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THE REVISION OF ISO 5167 - AN UPDATE

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THE REVISION OF ISO 5167 - AN UPDATE

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1 INTRODUCTION

ISO TC30/SC2 WG11, of which I am Convener, is responsible for producing a draft of ISO 5167 which takes into account the work on differential pressure flowmeters which has been done in recent years and will provide a thorough revision of ISO 5167-1 [1]. WG11 was established in 1996 and a Committee Draft was issued to SC2 for discussion at ISO TC30/SC2 in June 1998. A revised Committee Draft incorporating comments should be issued at the end of this year with the expectation that a revised International Standard will be published in 2001.

This paper brings the North Sea Workshop up to date on the current situation with regard to the revision of the standard. It describes the planned changes and gives at least an indication of the research on which the changes are based.

ISO 5167 will be 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). Many users will only require the general part and one other part. The most significant areas of change from the existing ISO 5167-1 are given below. It is likely that further changes will take place in the next three years.

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 [2]. Revised straight lengths based on an analysis of these data, initially undertaken by API but in which representatives of countries outside North America are now involved, will be included. The data downstream of two bends were taken with various distances between the two bends, and the required distances between pairs of bends will be included in the table. The table of required upstream straight lengths included in the Committee Draft of ISO 5167-2 discussed at SC2 in June 1998 is given here as Table 1. Only thermowells smaller than 0.03*D* will be permitted upstream of the orifice. 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 much smaller shifts. It will be recommended that if there is a possibility of a fitting creating severe swirl, as two bends in perpendicular planes with separation less than 5*D* or a header may do, a flow conditioner should be used.

In ISO 5167-1 a compliance test for flow conditioners will be included: using a primary device of diameter ratio 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 four situations:

a) in good flow conditions,

b) downstream of two bends in perpendicular planes (ratio of bend radius to pipe diameter approximately equal to 1.5; separation between curved portions of bends less than two pipe diameters)

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

d) in conditions of high swirl downstream of a device producing a high swirl (the device should produce a maximum swirl angle across the pipe of 25 - 30° 18*D* downstream of it or 20 - 25° 30*D* downstream of it).

Table 1 - Required straight lengths between orifice plates and fittings without flow conditioners

Values expressed as multiples of internal diameter D

Diameter ratio β	Upstream (inlet) side of the orifice plate																								Downstream (outlet) side of the orifice plate	
	Single 90° bend Two 90° bends in same plane ($S > 30D$)* Two 90° bends in perpendicular planes ($S > 15D$)*		Two 90° bends in the same plane: S-configuration ($10D \geq S$)*		Two 90° bends in the same plane: S-configuration ($30D \geq S > 10D$)*		Two 90° bends in perpendicular planes ($5D > S$)*		Two 90° bends in perpendicular planes ($15D \geq S \geq 5D$)*		Single 90° tee		Single 45° bend Two 45° bends in the same plane ($S > 22D$)*		Reducer $2D$ to D over a length of $1{,}5D$ to $3D$		Expander $0{,}5D$ to D over a length of D to $2D$		Full bore ball valve fully open		Abrupt symmetrical reduction		Thermometer pocket or well** of diameter $\leq 0{,}03D$		Fittings (columns 2 to 11) and the densitometer pocket	
1	2		3		4		5		6		7		8		9		10		11		12		13		14	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
0,20	6	6	10	10	10	10	***	***	19	19	9	9	****	****	5	5	16	8	12	6	30	15	5	3	4	2
0,40	18	9	10	10	13	10	50	30	44	19	9	9	30	18	5	5	16	8	12	6	30	15	5	3	6	3
0,50	30	16	30	10	18	16	95	60	44	19	19	9	30	18	6	5	18	9	12	6	30	15	5	3	6	3
0,60	44	30	44	30	30	18	95	50	44	19	29	18	30	30	9	5	22	11	14	7	30	15	5	3	7	3,5
0,67	44	30	44	30	44	30	95	50	44	30	36	30	44	30	12	6	27	14	18	9	30	15	5	3	7	3,5
0,75	44	35	44	30	44	30	95	65	44	30	44	30	44	30	22	11	38	19	24	12	30	15	5	3	8	4

*) S is the separation between the two bends measured from the downstream end of the curved portion of the upstream bend to the upstream end of the curved portion of the downstream bend.

