Latest Developments in Flow Measurement using Differential-Pressure Meters

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Abstract

After many years of discussion ISO 5167:2003 was published with several new features: it is divided into 4 parts; it has a new expansibility equation for orifice plates; the straight lengths required upstream of orifice plates and of Venturi tubes have been revised; a compliance test for flow conditioners is included, which leads to significant reductions in upstream straight length; an isenthalpic temperature correction between the downstream temperature measurement and the required upstream temperature is included; permissible steps in the upstream pipe diameter have been increased; and roughness limits have been revised. These changes are described and the reasons given.

ISO/TR 9464 (the Code of Practice), ISO/TR 12767 (effects of departure from ISO 5167), and ISO/TR 15377 (nozzles and orifice plates beyond the scope of ISO 5167) all now require revision, to take account of both the changes to ISO 5167 and data collected recently. Some issues in these revisions are described. One area where work has been carried out in recent years is the performance of Venturi tubes in high-pressure gas. Some of the research work is briefly described.

Keywords: Differential pressure meters; orifice plates; Venturi tubes; international standards

1. Introduction

ISO TC30/SC2 WG11, of which I am Convener, was established in 1996 to revise ISO 5167-1:1991. Committee Drafts were issued to SC2 for discussion in 1998 and 1999. The Draft International Standard was published in 2000, the Final Draft International Standard in 2002, and the new International Standard itself in 2003.

ISO 5167 was divided into 4 parts: general (ISO 5167-1), orifice plates (ISO 5167-2), nozzles and Venturi nozzles (ISO 5167-3), and Venturi tubes (ISO 5167-4) [1-4]. Many users will only require the general part and one other part. The most significant areas of change from ISO 5167-1:1991 [5] are given below.

2. Orifice Plate Installation Effects and Flow Conditioners

A very large amount of data on installation effects on orifice plates has been collected in recent years, particularly in the USA and in Canada but also in the UK and in Germany. The data and the methods of analysis are given in [6]. Revised straight lengths based on an analysis of these data, initially undertaken by API but in which representatives of countries outside North America were later involved, are included in ISO 5167-2. The data downstream of two bends were taken with various separations between the bends, and the required separations between pairs of bends are included in the table together with the corresponding lengths between the bends and the orifice plate. Some pairs of bends in perpendicular planes with small separation gave very large shifts in discharge coefficients, although other pairs of bends with the same separation gave smaller shifts. It is recommended that if there is a possibility of a fitting creating severe swirl, as two bends in perpendicular planes with separation less than $5D$ or a header may do, a flow conditioner should be used. The importance of separating pairs of bends in perpendicular planes in order to reduce swirl has become increasingly clear: $5D$ separation will normally reduce the required length between the bends and the orifice. Thermowells larger than $0.03D$ in diameter are not recommended upstream of the orifice.

In ISO 5167-1:2003 a compliance test for flow conditioners is included: using a primary device of diameter ratio, β, equal to 0.67 the shift in discharge coefficient from that obtained in a long straight pipe must be less than 0.23 per cent when a flow conditioner is used in each of three situations:

a) in good flow conditions,

b) downstream of a half closed gate valve (or D-shaped orifice)

c) downstream of a device producing a high swirl (the device should produce a maximum swirl angle across the pipe of at least 24° $18D$ downstream of it or at least 20° $30D$ downstream of it).

This last installation was included to give a swirl similar to that found downstream of a header tested at NEL. As well as the test for β = 0.67 the flow conditioner must pass the high-swirl test (c) for β = 0.4. This test for β = 0.4 is included because, although for non-swirling flow shifts in discharge coefficient increase with β, this is not necessarily true for swirling flow. Most of the test must be passed for one pipe diameter and Reynolds number, but additional test work must be undertaken for a second pipe diameter and for a second Reynolds number. If a flow conditioner passes this test it may be used with no additional uncertainty downstream of any fitting with any diameter ratio up to 0.67. If it is desired to use the flow conditioner for $\beta > 0.67$ additional testing is required. The range of distances between the upstream fittings and the flow conditioner and that between the flow conditioner and the primary device which are used in the tests will determine the acceptable ranges of distances when the primary device is used. The compliance test is included in ISO 5167-1 because the same test is applied whichever primary device is used; however, passing the test with one type of primary device does not imply that the test would have been passed with all types of primary device at the same ranges of distances.

Tests have been undertaken with orifice plates to determine the required distances between fittings and flow conditioner and flow conditioner and orifice plate. Within ISO 5167-2 examples of flow conditioners giving an overall length of 17D from fittings to orifice plate are given (see also [7,8]). This length is not the minimum upstream length requirement using flow conditioners; it is the length at which successful tests were carried out.

The installation requirements given in Part 2 of API 14.3 [9] are similar to those in ISO 5167-2. This is significant progress.

3 Expansibility Equation for Orifice Plates

The orifice plate discharge coefficient equation has been revised on the basis of data collected in the last 20 years, and it was appropriate that the same process of revision should occur for the expansibility factor.

The equation for the orifice expansibility factor in ISO 5167-1: 1991 was derived by Buckingham [10] largely based on data collected at tests in Los Angeles in 1929. Using these data Buckingham derived the equation:

$$\varepsilon = 1 - (0.41 + 0.35\beta^4)\frac{\Delta p}{\kappa p_1} \qquad (1)$$

where ε is the expansibility factor, Δp is the differential pressure across the orifice plate, p_1 is the static pressure at the upstream tapping and κ is the isentropic exponent.

