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This paper is to be delivered at a Sessional Meeting of the Institution on Thursday, 11th November. The meeting will take place at the National College, Skipton Street, Elephant & Castle, London, S.E.1, at 5.30 p.m. for 6 p.m. All members are entitled to attend, to take part in the discussion and to put questions to the authors. The discussion on this paper will be printed in a later edition of the *Journal.*

BALANCING AIR FLOW IN VENTILATING DUCT SYSTEMS

*by E. Harrison, A.M.I.H.V.E., M.A.S.H.R.A.E., and N. C. Gibbard, M.A. (Cantab), M.I.H.V.E.**

Synopsis

The paper is concerned with the procedure for regulating or balancing a ventilation system, so as to ensure that each outlet supplies or extracts its proper quantity.

It deals with methods, organisation, instrumentation and recording of results; it also discusses the accuracy required and attainable in ordinary practice. The advocated method of regulating consists of balancing the proportions of air flow at the junctions of a system, dealt with in a routine order, followed by the adjustment of the complete system to design volume by a final setting at the fan. The main discussion is related to low-velocity systems, but the extension of the method to high-velocity induction systems is also covered.

The paper deals with aspects of design for air handling plant and air ducting that is required to facilitate the regulation of systems to achieve design performance.

The paper does not deal with the performance of equipment for heating, cooling, filtering or humidifying the air, nor with any aspect of noise measurement and control.

1. The need for an air balancing routine

1.1

Air duct systems have been regulated in the past generally by a variety of trial-and-error methods, satisfactory results often being obtained only when a tester of experience and judgment has the necessary time and determination. Today new buildings are larger and higher and most of them must have mechanical ventilation of some sort, many of them air-conditioning. As an industry we need to raise our standards of efficiency while at the same time we are faced with a shortage of trained and experienced technicians. For all these reasons the need is pressing for a method of balancing that may be performed efficiently by a simple and logical step-by-step routine on jobs of any shape or size.

1.2

The need for balancing an air duct system is even greater than the need to balance a hot-water distribution system. In the first place a hot-water heating system specification usually calls for a stated room temperature to be achieved in given conditions, but leaves water flow rates unspecified. In a hot-water heating system an increase in water flow rate of 50% produces only a 7% change in heat output. In a ventilation system on the other hand the air quantities delivered into or extracted from the room are themselves of prime importance, and sometimes rates of air flow are the criteria by which the performance of the system is evaluated.

Secondly, it is easier to design and install a water piping system for a precise performance than it is to design and install a ventilation ductwork system which will result in known air quantities. This is partly because pipe and fittings for water are standardized and have resistances to water flow which are known with some degree of precision, whereas ductwork is almost all 'tailor made' with various constructional details, shapes and qualities, the effects of which on the resistance are only vaguely known beforehand. It is also because the greater size of ductwork means that planning alterations and unforeseen site conditions will often demand site alterations in the ductwork, which may have a large effect on the relative resistance of parts of the system.

These circumstances mean that a considerable lack of balance often exists in a duct system as installed, and site testing and sometimes drastic dampering is required in order to provide the air quantities desired.

2. The problem

The difficulty about balancing an air duct system is that the alteration of the setting of any one damper in the system alters the rate of flow not only in that branch which the damper controls but also in all the other branches in the system.

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The consequence of this is that, however great the skill and experience of the tester, it is very seldom possible to set any damper so as to achieve the correct absolute values of air flow without several (perhaps very numerous) subsequent corrections.

The procedure of making successive physical changes in a duct system (i.e. damper alterations) in order to produce appropriate corrections in the air flow rate, can be likened to the arithmetical procedure in Southwell's relaxation method of solving mathematical equations. Southwell himself has likened his method to that of hand-scraping a bearing surface by removing successively the high spots until the marking blue uniformly covers the whole. The trouble about applying such a method to ventilation balancing is that the result of any correction cannot be simply and easily revealed by means of some sort of 'marking blue'; nor can it be calculated by simple arithmetic unless accurately calibrated dampers are employed. The result of any change can only be ascertained by measurement, so that a time-consuming process is rendered that much slower and more cumbersome by the need to measure the effect of each change.

The trial-and-error method is perhaps workable with highly skilled testing personnel, but it is appallingly wasteful of time even when operated with skill and judgment.

Fortunately there is a better alternative. This is systematic proportional balancing at each branch so that the ratio (but not necessarily the absolute values) of the two flows is correct. In order to preserve this ratio in spite of all subsequent alterations of other dampers it is necessary to make each junction balance in its proper order. The method is explained in greater detail below.

3. Review of previous literature on the subject

3.1

Somewhat surprisingly, very little has been written on this subject although it is one of some importance to ventilating and air-conditioning engineers.

There is a lot of information in books and papers on the subject of air flow measurement and on the instruments and methods used for determining it in laboratory and field, but hardly anything has been published on the related, but really quite distinct, subject of the organisation and method required for regulating the air quantities.

3.2

In the Institution's *Journal* a paper by Knox[1] discusses the subject in some detail and advocates the use of the pitot tube and inclined gauge. As to method it says no more than that one should start at the fan and work down the system.

3.3

Twenty years later Ramsay[2], in a paper of wider scope, discusses instrumentation in considerable detail from a wide knowledge of instrument practice, but dismisses the subject of method in a paragraph in which he points out that every alteration of a damper alters the flow rate in every part of the system and concludes therefore that there is no relief from the necessity of proceeding by a method of trial and error with successive corrections of damper settings, in which, it is implied, only the skill and experience of the tester avails to reduce the infinite time and labour required.

3.4

An American paper by Dauphinee & Argentieri[3] of 1949 appears to be the first attempt to overcome the difficulty and labour of trial-and-error damper adjustments. The method described depends on having calibrated outlet dampers. In the case of the diffuser outlets used by the authors the calibration is given by the manufacturers as a percentage of wide-open flow at various damper settings. The authors' method comprises (1) a measurement of the flow at each outlet with all dampers wide open; (2) a calculation of the correct damper setting to give the flow at each outlet as a percentage of the flow obtained at the base, or least-favoured outlet; (3) the setting of all dampers to the calculated opening and (4) a final check of the air flow rate at several selected outlets. This method is obviously a very great improvement on the trial-and-error methods. If the calibration of outlet dampers can be relied upon, it eliminates all checking of outlet velocities except for the recommended sample check at the end. The main obstacle to the adoption of such a method is that few calibrated dampers of any kind are available. The authors themselves point out that trial-and-error methods are necessary when balancing main branches, because the main dampers (in contrast to the individual outlet dampers) are not calibrated.

Calibrated dampers are rare in this country. A diffuser manufacturer offers a range of diffusers with built-in calibrated air volume dampers. These are quoted as passing from 100% to 40% of fully open delivery against specified numbers of turns of the adjusting screw. The calibration appears to be reasonably accurate, but the number of turns of the screw becomes inconveniently large with large diffusers. Another maker offers multi-blade main dampers calibrated in resistance coefficients $\left(K=\dfrac{\Delta p}{\left(\dfrac{v}{4005}\right)^2}\right)$ against angular setting of blades (or of operating lever). It is not known to the present authors what variation between individual samples is to be expected. One suspects that manufacturing tolerances and other factors might make the variation rather large. Some support for this view is given by an article by Koch-Emmery[4] where variations of $\pm 50\%$ in the pressure drop through various types and sizes of multileaf damper are quoted. Fig. 1 is reproduced from this article.

One very important factor, affecting the behaviour of any damper, is the occurrence of upstream disturbances which alter the pattern of velocity distribution at the damper. These may make the damper calibration wildly inaccurate, and its performance inconsistent at different flow rates. This question is discussed later at greater length since it is one of the factors which can make difficulties whatever method of balancing is used.

3.5

A proportional balancing method was advocated by Bricker[5] at a Ventilation Symposium at the ASHRAE Semiannual meeting in Chicago, 1961. The description of the method, as reported in the technical press, is not very clear, but it appears to be a method in which attempts are made repeatedly to set absolute values at main branches before a proportional balance has been achieved in the whole system.

3.6

In 1963 the Ventilating and Air Conditioning Contractors' Association of Chicago published a 'Testing and Balancing Manual'[6]. This contains a reasonably clear step-by-step exposition of a proportional balancing method. The method described is unexceptionable, but the authors appear to lose confidence in it at the end and enjoin making 'final adjustments at each outlet so that each is delivering its design

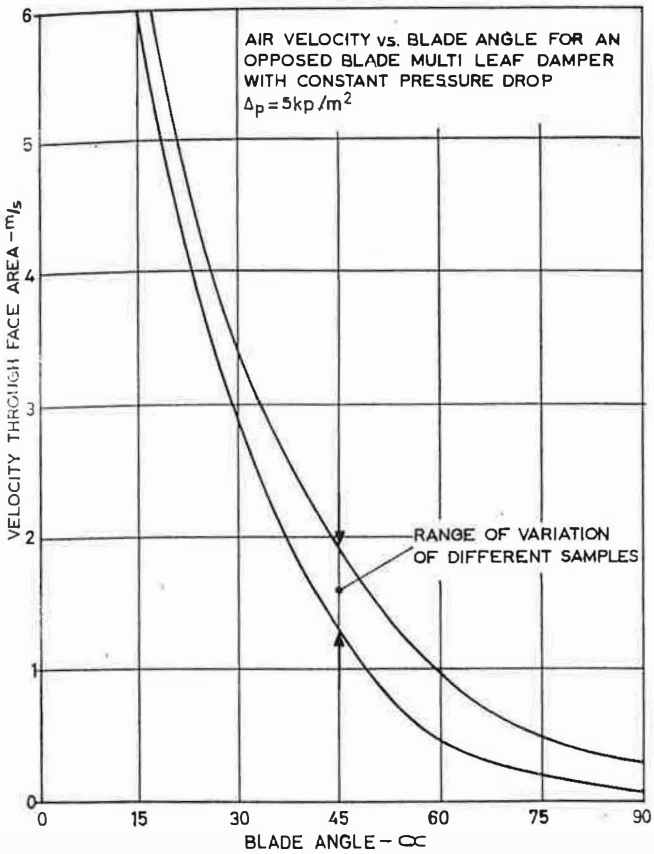


Fig. 1

ft³/min', but, curiously, state that velocity and ft³/min for each outlet should be recorded 'before final adjustments'.

4. The theoretical basis for a proportional method

4.1

The method of proportional balancing depends on the validity of the assumption that at a given branched (tee or Y) fitting the relative flow in each branch will remain unaltered with changes of total flow in the main duct. It is necessary to examine how far this assumption is valid.

4.2

Plainly the validity of the assumption depends on the ratio of the resistances of the two branches remaining constant. Now the total resistance of a normal run of duct usually comprises the frictional resistance to flow of all the straight lengths of duct plus the dynamic resistance due to changes of velocity or direction at various fittings. The dynamic resistance of a fitting is commonly expressed as a fraction or multiple K of the velocity pressure, and K is invariably accepted as remaining constant at least over a wide range of velocities. Hence if the resistance of each branch consisted solely of fittings the ratio of the flow in the two branches would remain constant over a wide range of variation of the total flow.

The case is somewhat different with frictional resistance. As is well known, the friction factor f is not a constant but varies with Reynolds' number. Moreover, the variation is rapid at low Reynolds' numbers and the value becomes practically constant at high Reynolds' numbers. Hence it is easy to see that, if the difference between the Reynolds' numbers in the two branches is considerable, a change in the total flow rate will produce a change in the two Reynolds' numbers which will result in a disproportional change in the respective friction factors, and hence a change in the ratio of one flow to the other.

Appendix A examines the extent of this departure from proportionality. It can be seen that even when there is a very large change in total flow rate the effect is not very significant. This is true even of a system in which all the resistance is frictional, and becomes less and less significant when more of the total resistance is made up of dynamic losses in fittings.

4.3

There is another difficulty which may arise in certain arrangements of ductwork, namely where there is a serious disturbance of the flow pattern immediately upstream of a branched fitting. It is possible in such a case that geometrical similarity of the flow pattern will not be preserved when the total flow rate is altered. Fig. 2a illustrates diagrammatically the condition which can occur at a high rate of flow. At a lower rate of the total flow (Fig. 2b), the disturbed region may not reach the branch and the proportionality of the divided flow rates may not be preserved, i.e. a re-setting of the dampers may be necessary. This is very undesirable whatever method of balancing is employed and should be avoided.

The remedy is either to remove the branch further downstream or, when this is impossible, to use a vaned elbow upstream of the branch (Fig. 2c).

4.4

It is worth pointing out that the maintenance of balance in a system with changes of total flow rate is very important in some cases on quite other grounds than that of making a proportional balancing procedure feasible. For example in systems designed for a larger rate of ventilation in summer than in winter, it would be quite intolerable if the balance were to be seriously upset by the change from winter to summer working and vice versa.

5. Outline of the proportional method

5.1

For the purposes of this discussion we shall consider a supply ventilation system.

We have shown that at a duct junction the *proportion* of air flowing in each branch will remain substantially constant when the total volume of air approaching the junction is varied, provided that no changes are made to the physical disposition of the branches downsteam (e.g. to damper positions or to grilles), or to the pressure in the room to which the ducts discharge. We can make use of this principle to balance a duct system by a proportional method which entails the least number of measurements of air velocity and no resetting of dampers that have once been set.

At any junction dampers may be set to direct a fixed proportion of the total air flow to each branch so long as no alterations at all have to be made downstream. Alterations upstream may be made affecting the total airflow but they will not affect the proportions of air flowing to the branches. In practice this means that the balancing procedure must begin at an end junction remote from the fan where no adjustments have to be made further downstream. This first junction is balanced by proportioning the air flow in each branch in the ratio of the design volumes in the two branches.