**) The installation of thermometer pockets or wells will not alter the required minimum upstream straight lengths for the other fittings.

***) There are insufficient data for this configuration: the lengths for $\beta = 0{,}4$ should be used.

****) There are no data for this configuration: the lengths for $\beta = 0{,}4$ are at least sufficient.

NOTES:

1. The minimum straight lengths required are the lengths between various fittings located upstream or downstream of the orifice plate and the orifice plate itself. Straight lengths shall be measured from the downstream end of the curved portion of the nearest (or only) bend or of the tee or the downstream end of the conical portion of the reducer or expander.
2. Lengths for $\beta < 0{,}2$ can be taken to be equal to those for $\beta = 0{,}2$.
3. Most of the bends on which the lengths in this table are based had a radius of curvature equal to $1{,}5D$, but it may be used for bends with any radius of curvature.
4. Column A for each fitting gives lengths corresponding to 'zero additional uncertainty' values (see 5.2.3).
5. Column B for each fitting gives lengths corresponding to '0,5% additional uncertainty' values (see 5.2.4).

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 $\beta = 0.67$ the flow conditioner must pass the high-swirl test (d) for $\beta = 0.4$. This test for $\beta = 0.4$ is included because, although for non-swirling flow shifts in discharge coefficient increase with β, this is not necessarily true for swirling flow. Provided that the flow conditioner is geometrically similar for all pipe diameters this compliance test need only be passed for one diameter, and if it is passed for a range of Re_D greater than 3×10^6 it will be taken to have been passed for all $Re_D > 3 \times 10^6$. 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. Required distances between the primary device and flow conditioner and between upstream fittings and flow conditioner will be determined in the course of the tests since a flow conditioner will not pass for all distances. 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.

Tests were undertaken with orifice plates at SwRI to check that there are flow conditioners which meet these compliance tests with reasonable overall distances between fittings and orifice plate. These showed that there is a location for a 19 tube bundle of defined geometry at which the compliance test is satisfied provided that the overall length between orifice plate and tube bundle is at least 30*D*; similarly there is at least one location at which both the Gallagher and the K-Lab Laws Nova 50 E pass this test provided that the overall length between orifice plate and either flow conditioner was at least 18*D*. These lengths are not minimum upstream length requirements using the tube bundle or the flow conditioners; they are lengths at which successful tests were carried out. Measurements of required length are taken from the orifice face to the upstream fitting itself so that the weld neck is included within the straight length; for example the distance is measured to the downstream end of the curved portion of a bend. The results are presented without naming the conditioners in [2], but names are given in [3].

The installation requirements given in the current draft of Part 2 of API 14.3 [4] are almost identical to those proposed in ISO 5167. 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 is appropriate that the same process of revision should occur for the expansibility factor.

The existing equation for the orifice expansibility factor was derived by Buckingham [5] on the basis largely of data collected at tests in Los Angeles in 1929. Using these data Buckingham derived the equation:

$$\varepsilon_1 = 1 - (0.41 + 0.35\beta^4)\frac{\Delta p}{\kappa p_1}, \tag{1}$$

where ε_1 is the expansibility coefficient, 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.

It is significant that in analysing the data Buckingham neglected the effect of Reynolds number on the grounds that above a throat Reynolds number, Re_d, of 2×10^5 the discharge coefficient is constant. It is now known that the discharge coefficient continues to change with Re_d above this value. The work of Bean and Buckingham has nevertheless set a pattern for subsequent workers in this area.

As part of the EEC Orifice Project data were collected on expansibility factors. At NEL on the 100 mm (4-inch) pipe run 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 [6]

together with references to the individual data sets from all the laboratories whose data were used. This work followed that of Kinghorn [7]. Gaz de France collected data on the 100 mm (4-inch) pipe run 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 on the 100 mm (4-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 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 on a 50 mm (2-inch) pipe run 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

$$\varepsilon_1\big|_{\text{calc}} = \frac{(C\varepsilon_1)_{\text{measured}}}{C_{\text{water}}} \tag{2}$$

where $(C\varepsilon_1)_{\text{measured}}$ 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

$$\varepsilon_1\big|_{\text{calc}} = 1 - a - b\,\mathrm{f}\left(\frac{\Delta p}{p_1}, \kappa\right) \tag{3}$$

