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A NEW FLOW CONDITIONER FOR 4-PATH ULTRASONIC FLOWMETERS

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Abstract

There are benefits, limitations and trade-offs to be taken in to account when considering using a flow conditioner with an ultrasonic meter. These include aspects such as the pressure loss, the risk of fouling and the influence on meter repeatability, as well as the ease and reproducibility of manufacturing.

This paper discusses different flow conditioner designs and introduces a new design intended for use with 4-path ultrasonic flowmeters. Performance data and permanent pressure loss measurements are presented and compared with a commonly used form of perforated plate flow conditioner. It is shown that comparable flowmeter performance can be achieved while reducing the pressure loss significantly.

Keywords: conditioner, straightener, ultrasonic, multipath, flowmeter

Background

In the field of flow measurement it is often necessary to condition the flow upstream of a flow metering device in order that the flow meter will register flow with a minimal error. Bends, valves, filters and other forms of pipeline component distort the flow velocity profile and by changing the flow direction introduce non-axial velocity components or ‘swirl’ in the flow stream. It is well known that the calibration or flow coefficient of certain types of flow meters are affected by distortions of the profile and/or by the presence of swirl.

Flow conditioners have been employed for many years to partially rectify distorted and swirling flows upstream of flow meters. The various devices deployed to date differ in design with resulting differences in performance in terms of their ability to rectify flow versus the permanent pressure loss that they impose. Most conditioners have a single specified geometry or a constrained set of design parameters and cannot easily be adapted to suit the requirements of a particular situation.

Multipath ultrasonic meters are now widely employed for measurement of the flow of oil and natural gas. Although some models of ultrasonic flow meter, such as 8-path chordal meters, are able to perform accurately without the need for a flow conditioner, meters with three, four and five paths are typically used with flow conditioners when deployed in custody transfer applications. Most of the devices used with ultrasonic meters today were designed for use with turbine meter or orifice plates. Common examples are tube bundles and thick perforated plates. While these conditioners are generally suitable for use with ultrasonic meters, none of them were specifically optimised for this purpose and there are trade-offs to be taken into account during selection.

This paper discusses different flow conditioner designs currently used with ultrasonic meters and introduces a new design specifically intended for use with 4-path ultrasonic flowmeters and optimised for that purpose.

Flow Straigheners

By far the most common type of flow conditioner has been a ‘flow straightener’ of either the vane type or in the form of a tube bundle assembly. Flow straighteners essentially divide the flow into a number of passages that are long and straight and parallel with the axis of the pipe. The aim is that any rotational component of velocity is reduced or eliminated before the flow exits the conditioner.

The tube bundle is the most commonly employed form of flow straightener, having been standardised to some degree, and is essentially an assembly of tubes, typically between 7 and 55 in total, arranged either in a hexagonal or circular geometry, as illustrated in Figure 1. A tube bundle using 19 tubes of equal size arranged in a circular geometry is included in the International Standard for differential pressure flow meters, ISO5167 [1]. Tube bundles are typically made to be between two and three pipe diameters in length, with the result that the tubes may be 20 to 30 tube diameters long, though studies have shown that in terms of limiting swirl, a much shorter length of bundle can still be effective [2].

A recognised deficiency of the tube bundle design of flow conditioner is that while it is effective at removing swirl, the emerging axial velocity profile does not tend to be fully developed; that is it generally tends to be flatter than the profile that would be found downstream of a long straight length of pipe at the Reynolds number of interest. In order to try to overcome this limitation, Stuart [3] developed a tube bundle flow conditioner where the tube diameters used within the bundle were varied in order to produce a velocity profile shape closer to the desired fully developed profile. A disadvantage of this conditioner design in terms of manufacturing, which also applies to most tube bundle designs, is that when the pipe diameter is varied, the required tube diameters may not be readily available in standard sizes of tubing. The main advantage of tube bundles is that they have relatively low permanent pressure loss, having a loss coefficient for fully turbulent flow in the range of 0.65 to 1.2; see for example Baker [4].

A further disadvantage of the tube bundle is its variable design and quality. If not constructed to the ISO standard, the potential variations in number and size of tubes are almost endless, making it difficult to predict performance or relate experience from one design of tube bundle to another. Furthermore, variable manufacturing quality means that the tube alignment may vary, and in some cases, for example if the bundle becomes twisted during manufacture, the bundle can actually generate a swirling flow.

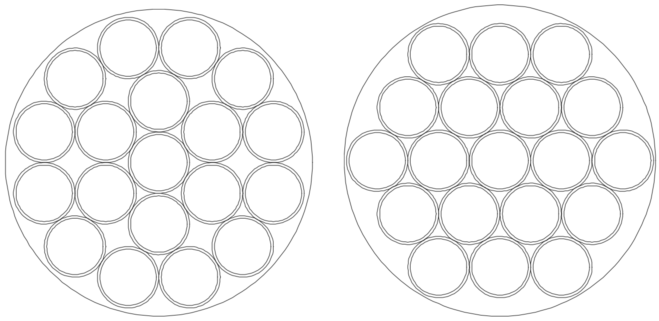

Fig. 1 Two typical arrangements of 19-tube tube bundle conditioners

Flow Conditioners

The need to shape the axial velocity profile as well as remove swirl was first addressed properly in the design of the Zanker flow conditioner [5]. The original Zanker conditioner comprises a thin plate with holes designed to produce a graded resistance to flow combined with a vane type straightener attached to the downstream side of the plate. In terms of the flow profile produced and level of swirl reduction achieved by this conditioner, it is recognised as being very effective. However, it is somewhat difficult to manufacture and has a high pressure loss coefficient of greater than 5.

