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Flow Measurement

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CONTENTS

INTRODUCTION

Page

Part ONE : Scope

1. General 5
2. Types of differential-pressure device 5
3. Coefficients, tolerances and overall accuracy 6
4. Methods of use 5
5. Requirements respecting fluid to be measured 6
6. Flow requirements 6
7. Correction factors for application to meter reading 6

Part TWO : Standard Pressure Difference Devices

A. ISA 1932 Nozzle

8. Size limitation 7
9. Construction 7
10. Accuracy of manufacture 7
11. Pressure tappings 7
12. Coefficients, corrections and tolerances 7

B. Orifice Plate with corner taps

13. Size limitation 7
14. Construction 7
15. Accuracy of manufacture 7
16. Pressure tappings 7
17. Coefficients, corrections and tolerances 8

C. Orifice Plate with D and D/2 taps

18. Size limitation 8
19. Construction 8
20. Accuracy of manufacture 8
21. Pressure tappings 8
22. Coefficients, corrections and tolerances 8

D. Venturi Tubes

23. Size limitation 8
24. Construction 8
25. Accuracy of manufacture 8
26. Pressure tappings 8
27. Coefficients, corrections and tolerances 9

Part THREE : Standard Methods of Installation of Pressure Difference Devices 9

Page

Part FOUR : Methods of Measurement, Application and Data required

A. Pressure Pipes from Differential Producer to meter 10
- *a.* Liquids 10
- *b.* Steam 10
- *c.* Air and gases 10
- *d.* Long pressure pipe runs 11
- *e.* Sealing chambers 11

B. Measurement of Pressure Difference 11

C. Measurement of Density 11

D. Viscosity of Fluids 12

Part FIVE : Working Formulæ 12

Part SIX : Reynolds Number 14

Part SEVEN : Coefficients of Discharge and Tolerances 15

Part EIGHT : Special cases of Orifice or Nozzle Installation 16

Part NINE : Choice and use of Pitot Tubes for Flow Measurement 18

APPENDICES

A. List of symbols 21

B. Derivation of working formulæ 23

C. Loss of pressure due to orifices, nozzles and venturi tubes 26

D. Moisture in gases 26

E. Viscosity tables and conversion factors 27

F. Correction factors for application to meter readings 33

G. Application of the working formulæ and examples 34

H. Density of steam, tables 43

NOTE : Sixteen Data Sheets are listed on page 3.

DATA SHEETS

1. STANDARD NOZZLES AND ORIFICES

Fig. 1. Standard I S A nozzle when $m < 0.45$
Fig. 2. Standard I S A nozzle when $m > 0.45$
Fig. 3. Orifice plate with corner tappings
Fig. 4. Orifice plate with D and $\frac{D}{2}$ tappings

2. STANDARD VENTURI TUBES

Fig. 5. Conical type Venturi tube
Fig. 6. Nozzle type Venturi tube

3. STRAIGHT PIPE REQUIREMENTS AFTER VARIOUS IRREGULARITIES

Fig. 7. Fully open sluice valves
Fig. 8. Right angle bend : two right angle bends in same plane
Fig. 9. Three bends at right angles : two bends at right angles : branches
Fig. 10. Fully open globe valves

4. PRESSURE PIPES TO METER

Fig. 11. Steam and liquids—meter above main
Fig. 12. Steam and liquids—meter below main
Fig. 13. Condensation chambers for non-horizontal steam main
Fig. 14. Condensation chambers
Fig. 15. Air and gases—meter above main
Fig. 16. Air and gases—meter below main

5. COEFFICIENTS AND TOLERANCES—ISA 1932 NOZZLE AND I S A NOZZLE VENTURI

Fig. 17. *a.* Coefficient of discharge ' C '
Fig. 17. *b.* Corrections for R_d.
Fig. 17. *c.* Corrections for ' D '
Fig. 17. *d.* Tolerance : basic
Fig. 17. *e.* Tolerance : extra for ' D '

6. COEFFICIENTS AND TOLERANCES—ORIFICE WITH CORNER TAPS

Fig. 18. *a.* Coefficient of discharge ' C '
Fig. 18. *b.* Corrections for R_d.
Fig. 18. *c.* Corrections for ' D '
Fig. 18. *d.* Tolerance : basic
Fig. 18. *e.* Tolerance : extra for ' D '

7. COEFFICIENTS AND TOLERANCES—ORIFICE WITH D AND $\frac{D}{2}$ TAPS

Fig. 19. *a.* Coefficient of discharge ' C '
Fig. 19. *b.* Corrections for R_d.
Fig. 19. *c.* Corrections for ' D '
Fig. 19. *d.* Tolerance : basic
Fig. 19. *e.* Tolerance : extra for ' D '

8. COEFFICIENTS AND TOLERANCES—VENTURI CONICAL TYPE

Fig. 20. *a.* Coefficient of discharge ' C '
Fig. 20. *b.* Corrections for R_d.
Fig. 20. *c.* Corrections for ' D '
Fig. 20. *d.* Tolerance : basic
Fig. 20. *e.* Tolerance : extra for ' D '

9. LONG RANGE mE CALCULATING CHART

Fig. 21

10. THERMAL EXPANSIONS OF PRESSURE DIFFERENCE DEVICES

Fig. 22. Temperature correction for the diameter of the orifice

11. VALUES OF ε FOR ORIFICES AND NOZZLES

Fig. 23. *a.* Values of expansion coefficients for diatomic gases
Fig. 23. *b.* Values of expansion coefficients for superheated steam

12. COMPRESSIBILITY CORRECTION K

13. MOISTURE CORRECTION FACTOR N

Fig. 24

14. PITOT TUBES

Fig. 25. N.P.L. hemispherical head
Fig. 26. N.P.L. sharp-ended head
Fig. 27. Single hole Pitot tube with separate static tapping
Fig. 28. Illustrating test points required for 96 in. × 48 in. duct
Fig. 29. Showing additional points to be taken in corner rectangle

15. PITOT TUBES—CIRCULAR MAINS

Fig. 30. Methods of making traverses.

16. PRESSURE LOSS.

Fig. 31. Approximate nett pressure loss curves for nozzles, orifices and Venturi tubes giving equal differential at equal flows

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THIS BRITISH STANDARD having been approved by the Solid Fuel Industry Committee and endorsed by the Chairman of the Engineering Divisional Council was published under the authority of the General Council on 13th February, 1943.

In order to keep abreast of progress in the industries concerned, the British Standards are subject to periodical review. Suggestions for improvements will be recorded and in due course brought to the notice of the Committees charged with the revision of the standards to which they refer.

A complete list of British Standards, indexed and cross-indexed for reference, will be found in the Institution's Handbook.

CO-OPERATING ORGANISATIONS

The Solid Fuel Industry Committee under whose supervision this British Standard was prepared consists of representatives from the following Government Departments and Scientific and Industrial Organisations :

Admiralty
Board of Trade
Fuel Research Board
Mines Department
Ministry of Health
*British Cast Iron Research Association
British Chemical Plant Manufacturers' Association
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*British Electrical and Allied Manufacturers' Association
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*British Iron and Steel Federation
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*Institution of Gas Engineers
Institution of Heating and Ventilating Engineers
*Institution of Mechanical Engineers
*Iron and Steel Institute
London and Counties Coke Association
Mining Association of Great Britain
National Association of Heating and Ventilating and Domestic Engineering Employers
*Society of British Gas Industries
*Society of Chemical Industry

The Government Departments and Scientific Industrial Organisations marked with an asterisk in the above list, together with the following, were directly represented on the Committee entrusted with the preparation of this British Standard :

H.M. Fuel Research Station
Ministry of Supply
National Physical Laboratory
Association of Consulting Engineers
British Compressed Air Society
British Refractories Research Association
British Scientific Instruments Research Association
Institute of Metals
Institute of Petroleum
Institution of Civil Engineers
Institution of Electrical Engineers
Society of Glass Technology
Makers of Scientific and Industrial Instruments

BRITISH STANDARD CODE FOR THE MEASUREMENT OF FLUID FLOW IN PIPES AND DUCTS

Introduction

The purpose of this British Standard Code is to set out fully the conditions governing the design, installation and use of standard pressure-difference devices in order to obtain consistent and generally-acceptable results within a specified tolerance, from commercial instruments. The 'secondary devices' or flowmeters are not dealt with, since if all the relevant rules of this Code are followed, the slight additional tolerance required for any individual flowmeter, under any given conditions, can easily be established by a manometric check. (See paras. 45 to 47.)

Methods of volumetric measurement of fluid flow by weirs and notches are dealt with in B.S. 599. It is intended to incorporate Sections dealing with these methods in the first revision of this Code.

The methods set out in this Standard Code are not intended to replace the methods laid down in the following British Standards which were prepared for application to specific fluids or for use under specific and restricted conditions :

B.S. 599. Pump Tests.
B.S. 707. Testing of Mine Fans.
B.S. 722. Borehole and Well Pump Tests.
B.S. 723. Sewage Pump Tests.
B.S. 724. Vaporizing Liquid Pump Tests.
B.S. 726. Measurement of Air Flow and the Free Air delivered by Compressors.
B.S. 752. Acceptance Tests for Steam Turbines.
B.S. 848. Testing of Fans for General Purposes.