The adjacent junction upstream is next tackled and the proportions of flow in its branches are adjusted in the same way. This procedure continues, working in order back to the

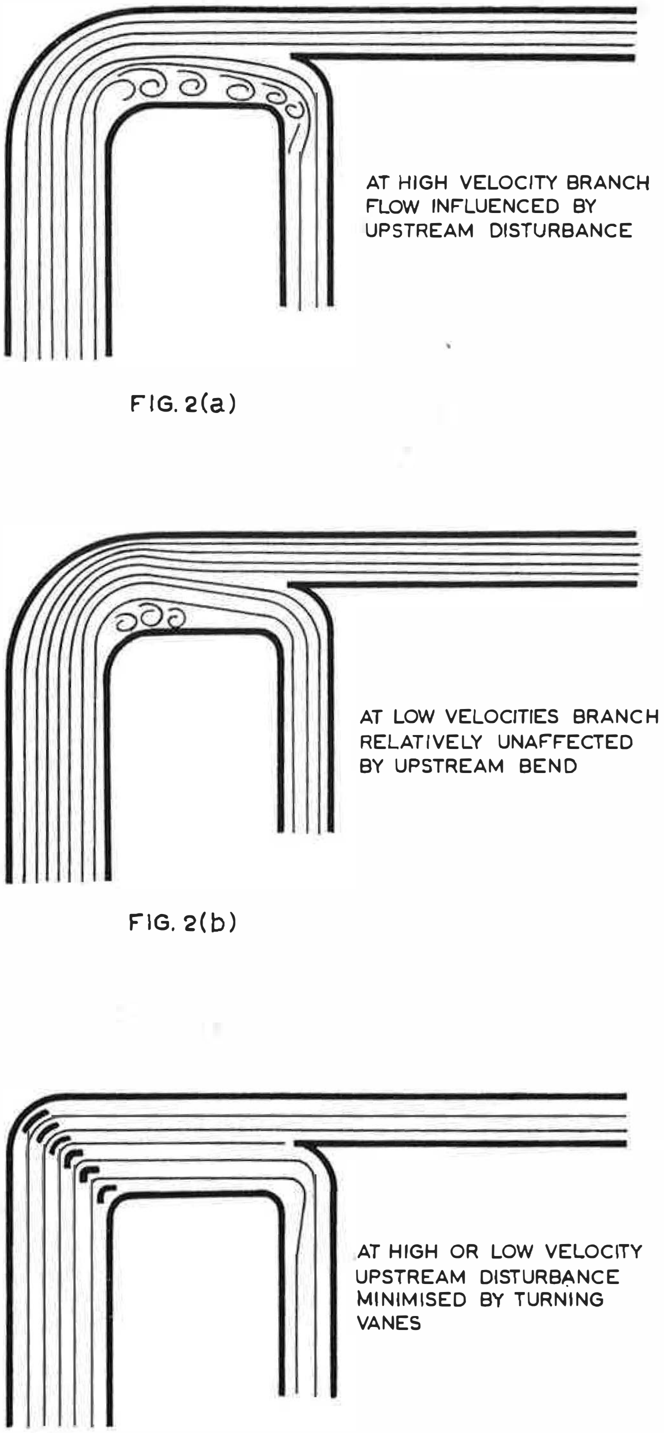


Fig. 2.—Flow pattern at junction.

fan, the golden rule being that before any junction is balanced every other junction on the side away from the fan, on either branch, must first have been dealt with. Branch dampers are set in each case (without regard for absolute values of velocity or volume) so that the proportion of air flow in each branch is in the ratio of the design volumes in the branches. When the junction nearest to the fan has been proportioned the balancing procedure is complete.

The actual volumes throughout the system can then be adjusted to their design values in one operation by finally setting the total quantity at the fan by means of the main damper or by adjusting the fan speed. In accordance with the basic principle, variation of the total volume in the system will not affect the proportion of air flowing in each branch. Hence if the air flow in every duct is the same proportion of its design volume and the fan is then set to the system total design volume, each branch and terminal will then also pass design volume.

5.2

The detailed procedure for balancing a typical system is described in Appendix B.

From this, some important features will be seen. A damper once set never needs to be re-adjusted because of alterations to dampers in other parts of the system. This eliminates the most time-wasting and frustrating aspect of trial-and-error methods.

It will also be seen that there will be, at the completion of the balancing operation, at least one path through the ducts to a terminal having every damper fully open. All unnecessary closing of dampers is thus avoided. Such a result is difficult to achieve by trial-and-error methods, when often quite unnecessary resistance may be imposed on the fan.

5.3

Extract systems may be balanced using the proportional method in the same way. Dampers are adjusted as for a supply system, starting at an end junction and working back to the fan.

6. Practical aspects of the proportional balancing method

6.1 Instruments and instrument errors

6.1.1

The pitot static tube is used for measuring the total air flow rate in the main duct and setting the main damper as the final operation to give the design air flow rate at the fan. No other instrument is so universally accepted as accurate and it is a simple instrument with nothing to go wrong, no moving parts to stick, no periodical recalibration needed, and so on.

Its weaknesses are:

(1) It cannot measure accurately very low pressure, say below 0·005in w.g. corresponding to about 300ft/min. Various instruments are available which are capable of measuring to 0·001in w.g. or less, but all of these are laboratory instruments and scarcely suitable for site use.

(2) It is inaccurate and difficult to read with pulsating or very turbulent air flow.

(3) It measures velocity only at one point, and hence total flow in a duct can only be obtained by averaging a number of readings.

However, for a reasonably accurate setting of final fan volume, a carefully chosen inclined gauge and pitot tube is the best (Fig. 3). This is the only measurement in the proportional balancing process that needs to be accurate in absolute terms. The accuracy attainable depends on the number of readings taken across the face of the duct, since the velocity distribution across the duct will usually be far from uniform. The standard to be aimed at will be determined by economic factors (how much time can you afford to spend on it?) and practical ones (does it matter whether the volume is 100% or 120%?). These considerations are developed in Section 7 and Appendix C which also proposes standards for duct measurements by pitot tube.

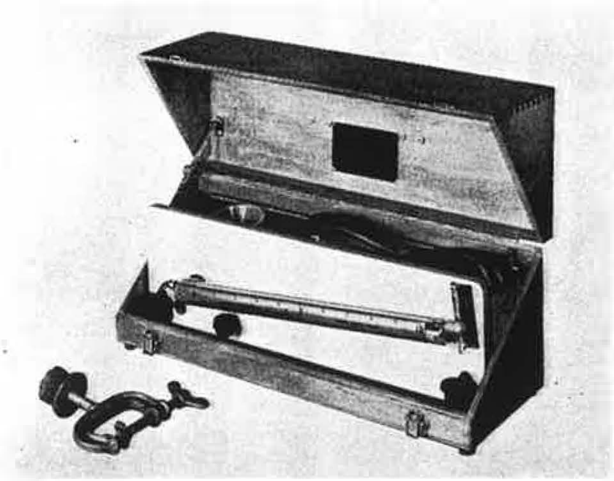


Fig. 3.—Portable inclined gauge set.

A further possibility with this instrument is to use a pair of matched heads with a ratio-indicating circuit substituted for the normal frequency-measuring arrangement calibrated in absolute values of velocity. For the proportional balancing method advocated in this paper, such an instrument could be used with one head fixed at the reference outlet while a whole series of subsidiary outlets was balanced against it, with an obvious saving in time and trouble.

At diffusers a vane anemometer may be used in conjunction with a lightweight hood similar to that shown in Fig. 5. Such a hood designed for use with diffusers (or grilles) can be used for comparing their performance by readings taken on an anemometer mounted in the neck. The coefficient of discharge of the hood and its resistance need not be known since they affect equally similar diffusers being compared. In designing the hood its resistance should nevertheless be kept to a minimum so that the performance of a terminal is measured as near as possible to its true performance. For most normal diffusers a velocity up to 1000 ft/min in the neck of the hood may be used.

Fig. 4.—Electronic anemometer

6.1.2

At grilles and diffusers, however carefully instruments suitable for use at site are used, it is inevitable that readings obtained cannot give an accurate indication of air volume[7]. Accuracy is affected by such factors as the mechanical condition of the instrument, the calibration of the instrument, the angle at which the instrument head is held, and the velocity profile at the terminal. The operator of the instrument himself introduces further errors in his particular way of handling the instrument, of reading the scale, or of making a timed measurement. But beyond all these, the type of grille and its effect on the air flow have a very large and unknown influence on any air measurement made at its face. This means in practice that absolute measurements at grilles are very difficult if not impossible to make.

The proportional method of balancing does not depend at all on absolute values of either velocity or volume but compares the performance of one branch with another. The effect of this is that errors which are factors common to two readings being compared will be cancelled out. Relative errors may, of course, still arise, but they will invariably be smaller than the absolute errors. To take the greatest advantage of this feature some simple rules should be followed:

(*a*) A group balance should be performed by the same operator using the same instrument at each terminal or at each duct test point.

(*b*) A number of terminals or test points should be balanced as a group only if the terminals comprise similar grilles or diffusers of a single type. Should there be a mixture of grille or diffuser types in a single balancing group, readings made at the different types may not be comparable. It will then be necessary to relate one to the other by, for example, taking duct readings with the same instrument.

For *relative* flow at terminals the instrument used need not be accurate in absolute terms but it must be *consistent*. A vane anemometer has weaknesses but is probably the simplest and most suitable instrument for readings at most terminals.

A variant of the ordinary vane anemometer is the electronic indicating instrument illustrated in Fig. 4. In this the usual gear train is entirely eliminated, the indication being given by the series of impulses as each blade passes a capacity transducer built into the housing of the instrument. All friction except that of the rotor bearings is thus eliminated, and the indicating dial which gives continuous indication of velocity can be remote from the head.

6.1.3

Duct measurements can very largely be eliminated. This is a considerable advantage since by the time a system can be balanced ducts are often concealed and access is difficult or impossible—and the cost of providing test points in ducts can be saved. However, where it is more convenient to take readings in ducts rather than at inaccessible terminals, the best instrument to use is again a pitot tube. (See 6.1.1. above.)

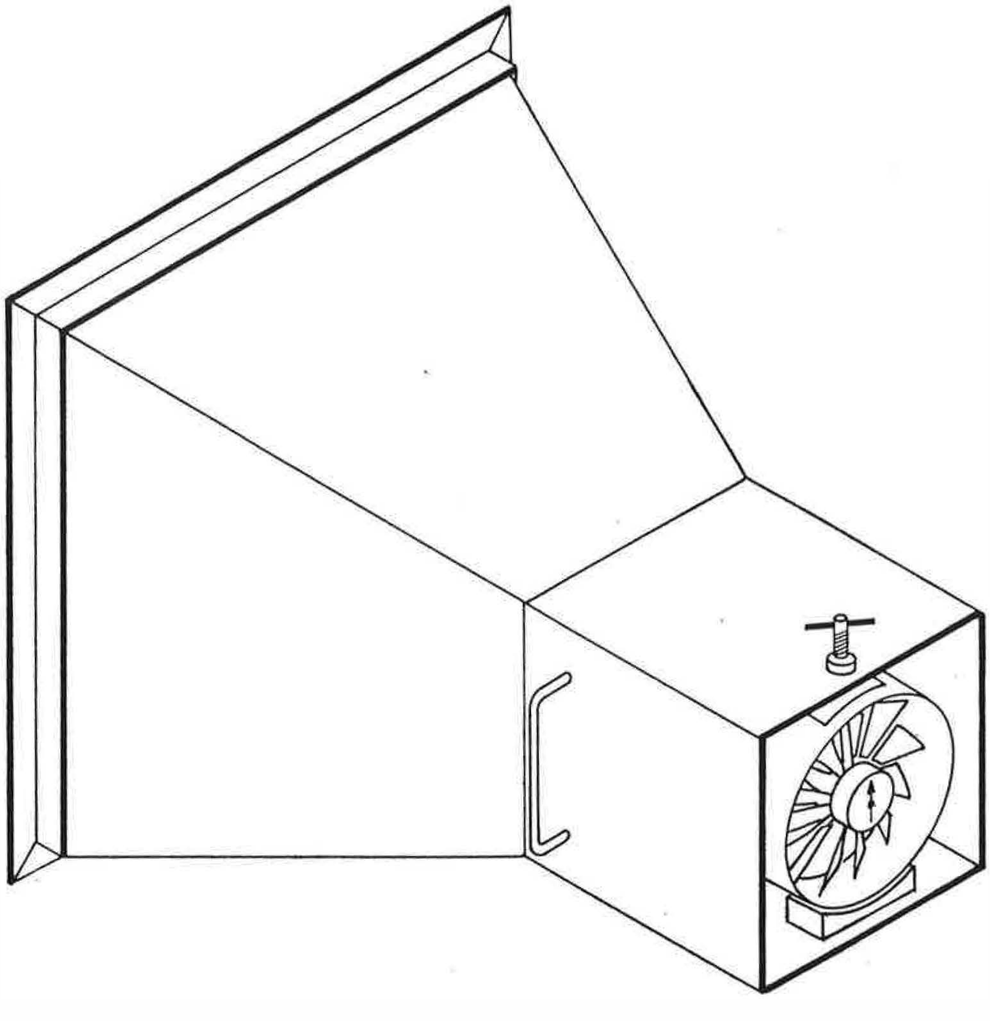


Fig. 5.—Hood for diffused measurements

6.1.4

Ventilating ceilings and linear diffusers present a problem in measurement and balancing which is again more easily handled by the proportional method. Whatever method is used for measuring the air flow, whether it be by direct instrument reading, by hood or by duct measurement, errors will inevitably occur in attempting to determine absolute values for air flow. The proportional method eliminates the need for absolute values in these readings and cancels out the errors common to all measurements.

6.1.5

Corrections for varying air density because of altitude or temperature are eliminated by the proportional balancing method, provided the branches being balanced at any time are being fed by air at the same condition. Any interposed reheat batteries, for example, must be out of action. (Corrections should of course be applied as necessary for the final fan volume).

6.2 The effect of variations in the system during the balancing process

Variations in conditions affecting the system during balancing are minimized and some are eliminated by the use of the proportional method. This is because each group is balanced as a small complete operation, without relation to conditions elsewhere in the system perhaps on another day. A very large system may take several days or even weeks to balance completely and during this time there may be variations in wind and weather, fan-motor voltage, stack effect, plant resistance and air temperature. The effect of variations in these is limited with the proportional method, to the variation occurring only during the balancing of any one group which would normally take quite a short time. Should it be necessary to interrupt the balancing of a group, the effect of such variations is no more serious since each branch (or terminal) is always balanced in relation to the current performance of the reference point.

The factors which govern the magnitude of wind and stack effects and the methods used to minimize their results are discussed in the literature (e.g. Rydberg[8]) and need not concern us here. The important fact in relation to the question of balancing an air system is that these disturbances are almost always present in some degree and that they can vary from day to day, perhaps even from hour to hour, in a given case. Two cases may be distinguished: (*a*) in which the total flow rate in the system is altered by wind or other disturbance but the distribution or balance remains undisturbed, and (*b*) in which the balance of the system is disturbed with or without an alteration in the total flow rate. In general a sealed building with inlet and extract ventilation is usually not subject to disturbances of the second category. Wind (producing under- or over-pressure at the fresh air intake or exhaust outlet) may increase or reduce the total quantity of air handled by a system but will not materially alter the proportions of flow in its several parts.

A system, however, with natural inlet or exhaust openings (e.g. through windows or grilles behind radiators) at different levels and orientations is particularly prone to disturbances affecting its balance. Both classes of disturbance may be undesirable on various grounds, and both can produce serious difficulties when attempting to balance a system.

The difficulties, however, are much more serious when the balancing method is one in which the attempt is made, by trial-and-error or any other means, to set absolute values at every branch at each stage of the procedure. If, because of wind, for example, the total air quantity handled varies from day to day and several days are needed to complete the testing and balancing, it is plain that an accurate final balance is impossible by such a method; any branch or branches set up correctly on one day will handle incorrect absolute air quantities on subsequent days. If, on the other hand, the proportional balancing method is adopted, changes in the total flow rate from day to day do not matter in the least. The absolute value of the total air flow is set once only at the main damper near the fan as a last operation. The total air quantity set will be correct only for that one day on which the testing and regulating of the main or total rate of air flow is carried out. If the system is badly affected by wind it may be advisable to re-check the total flow under different weather conditions, but the balancing of the system can be achieved however much the total air handled may vary from day to day during the balancing process.

In systems in which separate parts of the system (e.g. an east and a west wing) are differently affected by weather conditions, the proportional balancing method again shows clear advantages, since the balance (as distinct from the total quantity) can be correctly set up within each section, leaving only the balance *between* the sections and the total quantity to be set up (and if necessary re-checked under various conditions).