More commonly used today are thick-plate type conditioners. In these designs a graded resistance to flow is achieved by means of making circular passages in a fairly thick plate. By varying the number, spacing and size of the circular passages, the desired graded resistance is achieved. Examples of this type of conditioner include those by Laws (most common in the Nova/CPA 50E variant) [6, 7], Spearman [8] and Gallagher [9], in addition to the thick plate version of the Zanker conditioner [10], where the thicker plate negates the requirement for the downstream vane-type straightener. Common thick-plate conditioners are illustrated in Figure 2.

These thick-plate conditioners with circular passages are considered the current state-of-the-art but still have certain deficiencies. Pressure loss coefficients are typically in the range of 2 to 5, greater than that available with a tube bundle. Attempts to produce plates of higher porosity and hence lower pressure loss have generally resulted in a reduction in flow conditioning performance [11].

Optimisation of the design of a thick-plate conditioner with circular passages is also complicated by particular issues associated with the chosen circular hole geometry. An irregular number of holes and the circular shape of the passages result in a complex 'water-shed' between adjacent rings of holes, and hence make the calculation of the effective porosity in different areas difficult, as the water-shed defines the blockage area associated with each hole. Optimisation is further complicated in cases where the circular passage size is varied, as for a given thickness of plate as this results in variation of both the porosity and the ratio of the length of the passage to its hydraulic diameter. As a consequence, the steps that must be taken to optimise a conditioner with circular passages are not obvious, as when changes are made the shape of the water shed varies as well as the porosity and the length/hydraulic diameter ratio. This likely explains some of the observations made by Karnik [7], where in some cases small changes in hole size resulted in significant variations in performance, but for other cases it did not.

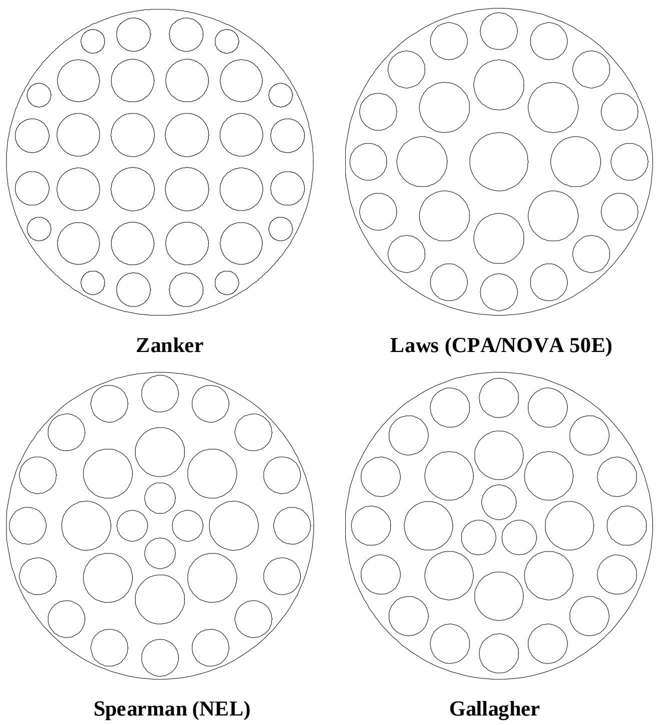

Fig. 2 Typical thick plate conditioner geometries

A particular advantage of the thick-plate conditioner is that the manufacture and geometric scaling to different sizes of pipe can be achieved very easily, which overcomes the manufacturing and quality limitations associated with the tube bundle type of conditioner.

As mentioned previously, the effectiveness of thick-plate conditioners has been found to diminish when the porosity is increased too much, with the result that most thick-plate conditioners in use today have porosity in the region of 50 %. When porosity has been increased, the investigators have not tended to increase plate thickness to compensate for the reduction in the length/depth of the hole relative to its hydraulic diameter, which may partly explain the diminished performance. This has led some designers to add straightening vanes to the conditioner or to employ two stages of conditioning, the first being a straightening vane and the second a graded thick-plate conditioner [9, 12].

A New Conditioner Design

Some types of flow meter are more sensitive to the condition of the incoming flow velocity field than others. In the case of multi-path ultrasonic flow meters, it is often the case that if swirl is removed effectively then the meter will be able to perform with high accuracy in a variety of different installation conditions. Therefore it is common for tube bundles to be used with ultrasonic meters, owing to their lower pressure loss characteristics. However, this does not offset three of the disadvantages of tube bundles: first, that they alter the axial velocity profile in an adverse way; secondly the fact that they are generally manufactured to be between 2 and 3 diameters long, and thirdly the manufacturing issues mentioned above that can result in poor quality conditioners. To overcome these limitations the intention of the work described here was to produce a new low pressure loss flow conditioner for use with ultrasonic flow meters. In addition to having a low permanent pressure loss, the requirement was set that conditioner should be easy to manufacture in a reproducible way and that it should be possible to obtain a desirable shape of axial velocity profile.

