THESIS FOR THE DEGREE OF PH.D.

(1) THE PROPELLER TYPE OF FAN AND ITS APPLICATION TO THE VENTILATION OF MINES.

(2) AN INVESTIGATION OF:

(a) THE RELATIONSHIP BETWEEN PRESSURE AND VOLUME IN THE FLOW OF AIR IN MINES;
(b) THE EFFECT OF NATURAL VENTILATION;
and (c) THE RESISTANCE OF MINE VENTILATORS.

by

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General Terms and Definitions.

Figure 9 gives, in several views, a general idea of an ordinary two-bladed air-screw. F shows what is generally called the projected plan form of the air-screw. G is a side elevation, and H gives sections through the blade at AA, BB, etc. The various terms in air-screw design and manufacture are here briefly explained.

**The Boss.** - The central portion of the air-screw from which the blades project on either side.

**The Centre Bore.** The hole through the centre of the boss.

**The Root.** A general term applied to that portion of the blade adjacent to the boss.

**The Tip of the Blade.** The point of the blade most distant from the centre.

**The Radius.** - The distance from the tip of the blade to the centre of the axis of rotation is termed the radius of the air-screw.

**The Diameter (D).** - The distance from tip to tip.

**The Blade Angle.** - At any section the angle made by the face of that section to the plane of rotation.

**The Leading and Trailing Edges.** are self-explanatory in terms of the direction of rotation.

The cambered side of the blade is the "back", and the flat side is termed the "working face".

**The Pitch of a Propeller.** - A propeller may be regarded as a screw which for every complete turn travels forward a certain distance. In our case the air will take the place of the nut, while the air-screw will represent the bolt. The air-screw is rotating at \( n \) revolutions per second and the air travelling with a velocity of \( V \) feet per second. Due to these two motions the path traced out by the air at any point will be a helix on the cylindrical surface whose radius is \( r \). Fig. 1; or if we imagine this surface to be laid out flat, the path of this point will appear as \( \text{AB} \), Fig. 2.

\[
\text{OA} = 2\pi r
\]

\[
\text{OB} = \text{the pitch} = 2\pi r \tan \theta
\]

If
Figure 3
Figure 3
If all sections have the same pitch equal to \( OB \), the faces of the sections will lie along these respective helicoidal paths as shown in Fig. 5.

**Tip Speed.** - The tip speed of an air-screw is the distance travelled in unit time in the plane of rotation by the tip of the air-screw.

Therefore tip speed = \( 2 \pi Rn \).
INTRODUCTION.

Mr. Samuel Hare, in his Presidential Address to the North of England Institute of Mining and Mechanical Engineers on 13th October, 1923, referring to the ventilation of mines in this country, said:

"With regard to the ventilation of our mines, however, I should like to see more progress reported. While the general ventilating arrangements are quite satisfactory, I am of the opinion that sufficient interest is not taken in connection with the economic side of the question."

"I have been astonished at some of the results obtained on the testing of fans, having met with instances where the efficiency has been as low as 20 per cent., and in all cases of this kind I have found that the fan has been much too large for the work/"
"work it was required to perform?"

"It is obvious that the 'equivalent orifice' of a mine must vary, sometimes to a small but occasionally to a large extent, both as the mine develops and as it is being exhausted; and as a fan, to be efficient, must be designed for a certain equivalent orifice, it naturally follows that it must be inefficient when working on an equivalent orifice which differs from the original. I hope it is not too much to expect that the makers of fans may some day be able to design a fan which will be so constructed that it may be regulated to work efficiently on different equivalent orifices."

Again, in the discussion of the paper by Professor Henry Briggs and Mr. James N. Williamson on "Experiments on the Distribution of Air in Centrifugal Fans and on Re-Entry Phenomena", the same gentleman further remarked:-

"It has been necessary very often to put in smaller fans, and my contention is that instead of having to go to the expense of substituting another fan it should be possible to regulate the fan to suit a different orifice from that for which the fan was first designed."

The above remarks are convincing proof of the inefficiency of the Centrifugal type of fan as a ventilator working under actual mine conditions, in spite of the high efficiencies claimed by most fan makers. In the choice of a suitable ventilating machine for a mine consideration must be given to:

(1) the resistance of the mine against which the fan will operate, and how this resistance will probably vary during the lifetime of the colliery;

and (2) the behaviour of the fan throughout any change or modification in the resistance of the mine.
Figure 4. - Variation in Efficiency of Fan with Mine Resistance.
With regard to (1), it has been generally assumed that the resistance of a mine increases as the workings extend and there is no doubt that in many cases this assumption is justified. The most recent work on the problem indicates, however, that, under modern conditions, the mine resistance will be a maximum during opening out and that, owing to increase in the number of splits, enlargement and duplication of airways, it falls off after full development has been reached.

In connection with the enlargement of airways, etc., Mr. R. Clive has shown that the total cost of ventilation at Bentley Colliery, Yorkshire, worked out at £6,734 per annum, but if the equivalent orifice of the mine had been 50 square feet (1.1 atkinson approx.) instead of 77.5 square feet (0.5 atkinson approx.), the volume remaining the same, the annual cost for power would be £16,170, i.e. a saving of nearly £10,000 per annum, apart from the increased capital cost required for the larger plant, has been effected.

Centrifugal fans, as at present designed, are only capable of working at their highest efficiency against a definite mine resistance. The ratio of the mine resistance to the fan resistance must have a definite value, which will vary slightly for different types of fans. This value is independent of the volume of air passed or the speed of the fan. Figure 4 shows how the efficiency of a fan varies with the resistance of the mine. It will be seen that if this resistance rises above the value OB or falls below the value OA the fan will then operate below a reasonable efficiency. As the size of

---

the fan installed is generally such that it will be most efficient when working against the resistance of the mine at full development, then as regards (2) the fan will operate at a much lower efficiency during opening out and during the final stages of its life.

Various methods have been suggested for regulating this resistance so that the fan will always operate at its highest efficiency. For instance, Dr. J. Parker has suggested that the effect of a varying mine resistance could be counterbalanced by running two fans in parallel where the resistance materially decreases or in series when it increases. Another suggestion of Dr. Parker's is that if a centrifugal fan could be designed with a flat-topped efficiency curve, that is one which would operate against a varying resistance with little or no change in its efficiency, then the difficulty presented by (2) would be overcome and the expenditure considerably reduced.

There is, however, another direction in which we may find a solution of the problem raised by Mr. Hare's remarks, and that lies in the application of Propellers or Air-screws placed in series on the same shaft, instead of improving the present type of centrifugal fan. Although the application of propellers to the ventilation of mines probably dates back to the earliest attempts to ventilate mines by mechanical means, this type of fan has never come into general use, chiefly because, as generally constructed, it is incapable of creating the pressures needed to circulate the large volumes of air required to ventilate modern mines. It would appear, however, from the work of Mr. F. A. Steart, South 1.

South Africa, that the chief reason for this lies in the method of application rather than in the principle of the propeller itself. From the result of his research the propeller fan appears to have a number of special features which should enable it to become a serious rival to the present form of mine ventilator and may ultimately oust it from the field. A résumé of Steart's work and contentions is given in a later section.

The writer's investigations have been concerned with the application of air-screws to the ventilation of mines with particular reference to their efficiency; Part III contains the record of this work together with conclusions arising therefrom. Part I is devoted to the theory of the machine, while in Part II a résumé of recent work connected with our subject is attempted.

At the outset, I should like to record my indebtedness to Professor Henry Briggs, under whose direction and supervision the experimental work described was carried out. He afforded me every facility in the performance of the work, and his interest, criticism, and advice have been most helpful. My thanks are also due to various members of the staffs of the Engineering Departments of Heriot-Watt College for practical assistance in many ways.
PART I.

THE PROPELLER VENTILATING FAN IN THEORY AND PRACTICE.
Figure 5.- Velocity Diagram for Propeller Fan.
PART I.

THE THEORY of the PROPELLER VENTILATING FAN.

The propeller is the simplest type of fan; it requires neither diffuser nor volute, although it may be provided with an evasee. Its complete theory, however, is extremely complicated, mainly because each particle of air does not keep to a cylinder concentric with the axis of the fan. In the following approximate theory we shall suppose that each particle of air moves on a cylindrical surface and that the axial component of inflow is the same as that of outflow. In Fig. 5 is shown a section of a blade, A2. $\phi$ is the relative angle of inflow, and $\theta$ that of outflow, assuming that the angle of flow coincides with the angle of the vane. In the diagram the axial velocity is represented by $U$, and the peripheral velocity by $V$. From the triangle of velocities at outflow, $a$, will be the component of the absolute velocity perpendicular to both radius and axis, or in other words, the velocity of whirl, since

$$a = KG - HG = V - U \cot \alpha \ldots \ldots \ldots \ldots \ldots \ldots (1)$$

WORK done on the Air in its Passage through a Propeller Fan.

One of the most important of the mechanical laws that applies to a fan is that the change of moment of the momentum of a mass acted upon by forces is equal to the moment of the impulse of the external forces, or to their angular impulse.

If a cubic foot of fluid be delivered by the machine per second and the weight of one cubic foot be \( w \) lbs., then the mass delivered per second = \( \frac{qw}{g} \). Take any point \( P \) (Fig. 5) located on a blade at a distance \( r \) from the centre of the fan; let its axial and circumferential velocities be represented by \( u \) and \( v \) respectively, and the angle of the blade at this point be \( \phi \). From equation (1) we have:

\[
\text{Velocity of air at } P = v - u \cot \phi
\]

\[
\text{Momentum of air at } P = \frac{qw}{g} (v - u \cot \phi)
\]

where \( g \) = the gravitational constant.

\[
\text{Moment of Momentum about the fan centre} = \frac{qwR}{g} (v - u \cot \phi)
\]

\[
\text{Total Moment of Momentum about the fan centre} = \frac{qwR}{g} (v - u \cot \phi)
\]

\[
\text{Moment of Resultant Momentum} = \frac{qwR}{g} (V - U \cot \phi)
\]

\[
\text{Angular Velocity} = \frac{qWR}{g} (V - U \cot \phi) \omega
\]

And since \( \omega = 2 \pi n \) and \( V = 2 \pi R n \), where \( n \) = number of revolutions of the fan per second, we have:

\[
\text{Work done on Air by Fan} = \frac{qw}{g} (V^2 - UV \cot \phi) \text{ ft.lbs/sec.}
\]

**The Theoretical Depression.** - (H)

In the perfect propeller fan, the whole of the energy evaluated by equation (2) would be utilised in doing useful work, it would all be expended in giving pressure energy to the air. Hence the fan would support an air column, \( H \) feet in height, equivalent to a pressure of \( H w \) pounds per square foot, and when it is delivering a cubic foot of air per second we have:

\[
\text{Work done on Air by Fan} = Hqw \text{ ft. lbf/sec}.
\]

Therefore from equations (2) and (3),

\[
Hqw = \frac{qw}{g} (V^2 - UV \cot \phi)
\]

or

\[
H = \frac{(V^2 - UV \cot \phi)}{g}
\]

From
8.

From this formula the so-called theoretical water-gauge can be calculated by simply converting the units of $H$ (i.e. feet of air column) into inches of water column, thus:

$$\text{Theoretical water-gauge} = \frac{12HW}{W_1} \text{ inches}$$

where $W_1 =$ weight of a cubic foot of water, or taking $W_1 = 62.4 \text{ lbs}$, the theoretical water gauge $G = \frac{HW}{5.2}$ \hspace{1cm} (5)

From equation (4) it will be observed that the value of $H$ is affected by the quantity of air passing through the fan. In this respect the propeller fan is similar to the centrifugal ventilator with backward trending blades. With both these types of fans the depression will decrease as the quantity of air which the fan is delivering increases. This feature of the propeller fan is illustrated by the characteristic curves given in Figures 23 to 26.

In the foregoing discussion, we considered the case where the theoretical depression, $H$ feet of air column, was created when delivering a volume of $g$ cubic feet per second. An actual propeller fan, however, when delivering the same volume, would produce a depression less than $H$, due to imperfections in its design. Let $H_A$ be the actual depression measured in the fan drift by the water-gauge. Then the connection between $H_A$ and the effective depression, $h$, is:

$$H_A = \frac{5.2 h}{W} \hspace{1cm} (6)$$

The loss of depression or head $H - H_A$ is an overall assessment of the imperfections in the design of the propeller.

---

1. "Water-gauge" is the routine method of measuring depression; the Ventilation of Mines Committee recommend that depressions be stated in "pounds per square foot".
Figure 6. - Flow Diagram of Power.
Fan Efficiencies.

By the efficiency of a machine is meant the ratio of the output to the input. It is usually expressed as a fraction of unity or as a percentage, i.e., it is the numerical value of the effectiveness. In Figure 6, what may be termed the "Flow Diagram of Power" is given, together with the various losses involved. The efficiencies which concern us most are:

(a) \( \frac{B}{D} = \frac{\text{Horse-Power in Air in Fan-Drift}}{\text{Horse-Power entering Fan}} = \text{Manometrical Efficiency.} \)

(b) \( \frac{B}{C} = \frac{\text{Horse-Power in Air in Fan-Drift}}{\text{Horse-Power entering Fan Shaft}} = \text{Mechanical Efficiency.} \)

(\text{and}) (c) \( \frac{B}{A} = \frac{\text{Horse-Power in Air in Fan-Drift}}{\text{E.H.P. of motor or I.H.P. of Engine}} = \text{Overall Efficiency of "Useful Effec.} \)

In order to evaluate the "Horse-Power in Air in Fan-Drift", i.e., the total energy possessed by the air at that part of the ventilating system, we require to determine:

(1) the effective depression

and (2) the volume passing.

In connection with these two measurements much controversial matter has been written and much otherwise careful experimental work rendered practically valueless for the lack of recognised standard methods. It is to be hoped that the Committee set up in 1923 by the Council of the Institution of Mining Engineers for the purpose of revising the existing theory of mine ventilation will issue a decision on this important question.
Figure 7. - Forms of Pressure-gauge Tubes in Fan Drift.
MEASUREMENT of the EFFECTIVE DEPRESSION.

The usual method of obtaining the effective depression, as distinct from the theoretical depression, is by means of the well-known water-gauge, which measures the "head" in inches of water-column. The committee referred to above recommend that this method of expressing the "head" be abandoned and replaced by one of "pounds per square foot".

There are three distinct forms of "pressure head" which can be measured in the air flowing along the fan-drift, viz:-

(i) **Velocity "Head".** This is the "pressure head" required to accelerate a mass of air from a state of rest to its final velocity - the kinetic energy possessed by the air.

(ii) **Static "Head".** This "head" overcomes the resistance to air flow and is the pressure which would be measured by the difference between two barometers, one placed inside the fan drift, and the other in the external atmosphere.

(iii) **Dynamic or Total "Head".** This is the pressure required to overcome the resistance to flow, and to create the velocity of flow. It is the algebraic sum of (i) and (ii).

We are concerned with the measurement of this last "head". Numerous otherwise useful fan tests have been spoilt by taking the depression incorrectly. Most engineers now realize, however, that the end of the gauge tube should be pointing directly to windward as in Figure 7 (a); otherwise the depression or pressure registered becomes affected by the velocity of the air. The static pressure may be measured by form (b) or by (c) provided, /

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provided, in the latter case, the opposite walls of the
fan drift are parallel and smooth-sided, especially on
the upstream side of the tube. The forms shown by (d) and (e) do not measure the static gauge correctly; both exaggerate the gauge reading with a suction fan and reduce it with a forcing fan. Form (c) is the one favoured by makers of exhaustive fans for obvious reasons, and is the form generally in use at mines. For instance, in fan drifts where the velocity is very high, exaggerated depressions will be registered by this form of gauge tube.

In the discussion on a recent paper by Mr. R. Clive, Professor H. Briggs stated "that if forcing fans had been more frequently adopted as main ventilators in this country, the mistake of using the static gauge readings would have been less common, for by sufficiently restricting the delivery-passage of such a fan it is not difficult to get a negative (apparent) efficiency."

The shape or size of the dynamic or impact tube (a) does not matter as long as its axis is parallel to the axis of flow. A small error in setting the direction of the tube has no appreciable influence.

While the static "pressure" is uniform over any cross-sectional area, the dynamic "pressure" is, due to difference in velocity at various points of the cross-section, very variable. The position of the impact tube in the fan-drift is, therefore, all important.

Professor H. Briggs and Dr. J. N. Williamson have found the position inside the drift at which the tube will register a mean value for any cross-section to be at a point one-seventh of the width of the duct or drift measured from either side, along a central axis.

Regarding /

2. Vide E.A. Griffiths in "Engineering Instruments and Meters" (1920) p. 96.
12.

Regarding the best form of manometer for use, there are varied opinions. The ordinary vertical U-tube in use at mines, while being good enough for comparative purposes, such as checking the daily performance of the fan, is hopeless for accurate determination. Several micromanometers, on the differential principle, have been introduced from time to time, such as the Cathetometer but these are chiefly for laboratory purposes. One of the latest of this class of instrument is the Wahlen Micromanometer for which is claimed a sensitivity as high as 0.0001 inch of water column. A more recent form is that developed by Mr. J.L. Hodgson and Prof. D. Hay which consists essentially of a combination of the U-tube and inclined manometers reading on a null method. The instrument is filled with paraffin oil to the centre of the inclined tube, where there is a zero mark, the meniscus being read from above either by a microscope fitted with cross hairs or by means of a zero pointer, while axial illumination is provided from below by an electric cap lamp. It is claimed that the instrument has an accuracy of 1 in 50,000.

The manometer used in all our experimental work in the Mining Laboratory of this University and at all the colliery tests consisted of an inclined glass U-tube, 5 feet long, ½ inch in diameter and graduated in tenths of an inch. Instead of water, petrol was used (sp. gr., 0.758), which, besides keeping the inside of the tube clean and giving a good meniscus, added to the sensitivity of the instrument. It was inclined at an angle of 7°35' which gave a multiplication factor of 0.1 when converting the inclined observations into the desired vertical measurements. The inertia of the comparatively large volume of liquid greatly assisted in minimising the oscillatory

oscillatory effects due to fluctuating flow. With this form of instrument it is easy to read, accurately, pressure "heads" as low as 0.002 inch of water column (i.e. 0.0104 pound pressure per square foot).

In the discussion of the First and Second Reports of the Midland Institute on the Ventilation of Mines, Professor Henry Briggs, University of Edinburgh, emphasised the necessity of an early report upon "the all important question of taking water-gauge readings". He contended, and rightly so, that the settlement of so important a point should be a primary consideration.

MEASUREMENT OF RATE OF FLOW OF AIR.

The volume of air flowing through a duct or gallery is usually stated in cubic feet per minute or thousands of cubic feet per minute or thousands of cubic feet per second. In present practice, to obtain the volume or quantity, two measurements are necessary, namely, the velocity of flow and the cross-sectional area of the gallery. Care should be taken in the selection of a suitable place for the measurement of the velocity, and such a place should be straight, parallel and smooth-sided and not too near the fan.

There are two instruments in use for measuring the velocity of flow, namely, the anemometer and the Pitot tube. The former, which is more often used than the Pitot tube, has often been adversely criticised and, indeed, there is at present an outcry for its abandonment. However, the ordinary wind-vane anemometer will give fairly accurate results provided sufficient attention is given to certain conditions. These conditions are:

(i) that the instrument has been recently calibrated;
(ii) that the velocity of flow lies between 200 and 1100 feet per minute;
(iii)/

(iii) that nearly streamline flow exists at the point of measurement - neither turbulent nor pulsating;

(iv) that the instrument is kept normal to the axis of flow and either a slow and uniform rate of motion in a zig-zag fashion be adopted or the method of approximately uniform sectional measurements employed;

(v) that several readings be taken.

The limits given in (ii) above refer to the present type of anemometer. Recently, however, Mr. E. Ower of the National Physical Laboratory, has designed a vane anemometer, which, he claims, will give satisfactory and consistent results at speeds down to about 0.6 feet per second or 36 feet per minute.

Anemometers are simply integrating machines and record the mean velocity of flow over the period of time chosen. They are not very robust and have, therefore, to be carefully handled. When in constant use their accuracy decreases, hence the need for frequent calibration. Nevertheless, provided the conditions stated above are given sufficient attention, the mean value of several readings cannot be far wrong. A 2¼ inch zero-setting anemometer and a 2½ inch Davis anemometer were satisfactorily used in the experimental work subsequently to be described, both instruments being frequently calibrated by means of the "anemometer table" in the Mining Laboratory of this University.

The Pitot tube as a reliable instrument for measuring velocity air in a drift is, like the anemometer, not without its critics. Many forms of this tube have been used, but the one most favoured, since it has unity factor, is that produced by the Cambridge Scientific Instrument Company. One serious drawback of this form of tube where the static holes are very small in diameter (usually 1/50 inch) is that, where there is moisture in the air, a film of water forms over the holes and renders the/

the instrument useless. This has been observed by various workers and was noticed also during our own experimental work at collieries. Moreover, this form of Pitot tube gives a very small "head" at low velocities, one inch of water column being equivalent to a velocity of approximately 4000 feet per minute, and is, indeed, really a laboratory instrument. Recently, however, Mr. J. L. Hodgson developed a modification of the standard Pitot tip, consisting of a cylinder bored with pressure holes on its upstream and downstream faces. It is claimed that the differential pressure produced by this form of Pitot tip follows the square law down to air velocities of 2 or 3 inches per second. As the suction as well as the impact effect is used, a good magnification is obtained. The gauge is held in front of the observer on the end of a stick. Barometric and hygrometric observations in the gallery are necessary with this method of measurement, in order that the weight of a cubic foot of air may be obtained. The whirling hygrometer, such as Storrows, should be used in this connection. The Pitot tube method thus involves much laborious calculation where several readings of the gauge are necessary, although tables covering a practical range of velocity "heads" and atmospheric conditions could be compiled to eliminate this. The position of the Pitot tube inside the gallery, so as to record the mean velocity, has been theoretically proved to be one-eighth of the diameter or width of the gallery measured.

measured from either side. This position, however, does not agree with that determined by the Prussian Firedamp Commission in 1884; the position given by that body being one-sixth of the diameter measured from either side.

Hot-wire anemometers, or electrical velocity meters - which seemed at one time promising - have proved too expensive, and as at present constructed are unreliable at low velocities. There is, however, at least one direct-reading hot-wire device for the measurement of the velocity of flow. This is the instrument designed by Professor MacGregor Morris of East London College. The Committee on the Control of Atmospheric Conditions in Hot and Deep Mines are at present trying out one of these instruments and are likely to issue a report thereon very soon. It is claimed that this instrument, in its most recent form, can record a velocity of 20 feet per minute with only a 4 per cent. error.

The modern desire is for an accurate form of flow-meter which will replace the instruments just discussed. The introduction of such a meter which has a high degree of accuracy and is suitable for mining work cannot come too soon.

With regard to the remaining measurements required before the three efficiencies referred to can be determined, there is little or no conflict of opinion. To estimate the manometric efficiency, it is necessary to know the weight of unit volume of the air passing through the fan. This is determined in the manner indicated.

---

2 See "The Fan" by C. H. Innes, p. 93.
indicated when discussing the Pitot tube. It is also necessary to know the speed of the fan and this is best ascertained by means of a tachometer, especially with modern quick-running fans. In determining the Mechanical Efficiency we require to know the horse-power entering the fan shaft. This presents a difficult practical problem. In the laboratory, this horse-power can be got by cutting the fan shaft in two and measuring the torsional effort on the shaft. In practice, however, we have to be content with a brake-horse-power measurement and this should be taken so as to eliminate all losses between the prime-mover and the fan shaft. If the fan is electrically driven the power input to the motor is easily obtained; if steam driven, the power input must be determined by some reliable form of indicator.