Of course it is better so to design a ventilation system that it will not be seriously affected by wind. But such effects are difficult to eliminate entirely and the advantage of a method of testing and balancing which can produce accurate results despite the influence of wind is quite evident.

6.3 Noise

The generation of noise at positions near the occupied space by severe throttling is limited by the proportional method. This is because terminal dampers have only to absorb the out-of-balance resistance between terminals in the same group. Out-of-balance resistance between various parts of the system is taken up by branch and main dampers but never at terminals.

7. Accuracy and working limits

7.1

Some consideration is required of the degree of accuracy that may reasonably be expected from a balancing operation. The aim must be to make the correct compromise between the demands of the system design and what is practically possible, taking into account the various sources of error and the cost of the work involved.

To set too high a standard of accuracy is unrealistic. Readings taken on site instruments can rarely be repeated to within 5%. The operator himself introduces his own particular inaccuracies. There is inevitably inaccuracy in measuring the total volume or average velocity in a duct (or at a grille) having, as it will, an irregular velocity profile. Accuracy can be increased only by taking a large number of readings in a traverse at a test point.

7.2

Even with a consistent and reliable instrument such as a pitot tube, accurate measurements of flow in ducts are not easy to make under site conditions, and the standards which are appropriate for laboratory measurements are quite inappropriate and unattainable except at extravagant cost, for measurements taken on site. A ductwork system cannot always (or often) be laid out so that measuring stations are sufficiently far downstream of disturbances occasioned by bends, dampers and the like, while, at the same time, the accurate placing of the pitot tube at a multiplicity of hopefully prescribed gauging points in order to make a '10-point traverse' according to BS: 1042 or whatever, is so difficult (except perhaps in very large ducts) as to render the supposed accuracy completely illusory under site conditions.

Appendix C discusses the question of the accuracy attainable under site conditions and proposes certain standard methods of making a pitot tube traverse of ducts of various sizes. In as much as the absolute values of air flow are of interest only when measuring the total fan volume at the conclusion of the proportional balancing of all the terminals on the system, it is only the larger sizes of duct that are of interest; but the proposed standard does cover all ducts down to the smallest in which a standard pitot tube can be used, for use when balancing by duct measurement rather than terminal measurement has to be adopted.

7.3

The number of readings at a grille called for on the test sheet illustrated in Fig. 6 is again a practical compromize for site conditions. This procedure carried out on grilles having roughly similar air flow patterns may be expected to give variations between readings of perhaps $\pm 5\%$. Of course, the absolute error may be more but as long as similar terminals are being compared in the same manner this is of no consequence.

7.4

The type of system to be balanced has an important bearing on the accuracy to be achieved. Clearly there can be a difference in the treatment given to a concert hall where conditions of temperature, air movement and noise are critical compared with, say, a factory where the balance between one zone and another may be of little moment. Air-conditioning supply grilles to partitioned offices need much more precise adjustment than general extract grilles from an open area.

7.5

From the above it will be clear that the degree of accuracy required in balancing must be variable according to the type of system and the position in the system, but that to set too close limits is unrealistic. Further, the cost of the balancing operation is directly related to the degree of accuracy required. Close limits of accuracy will increase the number of damper movements and readings taken for each damper setting. Wide limits of accuracy on the other hand result in more points where no adjustment at all is needed.

Table I, Column (*a*) gives typical working limits of accuracy that may be regarded as appropriate for normal installations given good practice in balancing by the proportional method. Some jobs may not demand such a standard and wider limits for balancing may be permissible. On other systems the

AIR TEST F
SHEET No ....

VELOCITIES AT INLET GRILLES WITH FIXED OR CLOSELY SPACED ADJUSTABLE BLADES MEASURED BY ANEMOMETER

JOB ______________________ PLANT ______________________
INSTRUMENT No ______________ CALIBRATION CHART PROVIDED ☐

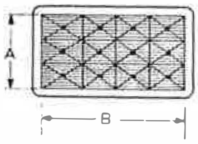
METHOD:-

1 CHECK DIMENSIONS 'A' & 'B' ON GRILLE AND NOTE

2 SHUT DOORS AND WINDOWS RUN ANY EXTRACT PLANT AND OPEN EXTRACT GRILLES

3 FRONT BLADES MAY BE SET OR NOT, TO GIVE DEFLECTION

4 ADJUST REAR BLADES TO GIVE APPROXIMATELY CONSTANT VELOCITY OVER GRILLE FACE



No. OF POSITIONS
(AND TIME AT EACH)

B →	6in	$6\frac{1}{2}$in-12in	$12\frac{1}{2}$in-18in	>18in
A ↓	No(secs)	No(secs)	No(secs)	No(secs)
6in	1(60)	2(30)	3(20)	4(15)
$6\frac{1}{2}$in-12in	2(30)	2x2(15)	2x3(10)	2x4(7-8)
$12\frac{1}{2}$in-18 in	3(20)	3x2(10)	3x3(6-7)	3x4(5)
>18in	4(15)	4x2(7-8)	4x3(5)	4x4(3-4)

5 SLIDE ANEMOMETER OVER BLADES WITHIN DIMENSIONS 'A' & 'B' WITH DIAL FACING OPERATOR, WITH APPROXIMATELY EQUAL TIME AT EACH POSITION (SEE TABLE) NOTE TOTAL READING AFTER EXACTLY 1 MINUTE. TAKE A MINIMUM OF TWO SUCH READINGS

6 WRITE DOWN READINGS AFTER CORRECTION FOR CALIBRATION WHERE TEMPS. ARE OUTSIDE THE RANGE 45 °F-85 °F OR ALTITUDE ABOVE 2 500 ft A CORRECTION FOR AIR DENSITY MAY ALSO BE NEEDED. AVERAGE THE READINGS

7 DESIGN VELOCITY = $\frac{\text{DESIGN VOL.}}{\frac{1}{2}(\text{FREE AREA} + \text{GROSS AREA})}$ GRILLE GROSS AREA = $\frac{A \times B}{144}$ ft^2
FREE AREA: FOR FIXED BLADES FROM MAKERS CATALOGUE (N.B. ADJUSTABLE BLADES, USE FREE AREA CONVERTED TO BLADES SET AT RIGHT ANGLES)

8 WHEN REGULATING A SYSTEM NOTE (FROM AIR TEST 1) THE LIMITS FOR BALANCING:- 0+ ....%

GRILLE No.	GRILLE A	GRILLE B	GRILLE NET AREA ft²	GRILLE GROSS AREA ft²	GRILLE MEAN AREA ft²	DESIGN VOL ft³/min	DESIGN VEL. ft/min	TEST READING ft/min	TEST MEAN ft/min	TEST % OF DESIGN	COMMENT
(1)	(2)	(3)	(4)	(5) = $\frac{(2)\times(3)}{144}$	(6) = $\frac{(4)+(5)}{2}$	(7)	(8) = $\frac{(7)}{(6)}$	(9)	(10) = $\frac{(9)1+(9)2}{2}$	(11) = $\frac{(10)}{(8)} \times 100$	(12)
CHECK								1 2			
CHECK								1 2			
CHECK								1 2			

Fig. 6

reverse will apply. In any case, working limits must be selected to suit the type of job and specified to the test engineer.

The variation in air quantities over a small group of points after balancing may of course be wider than the tolerance for balancing one junction. If for example we consider a group of outlets in a partitioned area on a supply system, Table I calls for the balancing to be done at each outlet to $+15\%$, -0% of the reference outlet. If there were a statistically large number of outlets in this group the maximum variation from the group mean would be $\pm 7\frac{1}{2}\%$. On the other hand if there were only a few outlets in the group the maximum possible variation from the group mean could tend to $\pm 15\%$. In practice the actual variation will lie between $\pm 7\frac{1}{2}\%$ and $\pm 15\%$. Column (*b*) has been arrived at from experience—without any statistical corroboration—as the likely final variation between points in groups of 6 to 50 points after balancing.

Table I—Limits of accuracy for balancing

System	(a) Balancing done to within the limits below	(b) Likely variation in air quantities at completion of balancing
General Supply Systems		
Outlets on any one branch when *not* all are in the same room ..	$+15\%$, -0% of reference outlet	$\pm 10\%$ of branch mean
Outlets on any one branch when all are in the same room ..	$+20\%$, -0% of reference outlet	$\pm 15\%$ of branch mean
Main branches	$+5\%$, -0% of reference branch	$\pm 5\%$ of mean of branches
General Extract Systems		
Intakes on any one branch when *not* all are in the same room ..	$+20\%$, -0% of reference intake	$\pm 15\%$ of branch mean
Intakes on any one branch when all are in the same room ..	$+33\frac{1}{3}\%$, -0% of reference intake	$\pm 25\%$ of branch mean
Main branches	$+15\%$, -0% of reference branch	$\pm 10\%$ of mean of branches

7.6 Time for balancing

The time required for balancing is variable depending on the type of system and the accuracy required. From records taken of over a hundred systems the variation ranges from 0.2 team hours per terminal to 20 team hours per terminal, though most systems lie between 0·4 team hours and 7 team hours per terminal. A typical figure for a straightforward supply system (of the type shown in Fig 14) might be between 0·5 and 1 team hours per terminal (with accuracy required as given in Table I for supply systems). A team in this context consists of a tester and an unskilled helper. It will be evident that in reducing the balancing process to a fixed routine by the use of the proportional method, the tester will need limited training and experience.

Time required for balancing may alternatively be expressed in terms of the system air volume. This gives rather less variable results, ranging over the same installations quoted above from 1–20 team hours per thousand ft³/min. On this basis the typical time taken for the system above would be about 5–10 team hours per thousand ft³/min.

Emphasis is placed on the importance of the installation being complete to the necessary degree (see Appendix B, section B.4) and in mechanically working order, and of all design information being in the hands of the test team in the right form, if the balancing operation is to be carried out smoothly and in good time.

8. Test Sheets and Records

8.1

Fig. 6 shows a typical test sheet for use when measuring grille velocities with a vane anemometer. The design figures should be filled in by the office before testing is started. Similar sheets can be devised for other situations and other instruments.

AIR TEST I

AIR FLOW BALANCING TEST DATA

Job... Plant..

SYSTEM DATA

Fan No. .... Vol:............ft³/min S.P. ....in w.g. Speed......... r.p.m.

Motor .... H.P. Speed.........r.p.m. F.L. Current..................... amps.

Filter Type............... Vol:..............ft³/min Resistance clean..................... in w.g.

dirty........................ in w.g.

Design air temp. at fan discharge.....................°F.

Heater Vol:..............................ft³/min Resistance..............................in w.g.

Cooler Vol:..............................ft³/min Resistance..............................in w.g.

Washer Type.................. Vol:..................ft/³min Resistance.....................in w.g.

State setting of plant dampers, automatic and manual, for test, and any special instructions.

....

....

Approx. no. of outlets or intakes ....

Layout Drawings Nos. ....attached ☐

Diagram (drawing No...................) lettered and numbered,...........................attached ☐

Tests required

	No. of Sheets	*Sheet (A to S)*	*Test Points*	*Limits for Balancing* *Single Points*	*Limits for Balancing* *Open Areas*	*Design figures entered and attached*
Velocities at	....	....	....	$^{+}_{-}$ 0 %	$^{+}_{-}$ 0 %	☐
Outlets or	....	....	....	$^{+}_{-}$ 0 %	$^{+}_{-}$ 0 %	☐
Intakes (or in	....	....	....	$^{+}_{-}$ 0 %	$^{+}_{-}$ 0 %	☐
minor branch ducts).........		....	....	$^{+}_{-}$ 0 %	$^{+}_{-}$ 0 %	☐
Velocities in	....	....	....	$^{+}_{-}$ 0 %		☐
Main Branch	....	....	....	$^{+}_{-}$ 0 %		☐
Ducts	....	....	....	$^{+}_{-}$ 0 %		☐
Static Pressures	....	P	....	$^{+}_{-}$ 0 %		☐
	....	D	....			☐
Fan Volume	....	....	....	$^{+10}_{-\ 0}$ %		☐
Noise	....	N	....	....../.........N.C.		☐

Sheet completed by.. Date.................................

Fig 7

AIR TEST II

AIR FLOW BALANCING TEST

Job.. Plant..

STATE OF SYSTEM — Fill in ☐ with ✓ or no

Fan and Motor. Lubrication O.K. ☐ Rotation O.K. ☐

Drive O.K. for tension and alignment ☐ Ammeter set for full load ☐

Running conditions. Noise O.K. ☐ Anti vibration O.K. ☐

Duct System Complete ☐ All grilles/diffusers fitted ☐

Access doors on ☐ All dampers open ☐

Filter. Viscous or electrostatic working ☐ Fabric clean ☐

Washer. Capillary cells fitted ☐ *Heaters/coolers* fitted ☐

Plant Dampers set as Air Test I ☐

State anything incomplete in system affecting test ....

....

TEST

1. *Initial fan load.* Adjust speed or main damper to give any fan volume up to about 25% more than design volume provided the motor is not overloaded. Read:

   Motor. Current..................amps. Speed..................r.p.m.

   Fan. S.P. Inlet ±..................in w.g. Discharge ±..................in w.g.

   Air temperature at fan discharge..................°F. Fan speed........... r.p.m.

2. *Outlets or Intakes.* Balance velocities against reference points to limits stated on Air Test I. Enter readings on sheets.

3. *Branch Ducts.* Balance branch velocities against reference branch to limits stated on Air Test I. Enter readings on sheets.

4. *Noise.* Check noise level by ear in rooms adjacent to plant and comment............................

   ....

   Measure noise level where required on Air Test I and enter readings on sheets.

   Noise level O.K. ☐ Remedial action wanted ☐

5. *Final fan load.* Plant and system now complete ☐

   Motor. Current.....................amps. Speed.....................r.p.m.

   Fan. S.P. Inlet ±.....................in w.g. Discharge ±....................in w.g.

   Volume....................ft³/min Speed.....................r.p.m.

   Air temperature at fan discharge.................°F. Filter resistance..................... in w.g.

   Other readings required ....

6. *Dampers.* Fixed ☐ Marked/and pinned ☐ Method...............................

7. *Test sheets* (as Air Test I) completed and attached ☐

Date of Test Begun................................ Completed ....

Tested by .... Approved by ....

Fig. 8

Fig. 7 shows a summary sheet to be completed by the office for each air-handling system giving data required by the test team. Fig. 8 provides a check list of the preliminaries and constitutes a reminder of test procedure for the test team. When completed it forms a record of the plant performance during the balancing. Fig. 9 forms a summary of the test sheets for individual groups, and forms a record of the final state of the system after balancing and after setting the fan volume.

Coupled with the proportional balancing method, the provision of such test sheets showing design information together with an outline sketch of the system (see Appendix B, para. B.4.1 and Fig. 14) and the layout drawings, makes the balancing procedure at site speedy and reliable and results in a fully documented record of performance.