In the compilation of this Code references have been made to the following existing reports :

V.D.I. Rules (Germany) DIN 1952, Fourth edition, 1938.
A.S.M.E. Fluid Meters Report (U.S.A.), Fourth Edition.
I S A (International) Bulletins 9 and 12 (Nozzles and Orifices).
I S A Bulletin 30 (Venturi Tubes).
V.D.E. (Germany) Bulletin 156, Flow Calculations.
U.N.I. (Italy) Bulletin 0183 (Nozzles and Orifices).
U.N.I. (Italy) Bulletin 0207 (Venturi Tubes).
Report of the Heat Engine Trials Committee of the Institution of Civil Engineers.
Reports of the British Compressed Air Society.

See Amdt. No. 1

The fullest acknowledgment is tendered to the authors of the codes, and particular thanks are given to the American Society of Mechanical Engineers and others for their co-operation in deciding small points of discrepancy between the British draft and their codes, and for communicating the results of their most recent researches. None of the material in this Code will be found to conflict to any serious extent with any of the codes mentioned.

A very full bibliography is given in the codes mentioned, which it has not been thought necessary to repeat in this specification.

After a review of the existing sources of information, the V.D.I. rules, DIN 1952, were selected as a pattern, and the arrangement of paragraphs in any particular section has been left unchanged in order to facilitate cross reference.

The order of the sections, however, has been changed, and any points or values considered not internationally substantiated have been omitted, or subjected to increased tolerances. Multiplying factors to compensate for avoidable faults such as bluntness of orifice edge, non-standard installation conditions, etc., have been excluded.

At the outbreak of the war the preparation of this Code was well advanced but it had not been finalised. The Committee would have wished to devote further time to its completion but the exigencies of the war have, however, made this impossible. In view of the numerous requests received for its publication it has been decided to issue the present provisional Code in the hope that industrial users will themselves remedy its defects and make suggestions for its improvement to the Committee which will resume its labours as soon as international conditions permit.

It is hoped that this Code will meet the needs of a large number of users, and that by continued co-operation between the countries concerned, the few points still under discussion may be decided and embodied herein.

Part One : Scope

GENERAL

1. It often occurs that the only practical method of metering pipe flows is to measure the pressure drop caused by a throttling device (nozzle, orifice plate, or Venturi tube) placed in the pipe line. Moreover, it is an accurate and convenient means in general use with any flowing liquids, gases and vapours at any pressures and temperatures.

The method of measurement depends on the change of pressure energy which occurs when a fluid flows through

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a path of varying cross section. The law of energy (Bernoulli) gives a definite relationship between the pressure energy and the flow energy. To make use of it, a form of throttling device is inserted in the main which reduces its cross sectional area, and thereby causes the conversion of a portion of the pressure energy into kinetic energy as the fluid accelerates.

TYPES OF DIFFERENTIAL-PRESSURE DEVICE

2. This code makes it possible, if the rules are followed, to use in circular pipes without calibration the following methods of measurement :

1. I S A 1932 nozzle.
2. Orifice plate with corner taps.
3. Orifice plate with D and D/2 taps.
4. Venturi tube, I S A 1932 nozzle type.
5. Venturi tube, conical type.

Measurement by Pitot tube is also specified, though in that case a calibration is implied in the actual method of use.

It will be found that coefficients are given for orifices up to $m=0{\cdot}70$, and for nozzles and Venturis, up to $m=0{\cdot}55$. These ratios may be exceeded when the installation is specially calibrated against a second orifice, nozzle, etc., of a ratio coming within the code, or under other conditions by agreement between the parties concerned. Differential pressure devices exceeding the specified ratios, however, shall not be referred to as 'British Standard'.

The following are excluded from this code :

Circular eccentric orifices.

Thick plate orifices.

Orifices with 'flange taps' (taps at a fixed non-proportional distance from the plate).

Chord or segmental orifices.

Orifices with 'vena contracta' taps (though it should be noted that the discharge coefficient for such tappings is the same as for the 'D and D/2 taps' up to $m=0{\cdot}5$.)

Elliptical A.S.M.E. nozzles.

COEFFICIENTS, TOLERANCES AND OVERALL ACCURACY

3. The coefficients to be used are given for each type of differential pressure device, in the particular section devoted to that type, together with the tolerances applicable. The tolerances there given do not apply to the total measurements, but only to the flow coefficients. In calculating the overall accuracy of any given measurement, reference must also be made to the following paragraphs :

Differential pressure	Paras. 45 to 47
Density	Paras. 48 to 50
Expansibility	Para. 63.

For each type of differential pressure device, two tolerances are given (one for basic coefficient and one for pipe size) so that the overall accuracy may depend on as many as five separate tolerances. If these tolerances are expressed as percentages, by x_1, x_2, x_3, etc., the total probable error (but not the maximum possible error) is given by :

$$x = \sqrt{x_1^2 + x_2^2 + x_3^2} \text{ .....etc.} \qquad (1)$$

care being taken that for density and differential pressure the corresponding *flow* tolerances are used.

At this point reference should be made to the examples in Appendix G showing the application of these tolerances to flow calculations.

METHODS OF USE

4. The Code deals not only with the differential pressure devices, but also with conditions of installation, laying of pressure piping, use of drains, condensation chambers, and all details necessary to ensure accurate measurement.

REQUIREMENTS RESPECTING THE FLUID TO BE MEASURED

5. Flow measurements can only be undertaken when the density of the flowing substance in front of the throttling device is accurately known, for this always affects the result.* On the other hand it suffices, as a rule, to know the viscosity approximately, in order to calculate approximately the Reynolds number for the flow to be measured.

To make use of the rules it is essential that the fluid to be measured should be either a homogeneous liquid free from solid matter or gas, or else a homogeneous gas or vapour free from liquid or solid matter, as it flows through the throttling device. This requirement shows that flowing liquids can only contain gases or solids in the dissolved state, and that no separation of the gases or vapours may occur. Solid undissolved substances, as in a slurry, must not be present.

This stipulation is of special importance in the measurement of gases and vapours near saturation point, and of liquids near their boiling point. In cases of doubt it is necessary to make sure that the vapour remains superheated in spite of any change in pressure at the throttling device and that no liquid is vaporized.

The rules also apply to colloidal solutions, if the degree of dispersion and the physical properties are such that the solution, e.g. milk, only differs slightly from a single phase liquid. There is not sufficient experience yet on which to base reliable rules for unevenly dispersed solutions.

FLOW REQUIREMENTS

6. The pipe must be flowing full bore at all parts near the throttling device.

The flow equations apply only to steady flows; there must be no flow pulsations such as are produced, for example, by piston type pumps. The errors which arise in the measuring of unsteady flows are further dealt with in Part 8.

CORRECTION FACTORS FOR APPLICATION TO METER READING

7. The characteristics of a fluid under working (metering) conditions may vary from those under basic or the designed conditions and the scale markings must be multiplied by a factor F to reduce them to the standard conditions for which the scale is graduated. (See Appendix F.)

* Except when the flow is measured in volume units and the metered fluid is used as the manometric fluid.

Part Two : Standard Pressure Difference Devices

Referred to as 'p.d.' devices

A. I S A 1932 NOZZLE

8. *Size limitation.* The use of the type of nozzle specified below is limited to mains not less than 2 in. internal diameter and values of the ratio 'm' $\left(\frac{\text{Area of throat}}{\text{Area of pipe upstream}}\right)$ between 0·2 and 0·55.

9. *Construction.* This device was originally known as the I.G. Nozzle and is identical with the German and Italian Standard Nozzles. Tests have been made for sizes between 2 in. and 20 in. The key dimensions are given in Figs. 1 and 2 (Data Sheet 1) in terms of the diameters 'D' of the main and 'd' of the throat, and these are sufficient to determine the construction of the nozzle for any given condition. The inner contour has two radii, one d/3 tangential to the cylindrical part and the other 0·2d tangential to the flat face, which run smoothly into each other. When m reaches the value 0·45 the curved part of the nozzle extends beyond the diameter 'D' of the main. In such a case, a flat surface is to be turned on the upstream side as shown in Fig. 2.

10. *Accuracy of manufacture.* The inner contour should be checked by means of a template, the permissible error in radii being 10 per cent for values of $\frac{d^2}{D^2}$ ('m') less than 0·3, and 3 per cent for values of 'm' above 0·3. The contour must however be smooth in any case, especially with the smaller sizes of nozzle.

The diameter d of the cylindrical portion of the nozzle must be accurate within plus or minus $0{\cdot}001d$ for values of m below 0·45 and within $0{\cdot}000\,5d$ for values of m above 0·45. In checking this dimension, the value of 'd' is to be taken as the mean of four diameters measured at the outlet end, and the variation from this mean of any diameter measured here or at the inlet end must not exceed the limit $0{\cdot}001d$. Hand finishing of the cylindrical portion should, therefore, be avoided.