**Manometrical Efficiency.**

It is now generally agreed that the manometrical efficiency of a fan is the ratio of the useful to the theoretical depression, or

\[
\text{Manometrical Efficiency} = \frac{\text{Actual Depression Realised}}{\text{Theoretical Depression}}
\]

\[
M = \frac{h}{G} \quad \text{...............(7)}
\]

Converting the numerator and the denominator into horse-powers, we can re-write equation (7) as under:

\[
M = \frac{h \times 5.2 \times 9}{550}
\]

\[
= \frac{G \times 5.2 \times 9}{550}
\]

\[
= \text{Horse-Power in Air in Fan-Drift} \quad \text{Horse-Power entering the Fan.}
\]

The manometrical efficiency is thus seen to conform to the accepted idea of an efficiency. It is the ratio of power given out to power put in. By the phrase "horse-power entering the fan" is meant the net power actually received by that machine without consideration of/
of any external loss of power; it is the power that would be recorded by, say, a torsion dynamometer on the fan shaft less the power lost by friction at the fan bearings. This latter, in most cases, is generally very small indeed.

The manometrical efficiency may also be expressed as a ratio of resistances (in Atkinsons). Professor Briggs in a recent article deals with the manometrical ratio as a means of determining the relation between the resistances, in Atkinsons, of the mine and fan. He expresses the manometrical efficiency,

$$M = \frac{R}{R + r} \ldots \ldots \ldots \ldots (8)$$

where $R$ is the mine resistance in Atkinsons and $r$ the resistance of the fan in the same units.

From equation (8) it is obviously desirable to have a fan with a small resistance in comparison with that of the mine, and a high manometrical efficiency is the best assurance we can possess in this respect.

**Mechanical Efficiency.**

This efficiency is frequently confused with the efficiency of the whole plant. The mechanical efficiency of a fan is measured by the ratio of the useful work done in circulating the air in the mine to the horse power supplied to the fan shaft. This latter is the power that would be recorded by a torsion dynamometer on the fan shaft. As already pointed out, this power is difficult to measure in practice. However, the only difference between the mechanical and manometrical efficiencies will be/  

1. The "Atkinson" is defined as that resistance which absorbs a pressure of 1 lb./per square foot when a volume of 1 Kilolusec of dry air at 60 degrees Fahr. and 30 in. barometer is flowing.  
be due to losses at the fan shaft bearings.

Overall Efficiency.

This is the ratio between the horse-power put into the air and that supplied to the engine or motor. It includes losses in the drive, bearings, and other places, thus giving the "Useful Effect" of the ventilating system, and is of the utmost economical importance.

During the period of my research, I assessed the overall efficiency of a Sirocco propeller fan installed under normal working conditions, during a period in summer when natural ventilation was practically negligible. The overall efficiency at 480 revolutions per minute (normal speed of fan) was only 15.3 per cent., while at 522 revolutions the efficiency rose to 25 per cent.

Aerodynamical Efficiency.

Those who advocate this efficiency seek to differentiate between losses due to bad design and blade slip on the one hand and losses due to the internal friction of the fan on the other. If \( G \) be the theoretical depression produced by the perfect fan, and \( h_a \) that which it does produce, neglecting internal frictional losses, then

\[
\text{Manometrical Efficiency} = \frac{h_a}{G}
\]

If \( h_f \) be the head lost due to internal friction of the fan, then

\[
\text{Aerodynamical Efficiency} = \frac{h_a - h_f}{h_a} \quad \ldots \ldots \ldots (9)
\]

Since the imperfections in the design are certainly the cause of the chief sources of loss, namely eddying and stream collisions, we consider it preferable to stick to the conception of manometrical efficiency as being the best criterion regarding the performance of the actual fan.

Volumetric Efficiency.
Volumetric Efficiency.

This is the ratio of the actual volume discharged per revolution to the cubical capacity of the fan, or

\[
\text{Volumetric Efficiency} = \frac{\text{Volume of air discharged per revolution}}{\text{Cubical capacity of the fan}}.
\]

This does not conform to the accepted idea of an efficiency. It is of more value to manufacturers of fans than mining men, hence its comparative disuse.
PART II.

A REVIEW OF RECENT RESEARCHES RELATING TO PROPELLERS.
PART II.

INTRODUCTION.

In the following brief survey of recent work bearing upon the propeller fan, our chief purpose will be to summarily review the main conclusions arrived at by the foremost workers. With the advance of aviation, much work has been done relating to the performance of air-screws for air-craft purposes. On the other hand very little data are available as to the application of propellers or air-screws to the ventilation of mines. Since our own investigations are primarily concerned with this latter phase, precedence is given to work of this character, although the field for review is limited

Walker's Work:— During 1895-6, Mr. W.G.Walker conducted experiments on propeller fans and the results of these tests were contained in a paper read before the Institute of Mechanical Engineers on 30th. April, 1897. Mr. Walker's primary purpose was to ascertain:

(a) whether this kind of fan follows the ordinary laws respecting the mutual relations of speed of fan, power absorbed, and the volume of air discharged;

(b) the general characteristics regarding the speed of fan; power absorbed, and the volume of air discharged, with different angles of blades;

and (c) the effect of fans differing from one another only in the cross-section of their blades.

Seventeen three-bladed fans of 23½ inches diameter over the tips were tested. The fans, which were direct driven by an electric motor, forced the air through a tube of 24 inch bore and 4-feet long. In some of the later tests the fans were arranged so as to exhaust the air.

The experiments showed that the ordinary laws hold good and that the propeller fan is adapted to the discharged of large volumes of air at small pressures and further that the volumetric efficiency is more important than/
than even the mechanical efficiency. Briefly, Walker's conclusions for a constant angle of blade were:

(a) Air discharge varies as speed of revolution;
(b) Horse-power varies as \( \text{speed of revolution}^3 \);
(c) Horse-power varies as \( \text{discharge}^3 \);
(d) Torque varies as \( \text{speed of revolution}^2 \);
(e) Torque varies as electric current.

He states, and his results would tend to show, that the best angle of blade is some angle between 25 and 35 degrees. His mechanical and volumetric efficiencies were highest when the angle of blade was between these limits. He obtained mechanical efficiencies in the region of 40 per cent. and volumetric efficiencies of 75 per cent. and in some cases over 100 per cent.

As already stated, he tested several fans exhausting and forcing, and found that the propeller fan is more efficient when exhausting than when producing pressure.

The velocities were measured by means of the anemometer, but since many of the velocities given were over 1000 feet per minute we think that the results quoted are too high as this speed is outside the range of an anemometer for anything like accurate work. The pressure was measured by means of an impact tube set facing upstream.

**Stott's Work:** Experimental work on propeller fans has also been carried out by Mr. Oswald Stott, the results of which were embodied in a paper on "The Characteristics of the Propeller Fan" read before the Institute of Heating and Ventilating Engineers on 4th July, 1911.

The object of Mr. Stott's work was to ascertain the behaviour of the propeller type of fan under varying conditions of speed and resistance; primarily with respect to the power taken under these varying conditions and also noting the output.
The fan was 24 inches diameter over the tips, belt driven by an electric motor and discharging the air through a duct 3 feet by 3 feet in section and 25 feet long.

Briefly, Mr. Stott's conclusions were that under varying conditions of speed and resistance:

1. The output of air was proportionate to the revolutions of the fan;
2. the power taken was proportionate to the "cube" of the revolutions of the fan;
3. the pressure produced was proportionate to the "square" of the revolutions of the fan.

It will be observed that in the above tests the fan was forcing. The tests carried out by Mr. Walker and also those conducted in the Mining Laboratory of this University show that this type of fan gives better results when exhausting than when it is forcing the air through the duct.

In the paper referred to no mention is made as to how the pressure tube is placed inside the duct - whether facing upstream or downstream. This is a common fault with published results on fan work and one is somewhat sceptical of accepting results when this important point is omitted. The velocity of the air was measured at the outlet by means of an anemometer.

However, the results obtained in our own tests are in agreement with those obtained by Stott.

The foregoing researches on the propeller fan have been conducted under conditions somewhat resembling the ventilation of buildings, factories, ships, tea-drying, etc. and can scarcely be compared to the conditions under which a propeller fan would have to operate if installed to ventilate a large modern mine.
We have already mentioned that the propeller fan, (page 4) as generally constructed, has certain defects - principally, its inability to produce the pressures necessary to circulate large volumes of air through the workings of a mine. From the researches of Mr. F. A. Steart it would seem, however, that the fault lies not in the propeller itself but in its method of application. In a recent paper, "The Application of Air-Screws to Mine Ventilation", he provides strong evidence regarding the application of air-screws for this purpose.

During the past two years this energetic worker has conducted practical tests at the Northfield Colliery and the Transvaal and Delagoa Bay Colliery, South Africa, with Curtis air-craft propellers, 100 inches in diameter and of varied pitch. The air-screws were mounted symmetrically, and in series on the same shaft, and spaced 9 inches apart. The pitch of the air-screws was adjustable. The fan was arranged to exhaust the air from the mine, and in the earlier tests was fitted with an evasee, but this was discarded in later work, since it was constructed with a right-angled bend and tended to impede the air at discharge. It is worthy of note that at the Delagoa Bay mine the fan is placed vertically in the upcast shaft. By doing so, no fan drift is required and the loss of energy occurring at the fan drift inlet disappears; but, on the other hand, the use of the upcast shaft as an auxiliary or emergency winding shaft, is prevented.

Steart's work, if it can be accepted as reliable, and removed from commercial bias, would dispel the old idea.

idea regarding the inability of the propeller fan to create pressure differences sufficient for modern mine requirements. Moreover, he contends that the screw-propeller type of ventilator is not only applicable to mine use, but that it can be operated under such conditions with a considerable degree of efficiency. He also tested the difference in pressure existing between the various air-screws of the fan while it was working and found that air-screws in tandem build up the pressure stage by stage. His observations would appear to indicate that a definite relationship exists between the number of air-screws, their pitch and the mine resistance, viz:

(1) the resistance of the mine in Atkinsons is almost directly proportional to the number of air-screws;

(2) assuming the approximate truth of (1), when the number of air-screws is constant, the pitch varies inversely as the square root of the mine resistance (or directly as the equivalent orifice).

If these relationships are confirmed by further practical large scale tests, it will become a fairly simple matter to select a suitable air-screw combination to deal with any particular or varying mine resistance.

Steart observed that, the air passing through air-screws in tandem is not given a rotary motion, but that it appears to pass from stage to stage straight through the fan; this fact he observed by allowing small pieces of paper to be drawn through the fan.

Mechanical efficiencies exceeding 75 per cent. and volumetric efficiencies of over 80 per cent. were obtained with 2 bladed air-screws set in tandem.

In more recent work it would seem that the number of air-screws could be reduced by using four blades fitted to each boss instead of two blades. The relationships stated above would also seem to hold good for this type. These results also indicate that, as the number of 4-bladed air-screws is increased, the graph, obtained/
obtained by plotting efficiencies as ordinates and resistances or "equivalent orifices" as abscissae, moves towards the right and becomes more dome-shaped. This means that a 3-stage air-screw will maintain a given efficiency over a greater variation of mine resistance than a 2-stage fan of the same diameter, the pitch being constant.

In the tests at Northfield Colliery, Steart used the static gauge, and for this reason the efficiencies given in his paper are rather too high.

Again, in the tests with 4-bladed air-screws, the impact tube is set facing upstream, i.e. in the proper direction for recording the dynamic pressure. So far so good, but the tube is placed immediately behind the innermost impeller. As already stated, Steart found that the flow of air through the fan was practically streamline. If this be so, there would be some justification for placing the impact tube so close to the fan. In our own tests we found that there was a decided whirl when the distance between the impellers was small compared to the pitch. Since the smallest pitch used by Steart was 50 inches, it is very probable that rotational flow would always be present under such conditions. For this reason, it would seem advisable to place the impact tube some distance away from the fan. In all our tests the tube was fixed 7 feet away from the fan. Had it been set immediately behind the innermost air-screw the overall efficiencies obtained would have been almost doubled.

Again, Steart does not state the exact position of the impact tube with relation to the diameter of the drift. This is also an important point since it affects the efficiency of the machine.

Although Steart has carried out numerous tests with the air-screw fan working against a varying mine resistance (the resistance was varied by means of doors in/
in the fan drift) he does not appear to have noticed that with this type of fan the direction of outflow of the air changes with increased mine resistance. In fact, if the resistance be high enough, the direction of outflow is radial. Since this undoubtedly holds good for small fans of the propeller type, as was proved in our own tests, we do not know of any reason why it should not hold good for a large diameter air-screw. Further comment will be made upon this in Part III.

Nevertheless, although Steart's method of measuring the difference in pressure was undoubtedly wrong, the results obtained by him with air-screw fans will, we believe, compare favourably with those obtained with a centrifugal fan under similar conditions.

As remarked by Professor Henry Briggs during the discussion on Steart's paper, the air-screw type of fan would seem to be the fan of the future.

Figure 8. - The Effect of Diameter on the Efficiency of the Propeller.
The Aeronautical Research Committee's Work.

This Committee recently carried out an investigation with a Family of Air-Screws. The air-screws were all made with constant pitch, i.e. the undersurface of each blade was designed to form a true helicoid. This favours a good all-round performance. Briefly, the conclusions arrived at were:

1. that the performance depends on any variable such as the pitch, number of blades, area of plan form and maximum blade width;

2. that the efficiency of an air-screw of constant blade shape decreases as the number of blades increases from 2 to 6;

3. that the efficiency of an air-screw increases as the maximum blade width decreases from 0.123 D to 0.041 D, the number of blades remaining constant (where D is the diameter of the air-screw);

and 4. that with air-screws of constant blade area and constant pitch, the efficiency decreases, though slightly, with either a decrease of maximum blade width or an increase of the number of blades.

The highest maximum efficiency of the normal air-screws was obtained with a two-bladed air-screw with maximum blade width of 0.041 D. The air-screws used in our experiments were designed with maximum blade width of 0.041 D.

The Effect of Diameter on the Efficiency of the Propeller.

According to Dr. H. C. Watts, the biggest possible diameter of air-screw should be aimed at and such appears to be the common opinion in the aeronautical world. No experiments in a wind channel have been made to prove this point, but Dr. Watts gives a graph — reproduced in Figure 8 — plotted from Froude's theory to substantiate this/

1. Reports and Memoranda, No. 829
2. See "The Design of Screw Propellers for Aircraft" by H. C. Watts, p. 86.
this statement. The figure in question shows the variation of the efficiency with the diameter of the air-screw. Although this is intended to apply to aircraft yet it seems reasonable to suppose that a similar state will exist with air-screws applied to the ventilation of mines.

Again, according to the Aeronautical Research Committee, other things being equal:-

(i) an air-screw with tapering blades must have a larger diameter than one with broad blades to give the same performance?

(ii) the larger the diameter of the air-screw the smaller the reduction of the efficiency.

From the results of numerous tests on various sizes of propellers, W. G. Walker found that for free discharge of the air the horse-power required to drive the fan varied inversely as the fourth power of the diameter of the fan.

The diameter of the air-screw, therefore, would appear to have a big influence on its performance. We are inclined to the opinion that the low efficiencies obtained by us were principally due to the small diameter of the propellers.

Flow of Air Past an Air-screw.

As regards the inflowing column of air on an actual air-screw, each of the annular elements of the inflowing stream will possess the dynamic head of the surrounding air in the drift, and since the flow is convergent towards the air-screw there will be no tendency for the head to be dissipated in eddies, that is, there will be no turbulence, and it would be reasonable to suppose that /

---
1. Reports and Memoranda No. 829.
2. Reports and Memoranda No. 442.
that on reaching the air-screw the total head of each stream would still be the same and equal to the surrounding air in the drift.

In the outflowing column of air, however, the conditions for streamline flow would appear not to be so favourable since, the head of each annular stream having been increased by the blade to amounts differing in value, the probable result will be that on emerging from the air-screw radial current will be set up causing turbulent flow and consequently loss of head.

Recently, experiments were conducted by Dr. 1 Stanton and Miss Dorothy Marshall at the National Physical Laboratory in an endeavour to establish the conditions of the inflowing and outflowing columns of air past an air-screw.

A model air-screw was run in a four-foot wind channel. A pitot tube (A) was set up at a point 7 feet upstream from the air-screw and another (B) set 3 inches in front of the air-screw on the inflow side. Sensitive tilting manometers measured the difference in pressure between these two points. In no case was there found to be any difference between the values of the total head at B and A, sufficiently great to throw any doubt on the assumption of streamline inflow.

Observations were also made with the channel current stopped and the air-screw acting as a fan. In this case the conditions of streamline inflow were also satisfied.

Experiments were also made to compare the static pressure between the two sides of the airscrew, with the result that the loss due to friction and eddies was found to be very small and hence these workers concluded that the flow of air through an airscrew was approximately streamline flow.

1. See "The Design of Screw Propellers" by H. C. Watts, p. 90.
flow.

It is worth noting at this point that only a single air-screw was used in these tests and that no attempt was made to obtain information as to the flow of air through, say, 3 or 4 air-screws placed on the same shaft.
PART III.

EXPERIMENTAL WORK ON PROPELLER FANS.
One of the difficulties of carrying out tests on fans in a Mining Laboratory is that the resistance of the air-passage - unless this is provided with a shutter - is generally small compared with the resistance of the fan itself. Moreover, the velocities used in order to obtain a measurable drop of pressure are as a rule higher than those common in mine airways. It is usual, therefore, in tests on laboratory fans to increase the resistance of the air-passage by inserting a shutter or regulator.

In a recent paper by Dr. D. Pemban and the writer on "Experiments on the Flow of Air in Mines" it was pointed out that the orifice in a thin plate could not properly be substituted for the resistance to the passage of air in a long tube, or in a mine airway, or in a whole mine, in which there may be more than one condition of air-flow, and in which the index of velocity (or quantity) may have any value between 1 and 2. Therefore, in order that the propeller fans used might be tested under conditions closely resembling those of an actual mine, a gallery was erected at the Mine Rescue Station attached to the Heriot-Watt College. The total length of the gallery was 430 feet and had a total rubbing surface of approximately 8,800 square feet. The earlier tests on the propeller fans were conducted on this gallery, but as the results were very unsatisfactory, the later tests were carried out on a gallery 52 feet long erected in the Mining Department of this University. The gallery was provided with a shutter so as to vary the resistance against which the fan was working.

**Particulars of Fans Tested.**

Figure 9.- Typical Views of Air-Screw
FIG. 10.—BOSS FOR AIRSCREWS.
Experiments were carried out on five fans; two (a and b, page 33) were tested at the Mine Rescue Station; four (a, b, c and d, pages 33 and 34) on the gallery in this Laboratory and the other was a 6 feet diameter Sirocco propeller fan used in the ventilation of the Easthouses Colliery, Midlothian.

(a) Keith-Blackman Propeller Fan.

This fan has an overall diameter of 24\(\frac{3}{4}\) inches. It is fitted with six "spade-shaped" blades, which are set at an angle of 30° to the axis of rotation. The pitch of the blades, which is fixed, is 2 feet, giving a pitch:diameter ratio of 1 (approx). By means of additional bosses fixed to the same shaft, the fan was tested when working as:

1. A single fan having six blades;
2. Two fans in series, each having three blades and
3. Three fans in series, each having two blades.

(b) Air-Screw Fan.

This fan was kindly supplied by Messrs Walker Brothers, Wigan. The blades had a diameter of 24 inches over the tips and were of the same design, on a smaller scale, as those used by the Aeronautical Research Committee and which they found gave the highest efficiency. The shape of the blade is illustrated by Figure 9. The boss (Figure 10) which was designed in this University, was so arranged that it could hold four blades, and that the blades could be set at any desired angle to the axis of rotation, thereby altering their pitch. Air-screws, with two, three or four blades could be put on in tandem up to six stages, each stage having two blades.

(c) Helical-Screw Propeller Fan.

This/
Figure 11. - Helical Screw Blade.

Figure 12. - Helical Screw Blade
Fig. 13.
This fan was kindly sent by Mr. F. A. Steart, South Africa, and is 13½ inches in diameter. The blades, which are shown in Figure 11 were cut from a four-inch-diameter sheet steel pipe, and are six in number.

(a) Figure 12 shows a somewhat similar blade, the only difference being that both edges are straight.

All the machines tested acted as suction fans and were belt-driven from a 1 B.H.P. direct-current motor. The b.h.p. of this motor was obtained by means of the Prony Brake Method. The determination of the b.h.p. was carried out several times and mean values obtained from which Figure 13 is constructed. The motor was an old one.

Attention is directed to the fact that all the fans tested discharged direct into the atmosphere, none of them being fitted with an evasor.

Methods of Measurement.

1. Depression:— The difference in pressure between the outside atmosphere and a point inside the drift was obtained by means of a sloping manometer, which could easily read accurately to 0.002 of an inch of water. Petrol of specific gravity of 0.758 was used in the manometer throughout all the tests. The impact tube in the drift was placed at a point some distance from the fan where the flow was nearly streamline and the open mouth of the tube pointed up-stream. The tube was set on a centre line and 1/7th of the diameter of the duct measured from the side.

2. Velocity of Flow.

In the tests conducted at the Mine Rescue Station the velocity was measured inside the drift, while in subsequent tests the point of measurement was at the inlet to the duct. The observations were taken by means of a wind vane anemometer, 2½ inch. diameter. The instrument /
instrument was carefully used in a zig-zag manner over the cross-section area of the airway. It was frequently calibrated during the period of my research.

3. Input to Motor:

This was obtained by means of an Ammeter and Voltmeter, both instruments being new at the commencement of this research. They were calibrated in the Electrical Department of the Heriot-Watt College.

4. Speed of Motor.

Although not an essential factor for the calculation of the Overall Efficiency, the speed was always noted and was obtained by a reliable tachometer.
Figure 14.
Figure 14. - Variation of Efficiency with Resistance for Keith-Blackman Propeller.
In the tests carried out on the Keith-Blackman Fan, which was the first fan tested at the Mine Rescue Station Gallery, it was observed that the highest efficiency was not obtained with lowest resistance. Working against resistance there is, for a given speed, a point at which the efficiency is highest (Figure 14). We found that this point occurred when the resistance of the gallery amounted to about 1100 atkinsons (Figure 16). This fan appears to be suited to discharge fairly large volumes of air against not a high, but a reasonable resistance, although we doubt if it would create the depression necessary to ventilate a large modern mine.

In the earlier work this fan was tested:

1. with 2 blades - single stage
2. with 3 blades - single stage
3. with 6 blades - single stage
4. with 3 blades - two stages
5. with 2 blades - three stages.

From the results given in Table I it will be observed that there is not much difference in efficiency between three and six blades, single stage. From his experiments, W. G. Walker arrived at a like conclusion. It was also found that when this fan was run in combination, either two 3-bladed propellers or three 2-bladed propellers, the efficiency fell below that obtained when the fan was a single stage one of 3 or 6 blades.

<table>
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<th>No. of blades</th>
<th>Stages</th>
<th>W.G. in (ins. of water)</th>
<th>Qty. (cu. ft./min)</th>
<th>E.H.P. Overall Eff. per cent</th>
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</thead>
<tbody>
<tr>
<td>6</td>
<td>1</td>
<td>.2153</td>
<td>4550</td>
<td>1.61</td>
</tr>
<tr>
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<td>1</td>
<td>.1716</td>
<td>3650</td>
<td>1.11</td>
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<td>.13</td>
<td>3360</td>
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<tr>
<td>2</td>
<td>3</td>
<td>.1728</td>
<td>4500</td>
<td>1.73</td>
</tr>
</tbody>
</table>

Figure 15. - Direction of Effluent Air for Keith-Blackman Propeller operating against high Resistance.
Figure 16. - Resistance Curves for Keith-Blackman Propeller Fan.
We think that this is probably due to the fact that the direction of outflow is never axial with this fan, but generally about 45 degrees. Hence the impellers inside the fan casing would discharge against the inside of this casing, with subsequent loss of energy. It may also be partly due to increase in resistance offered to the passage of the air by having the fans in series. It would appear, however, that the former reason is the more probable.