8.2 Checking system performance

It is important to realise that when the performance of a system is to be checked for any reason after it has been balanced, it is quite misleading to attempt to take absolute readings of air volume or velocity at terminals or ducts, because of the inevitable errors in measurement mentioned above. Checking should only be done on a comparative basis between groups following the same pattern in which balancing would be done. Thus if, for example, the performance of a grille is suspect, its performance should be compared, by the same instrument and operator, with other grilles in the same group. If it is found to vary, outside the limits of accuracy required, from the other grilles in the same group, then the group may be balanced to bring the suspect grille into line. If on the other hand the suspect terminal does not vary from its neighbours then more harm than good is done by adjusting its damper. The next step is to compare the performance of the branch duct feeding the group with the other branch ducts of the same order. As soon as a discrepancy is found in a group then adjustments may be made. If no discrepancy is found between neighbours within groups right back to the fan it may be deduced that the system is in balance and if the fan volume is correct that each terminal is passing, within the working limits, its design volume. This statement implies that there is continuity within the ducting and that there is no major leakage at any point in the system. If there is suspicion of leakage the only course is to make a physical check.

9. Experience in use

The discussion so far will in itself demonstrate the benefits to be derived from the proportional balancing method. The development of the system to its present stage is a result of its application extensively on some recent very large contracts. The method is now in general use in the authors' company, resulting we believe in a general improvement in the performance of systems.

10. Design Features

10.1 Ductwork design

One of the reasons why ductwork systems (in contrast to water piping systems) require considerable regulation by means of dampers in order to balance them, is that it is seldom possible to design short branches near to the fan so that all or most of the available head is absorbed. Indeed, very often no attempt at all is made to do so, all ducts being designed either according to some pre-selected velocity range, or for constant pressure drop per 100ft run, or some other principles, none of which pays much regard to the available pressure at subsidiary branches. The limitation in ductwork systems which makes the absorption of available head by reduction of duct sizes impossible or undesirable is the noise resulting from the high velocities obtained. One way or another the head must be absorbed however, and the proper way to do this is by a damper as far upstream of the outlet as possible so that noise attenuation in the duct will reduce the damper noise before it issues into the room. There is no reason however, why the smallest possible duct should not be chosen for such branches consistent with the requirement that a safe velocity should not be exceeded. What can be regarded as a safe velocity in various circumstances is a large question which has been discussed elsewhere but probably could profitably be the subject of more research.

In the present context the advantages of careful attention to design in this question are pointed out since a little extra design time may save quite a large amount of testing and balancing time on site afterwards.

The need to adjust the air flow in a system places on the designer the responsibility of planning the system and laying it out so that the balancing may be done effectively. It is pointless to spend time on careful design and accurate calculations if the completed installation cannot be set up to give its required performance. Some of the most important features are mentioned below.

10.2 Dampers

Sufficient thought is seldom given to the demands made on the adjustability of a duct system resulting from the limitations discussed above and mentioned in para. 1.2. The numbers, the location and the effectiveness of the dampers provided too often leave very much to be desired. Sufficient dampers, in the right places, are an absolute necessity. Strictly dampers should be called for on all branches of a duct and at all grilles or diffusers. The nearer the fan the branch position is, the more important the damper becomes. In practice it is usually unnecessary to fit dampers to branches serving fewer than four terminals which are themselves fitted with dampers. On low velocity extract systems it may be permissible to omit dampers on a group of grilles or diffusers serving the same space, but the damper controlling the branch which supplies the group then becomes the more important. Dampers are not required on the main run of a duct except where the duct is itself a branch of another duct of higher order. A damper is needed on what is theoretically the index branch since it may not be so in practice. A main damper at the fan is essential.

Dampers must be positioned far enough upstream of a junction so that the variation in flow which they produce downstream does not interfere with the pattern of flow at the junction (see para. 4.3). In positions where a damper is liable to interfere with the pattern of flow at a junction, the effect will be reduced if the damper is of the opposed blade multi-leaf type or of iris pattern. Where dampers are well upstream of branches, butterfly dampers, or, in larger sizes, parallel multi-blade dampers may be sufficient unless much head has to be absorbed. An inspection opening should be specified at larger dampers, which are not visible through grilles or diffusers, so that a check can be made that the damper is correctly fixed on its spindle and functioning properly.

For control dampers and main plant dampers an access door or plate is needed.

Dampers should be substantially built and without sharp edges, have adequate bearings and mechanically sound adjusting and locking arrangements.

10.3 Duct layout

Ducts should be arranged so that junctions occur as far as possible downstream of bends, other branches, dampers or other disturbances to air flow. An elbow with properly designed turning vanes will produce much less disturbance

AIR TEST III

Fig. 9

AIR FLOW BALANCING TEST
SUMMARY OF APPROVED TEST SHEETS

Job.. Plant ....

Date of Test .... Fan No. ....

Test Points	Limits					Comment
	Design		Test			(Tick if satisfactory)
	For balancing (1)	Final (2)	Balanced (3)	Final Check (4)	Any points outside (4) (5)	
Outlets/Intakes (partitioned areas)						
Outlets/Intakes (open areas)						
Main branches						
Fan volume, after balancing						
Fan volume, after plant dampers set						

Remarks on state of system as left.
(Tick if satisfactory)

Accepted:

Date:

downstream than an ordinary bend. Such a fitting need not be expensive and it saves space and pressure loss. Care in the design of branches—as well as of dampers mentioned above—is also necessary to avoid difficulties with measuring and balancing.

10.4 Duct test points

Usually few duct test points are required. Where necessary their location should be carefully selected. Appendix C gives some guidance.

10.5 Grilles

Supply grilles are often fitted on the side of a duct with a very short branch or neck connection. Unless the duct velocity is very low, this commonly leads to the very undesirable condition that air issues at high velocity from the downstream edge of the grille and is entrained (giving a negative velocity reading when measured) at the upstream edge. Such a pattern of air distribution is not only likely to be objectionable on grounds of comfort and noise, but also makes testing and balancing difficult. Ways of avoiding such troubles are described in the literature (see for example Nelson and Smedberg's paper[9]). Turning vanes behind the grille are an obvious solution but expensive. Almost as good results can be obtained with fixed vertical blades behind the grille. The ratio of width of blade to spacing must be 1·5 or higher to make them effective, and an opposed blade damper behind the grille (if used) should have vertical blades rather than horizontal so as to help in straightening the air.

10.6 Plant design

Plant output conditions frequently depend on the operaion of controls regulating mixing or face and bypass dampers. To set up these controls the air flow in the various paths through the plant must be proportioned and measured either by direct means or differential pressure measurements. The layout of plants must be such that these measurements are possible. It is illogical to call for expensive automatic controls capable of maintaining conditions within close limits if the basic air flow on which they depend can be adjusted only by guesswork.

The operation of automatic dampers can often be the cause of considerable variation in fan volume. In the balancing process such dampers must anyway be kept fixed in one position (see Appendix B, para. B.4.2) but, when functioning automatically, unless carefully designed to avoid such troubles they can have the effect of changing the set-up fan volume as they modulate through their intermediate positions(4).This means that the fan volume will be as intended only when the dampers are in the position at which they were set when the fan volume was regulated. This must be borne in mind when making subsequent checks. Also it may cause trouble with fan motor overloading and with room conditions. More thought is needed on the design and operation of automatic dampers to prevent these difficulties.

10.7 Fan performance

Fan performance curves can be invaluable to the test team as an alternative means of estimating the performance of both fan and system. On fan motors ammeters are a valuable aid particularly in the first adjustment of the fan volume.

11. Application of proportional method of balancing to high-velocity induction systems

The proportional method described can be applied successfully to high-velocity induction systems. Because of the properties of induction systems, in particular the terminal resistance of the induction units, the procedure can generally be considerably shortened. This is because the adjustment of one damper has only little influence on the air flow at other parts of the system even in the same group.

Consequently a single measurement at the reference terminal for a group may be used to balance a number of units in the group without repeated reference back to the reference terminal. This simplification may be possible on exceptionally large and regular low-velocity systems, but it may be considered as normal procedure on high-velocity induction systems. Another point of dissimilarity with low-velocity systems is that, for the same reasons above, the procedure can be performed with dampers only on the most important main ducts: dampers are not needed on branch ducts. The routine for balancing applied to high-velocity induction systems is given in detail in Appendix D.

12. Conclusions

The proportional method of balancing described in this paper is believed to be the most economical and most accurate practical method of setting up the performance of a ventilation system. It can be carried out with instruments which are suitable for use on site. Its accuracy is least affected by conditions normally to be expected on building sites, such as breaks in continuity, the interaction of other trades and activities, variability of access, occupancy. It is the one method of balancing which is almost completely unaffected by variations in such things as wind and temperature, stack effect and supply voltage. The method can be operated by staff with comparatively little training and experience.

The extended use of the method can make a significant contribution to the improvement in performance of ventilating and air conditioning systems.

Looking to the future, there is room for the further development of the system in conjunction with reliable and accurately -calibrated dampers. We may also perhaps look for the advent of an economical self-adjusting constant volume unit suitable for low-velocity systems, analogous to constant volume boxes common on high-pressure systems.

Appendix A

THE EFFECT OF A CHANGE OF TOTAL AIR FLOW RATE ON THE PROPORTION OF THE WHOLE FLOWING ALONG EACH BRANCH OF A Y OR TEE

A.1. List of symbols

$C_{1,2}$, etc.	=	constant conversion factors for equations in non-consistent units.
D	=	dia. of duct in feet.
d	=	dia. of duct in inches.
f	=	coefficient of friction $= \dfrac{2gDH}{4Lu^2}$ (dimensionless).
g	=	acceleration due to gravity, ft/sec^2.
H	=	head lost in feet of fluid flowing.
h_f	=	frictional head lost (in w.g.) at 62°F in straight pipe per foot.
h_d	=	dynamic head lost (in w.g.) at 62°F in fittings.
K	=	velocity head factor for dynamic loss in fittings (dimensionless).
K_s	=	absolute surface roughness in inches.
L	=	length of duct in feet.
Q	=	air flow in ft^3/min.
R_e	=	Reynolds' Number $= \mathrm{Du}\dfrac{\rho}{\mu}$ where $\dfrac{\mu}{\rho} =$ kinematic viscosity of fluid flowing ft^2/sec.
u	=	velocity of air ft/sec.
v	=	velocity of air ft/min.

A.2

Consider a branched fitting with air flowing in the main and branches as shown (Fig. 10), the branches discharging into a space at a common pressure. The flow in each branch depends on its resistance. The pressure lost in each branch due to frictional and dynamic losses must be equal at the equilibrium flow rates.

The frictional losses may be expressed by the fundamental fluid flow formula:

$$H = \frac{4f\,Lu^2}{2gD}$$

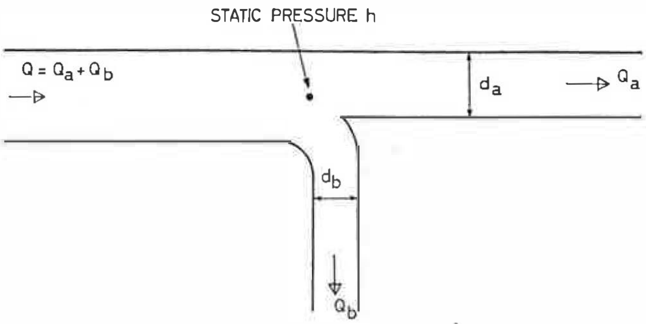


Fig. 10. Air flow at a junction

or in terms of Q, d and h_f as

$$h_f = (4f)\, L\, \frac{Q^2}{d^5} \times C_l \qquad \ldots (1)$$

(where C_l has the value of 0·02513 for air at 0·075 lb/ft³ and when the water in the gauge is at 62°F).

The dynamic losses in fittings, etc., are given by:–

$$h_d = \sum k \left(\frac{v}{4005}\right)^2 \text{ in w.g.}$$

or, again, in terms of Q ft³/min as:–

$$h_d = \sum k\, \frac{Q^2}{d^4} \times C_2 \qquad \ldots (2)$$

(C_2 has the value 0·0021.)

If the flow rate Q in the approach duct is now varied, the proportion of air in the branches will not alter so long as the *relative resistance* of each branch remains the same. The relative resistance of the fittings in each branch will not alter. The resistance of a fitting is always assumed to be (and probably in fact very nearly is) proportional to the square of the velocity. But the frictional resistance in the two branches will not necessarily always remain in proportion because the friction factor f is not a constant but varies with Reynolds' number. Air flow in ducts at the usual velocities is almost always within the transition region between laminar flow and fully developed turbulence. For example, the lowest Reynolds' number (in a 2 in dia duct with air flowing at 200 ft/min) is of the order of 3400 and the upper limit of the transitional zone with normally smooth metal ductwork would occur at Reynolds' numbers of round 10^6 or higher corresponding to high velocities in ducts of more than 30in dia. The variation in friction factor f (in the transition zone between laminar flow and fully developed turbulence) is expressed by Colebrook and White as:–

$$\frac{1}{\sqrt{f}} = -4 \log_{10} \left(\frac{K}{3{\cdot}7d} + \frac{1{\cdot}255}{Re\sqrt{f}}\right)$$

Now, if the Reynolds' number in the two branches is not the same, the rate of change of the friction factor with change of flow rate will also be unequal and the proportionality of the respective resistances and hence of the flow rates in the two ducts will not be maintained.

The extent of this departure from proportionality will be examined by considering an extreme case. The duct arrangement is assumed to consist of two branches in which the smaller has a flow rate only 1 % of the flow rate in the larger. Furthermore the two branches are assumed to have no bends or other unit resistances, i.e. the whole of the pressure drop is due to duct friction.

Then, since the pressure drop in the branches must be equal:–

$$h = C_l\,(4f_a)\, L_a\, \frac{Q_a^2}{d_a^5} = C_l\,(4f_b)\, L_b\, \frac{Q_b^2}{d_b^5}$$

$$\text{Therefore } \frac{Q_a}{Q_b} = \left(\frac{L_b}{L_a} \cdot \frac{d_a^5}{d_b^5}\right)^{1/2} \left(\frac{4f_b}{4f_a}\right)^{1/2}$$

The expression in the first bracket is a constant for a given duct system, hence we may write:–

$$\frac{Q_a}{Q_b} = C_3 \left(\frac{4f_b}{4f_a}\right)^{1/2} \qquad \ldots (3)$$

We assume the following values for the initial condition when the total flow Q is at its design value:–

Q_a = *31 000* ft³/min d_a = 50in
Q_b = *310* ft³/min d_b = 14in
Hence R_{ea} = 10^6 and R_{eb} = $3{\cdot}51 \times 10^4$

Then Table II below shows the variation of the friction factors $4f_a$ and $4f_b$ and the consequent deviation of the ratio $\frac{Q_a}{Q_b}$ from its proper value of 100 for a wide range of variation of the total flow Q.

Table II— Q_a/Q_b for various values of Q when losses are all frictional

Q percentage of design	Reynolds' number		Friction factors		Q_a/Q_b
	R_{ea} $\times 10^{-6}$	R_{eb} $\times 10^{-4}$	$4f_a$	$4f_b$	
140	1·4	4·93	0·0112	0·0209	97·0
130	1·3	4·58	0·01125	0·0214	97·9
120	1·2	4·23	0·0113	0·0218	98·7
110	1·1	3·87	0·0115	0·0222	98·7
100	1·0	3·52	0·0117	0·0232	100
90	0·9	3·17	0·01175	0·0233	100
80	0·8	2·82	0·0121	0·0238	100·4
70	0·7	2·47	0·0123	0·0246	100·4
60	0·6	2·11	0·0127	0·0259	101·3
50	0·5	1·76	0·0132	0·02	101·5
40	0·4	1·42	0·0137	0·0282	101·8
30	0·3	1·06	0·0144	0·0305	103·4
20	0·2	0·704	0·0156	0·0340	104·8
10	0·1	0·352	0·0180	0·042	108·8

It will be seen that over the wide range of variation of the total flow rate from 40 % to 140 % the resulting unbalance of the system does not amount to more than 3 % or so. But this is an extreme case in which there are no fittings—the whole of the resistance is frictional, and the ratio of design flow rate in the branches is 100 to 1.

