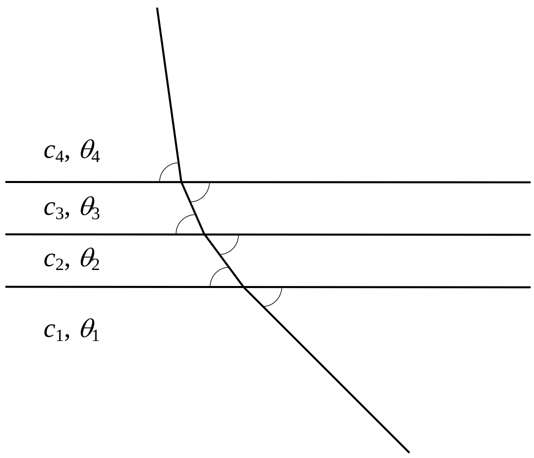


Figure 10 An illustration of Snell's law of Refraction

What this relationship shows is that owing to the transducer face being at an angle to the pipe axis (typically 45°), it must undergo refraction as it passes through the thermal gradient, which will be roughly parallel with the pipe wall. Snell's law also illustrates that is not important that the change in velocity of sound occurs as a gradient rather than a sudden change, the overall change in angle is only dependent on the sound velocity on either side of the layer.

In reality the effects are quite complex, particularly when the three-dimensional geometry of the transducer cavities are considered, as pictured in Figure 11. The refraction effect means that the effective location of the path with change both in terms of its angle to the pipe axis, and its lateral offset (height) in the cross section.

Figure 11 A photograph showing the complex geometry of a transducer cavity

In order to illustrate the effect with a very simple example, consider an ultrasonic wave that meets a change in sound velocity at an angle of 45º. If the sound velocity on the 'cold' side the interface is 1441 m/s and on the 'warm' side it is 1434 m/s, then the path angle would change by approximately 0.3º. While this might not appear to be a large change in angle, corresponding change in the transit time difference would be almost 1%.

The values of sound velocity in the above example correspond to a change of roughly 2 ºC in oil temperature. This illustrates that refraction is a physical mechanism that can easily explain effects of the magnitude seen in the experimental data.

Understanding that it is the mechanism of refraction that is responsible for meter factor changes in laminar flow also explains why the meter design with the reducing nozzle has a response that is very similar to the full-bore meter. Although the thermal boundary layer will be 'squeezed' as it goes through the reducer, ultrasound still has to pass from one side of the layer to the other, and the effects of refraction will be virtually the same.

The refraction mechanism also explains the thermal time delay effects shown in Figures 5 and 6. Consider that in turbulent flow the fluid is well mixed and the temperature is the same in the transducer cavities and in the bulk flow. If the ambient temperature is lower than the oil temperature when the flow then drops into laminar conditions, the fluid closest to the pipe wall begins to cool (and also flows downstream). This creates a layer of cool oil that encloses the fluid in the transducer cavity. It then takes some time for the thermal boundary layer to thicken and fluid trapped in the cavity to cool to the outside temperature of the boundary layer, by which time the refraction effect reaches its maximum.

6 CONDITIONS AFFECTING THE DEVELOPMENT OF THERMAL GRADIENTS

The data in Figure 9 above shows that thermal gradients develop gradually downstream of a point of where the fluid is isothermal. In practice this means that if the meter is a short distance from this point, or if there is repeated mixing of the fluid, then the effects on the meter may be less significant. So in some applications significant thermal gradients may develop and in others they may not. At present it is not easy to judge when thermal gradients will be a problem or not, as it is dependent on many factors including the pipe geometry, the fluid and ambient temperatures, the meter design, and potentially other factors such as velocity or Froude number.