The direction of outflow is about 45 degrees with no resistance. As the resistance is increased the direction changes, until, with sufficient resistance, it is purely radial flow. With still higher resistance the fan tends to discharge its air in a backward direction as shown in Figure 15. The cause of this phenomenon is somewhat difficult to explain, but it may be due to any of the following reasons or a combination of them:

1. the air coming off the blades at a velocity higher than the speed of the blade itself, and rebounding off the blade in front;
2. the deformation of the blade under load;
3. the air meeting the blade at an angle greater than 135 degrees when the tendency of outflow would be backward.

Figure 16 shows the relationship that exists:

1. between depression and volume with the fan running at constant speed against varying resistance;
2. between depression and volume with varying speed and constant resistance.

Following the lead of recent papers, notably by Dr. Parker and Prof. Briggs, the curve connecting the factors in (1) is spoken off as the Fan Characteristic and is represented by A S in the Figure, while (2) is termed the/
the resistance curve or, when applied in a practical sense, the Mine Characteristic.

The overall efficiency relationship is also given graphically in the same figure.

Taking a particular case, we have the resistance curve crossing the Fan Characteristic at the point F. Hence this fan running at the particular speed is able to cause a quantity $Q_1$ of air to pass through the drift at a pressure of $h_1$. Under the conditions denoted by $F$, the efficiency is measured by the intercept GH. It is clear that the fan is not at its best on this resistance, as it is not able to work at its maximum efficiency. It is more suited to operate against a higher resistance, such as that furnishing the curve $OC (P = 1106 Q^2)$, for which it would yield a smaller quantity at a higher pressure and at a n overall efficiency, K. L.

Suppose, for the sake of argument, that a fan with the characteristics shown in the figure was to be installed at a mine and that the minimum desired efficiency was 16 per cent. A horizontal line drawn through the 16 per cent. efficiency ordinate cuts the efficiency curve in the two points, M and N, which points determine the extreme limits of the desired efficiency range. Drawing verticals through M and N respectively, we determine the points B and R, on the Fan Characteristic, through which the curves of limiting resistance must pass to maintain the desired efficiency.

As already stated, the fan tested was best suited to operate against a resistance furnishing the curve $OC (P = 1106 Q^2)$, when it circulated 3050 cubic feet of air at a pressure of 0.57 inches of water column, and had an overall efficiency of nearly 17 per cent. Under these conditions the direction of the effluent air was radial.

The results show that the relationship between the pressure and the volume appear to follow the "square" law.
Figure 17. - Variation of Efficiency with Resistance of Helical Screw Fan (Fig. 11).
HELMICAL SCREW FAN.

The helical screw fan was 13 inches in diameter and the gallery had to be slightly altered to suit. The two fans tested - (c) and (d) p. 34 - gave practically similar results. The efficiencies obtained by these two fans were the highest recorded during this work.

Table II gives some of the results obtained and figure 17 shows how the efficiency changes with varying resistance. As with the Keith-Blackman, so with this type, the efficiency is highest when working against a certain definite resistance. This resistance, however, is higher than is the case with the Keith Fan.

The direction of the effluent air is similar to that in the Keith-Blackman type with the exception that there appears to be no tendency for backward flow when the resistance is relatively high.

During our research we tested a Sirocco propeller, 70 inches in diameter and ventilating a modern mine. Table III gives the results of this test. The efficiency and power factor graphs for the motor were obtained from the makers. Owing to the abnormal shape of the power factor curve it was subsequently found difficult to get the power input to the motor accurately when the power was low and for this reason no efficiencies are introduced on the table given for the first three tests.

<table>
<thead>
<tr>
<th>Fan Speed R.P.M.</th>
<th>Pressure in lbs/sq.ft.</th>
<th>Volume in cuft/min</th>
<th>H.P. in E.H.P.</th>
<th>Efficiency per cent.</th>
</tr>
</thead>
<tbody>
<tr>
<td>2300</td>
<td>1.734</td>
<td>1112</td>
<td>0.058</td>
<td>0.356</td>
</tr>
<tr>
<td>2660</td>
<td>2.168</td>
<td>1265</td>
<td>0.083</td>
<td>0.446</td>
</tr>
<tr>
<td>2930</td>
<td>2.556</td>
<td>1355</td>
<td>0.105</td>
<td>0.549</td>
</tr>
<tr>
<td>3200</td>
<td>2.985</td>
<td>1460</td>
<td>0.132</td>
<td>0.623</td>
</tr>
<tr>
<td>3400</td>
<td>3.479</td>
<td>1570</td>
<td>0.165</td>
<td>0.780</td>
</tr>
<tr>
<td>3670</td>
<td>4.074</td>
<td>1735</td>
<td>0.214</td>
<td>1.100</td>
</tr>
<tr>
<td>3740</td>
<td>4.246</td>
<td>1770</td>
<td>0.227</td>
<td>1.720</td>
</tr>
</tbody>
</table>

The condition under which the fan is normally run is that set forth in Test No. 4 when the volumetric discharge /
Figure 18
Figure 18. - Variation of Efficiency with Speed for 24 inch.
dia. Airscrew Fan.

4 Fans in Series Pitch 12 inches
4 " " " " 24 "
4 " " " " 36 "
Figure 19
Figure 19 - Variation of Efficiency with speed for 24 inch dia. Air-Screw Fan.

- 6 Fans in Series - Pitch 24"
- 4 Fans in Series - Pitch 24"
- 3 Fans in Series - Pitch 24"
discharge was 59280 cubic ft. per minute at a depression of 4.134 lbs. per square foot. The fan is apparently more efficient at higher loads though the efficiency is considerably lower than would be expected from the rival centrifugal machine. Undoubtedly the absence of an evasée is responsible, to some extent, for the low efficiency.

Airscrews.

The results obtained from the Air-Screws were particularly disappointing. It was by far the least efficient of the fans tested, even when working in six stages. Figures 18 and 19 give some idea of the efficiencies obtained with the fan installed at the Mine Rescue Station.

During tests on the flow of air through airscrews in series, it was observed that the distance between the impellers had a big influence on the direction of flow. When the airscrews (3 stages) were spaced three inches apart, rotary motion was predominant. This was observed by allowing confetti and fine dust to be drawn through the fan. Under these conditions the air appeared to pass through the fan with a cork-screw motion. When the distance was increased to eight inches, the rotary motion, though still present, was less noticeable. When the distance had been increased up to 20 inches, the direction of flow through the fan appeared to be purely streamline. The airscrews were 24 inches in diameter, the pitch being the same.

When the pitch was reduced to 12 inches, streamline flow appeared to take place when the impellers were separated /
Stage | Water-Gauge in ins.
--- | ---
1 | 0.036
2 | 0.051
3 | 0.140
4 | 0.290
5 | 0.410
6 | 0.436

Figure 20. - Positions and Readings of Gauges between the Stages of a Six-Stage Air-Screw Fan.
separated by 9 inches. With a pitch of 36 inches the rotary motion did not disappear when the distance apart was 21\(\frac{1}{2}\) inches. The three impellers then occupied the whole shaft.

Although we think it would be disadvantageous in practice to get rid of this whirl, due to the enormous length of shaft required if the fan was of large diameter, nevertheless, we contend that when testing a fan of this description, the dynamic tube of the pressure gauge should be set some distance from the fan, otherwise the gauge reading is bound to be affected by the whirl. Again, if the blades impart rotary motion to the air, Steart's statement that, with airscrews in combination, the increase in pressure-drop is almost directly proportional to the number of stages is not applicable.

Figure 20 gives the average results of several tests conducted in order to confirm, or otherwise, this contention made by Steart. The results appear to confirm the presence of whirl.

It is worthy of note that if, in our own tests, we had placed the impact tube at position 6, instead of 7 feet on the inflow side of the fan, the overall efficiencies obtained would have been doubled. From this it is evident that the whirl is causing an increase in the pressure registered and thence, in Steart's case, exaggerated efficiencies.

In all Steart's work the fan was 100 inches in diameter and the maximum distance between the airscrews in series was 9 inches. According to our results, rotary motion would not have disappeared until the distance apart of the impellers had been increased to something like 20 inches. This is for the smallest pitch tried, viz: 50 inches. For higher pitches the distance apart would require to be correspondingly increased.

Other points observed in these tests were mainly confirmations of Steart's results on a very small scale/
Figure 21
Figure 21. - Resistance Curves for 24 inch diameter Airscrew Fan - Pitch 12 inches.
Figure 22. - Resistance Curves for 24 inch Diam. Airscrew Fan - Pitch 24 inches.
scale with the exception that the fan failed entirely to produce a reasonable depression. Steart contended, and our laboratory tests tend to confirm, that:

(1) for a given resistance there are a definite number of airscrews and a definite pitch which give the best performance;

(2) any variation of mine resistance can be counterbalanced by varying the number of impellers or by altering the pitch of the blades.

Figure 21 has been plotted from the results obtained for a 6-stage air-screw fan of 24 inch. diameter having a pitch of 12 inches, while Figure 22 shows the results when the pitch is increased to 24 inches. The figures show:

(1) the characteristic curve for the fan running at a constant speed;

(2) the relation between pressure and volume with varying resistance and varying speed;

and (3) the overall efficiencies.

With regard to the last named, it should be noted that, although the values obtained are below those for the Keith-Blackman propeller fan, the graph is more dome-shaped and the fan would operate at nearly maximum efficiency against a large variation in resistance. It would appear that, as the pitch is increased, the efficiency curve becomes very flat-topped. This would tend to corroborate the earlier work of Steart.

The maximum overall efficiency obtained when the pitch was 12 inches, was 8.2 per cent. - the fan producing a pressure-drop of 1.248 lbs. per square foot and circulating 1950 cubic feet per minute. When the pitch was increased to 24 inches the efficiency increased to 13.3 per cent., the fan producing a depression of 1.33 lbs. per square foot and circulating 4,100 cubic feet of air per minute. In this case there was no restriction in the drift.

The direction of the effluent air was axial for resistances up to that represented by the curves which pass/
Figure 23.- Characteristic Curve for 6 stage 24 in.
diameter Airscrew Fan - Pitch 12 inches
Fan-speed 1760-30 R.P.M.
Figure 24
Figure 24. - Characteristic Curve for 6 stage 24" diam. airscrew fan - Pitch 24 inches. Fan-speed 1700-1720 R.P.M.
Figure 25
Figure 25.— Relation between Pressure and Volume for Airscrews 100 inches in diameter and a Pitch of 50 inches.—720-30 R.P.M.
Figure 26. - Relation between Pressure and Volume for Airscrews 100 inch in diameter with a Pitch of 70 inches - 720-30 R.P.M.
pass through C. As the resistance was further increased, the direction of outflow changed until, when working against the resistance represented by the curve at E, it was inclined at an angle of 45 degrees. When the drift was nearly closed the direction of outflow was radial.

Again, the relation between the pressure and the volume appears to follow the "square" law.

**Characteristic Curves of Propeller Fans.**

Figures 23 to 26 show the relation that exists between the Pressure and the Volume with variation in the mine or drift resistance, the speed being constant. This curve is termed the Fan Characteristic, or in other words, the effective depression produced by the fan against varying resistance. Figures 23 and 24 are plotted from our own results while figures 25 and 26 are plotted from results obtained by Mr. Steart, and are as yet unpublished.

The graphs are all slightly curved and fall very rapidly towards the right. It will be noticed that the Characteristic Curve is raised by adding air-screws in series. To illustrate the significance of this, suppose we take figure 25 which gives the Characteristics for a 100-inch diameter air-screw fan of 50 inch pitch and running at 720-30 r.p.m. Suppose the quantity required to ventilate a given mine was 100000 cubic feet per minute with a pressure-drop of 5.2 lbs. per square foot. A single stage air-screw would be sufficient for the work. Now imagine that the resistance of the mine has increased to such an extent that a pressure-drop of 9.88 lbs. per square foot is required to circulate the same quantity, i.e. 100000 cubic feet per minute. This duty could be performed by running two similar air-screws in series on the same shaft. Similarly, if the required pressure-drop was 15 lbs. per square foot a three-stage air-screw would be required to circulate the same total quantity.

**Fundamental Relationships.**
Figure 27. - Relation between Fan-speed and Pressure, Volume and Horse-Power for Keith-Blakemore Propeller.
Figure 28
Figure 28. - Relation between Fan-Speed and Volume, Pressure and Horse-Power for Helical Screw Fan (Fig. 11).
Figure 30. Relation between Fan-speed and Pressure, Volume and Horse-Power for Airseep
There is general agreement regarding the fundamental laws connected with the performance of centrifugal ventilators. The following laws have been firmly established, provided the mine resistance is kept constant.

1. Where $Q$ is the quantity of air delivered per minute, and $N$, the number of fan revolutions in the same time:

$$ Q \propto N. $$

2. Since $h$, the effective head, varies as the square of the quantity, from (1) it follows that:

$$ h \propto N^2. $$

3. From numerous experiments, it has been shown that the quantity $Q$ is proportional to the cube root of the horse-power supplied, or

$$ \text{HP} \propto Q^{\frac{1}{3}}. $$

Again, from (1) it follows that:

$$ \text{HP} \propto N. $$

In figures 27 to 30 we have plotted some of the observed results for the several fans tested. The graphs would appear to indicate that the above quoted laws hold good for all forms of the propeller fan. Since the laws themselves have been plotted, all the graphs shown in figures 27 to 30 must pass through the origin.

Although the greatest care was exercised in taking the necessary observations, it will be seen that the results are very unsatisfactory from every point of view, especially the overall efficiency of this type of ventilator. This low efficiency may be due to a combination of the following causes:

1. The fans tested were too small in diameter. (With experimental centrifugal fans the overall efficiency is generally somewhere between 30 and 35 per cent, even under ideal conditions).

2. The air-screw blades were too narrow for the work required of them. (Although the narrowest blade used by the Aeronautical Research Committee gave the highest efficiency (see page 28), this may not apply to mine work).

3. Leakage - this was particularly evident at the Mine Rescue Station gallery, but was entirely absent during the later tests carried out in this Laboratory.
(4) The deformation of the blade under load.

This last point requires a knowledge of the position of the centres of pressure, and shows that aerodynamic design is dependent on a knowledge of the stresses and strains of an airscrew.

Conclusions.

The following are the conclusions arrived at:

(1) The fundamental relationships hold good for all types of propeller fans, namely:

\[
Q \propto N^2 \\
F \propto N^3 \\
\text{and } HP \propto N
\]

(2) The propeller fan will not operate efficiently against even a relatively moderate resistance since the direction of the effluent air changes with varying resistance.

(3) Fans of the size tested would seem to be suitable only for circulating comparatively large volumes of air against very low resistances, i.e. they appear to be adaptable only for the ventilation of buildings, factories, ships, etc.

(4) When the impellers are too close together, the air is given a rotary motion by the airscrew and hence does not pass through the fan in stream-line flow. This is particularly the case with airscrews in combination.

(5) Only when rotary motion is entirely absent will the airscrews build up their pressure by approximately equal stages.

(6) With airscrews, both depression and volume are increased by (1) increasing the pitch, or (2) increasing the number of airscrews.

(7) As the pitch increases the efficiency curve tends to become dome-shaped.

(8) No benefit is derived by having the blades staggered when dealing with airscrews in combination. Equally as good results may be obtained by placing the airscrews in the same axial plane.

(9) In airscrew combinations, the proximity of one fan to another is of considerable import. Adjacent fans should be as far apart as is practicable on the same shaft.

(10) The largest practicable diameter of airscrew should be aimed at; see page 28.

(11) Air-screws operate more efficiently when exhausting air than when forcing it.
An evasee would tend to increase the fan efficiency where the direction of the effluent air is axial.

and Reversal of the air can be easily accomplished by:
(a) reversing the direction of rotation of the airscrews;
(b) reversing the airscrews themselves.

Despite the fact that our observations lead us to the above conclusions, there is no doubt that the work has been carried out on airscrews too small in diameter. Tests by Steart on large-scale fans would indicate that the diameter of this type of fan materially matters even more so than does that of the centrifugal fan. This opinion has been stated by several observers, notably Dr. Watts and W.G. Walker. (see page 28).

The advantages of the airscrew fan as demonstrated by Steart are:

1. **Flexibility** - The fan can always be readily adjusted to efficiently cope with variations in the conditions, either by the speed, pitch or number of screws. Compare this with the installed centrifugal fan which has only a speed adjustment.

2. **Simplicity** - Because of its lightness and simple character, the installation of such a ventilator is facilitated.

3. **Cheapness** - The difference in cost of such an installation as compared with that for a centrifugal ventilator must be considerable.

4. **Reversal** - In emergency, simply requires a reversal of rotation; for permanent reversal, the airscrews on the shaft would be reversed.

Even though experience should fail to establish Steart's claim for an efficiency equal to that of a centrifugal machine, it has to be remembered that, in this regard, a fan which can easily be kept at its maximum of performance may behave more creditably than one whose best performance is somewhat higher, but which is generally compelled to operate under conditions outside the range for maximum efficiency.
PART IV.

A REVIEW OF RECENT RESEARCHES RELATING TO

MINE VENTILATION.
PART IV.

INTRODUCTION.

The theory of mine ventilation is at present in the melting pot, and principles, both simpler and more scientific, are being introduced. Since the classical expositions of Atkinson and Murgue, very little advance had been made until quite recently. In a paper "A New Method of Measuring Ventilating Resistances" read before the Scottish Institution of Mining Engineers in August 1921, Dr. D. Penman advocated the abandonment of Murgue's theories of the "equivalent orifice" and "orifice of passage" and a substitution of more direct methods of measuring the mine and fan resistances to the flow of air through them. These methods will be discussed in the next section.

In 1923, the Council of the Institution of Mining Engineers deemed it expedient to set up a Committee for the purpose of revising the existing theory of mine ventilation. That committee issued a First Report in June, 1925, and proposed the adoption of the simplified variation of the Atkinson formula advocated by Penman.

Atkinson's well-known equation for the flow of air in mine passages, viz: \( p_a = k s v^2 \), was evolved from a consideration of various hydraulic formulae, particularly Chezy's. The formula was based on the assumption that the frictional resistance to the flow of air varied directly

(a) As the extent of the "wetted" or "rubbing" surface;

(b) As the roughness of the surface;

and (c) As the square of the velocity of flow.

As has been pointed out elsewhere, however, modern research has shown that some of the hypotheses, and particu:

ularly /

particularly that under subheading (c), are only correct under certain conditions.

Although the formula is especially applicable to mine airways, it is unwieldy in application as in many problems we are dealing with changes produced in the same set of galleries when $k$, $s$ and $a$ are assumed to remain constant, and the only variables are $p$ and $v$ (or $q$).

The Committee referred to above recommend that an equation of the general form:

$$ P = RQ^n $$

be adopted, where $R$ is the resistance in terms of dimensions, etc., of the airway or the mine, and measured in Atkinsons.

The value of the index $n$ has led to some controversy in the past. From the work of Storrow we know that when air is dragging through narrow passages, such as through a waste or goaf, the pressure varies directly as the quantity. On the other hand, from numerous experiments we are reasonably sure that, for conditions such as exist in the fan drifts of Mining Laboratories, the pressure varies as the "square" of the quantity. The actual index of $Q$, for a whole mine, however, would appear to have some value between 1 and 2, the index depending on the condition of the mine. The value of the index of $Q$, therefore, is a subject for practical investigation. Dr. Penman and the writer carried out a test at Wellesley Colliery, Fifeshire, and ascertained the relationship between the quantity and the pressure for the whole mine, under the then existing conditions. The equation for the mine was $P = RQ^{1.8}$. Dr. Penman and Mr./

2. The Atkinson is defined as that resistance which absorbs a pressure of 1 pound per square foot when a volume of 1000 cubic feet per second of dry air at 60°F. and 30 in. barometer is passing.
Mr. T.A. Wetherell have also conducted a similar test at Lodna Colliery, India, and found that when the quantity passing through the mine was 20,000 cubic feet per minute or less, the equation connecting the pressure and the quantity was \( P = R Q \). and when the quantity exceeded 20,000 cubic feet per minute, the equation became \( P = R Q' \).

It has been suggested by several authorities, notably Professors Briggs and Hay, that a more exact relationship between the pressure and volume for a whole mine would be

\[
2 \quad P = A Q + B Q - p_o
\]

where \( A \) and \( B \) are constants, the value of which would depend on the mine conditions and \( p_o \), the natural ventilating pressure. On the other hand, Professor Gibson suggests that an equation of the form

\[
3 \quad P = A Q + B Q + C Q
\]

may more correctly represent the conditions.

In an endeavour to throw some light on this somewhat clouded subject, we conducted several tests at twelve mines, eight of them being situated in Scotland and four in England. Before proceeding to describe these experiments we propose to give a brief survey of recent work bearing upon mine ventilation in general, our chief purpose being to summarily review the main conclusions arrived at by the foremost workers.

At the outset, I should like to record my indebtedness to the Managing Directors, Agents, Managers and officials of the various collieries at which tests were conducted for the facilities granted and the assistance given, and to Dr. J. N. Williamson, H. Hyde and others for able assistance in the carrying out of the tests.

   (b) See "Fan Problems", Colliery Engineering, June, 1925.
I am also indebted to Professor Henry Briggs, under whose direction and supervision the experimental work herein described was carried out, for able assistance and advice. Finally, I have to express my thanks to the Coalowners' Research Association for very materially assisting the work by a grant to meet travelling and certain other expenses.
As stated at the outset, the present tendency is towards a simplification of the complex problem of mine ventilation. Halbaum, Shaw and others have mentioned the direct method of measuring the mine resistance. The credit of the present move by the Institution of Mining Engineers towards this simplification however must be given to Dr. David Penman who, in 1921, ably advocated the adoption of a direct method of assessing the mine resistance and the abandonment of Murgue's indirect methods of "equivalent orifice" and "orifice of passage". Penman compared an electrical and a ventilating circuit and although this is not a new idea, he, nevertheless, drew more from the analogy than any previous writer and developed a number of useful and simple relationships based on this central conception of a specific unit of resistance. His suggestion was based on Ohm's law, when applied to a dynamo, thus:

\[ E = I (R + r) \]

where \( E \) is the electromotive force in volts, \( I \) the current flowing in the circuit in amperes, \( R \), the external and \( r \) the internal resistance in ohms respectively. The analogous equation in ventilation he proposed was

\[ P = \frac{Q^2}{(R + r)} \]

where \( P \) is the total head produced by the fan, \( Q \) the quantity flowing, \( R \) and \( r \) the resistances of the mine and fan respectively. If \( h \) is the effective head, then

\[ h = \frac{RQ}{2} \]

and if \( h_0 \) is the head absorbed by the resistance, etc., of the fan, then

\[ h_0 = r_0^2 \]

As Parker aptly expressed it, \( R \) in the above formula, "is the integral of such quantities as \( \frac{K^2}{a^3} \) (Atkinson formula) for very numerous small portions of the air-way."
The connection between the older orifice formula and those proposed by Penman is:

(1) Equivalent Orifice of Mine = \( \frac{K}{\sqrt{R}} \)

(2) Orifice of Passage of Fan = \( \frac{k}{\sqrt{r}} \)

where \( k \) is a co-efficient determined by the units used.

In his original paper, Dr. Penman demonstrated the utility of the direct method of resistance measurement when applied to the solution of problems related to the running of fans in combination. Of recent years, practical problems of this nature have frequently arisen, where it has been found necessary to augment the volume of air in circulation, and usually, when two fans in combination have been tried the results obtained were considerably below those anticipated.