A.3

If 50% of the resistance of each branch is dynamic pressure loss in fittings of various kinds then the static pressure drop in each branch is, from equations (1) and (2):–

$$h = h_f + h_d = \frac{Q_a^2}{d_a^4}\left[(4f_a)\frac{L_a}{d_a}C_1 + \sum K\,C_2\right] \quad \ldots (4)$$

$$= \frac{Q_b^2}{d_b^4}\left[(4f_b)\frac{L_b}{d_b}C_1 + \sum K\,C_2\right] \quad \ldots (5)$$

Now we postulate that at design conditions

$$\sum K\,(4f)\;C_2 = \frac{L}{d}C_1 \text{ for each branch;}$$

Taking the design conditions as before, namely:

$Q_{a1} = 31\,000$ ft³/min		$Q_{b1} = 310$ ft³/min
$d_a = 50in$		$d_b = 14in$
$R_{ea1} = 10^6$		$R_{eb1} = 3{\cdot}52\times 10^4$
$4_{fa1} = 0{\cdot}0117$		$4_{fb1} = 0{\cdot}0232$;

$$\text{Then } \sum K_a C_2 = 0{\cdot}0117\left(\frac{L_a}{d_a}C_1\right)$$

$$\text{and } \sum K_b C_2 = 0.0232\left(\frac{L_b}{d_b}C_1\right)$$

Then substituting these values in equations (4) and (5) and combining them to give $\frac{Q_a}{Q_b}$ we have:–

$$\frac{Q_a}{Q_b} = \left(\frac{d_a}{d_b}\right)^2\left[\frac{(4f_b)\frac{L_b}{d_b}C_1 + 0{\cdot}0232\frac{L_b}{d_b}C_1}{(4f_a)\frac{L_a}{d_a}C_1 + 0{\cdot}0117\frac{L_a}{d_a}C_1}\right]^{\frac{1}{2}}$$

$$= \left(\frac{d_a}{d_b}\right)^2\left(\frac{L_b/d_b}{L_a/d_a}\right)^{\frac{1}{2}}\left(\frac{4f_b + 0{\cdot}0232}{4f_a + 0{\cdot}0117}\right)^{\frac{1}{2}}$$

$$= \left(\frac{d_a}{d_b}\right)^{2\cdot 5}\left(\frac{L_b}{L_a}\right)^{\frac{1}{2}}\left[\frac{4f_b + 0{\cdot}0232}{4f_a + 0{\cdot}0117}\right]^{\frac{1}{2}} \quad \ldots (6)$$

At design conditions in our example:

$$\frac{Q_a}{Q_b} = 100 \text{ and } \begin{array}{l} 4f_b = 0{\cdot}0232 \\ 4f_a = 0{\cdot}0117 \end{array}$$

Then writing C_4 for $\left(\frac{d_a}{d_b}\right)^{2\cdot 5}\left(\frac{L_b}{L_a}\right)^{\frac{1}{2}}$ in equation (6):–

$$\frac{Q_a}{Q_b} = 100 = C_4\left[\frac{0{\cdot}0464}{0{\cdot}0234}\right]^{\frac{1}{2}}$$

therefore $C_4 = 71{\cdot}1$

Thus equation (6) becomes:—

$$\frac{Q_a}{Q_b} = 71{\cdot}1\left[\frac{4f_b \pm 0{\cdot}0232}{4f_a + 0{\cdot}0117}\right]^{\frac{1}{2}} \quad \ldots (6a)$$

Then Table III below shows the variation of the ratio $\frac{Q_a}{Q_b}$ (when the losses are 50% frictional and 50% dynamic) for various values of Q.

In this case the maximum unbalance of the branches from the desired condition is only 1·3% over the same range of total flow variation of 40% to 140%.

Appendix B

DETAILED DESCRIPTION OF PROPORTIONAL BALANCING PROCEDURE

B.1 Example of method for balancing a junction

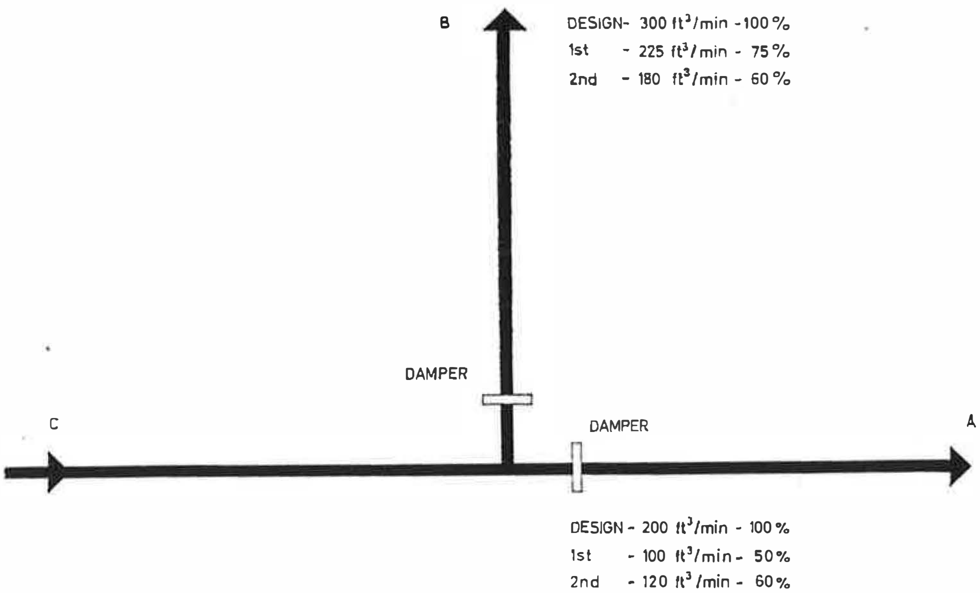
To simplify figuring, air flow measured in each branch is expressed as a fraction, or percentage of the design volume in the branch.

For a junction ABC with design volumes as shown (for example) in Fig. 11, the first step is to measure A and B with their dampers fully open. Suppose they are found to be passing

Table III—Q_a/Q_b for various values of Q when 50% of the losses are frictional

Q percentage of design	Reynolds' number		Friction factors		Friction factor plus dynamic loss factor		Q_a/Q_b
	R_{ea}	R_{eb}	$4f_a$	$4f_b$	$4f_a$ +0·0117	$4f_b$ +0·0232	
140	1·4×10⁶	4·93	0·0112	0·0209	0·0229	0·0441	98·7
130	1·3×10⁶	4·58	0·01125	0·0214	0·02295	0·0446	99·2
120	1·2×10⁶	4·23	0·0113	0·0218	0·0230	0·0450	99·4
110	1·1×10⁶	3·87×10⁴	0·0115	0·0222	0·0232	0·0454	99·4
100	10⁶	3·52×10⁴	0·0117	0·0232	0·0234	0·0464	100
90	0·9×10⁶	3·17	0·01175	0·0233	0·02345	0·0465	100
80	0·8×10⁶	2·82	0·0121	0·0238	0·0238	0·0470	100
70	0·7×10⁶	2·47	0·0123	0·0246	0·0240	0·0478	100·3
60	0·6×10⁶	2·11	0·0127	0·0259	0·0244	0·0491	100·9
50	0·5×10⁶	1·76	0·0132	0·0270	0·0249	0·0502	101
40	0·4×10⁶	1·42	0·0137	0·0282	0·0254	0·0514	101·1
30	0·3×10⁶	1·06	0·0144	0·0305	0·0261	0·0537	101·9
20	0·2×10⁶	0·704	0·0156	0·0340	0·0273	0·0572	103
10	0·1×10⁶	0·352	0·0180	0·0420	0·0297	0·0652	105·4

Fig. 11.— Balancing a junction.



100ft³/min (A) and 225ft³/min (B), that is 50% and 75% of their design volumes respectively, then damper B must be partly closed so that each branch, A and B, is passing the *same proportion* of its design volume. This might result in a balance of 120ft³/min at A (60% design) and 180ft³/min at B (also 60% design). It follows that 60% design volume is also passing C (300ft³/min).

Subsequent damper alterations above C (that is nearer the fan) will increase the total quantity in C, but once damper B has been set to divide the C quantity in the proper proportion it will never need resetting so long as no alterations below these branches are made. Thus no junction should be balanced until all dampers on the side remote from the fan have been set.

B.2 Example of method for balancing a group of junctions

The principle may be extended to a group of successive junctions such as shown in Fig. 12, as follows. Start with all dampers fully open. A and B are then balanced as described above. When A and B have been balanced, A, B and C all pass the same proportion of their design volumes and will continue to do so irrespective of changes in total volume flowing at C. This means that in balancing the junction CDE, D can be balanced against A or B and not only against C. To avoid cumulative errors, A is normally established as reference point and B, D and F are in turn balanced against the current reading at it.

In the above routine we have assumed that the end branch of A is initially receiving the smallest percentage of its design volume of air and that each successive junction can be balanced with it by closing down the damper on the other branch.

B.2.2

Should a branch other than the end branch be receiving the least percentage of design volume it would be necessary, in order to balance that junction by simple application of the routine, to have a damper on the main duct. It is evidently not a practical proposition to have a damper on main and branch at each junction and by a simple modification to the routine the need for dampers on the main can be dispensed with. This is effected by taking an initial measurement of the percentage of design volume flowing in each branch of the group. From this the least favoured branch is determined. The first operation in balancing is then to close the damper on the end branch until it is passing approximately the same percentage as the least favoured. The balancing may then proceed as before with the end branch the least favoured.

To avoid imposing unnecessary added resistance on the fan at least one damper in the group should be fully open when the group balancing is complete. This will be the case automatically when the end branch starts as the least favoured. When this is not so and the end branch must be dampered as

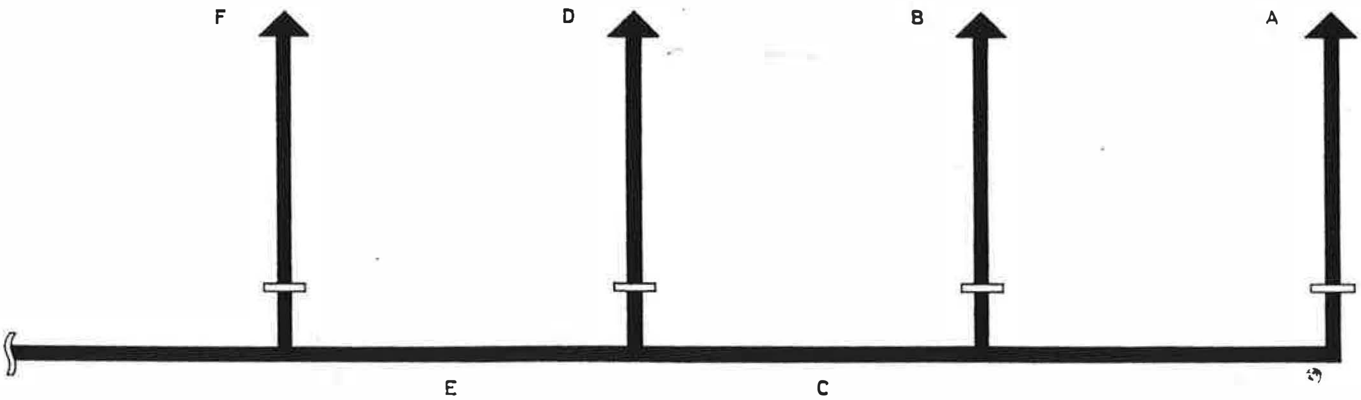


Fig. 12.—Balancing a junction.

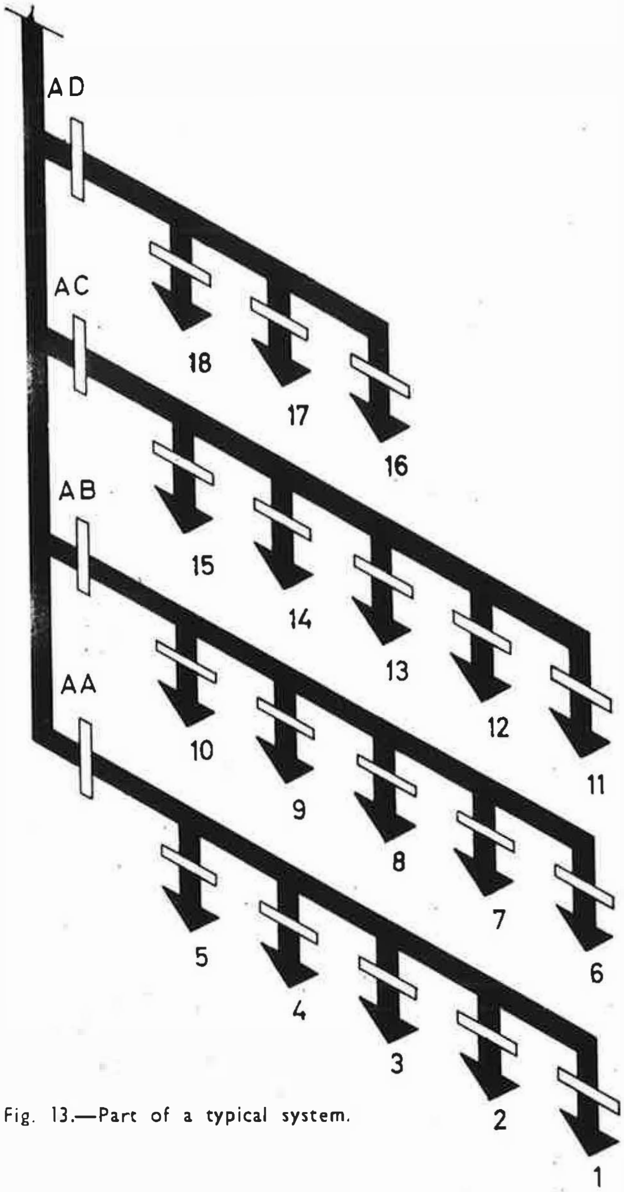


Fig. 13.—Part of a typical system.

described in the last paragraph, judgment and experience will indicate how nearly the end branch must be reduced to the branch with the least percentage. The least favoured will ideally be left wide open. If the end branch has been throttled too far it will be necessary to close partially the damper at the least favoured branch. Provided this is only slightly closed, little harm is done and it is better than if the end branch had been throttled too little. In the latter case it would be found that the least favoured was still below the end branch, when the balance might have to be repeated.

B.2.3

Note that the method of balancing a group of junctions applies equally to a group consisting of grilles, diffusers or other terminals fed by a branch duct, or to a group of branch ducts fed by a main duct.