The outside contour of the nozzle has no limitations other than the necessity for ensuring an unobstructed passage to the downstream tapping. The projection shown at the downstream end is merely for the protection of the end from mechanical damage, and is optional.

11. *Pressure tappings.* The pressure tappings may take the form of annular rings with slots opening into the main at each side of the flange of the nozzle, or of single holes (corner taps) drilled through the flanges of the main close to the nozzle flange. The dimension of 0·03D is intended to cover the general case with either of these arrangements, but where 'm' is 0·35 or smaller, gap widths for annular tappings may lie between 0·05 in. and 0·2 in. while single hole tappings may be between 0·25 in. and 0·5 in. diameter, or 0·1D, whichever is the lesser. When 'm' is greater than 0·35 the limit 0·03D applies.

This means in effect that for small pipes and large values of 'm' annular tappings are necessary, since single pressure connections must not be less than 0·25 in. diameter in ordinary circumstances, and not less than 0·5 in. when rising vertically from mains carrying condensing fluids.

12. *Coefficients, corrections and tolerances* are given in Figs. 17 *a.* to *e.*, Data Sheet 5.

B. ORIFICE PLATE WITH CORNER TAPS

13. *Size limitation.* This orifice may be used in any pipe not less than 2 in. internal diameter provided that 'm' does not exceed 0·7 and 'd' is not less than 0·25 in.

14. *Construction.* Fig. 3 (Data Sheet 1), shows the essential features and dimensions of the standard arrangement; the orifice is circular, and is concentric with the pipe. The edge of the orifice on the upstream face must be absolutely sharp: no rounding or burring is permissible, particularly when 'd' is below 3 in. The downstream edge is less important but no burrs must encroach into the bore of the orifice even on this side. A sharp edge may conveniently be made by taking a fine cut across the upstream face of the plate, from the centre outwards, after the orifice has been bored. Cleaning the plate with emery cloth is not permissible, owing to the danger of rounding the edge.

If the thickness of the plate is greater than 0·05d the downstream edge of the orifice must be bevelled as shown in Fig. 3 (Data Sheet 1). The necessity for bevelling can be avoided by using a thin plate if its strength and stiffness are adequate.

The orifice plate should be constructed of some material which will not be corroded by the fluid to be measured, otherwise the edge would rapidly become rounded to a sufficient extent to interfere with accurate measurement. The remarks in this paragraph apply equally to D and $\frac{D}{2}$ taps.

15. *Accuracy of manufacture.*

Bore. Below $m = 0{\cdot}45$, $\pm\, 0{\cdot}001\ d$
Above $m = 0{\cdot}45$, $\pm\, 0{\cdot}000\,5\ d$

Edge. To be perfectly square and free from all burrs or rounding.

Face. To be smooth for a diameter of at least 2*d* and to be flat within 0·02 in. per inch of pipe diameter when clamped between flanges.

16. *Pressure tappings.* It is common practice to have a number of pressure tappings distributed round the circumference of the pipe, and connected to an annular chamber or 'piezometer' ring. This arrangement has the advantage that the accuracy of measurement is less susceptible to disturbed flow conditions, due to bends or valves immediately preceding the orifice plate, and also that accidental obstruction of one pressure hole does not seriously interfere with the measurement. This arrangement, however, is not essential, and a single pressure hole is often used where the conditions are suitable. The arrangement described applies to each of the two pressure measurements, namely upstream and downstream.

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The stipulations with regard to pressure holes stated in para. 11 apply also in this case.

17. *Coefficients, corrections and tolerances* are given in Figs. 18 *a.* to *e.*, Data Sheet 6.

C. ORIFICE PLATE WITH D AND D/2 TAPS

18. *Size limitation.* This orifice may be used in any pipe not less than 2 in. internal diameter provided that '*m*' does not exceed 0·7 and '*d*' is not less than 0·25 in.

19. *Construction.* Experimental data on orifice coefficients is somewhat more complete for orifices with corner taps, and these have been considered primarily in this Code. Much data is however available for orifices with D and D/2 taps, and as this alternative position for the pressure taps presents certain obvious advantages from the manufacturing standpoint, particularly for metering in existing pipelines, the data is also included.

20. *Accuracy of manufacture.* The rules regarding the proportions and manufacture of orifice plates with corner taps (Section B preceding) apply equally for orifices with D and D/2 taps.

21. *Pressure tappings.* These should be of the form shown in Fig. 4 (Data Sheet 1) from which it should be noted that the essential feature is the maintenance of the smooth wall up to and flush with the pressure holes. The diameter of the hole at the point of breaking through should not exceed D/10. If this diameter exceeds that of the tapping for the cock or nipple, the hole may be tapped through into the main providing that all burrs and projections are removed by filing after the cock or nipple is screwed in.

The location of the pressure taps should be:

a. for the upstream taps: D±D/10 from the upstream face of the plate.

b. for the downstream tap: D/2±D/20 from the downstream face of the plate.

Allowance should be made in each case for the thickness of any jointing material if the distance from the flange face is required.

It is especially important that there are no rivet heads, projection ridges or other irregularities in the pipeline in the vicinity of the pressure holes. These would affect the coefficient, and if unavoidable, special tests would be required to determine their effect.

22. *Coefficients, corrections and tolerances.* These are given in Figs. 19 *a.* to *e.* (Data Sheet 7) and are slightly different from the corresponding data for corner taps, particularly at the higher values of '*m*'.

D. VENTURI TUBES

23. *Size limitation.* The standardised Venturi tubes (Figs. 5 and 6, Data Sheet 2) are applicable for pipe diameters 'D' not less than 2 in. and for ratios of '*m*' between 0·05 and 0·55 noting that in all cases '*d*' is not less than 0·8 in.

24. *Construction.* Two standard designs of Venturi tube are shown in Data Sheet 2 as follows:

a. Conical Entrance Type—Fig. 5.

b. I S A 1932 Nozzle Type—Fig. 6.

Type *b.* is almost identical as far as flow measurement is concerned with the corresponding flow nozzle. Type *a.* does not correspond to any existing design of nozzle. The coefficients of discharge of these tubes of design *b.* are identical with those of the corresponding nozzles. Owing to the uncertain value of the expansion coefficient, however, there may be an error if the differential exceeds 2 per cent of the absolute pressure (for compressible fluids only).

The Venturi tube differs from the nozzle in having a conical enlarging portion or tail following the throat, for the purpose of minimising the energy loss (due to eddies and friction) which occurs where the flow stream expands to fill the pipe. The dimensions of the tail do not affect the coefficient of discharge, provided that the throat is continued as a true cylinder beyond the throat tapping for the distance shown in the appropriate figure, and that no portion of the tail protrudes within the cylinder.

The angle of taper of the tail determines the pressure loss caused by the Venturi tube. Where the minimum pressure loss is desired, an included angle of five or six degrees is usual; but where pressure loss is less important, the tail often has a larger included angle (up to 14°) and is sometimes truncated.

Figs. 5 and 6 (Data Sheet 2) show the geometrical internal shape of the Venturi tube. While the method of construction is not specified, it is usual where a tube is used to measure water or other liquids, to have the tube made of cast-iron or steel, and in this case, liners of bronze or some other non-corrodible material are inserted at the throat and usually also at the entrance. Occasionally, the complete converging portion is lined throughout with a non-corrodible alloy. The object of this arrangement is to maintain the shape of the internal surface by avoiding corrosion and incrustation. The whole of the converging portion and throat must be finished smooth, as any roughness influences the coefficient of discharge.

The entrance portion of the Venturi tube must have the same diameter as the adjacent pipe, and must be installed concentric with it. No jointing material must be allowed to project inside, nor must any projection raised by manufacturing methods, e.g. welding, be permitted to exist.

25. *Accuracy of manufacture.* The rules regarding the proportions and manufacture for the I S A nozzle in Sub-Section A are applicable.

26. *Pressure tappings.* It is usual to have a number of pressure tappings distributed round the circumference of the tube, and connected to an annular chamber or 'piezometer ring'. This arrangement has the advantage that the accuracy of measurement is less susceptible to disturbed flow conditions due to bends or valves immediately preceding the Venturi tube, and also that accidental obstruction of one pressure hole does not seriously interfere with the measurement. This arrangement, however, is not essential, and a single pressure hole is often used where the conditions are suitable. The arrangement described applies to each of the two pressure measurements, namely, entrance and throat.

The diameter of the pressure hole or holes at the entrance should not exceed 0·1D and at the throat 0·1*d*.

27. *Coefficients, corrections and tolerances.* These are given on Figs. 20 *a.* to *e.* (Data Sheet 8) for Conical Entrance Type Venturis and on Figs. 17 *a.* to *e.* (Data Sheet 5) for the I S A 1932 Nozzle Type Venturi.

Part Three : Standard Methods of Installation of Pressure Difference Devices

28. In order to maintain the accuracies specified in Part 7, great care must be taken in fitting the pressure difference device into the main. If this is not done, errors in metering may occur which exceed the tolerances to a considerable extent. There are no reliable data for the estimation of such errors; in all cases, however, metering with large values of the area ratio '*m*' is much more sensitive to installation faults than with small values, as can be seen from the diagram of straight main requirements. (Data Sheet 3.)

