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The averaging pressure tubes flowmeter for the measurement of the rate of airflow in ventilating ducts and for the balancing of airflow circuits in ventilating systems‡

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Synopsis

This paper describes an original investigation of a new flowmeter and a method of balancing of airflow circuits in low pressure ventilating systems.

The flowmeter is simple and robust in its construction, imposes virtually no resistance to the airflow circuit, requires only short lengths of duct before and after the measuring point and is consistent in its performance. The accuracy of the flowmeter from laboratory tests was found to be better than ± 6.5% for air velocities from 600-1400 feet per minute.

The method of balancing using precalibrated air volume dampers has been shown to be reliable and quick. Only two adjustments were required to bring the airflow circuits tested in the laboratory to balance. This method took only one quarter of the time required by a 'Systematic Trial and Error' method in balancing an airflow circuit with four branches in the laboratory.

Field tests on the flowmeter and the method of balancing confirmed the above findings.

Introduction

A survey of the literature describing methods for the measurement of the rate of airflow in ventilating systems and of methods for the balancing of airflow circuits in ventilating networks was made by the author on behalf of the Hospital Engineering Research Unit at the University of Glasgow in 1962. The aim of this survey was to find suitable methods for the Unit's Hairmyres Experimental Ward Unit which was at its design stage at that time. This experimental ward unit was planned with a flexible air-conditioning system for the study of air transfer from bed to bed within a room, and from room to room within a ward unit and for the investigation of air movement control. In these studies, the assessment of the rate of airflow and the balancing of airflow circuits were of fundamental importance.

This survey showed that the available methods were unsuitable for the field measurements envisaged for the experimental ward unit. At the same time, it was realized that the importance of measuring the rate of airflow and of balancing of airflow circuits was not confined to the studies to be carried out at the experimental ward unit. It applied to the ventilating industry as a whole.

In modern air-conditioned buildings, such as laboratories, aerospace factories and hospitals, the air-conditioning or ventilating systems are called upon not only to meet the heating and cooling loads in the buildings but to pressurize certain parts of the building complex and to maintain a certain type of air distribution pattern within certain rooms. These requirements cannot be properly fulfilled unless there are reliable and simple methods by which the rate of airflow can be metered and the airflow circuits can be balanced.

A further enquiry was therefore conducted which included visits to a number of large heating and ventilating consulting and contracting firms in this country. The aim was to find out the current practice in measurement and balancing.

This enquiry revealed that measurements were made either by the laborious method of pitot-static tube traversing or by the unreliable method of measuring at the air supply diffusers and at the air extract grilles. Balancing was done by trial and error methods which were very time consuming. The result was that either a long period of balancing time was required or that balancing was largely neglected.

The need for a simple and reliable method for the measurement of the rate of airflow and the need for a technique which could simplify the process of balancing of airflow circuits were clearly indicated for the ventilating industry as a whole.

It was to meet the first of these needs that the averaging pressure tubes flowmeter was devised and tested in the laboratory. Continuing this work the basic principle of balancing of airflow circuits was generalized and applied together with the flowmeter and calibrated air control dampers in the laboratory tests. Results from these laboratory tests showed an encouraging practicability. Hence the flowmeter and calibrated dampers were exclusively installed in the experimental ward unit. Further field tests were carried out at the experimental ward unit. The results confirmed those established in the laboratory.

PART 1. THE AVERAGING PRESSURE TUBES FLOWMETER.

1. A survey of methods for the measurement of the rate of airflow in ventilating systems

This section describes the results of a survey made on the practicability of the generally available methods for measuring the rate of airflow in ventilating systems. The survey has been conducted with emphasis on the accuracy, the pressure loss and the requirements of straight length of duct of the methods.

The various methods can be divided into the following three groups.

1.1 Pressure difference caused by a constriction

A constriction in a duct is frequently used as a device for measuring the rate of airflow. The constriction causes a pressure variation which can be directly related to the rate of airflow. The most commonly used devices of this kind are the venturi meter, nozzles and orifices. (1), (2), (3)

The relationship between the rate of airflow and the pressure difference can be predicted theoretically by assuming an ideal flow and applying the Bernoulli principle and the law of continuity simultaneously. However in a real fluid because of frictional resistance and the energy losses occurring across the constriction and the mechanical imperfection of the constriction, the theoretical prediction has to be corrected by an experimental coefficient to bring it into conformity with reality.

These devices can give a high degree of accuracy (about 1% under favourable conditions) with an inclined type of manometer when constructed and installed properly. However, they have very limited use for field work, as they impose a high resistance pressure loss on the air circuit. They also require a long straight length of duct before the measuring point and are expensive.

Similar to this group of device is the elbow meter (4) which utilizes the pressure difference of the fluid on the inside and outside curves of an elbow. The advantage of this meter is that there has been no additional resistance to flow due to the elbow being converted into a meter. However, here again a long straight length of duct is required (about 25 duct diameters in length before the meter and 10 duct diameters after it) to give an accuracy in the region of $\pm 10\%$ without actual calibration in situ.

1.2 Velocity pressure

The velocity pressure of a moving air stream can be obtained by measuring the difference of the total pressure and the static pressure caused by the air stream. The total pressure can be accurately determined by means of an open-ended tube facing the air stream (5) and the static pressure can be measured by holes equally spaced on the circumference of the tube at some distance from the open-end, or by tube(s) flush to the internal surface of the duct. The velocity pressure can be converted into velocity in accordance with a power law. The rate of airflow is then the product of the mean air velocity in the duct and the area of the duct.

Instruments of this kind impose virtually no resistance on the airflow circuit. However, the velocity pressure of the air stream is usually small, hence a sensitive manometer is required to give a good accuracy.

The following describes some of the most common devices of this kind.

The pitot-static tube: This consists of an open-ended tube for the measurement of total pressure and a number of holes equally spaced around the circumference of the tube at a distance away from the open-end for the measurement of static pressure. Standards for the construction of several types of tube have been specified in the British Standard Code (6). The correction factor to be applied to one of these, the National Physical Laboratory Hemispherical Head Pitot-static Tube, is unity over a large range of Reynolds number (from 330 to 360 000) (7). If the tube is used with a micromanometer, an accuracy of one per cent can be obtained down to an air velocity of 10 feet per second.

In the measurement of the rate of airflow by the pitot-static tube, a traverse has to be made with the tube taking measurements of the velocity pressure along a diameter of the duct at predetermined intervals. The rate of airflow is then the integration of the product of the velocity and the area through which the velocity exists. This integration can be done numerically or graphically.

This method is very reliable. In fact it has been used as a standard for the calibration of other methods in the measurement of the rate of airflow. However, it is laborious in its application as at least 16 point measurements have to be made at the predetermined location across the duct (8) and the measured velocity pressures have to be converted into velocities individually. The arithmetic mean of the velocities is then multiplied by the cross-sectional area of the duct to give the rate of airflow.

In order to simplify the process of traversing, *a central pitot-static tube method* has been proposed. In this method only one reading of the velocity pressure from a pitot-static tube located at the axis of the duct is required in order to calculate the rate of airflow. This method is based on the fact that in a fully developed turbulent flow with a certain Reynolds number in a duct with a certain roughness factor, there is a fixed ratio between the mean velocity of the flow and the velocity at the axis of the duct (9), (10). In order to establish a fully developed turbulent flow, a straight length of duct in the region of 90 equivalent diameters of the duct is required upstream from the measuring point. This requirement normally excludes the use of this method for field measurements. Two other methods have been devised to simplify the process of converting the velocity pressures into velocities individually; one of them is called a *Multi-tube Differential Pressure Manometer* (11). This device converts the point measurements of velocity pressure in a traverse into the mean of velocities by curved small bore cylindrical tubes. This is again a delicate instrument and is not easily portable. The other is called a *Three-Quarter Radius*

Pitot-Tube Flowmeter[12]. The theory of this meter assumes that the average velocity of a wide range of velocity distribution patterns in accordance with a power law can be obtained by four pitot-tubes equally spaced at three-quarter radius of a cylindrical duct and four static tubes flush to the internal surface of the duct also equally spaced around the circumference of the duct at 45 angular degrees from the pitot tubes. If a straight length of duct 15 diameters long is provided before the flowmeter, an accuracy of $\pm 0.6\%$ is claimed to be possible at velocities from 100-450 ft/sec. This is outside of the range used in ventilating systems but no information is given regarding the accuracy at lower velocities.

1.3 Direct measurement

The velocity of an air stream can be measured directly without the intermediate step of converting velocity pressures into velocities. There are two basic direct measuring instruments.

Vane anemometer. It is in effect a windmill consisting of a number of light vanes mounted on a spindle. The motion of the spindle is communicated to a number of pointers moving over graduated dials. The dimensions, vane inclinations and communicating gear-ratios are so proportioned that the number of feet indicated on the dials in unit time is approximately equal to the distance travelled by the air in the same time or to the air velocity.

This type of instrument is not very consistent in its performance. It has been shown in an experimental work[13] that there were deviations of 6% in indicated velocity between three different sizes of vane anemometer —3 in, 6 in and 10 in— when they were used to measure the air velocity of the same air stream at the end of a long duct. Another report[14] stated that "the change in calibration that can occur in the less consistent instruments is some 5% of the true reading, which may be due to nothing more than a need to clean the instrument, but this is not always the case". In the same report, it has been stated that there was an even greater error when the anemometer was used in conditions other than those in which it was calibrated. This error has been shown[15] to be in the range of +5 to +23% when the anemometer readings of an airflow were compared with those given by an Orifice plate measurement.

The velometer. This device consists of a balanced, dampered pivoted vane in a case. The vane can be deflected by a very light current of air. The degree of deflection is indicated over a graduated scale. There is very little information available on the performance of this type of instrument. It is, however, believed that it would be similar or inferior to that of the vane anemometer.

These two types of instrument have been widely used in measuring the rate of airflow at the face of air supply diffusers and extract grilles in the ventilating industry. The accuracy of these types of measurement depends on the position where the instrument is held in relation to the airstream, the irregularities before the supply outlet, the design of diffuser and grilles. Although different manufacturers in the United States have given factors to be applied to the measurements of the rate of airflow for each individual type of diffuser[16], and a test code has been established[17], accuracy of field measurements using these types of instrument is likely to be in the region of $\pm 15\%$.

From the survey described above, it can be seen that the available methods are unsuitable for the measurement in field. The methods in the first group are impracticable on the ground of high cost, high pressure loss and the requirement of long lengths of straight duct before and after the measuring point. The methods in the second group are either laborious in their application or requiring a long length of straight duct before the measuring point. The methods in the third group have inherent uncertainties about their performance.

2. The averaging pressure tubes flowmeter

In the previous section, it has been shown that the available methods are unsuitable for the measurements of the rate of airflow in ventilating systems. Therefore there is a need for a better method for field measurements. This method has to be simple and robust in its construction, require short lengths of straight length of duct before and after the measuring point, impose a low resistance to the airflow circuit and have a consistent performance.

It was to meet these requirements that the idea of the Averaging Pressure Tubes Flowmeter was conceived. The principle of the flowmeter is closely related to the velocity integration method in which the rate of airflow in a duct is an integration of the product of the velocity and the area through which the velocity exists. This is given by

$$Q = \int^{A} V dA$$

where Q — rate of airflow,
V — point velocity,
A — cross sectional area of duct.

The velocity can be measured by the pressure difference between the total pressure and the static pressure. This relationship takes the form,

$$P_V = P_T - P_S = \tfrac{1}{2}\rho V^2$$

where P_V — velocity pressure,
P_T — total pressure,
P_S — static pressure,
V — mass density of the fluid,
ρ — velocity.

$$\text{or } V = \sqrt{\frac{2P_V}{\rho}}$$

If the velocities are measured at small discrete intervals of distance across the duct, a velocity profile can be constructed and the integral for the rate of airflow can be evaluated by a graphical method. However it has been proved[18] that it is easier and equally accurate to evaluate the rate of air by the product of cross-sectional area of the duct and the mean of the velocities. This can be shown in mathematical form as follows:

$$Q = A\overline{V} \text{ ——— EQ. (1)}$$

where $\overline{V}$ — mean air velocity,
Q — rate of airflow,
A — cross-sectional area of duct.

and

$$\overline{V} = \frac{V_1+V_2+V_3+\ldots}{n} = \frac{\sum_{i=1}^{n} V_i}{n} = \frac{\sqrt{\frac{2}{\rho}}\sum_{i=1}^{n}\sqrt{P_{Vi}}}{n} \text{ —EQ.(2)}$$

where V_i—velocities measured at small discrete distance intervals across the duct,
n — number of velocity measurements.

This is a laborious process in evaluating the rate of airflow because for each value of the rate of airflow a large number of point measurements of velocity have to be made and a lengthy arithmetical operation is involved to produce the mean velocity. This process can be simplified if it is assumed that the mean velocity of airflow in a duct is the square root of the average of the velocity pressures instead of the average of the individual square roots of the velocity pressure. In other words, Eq.(2) is now written as

$$\bar{V} = \sqrt{\frac{2}{P}} \sqrt{\frac{\sum_{i=1}^{n} P_{vi}}{n}} \qquad \text{EQ. (3)}$$

This averaging of velocity pressures can be done by mechanical means. The Averaging Pressure Tubes Flowmeter is based on this assumption. The flowmeter consists of two tubes crossing each other with small holes drilled equally spaced on the longitudinal side of the tubes facing the airstream and four static pressure holes on the circumference of the duct flush with the internal surface of the duct as shown in Fig. 1. The two cross tubes measure the average of the total pressures and the four tubes on the circumference of the duct measure the static pressure in the duct. The difference in these two pressures gives the average of the velocity pressure.

However, a number of errors and uncertainties are introduced with this assumption and this device.

(1) Whether the averaging pressure tubes will give a true average of the individual pressures?

This would seem to depend on the pressure differences imposed along the face of the tubes, size of the tubes, size of the drilled holes. One experimental work [19] on a tube with ten equally spaced holes and with different static pressures imposed on each hole has shown that the tube averaged to the arithmetic mean of the pressure imposed on each hole. However, it is still uncertain whether this holds true for velocity pressures in a real flow.

(2) The difference between the average of the individual square roots of the velocity pressures and the square root of the average of the velocity pressures as measured by the flowmeter.

This would seem to depend on the pattern of velocity distribution. The difference can be evaluated mathematically if the distribution is known. However, uncertainties exist about the relationship between the velocity and the distance from the wall of a duct at which the velocity is taken. It has been suggested that this relationship follows a power law [20] in the following form for a symmetrical velocity distribution.

$$\frac{V}{Va} = \left(\frac{Y}{R}\right)^{\frac{1}{m}}$$

where V — point velocity at a distance Y from the wall,
Va — velocity at the axis of duct,
R — radius of duct,
m — a power factor.

Another author [21] suggests a Log-linear relationship for a fully developed and symmetrical flow which takes the following form.

$$V = A + B \log (Y/D_a)$$

where D — diameter of duct,
A & B — constants.

For a turbulent but not fully developed and asymmetrical flow at a short distance from a right angle bend, the relationship becomes more uncertain. At this position, the flow is subjected to the effect of swirling created by the bend. In other words, the flow is not parallel to the axis of the duct but is a three-dimensional flow having three velocity components perpendicular to each other. The degree of influence of swirling on the performance of the flowmeter would depend on the mean air velocity, the size of duct, the length of straight duct before and after the flowmeter and the different configurations of bends upstream from the flowmeter.