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_1 is given by

$$\varepsilon_1\big|_{\text{calc},2} = \frac{\varepsilon_1\big|_{\text{calc}}}{1-a} = 1 - \frac{b}{1-a}\,\mathrm{f}\left(\frac{\Delta p}{p_1}, \kappa\right) \tag{4}$$

which is equivalent to fitting $(C\varepsilon_1)_{\text{measured}}$ 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_1 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 [6] 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 $C\varepsilon_1$ 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 [8] was assumed.

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

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

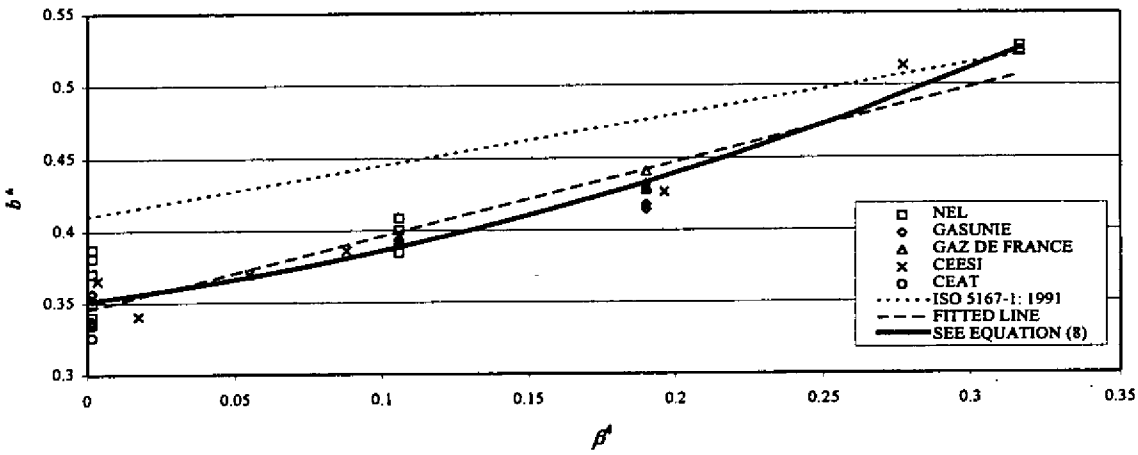
and

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

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 $\beta \leq 0.66$ 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:

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

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



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

$$\varepsilon_1 = 1 - (0.351 + 0.256\beta^4 + 0.93\beta^8)\left\{1 - \left(\frac{p_2}{p_1}\right)^{1/\kappa}\right\}. \tag{8}$$

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_1 is considered in Reference [6] and it is proposed that the relative uncertainty of ε_1 given in ISO 5167-2 should be

$$3.5\frac{\Delta p}{\kappa p_1} \text{ per cent.} \tag{9}$$

Because the present equation for ε_2 is in error and it is little used it will be omitted.

3 ECCENTRICITY LIMITS FOR ORIFICE PLATES

Debate on the requirements continues. There is a difference between the ISO and API standards, because there are small differences between the sets of data [9, 10] 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 proposed ISO requirements have been determined so that shifts using either set of data are less than 0.1 per cent. The present ISO requirements are sufficient to ensure good measurement. Under certain circumstances they are, however, more demanding than is necessary. To obtain a clause which describes the experimental data more accurately it is necessary to consider the eccentricity in its two components, parallel to a pressure tapping and perpendicular to the pressure tapping, since eccentricity parallel to a pressure tapping has more effect on the discharge coefficient measured using it than eccentricity perpendicular to it.

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, and 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 is determined. If there is to be no additional uncertainty e_{cl}, the component in the direction parallel to the pressure tapping, must for each pressure tapping be such that

$$e_{cl} \leq \frac{0.002\,5D}{0.1+2.3\beta^4}, \tag{10}$$

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

$$e_{cn} \leq \frac{0.005D}{0.1+2.3\beta^4}. \tag{11}$$

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

4 FLATNESS TEST FOR ORIFICE PLATES

Clause 8.1.2.1 in the existing ISO 5167-1 in which the slope of a straight line connecting any two points of its surface must be less than 0.5 per cent relative to a plane perpendicular to the centre-line of the orifice plate bore is too demanding since measurements can be made with the two points very close together. A simplified test based on that in the API standard (the current draft revision is reference [4]) is proposed 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 will ensure 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 can still be less than 1 per cent.