The installation rules which follow are those necessary to ensure that the standard metering errors are not exceeded.

29. The pressure difference device, manufactured in accordance with the appropriate specification, is preferably mounted between two flanges of a straight run of main. For mains of less diameter than 2½ in. it is recommended that a short length of pipe containing the pressure difference device be supplied by the makers as an integral part of the apparatus.

The device must be installed so that the flow through it is in the proper direction, and the throat must be concentric with the bore of the main. The latter is generally assured for orifice plates or nozzles by making the plate or carrier with an outside diameter equal to the diameter of the flange bolt pitch-circle less the diameter of the bolt holes, but concentricity should be checked by measurement. Gaskets, rivets or weld seams must not project into the main near the point of installation, and the former are usually made with an inner diameter about 10 per cent larger than the bore of the main for this reason. Where drain holes are provided through orifices or nozzles they must be at the bottom of the main when metering steam or gases and at the top for liquids. They must be at least 90° displaced from the differential pressure tappings.

When metering steam or gas in vertical mains it is necessary to take special care that the pressure difference connections do not become blocked by condensation (see details in Part 4).

30. When metering gases carrying dust or tar, means must be provided for removing deposits from the pressure difference device, so that for example the necessary sharpness of the edge of an orifice plate is maintained.

The main must be truly cylindrical for a distance of 2D on the upstream side of the pressure difference device. All diameters measured in this distance must lie within 0·5 per cent of one another. This average diameter, which frequently differs appreciably from the nominal bore of the pipe, must of course be used in the calculation of pressure difference devices. Orifice carriers, if used, may be 3 per cent larger than the actual bore of the main, provided the recess thus formed is not wider than 0·1D.

31. The roughness of the inside surface of the main has an effect on the discharge coefficient (see Part 7). The degree of roughness is difficult to define, as it depends on the dimensions of the surface irregularities in relation to the pipe diameter. As a basis, it is assumed that a new cast iron pipe tarred inside can be regarded as smooth if it is 8 in. in diameter or larger. The present assumption is that the pipe has no more than the usual roughness of a cast iron pipe, i.e., it may be rusty but not encrusted with scale or deposit.

32. The main must be straight and co-axial for a suitable distance before and after the pressure difference device. The upstream distance must be such that any disturbances of the flow have had time to subside before the measuring position is reached, and the downstream distance sufficient to prevent any reaction being propagated back to the pressure difference device. Data Sheet 3 shows the upstream (S) and the downstream values, in terms of the diameter of main 'D_1' of the minimum straight lengths necessary for various common arrangements of fittings.

It is to be noted that the straight pipe requirements become smaller as '*m*' decreases, so that in cases where a sufficient length is not available, a smaller value of '*m*' may improve the situation, if the additional pressure difference and pressure loss can be tolerated.

33. The installation of a pressure difference device immediately after an enlargement or contraction of the main is not permissible. A straight length of uniform diameter must be inserted between, and this length increases as '*m*' and the degree of enlargement increase. Sudden reductions of diameter without any taper piece cause important errors, the meter readings being too low, and should be avoided.

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Any device or combination of bends which produces a helical motion of the fluid in the main should be placed remote from the pressure difference device, as the helical motion subsides very slowly (see Data Sheet 3).

34. Flow straighteners (i.e., plates or vanes parallel to the axis of the main) are only of use if helical motion exists. To be effective they should have a length of from 1D to 2D. Placed after irregularities where helical motion does not exist a flow straightener usually does harm by preserving the abnormal velocity distribution due to the bend which would otherwise normalise itself. A length of straight pipe equivalent to that after a single bend is also necessary between any such flow straightener and the pressure difference device (see Data Sheet 3).

35. Regulating valves of any description should always be placed after the pressure difference device, but if this is impossible should be not less than 100 diameters before it. The error due to valves placed closer than this varies widely with the rate of flow and the valve opening.

Abnormal velocity distribution has less effect when annular pressure chambers are used than with single tappings for the pressure difference. Experience indicates that the efficiency of the annular chamber increases the larger its cross section and the smaller the area connecting it to the inside of the main.

36. The pressure difference device can only be said to have been installed correctly if all the conditions set out in this section have been complied with.

Part Four:

Methods of Measurement, Application and Data required

A. PRESSURE PIPES FROM DIFFERENTIAL PRODUCER TO METER

37. It is of the first importance that no back pressure or false differential shall be set up in the pressure pipes, as this would partially balance or resist the differential produced. The guiding principle, therefore, in the laying of pressure pipes, for any fluids, is that the slope shall be adequate (not less than 1 in 20) to carry away adventitious gas, liquid or solid to an 'air vent,' 'moisture sump' or other reservoir, as applicable, so that the fluid in the pressure pipes is homogeneous throughout, except in the special cases where sealing chambers are employed. The two pressure pipes must be at equal temperatures and where risk of outside radiation or cooling occurs they must be lagged together. The following rules elaborate this principle for various fluids.

a. LIQUIDS

38. The flowmeter should, whenever possible, be below the differential pressure device, and the slope of the pipes so arranged as to dissipate released gas or air back into the main. (Fig. 12, Data Sheet 4.)

If the flowmeter is situated on a higher level than the differential pressure device, then gas collecting chambers must be fitted at the highest point in the pressure piping. (Fig. 11, Data Sheet 4.)

Dirt sumps at the lowest points of the piping are optional, and are decided by the nature of the liquid.

b. STEAM

39. The flowmeter should whenever possible be at a lower level than the differential pressure device, and if the pressure in the main is sub-atmospheric, the meter must be sufficiently lower to ensure a positive pressure in the 'downstream' side of the instrument. (Fig. 12, Data Sheet 4.)

If the flowmeter is placed at a higher level than the differential pressure device, air vents must be provided at the high points of the piping (air chambers in the meter itself being sufficient if the meter is the highest point). The pressure must in no case be lower than atmospheric in the 'downsteam' side of the meter. (Fig. 11, Data Sheet 4.)

40. *Condensation chambers.* These must be fitted as close to the steam pipe as possible, and must have sufficient reserve of volume to permit maximum travel of the flowmeter mechanism without introducing perceptible false differential. Type A (Fig. 14, Data Sheet 4) achieves this by having a large surface area; type B (Fig. 14, Data Sheet 4) supplies the reserve from a column of condensate above the datum line of the pressure tapping.

41. *Condensation chamber connections on non-horizontal steam lines.* Where there is any difference in elevation between the two pressure tappings the two condensation chambers must both be placed at the same level, which may conveniently be the level of the upper tapping, but must not be lower. The connection from the lower tapping must be of at least ½ in. bore throughout, smooth and free from pockets, etc., which might trap condensate. It must be properly lagged, from the main up to the level of the condensation chambers and must be of adequate slope throughout its length to drain condensate freely back into the main. (Fig. 13, Data Sheet 4.)

c. AIR AND GASES

42. The flowmeter should preferably be above the differential pressure device, the pressure pipes falling smoothly to drain condensed liquid back into the main. (Fig. 15, Data Sheet 4.)

If the flowmeter is at a lower level than the differential pressure device, drains must be provided at the lowest point (water seals or catchpots according to pressure) and the connections to the meter must rise from these drains. (Fig. 16, Data Sheet 4.)

The two pressure pipes should in all cases be kept as close together as possible throughout their run. (Figs. 15 and 16, Data Sheet 4.)

When dealing with hot gases, great care must be taken to avoid false differential arising from a difference in temperature between the two pressure pipes.

Vertical runs of pressure pipe in close proximity to the hot main should be avoided, or if this is impossible the two pipes should be lagged together for that part of their run to avoid temperature inequalities.

The pressure tappings should, for air and gases, be taken from the top of the main.

d. LONG PRESSURE PIPE RUNS

43. Where the distances run are very considerable, or obstructions have to be avoided, it is permissible (disregarding speed of response of the meter) to run the pipe in a series of slopes, provided the general rules are followed (i.e., for steam and liquids, air vents at all high points and dirt sumps at all low points; for air and gases, water seals or catchpots at all low points).

In all cases where air vents or catchpots are fitted, arrangements must be made for their systematic operation, daily or as often as found necessary.

e. SEALING CHAMBERS

44. When measuring corrosive liquids or gases, or liquids likely to congeal in the pressure pipes, sealing chambers can be interposed between each pressure tapping and the ordinary pressure piping to the meter. These serve as a reservoir for an intervening liquid (usually referred to as a sealing liquid) of a more suitable character, with which the pressure pipes to the meter are filled.

The usual sealing liquids are light oils, or glycerine mixtures. They must, of course, have a different density from, and not be miscible with, the fluid to be measured.

Sealing chambers should always be installed close to the differential pressure device in the pipeline and the connections therewith should be short and large in bore. In the case of viscous liquids, they should be lagged to the pipeline or heated independently.

The type of connection to the sealing chambers depends on whether the sealing liquid is lighter or heavier than the metering fluid and the meter manufacturer will normally provide the user with all necessary details and instructions for the operation of filling and equalising, also the correction factor involved by the use of the sealing chambers. If these are not less than 4 in. diameter, the correction factor (which is a fairly complex function of meter displacements and liquid densities) is usually less than 2 per cent.