Figure 12 shows results from a 12-inch meter with reducing nozzle tested at NEL over a range of temperatures [2]. Based on the earlier results shown in Figure 5, it might be expected that at the lowest Reynolds numbers in this test (below 3,000 Re), there would be some evidence of the detrimental effects of thermal gradients. However, in this case there was only a slight suggestion of thermal gradients in the diagnostic data. In all probability the lack of effects in this case was due to the fact that there was only a relatively short length of around 20 diameters of 12-inch pipe upstream and prior to that the flow came through an complex series of combining and dividing flow and bends, all in 8-inch diameter.

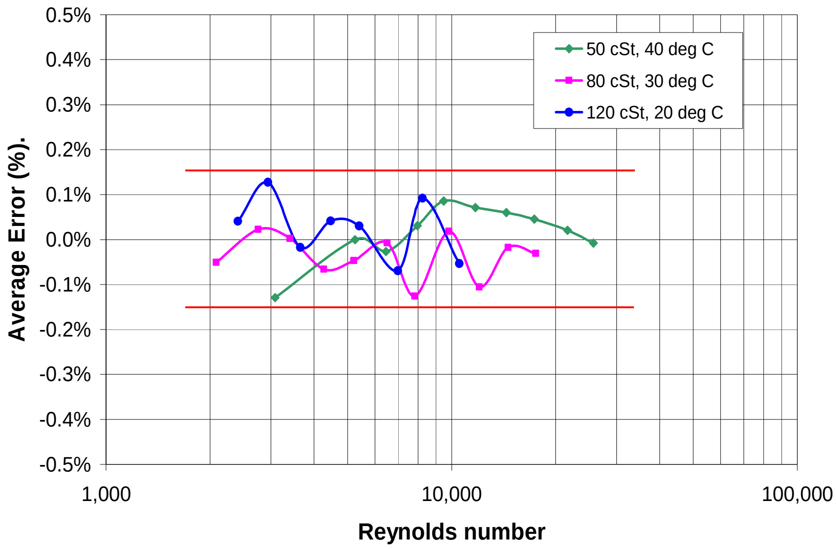


Figure 12 An example of calibration results absent of the effects of thermal gradients

Figure 13 on the other hand shows data for a 12-inch meter calibrated at SPSE in France under two different sets of conditions. The test lines are outside at SPSE and the tests were conducted in the months of December and January with the result that the ambient temperatures were low. Two different oils were used and in one instance the oil temperature was in the range of 10 – 17 ºC, and in the other case the oil temperature was in the range of 28 to 35 ºC. The data at Reynolds numbers between 4,000 and 6,000, showed a lack of reproducibility and clear evidence of the effects thermal gradients in the diagnostic data. This case is interesting as the line set up at SPSE is quite similar to many metering installations, comprising a header upstream of two parallel test lines.

Although laminar flow would not normally be expected at the relatively high Reynolds numbers shown in Figure 13, the meter diagnostics such as the flatness ratio clearly showed that the flow was laminar below 4,000 Re. In this case it appears that the heat transfer is also playing a part in where the transition occurs. It is stated by Schlichting that “the transfer of heat from the boundary layer to the wall exerts a stabilising influence by causing the critical Reynolds number to increase” [4], i.e. transition and hence laminar flow can occur at higher Reynolds numbers when heat loss through the pipe wall is present.