5 TEMPERATURE CORRECTION

Work at British Gas [11] and Gasunie [12] has shown 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 will be revised accordingly. The British Gas report covers a wide range of diameter ratios, whereas the Gasunie report has studied the flow through one orifice of diameter ratio 0.5 in more detail. The Gasunie report compares 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 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 upstream of the plate and the temperature measured at the usual temperature tapping location downstream of the plate can lead to significant error.

In many cases it can safely be assumed that the downstream and upstream temperatures of the fluid are the same at the differential pressure tappings. However, if the fluid is a non-ideal gas and 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 5*D* to 15*D* 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. Then the corresponding temperature drop from the upstream tapping to the downstream temperature location, ΔT, can be evaluated given the rate of change of T with respect to p at constant enthalpy:

$$\begin{aligned}\Delta T &= \left.\frac{\partial T}{\partial p}\right|_H \Delta\varpi \\ &= \frac{R_g T^2}{p c_p}\left.\frac{\partial Z}{\partial T}\right|_p \Delta\varpi,\end{aligned} \tag{12}$$

where T is the absolute temperature, R_g is the universal gas constant, c_p is the heat capacity at constant pressure and Z is the compressibility factor.

6 PIPE ROUGHNESS LIMITS UPSTREAM OF ORIFICE PLATES

New roughness limits for pipework upstream of orifice plates have been included. 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 new discharge coefficient equation 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:

$$\begin{aligned}C &= 0.5945 + 0.0157\beta^{1.3} - 0.2417\beta^8 \\ &\quad + 0.000514(10^6\beta / Re_D)^{0.7} \\ &\quad + (3.134 + 4.726A_p)\beta^{3.5}\max\{\lambda, 0.1704 - 35(Re_D / 10^6)\}\end{aligned} \tag{13}$$

where

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

Reference [13] gives the details.

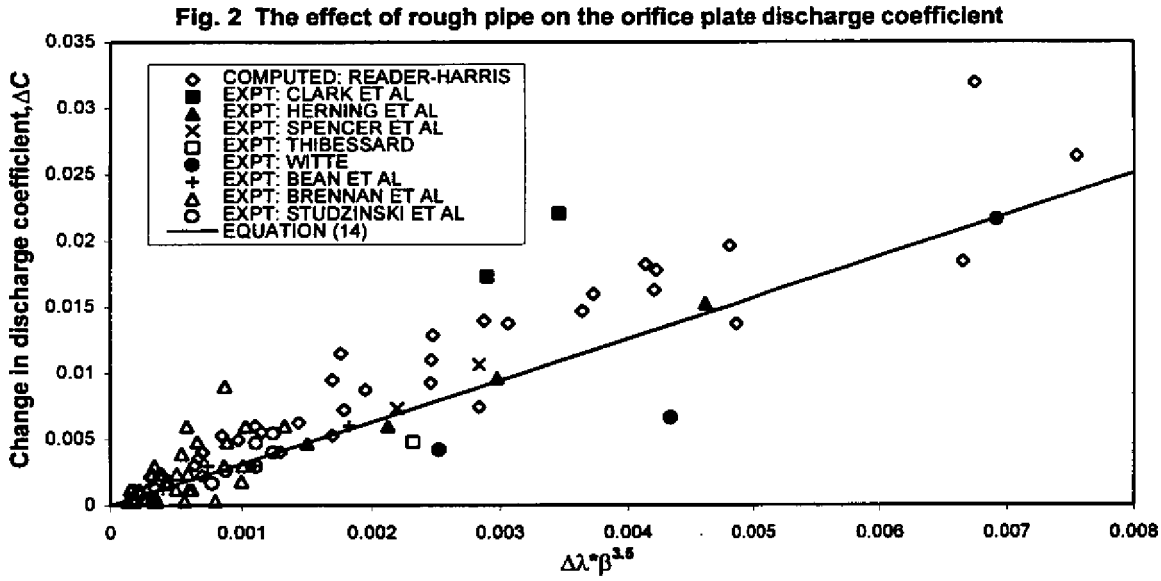
This gives the change in friction factor due to roughness, ΔC_{rough}, as

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

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 [14] 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 [14] 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 mean 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:

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

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 new 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 2. The values of friction factor are consistent with the values of relative roughness if the Colebrook-White Equation (see Reference [15]) is used.