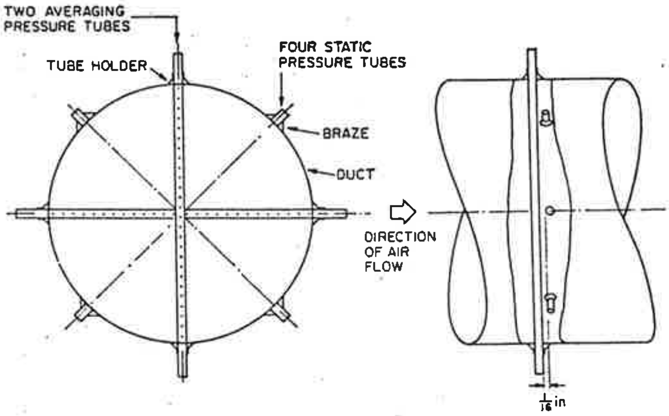


Fig. 1—Averaging pressure tubes flowmeter.

(3) The instrumental and the human errors.

The accumulated effect of these factors on the performance of the flowmeter can only be expressed by an experimental coefficient (C) which can be represented as

$$\bar{V} = C\sqrt{\frac{2}{\rho}} \sqrt{\Delta P_{vi}}$$

$$\text{and} \quad Q = C\sqrt{\frac{2}{\rho}} \sqrt{\Delta P_{vi}}\,(A)$$

$$\text{or} \quad C = \frac{Q_a}{Q_i}$$

where ΔP_{vi} — pressure difference measured by the flowmeter,
C — experimental coefficient,
Q_a — actual rate of airflow,
Q_i — indicated rate of airflow.

The following sections present the laboratory and field work on the assessment of these effects collectively and individually when applicable.

3. Laboratory layout

Fig. 2 shows the general layout of the apparatus in the laboratory. Figs. 4 and 5 illustrate some of the details of the layout.

An axial flow fan was used to provide the means of promoting airflow. The rate of air discharge was controlled by an airflow control box.

A 'Betz' manometer of an accuracy of 0.001 in of water gauge was used to measure the pressure differences from the flowmeter and the venturi meter. Two venturi meters, one 10 in and another 6 in, were used to measure the rate of airflow. These venturi meters were calibrated in the laboratory.

The barometric pressure was measured by a 'Fortins Principle' barometer with a vernier reading to 0.002 in of mercury gauge.

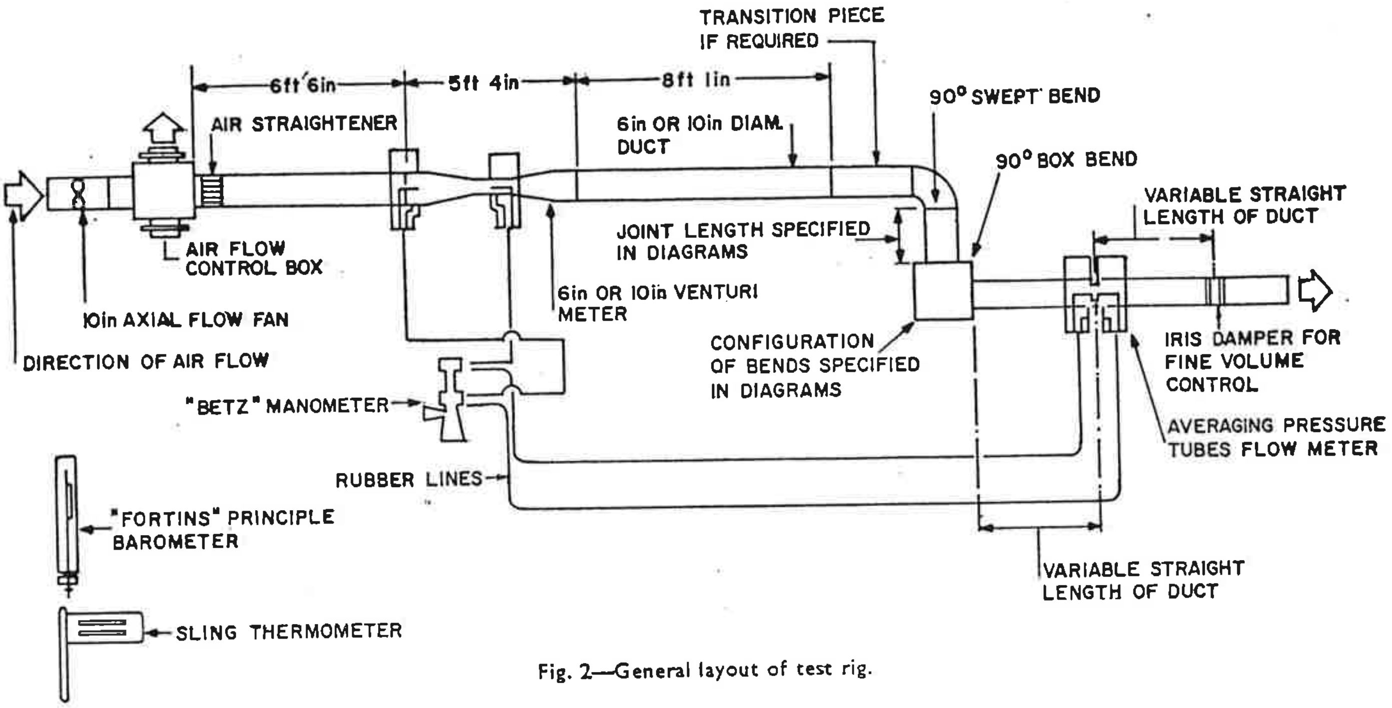


Fig. 2—General layout of test rig.

The wet and dry bulb temperatures were measured by a sling thermometer.

Spiral wound cylindrical ducts were selected for the test in preference to rectangular ducts for three reasons.

(1) There is only one variable to be considered in a cylindrical duct, namely the diameter of the duct.
(2) It is easier for the measurement of the rate of air flow by the flowmeter as there are no corners in the cross section of a cylindrical duct.
(3) The cost of spiral wound ducting is comparable to that of rectangular ducting. This is a point to be considered here as it will become a major factor when the flowmeter is to be used in large ventilating systems.

A special 'Box' bend was made and installed upstream from the flowmeter as shown in Fig. 2.

Fig. 3 shows a sample 'Box' bend. One side of this box can be removed for inspection and cleaning of the duct. The external dimension of the 'Box' bend is four inches wider than the diameter of the circular duct openings on it. The openings were placed at the centre of the faces of the 'Box' bend. The boxes were made of 22 swg galvanized metal sheet.

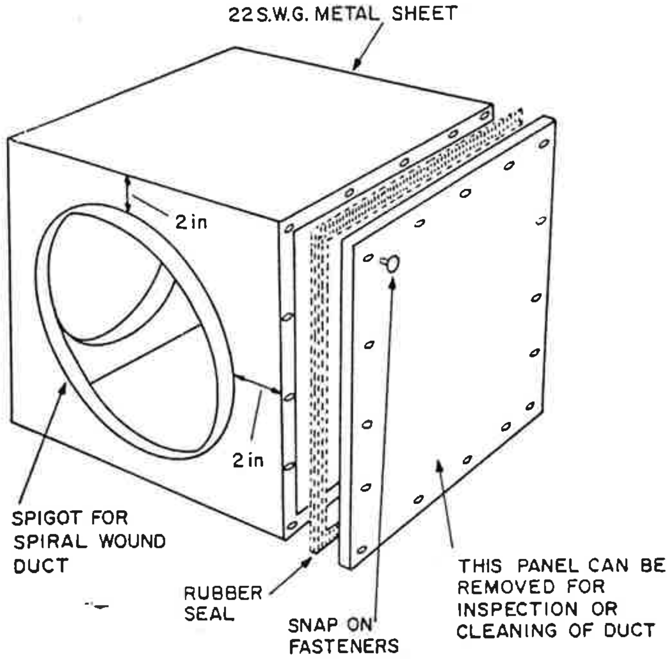


Fig. 3—A box bend.

There were two reasons for the use of the box bends. Firstly, to provide access to the internal surfaces of the ventilating ducts for inspection and for cleaning if required. Secondly, to act as a 'baffle' to randomize the different degrees of swirling created by different irregularities, such as bends, before the flowmeter.

The need for provision of access for cleaning has been the subject of much discussion, particularly in hospitals and especially for ducts serving critical areas such as operating rooms, burns dressing stations, treatment rooms and transplantation isolation cubicles. It is also a common practice to clean the internal surfaces of ventilating ducts handling greasy and corrosive vapours extracted from kitchen and other relevant industries.

4. Preliminary work leading to the construction of the flowmeter

An examination was carried out on the factors which may influence the characteristics of the averaging pressure tube(s), the static pressure tube(s) and the flowmeter in the measurement of the rate of airflow.

4.1 Characteristics of the averaging pressure tube

4.1.1 Size of the tube

It has been shown [22] that the size of the tube does not influence the measurement of total pressure of an airflow so long as the diameter of the tube is not greater than one-twelfth of the diameter of the duct. Hence a commercially available 16 swg copper tube of $\frac{3}{8}$ of an inch outside diameter was selected for all sizes of duct. For a duct of six inches diameter, the tube diameter is only one-sixteenth of that of the duct.

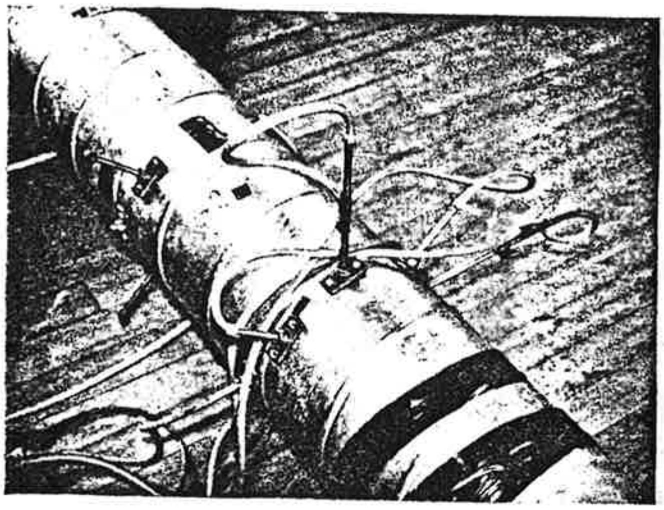


Fig. 4—A prototype flowmeter.

4.1.2 The opening(s) in the tube

As shown in the theory of velocity integration, it is ideal to take an infinite number of velocity readings in order to construct a true velocity profile or to calculate the mean velocity accurately. This is specially true when the velocity distribution pattern is asymmetrical with respect to the axis of the duct. It follows, therefore, that a tube with a slot across its longitudinal face would be ideal for the measurement of the average of the velocity pressures.

However, a slotted tube is less rigid in its physical construction and hence can be more easily distorted or damaged than a tube with holes drilled on it. It seemed therefore to be desirable to investigate the effect of the width of the slot and the difference between the slotted tube and the tube with drilled holes.

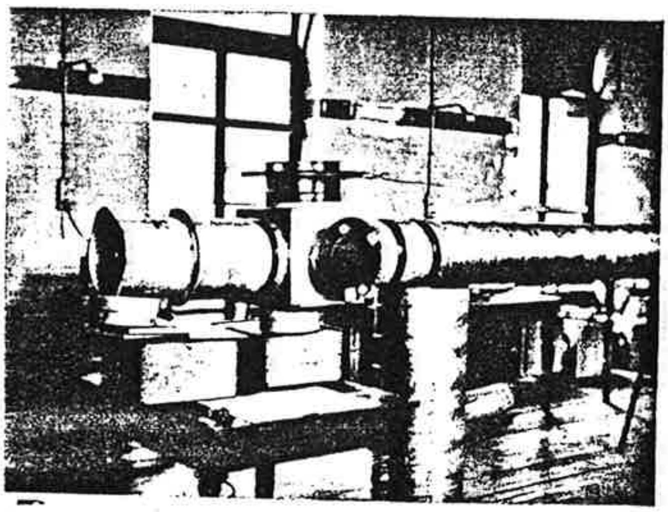


Fig. 5—Airflow control box.

Four tubes were made, three of them had slots of 1/16, 1/32 and 1/64 of an inch wide respectively and the fourth one had 1/16 of an inch diameter holes drilled at $\frac{1}{4}$ of an inch spacing as shown in Fig. 6.

Tests were conducted in a 10 in diameter duct after a Box bend. The tube was inserted along the diameter of the duct parallel to the plane of the bend at four equivalent duct diameters downstream from the bend and a static pressure tube was fitted flush to the internal surface of the duct, on the same circumference as the tube but at an angular position of 90° from the tube.

The air velocity as measured by the venturi meter was increased in five steps from 600 to 1 000 ft/min. At each step, the pressure differences between the static pressure tube and each end of the tube and a point to which both ends of the tube were connected were taken. Four readings were taken for each pressure difference and the arithmetic mean was recorded as the indicated velocity pressure. The

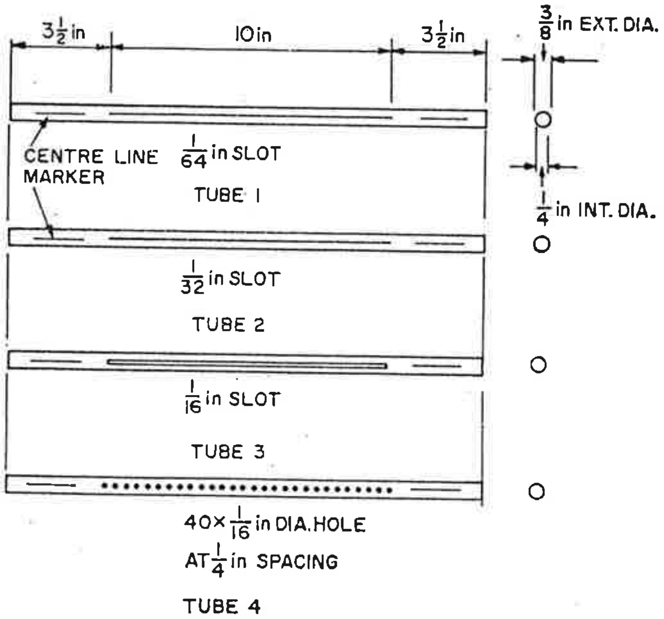


Fig. 6—Averaging pressure tubes.

indicated velocity pressures were converted into velocities. The actual air velocity in the duct was measured by the venturi meter.

This test procedure was carried out for each of the four tubes.

Fig. 7 shows a comparison of the indicated air velocities with their respective actual air velocities for each of the four tubes.

From this graph, the following conclusions can be drawn.

(a) When both ends of an averaging tube were connected to a common point, it gives a good average of the readings obtained from either end.