**The Series Combination.**

Considering the question from an increase in volume point of view, we have, for two identical fans in series

\[ Q = \sqrt{\frac{2h}{R + 2r}} \]

the necessary condition here being that the fans each produce the same total depression when in combination as when operating separately. When \( r \) is small compared with \( R \), the expression becomes

\[ Q = \sqrt{\frac{2h}{R}} \]

that is the quantity produced by two fans in series under these conditions is \( \sqrt{2} \) times the quantity \( (Q) \) produced by one. However, to realize this, the power required by the fans in combination would be greater than the sum of the powers developed by the fans separately since, when in combination, the power would be

\[ \sqrt{2} (2hQ_2) = 2.828 hQ_1 \]

instead of \( 2 hQ_1 \).

If on the other hand, the sum of the powers developed by the/
the fans is to be the same when they are running in series as when running separately, the quantity will not be increased $\sqrt[3]{2}$, but $\sqrt[3]{2}$ times, since it is well known that the quantity is proportional to the cube-root of the horse power. In other words, provided that the resistances of the identical fans are negligible in comparison with the mine resistance ($R$), and the fans are run so that the total power supply is equal to the sum of the separately-run power supplies, the maximum increase in volume which can be realised is 26 per cent. In practice, however, the fans in series are rarely identical and their resistances compared with that of the mine are not negligible, hence in the series arrangement, the increase would be much below 26 per cent.

Parallel Combination.

The equivalent resistance of two identical fans in parallel is $\frac{r}{4}$ so that the quantity circulated when both fans are running at the same speed is

$$Q = \sqrt{\frac{h}{R + \frac{r}{4}}}$$

as compared with

$$Q = \sqrt{\frac{h}{R + r}}$$

for one fan operating alone.

Comparing these formulae it is obvious that unless the resistance ($r$) of each fan is large compared with $R$, the increase in quantity resulting from the parallel arrangement will be very small. Penman showed, both theoretically and by actual experiment, that a slight decrease in the speed of one fan would result in the second fan sucking a portion of its air from the outside atmosphere, through the fan running at the lower speed.

Briefly, Penman's conclusions are:-

The relative resistances of the fans to the resistance of the mine determine which combination would be the better. In general, where increased quantity is the/
the chief consideration, better results will in most cases be obtained by combining fans in series than by running them in parallel. In the parallel arrangement efficient speed regulation is essential. (Parker, however, contends that power regulation is of more importance - see under Parker's Work).
Clive's Work.

Apart from his experimental work in the running of two fans in parallel at Bentley Colliery, in which he obtained results somewhat similar to Penman, Seymour Wood and others, Mr. Robert Clive has rendered much service towards the advance of the science of mine ventilation. Prior to his recent work, the effect of natural agencies on the ventilation of mines had been neglected. In the work referred to, the varying character of natural ventilation is made manifest. The volume of air produced by natural causes is solely dependent upon the difference in density of the air in the two shafts, and inclines and upon the resistance of the mine. Clive measured this natural effect by observing the depression or pressure at the pit-bottom, top of upcast, and in the fan drift, in four experiments and from these deduced the natural ventilation pressure for the period of the test. Since the seams at Bentley Colliery are practically dead-flat, we think that a more suitable method would have been to ascertain the natural ventilating pressure from the difference of density of the two air columns, when the fan was running. If the average upcast temperature is the higher, natural agencies will be assisting the fan; if the average downcast temperature is the higher, then natural ventilation will oppose the fan. Hence, the full effective pressure producing ventilation throughout the mine is:

\[ \text{The pressure (or depression) created by the fan + the natural ventilating pressure; the sign of the latter is positive when assisting the fan, and negative when it acts adversely.} \]

It follows from this argument that the total resistance:

resistance of the mine must be determined from that pressure and not from the observed fan drift pressure.

Clive's observations showed that the variation in the natural ventilation effect due to changes in the surface temperature may cause a wide variation in the total resistance against which the fan is called upon to operate, apart from any change in the mine resistance itself. At Bentley Colliery, for instance, it is estimated that the combined resistance under normal conditions varies from 0.854 Atkinsons in summer to 0.394 Atkinsons in winter, solely due to changes in temperature. That is to say that the equivalent resistance in winter is only about 44 per cent. of the mine resistance. The advantage, therefore, of having a fan with a flat-topped efficiency curve is obvious.

One of the most important things brought out by Clive's results was the abnormal resistance of the fan drift inset and the shafts. He showed that the relative resistances, expressed on a percentage basis, are:

- Fan-drift inset: 18.8 per cent.
- Shafts: 22.1 per cent.
- Mine: 59.1 per cent.

The figures are astonishing when it is realised that the area of the fan drift was 120 square feet (it has since been increased), while the shafts are 20 feet in diameter and 1970 feet deep.

The value of Clive's work is, in our opinion, that it has focussed the attention of the mining engineer upon:

(i) the magnitude of natural ventilation;

and (ii) the high resistance of fan-drift insets, shafts, etc.

Both these points appear to have been neglected in the past. Little or no attempts have been made to obtain the maximum assistance, economically from natural ventilation. The great majority of fan drifts in/
in this country are, we believe, too small in area and are generally constructed with a right-angle bend.
Parker's Work.  

Within recent years it is safe to say that no one has done more to advance the science of mine ventilation than Dr. Joseph Parker of the Fife Mining School. Since 1921 this fertile worker has contributed to the Institution of Mining Engineers four valuable papers, all bearing on the subject of mine ventilation. In dealing with his work we shall review it as briefly as possible.

The Characteristic Curves of Fans.

In this paper he excellently demonstrates the facility and utility of the direct method of measuring resistance. He also indicates the value of characteristic curves in the predetermination of the output and efficiency of fans working single and in parallel against various resistances. In later papers he illustrated the applicability of the curves to fans working in series with each other - natural ventilation being a particular case of a fan operating in series with the main ventilator. The key to these problems appears to lie in the conception of the "equivalent resistance". By this term is meant the resistance against which the fans in combination are operating as distinct from the resistance against which they operate when working singly. For example, if we can run two fans in parallel on a constant external resistance and each fan passes the same volume as it did when working alone, the total volume circulating will be doubled and the external pressure will be quadrupled. Hence, fans being in parallel under such conditions, each machine is passing the same volume of air as before, but at four times the former pressure. Or as Parker puts it, it is just as if the fan had been set to work on a mine having a/ 

Figure 31. - Equivalent Resistance for Fans in Series and in Parallel. Graphical Determination of Efficiency and Load at which a Fan would operate under various conditions.
a resistance four times as great as that of the actual mine.

He has formulated the following laws relating to equivalent resistance:

(1) The equivalent resistance against which fans running in parallel may be considered to operate is equal to $2^2$ times the resistance ($R$) of the mine when the fans and motors are duplicate; and for other cases is $n^2 R$, where $n$ is the number of times the fans are together more powerful than the fan running before being paralleled. (This law will be strictly fulfilled only if each fan-motor continues to develop the same power, after paralleling, as it did when acting separately).

(2) If $n$ be the number of times that the two fans in series are together more powerful than the first running fan, the equivalent resistance against which the fans may be supposed to act is the mine resistance ($R$) divided by $n$, i.e. $R_{e} = \frac{R}{n}$.

From these laws it will be observed that the effect of the series arrangement is to render it easier for the individual fan to circulate a given volume of air; whereas the opposite applies to the parallel combination. In connection with the satisfactory running of fans in parallel, Parker contends that the dictum enunciated by Penman regarding the necessity for speed regulation is wholly inadequate. Parker's contention is that the factor of primary importance in this arrangement is that the power input to the fan-motors shall either remain constant or shall not fall away too much when the speed of the fan increases. He attempted to verify his deductions by experimental evidence at Wellesley Colliery, but was unable to complete the work. Nevertheless, he did obtain some evidence supporting his contention.

To illustrate the advantage of the equivalent resistance idea we have taken Figure 31 from one of Parker's papers. Suppose AECDB represents the useful depression; OJFGHB the efficiency, and OFCR the pressure volume relation of

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of a fan operating against a mine resistance of 1 Atkinson. The pressure-volume and useful depression curves cross at $C$. Hence the particular fan running at the particular speed is able to cause a quantity, $Q$, of air to circulate at a pressure, $p_1$. Under the conditions denoted by OCR, the efficiency is measured by the intercept GL on the vertical through $C$, and shows that the fan installation is working with an efficiency of 50 per cent. Should it be desired to install duplicate fans and the question was, whether, under existing conditions, it would be better to arrange the fans in parallel or series, the answer is readily obtained from the graphs. For duplicate fans in series the equivalent resistance curve would be OPD; the efficiency of each fan if now represented by the intercept HM, and equals 40 per cent. The power input has been doubled, but the useful effect has increased in the ratio of 8 : 5 only. Under these conditions the volume circulating has been increased to only 117 per cent. of its former value, as compared with the 126 per cent. increase which the ordinary law connecting volume and power would indicate.

The curve OJE represents the equivalent resistance under a parallel arrangement of duplicate fans; the efficiency ($FK$) for each fan has increased to 65 per cent. in this case, which means that the volume has been augmented to 137.5 per cent. of that circulated by one fan when operating alone.

In his paper "Economy and Efficiency in Ventilation" Parker describes a "rapid" method of determining the full characteristics of a fan. This method consists of running the fan at constant speed on two different external resistances, and of making very careful measurements of the volume of air passed and of the useful ventilating /

ventilating pressure for each of these two resistances. He also deals with a variety of subjects such as the importance of the correct construction of the evasee, the position of regulators and underground booster fans, fan drives, surface leakage, and excessive resistance of splits. With regard to the latter, he reminds the mining engineer about a very important matter too often lost sight of, namely, that the ventilating pressure, and therefore the power required to pass a definite volume of air through an airway, varies inversely as the fifth power of the corresponding sides or diameters, where the shape of the airway remains constant. Thus, if we were expending 32 horse-power to pass a given volume through a certain airway, by doubling the height and breadth of the airway, keeping the same form of cross-section, the power required to pass the same volume would fall to 1 horse-power. The importance of maintaining as large airways as practicable is thus apparent.

Choice of a Mine Ventilator.

At one time it was considered that the basic factor upon which the choice of a ventilator depended was the equivalent orifice of the mine. Parker, however, shows that the selection of an efficient fan is not dependent on the equivalent orifice, orifice of passage, or on the shape of the blades, but upon two essentials, namely, (1) a consideration of the characteristics of the fan and (2) the limiting values of the external resistance against which it may have to operate. As already pointed out, (page 3) the principle cause of low efficiencies in connection with mine fans is that they have been designed to work against a particular resistance, which may not have been, or may never be, realised. As illustrated in Figure 4, when the efficiency curve begins to fall, it does so rapidly, so that, should fans be operating under conditions totally different from those for which the/
Figure 32
Figure 32.— Showing the Characteristic Curves of a Fan and how to determine the limiting value of $R$ within which the fan will give an Efficiency of not less than 60%. The values for this case show that $R$ must not exceed 7.5 Atkinson's and must not be less than 1.43 Atkinson's.
the fan was designed, then low efficiencies are inevitable. To efficiently meet this varying mine resistance, it is obvious that a fan possessed of a flat topped efficiency curve, is of first importance despite the fact that such a fan would have a high internal resistance. In his paper "The Choice of an Efficient Fan or Ventilator for a Mine", Parker indicates how the several mine resistances can be estimated after the general lay-out has been settled. The probable variations in the resistance due to development work and also due to seasonal changes are likewise dealt with in this paper. He takes Clive's figures for Bentley Colliery (see page 56) and shows that the variation of the effective resistance due to natural agencies may be much greater than any changes due to underground development.

To illustrate Parker's arguments on the choice of a fan, we have taken Figure 32 from his paper on the subject. Suppose we desire the overall efficiency of the plant to be 60 per cent., and that a fan having the characteristics shown in the figure has been offered. A horizontal line which represents an efficiency of 60 per cent. cuts the efficiency graph of the fan in the points A and B so that these points mark out the limits of the desired efficiency range. By drawing verticals EAC and FBD through the points A and B respectively, we obtain the points C and D on the pressure-volume curve, through which the curves of limiting resistance must pass. These resistances are numerically equal to CE/OE and DF/OF units respectively, and if the value of the mine resistance and the variations of that resistance fall between these limiting values, the fan will be suitable for the work and will give the required efficiency. If not, a fan having characteristics which satisfy the conditions of the case must be sought for.

As already mentioned (page 4), where the efficiency has fallen below an economic value, due to a
large reduction in the mine resistance, Parker suggests running another fan in parallel, which could boost up the efficiency. Similarly, should the mine resistance become too great for efficient operation, the installation of another fan in series is suggested. These obviously give alternatives to the replacement of the existing fan.
Brigg and Williamson's Work.

In an exhaustive research on the fan evasee, Professor Henry Briggs and Dr. J. N. Williamson have determined the practical standards in the design of such an adjutage. While future installations will doubtless be built to conform with the standards which they set forth, modern fan evasees indicate, by their variety of angle, length, and section, that hitherto, makers had not been guided by any authoritative work in their design. These workers conducted long and careful experiments on both diverging and converging ducts with angles having from 0 degrees to approximately 20 degrees. Tests on various evasees installed at large collieries were also made.

As is well known, the function of the evasee is to convert the kinetic energy possessed by the air effluent from the fan casing into pressure energy. Neglecting frictional losses, the evasee forms a good practical application of the Bernouillian Theorem. In assessing the efficiency of the various forms of divergent and convergent ducts, Messrs. Briggs and Williamson used this principle; the frictional loss in any duct is accurately measured by the mean difference in the dynamic-gauge readings between the throat and the mouth of the duct.

From their work these fertile workers arrived at the following conclusions:

(a) Pressure energy in air can be converted into kinetic energy with an efficiency approaching 100 per cent. by any form of converging duct.

(b) The efficiency of conversion of kinetic energy into pressure energy is in marked contrast to that of the reverse operation; with the best shaped divergent duct, the efficiency of conversion did not greatly exceed 80 per cent.

(c) Suggested Standards in Design:

(i) A chimney with four equally diverging sides should have its sides having at an angle between 2½ degrees and 4½ degrees.

Figure 33. - A Proposal to obtain a 4-to-1 Expansion without involving a large Vertical Chimney.
(ii) A chimney expanding in one direction only should have an angle of hade between 8 degrees and 14 degrees.

(iii) A chimney with three diverging sides should have:

(a) The wall nearest the fan, vertical;
(b) The wall farthest from the fan, to have at 7 degrees;
and (c) The side walls to have equally at 3$^\circ$ degrees.

This was their practical ideal.

(iv) Length of Evasee.

With angles diverging as above, the length should be continued until the ratio mouth area : throat area is as 4 : 1.

(d) The greatest loss of energy in an evasee occurs at the throat; hence this part requires care in design.

(e) An ill-designed evasee - even a parallel-sided one - is better than none.

(f) Other things being equal, the lower the depression created by the fan, the greater is the influence of the evasee on the efficiency of the plant.

Since, to incorporate the recommendations of these workers in the design of an evasee for a large fan installation would involve a huge vertical chimney (to secure the 4 : 1 ratio) if the modern methods of erection were to be followed, Messrs Briggs and Williamson proposed the arrangement illustrated in Figure 33. Here the greater part of the evasee is horizontal and is made by trenching from the surface; the walls and roof of the passage may be supported by concrete. There is no grave objection to the right-angled turn, as it occurs where the velocity of flow is low. Doors are provided to allow of the air being reversed in the usual manner.

In a series of experiments on fan-casings and fan-inlets, these workers set out to determine (as far as was possible under the limiting conditions of the laboratory and

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and the plant at their disposal) the merits and demerits of these fan adjutaries as at present constructed, and to ascertain the degree of improvement which could be effected in their practical design. Their tests were carried out on a Sirocco fan and casing, the latter being subsequently altered during the course of the experimental work. By means of a modified Pitot tube connected to a large inclined pressure gauge, over 200 measurements of velocity were taken at different points symmetrically arranged between the periphery of the fan runner and the sides of the casing while the fan was running at a constant speed against a constant resistance. Such measurements indicated the marked degree of variability in velocity of flow in the air space between the fan runner and the casing, and consequently, the considerable energy loss which occurs where excessive turbulence exists and where there are collisions between air-streams of different velocities. From this part of their work, these research workers pointed out that insufficient attention was paid to the design of fan casings, particularly as regards width and the shape of the spiral or volute. Small changes in the shape of the latter involve appreciable differences in efficiency. By reducing the width of the original Sirocco casing to less than half (it then cut off the less efficient, or as it sometimes is, the adversely efficient, part of the fan runner) and modifying the volute curve so that its contour is conformed to the true spiral (except at the beak, wherein present Sirocco designs, a right-angled corner), an increase of over 6 per cent. in overall efficiency was effected.

Again, the efficiency of the Rateau diffusion-ring as a means of reducing the excessive losses otherwise occurring through high-velocity air from the runner entering the casing is commended; this device also largely decreases kinetic energy losses due to radial velocity.
The merits of the Sirocco beak - the right-angled corner - were next investigated. This particular part of Sirocco casing is the subject of a patent by the late Sir S.C. Davidson (No. 10684/1905) in the specification of which he makes two claims, viz: (1) centripetal re-entry obviated; (2) the water-gauge remains constant with constant fan speed against any external resistance. Messrs Briggs and Williamson, as a result of their work, could not support these claims. On the contrary, their work showed that while the right-angled corner simplified construction it was seriously detrimental to the fan's efficiency. Indeed, according to their measurements, nearly 12 per cent. of the volume entering the fan-inlet was re-circulating in the casing, this volume re-entering the casing past the "beak" of the fan. By ingeniously varying the clearance between the fan casing and the runner at the "beak", a position was reached which gave maximum efficiency. Changing from the worst clearance to the best affected the overall efficiency by nearly 7 per cent. With the experimentally-determined "best" clearance, a certain amount of "circumferential" re-entry still occurred, but these workers contended that such was necessary to serve as "lubricant" and it would be impracticable to demarcate too finely the commencement of the casing.

Experiments with five forms of fan-inlet were made. From their comparative results, it was made clear that the inlet arrangement has an appreciable affect upon the fan efficiency. Indeed, the difference in overall efficiency which resulted from the use of the least and most efficaceous of the inlets they used amounted to as much as 9 per cent. They pointed out that it was highly probable considerable improvement could be gained in existing plants in this regard. A large inlet is not enough; to minimise losses due to shock and turbulence, the form of the inlet should be such that the entering air is/
is guided gradually towards the blades as is done in the case of the Rateau fan.
Figure 34. - Stream-line Flow

Figure 35. - Turbulent Flow
Conditions and Nature of the Flow of Air.

When a fluid which completely fills the duct in which it flows is set in motion in a long straight pipe without shock or disturbance which would produce turbulence in the fluid, continuous or stream-line flow is produced. As the velocity of flow is gradually increased, a point is reached at which internal disturbance takes place and the motion breaks up into turbulent flow. After the fashion of Unwin, stream-line and turbulent flow may be represented pictorially as in Figures 34 and 35.

The point at which the flow changes from stream-line to turbulent motion is called the "critical velocity" of the fluid. Professor Osborne Reynolds showed that the critical velocity depends upon:

(a) the density of the fluid,
(b) the viscosity
and (c) the diameter of the duct.

These relationships may be expressed in the form:

\[ \frac{\rho V_c D}{n} = \text{constant} \]  \hspace{1cm} (10)

where \( \rho \) = density, \( V_c \) = critical velocity, \( D \) = diameter of duct, and \( n \) = coefficient of viscosity.

This is known as the Reynolds constant. There are two values of the constant, one for the point of change from continuous or stream-line flow to sinuous or turbulent flow, and the other for the point of change from turbulent to stream-line flow. This latter form of change may occur when the fluid is initially disturbed and slows down, or when the rate of flow during turbulent motion slows down sufficiently to enable the fluid to revert to stream-line flow.

There are, therefore, two critical velocities in the flow of a fluid in a duct. These may be represented /
Figure 36
Figure 36. — Relation between Velocity and Pressure.
represented graphically as in Figure 36. Suppose the velocity to start from zero at the point A: as the velocity increases the friction loss increases along the line A - B. At the point B the flow changes to turbulent flow and the friction loss increases along the curve C - D. Suppose now that the velocity is gradually reduced, the friction loss decreases along the curve D - C until the point E on the line A - B is reached, when the flow changes into stream-line once more.

In stream-line flow the friction loss is due entirely to the viscosity of the fluid, and for a fluid whose viscosity is negligible the loss of pressure for any size or length of duct will also be negligible.

On the other hand, during turbulent motion, loss of pressure is incurred as the result of whirling and eddies, in addition to the viscosity loss. Indeed, when the flow is entirely turbulent, the resistance is wholly due to eddy formation.

The equation during the period A to B, Fig. 36, is

\[ P = KV \]

where \( P \) = loss of pressure due to frictional resistance to the flow of the air;
\( V \) = velocity
and \( K \) = a coefficient.

During the period C to D (or D, C, E) the equation is

\[ P = nFBI \]

Under the conditions in which the flow may be either stream-line or turbulent, it is necessary in determinations to assume the latter, since it results in the greater loss. In the working of a mine it is possible, we think, that the velocity may be stream-line at some parts and turbulent at/

1. Maxwell's definition of viscosity is as follows:— "The viscosity of a substance is measured by the tangential force on unit area of either of two horizontal planes of indefinite extent at unit distance apart, one of which is fixed, while the other moves with unit velocity, the space between being filled with the viscous substance." The viscosity of air varies slightly with the temperature and increases about 10 per cent. from 40° to 80° Fahr.
at others, while a third state may result in partial turbulence.

**Factors Influencing the Critical Velocity of Flow.**

From the Reynolds formula

\[
\frac{S V_c D}{n} = \text{constant},
\]

we have by transposition

\[
V_c = \text{constant} \frac{S}{D^n}
\]

Consequently the critical velocity is proportional to the viscosity and inversely proportional to the density and the diameter.

Lacey gives the following values of the critical velocity of flow of air in small pipes:

<table>
<thead>
<tr>
<th>Internal Diameter of Pipe</th>
<th>Critical Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inches</td>
<td>Feet per second</td>
</tr>
<tr>
<td>1/4</td>
<td>17.68</td>
</tr>
<tr>
<td>3/8</td>
<td>8.34</td>
</tr>
<tr>
<td>1/2</td>
<td>5.89</td>
</tr>
<tr>
<td>1</td>
<td>4.42</td>
</tr>
</tbody>
</table>

From these figures the following formula may be constructed:

\[
V_c = \frac{0.36}{D} \quad \text{feet per second},
\]

where \( D \) is in feet.

It appears to be established that this relationship between \( V_c \) and \( D \) is true for pipes of small diameter. It is unlikely, however, that it holds good for ducts of large dimensions such as mine airways, although at present there are little or no data available on the question.

From the work of Poiseuille, Lacey and Storrow, it appears to be established that, when air is flowing through capillary tubes, pipes of small diameter or narrow passages such as occur in the waste or goaf of a mine, the loss /

---

loss of pressure is directly proportional to the velocity (or quantity) i.e.

\[ P = KV \]

In a series of experiments carried out by Dr. Penman and the writer, on the flow of air through perforated wooden discs, we obtained the relationship

\[ P = KV \]

As already stated, page 70, at a certain velocity a state of instability exists and the motion changes from simple stream-line to turbulent flow. During this state the index of \( V \) (or \( Q \)) approximates to 2, that is, the friction, and consequently the pressure required to overcome it varies approximately as the square of the velocity (or quantity). Osborne Reynolds has shown that the loss of head due to friction of water is proportional to a power of \( n \) of the discharge, which varies according to the roughness of the walls of the pipe. Thus the index \( n \) will be equal to 1.7 for very smooth tubes, 1.72 for lead tubes and only attains the value of 2 for those tubes of which the walls are very rough. Professor H. Briggs obtained indices of 1.62 and 1.37 in tests of the uniform flow of air in Smoke-helmet tubes. Numerous workers have shown that the value of the index \( n \) approximates to 2 for the conditions that usually exist in connection with laboratory fans.