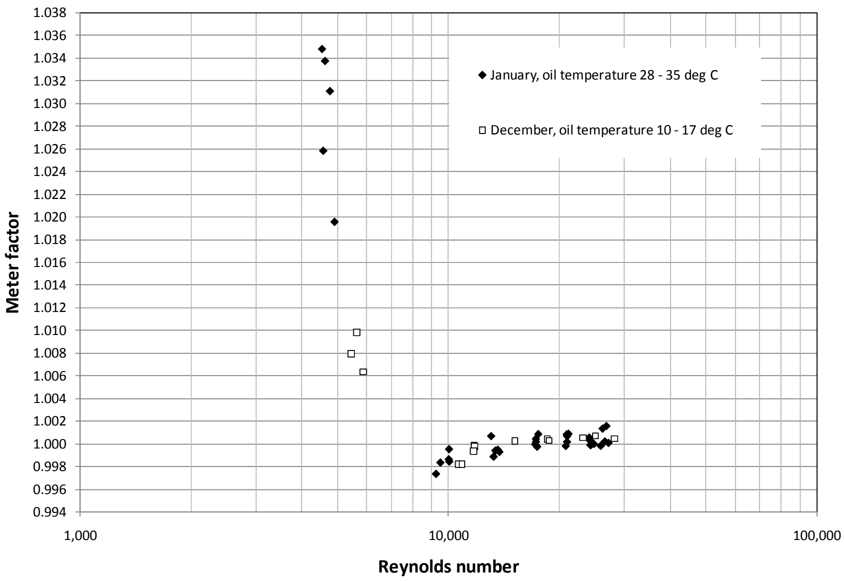


Figure 13 Thermal gradient effects observed during calibrations at SPSE

7 REDUCING THE IMPACT OF THERMAL GRADIENTS

7.1 Experimentation

Given the understanding that the effects observed experimentally are a result of thermal gradients and the resulting refraction of the ultrasonic beams, the question then is find ways to reduce or correct for these effects. This is an area in which Cameron have carried out a considerable amount of experimental work in recent years.

Initial experiments were designed to investigate various effects included testing meters with paths vertical rather than horizontal, testing with insulation over part of the installation, and testing with various conventional flow conditioners and mixers. A sample of the results of these tests are presented below.

<u>*Paths orientated vertically*</u>

Given that the experimental results showed evidence of buoyancy effects/stratification, a test was conducted where the flowmeters were reorientated such that their paths were vertical rather than horizontal. The tests were conducted without insulation or flow conditioning and the results for the two orientations compared under nominally identical test conditions. The results are shown in Figure 14 below for a 6-inch full-bore Caldon 240C ultrasonic meter. It can be observed that orientating the paths vertically produces virtually the same results in terms of the effect of temperature on the meter factor. This is due to the fact that the thermal boundary layer extends around the entire internal circumference of the pipe.

Similar results to those shown in Figure 14 were also obtained when the same test was performed on the meter design with reducing nozzle.

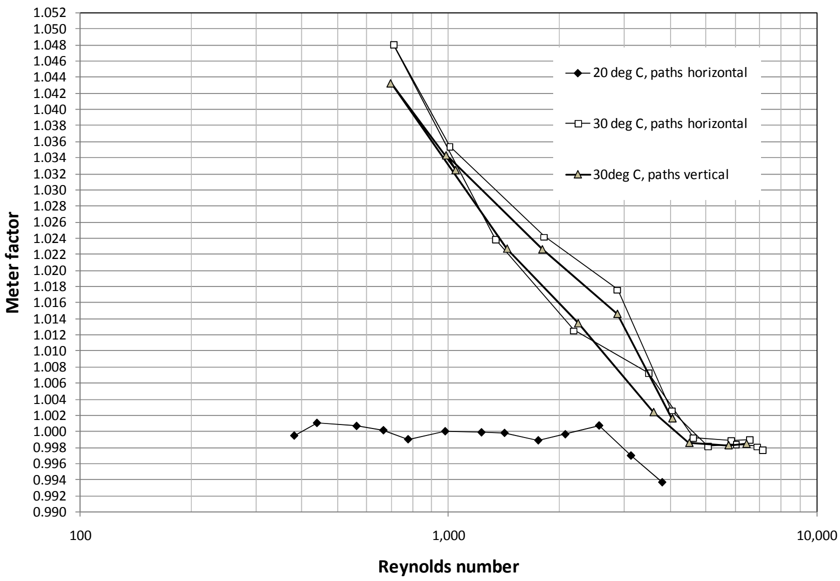


Figure 14 Results obtained with paths horizontal and vertical

Meter body and 20 diameters of pipe insulated

An obvious question to ask is: what is the effect of applying insulation to the upstream pipe and meter body? To answer this question a test was performed with approximately 70 diameters of straight pipe in total upstream, with 150mm glass mineral wool insulation applied to 20 diameters upstream of the meter and the meter itself.