B.3 Example of method for balancing a part of a duct system

Fig. 13 shows a number of branches with terminals fed from a main duct. By following the procedure described above terminals 1 to 5 on duct AA may be balanced. Moving towards the fan we come to the junction of AB and AA. Here our rule says no attempt must be made to balance this junction until all dampers downstream, on the side away from the fan, have been set; the next step therefore is to balance terminals 6 to 10 using the same procedure. Branches AB and AA could now be balanced, but it is best to wait until the other groups of terminals on the branches AC and AD have been balanced and then to deal with the branches AA, AB, AC and AD as a group.

The group of branches from duct A (AA, AB, AC, AD) can be balanced using the same procedure as for a group of terminals. First take check measurements for each branch to locate the least favoured. If this is not AA then AA must be throttled to approximately the same percentage as the least favoured branch (as described in paragraph B.2). Then proceed by balancing AB, AC, AD successively with AA.

In order to make a direct measurement of the air quantities at the branches AA to AD readings of duct velocity would be necessary. These are often difficult to perform because of limited access and are often inaccurate, because of the difficulties of choosing a suitable test point and errors associated with instruments suitable for use on site. An indirect measurement of air flow at the terminals is often more appropriate. Since the terminals on any branch have already been balanced we know that they handle the same fraction of their design air volume and that this corresponds with the fraction of design air volume that is passing into the branch from the main. This means that, instead of measuring, for example, the actual volume passing AB we could equally well measure the air volume passing at 6, 7, 8, 9 or 10 to determine this fraction. Normally the reference terminal for the branch (in this case 6) is chosen as measuring point. The procedure for balancing AA to AD is therefore performed by measuring 6 and balancing 6 with 1 by adjusting damper AB as required; next measuring 11 and balancing with 1 by adjusting damper AC; lastly measuring 16 and balancing with 1 by adjusting damper AD.

If it is inconvenient to use a terminal such as 6 as reference more than once or twice any other outlet in the group (e.g. 8) may be used as reference *after* (not before) it has been balanced (against 6) since the percentage at each of these terminals will then be equal. The reference terminal should not be changed more than once in a group to avoid introducing cumulative errors.

Where there are a large number of terminals on a branch, there will be a danger of error if the reference terminal only is measured to indicate the total flow at the branch. In such a case readings should be taken at two or more terminals together handling 10% of the total branch volume and the mean of their readings used to indicate the branch percentage.

When a terminal is being used as a reference point care must be taken that it is not an exceptional case that may be affected by variations in flow or turbulence caused by an obstruction upstream (such as a bend or a damper) and that consequently may not follow the proportional law. (See para. 4.3).

B.4 Routine for balancing a complete duct system

B.4.1 Preparation in the office

Design data that will be required by the test team must be prepared in the office to show design volumes throughout the system and the layout of the system.

Fig. 14 shows part of a line diagram showing the essential layout of a typical system. The main ducts are lettered and the terminals are numbered. This is a more convenient form

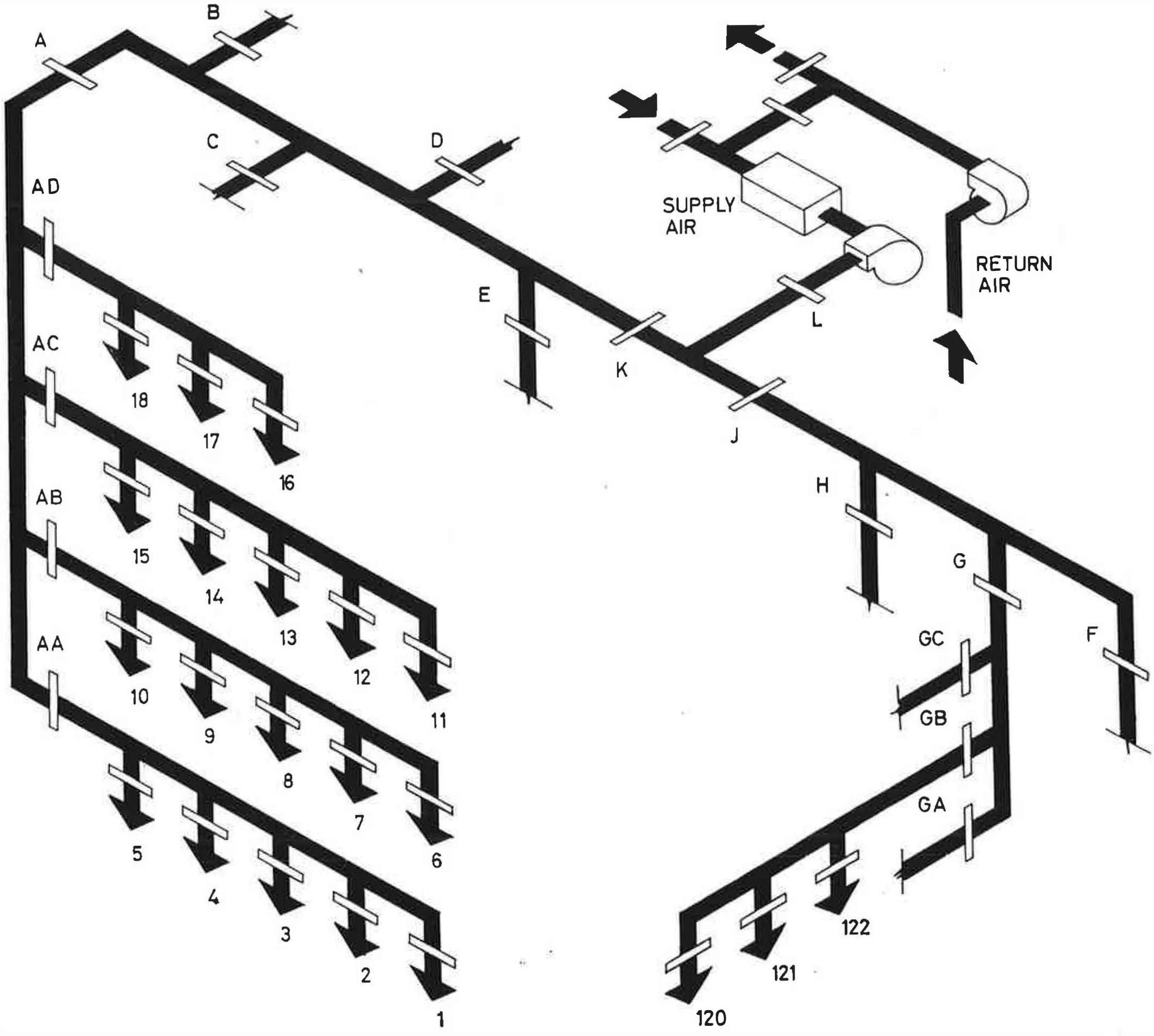


Fig. 14.—A typical system.

of drawing for the site than $\frac{1}{8}$in working drawings although the latter should also be made available to assist in the initial location of ducts, terminals and dampers. Design volumes for each group of terminals or branch ducts should be marked up on standard test sheets. (Typical test sheets are described in section 8.1).

B.4.2 Preparation of plant

Before the duct system can be balanced the plant must have reached a sufficiently advanced state of commissioning so far as the air-moving components are concerned. However, it is unnecessary for other items of plant, for example filters and cooler or heater coils, to be complete. The latter need not even be installed and it is certainly unnecessary for their automatic controls to be working. They must, of course, be in circuit in their working condition when the final fan volume is set. It may be necessary to compensate for the omission of plant items, by temporary blanking or throttling on the main fan damper, to prevent the fan motor tripping out on overload.

Any automatic plant dampers should be fixed in a constant position while the distribution ducting is balanced.

All necessary mechanical and electrical checks must be made on the plant before it is run. Points with an important bearing on the air balancing are:–

1. The plants must be air tight. For example, all access doors must be fitted securely. In air-conditioning plants drain connections must be examined to see that they are properly trapped and remain sealed with water.
2. The fan must be operating properly. For example, the bearings must be lubricated and the mountings and drive seen to be in order. Correct rotation must be checked. The fan motor ammeter is a useful check instrument and should be commissioned.

It is important that all of the above preparations be made beforehand and the plant conditions be unaltered during the balancing process. Any alterations made during the balancing, affecting the air supply to the system, will interfere with the setting of the group being balanced at the time.

B.4.3 *Preparation of duct system*

The duct system must be examined for continuity and air tightness. All access doors must be fixed. The system should be clean, blown through and checked for possible blockages.

All grilles, diffusers or other terminals must have been fitted.

During the inspection of the system all duct and terminal dampers must be opened.

Windows and doors to outside should be closed to minimize the effects of varying wind pressure and direction.

Where the plant to be balanced serves rooms which are normally also served by other systems (e.g. an extract system) then the other systems should also be running, even though not balanced, so that an artificial pressure or depression is not established in any room. Where a supply system is to be balanced which serves rooms where extract is by natural relief, then internal doors opening on to corridors or other rooms should be left open.

All of these preliminaries though somewhat tedious are important in themselves and also provide an opportunity for the test team to learn the layout of the installation.

B.4.4 *Initial fan volume*

The system fan volume must now be set approximately. The fan volume for test purposes is unimportant but should preferably be adjusted initially to around 100% to 120% of design. The only check that need normally be done is to ensure that the overload current is not exceeded, the main fan damper being adjusted accordingly against the fan motor ammeter.

B.4.5 *Balancing the system*

Attention can now be turned to the balancing itself. Note that it is pointless to take initial readings of branch or terminal volumes throughout the system before balancing begins. Preliminary readings are required only for each group as described for the balancing method. First the terminals are identified clearly into groups. These can be seen in Fig. 14. Thus numbers 1–5 and 6–10 etc., each form groups of terminals each of which must be balanced as a separate operation. Again branches must be identified into groups. Thus branches AA to AD form a group of branches which will be balanced as one operation. A similar group is found with GA, GB and GC and other such groups of course occur with the branches from the main ducts B to E and F and H. There are two groups of main branches, namely A, B, C, D, E and F, G, H. Lastly J and K will be balanced. After the system has been balanced the final operation will be to adjust the main fan damper a L for the design volume for the system.

Having identified the various groups of the system, the starting point for balancing must be settled. This will normally be at the terminal most remote from the fan, in our case terminal 1. The procedure then is to balance terminals 1–5 as described in para. B.2 above. This is followed by balancing terminals 6–10 followed by 11–15 and 16–18 in the same way.

Alternatively, it is not essential to start with terminals 1–5. It may happen, for example, that insufficient air is reaching these terminals to give a reading. A start could equally well have been made on any branch of the system, provided the terminal most remote from the fan were chosen for the starting point, preferably handling not less than 50% of its design volume, and not more than 150%. Thus terminals 11–15 could have been chosen and the other three groups of terminals on branch A dealt with subsequently in any order. The important thing to remember is that the group of branches AA, AB, AC and AD cannot be dealt with until all the terminals downstream have themselves been balanced in groups.

When this has been done AA, AB, AC and AD are balanced as described in para. B.2. above.

Working toward the fan, again it is important to delay the balancing of the main branches A, B, C, D, E until all the terminal groups and all the branch groups on these main ducts have been balanced in the same way as those on branch A. When this has been completed the main dampers, A, B, C, D, E can be balanced as a group.

In balancing the group A, B, C, D, E it may be possible to take duct readings at adjacent test points. Alternatively, since the system downstream of each main branch damper has already been balanced, it is usually more convenient to take representative readings at the previously selected terminal reference in the group. The average of these may be taken as representing the situation at the main branch. Thus, for example, in the balanced part of the system below A, measurements may be taken at terminals 1, 6, 11 and 16 expressed as a percentage of design volume and averaged to represent the percentage of design flowing at A. Indeed on small or regular systems it may be found that a single reading at one of these positions may be sufficient to indicate the situation at A.

In similar fashion the part of the system below J is balanced and finally J and K are balanced.

It will be observed that provided the balancing routine set out has been followed correctly there will be at the completion of the balancing operation at least one path, through the ducts to a terminal, having every damper fully open. This is the actual index circuit (which does not always correspond with the design index circuit).

With the duct system now in balance throughout we have the situation where all the branches and all the terminals will be passing the same fraction of their design volume within, of course, working limits of accuracy (discussed in section 7). The value of this fraction is not immediately known and is unimportant.

B.4.6 *Setting the system design volume*

The final operation in setting up the air flow in the system is to adjust the fan volume. This must be done after all the plant components in the air stream are installed and commissioned to function with their normal operating resistance. A careful measurement of fan volume is then made at a suitable test point near L by taking a traverse of measurements with a suitable pitot tube and inclined gauge. The main fan damper L is adjusted to the required volume. With this operation the volumes throughout the system and at all the terminals are varied in proportion and the terminal volumes are brought to their design value (again to the working limits discussed in section 7).

Following the final regulation of the main fan damper some spot check figures may be taken at selected terminals, but it is important to do this bearing in mind the considerations discussed in para. 8.2.

B.4.7 *Fixing damper settings*

To safeguard the balance of the system and the future proper operation of the installation, damper quadrants and operating levers should be marked as soon as each junction has been balanced and each damper set. This is best done by scribing rather than painting. When the complete system has been balanced the damper quadrants should be locked or pinned in their positions.

B.4.8 *Extract systems*

The above explanation is related to a supply air system.

Extract systems can be balanced in the same way using the proportional method. If the example above were an extract system, the dampers would be set in precisely the same order as for the supply system starting at an end junction and working back to the fan. The fact that the air flow is in the reverse direction does not affect the balancing procedure. In practice an extract system may be rather more sensitive to balance than a supply system, since there is seldom a large plant resistance associated with the extract fan and consequently changes in system resistance due to damper operation may tend to have a greater effect on air flow. It is also usually more difficult to measure accurately air volumes at extract grilles, for example, than supply grilles. In many cases, on the other hand, accurate setting of extract air volumes is less important than for supply volumes.

Appendix C

MEASUREMENT OF AIR FLOW IN DUCTS WITH THE PITOT TUBE

C.1

Methods of traversing ducts of circular or rectangular cross-section vary considerably. In every case the problem is to select a method which will give sufficient accuracy for the purpose in hand with a minimum expenditure of time and without calling for disproportionate precision in measurement of either position or velocity. The standards of precision which are appropriate in some circumstances may be quite unsuitable in others. For example, greater precision is plainly required in research or in laboratory work than for workshop use in industry. Measurements on a building site represent a case where the conditions under which the tester is required to work are often such that any kind of precision is very difficult indeed to attain. Fortunately the standard of precision usually necessary is not very high.

The purpose of the following discussion is to examine both the probable order or accuracy attainable on site without an extravagant expenditure of time and trouble and the methods necessary to achieve such standards.

C.2

BS1042: 1943 specifies (a) for circular ducts a traverse of 10 points (or more with disturbed flow) in equal annular areas, across each of two diameters at right angles, and (b) for rectangular ducts a reading at the centre of each of not less than sixteen rectangles into which the duct cross section is to be divided, plus additional readings in all the rectangles adjoining the walls of the duct when the total number of rectangles is less than one hundred. The number of rectangles is in all cases required to be an even number. (The purpose of this last requirement is obscure unless it be to avoid taking a velocity reading at the centre of the duct.) Four extra readings are required in each corner rectangle and two extra in each side rectangle. The effect of this is to make the total number of readings 48 in a duct divided into sixteen rectangles. These requirements both for circular and rectangular ducts appear to be intended for ducts of all sizes, the criterion for increasing the number of points of measurement being the irregularity of flow rather than the duct size.