(b) In comparing the readings obtained from either end of the tube, the 1/16 of an inch slotted tube is most sensitive to the velocity distribution pattern and the tube with 1/16 of an inch drilled holes is the least sensitive. However, there is very little difference between the readings from these tubes when both ends of the tubes were connected to a common point. In other words, the width of slots and the size of holes have little influence on the measurement of the rate of airflow when both ends of the tube were connected

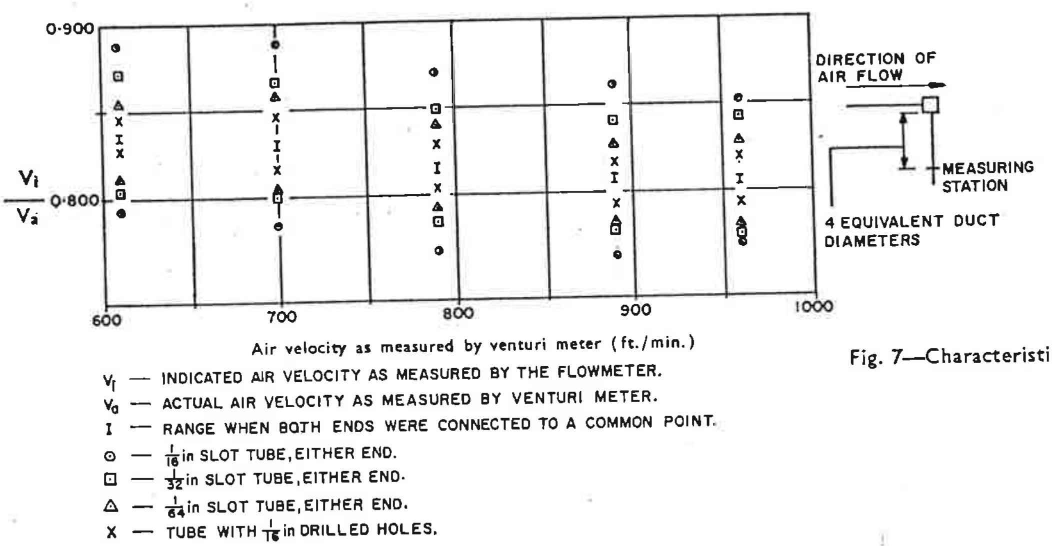


Fig. 7—Characteristics of averaging pressure tubes

to a common point over the range of the air velocities tested.

From these conclusions, it seemed that if both ends of the tubes were connected to a common point, the various openings would have equal merit in the measurement of the rate of airflow. This is in accordance with work on the pitot-tube[23] which has shown that variations in size and shape of the tube within wide limits were not accompanied by any errors in the readings of total pressure.

However, the tube with drilled holes offered better rigidity in its physical construction, hence it was chosen for the rest of the test.

4.1.3 Effect of alignment (or yaw) of the tube

The tube with drilled holes was placed in the duct at four equivalent diameters downstream from a Box bend. The pressure difference between the static pressure tube and the point to which both ends of the averaging pressure tube were connected was measured as the indicated velocity pressure. The indicated pressures were converted into indicated velocities. The actual air velocity in the duct was measured by the venturi meter and was maintained at 800 fpm. The tube was rotated at intervals of 2° from the plane where the axis of the holes was parallel to the axis of the duct.

Fig. 8 shows the effect of inclination of the tube when the indicated velocities were compared with the actual velocity in the duct. From this graph, it can be seen that appreciable errors arise when the tube was inclined t more than five degrees and the main air stream was abou 2° from the axis of the duct.

4.2 Characteristics of static pressure tube(s)

4.2.1 Size of the tube

It has been shown[24] [25] that in order to avoid th phenomenon of eddying, the size of the static pressure tu should not exceed 0.1 of the diameter of duct. Hence t size of tube used for the averaging pressure tube w suitable for the static pressure tube as well.

4.2.2 Number and location of the tubes

As stated previously swirling of air exists after an irreg larity, such as a Box bend. Hence the number and loc tion of the static pressure tubes should be such that the would give an average reading of the pressure around t circumference of the duct.

Tests were conducted with four static pressure tub flush with the internal surface of the duct and spac equally around the circumference of the duct at fo equivalent diameters downstream from a Box bend. constant static pressure at the inlet of the venturi me was maintained providing a static pressure of rougl 0.1 inch of water gauge at the measuring station. T pressure difference between the atmosphere and each the four static pressure tubes and the common point which the tubes were connected was measured. Fo readings were taken for each pressure difference and arithmetic mean was recorded as the static pressure.

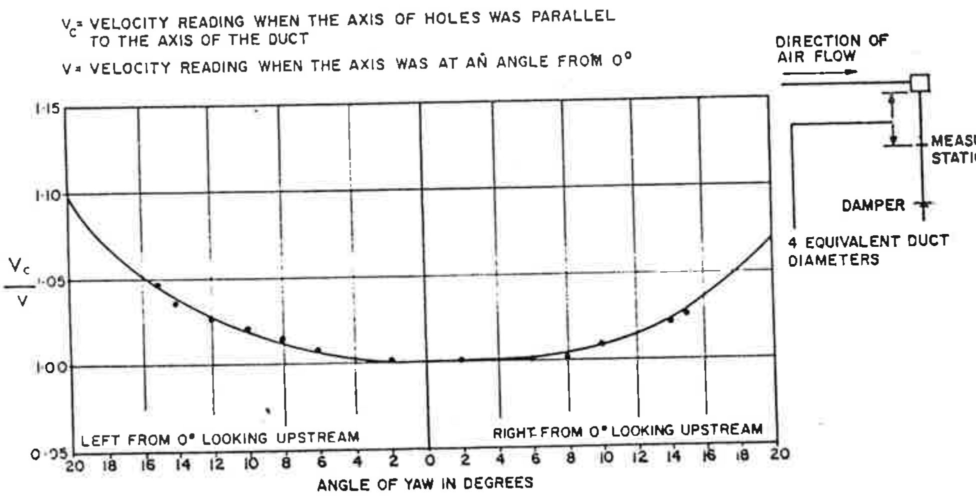


Fig. 8—Effect of yaw on velocity readings (air velocity at 800 ft./min.)

Table I shows the readings from the four tubes and the arithmetic mean of the readings. From this table it can be seen that the variation in the static pressure at these four locations from the arithmetic mean of them is in the range of 3%. The reading obtained from a common point to which the four static pressure tubes were connected agrees well with the arithmetic mean of the four individual readings.

Table I—Effect of the number and location of a static pressure tube(s) around the circumference of a ten-inch duct at four equivalent diameters downstream from a box bend when the static pressure was about 0.1 of an inch of water gauge

	Col. A				Col. B	Col. C
	0°	90°	180°	270°		
Ps / Ps(A.M.)	1·031	0·989	0·978	0·978	1·000	1·000

Col. A—Angular location of the static pressure tube around the axis of the duct. 0° being the location parallel to the plane of the bend and at the left-hand side when looking upstream.
Col. B—Arithmetic mean of the readings from the four static pressure tubes.
Col. C—Reading from a common point to which the four static pressure tubes were connected.
Ps—The static pressure readings.
Ps(A.M.)—The arithmetic mean of the four individual static pressure readings around the circumference of the duct.

A further test was conducted on the arrangement where the four static pressure tubes were connected to a common point when the duct was rotated around its axis with respect to the bend through 90° at four equal intervals. The result showed that there is no difference between the readings obtained at all these rotated positions.

On the evidence of the above results, the arrangement of four static pressure tubes at 90° apart around the circumference of the duct and connected to common points was adopted for the rest of the test.

4.3 Characteristics of the flowmeter with one and two averaging pressure tubes and four static pressure tubes

It has been shown in the previous sections that when both ends of an averaging pressure tube were connected to a common point, it would give a good average of the readings obtained from either end and that the reading obtained when four static tubes were connected to a common point agrees well with the arithmetic mean of the readings obtained from each of the four tubes. It would seem therefore that a flowmeter with one averaging pressure tube and four static pressure tubes as shown in Fig. 9a would be satisfactory. However, the effect of the angular position of the averaging pressure tube in relation to preceding bends had to be investigated. An alternative arrangement of the flowmeter was to have two averaging pressure tubes perpendicularly crossing each other as shown in Fig. 9b.

Tests were conducted on these two flowmeters. The flowmeters were placed at four equivalent diameters downstream from a box bend in a 10 in diameter duct system as shown in Fig. 2. The air velocity in the duct, as measured by the venturi meter, was maintained at 1 000 ft/min for all the tests. Six different configurations of box bends were used as shown in the sketches in Table II. For each configuration, the duct which contained the single tube flowmeter was rotated through 180° around its axis with respect to the box bend at 22½° angular intervals. The twin tube flowmeter was rotated in a similar manner through 90°.

At each rotated position, the pressure difference between the point to which the static pressure tubes were connected and the point to which the ends of the averaging pressure tube(s) were connected was measured. Four readings were taken for each pressure difference and the arithmetic mean was taken as the indicated velocity pressure. The indicated velocity pressure was converted into velocity.

The arithmetic mean of the readings for each configuration of bends was calculated. The deviation of each of the individual readings from the mean was calculated and expressed as the percentage of the mean.

Table II shows the maximum percentage deviation for each configuration of bends.

From this table it can be seen that the flowmeter with two averaging pressure tubes is superior to the flowmeter with one averaging pressure tube in that the angular

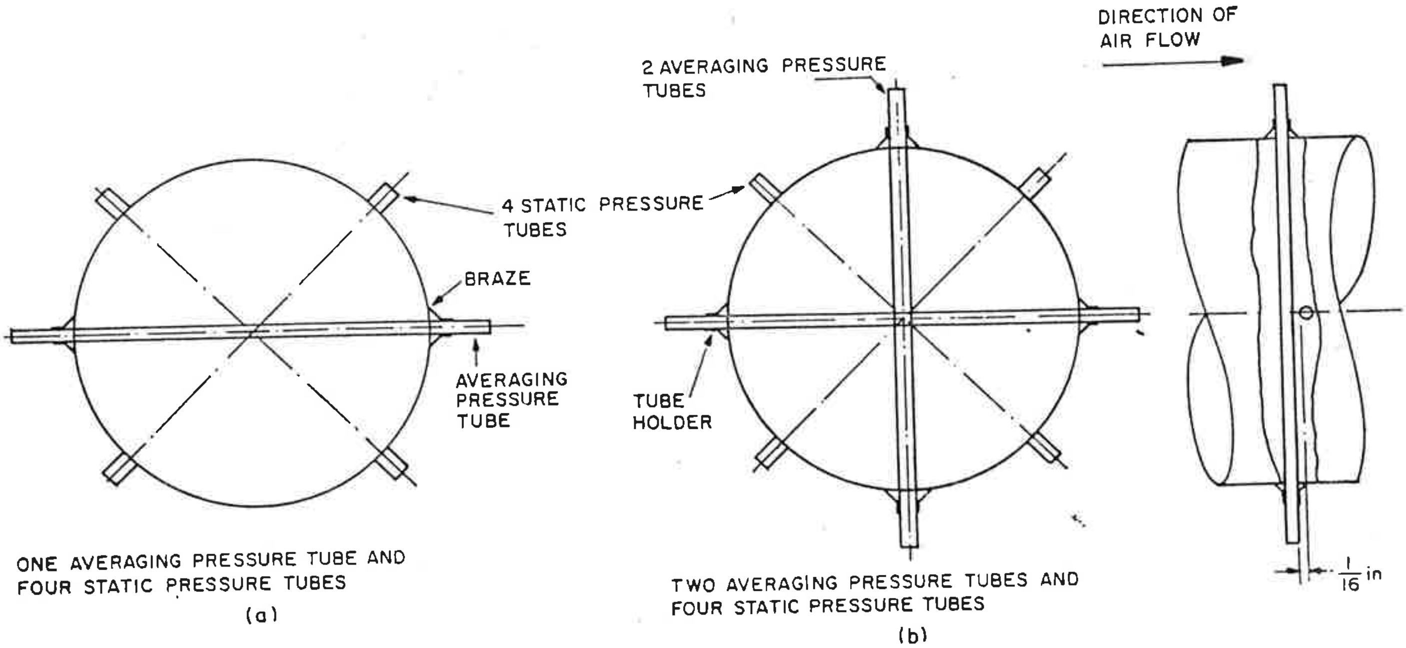


Fig. 9—Flowmeters.

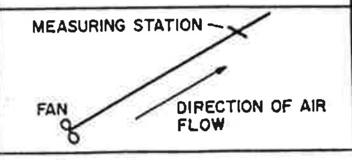
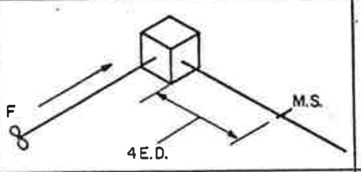
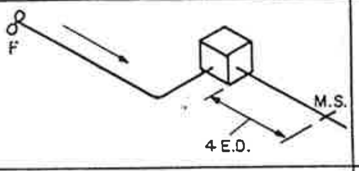
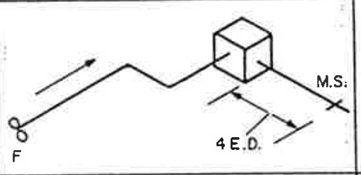
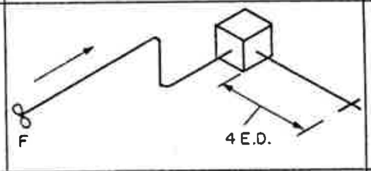
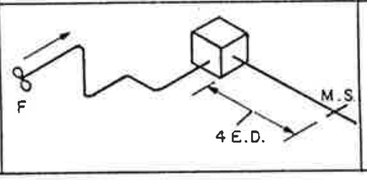
CONFIGURATION OF BENDS		COL. A.	
DESCRIPTION	SKETCH	COL. B.	COL. C.
NO BEND		0·00	0·00
ONE BOX BEND		0·82	0·00
TWO BOX BENDS		1·62	0·52
THREE BOX BENDS		3·10	0·00
THREE BOX BENDS IN THREE DIMENSIONS		3·10	0·00
FOUR BOX BENDS IN FOUR DIMENSIONS ..		2·30	0·00

Table II—Effect of the angular positions on the two flowmeters

Col. A—Maximum deviation of any one of the readings expressed as a percentage of its arithmetic mean of the individual readings.
Col. B—Flowmeter with one averaging pressure tube and four static pressure tubes. Measurements were taken at 22½° intervals when the duct was rotated through 180° around its axis with respect to the bend.
Col. C—Flowmeter with two averaging pressure tubes and four static pressure tubes. Measurements were taken at 22½° intervals when the duct was rotated through 90° around its axis with respect to the bend.

position of the former with respect to the box bend has little effect on the measurement of the rate of airflow. In other words, the different air velocity distribution patterns created by different configurations of box bends have very little effect on the flowmeter with two averaging pressure tubes in the measurement of the rate of airflow.

It was decided to use the flowmeter with two averaging pressure tubes and four static pressure tubes as shown in Fig. 9b for the rest of the test.

5. Tests on the flowmeter

The practicability of the flowmeter consisting of two averaging pressure tubes and four static pressure tubes as shown in Fig. 9b has been shown in the previous section. However, its performance can be influenced by a number of factors and therefore had to be investigated. This entailed firstly the determination of the straight length of duct required upstream and downstream from the flowmeter and secondly the assessment of the accumulated effect of the following factors on the performance of the flowmeter.