Table 2 - 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
$10^4k/D$	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 uncertainty of C. It is assumed here that the percentage shift, P, should not exceed

$$\begin{array}{ll} 0.5\beta & \beta \le 0.5 \\ 0.25 & 0.5 < \beta \le 0.6 \\ 0.5(1.667\beta - 0.5) & 0.6 < \beta \le 0.71 \\ 1.13\beta^{8.5} & 0.71 < \beta \end{array} \qquad (16)$$

This restriction ensures that for $\beta \le 0.5$, where other sources of error are dominant, $P/U < \beta$; for $0.5 < \beta \le 0.71$, 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 2, 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 has been 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. 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 use of R_a will apply for all primary devices. k will still be mentioned.) The maximum and minimum values of R_a/D for pipes upstream of orifice plates are given in Tables 3 and 4.

Table 3 - Maximum value of $10^4R_a/D$

Re_D / β	$\le10^4$	3×10^4	10^5	3×10^5	10^6	3×10^6	10^7	3×10^7	10^8
≤0.20	32	32	32	32	32	32	32	32	32
0.30	32	32	30	24	19	17	15	14	13
0.40	21	15	10	7.2	5.2	4.1	3.5	3.1	2.7
0.50	11	7.7	4.9	3.3	2.2	1.6	1.3	1.1	0.92
0.60	5.6	4.0	2.5	1.6	1.0	0.73	0.57	0.46	0.36
≥0.65	4.2	3.0	1.9	1.2	0.78	0.56	0.43	0.34	0.26

Table 4 - Minimum value of $10^4R_a/D$ (where one is required)

Re_D / β	$\le3\times10^6$	10^7	3×10^7	10^8
≤0.50	0.0	0.0	0.0	0.0
0.60	0.0	0.0	0.003	0.004
≥0.65	0.0	0.013	0.016	0.012

It will be stated that the requirements of this section are satisfied in both the following cases:

$1\ \mu m \le R_a \le 6\ \mu m$, $D \ge 150$ mm, $\beta \le 0.6$ and $Re_D \le 5 \times 10^7$.
$1.5\ \mu m \le R_a \le 6\ \mu m$, $D \ge 150$ mm, $\beta > 0.6$ and $Re_D \le 1.5 \times 10^7$.

The roughness shall meet requirements given above for 10*D* upstream of the orifice plate. The roughness requirements relate to the orifice fitting and the upstream pipework.

The minimum values represent exceedingly smooth pipes. The tables prescribe $k/D \leq 0.01$ even if the calculated value is higher. The limits for $Re_D = 10^4$ can be used for smaller Re_D. 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, an unsurprising agreement since the limits in ISO 5167-1:1991 were probably derived from data collected at around that Reynolds number.

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.

7 PULSATION EFFECTS

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

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

where $\overline{\Delta p}$ is the time-mean value of the differential pressure and $\Delta p'_{\mathrm{rms}}$ is the r.m.s. value of $\Delta p'$, the fluctuating component of the pressure. This clause is consistent with ISO/TR 3313 [16]. $\Delta p'_{\mathrm{rms}}$ 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.

8 THROAT THICKNESS FOR NOZZLES

For long radius high-ratio nozzles 9.2.2.5 of ISO 5167-1: 1991 states that the thickness *F* of the throat shall be between 3 mm and 13 mm. However, it has been pointed out that for large pipe diameter this may be too small to prevent distortion due to machining stresses. ASME MFC-3M: 1989 [17] gives (in ISO 5167-1 nomenclature) $2F \leq D - d - 6$ where *F*, *D* and *d* are expressed in millimetres. There is also a problem for small pipe diameters: if $D = 50$ mm and $\beta = 0.8$ it is not possible to satisfy 9.2.2.4 and 9.2.2.5.