The results are shown in Figure 15 below, for a meter of the reducing nozzle design (6-inch meter with a 4-inch throat). In this case the insulation was found to have a beneficial effect, reducing the overall magnitude of meter factor variability.

However, the insulation was certainly not 100 % effective. This can be explained if we consider what insulation can and cannot do. It can minimise heat transfer between the pipe wall and the outside environment, but it cannot prevent heat transfer to or from the fluid itself. If the pipe wall is at the fluid temperature in turbulent flow, then when the flow changes to a laminar condition the uninsulated pipe will begin to cool. The cool boundary layer will flow downstream into the uninsulated section and will itself have a cooling effect on the pipe wall and meter body under the insulation. In fact, it is possible to imagine that given sufficient time the temperature under the insulation could equalise with the temperature of the exposed pipe.

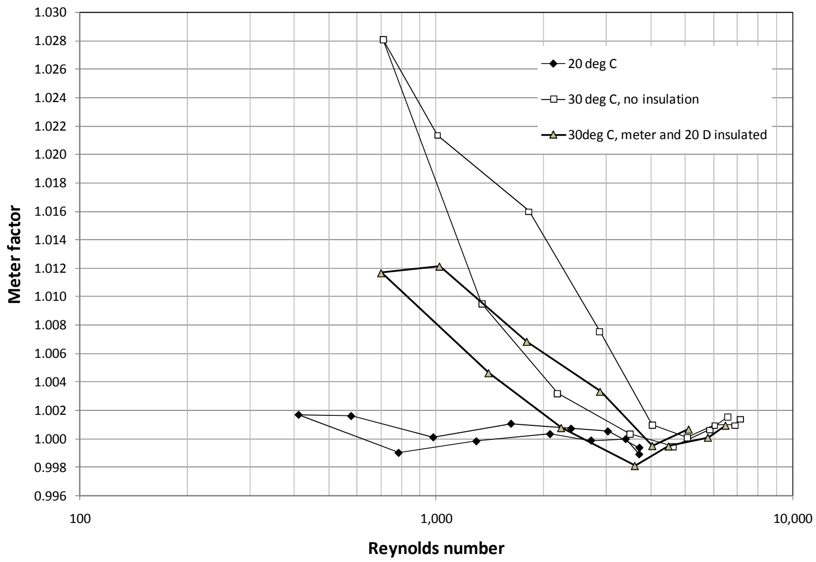


Figure 15 Results obtained with and without insulation

<u>Tests with a Spearman perforated plate flow conditioner</u>

Tests were carried out with various conventional forms of flow conditioning and mixing devices. Encouraging results were sometimes obtained by using two different devices in series, for example a mixer followed by a flow conditioner, but with the penalty of high pressure loss. Of the conventional flow conditioners, one which produced a fair degree of improvement on its own was a Spearman perforated plate. The plate was used at various distances from the meter, with similar results being obtained with the conditioner at distances of between 3 and 10 diameters upstream of the meter.

Figure 16 shows the results of tests with the Spearman plate just 3 diameters upstream of a meter with reducing nozzle, again with oil temperatures of 20 and 30 ºC. It can be observed that the results are in very close agreement down to a Reynolds number of around 3,000 and then there is some divergence. However, at a Reynolds number of 1,000, the difference in variation is meter factor is only about 0.5 %, as opposed to more than 2 % for the same meter with no flow conditioning.