The new flow conditioner design is based on an arrangement of segmented annular passages, arranged symmetrically around the centre-line of a circular conduit as illustrated schematically in Figure 3. The choice of segmented annular passages allows the cross-sectional area of the pipe to be easily divided into a predetermined number of annular rings with the width and separation of the passages to be freely varied in both radial and tangential directions to obtain a desired value of hydraulic diameter and porosity in each ring. Combined with control over the length of the passages via selecting the thickness of the conditioner, this arrangement of segmented annular passages can be optimised to produce a conditioner that will retard swirl *and* have a desired radial distribution of resistance, in combination with a specified overall pressure loss.

Circular ligaments separate the annular areas from one another and subdividing ligaments partition each area into two or more segmented annular passages through which the fluid can flow. The number and width of circular ligaments combined with the number and width of subdividing ligaments determines the overall porosity of the conditioner. Furthermore, by varying the number of subdividing ligaments, or the thickness of these ligaments from one area relative to another, the hydraulic diameter and porosity can be controlled in each area, and hence the flow resistance can be graded to produce a desirable flow profile. It is this variation of the thickness of the circular and subdividing ligaments to obtain a particular shape of flow profile that puts this new design into the 'flow conditioner' category, otherwise the device would fall into the more basic 'flow straightener' category.

The outer segmented-annular ring of passages have an outer bounding wall included in the design of the conditioning element, but could be open at the outer circumference such that the inside wall of the pipe forms in the outer wall of each of those passages.

A benefit of use of a segmented annular geometry over circular passages is easily explained by reducing one of the problems of circular holes down to a simple example. Consider a conditioner with 19 circular passages of equal size arranged in a hexagonal pattern. The maximum size that these passages can be before they merge together is one fifth of the inner diameter of the pipe. Therefore the maximum free area would be approximately 76 % of the pipe area. With segmented annular passages the free area in any portion of the conditioner can be larger than this limit, whilst still having sufficient ligament thickness for mechanical strength of the conditioner.

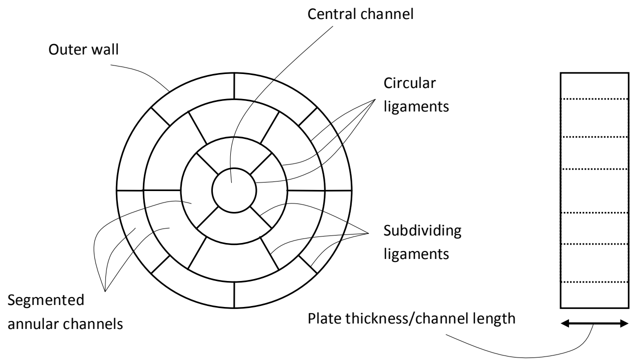

Fig. 3 Segmented annular conditioner geometry

For practical purposes it is convenient to make the length, *l*, of each segmented annular passage is equal to the others and to the thickness of the plate, though variations with different passage lengths can also be conceived by changing the plate thickness associated with each ring. It is however, important to consider the overall length of the passages and their hydraulic characteristics. Passages that are too short in length will be ineffective at preventing the passage of swirl, whereas passages that are too long may increase the pressure loss or increase the size of the conditioner unnecessarily, with consequences for manufacturing costs. Furthermore if the ratio of the length of the passage to its hydraulic diameter differs from one set of holes to another, then the characteristics of the flow through the holes may differ significantly as a function of the pipe Reynolds number.

The importance of the ratio of hydraulic diameter of the hole to plate depth (or passage length) is worth further consideration and is rarely mentioned in the flow conditioner literature, which may help explain some of the observations of Karnik [7] and Langshot & Erdal [11].

For passages that have a very short length to hydraulic diameter ratio (l/d) of less than 0.5, swirl will pass easily, and after the flow separates at the entrance to the passage, it will not reattach inside the passage (this is termed fully separated flow). For passages of intermediate l/d in the region of 0.5 to 1, swirl may still pass, and the flow may or may not reattach inside the passage depending on the flow conditions prevailing (this is termed marginally separated/attached flow). For passages of relatively long l/d, greater than 1, swirl will be suppressed and the separated flow at the entrance to the passage will re-attach (this can be termed fully reattached flow). In terms of the pressure loss for flow through a passage between two sections of pipe it can be shown that the pressure loss is greatest for fully separated flow, and reduces to a minimum once the flow is fully reattached [13]. Beyond the minimum pressure loss point the pressure loss will increase again owing to increased frictional losses in the passages of the conditioner. Therefore it is possible to optimise a flow conditioner in terms of the length to hydraulic diameter ratio.