As part of the EEC Orifice Project data were collected on expansibility factors. At NEL in the 100 mm (4-inch) pipe data were collected for three diameter ratios, 0.2, 0.57 and 0.75, in air with 140 kPa < p_1 < 800 kPa. Details of the analysis of the data are given in Reference [11] together with references to the individual data sets from all the laboratories whose data were used. This work followed that of Kinghorn [12]. Gaz de France collected data in the 100 mm (4-inch) pipe for a diameter ratio of 0.66 in natural gas at $Re_D = 1.2 \times 10^6$.

Gasunie did not collect data on expansibility factor directly, but, within the data collected by them in the 100 mm (4-inch) pipe for the discharge coefficient database, it is possible to identify sets of data taken over both a significant range of static pressure and a small range of Reynolds number. These data are for diameter ratios 0.2, 0.57 and 0.66. Similarly CEAT did not collect data on expansibility factor directly, but, within the data collected by them on the 250 mm (10-inch) pipe run for the discharge coefficient database, it is possible to identify sets of data taken over both a significant range of static pressure and a small range of Reynolds number. These data are for a diameter ratio of 0.2.

In addition to the European work CEESI in a 50 mm (2-inch) pipe collected data for six diameter ratios, 0.242, 0.363, 0.484, 0.5445, 0.6655 and 0.726 in air with 115 kPa < p_1 < 2150 kPa.

In order to analyse the NEL (and similar) data a common method has been to calculate

$$\varepsilon\Big|_{\text{calc}} = \frac{(C\varepsilon)_{\text{measured}}}{C_{\text{water}}} \qquad (2)$$

where $(C\varepsilon)_{\text{measured}}$ is taken from the gas tests and C_{water} from a previous water calibration of the same orifice plate and then to use the method of least squares to determine the constants in

$$\varepsilon\Big|_{\text{calc}} = 1 - a - b\,\mathrm{f}\left(\frac{\Delta p}{p_1}, \kappa\right) \qquad (3)$$

and then to fit the slope terms, b.

The problem with this method is that there is always some bias between a gas flow laboratory and a water flow laboratory. So, a better estimate of ε is given by

$$\varepsilon\big|_{\text{calc},2} = \frac{\varepsilon\big|_{\text{calc}}}{1-a} = 1 - \frac{b}{1-a}\,\mathrm{f}\left(\frac{\Delta p}{p_1}, \kappa\right) \qquad (4)$$

which is equivalent to fitting $(C\varepsilon)_{\text{measured}}$ as $C_{\text{incompressible}}(1 - b^{*}\mathrm{f})$ without assuming a value for $C_{\text{incompressible}}$.

In the case of the Gaz de France data the value of C obtained when ε is as close as possible to 1 had been used as the reference value (equivalent to C_{water} in Equation (2)); this reference value is a single measurement, and so in [11] all the data including the reference value were fitted and values of $b/(1 - a)$ calculated. The data from CEESI had already been analysed in a similar manner to that used here and so in most cases a was calculated to be 0.

For the data from Gasunie and CEAT the Reynolds number was not constant and so it was necessary to apply corrections to the measured values of $C\varepsilon$ so that all the values are effectively taken at one Reynolds number. To do this, the dependence of C on Reynolds number given in the Reader-Harris/Gallagher Equation [2] was assumed.

Calculations were performed for each set of data for two functions, f:

$$\mathrm{f}\left(\frac{\Delta p}{p_1}, \kappa\right) = \frac{\Delta p}{\kappa p_1} \qquad (5)$$

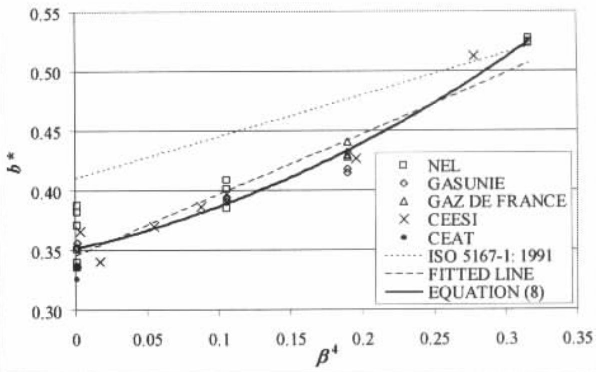
and

$$\mathrm{f}\left(\frac{\Delta p}{p_1}, \kappa\right) = 1 - \left(\frac{p_2}{p_1}\right)^{1/\kappa}, \qquad (6)$$

where p_2 is the static pressure at the downstream pressure tapping. The first equation is simpler but the latter is based on the best physical understanding. It was clear that, where there is a wide range of values of p_2/p_1, that is in the NEL, Gaz de France and CEESI data, using Equation (6) gives a smaller standard deviation of the data about the obtained fit, and so Equation (6) is used to reduce bias in the final equation. More complex forms of equation than Equations (5) and (6) were tried but only gave marginal improvements in quality of fit. The values of b^{*} (for use with f in Equation (6)) are dependent on β and are shown in Figure 1. For $\beta \le 0.66$ there is a linear dependence on β^4. At higher diameter ratios the measured values lie above a fitted line and so, to avoid bias at small β due to the points for large β, b^{*} has been fitted as follows:

$$b^{*} = a_1 + a_2\beta^4 + a_3\beta^8 . \qquad (7)$$

Fig 1 Experimental data: coefficients of f in equation (4) (f as in equation (6))



Different functions of β did not give a significant improvement in fit. On fitting the data the following equation was obtained:

$$\varepsilon = 1 - (0.351 + 0.256\beta^4 + 0.93\beta^8)\left\{1 - \left(\frac{p_2}{p_1}\right)^{1/\kappa}\right\}. \qquad (8)$$

The standard deviation of the values of b^{*} about the quadratic fit in β^4 was 0.0148.