B. MEASUREMENT OF PRESSURE DIFFERENCE

45. For acceptance and laboratory tests, a single or double tube manometer is generally used to measure the pressure difference. If the pressure difference is varying, it is preferable to use a single tube manometer, the short leg of which is connected to a vessel, the cross sectional area of which is about a hundred times that of the long leg. A direct reading from the scale fixed behind the glass tube is only accurate when the pressure difference is steady. A cursor is used, so that the meniscus can be observed without parallax error, and the reading can then be taken easily from a scale at the side.

To avoid errors due to capillary attraction, it is advisable to use glass tubes having a bore of at least ⅜ in. for mercury and ½ in. for water or oil. For differences in level of less than 2 in. it is advisable to use special manometers or micromanometers, e.g., [illegible] or curve tube type, etc., giving a large reading.

46. Differential pressure meters, with the exception of the simple glass U tube, must be calibrated against a water column or other standard. Because of the quadratic relationship between the pressure difference and the flow small percentage errors in measuring the pressure difference correspond to about half the percentage errors in the flow measurement.

47. Several aspects of metering technique must be considered in the selection of the most suitable type of flow meter. The working pressure and the pressure difference are the primary factors which govern the type of meter; i.e., whether a float manometer, ring balance, or liquid sealed bell type is to be used. The maximum pressure difference must be selected so as to suit the area ratio of the throttling device, for the existing plant conditions. While on the one hand, it may seem desirable to use a small pressure difference, and consequently a large area ratio in order to lessen the effect due to expansion, it is on the other hand recommended that a small diameter ratio and a large pressure difference be selected, because the tolerances for the flow coefficient and the errors due to irregularities of installation will be smaller, and also because the force operating the flow meter will be greater. It must always be remembered however that some of this differential pressure is not recovered. The permitted magnitude of this pressure loss depends upon the plant conditions and the choice of the p.d. device may be influenced by this consideration. The net pressure loss due to various p.d. devices is given in Data Sheet 16 and Appendix C.

C. MEASUREMENT OF DENSITY

48. The density required is that at the position of the upstream pressure tapping. A measurement of temperature suffices for incompressible fluids, while for compressible fluids the pressure is required in addition. This enables density to be derived from standard tables for water, air and steam; while for gases, oils, etc., a chemical analysis or an actual measurement of the density must be made.

The pressure is to be measured in the plane of the upstream pressure tapping. In cases where the upstream pressure cannot be read at the plane of the tapping point but is read at some adjacent point, due allowance must be made for the difference of pressure between this and the orifice position. Table 7, Appendix E, gives the densities of gases important in industry, and Appendix H gives the

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densities of saturated and superheated steam at various pressures.

The temperature is to be measured at a point as near as possible to the throttling device, but care must be taken that the position of the thermometer has no effect upon the flow.

When metering continuously and the density is varying, the latter must be determined at frequent intervals and the current figure applied to flow readings taken at any particular time.

49. *Departures from Boyle's Law.* At high pressures (10 atmospheres and over for air and pure gases), especially when near saturation point, the departure from the ideal gas law should be taken into account. This is taken care of in the formula by including the compressibility coefficient 'K'; 'K' is the ratio of the density calculated from W_n using the ideal gas law, to the true density.

The value of 'K' may be taken from the Landolt-Bernstein physical-chemical tables, or more conveniently from graphs of the type shown on Data Sheet 12, based on one atmosphere absolute pressure.

50. *Density of moist gases.* This should be calculated from the formulæ given in Part 5, using Data Sheet 13. (See also Appendix D.)

D. VISCOSITY OF FLUIDS

51. Absolute viscosity is defined as the force which exists when unit area of a fluid is moving at unit velocity relative to that of a similar area unit distance from it and parallel to it.

It has the dimensions $\frac{M}{LT}$ in the fundamental units of mass, length and time.

The unit of absolute viscosity in the C.G.S. system is known as the poise, the centipoise representing 0.01 C.G.S. units.

The ratio of absolute viscosity to density is known as kinematic viscosity. Density must be expressed in units compatible with the system of units employed; that is to say, either as grammes per cubic centimetre or as pounds per cubic foot. The fundamental dimensions of kinematic viscosity are $\frac{L^2}{T}$. The unit of kinematic viscosity in the C.G.S. system is known as the Stokes, the Centistokes representing 0.01 C.G.S. units.

Viscosity changes rapidly with temperature. It increases with increasing temperature in the case of gases and decreases with increasing temperature in the case of liquids. Kinematic viscosity must always be calculated with reference to the viscosity and density of the fluid at the temperature concerned. Kinematic viscosity is of great importance in all problems connected with the flow of fluids, both liquid and gaseous, since it is one of the factors on which the pressure drop depends.

At pressures in the neighbourhood of atmospheric, the kinematic viscosity of a gas is inversely proportional to its absolute pressure. Thus if the kinematic viscosity be required at a pressure of 1298 millimetres of mercury (absolute) the tabulated value must be multiplied by $\frac{760}{1298}$.

52. Although it is not possible accurately to calculate the viscosity of a gaseous mixture from a knowledge of the viscosity of the individual constituents and the percentage composition, approximate values for various fuel gases may be obtained by use of the following formula (Zipperer and Muller, Das Gas-und Wasserfach, 1932, 75, 623, 641, 660, 989):

$$\left.\begin{array}{l}\text{Kinematic Viscosity}\\ \text{(at 20°C, 760 mm.)}\\ \text{in C.G.S. Units (Stokes)}\end{array}\right\} = \frac{107{\cdot}8}{7(O_2 + CO + CH_4 + N_2) + 14(CO_2 + C_nH_m) + H_2}. \quad (2)$$

0·6 per cent is to be added for each 1°C. rise of temperature, 0·6 per cent is to be subtracted for each 1°C. fall in temperature.

The formula is applicable over the range −10° to +40°C.

By use of the above formula and the data given the characteristics have been calculated for various fuel gases and are given in Appendix E.

Part Five : Working Formulæ

NOTE. The derivation of the formulæ in this Part is given in Appendix B.

53. The following formulæ are to be used for calculating the rate of flow when the orifice or throat diameter is known.

$$L = 359{\cdot}1 \times C.Z.d^2E. \times \sqrt{h} \times \sqrt{W} \quad (3)$$

$$G = 2239 \times C.Z.d^2E. \times \sqrt{h} \times \sqrt{\frac{1}{W}} \quad (4)$$

$$Q = 7859 \times C.Z.d^2E. \times \sqrt{h} \times \sqrt{\frac{P_1}{T\delta}} \times N \quad (5)$$

See Annex No. 1.

54. The following formulæ are to be used for calculating the diameter of the orifice or throat when the rate of flow is known.

$$C.Z.m.E = \frac{L}{359{\cdot}1 \times D^2 \times \sqrt{h} \times \sqrt{W}} \quad (6)$$

$$\text{or } C.Z.m.E = \frac{G}{2239 \times D^2 \times \sqrt{h} \times \sqrt{\frac{1}{W}}} \quad (7)$$

$$\text{or } C.Z.m.E = \frac{Q}{7859 \times D^2 \times \sqrt{h} \times \sqrt{\frac{P_1}{T\delta}} \times N} \quad (8)$$

Equations 6, 7 and 8 are solved to find C. Z. *m*. E.

Taking Z = unity for a first approximation, the value of *m* approximately is then read directly from the curve at the top of each data sheet.

Formulæ 3 and 6 apply to any liquid, gas or vapour. Formulæ 4 and 7 apply to liquids only. Formulæ 5 and 8 apply to air and gases only, and give the rate of flow in 'standard cubic feet' (for definition see below).

The application of the working formulæ 3 to 8 and suitable examples are given in Appendix G.

In the above formulæ:

L = Rate of flow in lb. per hour.

G = Rate of flow in gallons per hour.

Q = Rate of flow in standard cubic feet per hour.

C = Coefficient of discharge of the orifice, nozzle or Venturi tube; the ratio of the actual discharge to the theoretical discharge calculated from the elementary energy equation (Bernoulli's theorem, Appendix B).

$E = \frac{1}{\sqrt{1-m^2}}$ Velocity of approach factor. See Data Sheet 9.

Z = a combined multiplier: the product of three separately specified multipliers for viscosity R_d, pipe size D and expansion ε.
'Ideal conditions' correspond to high Reynolds numbers, large pipe sizes and negligible density change by expansion.

m = Area ratio $\frac{d^2}{D^2}$

d = Diameter in inches of orifice or throat of nozzle or venturi at the working temperature.

D = Diameter in inches of pipe or venturi tube at high pressure tapping.

h = differential pressure, inches of water at 60°F. (under air).

W = density of fluid, lb. per cu. ft., under working conditions at the position of the high-pressure tapping.

P_1 = absolute pressure of gas or air, lb. per sq. in., measured at the position of the high-pressure tapping.

See Annex No. 1. T = absolute temperature of gas or air, deg. F. + 460 ~~459·63~~ measured at the position of the high-pressure tapping.