In BS848 : Part 1 : 1963, 'Methods of testing fans', the gauging positions for circular ducts of all sizes are those for a six-point log-linear traverse, the traverse being made across two diameters at right angles for ducts up to 4ft dia. and across four diameters at 45° for ducts exceeding 4ft dia. For rectangular ducts of all sizes this standard prescribes a subdivision of the cross-section into sixteen geometrically similar rectangles with a centre reading to be taken in each plus extra readings in all edge and corner rectangles as called for in BS1043, i.e. a total of 48 readings in all.

Neither of these standards is specifically intended for site testing, the former being (it is presumed) of general application in laboratory work and where results of the utmost accuracy are required, while the latter is plainly aimed at manufacturers' bench-testing of fans.

C.3

Scott *et al.*[7] in a very interesting recent paper discuss the different methods of traversing circular and rectangular ducts and, *inter alia*, endorse the greater accuracy of the log-linear rule (as adopted in BS848: 1963) as against the tangential rule of the much earlier BS1042: 1943. It is believed that the Part II of BS1042 which is under revision will also adopt the log-linear rule. Another finding of that paper, however, throws considerable doubt on the value of the extra pitot-tube measurements in corner and side rectangles of a rectangular duct, pointing out that in a series of seven experiments in a duct subdivided into 25 rectangles the improvement in accuracy by making the 40 additional readings never exceeded 2% and averaged only 1·25%. A further comparison of the 25-point traverse of a rectangular duct with a 10-point log-linear traverse of a circular cross-section showed an average difference in 15 experiments of only 2·35%. Their conclusion is that the 25-point traverse of a rectangular duct can be recommended.

C.4

For site testing of ventilation systems it would be an absurdity to require greater precision than that prescribed for the manufacturers' workshop tests of the fans. Hence BS1042 with its 10-point traverse of circular ducts can be excluded, and a 6-point log-linear traverse may be regarded as the utmost accuracy that should be considered. For rectangular ducts the finding of Scott's paper suggests that a 25-point traverse (i.e. a single reading in each of 25 geometrically similar subdivisions) is likewise the extreme requirement. But it is also clear that in ventilation practice on both economic and practical grounds, greater precision of measurement is needed in large ducts when large rates of air flow are in question than in small ducts where perhaps 20% or 25% error is of small consequence.

It is therefore of interest to seek to make some assessment of the possible decline of accuracy when still fewer measuring stations are used. Also of interest is the question of the effect on accuracy of imprecise measurement of the location of test points.

C.5

With the object of making some direct comparisons on these two related questions, two careful plots of velocity distribution across pipes downstream of disturbances were made use of. These were published in a paper by Norman[10] and show the velocities in two 10 inch pipes, one downstream of a bend and the other downstream of a partly-open butterfly damper. Both these show a severely disturbed flow pattern, as may be seen from Fig. 15 where the velocity profiles across four diameters are reproduced.

Table IV below shows the result of measurements made on these diagrams. All the resulting average velocities are expressed as a percentage of the graphical integration of all four diametral traverses.

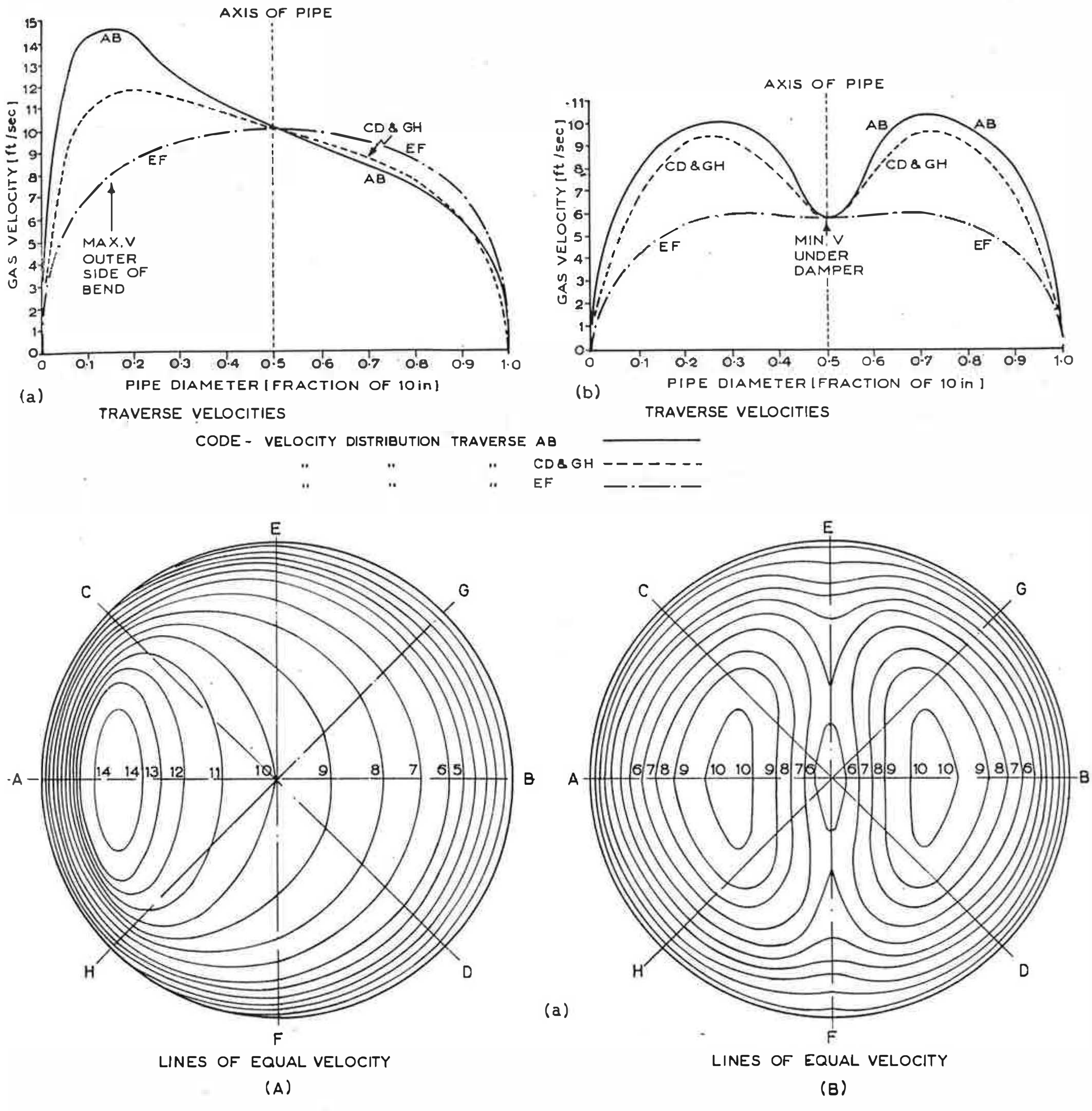


Fig. 15

The noteworthy feature of this result is that the results obtained with a log-linear 4-point traverse are only one or two per cent less accurate than the log-linear 6-point traverse which is required for fan testing according to BS848. Moreover taking only two readings per diameter (the Gauss-Sherwood 2-point traverse) produced results within $-2{\cdot}6\%$ and $+9{\cdot}4\%$ of the true value.

It seems, therefore, that for site use a 4-point log-linear traverse of circular ducts would give almost as great accuracy as an ostensibly much more precise survey, and except with violently disturbed flow might be expected to be well within $\pm 10\%$ of the truth. In cases of very disturbed flow traversing 4 diameters spaced at 45° would in most cases ensure the result being within $\pm 5\%$. For smaller ducts where a lower standard of accuracy is probably acceptable there seems no reason to attempt more than a 2-point traverse. And with the smallest duct sizes, given reasonably undisturbed flow, a single centre-point reading multiplied by 0·8 is all that is required.

Table IV—Percentage of mean velocity by various methods

Method of Making Traverse	Downstream of Bend		Downstream of butterfly damper	
	Traverse of AB/EF diameters	Traverse of all four diameters	Traverse of AB/EF diameters	Traverse of all four diameters
Graphical integration	100·4	100	93	100
Tangential 10-point	102·1	101·2	94·5	101·3
Log-linear 6-point	99·4	98·5	91·8	98·0
Log-linear 4-point	97·6	96·6	91·6	97·5
Gauss-Sherwood 2-point	109·3	109·4	97·4	105·5
Centre reading × 0·8	—	100·2	—	68·5

C.6

The comparisons in Table IV above and indeed all discussion of the relative merits of this or that method of traversing a duct with a pitot tube presuppose very precise location of the pitot tube at each point. But under site conditions it is more than doubtful if very precise location of a pitot tube is possible.

For example, with a log-linear 6-point traverse, the first three points have to be at the following distance from the duct-wall:

0·032D, 0·135D and 0·321D

or with a 10in dia. duct:

0·32in, 1·35in and 3·21in.

The best kind of pitot tube is one provided not with a fixed graduated scale on the stem but with movable spring clip markers which can be set to the position required. But even with the use of these, the location of the pitot tube (under site conditions) to within even 0·1in or so is far from easy. It can be shown that an error in placement of ± 0·1in can lead to variations of the order of ± 5% in the result with a six-point traverse. With a coarser (4 point) traverse the result is rather less sensitive to errors of location of the pitot tube. It is evident, therefore, that unless the pitot tube can be located with the necessary precision (which on site is not often possible) the advantage in accuracy of a 6- or 8-point traverse may be to some extent illusory.

Similar considerations apply to measurements in rectangular ducts. There is both less need for precision in small ducts than in large, and at the same time greater difficulty in locating the pitot tube with the required accuracy if the attempt is made to work to the same degree of precision.

C.7

With these considerations in mind, a proposed standard for traversing circular and rectangular ducts and for location of measuring points in relation to upstream disturbances is appended in the following diagrams.

Fig. 16 shows the recommended minimum distances for duct test holes downstream of various disturbances, while Fig. 17 gives details of the size and closure for a standard test hole.

Figs. 18 and 19 show the proposed location of measuring points in making a traverse in rectangular and circular ducts respectively. In both cases it will be seen that, in accordance

WHEN TESTS HAVE TO BE MADE DOWNSTREAM OF DISTURBANCES SUCH AS DAMPERS AND BENDS THEY SHOULD BE MADE AT THE POSITIONS SHOWN BY THE SKETCHES. WHERE THIS IS NOT POSSIBLE AND READINGS MUST BE TAKEN CLOSER TO A DISTURBANCE MORE READINGS MAY BE NEEDED. WHERE TESTS CAN BE MADE UPSTREAM OF DISTURBANCES THERE ARE NO LIMITING POSITIONS

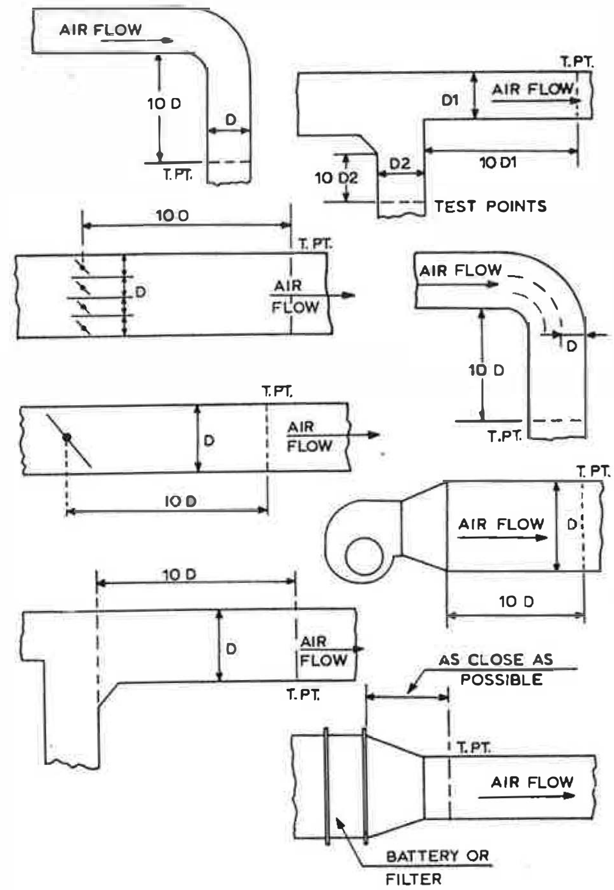


Fig. 16

with the principles stated above, a larger number of measuring points is required in large ducts than in small ones.

In all cases it is recommended that a rough preliminary traverse should be made to disclose whether the velocity distribution is very uneven or not.

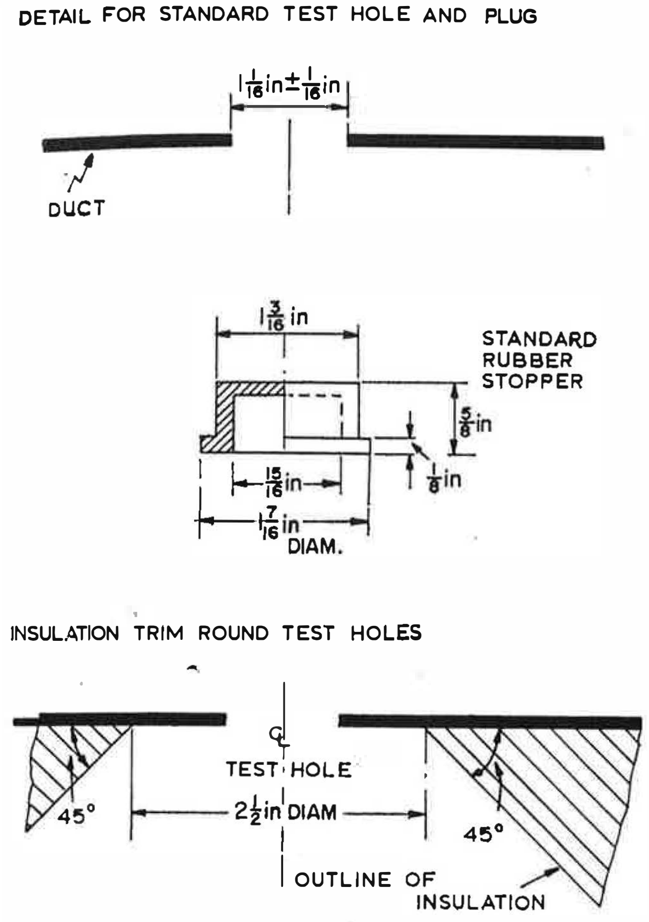
In cases where the maximum velocity is more than 1·5 times the centre-line velocity in the duct, or where the velocity pattern is in any other respect very uneven, it is recommended that more measuring points should be used, i.e. a 6-point linear traverse across 2 diameters, or, in extreme cases, across 4 diameters as in BS848: Part 1: 1963. From experience it seems unlikely that resort to still more measuring points or graphical integration is ever justifiable under site conditions. The theoretical increase in accuracy is not very great, and in practice may well be less than supposed for the various reasons discussed above.