The uncertainty of the value of ε is considered in Reference [11] and so the relative uncertainty of ε given in ISO 5167-2 is

$$3.5\frac{\Delta p}{\kappa p_1} \text{ per cent.} \qquad (9)$$

Because the previous equation for ε_2 was in error and it was little used it has been omitted.

Additional expansibility-factor data are now being collected in the USA.

4 Eccentricity limits for orifice plates

There is a difference between the ISO and API standards, because there are small differences between the sets of data [13, 14] on which the two standards are based; the data are in good agreement in general, but since the permitted eccentricities are based on giving shifts in discharge coefficient of less than 0.1 per cent it is not surprising that different data sets give different permitted eccentricities. The ISO requirements have been determined so that shifts using either set of data are less than 0.1 per cent. The previous ISO requirements were sufficient to ensure good measurement. Under certain circumstances they were, however, more demanding than is necessary. To obtain a clause which described the experimental data more accurately it was necessary to consider the eccentricity in its two components, namely parallel to a pressure tapping and perpendicular to the pressure tapping, since eccentricity parallel to a pressure tapping has more effect on the measured discharge coefficient than eccentricity perpendicular to the pressure tapping.

To determine the eccentricity, the distance e_c between the centre-line of the orifice and the centre-lines of the pipe on the upstream and downstream sides is measured. For each pressure tapping the components of the distance between the centre-line of the orifice and the centre-line of the pipe in which it is located in the directions parallel to and perpendicular to the axis of the pressure tapping are determined. If there is to be no additional uncertainty, e_{cl}, the component in the direction parallel to the pressure tapping, must be such that for each pressure tapping $e_{cl} \le \dfrac{0.002\,5D}{0.1+2.3\beta^4}$

$$e_{cl} \le \frac{0.0025D}{0.1+2.3\beta^4} . \qquad (10)$$

and e_{cn}, the component in the direction perpendicular to the pressure tapping, must be such that for each pressure tapping

$$e_{cn} \le \frac{0.005D}{0.1+2.3\beta^4} . \qquad (11)$$

If the eccentricity is purely parallel to a tapping this is the same requirement as in ISO 5167-1:1991; otherwise it gives a larger tolerance.

5 Flatness test for orifice plates

The requirement in ISO 5167-1:1991 that the slope of a straight line connecting any two points of its surface had to be less than 0.5 per cent relative to a plane perpendicular to the centre-line of the orifice plate bore was too demanding, since measurements can be made with the two points very close together. A simplified test based on that in the API standard [9] has been adopted in which a straight edge of length D is laid across any diameter of the plate and the maximum gap between the plate and the straight edge is measured. The plate is considered to be flat when the gap is, for every diameter, less than $0.005(D-d)/2$. This ensures that the mean slope from the pipe wall to the orifice edge is less than 0.5 per cent. The API standard allows twice as large a gap; the ISO limit has been chosen so that, when the effect of deformation under pressure is added, the slope is still less than 1 per cent.

6 Temperature correction

Work at British Gas [15] and Gasunie [16] showed that, in a gas flow, to estimate the temperature upstream of the orifice plate from that measured some distance downstream, an isenthalpic correction is appropriate, and the standard has been revised accordingly. The British Gas report covers a wide range of diameter ratios, whereas the Gasunie report studied the flow through one orifice of diameter ratio 0.5 in more detail. The Gasunie report compared the change in temperature with the differential pressure; even better agreement with that predicted by an isenthalpic correction would have been achieved if the change in temperature between that upstream of the plate and that at the temperature tapping location downstream had been compared with the pressure loss, $\Delta\varpi$. The data show that an isentropic expansion is still correctly assumed from upstream of an orifice plate into the vena contracta. However, to use an isentropic correction between a location upstream of the plate and the usual temperature-tapping location downstream of the plate can lead to significant error.

In many cases it can be safely assumed that the temperature of the fluid is the same upstream of the plate and at the temperature-tapping location downstream. However, if the fluid is a non-ideal gas, the highest accuracy is required and there is a large pressure loss between the upstream pressure tapping and the temperature location downstream of the primary device, then it is necessary to calculate the upstream temperature from the downstream temperature (measured at a distance of $5D$ to $15D$ from the primary device), assuming an isenthalpic expansion between the two points. To perform the calculation the pressure loss $\Delta\varpi$ should be calculated from the differential pressure; the corresponding temperature drop from the upstream tapping to the downstream temperature location, ΔT, can then be evaluated given the rate of change of T with respect to p at constant enthalpy:

$$\begin{aligned} \Delta T &= \left.\frac{\partial T}{\partial p}\right|_H \Delta\varpi \\ &= \frac{R_u T^2}{p c_{m,p}} \left.\frac{\partial Z}{\partial T}\right|_p \Delta\varpi , \end{aligned} \qquad (12)$$

where T is the absolute temperature, R_u is the universal gas constant, $c_{m,p}$ is the molar-heat capacity at constant pressure and Z is the compressibility factor.