δ (delta) = specific gravity of dry gas, relative to dry air at the same pressure and temperature. (For air δ = 1.)

N = correction for water vapour in gas or air: see below. (For dry gas or air N = 1.)

55. The correction N for water vapour is evaluated from the formula below, or more conveniently in ordinary cases from Data Sheet 13.

Data Sheet 13 consists of two separate charts drawn together for convenience: the first gives the vapour pressure of water at various temperatures but any steam tables may be used instead; the second chart gives the factor N. The Data Sheet is used as follows:

1. From the saturation temperature on the top scale, read off the vapour pressure w on the left-hand scale, utilising the line for lb./sq. in. or for in. Hg as required.

2. Divide the absolute pressure of the gas P_1 by the vapour pressure w (both being in the same units). This gives the factor B.

3. Against the factor B on the right-hand scale, read across to the curved line corresponding to the specific gravity δ of the gas, and on the bottom scale read off the moisture correction factor N.

$$N = \sqrt{\frac{B-1}{B + \frac{0{\cdot}624\,B}{\delta(B-1)}}} \quad (9)$$

where $B = \frac{P_1}{w}$

P_1 = Absolute pressure of moist gas or air, lb. per sq. in., measured immediately before pressure difference device.

w = Absolute pressure of water vapour in gas or air, lb. per sq. in., under working conditions, which is the same as the pressure of saturated steam at the dewpoint temperature of the gas, and may be found from standard steam tables. (If the gas is saturated the dewpoint and actual temperatures are the same.)

NOTE. Formulæ Nos. 5 and 8 depend for their accuracy on the validity of the assumption that the gas obeys Boyles and Charles' laws. This assumption is correct, within sufficiently close limits for purposes of flow measurement, for air and all the common industrial gases at pressures up to 150 lb. per sq. in. For higher pressures or for vapours refer to para. 49.

For moist gases they also depend on the assumption that the water vapour follows the same laws. This is sufficiently accurate at dewpoint temperatures up to 200°F.

For any gas, vapour or gaseous mixture, whether following Boyles and Charles' laws or not, the expression

$\left(\sqrt{\frac{P_1}{T\delta}} \times N\right)$ may be evaluated as follows:

$$\left(\sqrt{\frac{P_1}{T\delta}} \times N\right) = \frac{0{\cdot}04569}{\sqrt{W_m}} \times \frac{W_a}{W_z}. \quad (10)$$

where W_m = density of mixture of gases or vapours under working conditions. (lb. per cu. ft.)

W_a = weight of substance whose quantity is to be determined, in one cu. ft. of the mixture under working conditions. (lb.)

W_z = weight of substance whose quantity is to be determined in one standard cu. ft. (lb.)

(Normally W_a would be the weight of dry gas in one cu. ft. of a mixture of gas and water vapour, while W_m would be the total weight of gas and water vapour in one cu. ft.)

STANDARD CUBIC FEET

56. In air and gas measurement the standard unit of volume is that quantity of air or gas which occupies 1 cu. ft. when at a pressure of 14·70 lb. per sq. in. absolute (30 in. of mercury at 60°F. under air) and a temperature of 60°F. and saturated with water vapour.

If it is desired to adopt a different unit, the conversion factors given in Table 1 may be used.

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TABLE 1

Conditions at which unit is defined			Number of standard cubic feet equivalent to one unit
Pressure	Temperature	Moisture	
30 in. Hg.	60°F.	Saturated	1·000
30 in. Hg.	60°F.	Dry	1·018
760 mm. Hg.	0°C.	Saturated	1·066
760 mm. Hg.	0°C.	Dry	1·073

DIFFERENTIAL PRESSURE: CONVERSION FACTORS

57. In the formulæ in Paras. 51 and 52 the differential pressure is required in inches of water at 60°F. under air. If the differential pressure is measured by a U-tube containing other fluids, the measured difference in level of the U-tube liquid must be converted into inches of water as follows:

$$h = \frac{J\,(W_g - W_f)}{62{\cdot}28} \qquad (11)$$

where h = differential pressure, inches water at 60°F. under air.

J = observed difference in level of U-tube liquid, inches.

W_g = density of U-tube liquid, lb./cu. ft., at working temperature of U-tube.

W_f = density of covering fluid above U-tube liquid, lb./cu. ft., at working temperature of U-tube.

The conversion factors given in Table 2 are often required.

TABLE 2

Sealing liquid in U-tube	Fluid over sealing liquid	Inches of water corresponding to 1 in. difference of level
Water at 60°F.	Air	1·000
Mercury at 60°F.	Air	13·59
Mercury at 60°F.	Water at 60°F.	12·59

If the covering fluid is air or gas at low pressure, it may be assumed that $W_f = 0$.

Part Six : Reynolds Number

NOTE. The derivation of the formula in this Part is given in Appendix B.

58. Reynolds Number is a dimensionless criterion which permits of comparison between the results of observations made with geometrically similar installations under different conditions and with fluids of different densities and viscosities provided that the difference in their other properties such as compressibility, surface tension and thermal conductivity can be ignored.

The physical conditions governing the flow will depend on:

(1) the diameter of the throttling device throat 'd.'

(2) the mean velocity 'V' corresponding to d.

(3) the density of the fluid 'W.'

(4) the absolute viscosity 'η' (eta).

Reynolds Number may be written as:

$$R_d = \frac{d\,V\,W}{\eta} \qquad (12)$$

and since R_d is a dimensionless number the numerical value will be the same for any given case provided the separate factors are expressed in self-consistent units.

When the values of Reynolds Number are sufficiently high the flow coefficients remain substantially constant for standardised nozzles and orifice plates, i.e., they remain unaffected by the velocity, density and viscosity of the flowing liquid. The values of R_d at which the flow coefficients can no longer be considered to remain constant are known as 'limiting values.'

It is important to note that in order to use Reynolds criterion to determine the value of the flow coefficient in any instance the throttling device under consideration must be geometrically similar to that with which it is being compared. The term geometric similarity covers such details as: Similarity of installation, of pressure tappings and surface conditions in the channels and throttling device.

REYNOLDS NUMBER

59. The following formulæ may be used to evaluate the Reynolds Number.

For steam, water, oil etc.

$$R_d = \frac{L}{15{\cdot}83\,d\,\eta} \qquad (13)$$

For air and gases.

$$R_d = \frac{Q\,\delta}{211\,d\,\eta} \qquad (14)$$

where L = rate of flow, lb. per hour.

Q = rate of flow, standard cu. ft. per hour.

δ = specific gravity of dry gas relative to dry air.

d = diameter of orifice, or throat of nozzle or Venturi tube, in.

η = absolute viscosity of fluid at working conditions, in C.G.S. units (poises).

60. In the above formulæ the absolute viscosity in C.G.S. units is used, in preference to the kinematic viscosity or to F.P.S. units, because the former values are more readily available from standard tables. However, the following alternative formulæ may be used if preferred.

For steam, water, oil, etc.

$$R_d = \frac{3{\cdot}94\,L}{W_m\,d\,\nu} \qquad (15)$$

For air and gases following Boyles & Charles' law.

$$R_d = \frac{Q\,T}{134\,d\,\nu_1} \qquad (16)$$

where W_m = density of fluid, lb. per cu. ft. under metering conditions.

T = absolute temperature in °F.

ν (nu) = kinematic viscosity, C.G.S. units at working temperature and pressure.

ν_1 = kinematic viscosity, C.G.S., units at working temperature and atmospheric pressure (14·70 lb. per sq. in. absolute).

Part Seven : Coefficients of Discharge and Tolerances

61. In the references given below the coefficient C is plotted against 'm' values only, and not against R_d. Corrections are given for various values of R_d and the additional tolerances required, when the R_d correction comes into use, are given on the same curve. To facilitate calculations values of C.m.E. are marked along the C.m. curves.

NOTE. In the B.S.-Code the coefficient C is used which excludes the velocity of approach factor E. In some other codes, however, the coefficient α (alpha) is used which includes E i.e., $\alpha = CE$.

62. *Corrections for pipe size.* The coefficient C is to a small extent dependent on the roughness of the pipe walls, increased roughness having the effect of increasing C. The curves of correction factors for pipe size are applicable to normal commercial clean cast-iron or mild steel pipe, and do not cover badly corroded, incrusted or noduled pipe. For exceptionally smooth walled pipes, such as drawn copper or brass, the correction factor will be less. The extra tolerance for pipe size should be taken into account, even if the correction factor for pipe size is not used, since the extra tolerance covers a number of additional uncertainties, which are proportionately greater on small sizes. (See Figs. 17(*e*), 18(*e*), 19(*e*) and 20(*e*).)

63. *Correction factor for expansion for air, gases and vapours.* The factor ε is introduced into the flow formulæ to allow for the effect on the flow due to expansion of volume occurring during the acceleration of the stream.

Data Sheet 11 gives the value of ε (epsilon) for various values of m for nozzles and Venturi tubes, and the corresponding values for orifice plates. Where very accurate continuous metering is required it must also be remembered that the coefficient of expansion 'ε' varies with the pressure difference, and this can be allowed for in the flow spacing on the meter scale. Where the variation is comparatively small it is common to assume a mean value corresponding to two-thirds of the maximum flow.