In connection with mine airways, or a whole mine, the results obtained by previous workers would appear to indicate:

indicate that the ventilating pressure does not vary
as the square of the velocity (or quantity), but that the
index of $V$ (or $Q$) obtains to a value somewhere between 1
and 2. For instance, J. Bouvat-Martin found that the loss
of head was proportional to $Q^{1.75}$ for the Creal Mine, France.
Again, Dr. Penman and the writer obtained a value of 1.8
for $n$ at Wellesley Colliery, Fifeshire, while at an Indian
mine the value of the index was found to be 1.78 (see page 48).
More recently, Professor D. Hay and Mr. W. Cooke, in a
series of carefully conducted experiments with delicate
instruments obtained indices increasing from 1.6 to 1.9
for the conditions existing in underground airways.

From published results, therefore, it would
appear that the value of the index $n$ is somewhere between
1 and 2. In most of the above quoted tests, the total
volume passing has been measured, but the observed pressure
only has been taken into account. It is necessary to
determine the total ventilating pressure as well as the
total volume in order to evaluate $n$. Again, the value
of the index appears to depend upon the period of
instability; if this period be short, $n$ may obtain to a
value much less than 2, but if this period be long and at
the same time a large volume is passing due to natural
causes, the value of $n$ may be well above 2. This will
receive further consideration at a later stage - see page 82.

2. "Underground Tests on the Flow of Air at Rockingham
PART V.

EXPERIMENTS ON THE FLOW OF AIR IN MINES.

(A) THE EQUATION OF THE MINE CHARACTERISTIC CURVE;
(B) THE ZONE OF UNSTABLE FLOW;
(C) NATURAL VENTILATION;
(D) RESISTANCE OF STATIONARY EXHAUSTING FANS.
PART V.

INTRODUCTION.

As already stated (page 49) the experiments to be described were carried out at twelve mines during the period June, 1925 to May, 1926. The object of this research was fourfold, viz.:

1. To determine the relationship between the total volume of air circulating per second, \( V \), and the ventilating pressure, \( p \);

2. To ascertain how closely or otherwise the observed results satisfy the orthodox relationship,

\[
p \propto \frac{1}{Q}
\]

3. To elucidate, in some measure, the influence of natural ventilation;

and

4. To find the resistance of mine ventilators when stationary.

The "total volume" is that quantity with which the fan has to deal; it includes surface leakage, and is expressed throughout in Kilocusecs. The ventilating pressure is the difference between the pressure in the fan drift and that of the external atmosphere. This is the "dynamic" gauge and is expressed in pounds weight per square foot. The curve connecting these two factors is termed the Mine Characteristic (see page 38). The method of constructing this curve consists of using volumes as abscissae and pressures, or depressions, as ordinates.

Table IV gives some particulars of the collieries at which tests were conducted. All the mines are operated by means of vertical shafts with the exception of the Easthouses Colliery which is approached by means of inclines from the surface. Two mines, namely, Arniston and Hylton, are provided with two downcasts and one upcast shaft. The table /

1. The Kilocusec is defined as a flow of 1000 cubic feet of air per second.
<table>
<thead>
<tr>
<th>No. of Line</th>
<th>Name and Situation of Mine</th>
<th>Type of Fan</th>
<th>Details of Fan</th>
<th>No. of Underground Fans</th>
<th>Connections to Other Mines</th>
<th>No. of Main Splits</th>
<th>Name of Seam</th>
<th>Height</th>
<th>Gradient</th>
<th>Method of Working</th>
<th>Downcast</th>
<th>Upcast</th>
<th>Approx. Weekly Tonnage</th>
<th>Further Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Armiton Midlothian</td>
<td>Sirocco</td>
<td>Double Inlet</td>
<td>77&quot; Diam. 65&quot; Wide</td>
<td>2</td>
<td>None</td>
<td>Great Seam</td>
<td>6' 3&quot;</td>
<td>1 in 5</td>
<td>Longwall</td>
<td>490' Deep</td>
<td>350' Deep</td>
<td>7,000</td>
<td>An Old Colliery having Extensive Workings</td>
</tr>
<tr>
<td>2</td>
<td>Easthouses Midlothian</td>
<td>Sirocco</td>
<td>Propeller</td>
<td>70&quot; Diam. 5 Blades</td>
<td>1</td>
<td>Connected to Newbattle, Colliey &amp; on old workings</td>
<td>Great Seam</td>
<td>6' 3&quot;</td>
<td></td>
<td>Bred and Pillar</td>
<td>Incline from Surface</td>
<td>Incline from Surface</td>
<td>1,500</td>
<td>A New Colliery</td>
</tr>
<tr>
<td>3</td>
<td>Polkemmet Lithgow</td>
<td>Sirocco</td>
<td>Propeller</td>
<td>105&quot; Diam. 84&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>Wilsonstown Main Coal</td>
<td>4'</td>
<td>1 in 30</td>
<td>Longwall</td>
<td>157' Deep</td>
<td>157' Deep</td>
<td>6,000</td>
<td>Extensive Workings</td>
</tr>
<tr>
<td>4</td>
<td>Preston Links Haddington</td>
<td>Howden</td>
<td>Double Inlet</td>
<td>156&quot; Diam. 81&quot; Wide</td>
<td>3</td>
<td>None</td>
<td>Great Seam</td>
<td>3' 0&quot;</td>
<td>1 in 4</td>
<td>Longwall</td>
<td>400' Deep</td>
<td>400' Deep</td>
<td>6,000</td>
<td>Old Colliery Extensive Workings</td>
</tr>
<tr>
<td>5</td>
<td>Dunnikier Fife</td>
<td>Waddle</td>
<td>Single Inlet</td>
<td>318&quot; Diam. 12&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>Jersey Longgelly Split Parrot Five Foot</td>
<td>2' 0&quot;</td>
<td>2' 11&quot;</td>
<td>1 in 4</td>
<td>Longwall</td>
<td>701' Deep - 16&quot; x 7'</td>
<td>Timber &amp; Rock Sides</td>
<td>Very Wet</td>
</tr>
<tr>
<td>6</td>
<td>Kinglassie Fife</td>
<td>Walker</td>
<td>Double Inlet</td>
<td>216&quot; Diam. 84&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>Dunfermline Split</td>
<td>3' 6&quot;</td>
<td>2' 9&quot;</td>
<td>1 in 8</td>
<td>Longwall</td>
<td>1350' Deep - 20&quot; x 14&quot;</td>
<td>Timber &amp; Brick Lining</td>
<td>Wet</td>
</tr>
<tr>
<td>7</td>
<td>Valleyfield Fife</td>
<td>Walker</td>
<td>Double Inlet</td>
<td>216&quot; Diam. 84&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>Jewel Diamond Five Foot Dunfermline Split</td>
<td>4' 6&quot;</td>
<td>4' 3&quot;</td>
<td>1 in 3</td>
<td>Longwall</td>
<td>1150' Deep - 400 sq. ft</td>
<td>Elliptical Brick Lining</td>
<td>Dry</td>
</tr>
<tr>
<td>8</td>
<td>Wellesley Fife</td>
<td>Waddle</td>
<td>Sirocco</td>
<td>252&quot; Diam. 15&quot; Wide</td>
<td>4</td>
<td>Connected to several mines</td>
<td>Barn Craig Chemiss Bowhouse Dysart Main</td>
<td>4' 6&quot;</td>
<td>7' 0&quot;</td>
<td>1 in 6</td>
<td>Longwall</td>
<td>1554' Deep - 27&quot; x 10'</td>
<td>Elliptical Brick Lining</td>
<td>Dry</td>
</tr>
<tr>
<td>9</td>
<td>Hylton Co Durham</td>
<td>Sirocco</td>
<td>Double Inlet</td>
<td>91&quot; Diam. 63&quot; Wide</td>
<td>2</td>
<td>None</td>
<td>Maudlin Hutton Harvey</td>
<td>5' 6&quot;</td>
<td>4' 6&quot;</td>
<td>1 in 20 to 1 in 30</td>
<td>Bred and Pillar</td>
<td>Bred and Pillar</td>
<td>176' Deep - 12&quot; Diam</td>
<td>Brick Lining - Dry</td>
</tr>
<tr>
<td>10</td>
<td>Silksworth Co Durham</td>
<td>Capell</td>
<td>Double Inlet</td>
<td>144&quot; Diam. 72&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>Five Quarter Maudlin Hutton</td>
<td>3' 0&quot;</td>
<td>4' 0&quot;</td>
<td>1 in 18</td>
<td>Bred and Pillar</td>
<td>1744' Deep - 16&quot; x 9&quot;</td>
<td>Brick Lining</td>
<td>14,000</td>
</tr>
<tr>
<td>11</td>
<td>Coventry Warwick</td>
<td>Sirocco</td>
<td>Single Inlet</td>
<td>175&quot; Diam. 55&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>Two Yate Bar Coal (Yo) Ryder ELL COAL Slate Coal</td>
<td>6' 4&quot;</td>
<td>6' 6&quot;</td>
<td>1 in 10</td>
<td>Modified Longwall, Retreating</td>
<td>213' Deep - 21&quot; x 4&quot;</td>
<td>Brick, Tubbing and Concrete Lining Semi-Dry</td>
<td>213' Deep - Otherwise Same as Co</td>
</tr>
<tr>
<td>12</td>
<td>Craven Warwick</td>
<td>Sirocco</td>
<td>Double Inlet</td>
<td>49&quot; Diam. 45&quot; Wide</td>
<td>None</td>
<td>None</td>
<td>255' Deep - 9 Diameter</td>
<td>255' Deep - 9 Diameter</td>
<td>145' Deep - 8 Diameter</td>
<td>Brick Lining</td>
<td>3,000</td>
<td>An Old Colliery</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**TABLE IV:** Particulars of Collieries
table includes collieries at different stages of development. Polkemmet, for instance, is a comparatively new concern (shafts completed in 1924). Coventry colliery, though recent, is approaching full output. Many of the others are at full development, while Craven and Dunnikier Collieries provide examples of mines approaching exhaustion.

Of the surface ventilators, six, all Siroccos, have their blades curved forward, and vary from 49 inches to 175 inches in diameter; six are of types having their blades bent back at the periphery of the wheel and ranging from 13 feet to 26 feet 6 inches in diameter, while that at No. 2 mine is a 70 inch diameter Sirocco propeller fan having five blades. By conducting the tests at week-ends and holidays it was found practicable to stop all underground fans during the test period, with the exception of two at Prestonlinks colliery which had to continue running. These, however, were stopped during the measurement of natural ventilation. At eight of the collieries the ventilating system is self-contained. Wellesley and Easthouses Collieries are in communication with neighbouring mines, while Easthouses and Dunnikier are connected to old workings from which a certain quantity of air enters the present airways.

The thickness and inclination of the seams worked show considerable variation, the former varying from 1 foot 11 inches to 18 feet 3 inches, while the latter ranges from 38 degrees to practically level. The methods of working the seams also vary considerably. Half the number of mines are operated by means of safety lamps, while in the other half naked lights are used.

Three of the mines have steam pipes in the shafts. In Nos. 6 and 7 they are in the downcast, while in No. 3 the pipe is in the upcast shaft.
Method of Conducting the Tests.

In carrying out one of these tests, five observers were required. The duty of the first of these was to record the pressure observations; the second to note the speed of the fan; the third to measure the rate of flow; the fourth as time-keeper for the latter measurements; while the fifth was occupied taking barometric and hygrometric observations underground. The second of these observers was responsible for changing the fan-speed on a pre-arranged plan, and the third also recorded barometric and hygrometric observations inside the fan drift and in the external atmosphere. The aneroid barometers and the thermometers used were calibrated against standards in the Mining Laboratory, or compared, previous to the test, with instruments which had been so calibrated. The whirling type of hygrometer was used and the desired information obtained by reference to Marvin's Tables.

After the preliminary arrangements had been completed and all watches synchronised, the person entrusted with the underground observations proceeded down the pit, and generally remained there throughout the test, noting the conditions from time to time. The first set of readings was taken with the fan running at normal speed; the speed was then altered to another rate and the manometer watched continuously until it had settled down to a steady reading, when another set of observations was taken.

Generally /

Generally on two occasions during the test period the fan was stopped in order to measure the effect of natural agencies acting alone. An interval of about 15 minutes elapsed between each set of readings and a set usually occupied about 10 minutes. Since a complete series of observations from which the "mine characteristic" could be drawn was intended to relate to a particular condition of the outside atmosphere, and that condition would change too much if the test period was extended over many hours, time was an important factor. In a perusal of the Transactions of the Institution of Mining Engineers one will find many authorities who question the accuracy of such observations taken only a few minutes after changing the speed of the fan. Fortunately, however, during preliminary work it was found that the rate of flow quickly adjusted itself after changing the fan speed, and, indeed, that the interval between any two sets of observations did not need to exceed 15 minutes. This conclusion has recently been substantiated by the work of Mr. R.A.H.Flugge-de-Smidt. During a test on an underground fan (63 inch diameter Sirocco) at the Modder Deep Mine, Witwatersrand, this worker was able to demonstrate that, even in a deep (3000 feet) and extensive mine, the ventilation readjusts itself very rapidly when the fan-speed is altered. Indeed, he found that, 15 minutes after stopping the fan, the rate of flow was uniform.

Measurement of the Rate of Flow.

The principal difficulty in regard to this measurement was that of obtaining a place in the fan drift at which the flow was sufficiently uniform. As is well known, pulsation and eddying occurs near the fan drift inlet and the inlet to the fan and measurements taken near these

these places are likely to be erroneous. Generally, the rate of flow was measured at a point 40 feet from the fan although at mines Nos. 1, 5, 9 and 12 the distance was somewhat less than this. Under the adverse conditions met with in most fan drifts, we felt that a highly elaborate method of measuring the rate of flow would lose time to no useful purpose and that the simplest method was likely to prove the best. It was, therefore, decided to use the anemometer throughout; to support it on the end of a thin wooden rod and move it in a regular manner following a zig-zag path over the area of the fan drift. It is well known that irregular flow causes the anemometer to read too "high", hence, if anything, the volumes obtained are too great. Each measurement was taken two or three times. A zero-setting anemometer was used and was frequently calibrated in the Mining Laboratory of this University.

Measurement of the Depression.

Pulsating and eddying flow also introduce difficulty in the measurement of pressure differences, though by taking precautions, it is possible to reduce the effects considerably. The depression was obtained by means of a delicate inclined manometer similar to that already described on page 12. The inclination was measured by means of a clinometer and was generally set to give a magnification of 10 to 1. During a set of readings the manometer was under continual observation; it was read at short intervals, first with one limb and then with the other limb connected to the fan-drift. The mean was taken of the considerable number of readings so obtained. The relatively large bulk and inertia of the liquid (petrol of specific gravity 0.758) in the tubes and the small diameter (¼ inch) of the "Compo" piping connecting the manometer to the point of observation in the fan drift had the effect of damping out much of the fluctuation. In the drift/
drift the "comp" piping was connected to a smooth-mathed tube pointing directly to windward, placed mid-high and at one-seventh of the width of the fan drift from one side; all readings were therefore of the "dynamic" or total gauge.

Fig. 37
Equations of the Mine Characteristic.

$P = 3Q^2$

$P = 2.19Q^2 + 7.75Q - 15$

$P = 2.1Q^{2.36}$

$P = 2.19Q^2 + 7.75Q - 15$

Figure 37.

Volume of Air in Kilocusecs.
SECTION (A)

THE EQUATION OF THE MINE CHARACTERISTIC CURVE.

In order to simplify description of Figures 38 to 61, which have been plotted from observed data, we have chosen the curve appertaining to the conditions existing at Silksworth Colliery, Co. Durham, on 20th December, 1925, for detailed consideration. The graph ABCD, Figure 37, was obtained in the manner already described on page 74. The normal working of the fan (speed 246 R.P.M.; quantity, 196200 cubic feet of air per minute against a pressure of 33.44 lbs. per sq. ft) is indicated by D. The quantities represented by the various points in the Figure are those due to the fan and natural ventilation acting together. The latter was assisting the fan. The point A was obtained when the fan was stationary and natural ventilation acting alone.

An inspection of the various points in the Figure shows that:

(i) Whatever be the relation between p and Q there is a change of law at the point B; the lower part of the graph being less steep than the top portion. This change in the relationship can be detected in most of the curves shown in Figures 38 to 49. It is more distinct in some cases than in others. The cause of this change of law affecting the flow of air through mine passages is discussed in the next section (page 88).

(ii) The effect of natural ventilation operating in series with the fan causes the mine characteristic to meet the horizontal axis to the right of the origin, C. In the case under consideration, 1 Kilocusec was passing when the manometer recorded zero pressure inside the fan-drift. If the lower part, BA, be produced it would meet the vertical axis below the origin. This means that a negative pressure would be.

1 Pressures below the horizontal axis are really positive, but for the present purpose we consider them as being negative, and pressures above that axis as being positive.
be required to stop the flow of air in the fan drift.

(iii) As the point A indicates, there was a small negative pressure in the drift when the fan was stationary; in other words, the pressure inside the fan drift was slightly greater than that of the external atmosphere. This is due to the fact that the fan had a certain resistance and that air was flowing through it (see Section D page 101).

The equation of the principal part of the curve, namely, BCD is of great importance. When it is obtained, the full ventilating pressure required to circulate a given volume of air can be determined for the then existing conditions of the mine. There are five different ways of writing this equation and we now propose to consider these various forms separately.

(a) Equations of the Types: - \( p = RQ \) and \( p = RQ \)^2

The Ventilation Committee set up by the Institution of Mining Engineers recommended the adoption of these two forms of equation (see page 48). From the graphs shown in Figures 38 to 61 it is obvious that these types can no longer be accepted, since all curves having these equations must pass through the origin. Such a curve as BCD, Figure 37, for instance, lies a long way to the right of that point.

The curve \( p = 3Q \) is, for the series considered (Figure 37), as good as any of its type; it is shown in red. The curve nowhere resembles that of the line BCD, and in fact, the disparity is so great that the equation is useless.

During the summer months, when natural ventilation may be negligible, data may be obtained which agree fairly well with \( p = RQ \). The summer curves for Prestonlinks (Figure 53) and Dunnikier (Figure 54), are almost \( p = 3.2Q \) and \( p = 3.9Q \) respectively. Generally speaking, however, only at a small number of the mines included in Table IV, and then only/
only when natural agencies are inoperative, does this
form of equation hold good even over a small portion of
the mine characteristic. Hence, for the purpose of
representing that curve, the form \( p = R^2 \) may be dismissed
as worthless.

Equations of the more general form \( p = R^n \) can
be found to fit short portions of any of the curves shown
in Figures 38 to 49, though the results, when obtained,
appear to be practically meaningless. In the few
available cases, where this relationship has been applied
by previous workers on the subject to data obtained on
mines as a whole, the value of the index \( n \) appears to have
been 1.7 and 1.9 (see page 73). The reason for the value
being less than 2 is usually attributed to the flow of air
through the network of passages constituting a mine
ventilating system not being wholly turbulent, but a
mixture of stream-line and turbulent flow. When considered
alone, however, it obviously does not explain why the
best equation of this form for the portion, CD, of the
curve covering the fans' working range in the Silksworth
series (see Figure 37), comes to be

\[
p = 2.1Q^2
\]

nor why the indices for similar parts of the winter
curves for Craven (Figure 49), Wellesley (Figure 45),
Polkemmet (Figure 40), Coventry (Figure 48) and Hylton
(Figure 46) are 2.3, 2.5, 3.2, 3.5 and 4.0 respectively.

The principal factors determining the value of
the index appear to be:

(i) the position of the "kink" at B, Figure 37, with regard to the origin;

and (ii) the steepness of the curve above that point.

The main part of the mine characteristic commences
at B, Figure 37. When this point lies well to the right
of the origin and we try to represent any part of the
curve by an expression of the form \( p = R^n \), a high value
for \( n \) is obtained. When, in addition, the curve is
steep, \( n \) becomes bigger still. The greater the mine
resistance
resistance the steeper the curve. There are two influences which affect the position of B, namely:

(i) the duration of the period of unstable flow (see BC, Figure 36);

and (ii) a large natural ventilation assisting the fan.

With regard to (i), if, for example, the curve BCD, Figure 37, had been shifted 1 unit of quantity to the left (i.e. had B been a point where \( Q = 0.65 \) instead of 1.65), the value of \( n \) in the expression \( p = RQ^n \) most suitable for the upper half of the curve would have been 1.54 instead of 2.36. If, on the other hand, BCD had been moved bodily 1 unit of quantity to the right, \( n \) would attain to a value of 3.25. The value of \( R \) decreases as \( n \) increases, when \( n \) is large because the graph is steep, \( R \) is small. Hence \( R \) in this form as equation cannot be a measure of the resistance of the mine, for cases can be selected in which a high value of \( R \) corresponds to a low resistance and vice versa.

Messrs T.M. Stanton and J. R. Pannel, dealing with the resistance to flow of water through pipes, have observed that for the exponential form of law "it may be taken as fully demonstrated that an index law for surface friction cannot be devised which will express the fact with any accuracy except over a comparatively small range in the value of \( \frac{Vd}{v} \),"

where \( V = \) mean velocity, \( d = \) diameter of pipe, \( v = \) viscosity.

From the foregoing it is obvious that neither the form \( p = RQ^n \) nor \( p = RQ \) is of much use in expressing the relationship between \( p \) and \( Q \) existing at most mines; and further, that when an equation of the latter type has been obtained to satisfy the observed data over a limited range, no useful meaning can be attached to \( R \) or \( n \).

(b) Equations of the Type: \[ p = Aq + Bq - C. \]

As the result of experiments on the spinning of vanes at different velocities \((V)\) in still air, J.C. Fairweather found that the force required to keep the vanes in motion could be represented by an equation of the form

\[ F = a_1 V + a_2 V^2 + a_3 V^3 + \ldots \]

where \(a_1, a_2\) and \(a_3\) are coefficients.

As already mentioned (page 49), Professor Gibson has suggested that an expression similar to Fairweather's may satisfy the observed data. Although it is possible to obtain an equation of the form

\[ p = Aq + Bq - C \]

to fit a few of the graphs shown in Figures 38 to 49 very closely, as, for example, the Winter and Spring curves for Kinglassie Colliery, we are convinced, after many trials, that the terms in the Fairweather series involving higher powers than the "square", may, for mining purposes, be disregarded. Again, if natural agencies are operating, the mine characteristic cannot pass through the origin. Hence, any equation, to be satisfactory, must contain a term which will include this important factor. These considerations lead us to the expression suggested by Professor H. Briggs, namely,

\[ p = Aq + Bq - C \]

where \(A\) and \(B\) are coefficients and \(C\) is the pressure due to natural causes. \(C\) will be negative when natural ventilation is assisting the fan and positive when this is acting adversely. This form of equation is, of all those we have tried, the one which most exactly satisfies the requirements and in many of the curves shown in Figures 38 to

2. See "Fan Problems II", Colliery Engineering, June, 1925.
Characteristic Curves for Arniston Colliery.

Figure 38.

Winter Average Surface Temp. (F) -41.2°F
Spring Average Surface Temp. (F) -48.6°F
Characteristic Curves for Easthouses Colliery

\[ \rho = 3.00Q^2 + 2.10Q - 0.94 \]

\[ \rho = 1.85Q^2 + 0.09Q - 0.1\]

Volume of Air in Kilolitres

Pressure in Pounds per Sq. Ft.