Appendix D

BALANCING AIR FLOW IN INDUCTION SYSTEMS

D.1

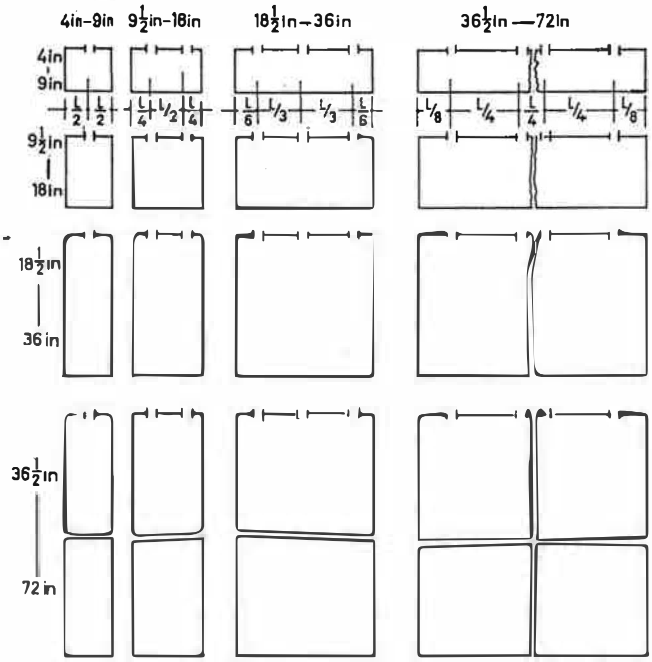
Many of the considerations governing the balancing of induction systems are the same as for low velocity air-ducting



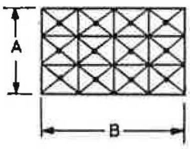
INSULATION TRIM ROUND TEST HOLES SHOULD HAVE ST STANDARD FINISH. FOR HOT DUCTS AND COLD DUCTS ABOVE AMBIENT DEW POINT THE TRIM NEED NOT NORMALLY BE FILLED IN. FOR COLD DUCTS BELOW AMBIENT DEW POINT THE TRIM SHOULD BE FILLED IN AFTER TESTING IS COMPLETE AND FINISHED WITH SUITABLE VAPOUR BARRIER

Fig. 17

STANDARD TEST HOLES REQUIRED IN ONE SIDE OF DUCT ONLY CHOOSE SHORTEST SIDE OF DUCT IF ACCESSIBLE. FOR DUCTS OVER 60 in. HOLES ALSO REQUIRED ON OPPOSITE SIDE OF DUCT



SUB-DIVIDE THE DUCT INTO A NUMBER (n) OF AREAS IN ACCORDANCE WITH THE TABLE AND TAKE A READING AT AT THE CENTRE OF EACH AREA



No. OF READINGS (n)

A↓ B→	9 in	$9\frac{1}{2}$in–18in	$18\frac{1}{2}$in–36in	>36in
9in	2	2	3	4
$9\frac{1}{2}$in–18in	2	2x2	2x3	2x4
$18\frac{1}{2}$in–36in	3	3x2	3x3	3x4
>36in	4	4x2	4x3	4x4

Fig. 18

systems. Familiarity with that method is needed to understand the basis of the method described here. This appendix is concerned only with aspects particular to induction systems.

In practice an induction system is more easily dealt with than a low velocity system. This is because the pressure loss at an induction unit is substantial in comparison with the loss in the distributing ducts and an alteration to one unit has only a very small influence on adjoining units. This has two results. Firstly, readings at the reference point need to be taken less often. Secondly, the method can be applied to systems without branch dampers.

The simplified method consists essentially of balancing the nozzle pressures (expressed as a percentage of design nozzle pressure) of batches of units against the most remote unit from the fan. This unit is used as a master for the whole operation.

In order to break the work down into manageable sections subsidiary reference units, themselves balanced with the master, are used for groups of the system.

D.2 *Balancing method*

Start with all unit dampers open. (If necessary in an occupied building the units may be throttled temporarily 'by ear' to limit noise.)

The fan should be set to give about design static pressure. The exact fan performance at this stage is not important and it is only necessary that the motor should not overload.

Successive moves in the balancing operation are shown diagrammatically in Fig. 20.

(1) The first step is to determine the initial pressure to which the master must be set. With a manometer make a preliminary traverse of nozzle pressures over the system. This need not cover every unit, but only sufficient typical units in each group (for example A, B, C, D, E) to find out which group is the least favoured and working at the lowest percentage of design pressure. Thence check this group to find the least favoured (or index) unit in the group. (Any isolated units showing a markedly lower pressure ratio than others should be discounted in determining the index unit and they should be checked for faults in installation.)

From the percentage of design pressure obtained above at the index unit decide the percentage of design to be applied to the master unit:

(a) If the index is on the same branch as the master, throttle the master to the *same* percentage of design as the index.

(b) If the index unit is on a branch closer to the fan selecting the initial pressure for the master is more a matter for judgment. If the master is throttled to the same percent-

DUCT SIZE (in)	No. OF POINTS ON EACH DIAMETER	LOCATION OF TEST POINT FROM DUCT WALL $\frac{y}{d}$ 1	2	3	4	5	6	No. OF DIAMETERS
<6 in	1(a)	0·5	—	—	—	—	—	—
6 in — 17½ in	2	0·12	0·83	—	—	—	—	2
18 in AND OVER	4	0·043	0·29	0·710	0·957	—	—	2
FOR DISTURBED FLOW IN ALL SIZES OVER 6 in	6	0·032	0·135	0·321	0·679	0·865	0·968	2 OR 4

(a) MULTIPLY SINGLE CENTRE READING BY 0·8 TO GIVE AVERAGE VELOCITY

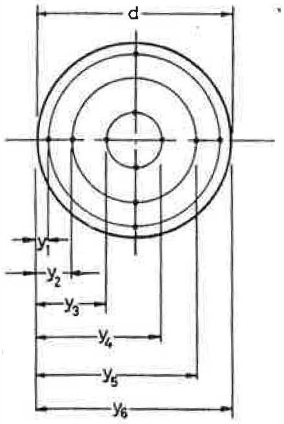


Fig. 19

age as the index this will be too much as throttling on units will ultimately increase the pressure at the index and this too will have to be throttled so adding unnecessary resistance to the fan. If the master is throttled too little, the index will remain below the master and the balancing will have to be repeated. As a guide first try throttling the master to a percentage mid-way between the index percentage and the approximate average percentage for units between the master and the index.

(2) Traverse the first batch of 10 units next to the master, working away from the master and adjusting the nozzle pressure at each unit in turn to give the same proportion of design pressure as set at the master. After each of the batch of 10 units has been set, take another pressure reading at the master. The pressure will probably have risen slightly because of some throttling on the other units. DO NOT reset its damper—the master and the first batch are now balanced—but record the new percentage of design at the master.

(3) Traverse the second batch of 10 units again, adjusting the nozzle pressure at each unit in turn to give the same proportion of design pressure just recorded at the master.

(4) Continue this routine until all units in group A have been balanced against the master.

(Note that the number of units in a batch dealt with between checks back to the master can vary above or below 10 according to the system. It is better to start with a small batch and then, if the master pressure is found to vary little as the units in the batch are regulated, the number of units in later batches can be judiciously increased.)

(5) After group A has been balanced, check the current design percentage at the master and set the end unit of group B to the same percentage. This unit now becomes the reference unit for group B.

(6) Traverse the first batch of units in group B adjusting the nozzle pressure at each in turn to the same design percentage as at the reference unit.

(7) Repeat for the second batch of group B balancing to the current design percentage at the reference unit.

(8) Continue until all units in group B are balanced with the group B reference unit.

(9) Check back to confirm that the group B reference unit and the master on group A show an equal design percentage (within the required limits).

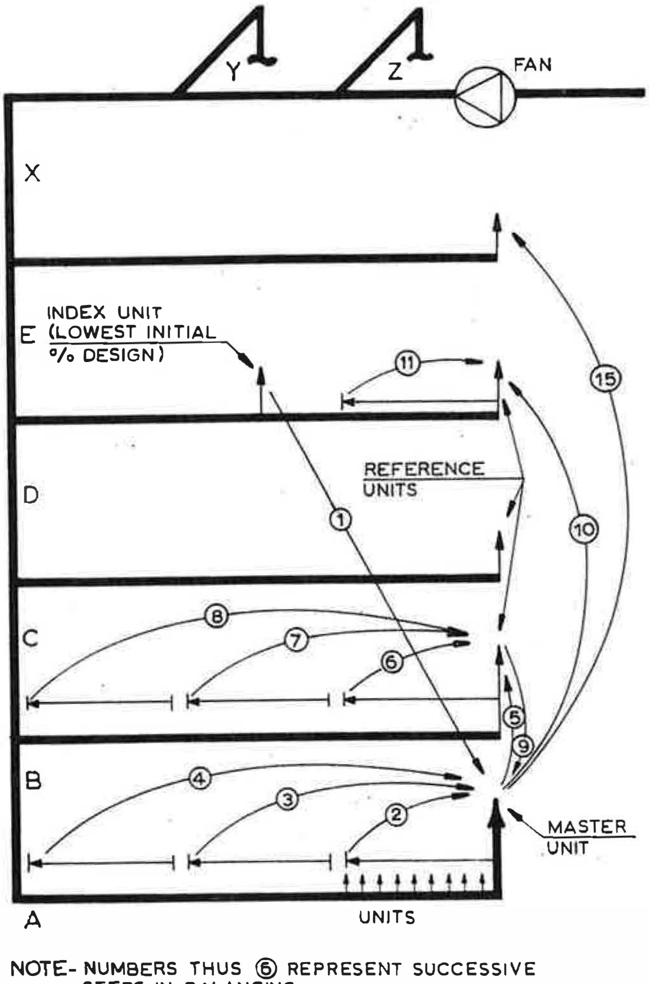
(10) Next set the end unit of group C (which becomes the reference for the group) to the current design percentage at the master (NOT the reference for group B).

(11) Balance batches of units in group C against the group C reference as before.

(12) Setting the reference unit for each group to the current design percentage at the master, complete the balancing for the whole system on the main duct X.

Where a system has a number of main ducts, e.g. X, Y and Z, each of these main ducts must be controlled by a damper. Each of the systems on X, Y and Z is then independently balanced with reference to its own master. Finally, the three are balanced with one another by adjusting the dampers at X, Y and Z to equalise the design percentage at each of the three masters.

Should the provision of dampers at X, Y and Z have been forgotten then the system on X can be balanced as described above, the master for X being set to a proportion midway between that of the *index unit for the entire system* and the average of units between the master and the index. Next a



NOTE- NUMBERS THUS ⑥ REPRESENT SUCCESSIVE STEPS IN BALANCING

Fig. 20

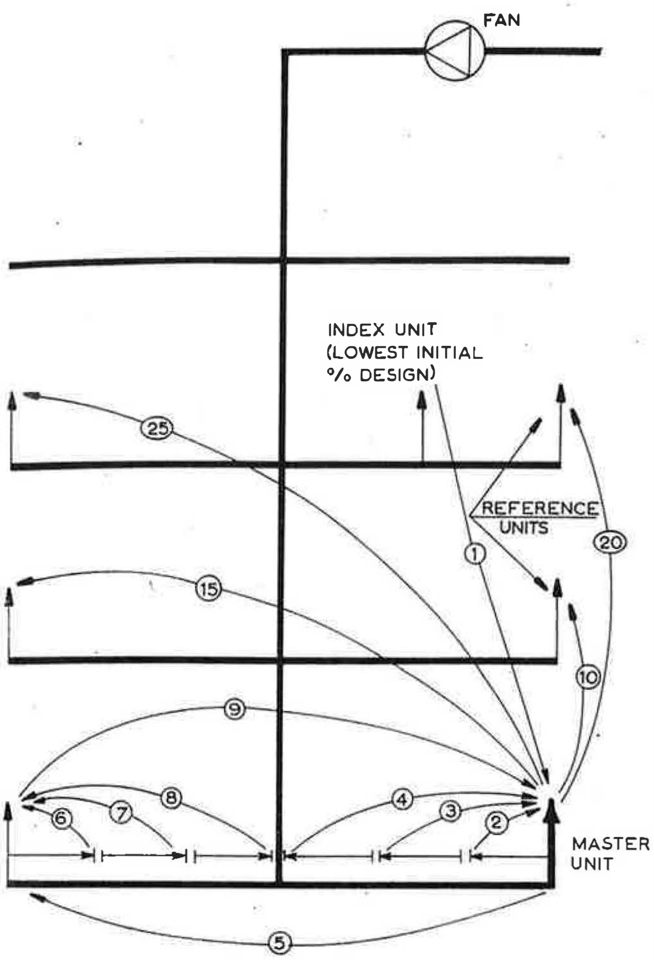


Fig. 21

sub-master for system 'Y' (its most remote unit) is balanced against the system 'X' master and system 'Y' is balanced with reference to the system 'Y' sub-master. Finally the sub-master for system 'Z' is balanced against the 'X' master and system 'Z' is balanced. This extension to three systems 'X', 'Y' and 'Z' is an approximate method and there is a definite possibility that after 'Y' has been balanced its sub-master may no longer correspond within the required limits to the 'X' master. If this is the case a second balance of 'Y' must be done.

For a similar reason if any of the group reference units fail to correspond within the limits with the master after the group has been balanced, then that group will have to be balanced again, repeating the same routine. This situation is, however, unlikely to occur with group balances.

Fig. 21 shows a diagram of a system having twin branches from the main duct. It shows the way to tackle this type—in fact, precisely as for a normal single branch system described above. The longest branch of a pair is first balanced. The second of the pair is then dealt with as if it were the next single branch above, using the end unit as reference after it has been balanced with the master. Proceed for all branches balancing the reference for each branch of a twin in turn against the master.

After all units on the system have been balanced to the same proportion of design nozzle pressure, within the specified limits, the fan performance is finally set up. This should be done accurately when all plant items are in circuit. First set the fan performance approximately by adjusting the main supply damper to give the design static pressure at fan discharge. Finally adjust the fan volume to design (within agreed working limits), having regard to the condition of the filter, by measuring duct velocity with a pitot tube and inclined gauge. Check and record the fan static pressure and the fan speed and motor current.

As a final check a few key readings of unit nozzle pressure should be taken.

D.3 Limits

Limits for balancing must be specified to the test engineer by the designer. Normal practice may be regarded as requiring $\pm 10\%$ between outlets on a system. This is a close requirement on systems not fitted with dampers on main ducts when $\pm 15\%$ may be acceptable in the final result. Individual damper settings during balancing should always be done to at least the nearest 0·05in w.g.

Acknowledgements

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Résume

COMMENT STABILISER LA CIRCULATION D'AIR DANS LES CONDUITS DES SYSTEMES DE VENTILATION

Cette étude traite des procédés de réglage ou de stabilisation d'un système de ventilation, assurant que chaque orifice fournisse ou extraie la quantité d'air convenable.

Cette communication étudie les méthodes, l'organisation, les instruments employés et l'enregistrement des résultats; elle examine également le degré de précision nécessaire et possible dans l'usage courant. La méthode recommandée pour régler le système consiste à égaliser les débits d'air à l'embranchement du système, avec ordre, suivi d'un ajustement du système complet jusqu'à obtenir le volume nominal grâce à un dernier réglage du ventilateur. L'étude principale traite des systèmes à vitesse peu élevée, mais elle couvre également l'application de la méthode aux systèmes à vitesse élevée.

La communication étudie certains aspects des projets d'installation de ventilation, et de canalisations d'air nécessaires au réglage facile des systèmes afin d'obtenir le rendement prévu.