The values of ε are not known with absolute accuracy, and it is therefore necessary to apply an additional tolerance equal to one-third the percentage correction in the case of gases and one-half the percentage correction in the case of vapours.

64. *Ultimate working coefficient.* This is the product of the basic coefficient and the correction factors for roughness, Reynolds number and expansion (where applicable), as separately defined above.

65. *Ultimate working tolerance.* See paragraph 3, part 2, the individual tolerances being given on the data sheets detailed in Table 3.

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TABLE 3

FIGURE NUMBERS OF CURVES GIVING COEFFICIENTS, CORRECTION FACTORS AND TOLERANCES

P.D. device	Coefficient C	R_d Corrections	Corrections for pipe size	Tolerances		
				Basic	Extra for pipe size	Extra for R_d.
I S A 1932 nozzle	17a	17b	17c	17d	17e	17b
I S A nozzle Venturi	17a	17b	17c	17d	17e	17b
Orifice corner taps	18a	18b	18c	18d	18e	18b
Orifice D & D/2 taps	19a	19b	19c	19d	19e	19b
Conical Venturi	20a	20b	20c	20d	20e	20b

Part Eight : Special Cases of Orifice or Nozzle Installation

Cases A, B, C.

66. The preceding parts have been concerned with orifices and nozzles inserted in a run of pipeline. Certain cases arise where it is more convenient to dispense with the pipeline on one or both sides of an orifice. These cases may be classified as below, and are applicable to pipes of circular cross section only and concentric orifices.

Case A. Flow from one space to another with orifice or nozzle in the partition plate between the spaces.

Case B. Flow from a pipe to an open space.

Case C. Flow from an open space into a pipe.

An open space may be defined as one in which the confining wall (if any) apart from that containing the orifice or nozzle, is not less than $4d$ removed from the centre of the orifice or nozzle. If the flow is approaching the orifice or nozzle from the open space it must do so uniformly, i.e., be accelerated from practically zero velocity, and be free from jet or deflection effects.

67. The coefficient applicable in the above three cases may be taken from the curves :

Case A. As for a pipeline, for $m = 0$.

Case B. As for a pipeline, for the value of m related to the pipe size on the inlet of the orifice or nozzle, providing this pipe is large enough to give a value of m less than 0·15.

Case C. As for a pipeline for $m = 0$, providing the area of the pipe on the downstream side is not less than 7 times that of the orifice or nozzle.

In cases B and C, if an orifice outside the above limits is required the coefficients must be determined experimentally.

68. The following also apply with regard to pressure tappings, etc.

Case A. Tappings should be in the wall of each vessel, preferably in that at right angles to the partition wall and close to the plane of this wall, or not more than d removed therefrom. If the inlet side of the partition wall is completely open, e.g., in a large room, the upstream tapping can be omitted if the corresponding manometer connection is situated in the same room and both orifice and manometer are shielded from draughts.

Case B. The upstream tapping should be as for a pipeline orifice or nozzle, while the proviso relating to upstream tapping under heading (A) applies in this case to the downstream side.

Case C. The downstream tapping should be as for a pipeline orifice or nozzle, while the proviso relating to upstream tapping under heading (A) applies to the upstream side.

The Reynolds Number corrections applicable are those corresponding to the value of m calculated from the upstream area only.

Case D. Flow Measurement above the Critical Pressure Drop

69. If a sufficiently large pressure difference is permissible, it is possible to measure the flow of gases and vapours by a knowledge of the upstream pressure and temperature only. For this purpose only a nozzle is suitable, and the ratio of downstream to upstream (absolute) pressure must be below the critical value. At this critical value the velocity of the flow at the throat of the nozzle is equal to that of sound under the same conditions as those ruling at that point. When the pressure difference increases, i.e., when the downstream pressure P_2 becomes less, the flow through the nozzle does not increase any further, because the pressure at the throat is no longer affected by the downstream pressure P_2.

70. If we take equation (3) neglecting pipe size and viscosity corrections and substitute for h the expression $\frac{1728}{62{\cdot}35}(P_1 - P_2)$ we get

$$L = 359{\cdot}1 \times C{\cdot}d^2 \cdot E \times \varepsilon \times \sqrt{\frac{1728}{62{\cdot}35}(P_1 - P_2)} \times \sqrt{W} \qquad (17)$$

which reduces to

$$L = 1890 \times C{\cdot}d^2 \cdot E \times \varepsilon \times \sqrt{1 - \frac{P_2}{P_1}}\ \sqrt{P_1 W} \qquad (18)$$

and dealing similarly with equation (5) we obtain

$$Q = 41380 \times C{\cdot}d^2 \cdot E \times \varepsilon \times \sqrt{1 - \frac{P_2}{P_1}} \times \frac{P_1}{\sqrt{\delta T}} \times N \qquad (19)$$

Now the quantity $\varepsilon \sqrt{1 - \frac{P_2}{P_1}}$ is dependent upon $\frac{P_2}{P_1}$, m and the specific heat ratio γ (gamma), and for given values of m and γ it increases as $\frac{P_2}{P_1}$ decreases until it reaches a maximum at the critical value of $\frac{P_2}{P_1}$ subsequently remaining constant. The critical values of the pressure ratio are given in terms of m and γ in Table 4 :

TABLE 4

γ	m		
	0	0·2	0·4
1·31	0·544	0·549	0·565
1·41	0·527	0·532	0·548

By writing

$$U = CE\,\varepsilon\sqrt{1 - \frac{P_2}{P_1}} \qquad (20)$$

we derive the final equations of flow

$$L = 1890\ Ud^2\ \sqrt{P_1 W} \qquad (21)$$

and $$Q = 41380\ Ud^2\ \frac{P_1}{\sqrt{\delta T}} \qquad (22)$$

where U is a number depending upon m and γ.

The moisture factor N has been taken as unity, and gases with substantial moisture content cannot, without special information, be measured below the critical pressure ratio.

For the ISA Nozzle, the variation with m is very small, so that U may be taken as 0·468 when $\gamma = 1{\cdot}31$ and 0·481 when $\gamma = 1{\cdot}41$, provided m does not exceed 0·4.

Case E. Notes on Pulsating Flows

71. In all cases where the pipeline to be metered carries a flow which is pulsating, due to reciprocating engines, pumps, blowers, or compressors, it is essential that sufficient circuit damping exists to reduce the flow wave form to a suitable extent.

For liquids the methods available consist in the use of adequate air chambers at or near the pumps, alternatively loaded piston accumulators or steam generators.

For air, steam or gases, a combination of volumetric capacity (pipeline and/or air receivers) with throttling (pipeline resistance or orifice or variable throttling) is necessary.

Reciprocating pumps cause pulsations in the flow of liquids from the pumps and unless eliminated will cause serious errors in the meter readings. In the case of 'open' feed circuits these pulsations may be 'damped' out by the use of air vessels in the feed line between the pumps and the differential pressure device.

In order to reduce the time necessary to charge the air vessel, compressed air may be used, but if this is not available a charging chamber fitted to the side of the air vessel will permit the vessel to be charged from the atmosphere.

The fitting of air vessels cannot be advised in the case of 'closed' feed circuits since the air used to charge the vessel will be dissolved into the feed water and carried into the generating unit and so defeat the object of fitting de-aerators. To damp out pulsations in a 'closed' feed circuit, therefore, loaded piston accumulators may be used or preferably a chamber similar to an air vessel but fitted with an internal superheated steam heated coil. The steam passing through the coil will generate a cushion of steam above the water and result in the elimination of the pulsations.

For pulsating flows of air, steam or gases, providing the flow wave form is of the simple sine-wave type investigated, application of the 'Hodgson criterion'* will determine where sufficient flow damping exists to ensure accurate differential pressure metering.

In doubtful cases, e.g., where the flow wave form is liable to be complex, or the effectiveness of the flow damping is uncertain, a reliable indication or check can be applied by noting the change of meter reading when additional throttling is given by means of a control valve.

To avoid disturbing the orifice approach flow, the valve should be downstream of the meter orifice and no less than 5 pipe diameters away. A 'U' tube manometer or pair of accurate pressure gauges across the valve should be used to measure the amount of throttling applied. This amount can be, say 3 or 4 times greater than the meter orifice differential.

If the meter reading is observed to drop *severely*, it may be inferred that considerable pulsation error existed, because the actual flow should not be reduced appreciably by this additional throttling.

The percentage reduction of actual flow should not normally exceed the percentage reduction of absolute pressure and in the case of throttle governed engines will be much less.

* I.C.E. Selected engineering papers No. 31 'The Orifice as a basis of Flow Measurement' 1925.

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Part Nine :
Choice and use of Pitot Tubes for Flow Measurement

A. PRINCIPLE

72. The pitot tube consists essentially of a tube with an open end placed in a direction to face a stream of fluid. An impact pressure is produced in the tube derived from the loss of kinetic energy of the stream striking it.

Since a single tube gives the sum of the impact and static pressures, a pitot tube must work in conjunction with a means of measuring the static pressure. This may be a simple connection in the wall of the main, or may be incorporated in the pitot tube itself as seen in the examples below.