For the common thick-plate conditioners available today that use circular passages, the thickness of the plate is constant and normally in the range of 0.12 to 0.15 pipe diameters. The hole diameters are typically in the range of 0.1 to 0.19 relative to the pipe diameter, with resulting l/d values in the range of 0.63 to 1.5. As pointed out earlier, the range of l/d corresponding to marginally separated/attached flow is typically between 0.5 and 1. In the likes of the Laws, Gallagher and Spearman plates, a range of values of passage diameter are used, with the result that under certain flow conditions some passages may have separated flow whereas others may have reattached flow.

It is desirable to avoid the possibility of having both separated and reattached flow conditions occurring in different passages of the same conditioner at the same time. One solution to this would be to increase plate thickness until fully reattached conditions occur in the passages of largest hydraulic diameter. However, when this is done the pressure loss coefficient would then increase undesirably in the passages of smaller hydraulic diameter. It is therefore attractive to be able to produce a conditioner design where the value of hydraulic diameter of all passages is the same.

When the hydraulic diameters of each passage are the same, then with a conditioner of constant thickness the values of l/d will also be the same, and consequently the coefficient of frictional pressure loss through the hole should also be the same. For circular passages, the hydraulic diameter is simply equal to the diameter, and therefore for l/d to be constant the passages must all have the same diameter, which imposes unwanted restrictions on the geometric arrangement of the holes in terms of producing the desired graded resistance. For segmented annular passages, the hydraulic diameter is equal to four times the cross-sectional area divided by the perimeter. Therefore the hydraulic diameter is a function not only of the cross-sectional area of the passage, but also the aspect ratio of the passage. This provides greater flexibility in design when it is desirable to vary or optimise the conditioner design in terms of both porosity and hydraulic diameter.

The segmented annular conditioner can be manufactured from a variety of materials by a method chosen to suit the materials of construction. For conditioners that are to be used in small pipes it is most likely that the conditioner would be manufactured from a solid part made of metal or plastic with the passages cut into the material using machine tools. Techniques such as water-jet cutting are appropriate for some materials up to a certain thickness or conventional drilling and milling techniques can be employed. In the design process the number of rings and number of subdividing ligaments can be constrained in order to produce an appropriate balance between material and manufacturing cost.

The new conditioner can be designed either to fit fully inside a pipe section with some means of securing it in place, or it can be designed to fit between pipe flanges. For increased mechanical strength it is preferable that the segmented annular passages be designed with rounded internal corners.

In terms of practical preference, the conditioner would be installed as a single unit. However, two or more units could be installed in series with some separation in between in order to perform more effective flow conditioning. In terms of conditioning performance versus overall pressure loss, this may be preferable to using a single unit.

The Design Process

The design and optimisation of a particular conditioner geometry according to these principles begins with defining the required characteristics of the conditioner in terms of overall pressure loss and desired axial flow profile shape. At this stage any other constraints or requirements can be added such as the overall thickness of the plate, the minimum length to hydraulic diameter ratio, the minimum width of ligaments, the minimum radius of the inside corners and/or a specification that all passages have the same hydraulic diameter. Next the general characteristics of the conditioner are considered in terms of the approximate total number and size of the passages. Once the number of annular rings and the number of segments per ring has been determined, values are chosen to produce an initial design and then the optimisation of the conditioner can begin (or indeed, it is also conceivable that the optimisation process could include varying the number of rings and segments per ring).

The thickness of the circular ligaments between annular rings and the thickness of the subdividing ligaments are set to initial values chosen from a practical and cost perspective. The radial width of the segmented annular passages are set to initial values (for example approximately equal) such that the total radial width of the passages plus the circular ligaments sums to the diameter of the conditioner. The thickness of the plate is set to an initial value.

The porosity and hydraulic diameter are then calculated for each ring. This, in addition to knowledge of the thickness of the conditioner, allows the pressure loss coefficient and relative velocity to be estimated for each segmented annular ring. In practice this can be achieved using semi-empirical pressure loss models that relate these terms, such as those described by Ward-Smith [13] or Idelchik [14]. Alternatively, the profile and pressure loss characteristics can be determined by means of computational fluid dynamics or by experimental testing. The geometry is then iteratively adjusted until the desired velocity profile and other optimal characteristics are achieved.

Some trial and error is required in terms of the starting conditions and constraints in order to obtain convergence and produce a solution that has the required characteristics, and when computational methods are used in the design, final testing and some fine tuning may be required.

A New Flow Conditioner

This design process will now be described in detail as applied to produce a new conditioner with the desirable characteristics in the form of a pressure loss coefficient of less than 1, and each passage in the conditioner having the same hydraulic diameter.

Common thick-plate type conditioners have between 25 and 32 circular passages, with the outer holes typically being sized at approximately 10 % of the pipe diameter. Therefore for a conditioner design with a broadly similar number and size of passages the process began by dividing the pipe into a central circular passage and three annular rings, the radial width of each ring being less than 14 % of the diameter. Then using radial subdividing ligaments, and taking into consideration a desire for symmetry, the inner annulus was partitioned into six segments, and each of the outer annuli into 12 segments, resulting in 31 passages in total. The thickness of the circular and subdividing ligaments were set to an arbitrary value to start, at 1 % of the pipe diameter. As an initial condition a width of 0.12 D was chosen for each of the annular sections, with the result that the starting diameter of the central circular passage was 0.2 D (given the constraint that the outer wall and circular ligaments plus the width of the passages must sum to equal the pipe diameter).