(a) The effect of different configurations of box bends upstream from the flowmeter.
(b) The effect of different lengths of connecting duct between bends.
(c) The effect of different sizes of duct.

In addition, the use of air straighteners in attempt to smooth out the swirling of air created by bends was examined and the effect of configurations of bends without a box bend was also investigated in order to assess the 'baffle' effect of the box bend. Field tests on the performance of the flowmeter were also carried out at the Hairmyres Experimental Ward Unit.

5.1 Test procedure and scope of the test

For each run of test—such as a test on one configuration of bends—about 21 pairs (or sets) of readings of the rate of airflow were taken. Each pair of readings consisted of one from the venturi meter and one from the flowmeter at the same rate of airflow. The rate of airflow was increased and then decreased in steps within the range from 600 to 1400 ft/min air velocities.

Wet and dry bulb temperatures and barometric pressure were taken before and after each run. The mean of each pair of readings was used for the calculation of the density of air for that run.

In order to take account of human errors, two observers were employed in taking all the readings, each took one half of the readings.

Table III shows the scope of the test. There were 32 runs consisting of 564 sets of readings excluding those in the pilot runs and those of subordinate interests to the test.

5.2 Treatment of data

From a number of pilot runs in the laboratory on different configurations of bends, it had been shown that a linear relationship existed between the indicated rate of airflow as measured by the flowmeter and the actual rates of airflow as measured by the venturi meter over the range of air velocities considered in this test. A linear regression line was therefore fitted in accordance with the least square rule for each run and for the combinations of runs (Fig. 10). This provided an equation which enabled estimates of the actual rates of airflow to be calculated from indicated rates of airflow. From these rates the experimental coefficient was calculated.

The Confidence intervals applicable to the estimates of the actual rate of airflow for the various indicated rates [27] were also calculated. This gave the degree of error involved in the estimates which was used as a means of assessing the effect of the different factors.

The influence of different densities of air during the tests on the measurement of the rate of airflow was eliminated by correcting the readings to a standard condition (0.075 lb/ft³). This rendered the test results universally applicable. The KDF9 computer at the university was used for these analyses.

In order to simplify the calculations, a conversion table, for air with a density of 0.075 lb/ft³ was established and is shown in Appendix I. From this table, the rate of airflow in ft³/min can be read by entering the velocity pressure and the size of duct.

DUCT SIZE	CONFIGURATION OF BENDS	ARRANGEMENT	NO. OF SETS
10 in	ONE BOX BEND, M.S. 1 E.D.	V; 1 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 2 E.D.	V; 2 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 3 E.D.	V; 3 E.D.; M.S.	22
10 in	ONE BOX BEND M.S. 4 E.D.	V; 4 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 5 E.D.	V; 5 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 6 E.D.	V; 6 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 7 E.D.	V; 7 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 8 E.D.	V; 8 E.D.; M.S.	21
10 in	ONE BOX BEND M.S. 9 E.D	V; 9 E.D; M.S.	24
10 in	TWO BOX BENDS NO JOINT, IN TWO DIMENSION	V; M.S.; 4 E.D.	12
10 in	TWO BOX BENDS 12 in JOINT, IN TWO DIMENSION	V; 12 in; M.S.; 4 E.D.	12
10 in	TWO BOX BENDS 36 in JOINT, IN TWO DIMENSION	V; 36 in; M.S.; 4 E.D.	12
10 in	TWO BOX BENDS 48 in JOINT, IN TWO DIMENSION	V; 48 in; M.S.; 4 E.D.	11
10 in	TWO BOX BENDS 60 in JOINT, IN TWO DIMENSION	V; 60 in; M.S.; 4 E.D.	11
10 in	TWO BOX BENDS REVERSED 12 in JOINT, IN TWO DIMENSION	V; M.S.; 12 in; 4 E.D.	14
10 in	TWO BOX BENDS REVERSED, 36 in JOINT, IN TWO DIMENSION	V; M.S.; 36 in; 4 E.D.	12

Table III—Scope of test.

V = Venturi meter.
M.S. = Measuring Station.
E.D. = Equivalent duct diameter.
→ = Direction of airflow.
8 = Fan.
□ = Box bend.
⌐ = 90° swept bend.

5.3 Tests

5.3.1 Determination of the length of straight duct required before and after the measuring station

In actual installations for ventilating buildings, it is impracticable to have long straight lengths of ducts due to space limitations and cost considerations. It was therefore desirable to select the shortest straight lengths of duct before and after the measuring station which provided a stable characteristic for the flowmeter.

Tests were conducted on a ten-inch diameter duct system and one box bend as shown in Fig. 2. Readings were taken with the flowmeter at different equivalent diameters of straight length of duct downstream from the box bend over the range of 600 to 1 400 ft/min air velocities. Fig. 11 shows the results which reveal that the experimental coefficients stabilize at three, four and five equivalent diameters downstream from the box bend where the confidence intervals are about 3 to 4%.

Further tests with a half closed iris damper at one, two and three equivalent diameters downsteam from the measuring station showed that the existence of the damper had no influence on the performance of the flowmeter.

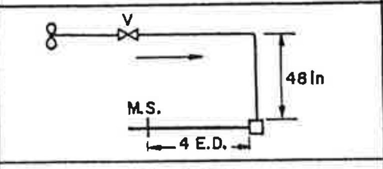
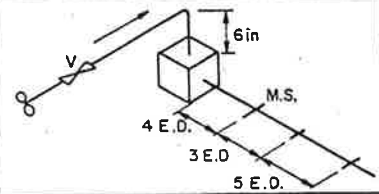
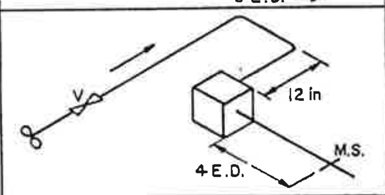
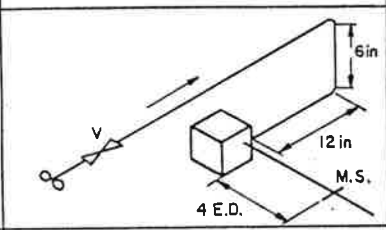
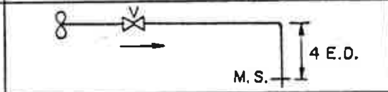
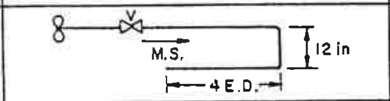
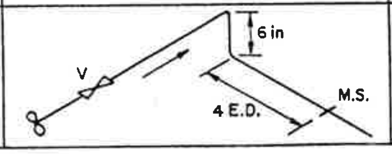
On the evidence of the results, a straight length of duct of four equivalent diameters long between the box bend and the flowmeter and two equivalent diameters after the flowmeter were selected for the rest of the test.

5.3.2 Tests on air straighteners

The major cause of the error in the measurement of the actual rate of airflow by the flowmeter is the swirling of air created by the box bend. It has been shown (28) that air straighteners can reduce the swirling.

Tests were conducted on three types of air straighteners as shown in Fig. 12 in order to assess their effects on the characteristic of the flowmeter. The air straighteners were placed at the outlet of a box bend in a 10 in duct system.

The results showed that the use of air straighteners does not improve the accuracy of the flowmeter. Hence no further tests on air straighteners were conducted.

DUCT SIZE	CONFIGURATION OF BENDS	ARRANGEMENT	NO. OF SETS
10 in	TWO BOX BENDS REVERSED, 48 in JOINT, IN TWO DIMENSION		12
10 in	TWO BOX BENDS IN THREE DIMENSION 6 in JOINT		13
10 in	THREE BOX BENDS IN TWO DIMENSION, 12 in JOINT		13
10 in	THREE BOX BENDS IN THREE DIMENSION, 6 in AND 12 in JOINTS		14
10 in	ONE 90° SWEPT BEND		14
10 in	TWO 90° SWEPT BENDS IN TWO DIMENSION, 12 in JOINT		15
10 in	TWO 90° SWEPT BENDS IN THREE DIMENSION, 6 in JOINT		14

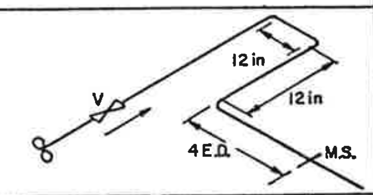
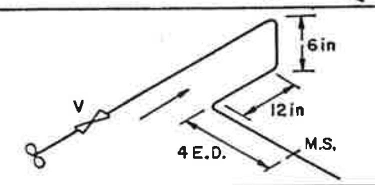
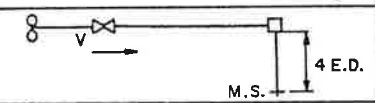
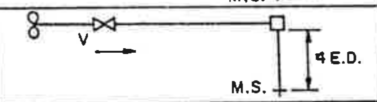
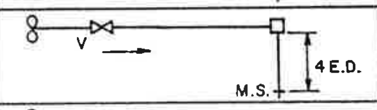
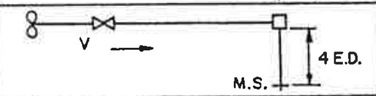
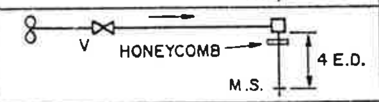
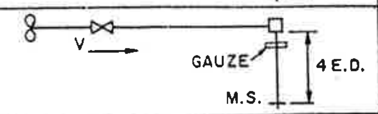
DUCT SIZE	CONFIGURATION OF BENDS	ARRANGEMENT	NO. OF SETS
10 in	THREE 90° SWEPT BENDS IN TWO DIMENSION, 12 in JOINTS		14
10 in	THREE 90° SWEPT BENDS IN THREE DIMENSION, 6 in AND 12 in JOINTS		13
6 in	ONE BOX BEND		20
8 in	ONE BOX BEND		24
12 in	ONE BOX BEND		21
14 in	ONE BOX BEND		40
10 in	ONE BOX BEND HONEYCOMB AIR STRAIGHTENER.		20
10 in	ONE BOX BEND PERFORATED PLATE AIR STRAIGHTENER		19
10 in	ONE BOX BEND GAUZE AIR STRAIGHTENER		21

Table III, continued

5.3.3 Tests on different lengths of connecting ducts between and different configurations of bends

Tests were conducted on the effects of different lengths of connecting duct between bends and different configurations of bends on the performance of the flowmeter in a 10 in diameter duct system as shown in Fig. 2. In each case, the last bend was a box bend and the preceding ones were 90° swept bends of a mean radius equal to the diameter of the duct. The flowmeter was located at four equivalent diameters downstream from the box bend.

The results obtained led to the conclusion that the different length of connecting ducts between bends and the configuration of bends in a 10 in duct system with a box bend before the flowmeter had no significant effect on the performance of the flowmeter and that the confidence intervals of the estimate of the actual rate of airflow were in all cases less than $\pm 6.56\%$ for air velocities from 600 to 1400 ft/min.

5.3.4 Tests on different sizes of duct

Tests were conducted on 6, 8, 10, 12 and 14 in diameter ducts with a box bend. The flowmeter was located at four equivalent diameters downstream from the box bend as shown in Fig. 2.

It was concluded from the results that the different sizes of duct had no significant effect on the performance of the flowmeter.

However, whether there was a significant difference between the effects of the different configurations of bends and the different sizes of duct on the performance of the flowmeter had to be investigated. Since these effects appeared to be of a random nature, a statistical test of the difference of the means was carried out to compare the numerical results of the tests on different configuration of bends with the results of the tests on different sizes of duct. From this test it was concluded that the test results of the combination of all the configurations of bends in

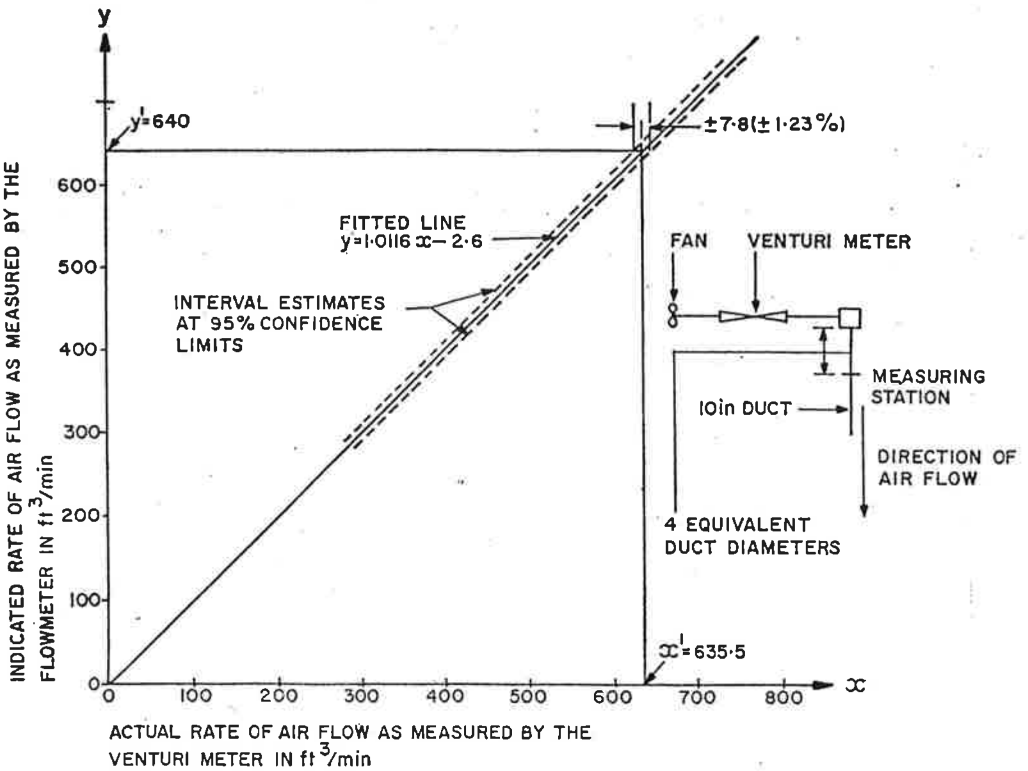


Fig. 10—Linear regression line and the confidence interval of the results from a test.

a 10 in diameter duct system, as shown in the preceding section, could be applied universally to the different sizes of duct.

Further tests on the different configurations of bends with different sizes of duct gave a similar confidence level for the point estimates of the actual rate of airflow. These demonstrated that the conclusions drawn from the results on the 10 in diameter duct were equally applicable to other sizes of duct.

5.3.5 Tests on different configurations of 90° swept bends
It has been stated previously that one of the reasons of using a box bend was because it would act as a 'baffle' to randomize the different degrees of swirling created by various irregularities, such as bends, before the flowmeter. In order to assess this 'baffle' effect, tests were conducted on different configurations of 90° swept bends in a 10 in diameter duct system as shown in Fig. 2 but without box bends. The mean radius of the 90° swept bend was equal to the diameter of the duct.