REASONS FOR USE

73. (1) The pitot tube, unless made excessively large in relation to the size of the main, produces no appreciable pressure loss.

(2) It can be inserted through a comparatively small hole into the main without the necessity for shutting down the main.

(3) It can be used to measure local velocities in the measurement of waste heat, etc.

LIMITATIONS

74. (1) The differential produced is far smaller and is also less steady than with orifices, nozzles and Venturi tubes.

(2) Only one filament of the stream is being measured and therefore a 'traverse' of the stream cross section has to be made in order to relate the readings to average velocity.

(3) For these reasons and others, the accuracy is not so high as with orifices, etc.

B. TYPES AND CONSTRUCTIONAL DETAILS

75. (1) *N.P.L. Hemispherical end type.* This is the type to be regarded as the standard, viz., the construction is shown in Fig. 25 (Data Sheet 14) and its coefficient may be taken as unity for values of R*d* above 3000 (taking *d* as the diameter of the pitot tip).

This type is less sensitive to 'yaw' (inclination of the tip at an angle to the streamlines); and the tip may be inclined at an angle of 20° to the stream without inaccuracies exceeding 2½ per cent in velocity.

In construction and in use, care must be taken that the tip is truly hemispherical and that the impact hole is not bell-mouthed, burred, nor eccentric with the hemisphere.

(2) *The N.P.L. sharp-ended type.* Fig. 26 (Data Sheet 14) may also be used if maintained in good condition; the tip must not be allowed to become blunt, distorted or burred in any way, and the tube must be free from damage, kinks or flattening before the static pressure holes and for a distance of five tube diameters after.

Other shapes of impact tube may be used with a separate static pressure connection in the wall of the pipe just upstream of the impact tube section. The impact tube or one of the same type should be calibrated by a recognised testing authority, though this calibration may be dispensed with by agreement between the parties.

Other types of pitot tube such as *a.* combined impact and suction tube (reversed tube), *b.* diametral rod type (single or reversed tube) may be used by agreement, provided that in cases where the suction head is measured, the tube is not used for traversing. Such tubes are useful where only one tapping is available on the pipe, or where circumstances require an increased differential. The coefficient depends on the position of the suction hole, and on the rate of flow. (See Fig. 27, Data Sheet 14.)

The type of pitot tube in which the suction hole is not diametrically opposite to the impact hole should not be used industrially, since small inaccuracies of the suction hole will cause large changes in the coefficient.

C. METHODS OF USE

76. Since the velocity is not uniform over any normal section of a stream, the velocity must be determined at enough points to give a fair average. In deciding the number of readings necessary, the extent to which the velocity distribution is likely to be irregular must be considered, and the apparatus must be arranged so as to ensure that readings are taken at all points where the velocity is likely to be abnormal.

77. *Circular ducts. Method* 1. If the cross section of the pipe is circular, it is presumed to be divided into a number of concentric rings of equal areas. Each of these rings is then sub-divided into two equal areas, by circles. (See Fig. 30, Data Sheet 15.) The inter-sections of these circles with a diameter determine the points at which the pitot tube is to be placed: thus on any one diameter twice as many readings as rings are obtained, half of the readings being made on either side of the centre, but none at the centre. Readings should be taken along two diameters at right-angles, one diameter being in the plane of any upstream irregularity (such as a bend) and if the values of :

$$\frac{\text{V mean}}{\text{V centre}}$$

differ by more than 5 per cent for the two traverses, additional intermediate diameters should be examined.

In all cases the number of concentric rings shall be five or more, so that at least ten determinations of the velocity on each diameter are required. Fig. 30 (Data Sheet 15) shows the positions of the points at which readings have to be taken, for the case of five rings.

Provision must be made for a continuous indication of the rate of flow; if this does not remain constant while the velocity head readings are being taken, each reading must be corrected to compensate for variations of flow, so that the values used in the calculation of the average velocity all correspond to the same rate of flow. This is most conveniently done by measuring the differential produced by a 'control' pitot tube, or other fixed obstruction, and multiplying the individual velocity head readings by :

$$\left(\frac{\text{'Control' differential taken as basis}}{\text{'Control' differential at moment of reading velocity head}}\right)$$

But where the flow is indicated by a meter reading directly in flow units, the observed velocity head readings must be multiplied by :

$$\left(\frac{\text{Rate of flow taken as basis}}{\text{Rate of flow at moment of reading velocity head}}\right)^2$$

This method is suitable for clean pipes where the diameter is known in advance and the correct positions can be worked out with certainty.

78. The average velocity is obtained not by averaging the individual velocity heads, but by averaging the square roots of the individual corrected velocity heads, and using this average value to compute the average velocity by the relation :

$$\text{V av.} = \sqrt{\frac{2g}{W}} \times \sqrt{\Delta P}\ \text{av.} \qquad (23)$$

For liquids this may be put in the form

$$\text{V av.} = \sqrt{2g} \times \sqrt{H}\ \text{av.} \qquad (24)$$

where 'H' is expressed in feet head of the liquid being measured. For gases

$$\text{V av.} = \frac{18{\cdot}29}{\sqrt{W}} \times \sqrt{h}\ \text{av.} \qquad (25)$$

where '*h*' is expressed in inches of water gauge.

79. *Circular ducts. Method* 2. Instead of placing the pitot tube at the points given in Fig. 30 (Data Sheet 15) and taking the average $\sqrt{h}$, the tube is moved from one side of the pipe to the other by equal increments (e.g. in a 20 in. main the first reading might be 1 in. from the wall, the next 2 in., the third 3 in. and so on): the velocity is calculated from the velocity head for each point, and is plotted on a graph (two graphs if two traverses at right angles are taken): the points of Fig. 30 (Data Sheet 15) are then marked on to the graph, and the velocities at these points are read off and averaged. The calculation of velocity is similar to that made for Method 1:

Liquids. $\text{V local} = \sqrt{2gH}\ \text{local}$ (26)

where H = feet head of liquid being measured.

Gases. $\text{V local} = \frac{18{\cdot}29}{\sqrt{W}}\ \sqrt{h}\ \text{local}$ (27)

where *h* = inches of water gauge.

One advantage of this method is that for accurate work a graph of the velocity distribution will in any case have to be prepared and as many extra readings as necessary will have to be taken in regions where abnormality is shown.

This method is particularly useful when the pipe is incrusted, as the additional readings taken near the incrustations establish easily the effective diameter of the pipe.

80. *Circular ducts.* (*Either method.*) After having made a traverse of a pipe and obtained the ratio between the centre velocity and the average velocity, the pitot tube may be set at the centre and this ratio used to give the average velocity from the centre reading.

While this simplifies future determinations of the average velocity, the result is subject to some uncertainty, since the ratio depends on the Reynolds number.

In long pipes (i.e. where the pipe is straight for 50 diameters or more) the value of

$$\frac{\text{V mean}}{\text{V centre}}$$

approximates to published values (which values vary with the Reynolds number) but in every case traverses should be taken, in order to determine this ratio, unless the objections to this are insuperable.

As an example of the extent to which the ratio

$$\frac{\text{V mean}}{\text{V centre}}$$

may vary, it has been found that in a long smooth drawn 6 in. pipe carrying water at normal velocity the ratio changes about 1 per cent when the flow is doubled or halved. The changes of

$$\frac{\text{V mean}}{\text{V centre}}$$

which occur when the pipe is *not* long and uniform are very much greater, and values ranging from 0·5 to 1·0 may occur.

81. *Rectangular ducts* (*Figs.* 28 *and* 29, *Data Sheet* 14). If the cross section of the pipe is rectangular, it is to be divided into a number of equal rectangles, geometrically similar to the cross section: the velocity shall be measured at the centre of each of these areas, and the average velocity calculated from the average of the square roots of the measured velocity heads.

The number of rectangles must not be less than 16, and must be even.

In all cases where the number of rectangles is less than 100, additional readings shall be taken in the areas adjoining the wall of the pipe. These readings shall be taken at points 1/6 and 5/6 of the width of the area from the wall of the pipe. In each of these areas, except those at the corners, the average of the three values $\sqrt{h}$ shall be taken as the average for that area. In the areas at the corners, each of which contains two sets of three readings (the reading at the centre of the area being included in both sets) the average value of $\sqrt{h}$ for the area is given by :

$$\text{Average for area} = \frac{\text{(average of one set)} \times \text{(average of second set)}}{\text{value at centre of area}} \qquad (28)$$

In finding the average for the whole pipe, the average values of $\sqrt{h}$ for the areas adjoining the walls found by the methods described above, are to be combined with the values measured at the centres of the other areas. Average velocity (as set out in method 1 for circular pipes) is calculated from the relations.

Liquids : $\text{V av.} = \sqrt{2gH}\ \text{av.}$ H = ft. of liquid (29)

Gases : $\text{V av.} = \frac{18{\cdot}29}{\sqrt{W}}\ \sqrt{h}\ \text{av.}$ *h* = inches of water gauge. (30)

* g = acceleration due to gravity = 32·19 ft./sec./sec.

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