7 Pipe roughness limits upstream of orifice plates

New roughness limits for pipework upstream of orifice plates have been included in ISO 5167-2. These are derived from a physical understanding of the effect of pipe roughness on orifice plate discharge coefficients and the knowledge of the roughness of the pipes on which the discharge coefficient equation in ISO 5167-2 is based. Pipe roughness limits have been calculated to ensure that roughness will not shift the discharge coefficient from that given by the discharge coefficient equation by more than an appropriate fraction of its uncertainty.

The arithmetical mean deviation of the roughness profile, R_a, or the friction factor, λ, (or both) was measured for each of the pipes used to collect the data to which the discharge coefficient equation was fitted. Using these data it is possible to fit a discharge coefficient equation containing an explicit friction factor term; this was done at the time when the PR14 equation was developed and, using the PR14 tapping terms, the following equation for the C_∞ and slope terms was obtained for $Re_D > 3700$:

$$\begin{aligned} C &= 0.5945 + 0.0157\beta^{1.3} - 0.2417\beta^{8} \\ &+ 0.000514(10^{6}\beta / Re_D)^{0.7} \\ &+ (3.134 + 4.726A_p)\beta^{3.5}\lambda \end{aligned} \tag{13}$$

where

$$A_p = \left(\frac{2100\beta}{Re_D}\right)^{0.9}.$$

Reference [17] gives the details.

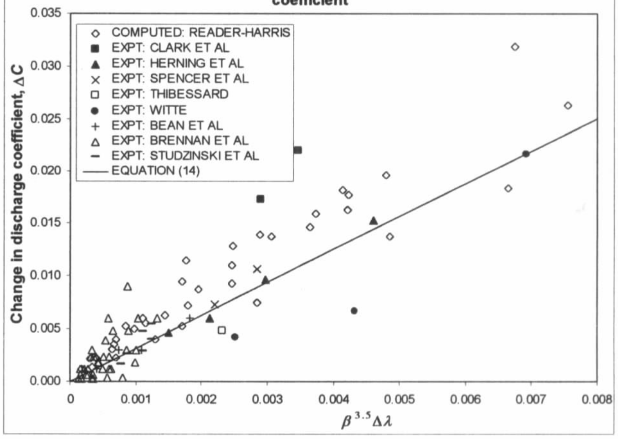
This gives the change in friction factor due to roughness, ΔC_{rough}, as

$$\Delta C_{\text{rough}} = 3.134\beta^{3.5}\Delta\lambda, \tag{14}$$

provided that Re_d is sufficiently large that A_p is negligible.

Figure 2 gives measured and computed (using CFD) values of ΔC as a function of $\beta^{3.5}\Delta\lambda$ (see Reference [18] for complete references). The computed values and the European experimental data were obtained using corner tappings. The North American experimental data (Bean et al, Brennan et al and Studzinski et al) were obtained using flange tappings. In Reference [18] the effect of roughness on discharge coefficients obtained with different pairs of pressure tappings is considered and it is shown that the effect of pipe roughness on the discharge coefficient using D and $D/2$ tappings is about 20 per cent less than on that using corner tappings. Since all the computational and most of the experimental data in Figure 2 were collected using corner tappings this may explain why equation (14) lies below the majority of the plotted data.

Fig. 2 The effect of rough pipe on the orifice plate discharge coefficient



Nevertheless there is a large scatter in the plotted data, and so a single equation is used to describe the effect of pipe roughness for all tappings. The equation used to determine limits of pipe roughness is again taken from Equation (13), but the A_p term is included:

$$\Delta C_{\text{rough}} = (3.134 + 4.726A_p)\beta^{3.5}\Delta\lambda. \tag{15}$$

It is not known whether the effect of change in friction factor increases for small Re_D, but it is safer to include the term in A_p in calculating the limits of pipe roughness. Moreover, there is little disadvantage in its inclusion since it causes a slight reduction in the limits of pipe roughness in a range of Reynolds number where they are already wide.

In order to calculate the limits of pipe roughness for the discharge coefficient equation, it is necessary first to use the measured values of relative roughness for each pipe to obtain a typical pipe relative roughness for the data (and thus for the equation fitted to it). This is a function of Re_D: the estimates are given in Table 1. The values of friction factor are consistent with the values of relative roughness if the Colebrook-White Equation is used (see Reference [19]).

Table 1 - Values of k/D and λ associated with the database

Re_D	10^4	3×10^4	10^5	3×10^5	10^6	3×10^6	10^7	3×10^7	10^8
$10^4 k/D$	1.75	1.45	1.15	0.9	0.7	0.55	0.45	0.35	0.25
λ	0.031	0.024	0.0185	0.0155	0.013	0.0115	0.0105	0.010	0.0095

The maximum permissible shift in C depends on U, the stated uncertainty of C. It is assumed here that the percentage shift, P, should not exceed

$$\begin{array}{lll} 0.5\beta & \beta \leq 0.5 & \\ 0.25 & 0.5 < \beta \leq 0.6 & (16) \\ 0.5(1.667\beta - 0.5) & 0.6 < \beta \leq 0.71 & \\ 1.13\beta^{3.5} & 0.71 < \beta & \end{array}$$

This restriction ensures that for $\beta \leq 0.5$, where other sources of error are dominant, $P/U < \beta$; for $0.5 < \beta \leq 0.71$, where pipe roughness is a major cause of error, $P/U < 0.5$; for $0.71 < \beta$, where pipe roughness is one of the two largest causes of error, the maximum value of P/U increases from 0.5 at $\beta = 0.71$ to 0.55 at $\beta = 0.75$.