These tests led to the conclusion the box bend tended to randomize swirling created by any preceding bends, hence the different configurations of bends had little influence on the performance of the flowmeter. In the 90° swept

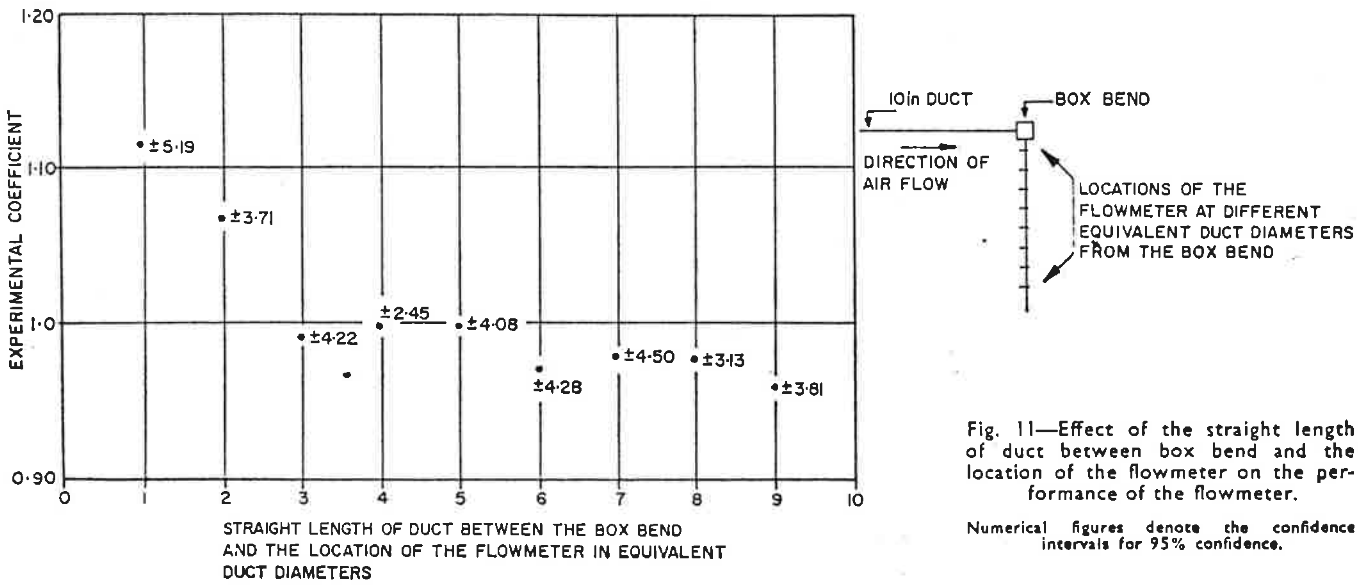


Fig. 11—Effect of the straight length of duct between box bend and the location of the flowmeter on the performance of the flowmeter.

Numerical figures denote the confidence intervals for 95% confidence.

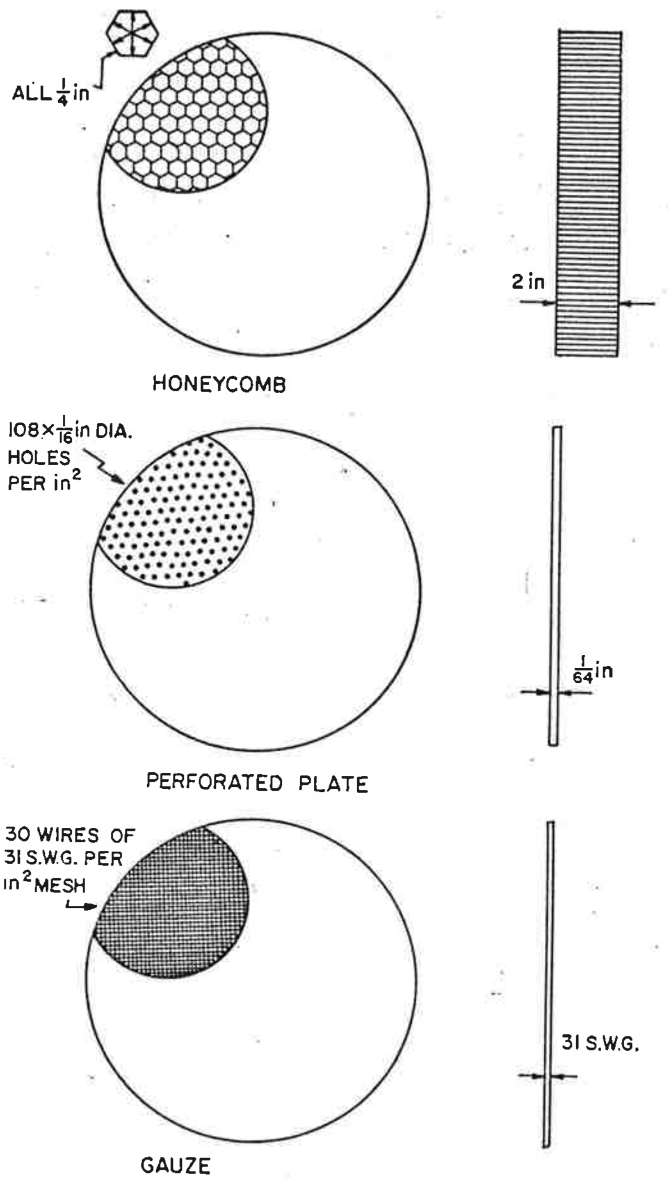


Fig. 12—Air straighteners.

bend system, the swirling was perpetuated and this caused the flowmeter to be less accurate.

5.3.6 Application of the test results

It has been shown in the previous sections that the test results on the combination of all the different configurations of box bends in a 10 in diameter duct system could be applied universally to the different configurations of bends in the different sizes of duct and that the confidence interval of the estimate of the actual rate of airflow was less than ±6.56% over the range of 600 to 1400 ft/min air velocities. This confidence interval seemed, with due consideration of the possible use of air straighteners, to be the best which could be achieved with the flowmeter located at four equivalent diameters downstream from a box bend. It was therefore decided to use these test results for the field trials.

Fig. 13 has been prepared in order to simplify use of the flowmeter. The indicated air velocity as measured by the flowmeter can be entered on the V co-ordinate on the graph, the experimental coefficient can be read from the H_1 co-ordinate and the appropriate percentage error (or confidence interval) can be read from the H_2 co-ordinate according to the percentage confidence limits selected. The indicated rate of airflow can be obtained from a prepared conversion table shown in Appendix I. The estimate of the actual rate of airflow is then

$$Q_a = CQ_i \pm \text{percentage error}$$

where C — the estimate of the experimental coefficient at the indicated air velocity.
Q_a — the estimate of the actual rate of air flow,
Q_i — the indicated rate of air flow.

The accuracy of this graphical method depends on the accuracy of reading of the graph. For Fig. 13, one small division on the V co-ordinate represents five fpm, the maximum error in reading the estimate of the experimental coefficient from the H_1 co-ordinate is about 0.1% and in reading the percentage error of the estimate from H_2 co-ordinate is about 0.2%. These do not seem to be large enough amounts to be considered.

If it is desired to use a single value of the experimental coefficient, irrespective of air velocity, this should be 1.016. If this figure is used then, over the whole range of air velocities tested, the percentage errors would be ±4.8% at 90% confidence limits; ±5.8% at 95% confidence limits and ±7.0% at the 98% confidence limits.

If further simplification, with less accuracy, is desired for the whole range of air velocities tested, it may be taken to be unity. If this is the case, the percentage errors would be ±6.4% at 90% confidence limits, ±7.4% at 95% confidence limits and ±8.4% at 98% confidence limits.

The choice of the confidence limits will depend on the degree of accuracy desired. For measurements in the field, 95% confidence limits would seem to be a reasonable choice.

In the application of the test results as shown above two corrections may be found to be necessary, for changes in the density of the air and for the accuracy of the manometer used.

As indicated previously, all the test results were corrected to a standard condition. In the application of the results, the indicated air velocity and the rate of airflow have to be corrected to the standard condition. For a condition where the barometric pressure does not exceed one half of an inch of mercury gauge from 29.92 in of mercury gauge, the hygrometric data tabulated in the 1959 edition of the *Guide* of the Institution of Heating and Ventilating Engineers (34)* can be used for the evaluation of the density of air at temperatures from 30°F to 136°F dry bulb. The error due to this degree of barometric pressure variation is about 0.8% in the evaluation of the density of air. Correspondingly the error in the calculation of the rate of airflow is about 0.4%. All the test results were obtained using a manometer of an accuracy of 0.001 in of water gauge. The percentage error should therefore be modified, using Fig. 14 in the following manner, if the accuracy of manometer used differs from 0.001 in of water gauge.

$$\varepsilon_m = \sqrt{\varepsilon^2 + \varepsilon_1^2}$$

* Similar tables appear in the 1965 edition of the *Guide* which has been published since this paper was written.

where ε_m — modified percentage error of the estimate,

ε — percentage error of the estimate based on the test results (Figure 13).

ε_1 — percentage error due to the manometer used, which is the difference between the percentage error of the manometer used and that of a manometer of an accuracy of 0·001 inch of water gauge.

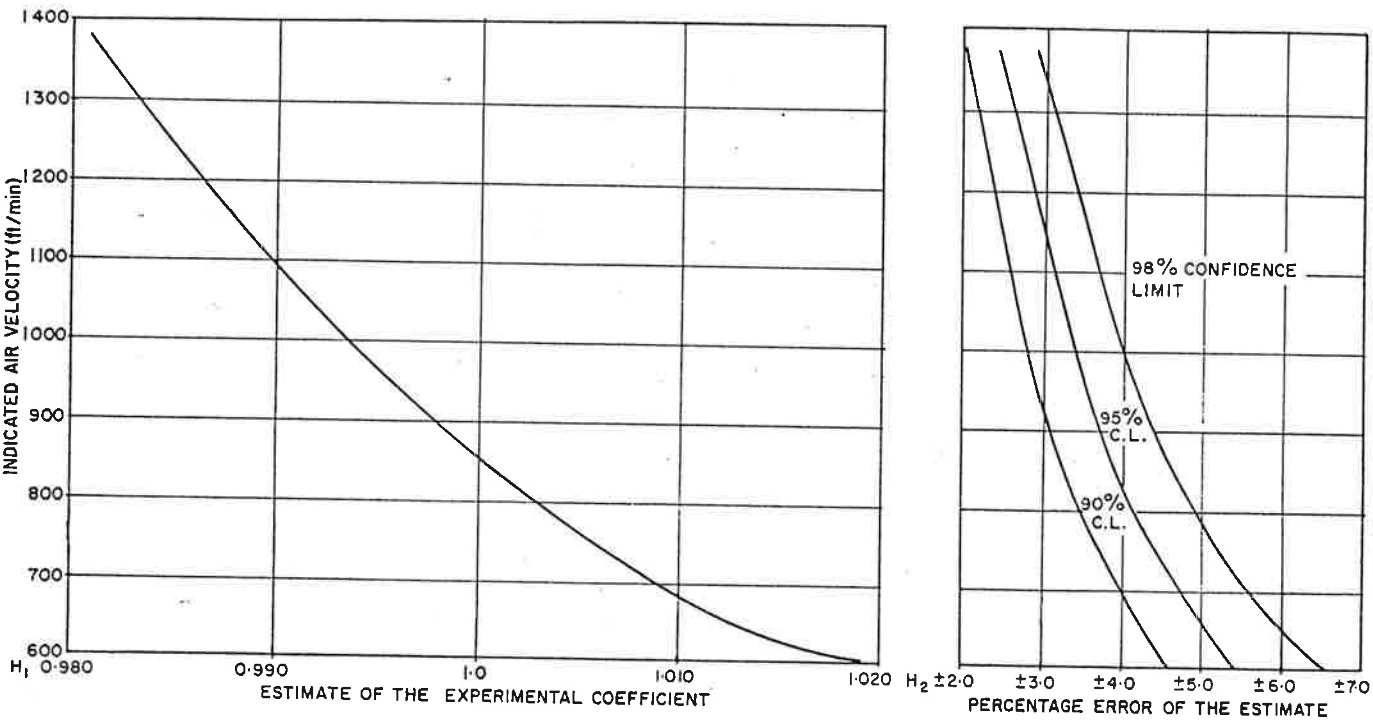
For an example, when a manometer of an accuracy of 0.002 in of water gauge is used instead of a manometer of an accuracy of 0.001 in of water gauge, the percentage error of the estimate of the actual rate of airflow at 600 fpm indicated velocity at 95% confidence limits would be

$$\varepsilon_m = \sqrt{5{\cdot}48^2 + (2{\cdot}35 - 1{\cdot}20)^2}$$
$$= \pm\, 5{\cdot}63$$

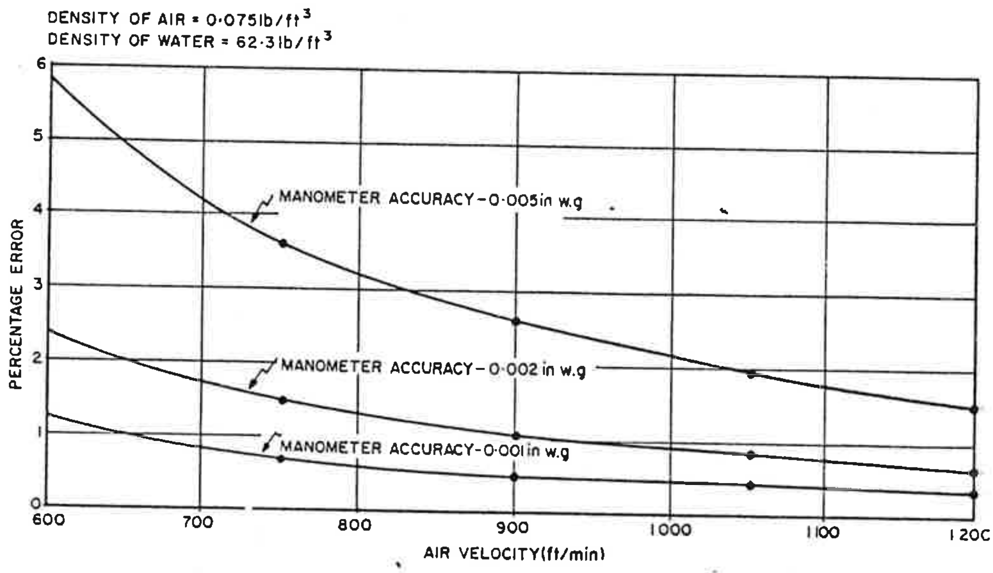
instead of $\pm$ 5·48.

5.3.7 Field check

Field checking of the flowmeter was conducted at the Hairmyres Experimental Ward Unit. Four separate sets of readings were taken for different sizes of duct—6, 8, 10 and 14 in diameter—as shown in Fig. 15. Fig. 16 shows a typical field installation of the flowmeter.



Above: Fig. 13—The experimental coefficient and percentage errors.



Right: Fig. 14—Error in the measurement of air velocity using different manometers.

The rate of airflow as measured by a pitot-static tube traverse was used as a standard to check the accuracy of the flowmeters. Two traverses in accordance with the ten-point log-linear rule were carried out along the diameter of the duct perpendicular to each other. Owing to the space limitation on site, it was convenient to measure the rate of airflow at four equivalent diameters downstream from the box bends where the flowmeter was located.

However, the performance of the method of pitot-static traverse at this location was uncertain. It was therefore decided to check this in the laboratory. These checks showed that the rate of airflow as measured by the pitot-static tube traverses should be decreased by 2.15% and the probable error would be $\pm 1.70\%$ at 95% confidence limits.