Average Surface Temp. = 35.0°F

Average Surface Temp. = 51.17°F

Figure 39
Figure 40
Characteristic Curves for Polkemmet Colliery

Volume of Air in Kilolitres

Winter Average Surface Temperature = 32°
  = 46.64°F.

Spring Average Surface Temp. = 57.79°F.
Characteristic Curves for Prestonlinks Colliery

$p = 1.20Q^2 + 80Q - 4.10$

$p = 1.40Q^2 + 4.90Q - 3.0$

$p = 1.25Q^2 + 4.50Q - 5.10$

Volume of Air in Kilocusecs

Figure 41

Summer — Average Surface Temp. = 65.5°F.  Temperature = 63°F.
Winter — Average Surface Temp. = 39.95°F.  "
Spring — Average Surface Temp. = 44.92°F.  "
Figure 42.
Characteristic Curves for Dunnikier Colliery.

Pressure in Pounds per Sq. Ft.

Volume of Air in Kilocusecs.

Figure 42.

Summer — Average Surface Temp. = 60.65°F.
Winter — " " " = 37.90°F.
Spring — " " " = 46.85°F.
Characteristic Curves for Kinglassie Colliery

\[ p = 7.16Q^2 + 0.5Q - 0.66 \]

\[ p = 7.03Q^2 - 0.06Q - 0.25 \]

Volume of Air in Kiloliters.

Figure 43

Winter – Average Surface Temp. = 40.70°F
Spring – “ U.” = 44.99°F
Characteristic Curves for Valleyfield Colliery.

\[ p = 2.3Q^2 + 1.04Q - 4.08 \]

\[ p = 2.0Q^2 + 2.14Q - 1.91 \]

\[ p = 1.6Q^2 + 4.2Q - 4.9 \]

Volume of Air in Kilolitres.

Figure 44.

Summer

Average Surface Temp. = 63.25°F

Winter

" " = 33.64°F

Spring

" " = 49.80°F
Characteristics Curves for Wellesley Colliery

\[ p = 4.5Q^2 + 2.15Q - 1.71 \]
\[ p = 4.4Q^2 + 3.28Q - 5.06 \]
\[ p = 5.0Q^2 + 0.07Q - 2.32 \]

Volume of Air in Kilocusecs.

**Figure 45.**

Summer  Average Surface Temp. = 65°F.
Winter   "       "       "       "       = 35.75°F.
Spring   "       "       "       "       = 44.26°F.
Characteristic Curve
For
Hilton Colliery.

\[ p = 2.55Q^2 + 0.062Q - 5.83 \]

Volume of Air in Kilocusecs.

Figure 46.

Winter — Average Surface Temperature = 35.4°F.
Characteristic Curve for Silksworth Colliery

$$p = 2.19 Q^2 + 7.75 Q - 15$$

Volume of Air in Kilocusecs.

Figure 47.

Winter —— Average Surface Temperature = X

= 36.16°F
Characteristic Curve for Coventry Colliery

\[ p = 12Q^2 + 1160Q - 1200 \]

Volume of Air in Kilocusecs

Figure 48

Winter — Average Surface Temperature

-41.3°F
Figure 49.
Characteristic Curve for Graven Colliery.

\[ p = 7.50Q^2 + 16.25Q - 6.55 \]

\[ p = 5.00Q^2 + 8.00Q - 2.10 \]

Figure 49

Winter — Average Surface Temp.

= 37.23°F.
to 49 inclusive, the agreement between it and the observed points is very close indeed. For Silksworth Colliery it takes the form
\[ p = 2.19q^2 + 7.75q - 15 \]
which is the equation of the full line BCD, (Figure 37). If produced downwards that curve will meet the vertical axis at \( F \), namely, at a point 15 units of pressure below the origin.

(c) Equations of the Type: \( p = R^q - C \).

As was recently pointed out by Professor Briggs, an exponential form of equation can be found to satisfy a set of readings very closely, if it be written:
\[ p = R^q - C. \]

The term \( C \) is the same as in the previous form of equation. In Figure 37, the curve represented by the expression
\[ p = 9.16q^{1.41} - 15 \]
satisfies the requirements almost as well as
\[ p = 2.19q^2 + 7.75q - 15 \]
and when produced downwards also meets the vertical axis at \( F \). In fact, the agreement is so close that for the part BCD, the full line may be taken as representing both these forms. The quadratic form of expression is the more logical and the easier to obtain from a set of observations. The exponential form has the advantage of providing a single factor \( R \) to take the place of the two coefficients \( A \) and \( B \). It must be noted, however, that if it is desired to use \( R \) as a measure of the resistance of the mine, the value of the index \( n \) must be given, otherwise it is practically meaningless.

(a) Equations of the Type: \( p = R^q - K \).

This form is a modification of the old orthodox "square" law, and although it does not satisfy the requirements/
CHARACTERISTIC CURVES FOR ARNISTON COLLIER Y

\[ p = 4.06 Q^2 - 1.84 \]

\[ p = 3.04 Q^2 - 2.36 \]

Volume of Air in Kilocusecs.

Figure 50.

Winter Average Surface Temp. = 41.2°F
Spring

\[ n \]
CHARACTERISTIC CURVES FOR EASTHOUSES COLLIERY.

\[ p = 3.25Q^2 - 0.58 \]

VOLUME OF AIR IN KILOCUSECS.

Figure 51

Winter Average Surface Temp. -15°F
Spring Average Surface Temp. -51.2°F
CHARACTERISTIC CURVES
FOR
POLKEMMET COLLIERY

Figure 52.

Winter — Average Surface Temp. = 46.6°F
Spring — Average Surface Temp. = 57.7°F

\[ p = 4.2Q^2 - 4.2 \]

\[ p = 3.68Q^2 - 3.24 \]
CHARACTERISTIC CURVES FOR PRESTONLINKS COLLIERY

\[ p = 2.8Q^2 - 0.90 \]

\[ p = 3.22Q^2 \]

\[ p = 2.32Q^2 - 0.78 \]

VOLUME OF AIR IN KILOGUSECS.

Figure 53.

Summer
Average Surface Temp. \( p = 65.5^\circ F \)
Winter
\( p = 39.99^\circ F \)
Spring
\( p = 44.92^\circ F \)
CHARACTERISTIC CURVES
FOR
DUNNIKER COLLIERY - FIFE.

PRESSURE IN POUNDS PER SQUARE FOOT.

VOLUME OF AIR IN KILOCUSECS.

Figure 54

Summer - Average Surface Temperature = 60.49°F

Winter -  40.76°F

Spring -  46.85°F

\[ P = 4.531Q^2 - 0.42 \]

\[ P = 3.875Q^2 - 0.02 \]
CHARACTERISTIC CURVES FOR KINGLASSIE COLLIEY - FIFE.

VOLUME OF AIR IN KILOCUSECS.

Figure 55.

Winter — Average Surface Temperature = 40.70°F.
Spring — Average Surface Temperature = 44.98°F.
Figure 56.

Summer  Average Surface Temperature=63.25°F.
Winter  Average Surface Temperature=33.64°F.
Spring  Average Surface Temperature=49.30°F.
Characteristic Curves for Wellesley Colliery - Fife.

Figure 57.

Summer — Average Surface Temperature = 6\degree C

Winter —

Spring —

Pressure in pounds per square foot.

Volume of air in kilocusecs.

\[ P = 5.25 Q^2 - 0.25 \]

\[ P = 5.45 Q^2 - 2 \]

\[ P = 4.67 Q^2 - 1.5 \]
Fig. 58.
Figure 58.

Winter — Average Surface Temperature = 35.4°F.
CHARACTERISTIC CURVES FOR SILKSWORTH COLLIERY - DURHAM.

\[ p = 3.91Q^2 - 6.8 \]

Volume of Air in Kilocusecs.

Figure 59.

Average Surface Temp. = 36.16°F.
CHARACTERISTIC CURVE
FOR
COVENTRY COLLIERY-WARWICK.

\[ P = 18.5Q^2 - 7.44 \]

Figure 60

Winter ——— Average Surface Temp. = 41
Figure 61
CHARACTERISTIC CURVE FOR CRAVEN COLLIERY—WARWICK

\[ P = 21Q^3 - 1.6 \]

Volume of Air in Kilorusecs.

Figure 61

Winter — Average Surface Temp. = 37°
<table>
<thead>
<tr>
<th>No.</th>
<th>NAME OF MINE</th>
<th>PERIOD OF YEAR</th>
<th>MEAN TEMP. OF EXTERNAL AIR</th>
<th>TYPES OF EQUATIONS</th>
<th>DEGREE OF AGREEMENT OF P = RG² – K WITH OBSERVED DATA</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>ARNISTON</td>
<td>WINTER</td>
<td>41.27</td>
<td>( p = 2.81Q^2 + 0.88Q - 3.16 )</td>
<td>( p = 3.57Q^2 - 3.16 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>49.03</td>
<td>( p = 1.44Q^2 + 2.02Q - 7.62 )</td>
<td>( p = 9.30Q^2 - 7.62 )</td>
</tr>
<tr>
<td>2</td>
<td>EASTHOUSES</td>
<td>WINTER</td>
<td>35.00</td>
<td>( p = 1.0Q^2 + 3.0Q - 4.19 )</td>
<td>( p = 5.92Q^2 + 4.19 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>51.17</td>
<td>( p = 2.00Q^2 + 1.10Q - 1.41 )</td>
<td>( p = 4.14Q^2 - 1.41 )</td>
</tr>
<tr>
<td>3</td>
<td>POLKEMMET</td>
<td>WINTER</td>
<td>46.64</td>
<td>( p = 0.42Q^2 - 0.92Q - 1.90 )</td>
<td>( p = 11.35Q^2 - 1.90 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>57.79</td>
<td>( p = 1.56Q^2 + 6.50Q - 8.15 )</td>
<td>( p = 6.02Q^2 - 8.15 )</td>
</tr>
<tr>
<td>4</td>
<td>PRESTONLINGS</td>
<td>WINTER</td>
<td>38.95</td>
<td>( p = 1.2Q^2 + 4.80Q - 4.10 )</td>
<td>( p = 5.94Q^2 - 4.10 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>44.92</td>
<td>( p = 1.25Q^2 + 4.50Q - 5.10 )</td>
<td>( p = 5.66Q^2 - 5.10 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SUMMER</td>
<td>65.50</td>
<td>( p = 1.40Q^2 + 4.90Q - 3.00 )</td>
<td>( p = 6.32Q^2 - 3.00 )</td>
</tr>
<tr>
<td>5</td>
<td>DUNNIKER</td>
<td>WINTER</td>
<td>37.90</td>
<td>( p = 3.44Q^2 + 0.83Q - 0.73 )</td>
<td>( p = 4.36Q^2 - 0.73 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>46.95</td>
<td>( p = 1.88Q^2 + 5.87Q - 3.55 )</td>
<td>( p = 7.73Q^2 - 3.55 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SUMMER</td>
<td>60.65</td>
<td>( p = 3.44Q^2 + 0.87Q - 0.41 )</td>
<td>( p = 4.36Q^2 - 0.41 )</td>
</tr>
<tr>
<td>6</td>
<td>KINGLASSIE</td>
<td>WINTER</td>
<td>40.70</td>
<td>( p = 7.16Q^2 + 0.50Q - 0.66 )</td>
<td>( p = 7.66Q^2 - 0.66 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>44.98</td>
<td>( p = 7.03Q^2 + 0.06Q - 0.25 )</td>
<td>( p = 7.06Q^2 - 0.25 )</td>
</tr>
<tr>
<td>7</td>
<td>VALLEYFIELD</td>
<td>WINTER</td>
<td>33.64</td>
<td>( p = 2.50Q^2 + 1.04Q - 4.08 )</td>
<td>( p = 3.68Q^2 - 4.08 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>49.80</td>
<td>( p = 1.60Q^2 + 2.00Q - 4.90 )</td>
<td>( p = 5.31Q^2 - 4.90 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SUMMER</td>
<td>68.25</td>
<td>( p = 2.00Q^2 + 2.14Q - 1.91 )</td>
<td>( p = 3.33Q^2 - 1.91 )</td>
</tr>
<tr>
<td>8</td>
<td>WELLESLEY</td>
<td>WINTER</td>
<td>35.75</td>
<td>( p = 4.60Q^2 + 3.28Q - 5.06 )</td>
<td>( p = 7.74Q^2 - 5.06 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SPRING</td>
<td>44.26</td>
<td>( p = 5.00Q^2 + 0.07Q - 2.32 )</td>
<td>( p = 5.39Q^2 - 2.32 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SUMMER</td>
<td>63.00</td>
<td>( p = 4.50Q^2 + 2.13Q - 1.71 )</td>
<td>( p = 6.63Q^2 - 1.71 )</td>
</tr>
<tr>
<td>9</td>
<td>HYLTON</td>
<td>WINTER</td>
<td>35.40</td>
<td>( p = 2.55Q^2 + 0.06Q - 5.83 )</td>
<td>( p = 2.60Q^2 - 5.83 )</td>
</tr>
<tr>
<td>10</td>
<td>SILKSWORTH</td>
<td>WINTER</td>
<td>36.16</td>
<td>( p = 2.19Q^2 + 7.75Q - 15.00 )</td>
<td>( p = 9.16Q^2 - 15.00 )</td>
</tr>
<tr>
<td>11</td>
<td>COVENTRY</td>
<td>WINTER</td>
<td>41.30</td>
<td>( p = 12.00Q^2 + 11.60Q - 12.40 )</td>
<td>( p = 23.93Q^2 - 12.40 )</td>
</tr>
<tr>
<td>12</td>
<td>CRAVEN</td>
<td>WINTER</td>
<td>37.23</td>
<td>( p = 7.50Q^2 + 16.25Q - 6.55 )</td>
<td>( p = 10.00Q^2 - 6.55 )</td>
</tr>
<tr>
<td>13</td>
<td>EAST ARDSLEY</td>
<td>WINTER</td>
<td>54.86</td>
<td>( p = 1.88Q^2 + 6.11Q - 4.49 )</td>
<td>( p = 7.91Q^2 - 4.49 )</td>
</tr>
<tr>
<td>14</td>
<td>LODNA</td>
<td>WINTER</td>
<td>80.00 (CHRP)</td>
<td>( p = 2.03Q^2 + 0.61Q - 0.44 )</td>
<td>( p = 2.74Q^2 - 0.44 )</td>
</tr>
</tbody>
</table>
requirements as well as the quadratic equation, it nevertheless offers, in every case we have tried, a very good approximation. Figures 50 to 61 show the curves following this relationship for all the mines included in Table 4. An inspection of these curves shows that, in many cases, this form of expression satisfies the observed points very closely. For examples, the curves represented by this form for the Winter, Spring and Summer conditions at Wellesley Colliery (Figure 57) satisfy the requirements almost as well as the curves following the quadratic law (Figure 45). The same also applies to the curves for Dunnikier (Figures 42 and 54), Kinglassie (Figures 43 and 55), Valleyfield (Figures 44 and 56), Hylton (Figures 46 and 58), Coventry (Figures 47 and 60) and Craven (Figures 49 and 61).

Providing that the limitations of this form of expression are recognised and that it is not expected to provide results outside the range over which it applies, this type is, we believe, of considerable utility. It has the great advantage of simplicity, is easily obtained from a set of observations and, unlike the previous form \( \bar{p} = R_q - C \), provides a coefficient, \( R \), which is not influenced by a variable index \( n \).

In Table V we have collected the three forms of expression which best satisfy the upper parts of the curves shown in Figures 38 to 61 inclusive. To these have been added those of the East Ardsley Colliery, Yorkshire (winter conditions) and the Lodna Colliery, India. The former observations were conducted by Dr. J. N. Williamsor while the latter were made by Dr. D. Penman and Mr. T. A. Wetherill. At the Lodna mine, three seams of coal, 27, 7 and 27 feet respectively, are being worked and the colliery is ventilated by a double inlet Waddle-Turbon fan.
6 feet 9 inches diameter capable of circulating 100,000 cubic feet of air when making 200 revolutions per minute with a ventilating pressure of 6.5 lbs. per square foot. The East Ardsley Colliery works three seams of coal, 4, 2½ and 3 feet respectively, and is ventilated by a double inlet "Leeds" fan 40 feet in diameter normally circulating 100,000 cubic feet of air per minute.
SECTION (B).

THE ZONE of UNSTABLE FLOW.

We have already dealt with recent experiments on the condition and nature of flow, in Part IV. These communications have familiarised the mining engineer with the conception of stream-line and turbulent flow, of the upper and lower critical velocities, and of the region of unstable flow lying between these critical velocities.

In the present instance, it would appear that we are chiefly concerned with turbulent motion and although stream-line flow may exist in certain parts of the mine where the velocity of flow is very low, or the passage is small in section - as, say, in leakage through a waste or gob - this latter state seems to have very little influence on the conditions as a whole. Assuming ordinary dimensions of fan-drifts, shafts, etc., it is pretty safe to say that, if stream-line flow ever exists, it can only be when the velocity of flow is about 5 feet per minute. Hence, the influence of stream-line flow upon the mine characteristic can be entirely neglected.

The first to recognise that the characteristic curve gave evidence of a "kink" were Messrs Penman and Wetherill (see page 49). These workers found that, for the mine tested, when the volume circulating was below 20,000 cubic feet per minute, the relationship applied, and that, when the volume exceeded 20,000 cubic feet the relationship became

Almost all the curves shown in Figures 38 to 49 a affected by a "kink" (e.g. B. Figure 37), indicating a change of law. Sometimes the change appears to be very sudden, as for example in the Silksworth (Figure 47) and the Folkemmet (Figure 40) series, while in others it is
<table>
<thead>
<tr>
<th>Colliery</th>
<th>Area of Ucast Shaft sq.ft.</th>
<th>Vol. in Ucast ft. per min.</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arniston</td>
<td>48</td>
<td>W. 720</td>
<td>Sp. 600</td>
</tr>
<tr>
<td>Easthouses</td>
<td>None</td>
<td>W. 566</td>
<td>Sp. 446</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Workings approached by inclines from surface.</td>
</tr>
<tr>
<td>Folkeemmet</td>
<td>363</td>
<td>W. 176</td>
<td>Sp. 180</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Leakage between shafts</td>
</tr>
<tr>
<td>Prestonlinks</td>
<td>240</td>
<td>W. 231</td>
<td>Sp. 325</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>S. 227</td>
</tr>
<tr>
<td>Dunnikier</td>
<td>128</td>
<td>W. 231</td>
<td>Sp. 250</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>S. 225</td>
</tr>
<tr>
<td>Kinglassie</td>
<td>161</td>
<td>W. {</td>
<td>Sp. }</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Indeterminate</td>
</tr>
<tr>
<td>Valleyfield</td>
<td>146</td>
<td>W. 460</td>
<td>Sp. {</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Indeterminate</td>
</tr>
<tr>
<td>Wellesley</td>
<td>Bottom Part 130 Top Part 150</td>
<td>W. 431 Top</td>
<td>200 Bottom</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Sp. 490 Top</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>227 Bottom</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>S. 395 Top</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>182 Bottom</td>
</tr>
<tr>
<td>Hylton</td>
<td>177</td>
<td>W. }</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Indeterminate</td>
</tr>
<tr>
<td>Silksworth</td>
<td>154</td>
<td>W. 638</td>
<td></td>
</tr>
<tr>
<td>Coventry</td>
<td>363</td>
<td>W. 114</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Narrow connection leading to bottom of ucast.</td>
</tr>
<tr>
<td>Craven</td>
<td>48</td>
<td>W. 552</td>
<td></td>
</tr>
</tbody>
</table>

W = Winter  
Sp = Spring  
S = Summer
more gradual, as in the case of Kinglassie (Figure 43) and Wellesley (Figure 45). At the commencement of these experiments we did not expect to find any sudden change in the relationship. While realising that a gradual reduction of fan-speed would alter the nature of flow through the various passages of the mine, we thought that, for the complex conditions existing in mines, the changes would be so widely distributed that their effect upon the mine characteristic would be gradual. This view, however, would seem to be erroneous, since, for each of the mines included in Table IV, there is a more-or-less definite critical fan-drift velocity, on the two sides of which the relationship between p and Q is entirely different.

We hesitate to suggest a reason for the "kink" which is a feature of most of the curves shown in Figures 38 to 49. To cause so sudden a change in relationship, would, it seems to us, indicate the presence of some single type of resistance which dominates the situation. It does not seem likely that the predominant factors will be the working faces and roads leading thereto, including inbye leakage, for, with such varying character, anything like a unanimous change of relation at a particular speed of the fan would appear improbable. Nor is it the shafts that dominate the situation. We have calculated the loss of pressure for a number of these and in general find them to be small in comparison with that of the whole mine. Again, had the resistance of the shafts been the principal factor, we might have expected that the "kink" would occur each time at a particular velocity of the air in the upcast shaft. The actual velocities in the upcast corresponding to the critical velocities in the fan-drift are set forth in Table VI. The variation is so great that obviously the shafts have very little to do with the change of law.
With regard to the main intake and main return airways; the former are generally fairly large and well maintained, while the latter are often very much neglected. Suspicion rests, therefore, on the return airways as being largely responsible for building up the total water-gauge, and for the "kink" in the mine characteristic. Much light will no doubt be thrown on this subject by the aneroid surveys of the airways in the mines belonging to the Powell-Duffryn Coal Company now being carried out under the supervision of Major E. Ivor David.

The effect of the duration of the period of unstable flow on the upper portion of the characteristic curve and therefore on its equation, has been referred to on page 83.
PART V (contd.).

SECTION (C).

NATURAL VENTILATION.

Natural Ventilation may be likened to a second fan operating in series with the surface ventilator. The effect is a varying one, chiefly depending upon the difference in temperature between the two shafts; if the average upcast temperature is the higher, natural agencies will assist the ventilation; if the average downcast temperature is the higher, then this natural ventilation will oppose the fan.

As the depth and temperature of our mines increase the problem of natural ventilation calls more urgently for solution. Natural causes ought never to be assumed negligible, even in shallow and cool mines.

The tests which first drew the attention of mining engineers to the magnitude of natural ventilation in a deep mine were those carried out by Mr. Robert Clive at Bentley Colliery, Yorkshire, to which reference has already been made (see Clive’s work, page 55). At the time of the tests the average downcast temperature was 20 Fahr. lower than the average upcast temperature. With the fan running at normal speed, 309,530 cubic feet of air per minute were circulating against a fan-drift pressure of 13 lbs. per square foot. When the fan was standing and the upcast doors wide open, 200,000 cubic feet per minute were yielded by natural agencies operating alone. This is to say, natural ventilation acting alone was able to produce about two-thirds of the air circulated by the fan and natural ventilation operating in series.