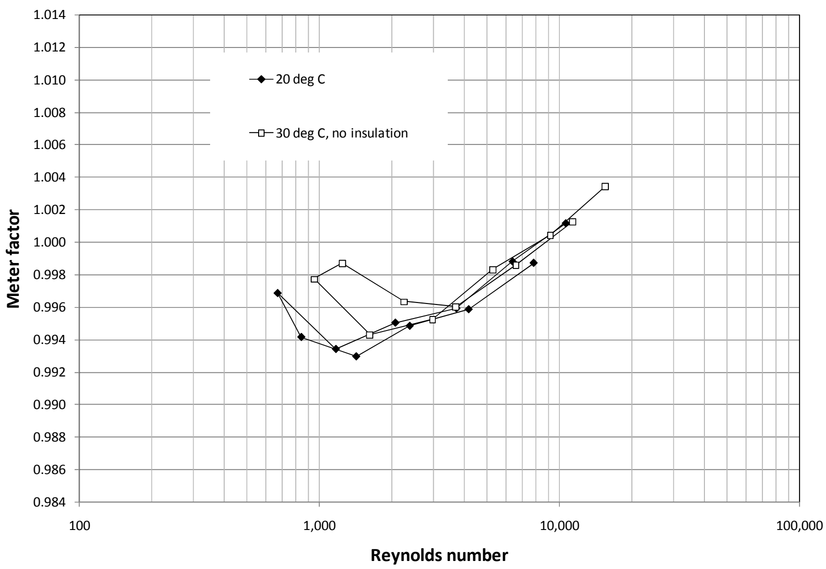


Figure 15 Results obtained with the Spearman plate for 20 and 30 °C

In general it was true of the flow conditioning devices tested that improvements were made but that below 1,000 Reynolds number the uncertainty tended to exceed the bounds required for custody transfer.

Empirical corrections applied in the meter

Analysis of various data sets showed that a meter could be calibrated or 'tuned' to reduce the effects of thermal gradients over a range of test conditions. However, the basis of such corrections is empirical and as such the applicability of the corrections outside of the specific calibration conditions would be questionable. For that reason it was deemed preferable to pursue a flow conditioning based solution.

7.2 Development of a New Design of Flow Conditioner

Following the period of experimentation described above, consideration was given to the design of a flow conditioner that would address the specific problem of thermal gradients in laminar flow. Various ideas were reviewed and then discarded as either as being ineffective or impractical. Eventually the idea was put forward of a device that would use ramps to direct the thermal boundary layer away from the pipe wall, whilst simultaneously directing fluid from beyond the boundary layer towards the pipe wall to replace the fluid displaced from there. A practical design based on alternating ramps extending from a tubular body of diameter smaller than the pipe internal diameter was then devised.

The photograph in Figure 17 shows a prototype of the 'laminar boundary layer flow conditonner' that comprises two sets of ramps in series, providing additional displacement and mixing of the fluid close to the pipe wall. This is the device that was used to obtain the test results described below.

The prototype laminar flow conditioner was tested at NEL using their Paraflex oil. The meter used in the test was a 6-inch 8-path full bore meter installed with approximately 80 diameters of straight pipe upstream. For the first set of tests the flow conditioner was not installed and the upstream pipe and flow meter were not insulated. For the second set of tests, the conditioner was installed 10 diameters upstream of the meter, and the upstream pipe was insulated with 150 mm glass mineral wool wrapped round the pipe from the mixer element to the meter, and including the meter itself. The purpose of the insulation was to prevent redevelopment of thermal gradients between the conditioner and the meter.