Given the value of P in Equation (16) and the value of λ associated with the data in the database (and thus with the equation) in Table 1, it is possible from Equation (15) to calculate the maximum and minimum values of λ and hence of k/D for use with the equation for each Re_D and β. It was decided to write ISO 5167 in terms of R_a rather than k for specification of pipe roughness, since this will be more convenient for users, although k is still mentioned. On the basis of a simple computation with a roughness profile in the shape of a sine wave it has been assumed that $k \approx \pi R_a$. The maximum and minimum values of R_a/D for pipes upstream of orifice plates are given in Tables 1 and 2 of ISO 5167-2.

It is stated in ISO 5167-2 that the roughness requirements are satisfied in both the following cases:

$1\ \mu\text{m} \leq R_a \leq 6\ \mu\text{m}$, $D \geq 150$ mm, $\beta \leq 0.6$ and $Re_D \leq 5 \times 10^7$.
$1.5\ \mu\text{m} \leq R_a \leq 6\ \mu\text{m}$, $D \geq 150$ mm, $\beta > 0.6$ and $Re_D \leq 1.5 \times 10^7$.

These are not additional restrictions, just a method of removing the need to use Tables 1 and 2 of ISO 5167-2 in some cases.

The tables prescribe $k/D \leq 0.005$ even if the calculated value is higher. Although the limits for very large Re_D are much tighter than those in ISO 5167-1: 1991, for $Re_D = 3 \times 10^5$ they are very similar; this is unsurprising since the limits in ISO 5167-1:1991 were probably derived from data collected at around that Reynolds number. The minimum values represent exceedingly smooth pipes.

It might be argued that an equation with a friction factor term included explicitly should have a lower uncertainty. Such an equation was rejected on the grounds that it would be difficult to use.

8 Pulsation effects

A short clause which clarifies the acceptable limits for pulsations is included: the flow is considered sufficiently steady for ISO 5167 to apply when

$$\frac{\Delta p'_{\text{rms}}}{\overline{\Delta p}} \leq 0.10, \tag{17}$$

where $\overline{\Delta p}$ is the time-mean value of the differential pressure and $\Delta p'_{\text{rms}}$ is the r.m.s. value of $\Delta p'$, the fluctuating component of the pressure. This clause is consistent with ISO/TR 3313 [20]. $\Delta p'_{\text{rms}}$ can only be measured accurately using a fast-response differential pressure sensor; moreover, the whole secondary system in undertaking this measurement should conform to the design recommendations specified in ISO/TR 3313. It will not, however, normally be necessary to check that this condition is satisfied.

9 Throat thickness for nozzles

For high-ratio long radius nozzles ISO 5167-1: 1991 stated that the thickness, F, of the throat shall be between 3 mm and 13 mm. However, it was pointed out that for a large pipe diameter this may be too small to prevent distortion due to machining stresses. ASME MFC-3M: 1989 [21] gave (in ISO 5167-1 nomenclature) $2F \leq D - d - 6$ where F, D and d are expressed in millimetres. There was also a problem with ISO 5167-1:1991 for small pipe diameters: if $D = 50$ mm and $\beta = 0.8$ it was not possible to satisfy the different requirements.

Following Japan's suggestion no maximum for F in has been specified in ISO 5167-3. The maximum is the same requirement as was given by ASME MFC-3M. The text has also been revised to take account of small pipe diameters and reads as follows: 'The thickness F of the throat shall be greater than or equal to 3 mm, unless $D \leq 65$ mm, in which case F shall be greater than or equal to 2 mm. The thickness shall be sufficient to prevent distortion due to machining stresses.'

10 Straight lengths upstream of Venturi tubes

The straight length requirements upstream of Venturi tubes in ISO 5167-1:1991 were shown by data collected in the UK and Germany to be too short, and revised lengths have been included in ISO 5167-4.

Three 150 mm (6-inch) Venturi tubes with machined convergent sections, of diameter ratios 0.4, 0.6 and 0.75, were calibrated in water at NEL (in addition to baseline measurements) with a contraction, an expansion, two bends in perpendicular planes, a single bend, and two bends in the same plane at various distances upstream of them (see [22] and [23]). The contraction and expansion were conical with an included half-angle of about 4°. The bends were of radius 1.5*D* and each had short weldnecks of approximately 50 mm so that the bends could be as closely coupled as possible. In each case the distance was increased until the magnitude of the shift in discharge coefficient from the baseline was less than 0.25 per cent.

The measurement of length between a fitting and a Venturi tube is the distance between the downstream end of the fitting and the upstream tapping plane of the Venturi tube. The former has been taken to be the end of the curved portion of a bend or the tapered portion of a contraction or an expansion; the weld neck has been considered to be part of the straight length. The pressure tappings were 4 mm in diameter and were connected in 'triple-tee' arrangements. The mean shifts are presented in Figures 3-7.