At this point the hydraulic diameter of the largest passage is 0.2 D. Given that it is desirable to target an *l*/*d* value of greater than 1, a plate thickness of 0.2 D was chosen, which ensures that the *l*/*d* requirement is met, though that could also have been adjusted as part of the optimisation process.

In the case of a single central passage its porosity is determined by its diameter and by the thickness of the circular ligament that separates it from the first annular ring. Therefore the first step of the optimisation was to adjust the other geometric parameters until the pressure loss coefficient of this passage is close to the target value for the conditioner as a whole. This first step was achieved by setting the constraint that the circular ligaments should be of equal thickness (with the exception of the outer wall, which was fixed at 0.01 D), and then gradually increasing the thickness of those, concurrently reducing the diameter of the central passage, until the desired loss coefficient was achieved.

The second step was to adjust the width of the radial ligaments in each ring until the desired velocity profile shape was achieved, whilst also considering the target for the overall loss coefficient. In this step the value for the width of the radial ligaments in each annulus was adjusted iteratively until the desired profile shape and loss coefficient was achieved. After this step the result is a design which that should produce the desired velocity profile, and have the intended overall loss coefficient, but may not yet meet some other requirements such as every passage having the same hydraulic diameter.

To obtain the same hydraulic diameter in each ring the next step was adjust the radial width of each of the segmented annular channels to satisfy the condition that all hydraulic diameters are the same, keeping all other dimensions constant, with the exception of the diameter of the central passage, which was also allowed to change. When the adjustment is made in this way, the relative velocities diverge again from their target values, requiring further iterations to be made.

In the next step, the widths of the circular ligaments were adjusted again to bring the loss coefficient for the central passage back closer to its target value. In the fifth and final step the desired velocity profile and loss coefficient was sought, again by means of adjusting the width of the radial ligaments.

Figures 4 and 5 illustrate the convergence of the velocity profile and the hydraulic diameters respectively towards their target values at each of these optimisation steps.

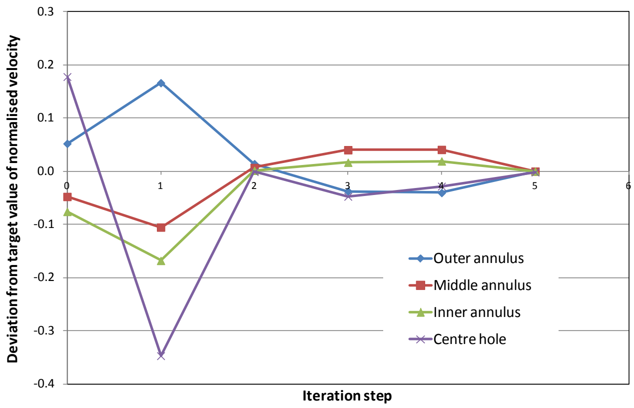

Fig. 4 Graph showing convergence of the design towards the desired velocity profile

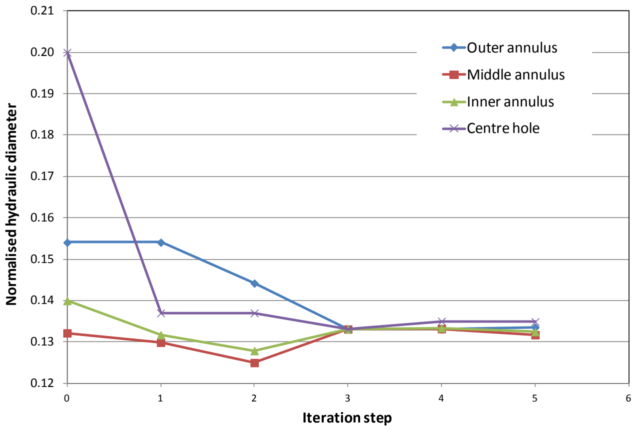

Fig. 5 Graph showing convergence of the design in terms of the hydraulic diameter of the passages

The resulting conditioner design was then manufactured complete with a flange for installation between pipe sections, as shown in Figure 6. Observe the variation in ligament width from inner to outer rings.

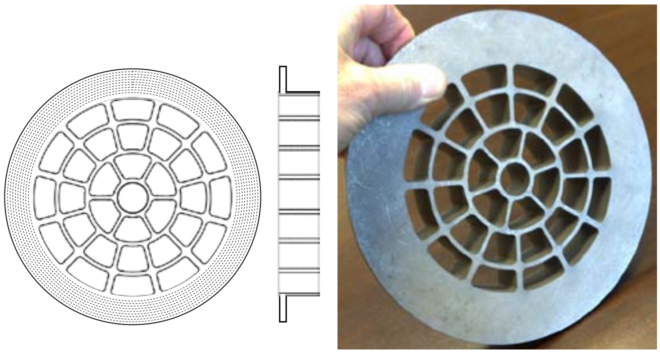

Fig. 6 New conditioner including rounded internal corners and flange

Performance Tests

In order to evaluate the effectiveness of the new flow conditioner, tests were carried out in the ISO 17025 accredited Cameron flow calibration laboratory at the Caldon Ultrasonics Technology Centre in Pittsburgh, Pennsylvania.