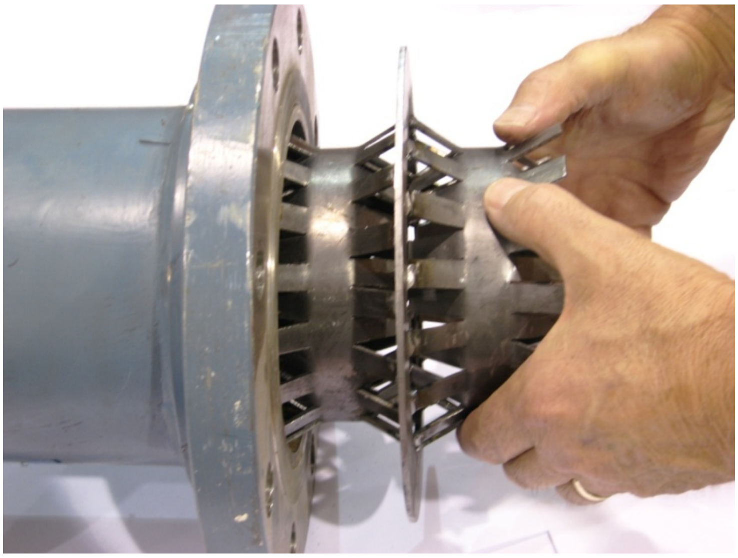


Figure 17 A photograph of the prototype laminar boundary layer flow conditioner

The two graphs shown below summarise the performance in terms of meter factor versus Reynolds number for the meter through laminar, transitional and turbulent flow, with and without the mixing device.

Figure 18 shows the behaviour of the meter without the flow conditioner in the line. This data is presented with no linearization applied in the meter. It can be observed that below around 3,000 Re, the calibration curves diverge as a function of temperature. It can also be observed that hysteresis is observed before below 3,000 Re, particularly at the elevated temperatures. As discussed earlier, the hysteresis is caused by the system not being at thermal equilibrium, and therefore meter factors change progressively over time as the fluid close to the pipewall and in the transducer cavities either heats or cools.

It can be observed that without the mixer the deviation between the 20 °C condition and the 40 °C condition at 800 Re, is approximately 2.7%.

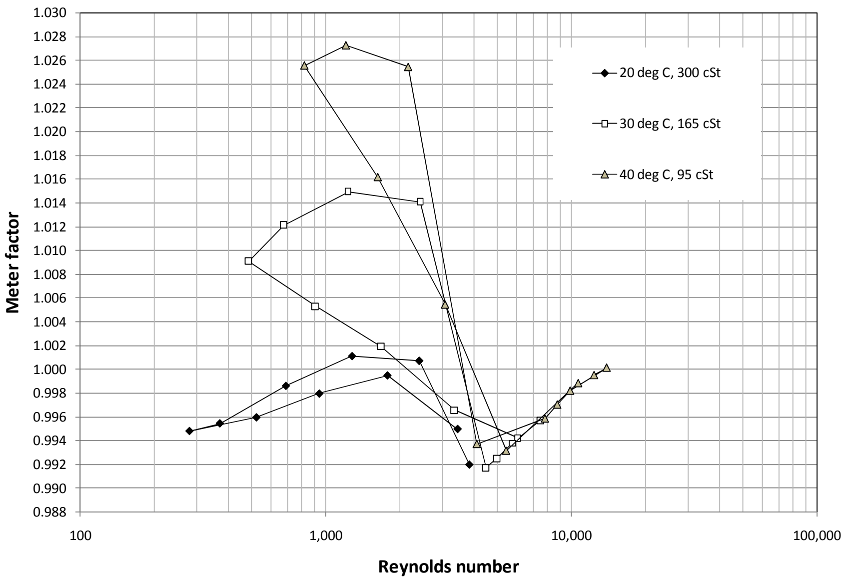


Figure 18 Full-bore 6-inch 8-path meter without flow conditioner (raw meter factor)

Figure 19 below shows the performance with the flow conditioning device installed 10 diameters upstream, and with the meter and pipework between insulated. This data is also presented with no linearization applied in the meter. It can be observed that the line of the turbulent flow meter factor now continues down to around 2,500 Reynolds number, where transitional behaviour was previously seen between 3,000 and 4,000 Re. As might be expected with a full bore meter design, there is a sharp change in meter factor in the transition between laminar and turbulent flow centred at around 2,000 Re. In applications that span the transition, this sudden change can be smoothed out by use of a reducing nozzle [2].

The benefits of the flow conditioning device can be observed in the improved reproducibility of the meter factor in the laminar region below 2000 Re. It can be seen that the maximum deviation in that region is now within +/- 0.15 %, i.e. there is an order of magnitude improvement. It is also useful to note, that above 5,000 Re, in the turbulent regime, the meter factor has the same value with and without the flow conditioner.