The readings from the flowmeter were taken before and after the traverse. In all the cases tested, these two readings showed no difference. The indicated air velocity was calculated with the aid of the conversion table shown in Appendix I. Fig. 13 was used to find the experimental coefficient. The estimate of the actual rate of airflow was calculated.

Since both of these measurements, i.e. by the flowmeter and by the pitot-static tube traverse, involved an error term, a check was made to find out whether there was any significant difference between the readings from these measurements.

Fig. 15—Layout of four flowmeters installed at the Hairmyres Experimental Ward Unit on which the field checkings were conducted.

M.S. = Measuring station where the pitot-tube traversings were carried out and where the flowmeters were located.
E.D. = Equivalent duct diameter.

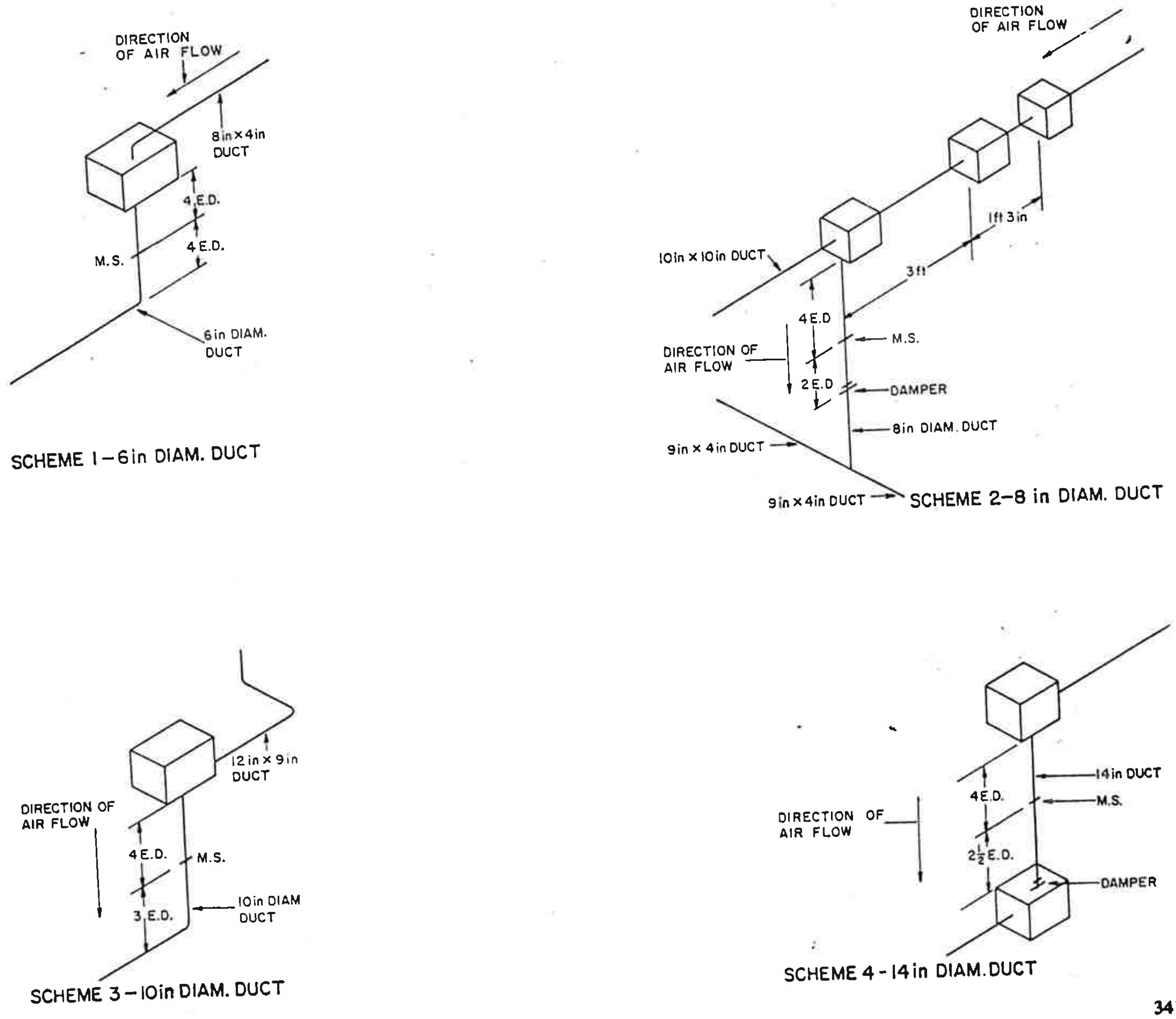


Table IV.—Results of field checking on the flowmeter

Scheme No.	Duct size in inch diameter	Indicated air velocity as measured by the flow-meter	Per-centage variation of the rate of air flow measured by the flow-meter in compari-son with those measured by the pitot-tube traverses	Per-centage error of the estimate of the actual rate of air flow based on the test results in the labora-tory at 95% con-fidence limits (Fig. 13)	Critical values of the per-centage variation at 95% confi-dence limits
1	6	760	+2·01	±4·40	±4·71
2	8	680	+3·52	±4·85	±5·14
3	10	825	+0·39	±4·05	±4·39
4	14	680	+2·50	±4·85	±5·14

Table IV lists the results which show that the percentage variations of the rate of airflow measured by the flow-meter in comparison with those measured by the pitot-static tube traverses are within the critical values at 95% confidence limits. In other words, the field measurements were in agreement with those established in the laboratory.

PART 2. THE BALANCING OF AIR FLOW CIRCUITS IN VENTILATING SYSTEMS

6. Basic principle of balancing of air circuits

The principle of balancing of airflow circuits is based on the phenomenon that the ratio of the rates of airflow delivered through each branch of duct in relation to each other remains constant when the total rate of air supply to them varies. This phenomenon can be shown as follows.

The pressure loss (or the static pressure difference) through a straight length of duct in a turbulent flow can be expressed as

$$h_{l1} = f \frac{l\, v^2}{d\, 2g} \qquad \text{Eq. (4)}$$

where

h_{l1} — pressure loss in length l,
l — length of the duct,
d — diameter of the duct,
v — air velocity,
g — gravitational acceleration,
f — friction factor.

The friction factor varies with the Reynolds number and the roughness factor. However, at Reynolds numbers above a certain critical level, the friction factors of rough ducts become constant, dependent wholly on the roughness of the duct and thus independent of the Reynolds number. This critical Reynolds number [35][36] is about 10^4.

For a 6 in diameter duct and a mean air velocity of

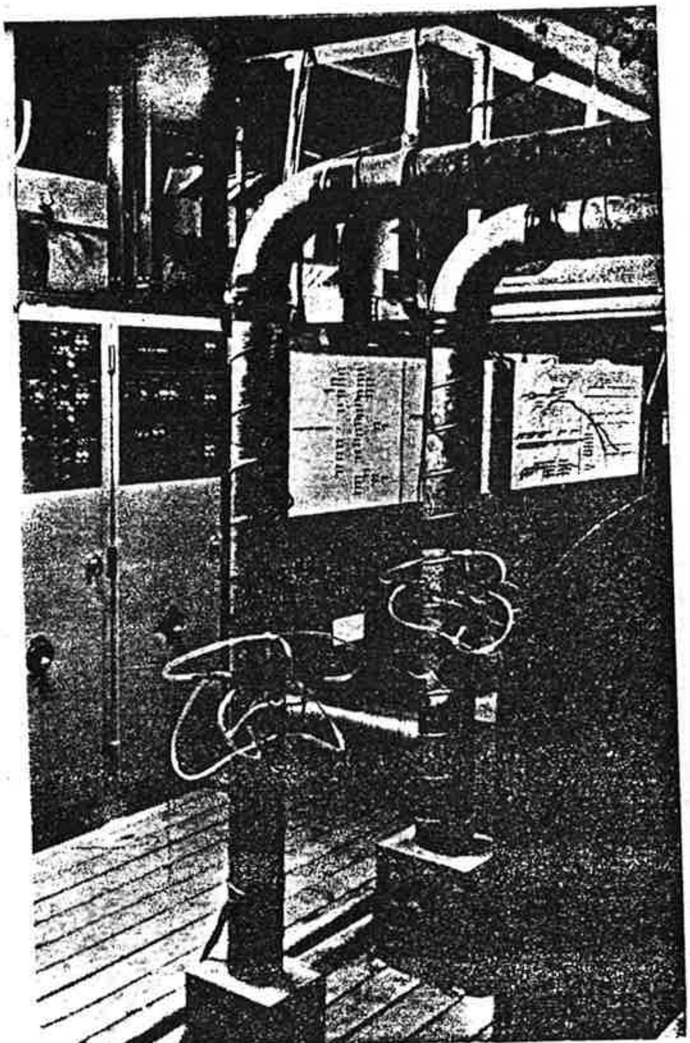


Fig. 16—Field installation of two flowmeters at the Hairmyres Experimental Ward Unit.

600 ft/min the Reynolds number at an ambient temperature of 70°F and a barometric pressure of 29.92 in of mercury gauge is calculated to be 2.97×10^4. Hence for low pressure ventilating work, the friction factor can be treated as a constant.

Eq. (4) becomes

$$h_{l1} = C \frac{l\, v^2}{d\, 2g} \qquad \text{Eq. (5)}$$

where C — a constant.

For bends and dampers, the pressure loss can be expressed in terms of velocity pressure as

$$h_{l2} = k \left(\frac{v^2}{2g}\right) \qquad \text{Eq. (6)}$$

where

h_{l2} — pressure loss across bends or dampers,
k — a resistance factor.

Fig. 17 shows the relationship between the resistance factor and the air velocity for a box bend and for a partially closed iris damper (the iris damper is described in the next section). This graph has been drawn from the results of a separate test.

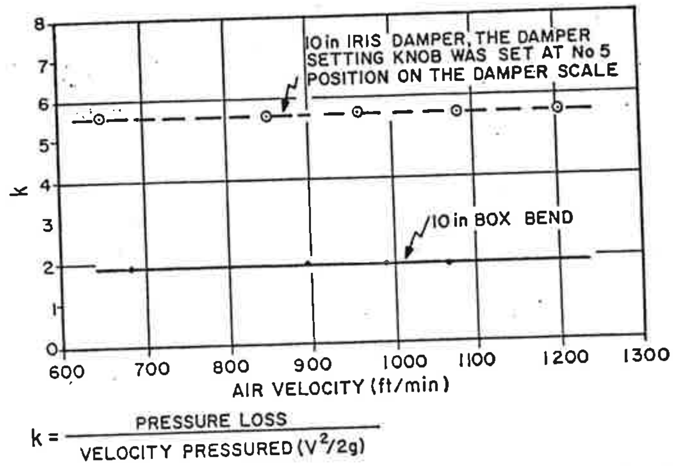


Fig. 17—Pressure losses across a box bend and a partially closed iris damper

From this graph it can be seen that the resistance factor for both the box bend and the iris damper remains substantially constant over the range of air velocities from 600 to 1 200 fpm.

From Eqs. (5) and (6) it can be seen that the total pressure loss (h_L) through an airflow circuit consisting of box bend, iris damper and connecting duct can be expressed as

$$h_L = h_{l1} + h_{l2}$$

$$= C\frac{l\,v^2}{d\,2\,g} + k\left(\frac{v^2}{2\,g}\right)$$

or $$h_L = K(v)^2 \qquad \text{Eq. (7)}$$

where

$$K = \frac{Cl + dk}{2dg} = \text{a constant.}$$

Since $Q = vA$

where

Q — rate of airflow,
A — cross-section area of the duct.

Eq. (7) can be expressed as

$$h_L = K(Q)^2$$

Take as an example an airflow circuit of two branches as shown in Fig. 18. The argument can be extended to airflow circuits with multiple branches if required.

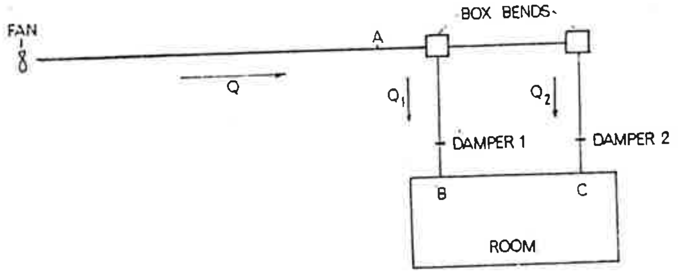


Fig. 18—A two branch air circuit
where Q The rate of total airflow
Q_1 The rate of airflow through branch I
Q_2 The rate of airflow through branch II

The pressure loss (h_{L1}) through Branch I (AB) is equal to the pressure loss (h_{L2}) through Branch II (AC). In other words,

$$h_{L1} = h_{L2}$$

or $$K_1(Q_1)^2 = K_2(Q_2)^2$$

where

K_1 — resistance factor associated with Branch I,
K_2 — resistance factor associated with Branch II,
Q_1 — rate of air flow through Branch I,
Q_2 — rate of air flow through Branch II.

and $$\frac{Q_1}{Q_2} = \sqrt{\frac{K_2}{K_1}} \qquad \text{Eq. (8)}$$

Since the resistance factors do not change with the rate of airflow, the ratio of (Q_1/Q_2) will remain constant when the total rate of airflow (Q) varies.

The above phenomenon indicates that in balancing an airflow circuit it is necessary; firstly to adjust the rate of airflow through branches in proportion to each other in accordance with the respective design rates of airflow; secondly to adjust the total rate of airflow to the design value. This process will be shown in detail in the section on the Procedure for Balancing.

7. Calibration of dampers

7.1 General consideration

From the general procedure for the balancing of an airflow circuit outlined in the previous section, it follows that it would simplify the procedure considerably if the volume control damper on each branch duct had been calibrated so that the relationship between damper opening and the rate of airflow was known.

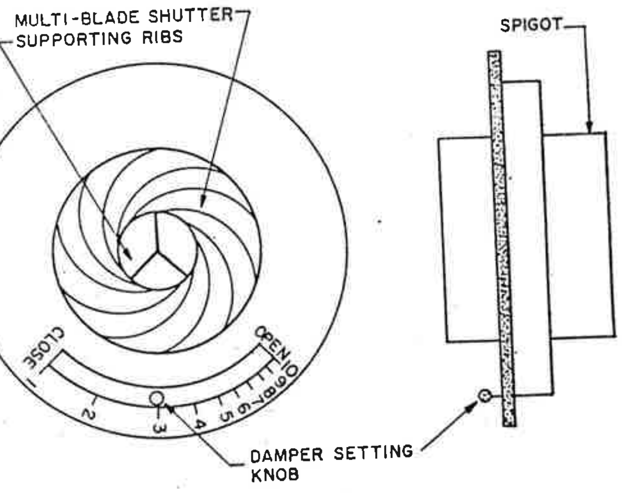


Fig. 19—Iris damper.

This relationship is influenced by the resistance factors of the connecting ducts. For an example, take the air circuit with two branches as shown in Fig. 18. If damper 1 is partially closed then an additional pressure loss is imposed on branch I, which is represented by a resistance factor (K_3) Eq. 8 becomes

$$\frac{Q_1}{Q_2} = \sqrt{\frac{K_2}{K_1 + K_3}}$$

This indicates that the change of the ratio of the rates of airflow through these two branches by adjusting the damper settings would depend on the resistance factors of the branches as well as the resistance factor of the damper when partially closed.