FIG. 3 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF A 203 mm - 152 mm CONTRACTION

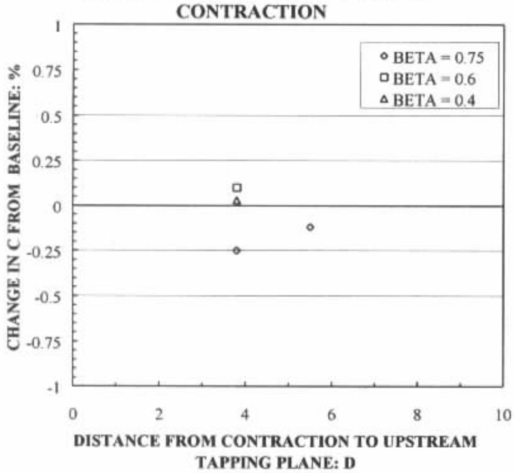


FIG. 4 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF A 102 mm - 152 mm EXPANSION

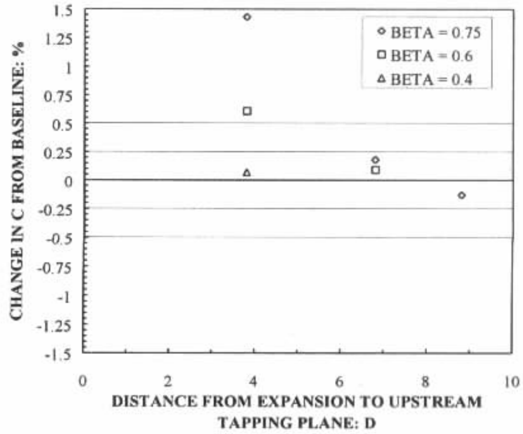


FIG. 5 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF TWO BENDS IN PERPENDICULAR PLANES

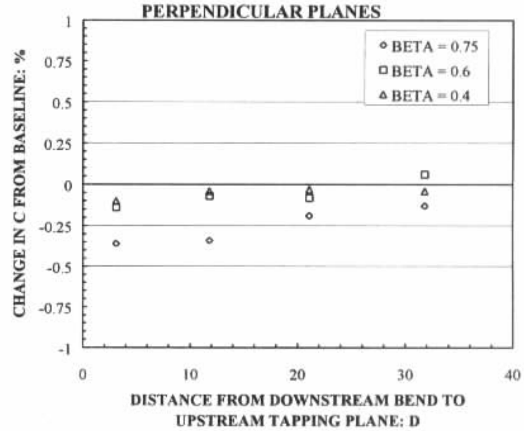
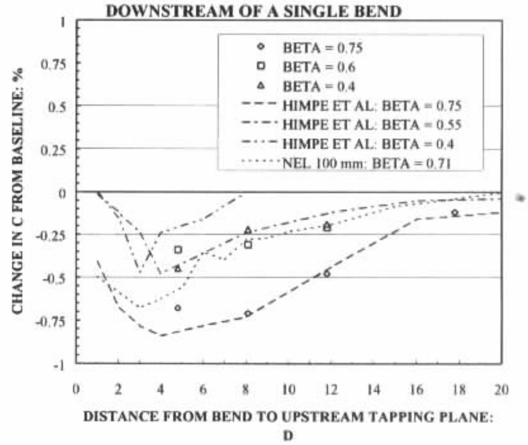
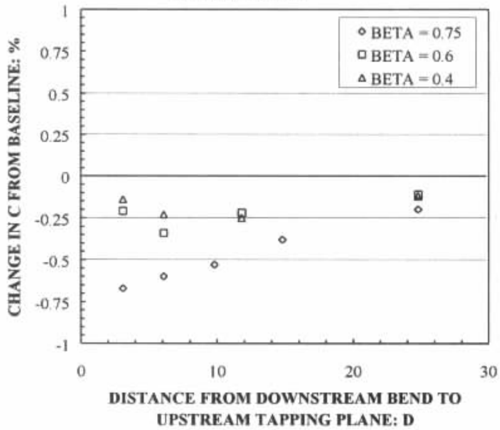


FIG. 6 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF A SINGLE BEND



The calibrations shown in Figure 6 are in good agreement with the data of Himpe et al [24], particularly for the larger values of β, and those of NEL (1985) quoted by Kochen et al [25]. Both Himpe et al and those described as 'NEL' had a pipe diameter of 100 mm.

FIG. 7 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF TWO BENDS IN THE SAME PLANE



Given the lack of multiple data sets for combinations of bends it was decided, in constructing Table 1 of ISO 5167-4, that the lengths from combinations of two bends should not be less than those from a single bend. It is known from the orifice installation-effects data that a small change in bend separation can have a significant effect on the shift in discharge coefficient. Moreover, it hardly seems sensible to encourage the installation of two bends in perpendicular planes upstream of a meter.

It is encouraging that there is good agreement between the data sets shown in Fig. 6 of this paper. In addition to those data shown Bluschke et al [26] obtained data for β = 0.71 downstream of a single bend and two bends in perpendicular planes: they are very similar to those shown here, although with slightly larger shifts downstream of two bends in perpendicular planes. The lengths required by ISO 5167-1:1991 are similar to those obtained by Pardoe [27]: he obtained positive shifts in C downstream of a single bend and stated that $6D$ was sufficient downstream of a single bend even for $\beta = 0.8$, but that downstream of two bends in perpendicular planes even $30D$ would not be sufficient (at least for $\beta >$ 0.55). An explanation for the difference between Pardoe's data and subsequent data is still required.