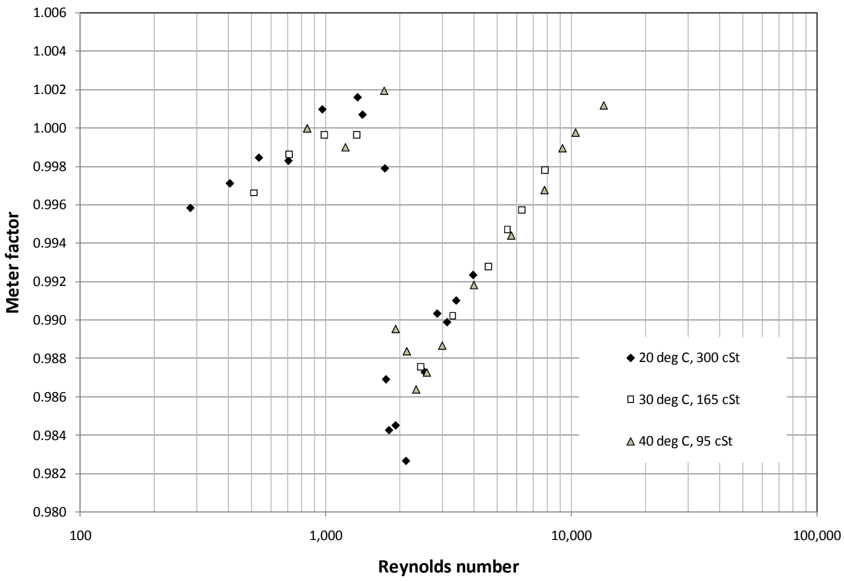


Figure 19 Full-bore 6-inch 8-path meter with laminar flow conditioner (raw meter factor)

The results shown in Figures 18 and 19 demonstrate that this conditioner has been effective at reducing the effects of thermal gradients to a sufficiently low for level custody transfer applications. Preliminary analysis shows that this improved benefit is obtained with significantly lower pressure loss than for a perforated plate conditioner. Further work is still required to optimise the design both in terms of its effectiveness and its pressure drop.

8 DISCUSSION AND CONCLUSIONS

The experimental data presented in this paper clearly demonstrates that thermal gradients in laminar flow can have significant effects on the performance of ultrasonic flow meters. The possibility of thermal gradients being present in the flow arises whenever the flow is laminar and there is a differential between the fluid and ambient temperatures.

The use of a meter design incorporating a reducing nozzle or convergent section is beneficial in terms of transitional flow and the generally ability to cope with higher viscosities but does not help to reduce the impact of thermal gradients.

The primary mechanism by which thermal gradients affect the performance of ultrasonic meters is not velocity profile, it is refraction, resulting from the fact that the ultrasonic paths must cross a sound velocity gradient.

CFD analysis has proven to be a useful tool for aiding understanding of the formation of thermal gradients, but has not been effective in predicting their effect on ultrasonic meters, owing to the fact that the effects of refraction have not been accounted for in the modelling process.

In practice the severity of thermal gradients, and hence the resulting effects on ultrasonic meters will be a function of many variables including:

- Oil properties
- Reynolds number and velocity
- Pipe diameter
- Fluid and ambient temperatures
- The quality and extent of insulation
- The effects of the upstream pipe configuration, including bends, flow conditioners, pumps and valves

It has been shown that insulating the pipe upstream of a meter can have beneficial effects, but is not sufficient to ensure custody transfer levels of uncertainty if thermal gradients are established in the pipe upstream of the insulation.

It has also been demonstrated that use of a perforated plate flow conditioner can reduce the impact of thermal gradients, but that at lower Reynolds number in the laminar regime, the effect on the meter factor can still be significant.

Tests of a prototype laminar boundary layer flow conditioner have demonstrated that it is effective in reducing the effects of thermal gradients to level consistent with the requirements for custody transfer. Even before the prototype has been optimised, this benefit is achieved with lower pressure loss than a conventional perforated plate flow conditioner.

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