A suitable calibration would relate the movement of the damper to the ratio of the rate of airflow at each intermediate position to the rate of airflow at the fully open position. This calibration would provide a means by which the degree of damper opening could be set in accordance with the required adjustments of the rates of airflow.

Iris dampers as shown in Fig. 19 were used.

Iris dampers operate in a similar manner to the variable aperture in a camera. The degree of opening can be regulated by moving the damper setting knob along a scale. The scale was marked in ten divisions between the 'close' position to the 'open' position on a logarithmic scale.

From the theory of orifices (which the iris damper resembles) in an imcompressible flow, the flow equation between the duct and the constriction is [37]

$$Q = \frac{C_v\, C_c\, A}{\sqrt{1 - C_c^2 \left(\frac{A}{A_1}\right)^2}} \sqrt{2g \frac{\Delta P}{w}} \qquad \text{EQ. (9)}$$

where

Q — rate of air flow,
C_v — coefficient of velocity,
C_c — coefficient of constriction
$= A_2/A_1$
A — area of orifice,
A_2 — area of vena contracta,
A_1 — area of duct,
g — gravitational acceleration,
Δp — pressure difference across orifice,
w — density of air.

If a coefficient of discharge (C), is introduced where

$$C = \frac{C_v\, C_c\, A}{\sqrt{1 - C_c^2 \left(\frac{A}{A_1}\right)^2}}$$

then Eq. (9) becomes

$$Q = C \sqrt{2g \frac{\Delta P}{w}}$$

The value of the coefficient of discharge depends on the following factors:

(a) Air velocity in the duct.
(b) The ratio between the area of the duct and the area of the damper opening.
(c) The construction of the damper opening.
(d) The roughness factor of the duct and,
(e) The straight length of duct before and after the damper opening.

7.2 Laboratory layout

In order to minimize the effects of the resistance factors of different branches of duct as found in the field, it seemed logical to calibrate not only the damper but the assembly of damper, box bend and the flowmeter as described in Part 1 of this paper. The flowmeter should be located at four equivalent diameters downstream from a box bend and obstructions, such as dampers, should be located at two equivalent diameters downstream from the flowmeter. For the calibration of dampers, it was therefore decided to install the damper at two equivalent diameters downstream from the flowmeter.

In this calibration of dampers, the effects of the factors b, c, d, e, as listed in the preceding section which influence the coefficient of discharge of the opening of the damper were included as constants. However, the effect of air velocity in the duct, the effect of different dampers and the effect of different sizes of duct on the calibration had to be investigated.

Fig. 20 shows the laboratory layout for the investigation of these effects and for the calibration of dampers.

Instrumentation was similar to that used for the tests on the flowmeter as described in Part 1.

7.3 Investigation of the effect of the different factors

(a) *Effect of air velocity*

Tests were conducted on a 10 in iris damper. The air velocity with the damper fully open was measured, using the venturi meter. The damper was then partially closed to setting number 5 and the air velocity measured again. Four pairs of these measurements were taken and the average used to calculate the ratio of the rates of airflow for the damper fully open and partially closed. Using the air volume control box (Fig. 20) the air velocity with the damper fully open was then altered and the process repeated to give four ratios of the rates of airflow.

The results showed a slight variation (about 2%) between the ratios of the rates of airflow at the different air velocities with the damper fully open. It was concluded that air velocity had very little effect on the calibration of dampers over the range of air velocities of 600 to 1 200ft/min.

(b) *Effect of different dampers*

Tests were conducted on four different 10 in diameter dampers. The dampers were tested separately at an air velocity of 1 000ft/min as measured by the venturi meter when the dampers were fully open. The rates of airflow were measured both when the damper was fully open and when the damper setting knob was set at No. 5 on the

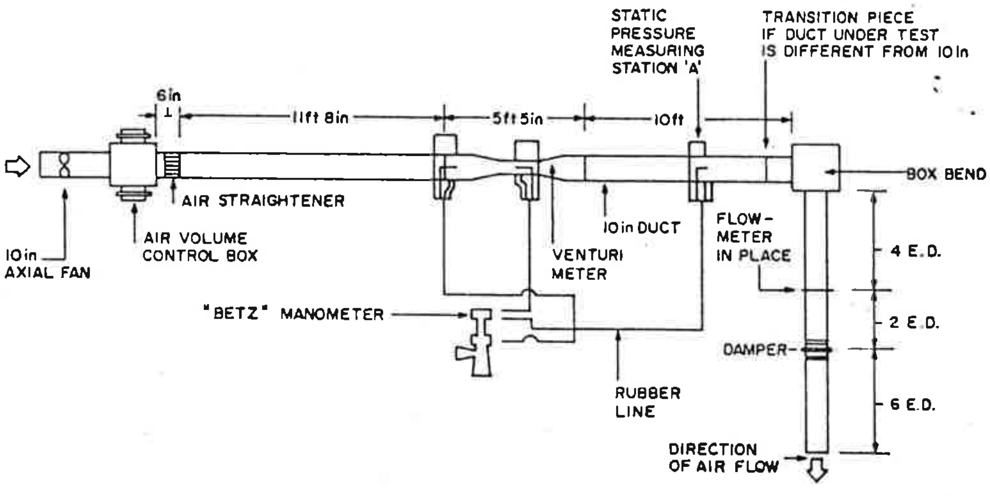


Fig. 20—Laboratory layout for the calibration of dampers.

scale of the damper. For each damper, four sets of readings were taken and the mean of them was used for the calculation of the ratio between the rates of airflow.

The results indicated that the maximum difference between the ratios associated with different dampers is about 2½% in comparison with the mean of the readings. This amount of difference might be due to errors in the marking of the scale, in setting the knob to the exact position and the small amount of backlash in the mechanism which controlled the openings of the dampers.

(c) *Effect of different sizes of duct*

Tests were conducted on 6, 10, 12 and 14 in diameter dampers in their appropriate sizes of duct. An air velocity of 1 000 ft/min in the duct when the damper was fully open was used. The rates of airflow were measured by the venturi meter, both when the damper was fully open and when the damper setting knob was set at No. 5 position on the scale of the damper. For each size of duct, four sets of readings were taken and the mean of these was used for the calculation of the ratio between the rates of airflow.

The results showed that the maximum difference between the ratios associated with different sizes of damper is about 3%, in comparison with the mean of the readings. This might be due to the same causes as quoted in the preceding sub-section.

From this evidence, it seemed to be feasible to use one calibration curve for the different sizes of damper within the range of air velocities of 600 to 1 400 ft/min for field use, as the effect of air velocity, the effect of different dampers and the effect of different sizes of damper were small. A further reason for the use of one calibration curve for the different sizes of damper was the effect of the different resistance factors of the connecting ducts indicated previously. In the field, it is impracticable to have the same resistance factor for all branches of duct, hence it would suffice to provide a calibration curve which is only sufficiently accurate to give a first approximation in the process of balancing. The final adjustment would depend on the readings from the flowmeter after the first adjustment. This is illustrated more clearly from the results in both the laboratory tests and the field tests on the balancing in the last two sections.

7.4 Calibration of the dampers

The following was the procedure for the calibration of dampers.

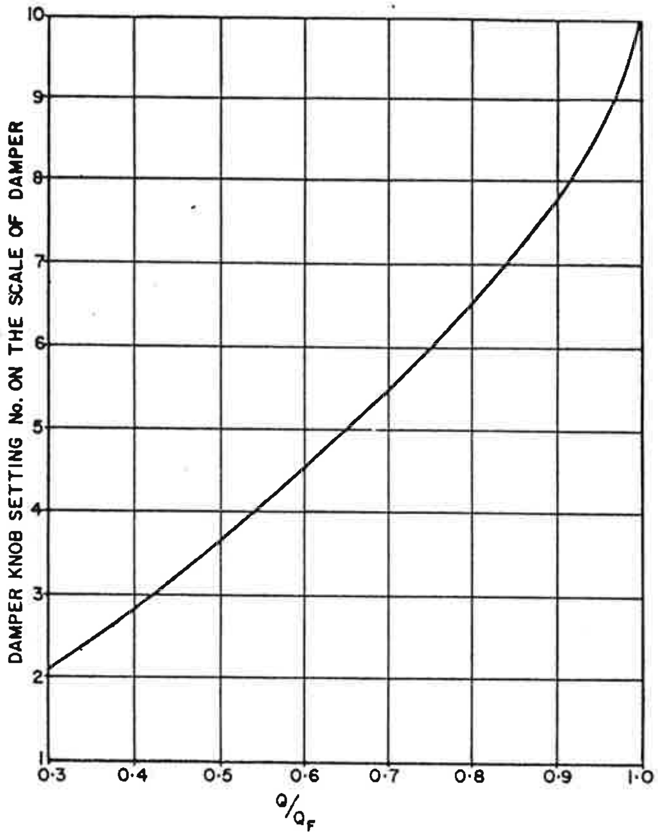
(a) The damper was set at its fully open position and the rate of airflow was adjusted by the airflow control box to give an appropriate rate of airflow between the air velocities of 600 to 1 400 ft/min. The rate of airflow as measured by the venturi meter was recorded. The static pressure at Pt. A (Fig. 20) was measured and recorded.

(b) The damper setting knob was then moved one division at a time on the scale of the damper from its 'open' position to its 'close' position. For each setting, the airflow control box was adjusted to bring the static pressure at Pt. A back to the same pressure as recorded initially (to avoid interference from the fan characteristic). The rate of airflow as measured by the venturi meter was then recorded.

(c) The ratio of the rates of airflow was calculated and plotted against the number of the damper setting.

Five dampers of each of the different sizes (6, 8, 10, 12 and 14 in diameter) were calibrated within the range of air velocities of 600 to 1 400 ft/min with the damper fully open.

Fig. 21 shows the combined calibration curve for all the 25 sets of tests.



Q = THE RATE OF AIR FLOW WHEN THE DAMPER WAS FULLY OPEN

Q_F = THE RATE OF AIR FLOW WHEN THE DAMPER SETTING KNOB WAS SET AT THE VARIOUS POSITIONS ON THE SCALE OF DAMPER

Fig. 21—Calibration curve for iris dampers.

8. Procedure for balancing

1. Measure the rate of total airflow delivered by the fan.
2. Adjust the main balancing damper or the fan to give approximately the design rate of airflow, say, $\pm 10\%$.
3. Measure the rate of airflow through each branch.
4. Calculate the ratio between the design rate of airflow and the actual rate of airflow, e.g.
$$\frac{\text{Actual rate of airflow}}{\text{Design rate of airflow}}$$ for all branches.
5. Select the lowest ratio.
6. Divide the lowest ratio by the other ratios individually. (This gives the proportion by which the rate of airflow must be reduced.)
7. Enter the ratios calculated in 6 individually into the calibration curve as shown in Fig. 21. Read from the curve the damper knob setting required.
8. Adjust the damper knob settings in accordance with the figure given in 7.
9. Measure the rate of airflow through each branch, and calculate the ratio of actual to design rates of airflow as in 3 and 4.

10. Divide the ratio, calculated in 9, of the branch selected in 5 by each of the other branch ratios, calculated in 9, in turn.
11. Multiply the new ratios from 10 by the respective figures of the damper knob settings obtained in 8.
12. Adjust the damper knob settings in accordance with the figures given in 11.
13. Check the rate of airflow after the second adjustment as from 3 and 4. If the ratios for all branches do not differ by more than the percentage error of the Flowmeter, proceed to 14. If not, repeat 10 to 13 and repeat if necessary.
14. Adjust the main balancing damper or the fan to the design rate of airflow. (All branches should be now in balance, that is to say they deliver the rates of airflow required by the design.)
15. Check the rate of airflow through each branch.

Note: Where a main trunk from the fan supplies a number of sub-trunks and each sub-trunk in turn supplies a number of branches, the branches supplied by individual sub-trunks can be proportionally balanced as 1 to 13 then the sub-trunks can be treated as branches and balanced as 1 to 15.

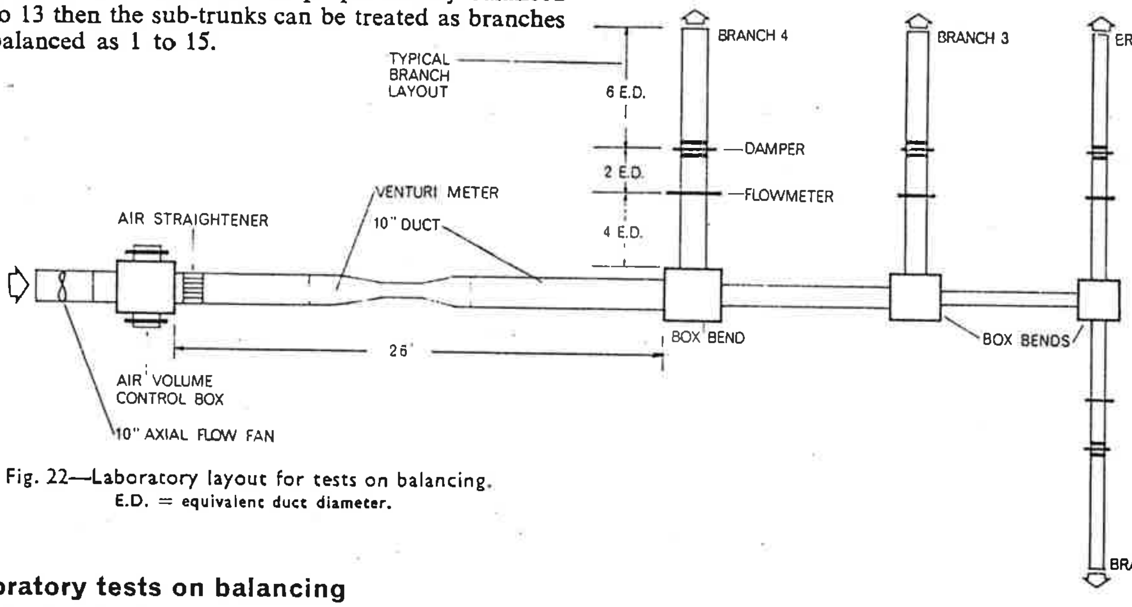


Fig. 22—Laboratory layout for tests on balancing.
E.D. = equivalent duct diameter.

9. Laboratory tests on balancing

An air circuit with four branches was used, as shown in Fig. 22, for the testing of the balancing procedure. The design rates of airflow assigned to the branches for four different schemes as shown in Table V.

Schemes 1, 2 and 3 were balanced with the aid of the calibration curve (Fig. 21). Scheme 4 was balanced by a 'Systematic Trial and Error' method without the aid of the calibration curve. In other words, it was balanced by comparing one pair of branches at a time, e.g. Branch 1 was firstly proportionally balanced with Branch 2, then Branch 2 was balanced with Branch 3 and so on.

Table VI shows a typical set of results of the balancing with the aid of calibration curve (Scheme 2). From the table it may be noted that only two adjustments are required to bring the airflow circuit into proportional balance. This applies to Schemes 1 and 3 as well. In all schemes the final readings were within $\pm 4\%$, which is the accuracy of the flowmeter.