A Caldon LEFM 280Ci ultrasonic flow meter with eight flow velocity measuring paths was employed to determine the effectiveness of the flow conditioner. The 8-path meter has been described in detail elsewhere, see for example Brown et al [15]. In its 8-path format the LEFM flow meter does not normally require a flow conditioner. However in this case, the data from the eight measurement paths was evaluated in the form of two 4-path combinations equivalent to the Caldon LEFM 240Ci flow meter, in order to determine the influence on the hydraulic correction factor of each 4-path meter. Furthermore, all eight paths were combined to give a measure of the average swirl in the form of a ratio of the tangential velocity to the mean axial velocity.

The new conditioner was tested downstream of a long straight pipe and then downstream of an arrangement of six out of plane bends known to generate swirl and produce asymmetric distortion of the axial velocity profile. The tests were conducted using a kerosene substitute fluid with a viscosity of approximately 3 cSt over a range of flowrates in the range of 74 to 740 m^3/hr in a 6-inch pipe. A Laws type conditioner (Nova/CPA 50E variant) was also tested in the same configuration. The arrangement of bends was kept the same for all tests. The meter factor data was obtained by calibration using a unidirectional ball prover with an uncertainty of +/-0.044 % as the traceable reference standard.

Test data is presented for the following installation combinations:

- Long straight pipe with no flow conditioning
- Long straight pipe with a Laws type conditioner ten pipe diameters upstream of the flow meters
- Long straight pipe with the new conditioner ten pipe diameters upstream of the flow meters
- Flow meters at ten pipe diameters downstream of six bends with no flow conditioning, with the measurement paths orientated horizontally
- Flow meters at ten pipe diameters downstream of six bends with no flow conditioning, with the measurement paths orientated vertically
- Laws type conditioner four pipe diameters downstream of six bend, with the flow meters ten pipe diameters downstream of the conditioner
- New conditioner four pipe diameters downstream of six bend, with the flow meters ten pipe diameters downstream of the conditioner

Figures 7a and 7b show the results for each of the 4-path meters (A and B) in the straight pipe configuration and at ten diameters downstream of the bends without flow conditioning, with the measurement paths in both horizontal and vertical orientations. It can be observed that under these conditions, with no flow conditioning, the swirl and distortion generated by the bends results in changes in meter factor having magnitude typically between 0.3 to 0.5 %.

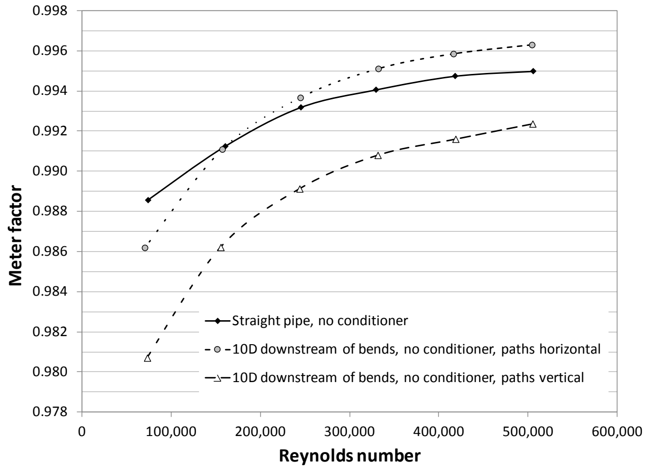

Fig. 7a Meter factor versus Reynolds number for meter A in straight pipe and downstream of bends with no flow conditioner

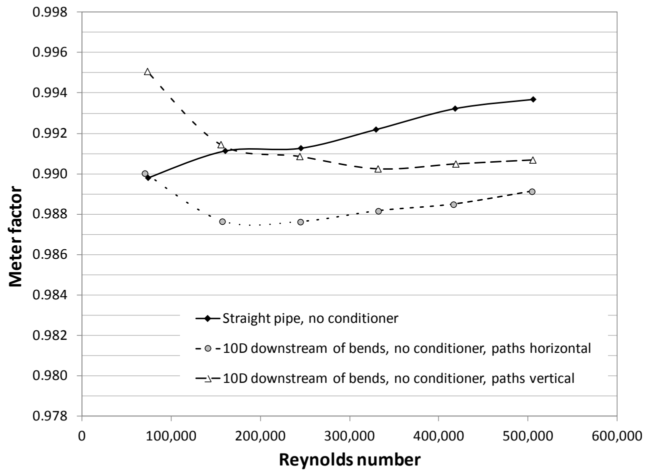

Fig. 7b Meter factor versus Reynolds number for meter B in straight pipe and downstream of bends with no flow conditioner

Figures 8a and 8b show the results for meters A and B installed ten diameters downstream of a Laws type thick-plate conditioner. The difference between the straight pipe case and the case where the conditioning plate is four diameters downstream of the bends is typically on the order of 0.1 % or less.