Following Japan's suggestion no maximum for *F* in the revision of 9.2.2.5 will be specified. The maximum can be deduced from 9.2.2.4 and will be the same requirement as was given by ASME MFC-3M. 9.2.2.5 will also be revised to take account of small pipe diameters and the second half of the draft clause 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.'

9 STRAIGHT LENGTHS UPSTREAM OF VENTURI TUBES

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

Three Venturi tubes, of diameter ratios 0.4, 0.6 and 0.75, were calibrated 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. 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.

References [18] and [19] describe the Venturi tubes and pipework used for this project and the measurements of installation effects obtained with them. All the experimental data were taken in water: over the complete database the throat Reynolds number, Re_d, was always in the range 1.9×10^5 - 2.0×10^6. Three classical Venturi tubes with a machined convergent section had been made by ISA Controls Ltd for Shell UK Exploration and Production to meet the requirements of ISO 5167-1. The pipe diameter was 154.0 mm. The divergent angle was 7½°. The pressure tappings were 4 mm in diameter and were connected in 'triple-tee' arrangements. Lengths of pipe upstream of the Venturi tubes were machined to ensure there were no significant steps near the Venturis. The lengths of pipe and the Venturi tubes were dowelled to ensure concentricity; O rings were used to ensure that there would not be recesses or protruding gaskets.

The three Venturi tubes were initially calibrated with 41*D* of straight pipe upstream of the upstream pressure tapping. Upstream of the straight pipe was a tube bundle flow conditioner. They were then calibrated immediately before the installation effects tests described here. At the end of all the calibrations (except for the calibration 17.8*D* downstream of a single bend which took place one week after the final baseline) all three Venturi tubes were recalibrated again. For each Venturi tube the slope of the calibration was positive for at least one calibration and negative for at least one calibration; so the mean value was used for each calibration. The mean value of discharge coefficient used as the baseline for each Venturi tube was the average of the mean values from the second and third calibrations, which preceded and followed the installation-effects calibrations.

The shift in discharge coefficient for a particular installation condition was determined by subtracting the mean discharge coefficient obtained with that installation from the baseline discharge coefficient. 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.

A contraction from 203 mm to 152 mm was installed upstream of each Venturi tube: the contraction was conical with a length of conical section of 350 mm, giving an included half-angle of about 4°. Upstream of the contraction there was 4.7 m of straight 203 mm pipe (23 diameters of 203 mm pipe) preceded by a tube bundle. The mean shifts are presented in Figure 3.

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

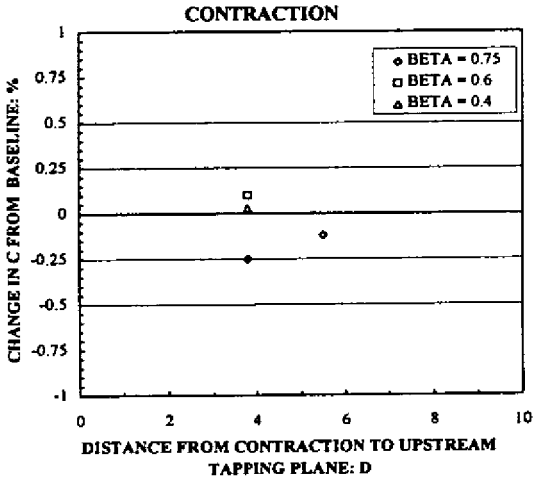
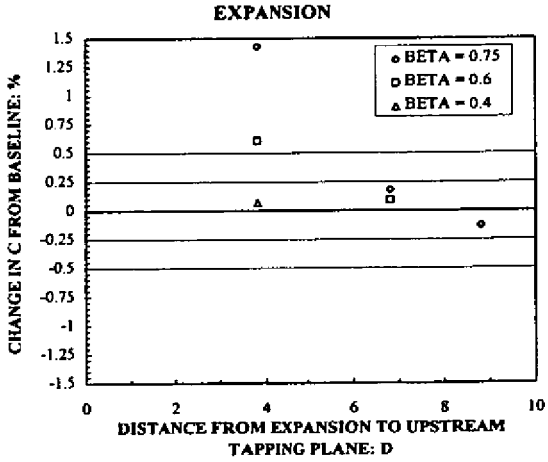


FIG. 4 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF A 102 mm - 152 mm EXPANSION



An expansion from 102 mm to 152 mm was installed upstream of each Venturi tube: the expansion was conical with a length of conical section of 375 mm, giving an included half-angle of about 4°. Upstream of the expansion there was 4.9 m of straight 102 mm pipe (48 diameters of 102 mm pipe) preceded by a tube bundle. The mean shifts are presented in Figure. 4. The shifts are positive because the velocity profile is peaked.