11 Surface roughness of Venturi tubes

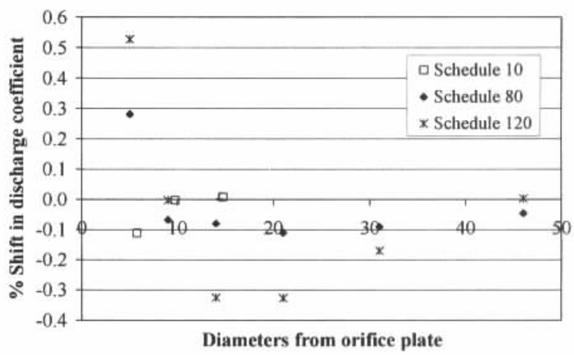
According to ISO 5167-1:1991 the surface roughness for the throat of a classical Venturi tube must be such that $R_a \leq 10^{-5}d$; so for a 50 mm $\beta = 0.4$ Venturi tube a surface roughness of $R_a \leq 0.2$ μm is required, which is very expensive to manufacture. Moreover, according to ISO 5167-1:1991 subsonic nozzles only required $R_a \leq 10^{-4}d$; so Venturi tubes had to be 10 times smoother than subsonic nozzles were required to be. NEL purchased 15 Venturi tubes from manufacturers and calibrated them [28], and found that, although they have roughnesses outside the existing standard but within $R_a \leq 10^{-4}d$, the mean discharge coefficients in water lay in each case within 1 per cent of 0.995. In ISO 5167-4 the surface finish of the throat of Venturi tubes must be such that $R_a \leq 10^{-4}d$.

12 Effect of upstream steps on an orifice plate

The requirements for the diameter of pipework upstream of orifice plates were very restrictive in ISO 5167-1:1991. It was stated that if there is to be no additional uncertainty there had to be no diameter steps in pipework greater than 0.3 per cent at any point within the upstream pipework, until the first fitting is reached. The first fitting could be more than $100D$ from the orifice plate, where D is the pipe internal diameter. If the pipe size was small it might have been necessary to machine the whole upstream length in order to meet this requirement. However, any requirement on steps which was the same at both $2D$ and $50D$ from the orifice plate was unlikely to be necessary at both locations. Moreover, it is important to know whether an existing orifice meter can be used in a different installation and what the pipework requirements should be for a laboratory calibrating an orifice meter.

So a 4-inch (100 mm) Schedule 40 orifice meter run made of several machined and dowelled sections was used: from various different distances from the orifice pipework of different schedules replaced the meter run. The shifts in discharge coefficient are shown in Figure 8 [29]. The effect of pipe reductions is, as expected, much smaller than the effect of pipe expansions. From the data shown here revised restrictions on upstream steps have been included in ISO 5167-2.

Figure 8 Mean effect of steps downstream of pipes of different schedules on an orifice plate ($\beta = 0.67$) in a Schedule 40 pipe



13 Venturi tubes

Not all areas of differential-pressure meter research have yielded answers which have gone straight into international standards. Work over the last eight years has shown that the performance of Venturi tubes in dry gas is very different from that in water. Some discharge coefficients in dry gas are greater than would have been expected by 3 per cent or even more. Work was presented by Jamieson et al. [30] and by van Weers et al. [31].

In response to this issue, as mentioned above, fifteen Venturi tubes were manufactured for NEL in a range of diameter ratios from 0.4 to 0.75, of diameters from 50 mm to 200 mm and of diameter ratios from 0.4 to 0.75 [28]. They all had machined convergents. They were calibrated in water and high-pressure air. The discharge coefficients increase with Reynolds number. CFD undertaken on Venturi tubes without pressure tappings does not show this significant increase in discharge coefficient. In fact the increase in discharge coefficient is due to static-hole error, the effect that pressure tappings of finite size do not measure the pressure which would have been measured using an infinitely small hole. The simplest presentation of the data is to define the Venturi throat-tapping Reynolds number

$$Re^* = \frac{d_{tap}}{d} Re_d . \qquad (18)$$

It was found [28] that if the mean discharge coefficient for any Venturi tube in water was C_{water} then the difference between the discharge coefficient for each point in gas and the mean value in water is given by Figure 9. An equation fitting all the air data from the standard Venturi tubes with an uncertainty of 1.23 per cent was derived and is given in [28].

It is possible to change the shape of a Venturi tube and to examine the effect of changes on the discharge coefficient. Several different changes have been tried [28, 32], and the best results (out of the changes tried) were obtained by changing the convergent angle to 10.5° and maintaining the other parameters as before. The data obtained in 100 mm pipe are shown in Figure 10. Further work in other pipe sizes to establish the benefits of halving the convergent angle has been undertaken and will be published in the next year.