The data shown in Figures 8a and 8b can be summarised quantitatively in terms of a flow weighted mean error shift. The results of this calculation are given in Table 1.

Table 1

Flow weighted mean error shift		
	Laws type thick-plate	New conditioner
Meter A	0.08%	-0.06%
Meter B	0.09%	0.10%

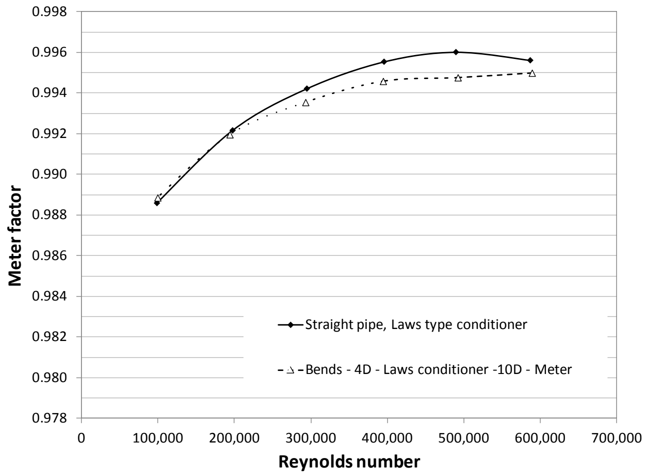

Fig. 8a Meter factor versus Reynolds number for meter A in straight pipe and downstream of bends with the 4D–Laws conditioner–10D–Meter arrangement

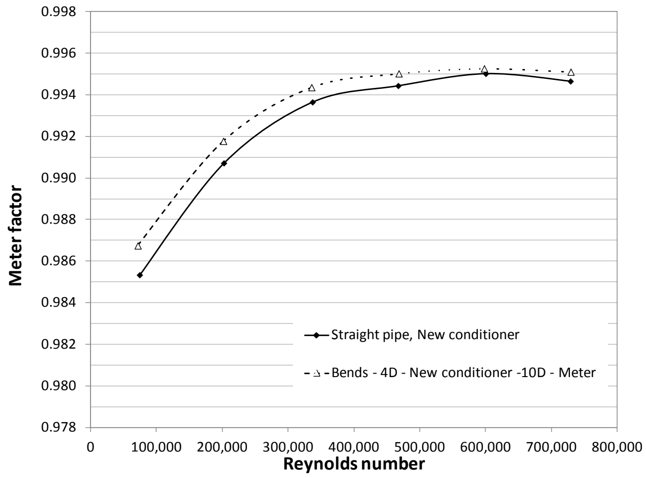

Fig. 9a Meter factor versus Reynolds number for meter A in straight pipe and downstream of bends with the 4D–New conditioner–10D–Meter arrangement

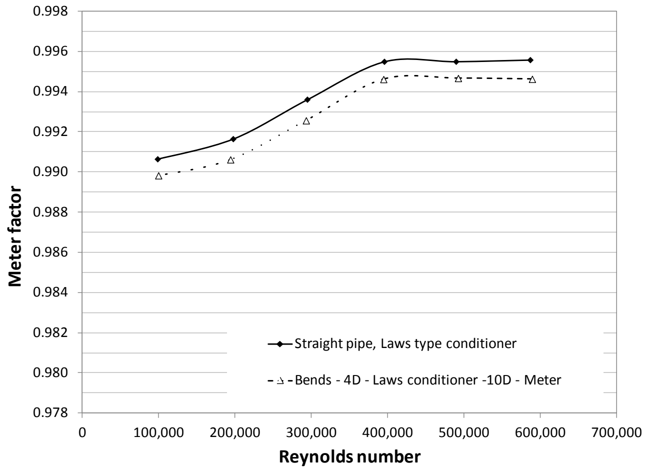

Fig. 8b Meter factor versus Reynolds number for meter B in straight pipe and downstream of bends with the 4D–Laws conditioner–10D–Meter arrangement

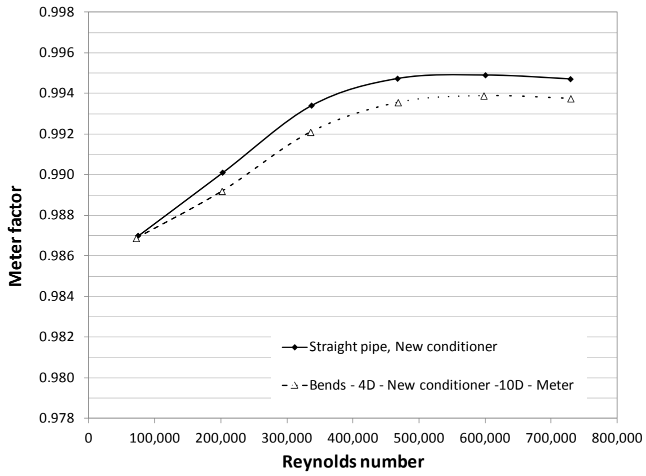

Fig. 9a Meter factor versus Reynolds number for meter B in straight pipe and downstream of bends with the 4D–New conditioner–10D–Meter arrangement

Figures 9a and 9b show the results obtained using the new conditioner previously described. It can be readily observed that the difference between the straight pipe case and the case where the conditioning plate is four diameters downstream of the bends is similar to the case for the Laws type conditioner, typically on the order of 0.1 % or less. These results can also be summarised quantitatively in terms of a flow weighted mean error shift, as reported in Table 3.