In Table VII we have collected the volumes passing when the fan is operating at normal speed together with those /
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Arniston</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>16.63</td>
<td>128,850</td>
<td>38,150</td>
<td>12.46</td>
<td>113,110</td>
<td>30,250</td>
</tr>
<tr>
<td>Easthouses</td>
<td>-</td>
<td>59,280</td>
<td>10,430</td>
<td>12.51</td>
<td>70,400</td>
<td>25,770</td>
<td>2.27</td>
<td>70,610</td>
<td>15,500</td>
</tr>
<tr>
<td>Flakemmet</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>19.3</td>
<td>105,570</td>
<td>49,290</td>
<td>10.52</td>
<td>103,740</td>
<td>47,960</td>
</tr>
<tr>
<td>Prestonlinks</td>
<td>-6.99</td>
<td>100,768</td>
<td>3,660</td>
<td>13.9</td>
<td>117,750</td>
<td>22,340</td>
<td>9.54</td>
<td>116,940</td>
<td>21,180</td>
</tr>
<tr>
<td>Dunnikier</td>
<td>0.9</td>
<td>78,390</td>
<td>6,600</td>
<td>12.49</td>
<td>77,220</td>
<td>12,325</td>
<td>6.97</td>
<td>73,450</td>
<td>11,430</td>
</tr>
<tr>
<td>Kinglassie</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>11.31</td>
<td>76,710</td>
<td>11,645</td>
<td>7.57</td>
<td>77,370</td>
<td>10,040</td>
</tr>
<tr>
<td>Valleyfield</td>
<td>-2.33</td>
<td>165,900</td>
<td>6,380</td>
<td>19.09</td>
<td>159,700</td>
<td>38,615</td>
<td>9.46</td>
<td>163,140</td>
<td>27,970</td>
</tr>
<tr>
<td>Wellesley</td>
<td>-2.78</td>
<td>121,100</td>
<td>-9,315</td>
<td>21.37</td>
<td>127,460</td>
<td>34,850</td>
<td>15.96</td>
<td>148,100</td>
<td>33,090</td>
</tr>
<tr>
<td>Hylton</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>22.43</td>
<td>173,590</td>
<td>70,500</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Silksworth</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>21.65</td>
<td>198,220</td>
<td>59,700</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Coventry</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>16.61</td>
<td>69,620</td>
<td>35,030</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Craven</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>13.93</td>
<td>31,000</td>
<td>12,050</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

where $T$ = Average Upcast Temperature  
$t$ = Average Downcast Temperature
those circulating when the fan is stationary and natural ventilation is acting alone, for different periods of the year at the collieries named in the Table. The difference between the average upcast temperature and the average downcast temperature is also given.

A brief inspection of the table shows that during the greater part of the year natural agencies play an important part in the ventilation of our mines. At Hylton Colliery, for instance, two-fifths of the total volume circulating during winter conditions was due to natural ventilation, while at Coventry and Polkemmet Collieries natural ventilation yielded nearly half of the total volume. Reference to the final column of the table shows that during Spring, and therefore Autumn, conditions, natural ventilation renders much assistance. At Arniston Colliery, for instance, 30,200 cubic feet of air (or over one-quarter of the total air-volume circulating at normal fan-speed) were yielded by natural causes when the average surface temperature was 48.6 °Fahr.

The figures given (Column 4) appear to indicate that during the summer months, however, no great assistance is to be expected from natural ventilation, since even at Valleyfield Colliery (shafts 1160 feet deep) the volume provided by natural causes amounted to just over 6000 cubic feet of air per minute, the average surface temperature being 63.25 °Fahr. At Easthouses Colliery natural ventilation yielded nearly 10,500 cubic feet of air per minute when the average surface temperature was 57.7 °Fahr. However, since this mine is connected underground to Newbattle Colliery, we cannot say with certainty that this volume was solely due to natural agencies. With regard to the reversed flow indicated for Wellesley Colliery, this was probably due to the heating of the air by the steam pipes in the downcast shaft.

From the above considerations it appears reasonable to assume that, during at least nine months of the year, the/
the fan is being assisted, more-or-less, by natural ventilation. The fan is required to run continuously, twenty-four hours per day, and, therefore, if the speed be kept constant the fan-motor or engine is required to develop more power when natural ventilation is assisting the fan in the greatest degree, i.e. when the conditions of the mine least require to make an increased demand for power. Fan installations should be so arranged, therefore, that full advantage can be taken of natural ventilation. If economy is to be effected in power consumption there must be a ready means of varying the speed of the fan-motor or engine. It is an easy matter to regulate the fan-speed where the driving machine is a steam engine or a variable speed motor. Where a constant speed motor is the prime mover, the problem is not so simple. For such a case, Dr. Parker has suggested a three-speed stepped-pulley arrangement fitted to the fan-motor as a means of utilising natural ventilation effects to secure the maximum economy in power consumption. With a large installation, however, the stepped-pulley arrangement is, we think, scarcely practicable and a variable speed motor appears to be a simpler solution to the problem.

It is interesting to note that at Nottalburgh (W. Va.) mine, belonging to the Fordson Coal Company, U.S.A., a motor of the variable speed type has been installed primarily to compensate for changes in the temperature of the outside air. The fan, in this instance, is situated about 2000 feet distant from the tipple and at an elevation of 350 feet, and provision is made for changing the fan-speed from inside the mine.

1. loc. cit.
2. Coal Age - 11th March, 1926, p. 368,
Factors Controlling Natural Ventilation.

Natural Ventilation is influenced by various factors, such as:-

(a) Changes in the temperature of the external atmosphere;
(b) The difference between the temperature of the air in the upcast and that in the downcast shaft;
(c) Workings to the rise and the dip. A ventilating system lying to the dip side of the shafts tends to increase natural ventilation, while such a system on the rise side tends to reduce it;
(d) The mine fan, when it is running.

The mine fan modifies natural ventilation in the following ways:-

(i) When the volume and velocity of the air is increased, due to increased fan-speed, there is a slight reduction in the temperature of the rock surfaces, which increases the difference between the temperature of the surfaces and that of the air itself. Where the workings are flat and extensive the airways will probably be long enough to ensure that the temperature of the air in the upcast shaft remains unchanged, and, hence, the effect on natural ventilation of an increased volume of air will probably be slight. If the workings are to the dip the tendency will be to increase the natural ventilation due to the cooling by the intakes, while with workings to the rise, or a mine in the development stage, the opposite effect is more likely.

(ii) When the fan is running it causes a reduction in pressure in the upcast shaft, which in turn rarefies.

2. The exhausting fan is being considered. The phraseology will require adjusting for a forcing fan, though the argument holds good.
rarefies the air in that shaft and therefore increases natural ventilation. This rarefaction of the air varies with the fan-speed and is independent of temperature changes. It will disappear when the fan is stationary. The reduction in pressure affects the whole mine, but as it dies out in the direction of the downcast it will be sufficient if we consider the shafts only. Since we found that, for the mines included in Table IV, the shaft resistances were small compared with those of the workings, these resistances may be neglected and the pressure-difference between the shafts taken as the observed fan-drift pressure, measured in \( p \) lbs. per square foot. The slight rarefaction of the air caused by the fan imposes, on the natural ventilating pressure, a positive correction, of an amount:

\[
a_1 = \frac{D}{A} \frac{P w_u D}{A} \text{ lbs. per square foot.}
\]

where \( w_u \) = Average weight of a cubic foot of air in the upcast shaft; 
\( D \) = Depth of shaft in feet; 
\( A \) = Normal atmospheric pressure = 2120 lbs. per square foot.

(iii) The volume of air passing up the upcast shaft is greater by perhaps 10 to 15 per cent. than that in the downcast shaft due to its increase in temperature and to the presence of mine gases. Since upcasts are usually smaller in sectional area than downcasts (see Table IV); the velocity of the air in the former is greater than that in the latter. The upcast air will, therefore, have a lower static pressure. Since static pressure is one of the factors affecting the density of the air and another correction is necessary. The difference of the static pressure in the two shafts due to the difference in velocity head is, very nearly:

\[
\frac{w_u (v_u^2 - v_d^2)}{2g}
\]

where \( v_u \) = velocity of air in the upcast in feet per second; 
\( v_d \) = velocity of air in the downcast in feet per second.
Therefore the correction is:

\[ a_2 = \frac{w_u^2 (v_u^2 - v_d^2) D}{4.240 g} \]

In most cases this correction will be so small that it may be considered negligible.
DETERMINATION of the NATURAL VENTILATING PRESSURE.

Only in the case of absolutely level seams can the pressure producing natural ventilation be determined from the densities of the air in the shafts. Where the workings are inclined, as is usually the case, it becomes impossible to determine that pressure from an air-density survey of the mine. The problem becomes more complex where the workings extend to the rise and the dip and where more than one mine is involved. Take, for instance, the case of Wellesley Colliery which has extensive workings to the rise and the dip and in addition is interconnected with at least five neighbouring mines. In such an instance it would be hopeless to attempt to evaluate the natural ventilating pressure by means of an air-density survey.

In the appendix we give the atmospheric observations taken at the surface, underground, and in the fan-drift, for the various mines. The method of reduction of the observations is set forth on page 106.

According to Murgue the most simple proceeding to render visible the motive column due to natural ventilation is to close the fan-drift by a stopping and allow the natural action to press with all its force against the obstruction. On conducting several tests he found that the motive column of a natural current was stronger when that current was stopped than when it was circulating and concluded that the method was wrong. We think, however, that this procedure would, if properly carried /

carried out, furnish a fairly accurate result provided the corrections contained on page 95 were made.

In a recent article, Professor Briggs suggested a direct method which would probably provide a reliable result no matter how complex the ventilating system might be. It consists of stopping the surface fan (and underground fans, if any) and keeping the upcast doors closed, allowing natural agencies to drive air through the mine and fan for, say, half an hour. Then setting the doors, shutters, etc. for reversal, start the fan very slowly, gradually increasing its speed until the air in the upcast shaft becomes stationary and read the reversed gauge in the fan drift. By producing a certain pressure the fan has been able to balance the effect due to natural ventilation; in other words, the fan drift pressure is equal and opposite to the natural ventilation pressure and so measures the latter. This method would, we think, involve a finer speed adjustment than most mine fans possess.

An indirect method which is the outcome of a recent paper by Dr. Parker consists of extrapolating the mine characteristic to the vertical axis after the manner shown in Figure 37. Does this process rest on a secure foundation?

It has been clearly demonstrated in a previous section of this work that the relation between $p$ and $Q$ for the lower portion of the mine curve (AB, Figure 37) is entirely different to that pertaining to the upper part (BD). The top and bottom parts of the curve can both be represented by an equation of the form

$$ p = A_Q^2 + B_Q - C $$

From the considerations in the section on the Zone/
Figure 62.
Volume of Air.

Figure 62.- Graphical Determination of the Natural Ventilating Pressure.
Zone of Unstable Flow (page 88) it is obvious that the position of the major portion of the mine curve is not solely dependent on natural ventilation and hence it would be illogical to extrapolate that part in order to evaluate the natural ventilating pressure.

With regard to the lower portion of the curve, it will be readily realized that, as this lies in a region of unstability, we cannot use the equation of this part to determine the desired result. Moreover, it is seldom possible to obtain a sufficient number of readings below the "kink" to construct, with any degree of accuracy, this part of the mine characteristic.

From these considerations and the fact that this method assumes there is no change in the relation between \( p \) and \( V \) throughout the length of the mine curve, we think it reasonable to conclude that the method of extrapolation is useless for determining the pressure due to natural ventilation. However, equations of the form

\[
p^2 = RQ - K
\]

may, for routine work, give a result sufficiently close to the correct value.

Since we are chiefly concerned with the value of the natural ventilating pressure over the working range of the fan, there is yet another method which offers a solution to this involved problem. Suppose the curves ABC and DEF (Figure 62) represent the major parts of the mine characteristics for Summer and Winter conditions respective. Assuming that the mine resistance has remained practically constant during the interval, the pressure due to natural ventilation can be determined by the vertical distance between the two curves. Suppose it is required to determine this pressure when a certain volume, say \( Q_1 \) is passing during winter conditions. This is evaluated by the vertical intercept BE. Although this method involves the use of two mine curves obtained at different periods of the year, we think, nevertheless, that it provides a
very probable solution of the problem of determining the natural ventilating pressure.

The Full Effective Ventilating Pressure.

When natural ventilation is assisting the fan, this pressure has been considered to be the sum of the observed fan-drift pressure \((p)\) and the natural ventilation pressure \((p_0)\), i.e.

\[
P = p + p_0
\]

This does not, however, give the desired result.

We have already seen that, when the fan is stationary, there is a reversed pressure in the fan drift, i.e., the fan drift pressure is greater than atmospheric. With a high resistance fan this pressure difference is not negligible. At Hylton Colliery (see pages of Appendix), for instance, this reversed pressure was 1.35 pounds per square foot. Now when the fan begins to rotate, it must neutralize this reversed pressure before it begins to register the so-called observed pressure and, therefore, the fan must be credited with this pressure. Hence, the pressure created by the fan is:

\[
\text{Pressure produced by fan} = p + p_r
\]

From this it follows that the full effective pressure producing ventilation throughout the mine is:

\[
P = p + p_r + p_0
\]

where \(P\) = full effective ventilating pressure and \(p_0\) = the natural ventilation pressure.

The sign of the latter is positive when natural agencies are assisting the fan and negative when they are acting adversely.
| No. | Colliery     | Type of Fan                      | Diam. | Width | Press. (p) lbs. per ft. | Volume (Q) Kilolitres | Resistance of Fan Atkins
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Arniston</td>
<td>Sirocco (Double inlet)</td>
<td>77</td>
<td>63</td>
<td>0.82</td>
<td>0.64</td>
<td>2.0</td>
</tr>
<tr>
<td>2.</td>
<td>Easthouses</td>
<td>Propeller (Sirocco)</td>
<td>70</td>
<td>-</td>
<td>1.10</td>
<td>0.44</td>
<td>5.8</td>
</tr>
<tr>
<td>3.</td>
<td>Polkemmet</td>
<td>Sirocco (Double inlet)</td>
<td>105</td>
<td>84</td>
<td>0.30</td>
<td>0.84</td>
<td>0.33</td>
</tr>
<tr>
<td>4.</td>
<td>Prestonlinks</td>
<td>Howden (Double inlet)</td>
<td>156</td>
<td>81</td>
<td>0.22</td>
<td>0.37</td>
<td>1.6</td>
</tr>
<tr>
<td>5.</td>
<td>Dunnikier</td>
<td>Waddle (Single inlet)</td>
<td>318</td>
<td>12</td>
<td>0.15</td>
<td>0.21</td>
<td>3.4</td>
</tr>
<tr>
<td>6.</td>
<td>Valleyfield</td>
<td>Walker (Double inlet)</td>
<td>216</td>
<td>84</td>
<td>0.13</td>
<td>0.64</td>
<td>0.32</td>
</tr>
<tr>
<td>7.</td>
<td>Wellesley</td>
<td>Waddle (Single inlet)</td>
<td>252</td>
<td>15</td>
<td>0.11</td>
<td>0.59</td>
<td>0.30</td>
</tr>
<tr>
<td>8.</td>
<td>Hylton</td>
<td>Sirocco (Double inlet)</td>
<td>91</td>
<td>63</td>
<td>1.35</td>
<td>1.15</td>
<td>0.93</td>
</tr>
<tr>
<td>9.</td>
<td>Silksworth</td>
<td>Capell (Double inlet)</td>
<td>144</td>
<td>72</td>
<td>0.33</td>
<td>0.97</td>
<td>0.35</td>
</tr>
<tr>
<td>10.</td>
<td>Craven</td>
<td>Sirocco (Double inlet)</td>
<td>49</td>
<td>45</td>
<td>0.24</td>
<td>0.21</td>
<td>5.53</td>
</tr>
</tbody>
</table>
101.

SECTION (D).

RESISTANCE OF STATIONARY EXHAUSTING FANS.

The resistance of the mine ventilator is a very involved problem. The relationship between the resistance of a fan when standing and when running is not constant for any given type of ventilator, but depends upon the circumstances of application. Our object, in this instance, is to deal with the resistance of stationary fans.

The figures in the final column of Table VIII were obtained by dividing the total pressure in the fan drift when the fan was stationary (column 6) by the square of the volume of air (column 7) passing through the fan due to the effect of natural agencies. We believe that, in this instance, $P_f$ is proportional to $Q_f^2$. Therefore the fan's resistance is:

$$ R_f = \frac{P_f}{Q_f^2} $$

where $R_f$ = resistance of the stationary fan measured in Atkinsons;
$P_f$ = pressure in the fan drift when the fan is standing;
$Q_f$ = volume of air passing through the fan due to natural causes.

In all the selected cases natural ventilation was assisting the fan when the latter was operating. Owing to the fact that natural agencies had to force air through the stationary fan, the observed fan drift pressures are above atmospheric; in other words, the pressures were negative. The corresponding points are, therefore, to be found below the horizontal axis of the graphs shown in Figures 38 to 61 inclusive. We have omitted a number of results involving pressures below 0.1.
0.1 lbs. per square foot as some were found to give widely varying values. Most of the data contained in the table were obtained during cold weather when natural ventilation produced air-volumes and pressures which were higher and more easily measured than those yielded during the summer months. This method of evaluating a fan's resistance suffers from the fact that it involves the use of observations lying in the zone of unstable flow, i.e. in the region where the relationship between \( p \) and \( Q \) is variable (see page 69). Observations taken after stopping the fan were sometimes different from those obtained under similar circumstances a short time afterwards. These changes probably account for the variation in the results obtained, as, for instance, at Prestonlinks and Valleyfield Collieries.

It is to be observed that the fan-resistance, as given in the Table, includes not only the frictional resistance of the fan and its adjutages but also all other losses of pressure (or energy) between the pressure-tube in the fan drift and the mouth of the evasee. The effect of friction is generally small compared with the losses due to eddying and turbulence in the approaches and inlets, in the runner and casing and in the chimney, and with those due to inefficient design of the casing and evasee.

As might be expected, the figures given in the Table show that, considering fans of the same type, the smaller the diameter of the fan the higher is its resistance. For instance, the 49 inch diameter Sirocco fan at Craven Colliery has a resistance of about 5.9 Atkinsons; that at Arniston (77 inch diameter) a resistance of about 2.2 Atkinsons; that at Hylton (91 inch diameter) a resistance of 0.93 Atkinsons, while the resistance of the 105 inch Sirocco at Folkemmet is about 0.81 Atkinsons. On comparing the Waddle fans at Dunnikier and Wellesley Collieries it would seem that the smaller the diameter the lower /
lower the resistance. It must be observed, however, that the fan at the former mine was in a very bad condition, being coated with at least three inches of mud, while that at the latter colliery was comparatively clean.

In the case of the Sirocco propeller fan at Easthouses Mine, the high resistance is, no doubt, due to two right-angle bends existing on the inlet side and to the absence of an evacuee. The resistance of this fan is considerably higher than that of the whole mine. The power consumed at Easthouses is comparatively small (see Table III page 40) and the present fan is only temporary. Speaking generally, however, we doubt whether the expense of running a high resistance fan is sufficiently realised. The approaches to the fan should be large and free from right-angle bends. The blades should be kept clean and no water or mud allowed to collect in the casing or pit below the wheel. Once or twice we have come across fans whose runners were dragging through water or mud, with an effect on their capacity and efficiency which can well be imagined. In one instance, cleaning the pit under the runner reduced the resistance of the fan to less than one-tenth of its previous value.
CONCLUSIONS.

Section (A).

1. The relation between the ventilating pressure \( p \) and the air-volume \( Q \) cannot be represented by equations of the form:
\[
2 \quad P = RQ_n
\]
or
\[
2 \quad P = RQ
\]
Hence, these forms should be abandoned forthwith.

2. The relation between \( p \) and \( Q \) can be very accurately represented by equations of the form:
\[
2 \quad p = A_Q + B_Q - C
\]
and
\[
2 \quad p = RQ - C.
\]
The former is the more easily obtained from a set of readings but the latter furnishes a single factor \( R \). The value of \( n \), however, must always be stated when using the latter form otherwise no useful meaning can be attached to the coefficient \( R \).

3. At all seasons of the year the mine characteristic curve can be very closely represented by an equation of the form:
\[
2 \quad p = RQ - K.
\]

This is simpler than the forms given in conclusion (2) and gives a factor \( R \) which is unaffected by a variable index \( n \). This form appears acceptable for routine work.

Section (B).

1. There is a point in the mine curve on the two sides of which the relation between \( p \) and \( Q \) is entirely different. This change is produced by the existence of some particular part of the ventilating system which dominates the situation.
1. Natural ventilation should never be considered negligible.

2. Much assistance is derived from natural ventilation during at least nine months of the year.

3. Fan installations should be so arranged that the fan-speed can be varied to utilize to the full the effect of natural ventilation.

4. The natural ventilating pressure cannot be determined, by extrapolation of the mine curve.

5. The pressure producing natural ventilation can be determined, with fair accuracy, by the vertical intercept between two mine curves obtained at different periods of the year.

6. The total pressure producing ventilation throughout the mine is the algebraical sum of:
   (a) The observed fan-drift pressure;
   (b) The reversed fan-drift pressure when the fan is stationary;
   (c) The pressure producing natural ventilation.

7. The pressure actually produced by the fan is the sum of (a) and (b) in Conclusion 6.

8. On stopping the fan, the air rapidly adjusts itself to the altered conditions.

Section (D).

1. Fan Drifts should be large and free of elbow bends between the fan-drift inset and the inlet to the fan.

2. The air should be guided gradually to the inlet of the fan instead of having to manoeuvre an angle of 90 degrees.

3. The fan blades and wheel should be periodically cleaned.

4. The fan casing should be so constructed that the pit below the wheel can be kept free from mud or water without stopping the fan.
PART VI.

APPENDIX.
In this part of our work we tabulate the observed data from which the curves shown in Figures 38 to 61 have been constructed. We also give in table form the barometric and hygrometric observations obtained at the surface, underground and in the fan-drift for the various mines in question. We have included the Dew Point, Relative Humidity and the density of the air in lbs. per cubic foot. These, except the last named, have been computed from Marvin's Tables. In the absence of suitable Tables the observed data may be reduced as follows:

**Vapour Pressure.**

From the formula

$$f = f_1 - 0.000367B (t-t_1) \left(1 + \frac{t-t_1}{1571}\right)$$

in which $f =$ vapour pressure in inches,

$f_1 =$ vapour pressure in saturated air at temperature $t_1$,

$t =$ temperature of the air in Fahr. degrees.

$t_1 =$ temperature of the wet bulb thermometer in Fahr. degrees,

$B =$ Barometric pressure in inches.

**Relative Humidity.**

This may be determined from the formula

$$\text{Relative Humidity} = \frac{f}{F}$$

where $f$ and $F$ are the maximum pressures of vapour corresponding respectively to the temperature of the dew point and $t$.

**Density of the Air in lbs. per cubic foot.**

In every case this has been computed from the formula

$$w = \frac{1.3255 (B - f)}{459 + t}$$

where /
where $w =$ weight of 1 cubic foot of air in lbs.
$f =$ vapour pressure in inches
$t =$ temperature of the air in Fahr. degrees.
$B =$ Barometric pressure in inches.
108.

Arniston Colliery.
24th January 1926.

Fan:- 77 inch Diameter Sirocco - double inlet - rope driven by D.C. Motor.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.F.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements Velocity ft/min.</th>
<th>Volume in Kilocusecs</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>-0.83</td>
<td>484</td>
<td>0.635</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>63 1/4</td>
<td>0.82</td>
<td>635</td>
<td>0.832</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>100 1/2</td>
<td>1.27</td>
<td>830</td>
<td>1.088</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>122.8</td>
<td>2.20</td>
<td>956</td>
<td>1.253</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>134</td>
<td>2.69</td>
<td>980</td>
<td>1.285</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>160.6</td>
<td>4.20</td>
<td>1132</td>
<td>1.434</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>186.6</td>
<td>5.81</td>
<td>1239</td>
<td>1.624</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>214</td>
<td>7.76</td>
<td>1371</td>
<td>1.798</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>-0.82</td>
<td>486</td>
<td>0.637</td>
<td>Normal Speed.</td>
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<td>10.07</td>
<td>1570</td>
<td>2.058</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>255.2</td>
<td>11.31</td>
<td>1638</td>
<td>2.148</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>279.8</td>
<td>13.66</td>
<td>1730</td>
<td>2.268</td>
<td></td>
</tr>
</tbody>
</table>

Area of Fan Drift = 78.66 square feet.