Figure 9 C - C_{water} **as a function of Re***

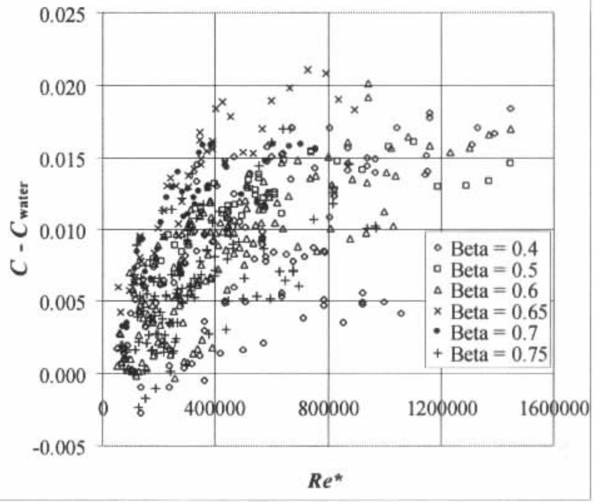


Figure 10 C - C_{water} **with 10.5 deg convergent**

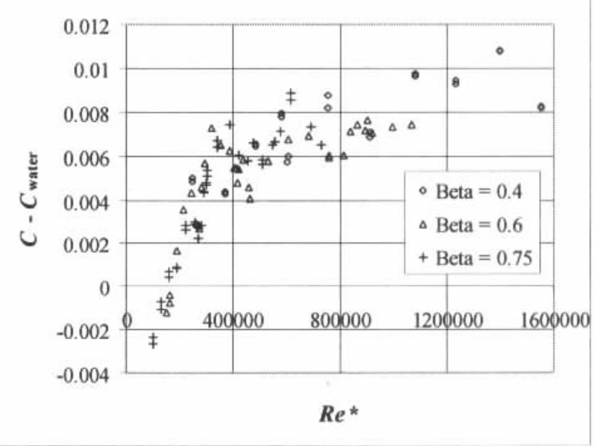
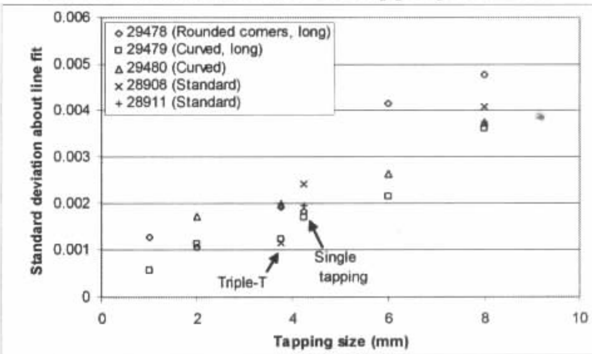


Figure 11 The standard deviation of the data about the line fit as a function of tapping diameter



In addition to the effect of the shape of the Venturi tube two important items which affect the Venturi tube discharge coefficients are the sharpness and the size of the pressure tappings. One way of obtaining sharp pressure tappings is to use spark erosion. When tappings of different diameters were put into five Venturi tubes of a range of shapes and diameter ratios and lines were fitted to the resulting data plotted against exp(-0.4$Re^*/10^5$) the standard deviations of the data were as given in Figure 11. Where possible 4 mm is a good choice for tapping diameter.

14 Differential-pressure meters outside ISO 5167

ISO/TR 9464 (the Code of Practice), ISO/TR 12767 (effects of departure from ISO 5167), and ISO/TR 15377 (nozzles and orifice plates beyond the scope of ISO 5167) all now require revision, to take account of both the changes to ISO 5167 and data collected recently. With a large open space upstream of the orifice plate the discharge coefficient equation is used with β equal to 0. The Stolz equation tended to a constant, the Reader-Harris/Gallagher Equation tends to a function of throat Reynolds number. It should be possible in ISO/TR 15377 to include work carried out on Venturi tubes with a convergent angle of 10.5°. It would also be useful to include an extension of the Reader-Harris/Gallagher (1998) equation for low pipe Reynolds number (below 5000) based on the data collected for the equation itself. In ISO/TR 12767 it would be useful to include new installation-effects data and new data on the effects of deposits on orifice plates.

One problem in the use of meters not standardized in ISO 5167 is how to assess their performance: API MPMS 5.7 [33] provides a standardized method for describing performance characteristics which helps to make possible the introduction of new flowmeters.

A modern patented differential-pressure meter which has found significant use is the V-Cone flowmeter. Reference [34] shows that it requires a shorter upstream straight length than an orifice plate [34]. Its expansibility factor is published in [35].

15 Conclusions

ISO 5167 has been revised to take into account the work on differential pressure flowmeters which has been done in recent years. This paper describes many of the changes: more use of flow conditioners, revised straight lengths upstream of orifice plates and Venturi tubes, a new expansibility equation for orifice plates, revised limits to pipe roughness, and better temperature correction all lead to improved accuracy. Flow conditioners and revised limits for eccentricity and flatness of orifice plates, for steps upstream of orifice plates, and for Venturi tube roughness all lead to lower costs.

A testing protocol for differential-pressure meters in general has been published. Now other differential-pressure meter standards need to be revised. Changes due to the revision of ISO 5167 and to other new work including that on Venturi tubes are required.

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Permission from Shell UK Exploration and Production to use their Venturi tubes is appreciated.

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NOTATION

C	Discharge coefficient
C_{water}	Mean discharge coefficient in water
$C_{m,p}$	Molar-heat capacity at constant pressure (J/(mol.K))
D	Pipe internal diameter (m)
d	Throat diameter (m)
d_{tap}	Diameter of throat pressure tapping (m)
e_c	Eccentricity (m)
F	Throat thickness (of a nozzle) (m)
H	Enthalpy (J/mol)
k	Uniform equivalent roughness (m)
P	Percentage shift in C
p	Static pressure (Pa)
p_1	Static pressure at upstream tapping (Pa)
p_2	Static pressure at downstream tapping (Pa)
Δp	Differential pressure (Pa)
R_a	Arithmetical mean deviation of the roughness profile (m)
R_u	Universal gas constant (J/(mol.K))
Re_D	Pipe Reynolds number
Re_d	Throat Reynolds number
Re^*	Venturi throat-tapping Reynolds number (equation (18))
T	Absolute temperature (K)
U	Percentage relative uncertainty
Z	Compressibility factor
β	Diameter ratio
ε	Expansiblity factor
κ	Isentropic exponent
λ	Friction factor
$\Delta\varpi$	Pressure loss (Pa)

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