It was noted that the time required to balance Scheme 4 using the 'Systematic Trial and Error' method was roughly four times of that required for Schemes 1, 2 or 3.

Table V—Schemes for the laboratory tests on balancing.

Scheme No.	Branch No.	Size of duct in inch	Design rate of airflow in ft^3/min
1	1	6	200
	2	6	200
	3	6	200
	4	6	200
2	1	6	200
	2	6	150
	3	6	200
	4	6	150
3	1	6	150
	2	6	200
	3	8	250
	4	6	150
4	1	6	140
	2	6	180
	3	8	230
	4	6	120

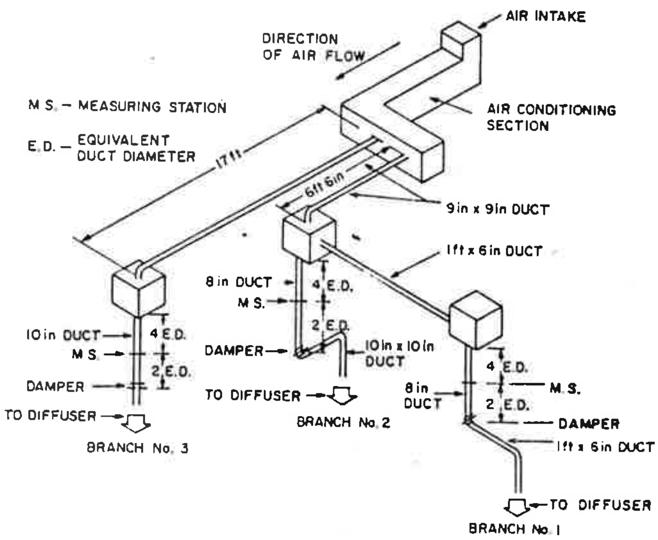


Fig. 23—An air circuit of three branches at the Hairmyres Experimental Ward Unit.

Table VI—Balancing sheet for Scheme 2

Branch No.	Size of duct in inches diameter	Design		Initial			1st adjustment				2nd adjustment			Final adjustment	
		ft³/min	ft/min	ft³/min	R_1	R_2	Damper knob setting No.	ft³/min	R_3	R_4	Damper knob setting No.	ft³/min	R_5	ft³/min	R_6
1	6	200	1020	134	·67	1·00	10	177	0·89	1·00	10	170	·85	197	·98
2	6	150	760	133	·89	0·75	5·9	127	0·85	1·05	6·2	131	·87	151	1·01
3	6	200	1020	202	1·01	0·66	5·1	178	0·89	1·00	5·1	173	·87	198	·99
4	6	150	760	228	1·52	0·44	3·1	128	0·85	1·05	3·2	133	·89	152	1·01

$R_1 = \frac{\text{Actual rate of airflow}}{\text{Design rate of airflow}}$ $R_3 = \frac{\text{Actual rate of airflow}}{\text{Design rate of airflow}}$ $R_5 = \frac{\text{Actual rate of airflow}}{\text{Design rate of airflow}}$

$R_2 = \frac{\text{Lowest } R_1}{R_1}$ $R_4 = \frac{R_3 \text{ of branch which had lowest } R_3}{R_3}$ $R_6 = \frac{\text{Actual rate of airflow}}{\text{Design rate of airflow}}$

For airflow circuits of more than four branches, the difference in term of time required would probably be increased exponentially.

10. Field tests on balancing

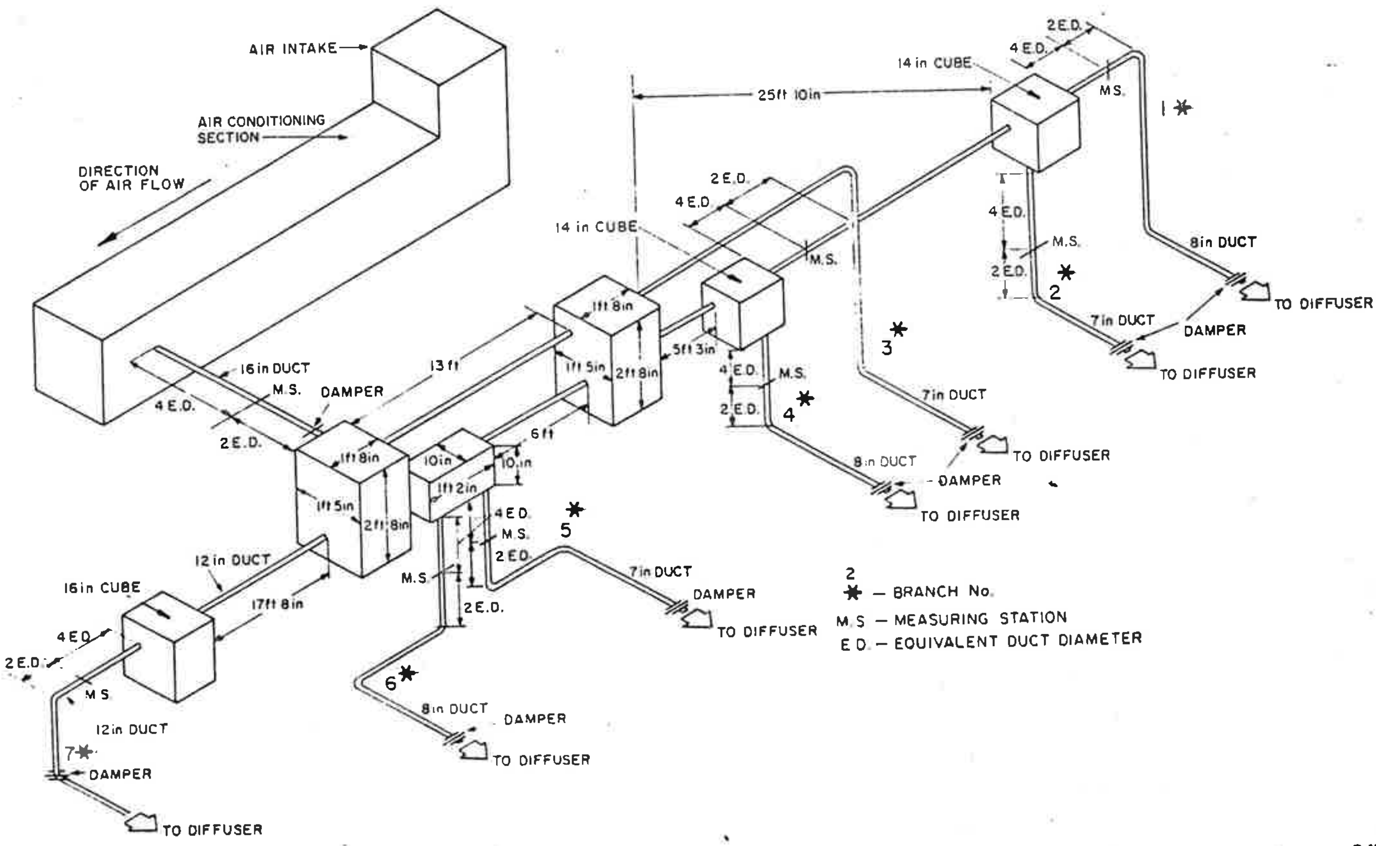
Figs. 23 and 24 shows two actual installations of air circuit at the Hairmyres Experimental Ward Unit. Figs. 25 and 26 show results of the balancing for these two air circuits. From the graphs, it can be seen that, as in the laboratory tests, only two adjustments were required to bring the air circuit to balance and the accuracy for the balancing was within $\pm 4\%$.

11. Acknowledgements

I would like to express my thanks to the Nuffield Provincial Hospitals Trust and the University of Glasgow for enabling me to carry out the study described. Opinions put forward in this paper are not necessarily those of the Trust or of the University.

It is a pleasant duty to acknowledge my indebtedness to my supervisor, my colleagues at the Hospital Engineering Research Unit and those who have given their assistance most generously; to Professor J. Small, D.Sc., Ph.D.,

Fig. 24—An air circuit of seven branches at the Hairmyres Experimental Ward Unit.



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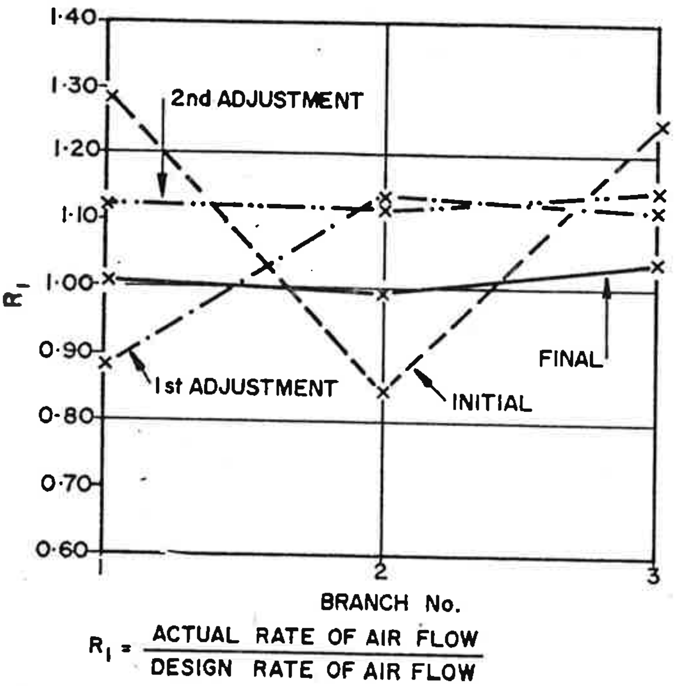

Fig. 25—Results of field test on balancing, layout as shown in Fig. 23.

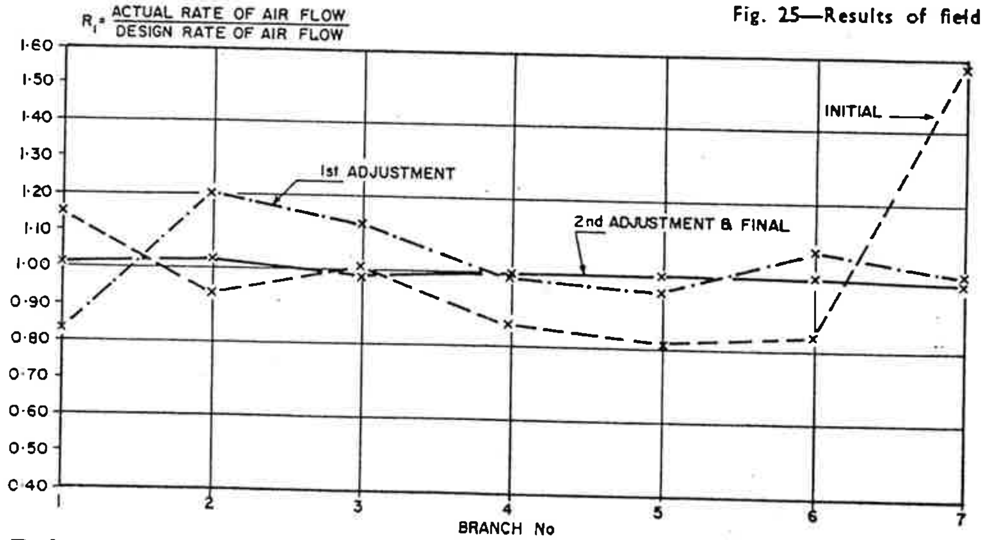

Fig. 26—Results of field test on balancing, layout as shown in Fig. 24.

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Appendix 1.

A Conversion Table. Velocity Pressure and Duct Size to the Rate of Airflow for air having a density of 0.075 lb/ft^3

Velocity Pressure		Duct size in inches diameter											
		4	5	6	7	8	9	10	12	14	16	18	24
Inches of water gauge	Millimetres of water gauge	Rate of airflow ft^3/min											
·002	0·05	16	24	35	48	62	79	97	140	190	248	314	559
·004	0·10	22	34	49	67	88	111	137	197	269	351	444	789
·006	0·15	27	42	61	82	108	136	168	242	329	430	544	968
·008	0·20	31	49	70	95	124	157	194	279	380	497	629	1117
·010	0·25	35	54	78	106	139	176	217	312	425	556	703	1250
·012	0·30	39	60	87	118	154	195	241	347	472	617	781	1388
·014	0·35	42	65	94	128	167	211	261	376	512	668	846	1503
·016	0·40	45	70	100	136	178	225	278	400	545	712	901	1601
·018	0·45	47	74	105	144	188	237	293	422	574	750	949	1688
·020	0·50	49	77	111	151	196	249	307	442	602	786	995	1768
·022	0·55	52	81	116	158	206	261	322	464	631	824	1043	1855
·024	0·60	54	84	121	165	215	272	336	484	659	860	1089	1935
·026	0·65	56	88	126	172	224	284	350	504	686	896	1134	2016
·028	0·70	58	91	131	179	232	294	363	523	711	929	1176	2091
·030	0·75	60	94	135	184	241	305	376	541	737	963	1218	2166
·031	0·80	62	97	140	191	249	315	389	560	762	996	1260	2241
·033	0·85	64	100	144	196	256	324	400	576	784	1024	1296	2304
·035	0·90	66	103	148	202	264	334	412	593	808	1055	1335	2373
·037	0·95	68	106	152	207	271	343	423	609	829	1083	1371	2436
·039	1·00	69	109	156	213	278	352	434	625	851	1111	1406	2500
·041	1·05	71	111	160	218	285	360	445	641	872	1139	1442	2563
·043	1·10	73	114	164	223	291	369	455	655	892	1165	1474	2621
·045	1·15	74	116	167	228	298	377	465	670	911	1190	1507	2678
·047	1·20	76	119	171	233	304	385	475	684	931	1216	1539	2736
·049	1·25	78	121	175	238	310	393	485	698	951	1242	1571	2794
·051	1·30	79	124	178	242	316	400	494	711	968	1265	1601	2845
·053	1·35	81	126	181	247	323	408	504	726	988	1290	1633	2903
·055	1·40	82	128	185	251	328	416	513	739	1005	1313	1662	2955
·057	1·45	84	131	188	256	334	423	522	752	1023	1336	1691	3007
·059	1·50	85	133	191	260	340	430	531	765	1041	1359	1720	2059
·061	1·55	86	135	194	265	346	437	540	778	1058	1382	1750	3110
·063	1·60	88	137	198	269	351	447	549	791	1076	1405	1779	3162
·065	1·65	89	139	201	273	356	458	557	802	1092	1425	1805	3208
·067	1·70	91	141	204	277	362	465	566	815	1109	1449	1834	3260
·069	1·75	92	143	207	281	367	471	574	827	1125	1469	1860	3306
·071	1·80	93	145	210	285	372	468	582	838	1141	1490	1886	3352
·073	1·85	94	147	212	289	378	484	590	850	1156	1510	1912	3398
·075	1·90	96	149	215	293	383	490	598	861	1172	1531	1938	3444
·077	1·95	97	151	218	296	387	497	605	871	1186	1549	1960	3485
·079	2·00	98	153	221	300	392	503	613	883	1201	1569	1987	3531
·081	2·05	99	155	224	304	397	509	621	894	1217	1590	2012	3577
·083	2·10	100	157	226	308	402	515	628	904	1231	1608	2035	3617