The data recorded in Table 3 shows that in terms of the flow measurement performance of a 4-path ultrasonic meter, the new conditioner matches the performance of the Laws type thick plate conditioner as the flow weighted mean error shifts are of a similar magnitude, all being less than 0.1 %.

Figures 10 and 11 show the swirl quantified in terms of the tangential velocity as a percentage of the mean axial velocity. In Figure 10 we show the bare straight pipe case plus the swirl generated by the bends and measured ten diameters downstream with no flow conditioning. It is clear that the bends generate a high level of swirl. Figure 11 shows the results for the two flow conditioners tested here, the Laws type conditioner and the new conditioner. Comparing Figures 10 and 11 it is clear that both conditioners substantially reduce swirl. At the higher Reynolds numbers it appears that the Laws type conditioner is slightly more effective at reducing swirl than the new conditioner. However, when the measurement results of Table 3 are taken into consideration, this appears to be an insignificant difference in terms of 4-path ultrasonic meter performance.

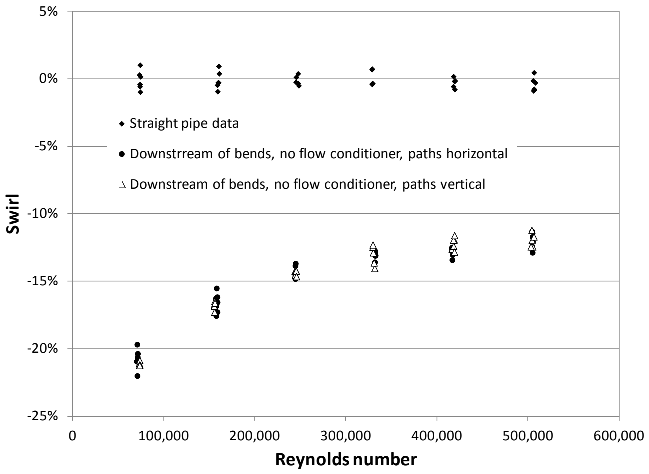

Fig. 10 Swirl measured using the 8-path ultrasonic meter in straight pipe and 10D downstream of bends with no flow conditioning

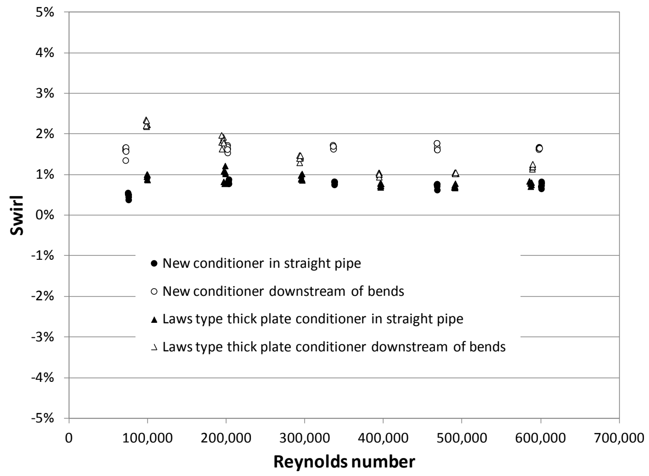

Fig. 11 Swirl measured using the 8-path ultrasonic meter in straight pipe and downstream of bends with the 4D – conditioner – 10D – Meter arrangement, for both the Laws type conditioner and the new conditioner

During the tests, the pressure loss across the new conditioner was measured. The measurements of pressure loss can be converted into a dimensionless loss coefficient, which is a useful non-dimensional measure of the energy lost when flowing through the conditioner. The pressure loss data is shown in Table 2.

Table 2 Pressure loss data

Flowrate	Density	Differential pressure	Velocity	Differential pressure	Loss coefficient, K
m3/hr	kg/m3	psi	m/s	Pascals	-
740	800	6.9	11.3	47574	0.937
607	800	4.5	9.2	31026	0.908
474	800	2.5	7.2	17237	0.827
340	800	1.5	5.2	10342	0.965
				Average	**0.91**

For the new conditioner, the average loss coefficient is 0.91. This is less than half of the pressure loss corresponding to the Nova/CPA 50E variant of the Laws conditioner, which has a loss coefficient of approximately 2 [1].

DISCUSSION AND CONCLUSIONS

A new design of flow conditioner using segmented annual passages has been described which by varying the number and width of circular and subdividing ligaments allows design targets to be reached via a relatively simple process of design and optimisation.

The general design is versatile and allows for variations in plate thickness, number of holes, pressure loss and velocity profile to suit different application requirements.

A conditioner based on the new design with 31 passages has been built and tested in an accredited calibration laboratory. The results of the tests show that for use with a 4-path ultrasonic meter, the measurement results with the new conditioner are equivalent to the Laws type plate, but are achieved with less than half the pressure loss.

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