Two bends in perpendicular planes were installed upstream of each Venturi tube. Upstream of the upstream bend there was 26*D* of straight pipe preceded by a Zanker flow conditioner. The mean shifts are presented in Figure 5. For all the calibrations downstream of bends 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.

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

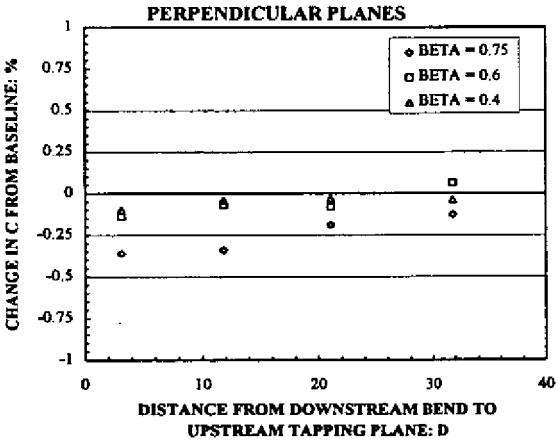
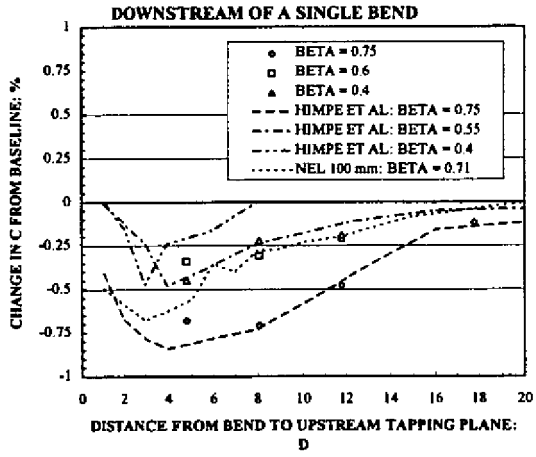


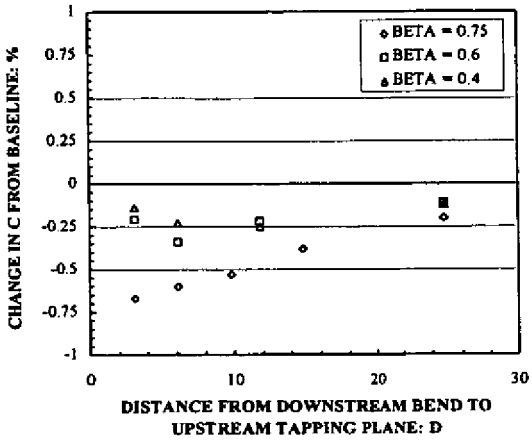
FIG. 6 CHANGE IN C FOR VENTURI TUBES DOWNSTREAM OF A SINGLE BEND



A single bend was installed upstream of each Venturi tube. Upstream of the bend there was the same 26*D* of straight pipe preceded by a Zanker flow conditioner as there was for the tests with two bends in perpendicular planes. The mean shifts are presented in Figure 6. These calibrations are in good agreement with the data of Himpe et al [20], particularly for the larger values of β, and those of NEL (1985) quoted by Kochen et al [21]. Both Himpe et al and NEL had a pipe diameter of 100 mm.

Two bends in the same plane were installed upstream of each Venturi tube. Upstream of the upstream bend there was 21*D* of straight pipe preceded by a tube bundle flow conditioner. The mean shifts are presented in Figure 7.

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



Two conditioners, the Spearman (NEL) conditioner and the tube bundle, were tested downstream of two bends in perpendicular planes and the single bend and upstream of the Venturi tubes. The results are given in References [18] and [19]. The Spearman flow conditioner gave excellent results.

On the basis that straight lengths should be calculated so that shifts in discharge coefficient should be smaller in magnitude than 0.25 per cent or alternatively, for greater uncertainty, 0.5 per cent, the proposed revision of Table 2 of ISO 5167-1 was obtained. It is given as Table 5. The column relating to upstream valves has been left unchanged because of a lack of new data. To provide lengths for values of β other than those measured an appropriate dependence of shift on β based on both the general pattern of the data and the specific installation was assumed. Given the lack of multiple data sets for combinations of bends it was decided 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 discharge coefficient shift. Moreover, it hardly seems sensible to encourage the installation of two bends in perpendicular planes upstream of a meter. The straight length is defined as it was for the tests; columns A and B are defined as in Table 1.

Table 5 - Required straight lengths for classical Venturi tubes

Values expressed as multiples of internal diameter D

Diameter Ratio β	Single 90° bend*)		Two or more 90° bends in the same plane or different planes*)		Reducer 1,33*D* to *D* over a length of 2,3*D*		Expander 0,67*D* to *D* over a length of 2,5*D*		Full bore ball or gate valve fully open	
1	2		3		4		5		6	
	A	B	A	B	A	B	A	B	A	B
0,30	8	3	8	3	4	4	4	4	2,5	2,5
0,40	8	3	8	3	4	4	4	4	2,5	2,5
0,50	9	3	10	3	4	4	5	4	3,5	2,5
0,60	10	5	10	5	4	4	6	5	4,5	2,5
0,70	14	10	19	10	4	4	7	6	5,5	3,5
0,75	16	12	22	12	4	4	7	6	5,5	3,5

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 [22] 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 existing lengths required by ISO 5167-1 are similar to those obtained by Pardoe [23]: he obtained positive shifts in C downstream of a single bend and stated that 6D was sufficient downstream of a single bend even for β = 0.8, whereas downstream of two bends in perpendicular planes even 30*D* would not be sufficient (at least for β > 0.55). An explanation for the difference between Pardoe's data and subsequent data is still required.

10 SURFACE ROUGHNESS OF VENTURI TUBES

According to ISO 5167-1 the surface roughness for the throat of a classical Venturi tube is that $R_a \leq 10^{-5}d$; so for a 50 mm β = 0.4 Venturi tube a surface roughness of $R_a \leq 0.2$ µm is required, which is very expensive to manufacture and unlikely to last long in service in the North Sea. Moreover, according to ISO 5167-1 subsonic nozzles only require $R_a \leq 10^{-4}d$; so Venturi tubes must be 10 times smoother than subsonic nozzles are required to be. NEL has recently purchased Venturi tubes from manufacturers who supply the North Sea and has found that Venturi tubes usually have roughnesses outside the existing standard but within $R_a \leq 10^{-4}d$. It has been agreed that the required surface finish of the throat of Venturi tubes should be such

that $R_a \leq 10^{-4}d$. Modern data to verify the existing values of discharge coefficients and to extend the results to higher Reynolds numbers are being taken using Venturi tubes that are typical of current practice.

11 CONCLUSIONS

ISO 5167 is being revised to take into account the work on differential pressure flowmeters which has been done in recent years. This paper describes many of the planned changes: revised straight lengths upstream of orifice plates and Venturi tubes, a new expansibility equation for orifice plates, revised limits for eccentricity and flatness of orifice plates and for pipe roughness upstream of orifice plates. More accurate temperature correction, clarification on acceptable pulsations, and more practical throat thickness for nozzles and Venturi tube surface roughness are all included. As regards uncertainty, minor changes to give consistency with the Guide will be included. Other smaller changes will also be made.

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NOTATION

C	Discharge coefficient
c_p	Heat capacity at constant pressure (J/(mol.K))
D	Pipe internal diameter (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_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_g	Universal gas constant (J/(mol.K))
Re_D	Pipe Reynolds number
Re_d	Throat Reynolds number
S	Bend separation (m)
T	Absolute temperature (K)
U	Percentage relative uncertainty
Z	Compressibility factor
β	Diameter ratio
ε_1	Expansiblity factor
κ	Isentropic exponent
λ	Friction factor
$\Delta\varpi$	Pressure loss (Pa)

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