Summary of Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb 0°F</th>
<th>Wet Bulb 0°F</th>
<th>Barometer inches of Mercury</th>
<th>Dew Point 0°F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu.ft. of air (lbs.)</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>42.03</td>
<td>39.6</td>
<td>29.105</td>
<td>36.72</td>
<td>81.4</td>
<td>0.076783</td>
<td>Surface at No. 1 D.C.</td>
</tr>
<tr>
<td>40.50</td>
<td>38.1</td>
<td>29.08</td>
<td>35.05</td>
<td>81.1</td>
<td>0.076965</td>
<td>Surface at No. 2 D.C.</td>
</tr>
<tr>
<td>47.10</td>
<td>45.03</td>
<td>30.233</td>
<td>42.87</td>
<td>85.1</td>
<td>0.078911</td>
<td>Bottom of No. 1 D.C.</td>
</tr>
<tr>
<td>45.25</td>
<td>43.65</td>
<td>29.49</td>
<td>42.00</td>
<td>88.0</td>
<td>0.078361</td>
<td>Bottom of No. 2 D.C.</td>
</tr>
<tr>
<td>61.15</td>
<td>60.5</td>
<td>28.91</td>
<td>60.10</td>
<td>96.8</td>
<td>0.075673</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>63.55</td>
<td>62.0</td>
<td>29.645</td>
<td>62.70</td>
<td>97.3</td>
<td>0.074631</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 2.78 lbs. per sq.ft
Arniston Colliery.
14th March 1926.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>11.49</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>253</td>
<td>11.49</td>
<td>1438</td>
<td>1.885</td>
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<tr>
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<td>0.531</td>
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<td>1.09</td>
<td>655</td>
<td>0.859</td>
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<tr>
<td>4</td>
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<td>-0.37</td>
<td>452</td>
<td>0.593</td>
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<tr>
<td>5</td>
<td>53</td>
<td>0.16</td>
<td>527</td>
<td>0.691</td>
</tr>
<tr>
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<td>2.03</td>
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<td>1.028</td>
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<td>7</td>
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<td>3.36</td>
<td>876</td>
<td>1.149</td>
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<tr>
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<td>16</td>
<td>4.96</td>
<td>973</td>
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<td>1075</td>
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<tr>
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<td>214.2</td>
<td>8.37</td>
<td>1139</td>
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</tr>
<tr>
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<td>0.477</td>
</tr>
<tr>
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<td>281</td>
<td>14.18</td>
<td>1521</td>
<td>1.994</td>
</tr>
</tbody>
</table>

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer inches Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Mt. of 1 cu. ft. of air (lbs.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>48.50</td>
<td>46.50</td>
<td>29.77</td>
<td>44.60</td>
<td>88.20</td>
<td>0.077466</td>
</tr>
<tr>
<td>48.75</td>
<td>45.95</td>
<td>29.78</td>
<td>43.17</td>
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</tr>
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<td>60.85</td>
<td>100.00</td>
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<tr>
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<td>64.25</td>
<td>30.54</td>
<td>64.25</td>
<td>100.00</td>
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</tr>
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</table>

Calculated Natural Ventilating Pressure = 2.15 lbs. per sq. ft.
Easthouses Colliery
5th December 1925

Fan:- 70 inch Diameter Sirocco Propeller - belt driven by A.C. Motor.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Velocity ft/min.</td>
</tr>
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<td>-0.45</td>
<td>480</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-1.10</td>
<td>325</td>
</tr>
<tr>
<td>3</td>
<td>493</td>
<td>3.06</td>
<td>874</td>
</tr>
<tr>
<td>4</td>
<td>80*</td>
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<td>381</td>
</tr>
<tr>
<td>5</td>
<td>526</td>
<td>3.71</td>
<td>919</td>
</tr>
<tr>
<td>6</td>
<td>441</td>
<td>2.29</td>
<td>789</td>
</tr>
<tr>
<td>7</td>
<td>405</td>
<td>1.76</td>
<td>742</td>
</tr>
<tr>
<td>8</td>
<td>350</td>
<td>1.09</td>
<td>684</td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>-1.05</td>
<td>315</td>
</tr>
<tr>
<td>10</td>
<td>303</td>
<td>0.47</td>
<td>623</td>
</tr>
</tbody>
</table>

Area of Drift = 80.55 square feet

* Fan being driven by Natural Agencies.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer in Inches of Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity</th>
<th>Wt. of 1 cu. ft. of air (lbs.)</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>35.03</td>
<td>32.83</td>
<td>29.92</td>
<td>30.00</td>
<td>79.75</td>
<td>0.08010</td>
<td>Surface</td>
</tr>
<tr>
<td>52.5</td>
<td>50.00</td>
<td>31.40</td>
<td>47.85</td>
<td>84.4</td>
<td>0.081383</td>
<td>Bottom of DC</td>
</tr>
<tr>
<td>53.35</td>
<td>52.5</td>
<td>29.91</td>
<td>52.00</td>
<td>98.00</td>
<td>0.076991</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>59.20</td>
<td>58.60</td>
<td>31.40</td>
<td>58.24</td>
<td>97.08</td>
<td>0.080185</td>
<td>Bottom of UC</td>
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</table>

Calculated Natural Ventilating Pressure = 4.41 lbs. per sq. ft.
Easthouses Colliery
13th March, 1926.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements Velocity ft./min</th>
<th>Volume in Kilocusecs</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>484</td>
<td>3.82</td>
<td>889</td>
<td>1.193</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-0.39</td>
<td>198</td>
<td>0.266</td>
<td>U.C. doors open</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>-0.14</td>
<td>255</td>
<td>0.342</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements Velocity ft./min</th>
<th>Volume in Kilocusecs</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>524</td>
<td>4.51</td>
<td>936</td>
<td>1.296</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>470</td>
<td>3.51</td>
<td>846</td>
<td>1.136</td>
<td></td>
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<tr>
<td>6</td>
<td>365</td>
<td>2.23</td>
<td>696</td>
<td>0.934</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>390</td>
<td>2.38</td>
<td>708</td>
<td>0.950</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>356</td>
<td>2.04</td>
<td>660</td>
<td>0.886</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>-0.34</td>
<td>187</td>
<td>0.251</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>233</td>
<td>0.66</td>
<td>460</td>
<td>0.618</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>262</td>
<td>0.195</td>
<td>500</td>
<td>0.671</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>342</td>
<td>1.68</td>
<td>626</td>
<td>0.840</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>412</td>
<td>2.64</td>
<td>733</td>
<td>0.984</td>
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</tr>
</tbody>
</table>

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb Temp. °F</th>
<th>Wet Bulb Temp. °F</th>
<th>Barometer Point Inches</th>
<th>Dew Point Temp. °F</th>
<th>Relative Humidity per cent</th>
<th>Wet. of 1 cu. ft. of air (lbs.)</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>51.16</td>
<td>48.50</td>
<td>29.92</td>
<td>46.07</td>
<td>82.82</td>
<td>0.077435</td>
<td>Surface</td>
</tr>
<tr>
<td>55.81</td>
<td>53.96</td>
<td>30.81</td>
<td>52.55</td>
<td>88.97</td>
<td>0.079196</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>52.69</td>
<td>51.49</td>
<td>29.88</td>
<td>52.25</td>
<td>98.40</td>
<td>0.077041</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>58.88</td>
<td>58.31</td>
<td>30.91</td>
<td>57.95</td>
<td>96.37</td>
<td>0.076663</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.911 lbs. per sq. ft.
Fan:- 105 inch Diameter Sirocco-double inlet - steam driven.

### Table

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure (lbs. per sq. ft.)</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Velocity (ft/min)</td>
</tr>
<tr>
<td>1</td>
<td>123.6</td>
<td>8.58</td>
<td>635</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-0.21</td>
<td>291</td>
</tr>
<tr>
<td>3</td>
<td>139.4</td>
<td>10.03</td>
<td>676</td>
</tr>
<tr>
<td>4</td>
<td>116.4</td>
<td>7.19</td>
<td>391</td>
</tr>
<tr>
<td>5</td>
<td>110</td>
<td>6.07</td>
<td>559</td>
</tr>
<tr>
<td>6</td>
<td>105.3</td>
<td>5.46</td>
<td>542</td>
</tr>
<tr>
<td>7</td>
<td>93</td>
<td>4.28</td>
<td>506</td>
</tr>
<tr>
<td>8</td>
<td>81.6</td>
<td>3.05</td>
<td>476</td>
</tr>
<tr>
<td>9</td>
<td>72.3</td>
<td>2.33</td>
<td>451</td>
</tr>
<tr>
<td>10</td>
<td>57.6</td>
<td>1.52</td>
<td>444</td>
</tr>
<tr>
<td>11</td>
<td>43.6</td>
<td>0.77</td>
<td>402</td>
</tr>
<tr>
<td>12</td>
<td>0</td>
<td>-0.23</td>
<td>302</td>
</tr>
</tbody>
</table>

Area of Fan Drift = 1664 sq. ft.

### Atmospheric Observations

<table>
<thead>
<tr>
<th>Dry Bulb °F.</th>
<th>Wet Bulb °F.</th>
<th>Barometer inches Hg</th>
<th>Dew Point °F.</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu. ft. of air (lbs)</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>46.64</td>
<td>44.30</td>
<td>29.69</td>
<td>41.87</td>
<td>83.45</td>
<td>0.077574</td>
<td>Surface</td>
</tr>
<tr>
<td>49.20</td>
<td>48.10</td>
<td>30.87</td>
<td>47.10</td>
<td>92.60</td>
<td>0.080211</td>
<td>Bottom of D. Fan Drift</td>
</tr>
<tr>
<td>68.58</td>
<td>68.56</td>
<td>29.61</td>
<td>68.54</td>
<td>98.84</td>
<td>0.083534</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>65.87</td>
<td>65.22</td>
<td>30.88</td>
<td>65.02</td>
<td>97.3</td>
<td>0.077353</td>
<td>Bottom of U.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 6.63 lbs. / sq. ft.
**Polkemmet Colliery**

**4th April 1926.**

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Velocity ft/min</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Volume in Killocusecs</td>
</tr>
<tr>
<td>Remarks</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| 1 | 126.2 | 7.85 | 624 | 1.729 | Normal Speed |
| 2 | 138.4 | 9.74 | 681 | 1.887 |
| 3 | 0     | -0.26| 287 | 0.795 |
| 4 | 0.33 | 0.72 | 383 | 0.953 |
| 5 | 4.70 | 3.35 | 531 | 1.061 |
| 6 | 2.48 | 3.15 | 452 | 1.471 |
| 7 | 3.42 | 6.14 | 577 | 1.336 |
| 8 | 5.92 | 6.76 | 591 | 1.252 |
| 9 | 6.62 | 1.38 | 415 | 1.500 |
| 10| 0    | -0.26| 290 | 0.804 |
| 11| 0    | -0.26| 290 | 0.804 |
| 12| 0    | -0.26| 290 | 0.804 |

**Atmospheric Observations.**

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer inches Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity %</th>
<th>Wt. of air per cu. ft (lbs.)</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>57.79</td>
<td>54.71</td>
<td>29.42</td>
<td>52.45</td>
<td>88.33</td>
<td>.070593</td>
<td>Surface</td>
</tr>
<tr>
<td>54.71</td>
<td>53.79</td>
<td>30.87</td>
<td>53.12</td>
<td>94.62</td>
<td>.079254</td>
<td>Bottom of D</td>
</tr>
<tr>
<td>73.80</td>
<td>73.07</td>
<td>29.37</td>
<td>72.75</td>
<td>96.80</td>
<td>.072303</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>59.75</td>
<td>58.60</td>
<td>30.86</td>
<td>57.88</td>
<td>93.52</td>
<td>.078406</td>
<td>Bottom of U</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = \( 1.37 \) lbs./sq.ft
Prestonlinks Colliery
9th August 1921

Fan: 13 Ft. Diameter Howden - double inlet + steam driven.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0</td>
<td>-0.01</td>
<td>31*</td>
<td>*by smoke</td>
</tr>
<tr>
<td>2</td>
<td>19.5</td>
<td>+0.10</td>
<td>86</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>68</td>
<td>+2.60</td>
<td>649</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>47</td>
<td>+1.58</td>
<td>224</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>88</td>
<td>+2.60</td>
<td>604</td>
<td>1.171</td>
</tr>
<tr>
<td>6</td>
<td>98</td>
<td>+5.86</td>
<td>679</td>
<td>1.31</td>
</tr>
<tr>
<td>7</td>
<td>109</td>
<td>+7.59</td>
<td>778</td>
<td>1.509</td>
</tr>
<tr>
<td>8</td>
<td>120</td>
<td>+9.20</td>
<td>866</td>
<td>1.679</td>
</tr>
<tr>
<td>9</td>
<td>132</td>
<td>+11.20</td>
<td>980</td>
<td>1.901</td>
</tr>
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</table>

Area of Fan Drift = 116.36 sq. ft.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer inches</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu. ft. air (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>65.50</td>
<td>58.45</td>
<td>29.64</td>
<td>53.55</td>
<td>65.60</td>
<td>.074720</td>
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<tr>
<td>63.10</td>
<td>58.85</td>
<td>30.01</td>
<td>56.25</td>
<td>84.15</td>
<td>.075770</td>
</tr>
<tr>
<td>56.85</td>
<td>56.65</td>
<td>29.52</td>
<td>56.52</td>
<td>98.7</td>
<td>.075413</td>
</tr>
<tr>
<td>27.77</td>
<td>27.45</td>
<td>29.90</td>
<td>57.29</td>
<td>98.06</td>
<td>.076247</td>
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</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.23 lbs. per sq.ft
### Prestonlinks Colliery

6th December 1925

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Velocity ft/min.</td>
</tr>
<tr>
<td>1</td>
<td>12</td>
<td>-0.10</td>
<td>240</td>
</tr>
<tr>
<td>2</td>
<td>54½</td>
<td>1.49</td>
<td>478</td>
</tr>
<tr>
<td>3</td>
<td>129</td>
<td>9.193</td>
<td>1073</td>
</tr>
<tr>
<td>4</td>
<td>146</td>
<td>12.9</td>
<td>914</td>
</tr>
<tr>
<td>5</td>
<td>123</td>
<td>7.42</td>
<td>844</td>
</tr>
<tr>
<td>6</td>
<td>116</td>
<td>7.38</td>
<td>582</td>
</tr>
<tr>
<td>7</td>
<td>105½</td>
<td>5.75</td>
<td>757</td>
</tr>
<tr>
<td>8</td>
<td>81½</td>
<td>3.71</td>
<td>640</td>
</tr>
<tr>
<td>9</td>
<td>70</td>
<td>2.61</td>
<td>572</td>
</tr>
<tr>
<td>10</td>
<td>17</td>
<td>0.02</td>
<td>45</td>
</tr>
<tr>
<td>11</td>
<td>0</td>
<td>-0.28</td>
<td>185</td>
</tr>
<tr>
<td>12</td>
<td>0</td>
<td>-0.22</td>
<td>192</td>
</tr>
</tbody>
</table>

Remarks:
- Underground Fans stopped
- Underground Fans running

Area of Fan Drift = 116.36 sq. ft.

### Atmospheric Observations

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer in Inches of Mercury</th>
<th>Dew Point in °F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of Air in 1 cu. ft.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>33.95</td>
<td>37</td>
<td>30.06</td>
<td>32.95</td>
<td>75.69</td>
<td>0.079705</td>
<td>Surface</td>
</tr>
<tr>
<td>39.00</td>
<td>37.8</td>
<td>30.46</td>
<td>36.22</td>
<td>89.4</td>
<td>0.080063</td>
<td>Bottom of D. Fan Drift</td>
</tr>
<tr>
<td>51.20</td>
<td>50.80</td>
<td>29.98</td>
<td>50.86</td>
<td>97.20</td>
<td>0.077531</td>
<td>Bottom of U. C.</td>
</tr>
<tr>
<td>55.55</td>
<td>55.00</td>
<td>30.30</td>
<td>54.60</td>
<td>96.40</td>
<td>0.077695</td>
<td></td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 1.07 lbs./sq. ft.
Prestonlinks Colliery.
21st March 1926.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Velocity fly/min.</td>
<td>Volume in Kilocusecs</td>
</tr>
<tr>
<td>1</td>
<td>122(\frac{3}{4})</td>
<td>8.50</td>
<td>1005</td>
<td>1.949</td>
</tr>
<tr>
<td>2</td>
<td>143(\frac{3}{4})</td>
<td>11.53</td>
<td>1166</td>
<td>2.261</td>
</tr>
<tr>
<td>3</td>
<td>69</td>
<td>-0.19</td>
<td>179</td>
<td>0.347</td>
</tr>
<tr>
<td>4</td>
<td>31(\frac{3}{4})</td>
<td>2.62</td>
<td>595</td>
<td>1.154</td>
</tr>
<tr>
<td>5</td>
<td>62</td>
<td>0.41</td>
<td>357</td>
<td>0.688</td>
</tr>
<tr>
<td>6</td>
<td>40(\frac{3}{4})</td>
<td>0.74</td>
<td>412</td>
<td>0.749</td>
</tr>
<tr>
<td>7</td>
<td>62</td>
<td>2.01</td>
<td>561</td>
<td>1.088</td>
</tr>
<tr>
<td>8</td>
<td>73(\frac{3}{4})</td>
<td>1.42</td>
<td>504</td>
<td>0.977</td>
</tr>
<tr>
<td>9</td>
<td>10(\frac{1}{4})</td>
<td>5.41</td>
<td>848</td>
<td>1.647</td>
</tr>
<tr>
<td>10</td>
<td>10(\frac{3}{4})</td>
<td>3.41</td>
<td>695</td>
<td>1.148</td>
</tr>
<tr>
<td>11</td>
<td>112(\frac{3}{4})</td>
<td>6.79</td>
<td>960</td>
<td>1.862</td>
</tr>
<tr>
<td>12</td>
<td>12</td>
<td>-0.22</td>
<td>182</td>
<td>0.168</td>
</tr>
<tr>
<td>13</td>
<td>0</td>
<td>-0.24</td>
<td></td>
<td>0.153</td>
</tr>
</tbody>
</table>

Note: Two underground fans running during tests Nos. 1 to 12 but stopped during test No. 13.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb (^{\circ})F</th>
<th>Wet Bulb (^{\circ})F</th>
<th>Barometer Inches of Mercury</th>
<th>Dew Point (^{\circ})F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu. ft. air (lbs.)</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>44.92</td>
<td>41.83</td>
<td>30.42</td>
<td>40.63</td>
<td>84.83</td>
<td>0.079770</td>
<td>Surface</td>
</tr>
<tr>
<td>43.20</td>
<td>41.10</td>
<td>30.61</td>
<td>38.58</td>
<td>83.76</td>
<td>0.081083</td>
<td>Bottom of Fan Drift</td>
</tr>
<tr>
<td>51.86</td>
<td>51.76</td>
<td>30.31</td>
<td>51.65</td>
<td>99.16</td>
<td>0.078277</td>
<td>Bottom of Fan Drift</td>
</tr>
<tr>
<td>55.35</td>
<td>54.87</td>
<td>30.75</td>
<td>54.55</td>
<td>96.80</td>
<td>0.078833</td>
<td>Bottom of Fan Drift</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.75 lbs. per sq. ft.
### Dunnikier Colliery, Fifeshire.

**2nd August, 1925.**

Fan: - 26½ feet diameter Waddle - steam driven.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Remarks</td>
</tr>
<tr>
<td>1</td>
<td>59</td>
<td>6.58</td>
<td>603</td>
</tr>
<tr>
<td>2</td>
<td>66</td>
<td>7.99</td>
<td>667</td>
</tr>
<tr>
<td>3</td>
<td>52</td>
<td>5.96</td>
<td>521</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
<td>-0.05</td>
<td>53*</td>
</tr>
<tr>
<td>5</td>
<td>46</td>
<td>3.74</td>
<td>447</td>
</tr>
<tr>
<td>6</td>
<td>40</td>
<td>2.70</td>
<td>384</td>
</tr>
<tr>
<td>7</td>
<td>31</td>
<td>1.45</td>
<td>291</td>
</tr>
<tr>
<td>8</td>
<td>16.4</td>
<td>0.52</td>
<td>150</td>
</tr>
</tbody>
</table>

Area of Fan Drift = 130 sq. ft.

### Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb°F</th>
<th>Wet Bulb°F</th>
<th>Barometered Inches</th>
<th>Dew Point°F</th>
<th>Relative Humidity</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>60.65</td>
<td>54.45</td>
<td>29.73</td>
<td>49.70</td>
<td>67.10</td>
<td>.075507</td>
<td>Surface</td>
</tr>
<tr>
<td>57.85</td>
<td>57.2</td>
<td>30.22</td>
<td>56.80</td>
<td>96.00</td>
<td>.077391</td>
<td>Bottom of D. Fan Drift</td>
</tr>
<tr>
<td>59.1</td>
<td>58.2</td>
<td>29.64</td>
<td>54.15</td>
<td>84.20</td>
<td>.075428</td>
<td>Bottom of U. Fan Drift</td>
</tr>
<tr>
<td>61.2</td>
<td>60.65</td>
<td>30.185</td>
<td>60.3</td>
<td>97.2</td>
<td>.076770</td>
<td></td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.64 lbs/sq.ft.
Dunnikier Colliery, Fifeshire.
12th December, 1925.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Volume in Kilolitres</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Normal</td>
</tr>
<tr>
<td>1</td>
<td>55.8</td>
<td>5.86</td>
<td>594</td>
<td>1.287</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-0.14</td>
<td>134</td>
<td>0.205</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>62</td>
<td>7.56</td>
<td>656</td>
<td>1.421</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>48.5</td>
<td>4.53</td>
<td>512</td>
<td>1.109</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>43</td>
<td>3.60</td>
<td>476</td>
<td>1.031</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>36.5</td>
<td>2.57</td>
<td>392</td>
<td>0.849</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>29</td>
<td>1.54</td>
<td>318</td>
<td>0.689</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>19.5</td>
<td>0.57</td>
<td>226</td>
<td>0.490</td>
<td></td>
</tr>
</tbody>
</table>

Area of Fan Drift = 130 square feet
* Area for Test No.2 = 92.13 square feet.

Atmospheric Observations:

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer Inches</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>37.9</td>
<td>35</td>
<td>29.905</td>
<td>32.2</td>
<td>79.2</td>
<td>0.079592</td>
<td>Surface</td>
</tr>
<tr>
<td>46.33</td>
<td>45.9</td>
<td>30.471</td>
<td>45.59</td>
<td>69.5</td>
<td>0.079339</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>50.5</td>
<td>50.5</td>
<td>29.830</td>
<td>50.5</td>
<td>100.0</td>
<td>0.077233</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>58.7</td>
<td>58.4</td>
<td>30.43</td>
<td>58.2</td>
<td>9810</td>
<td>0.077779</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 1.58 lbs/sq.ft.
### Dunnikier Colliery.

27th March 1926.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Volume in Kilocosecs</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>60</td>
<td>6.71</td>
<td>565</td>
<td>1.224</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>66</td>
<td>8.05</td>
<td>643</td>
<td>1.393</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>-0.01</td>
<td>126</td>
<td>0.193</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>5.6</td>
<td>0.03</td>
<td>196</td>
<td>0.301</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>15.2</td>
<td>0.39</td>
<td>276</td>
<td>0.424</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>23</td>
<td>0.90</td>
<td>256</td>
<td>0.555</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>29.8</td>
<td>1.61</td>
<td>320</td>
<td>0.693</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>34</td>
<td>2.07</td>
<td>357</td>
<td>0.774</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>40</td>
<td>2.97</td>
<td>394</td>
<td>0.854</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>47.8</td>
<td>4.32</td>
<td>470</td>
<td>1.018</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>54.3</td>
<td>5.54</td>
<td>528</td>
<td>1.144</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>44</td>
<td>3.66</td>
<td>429</td>
<td>0.930</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>0</td>
<td>-0.002</td>
<td>122</td>
<td>0.187</td>
<td></td>
</tr>
</tbody>
</table>

**Note:** For tests 3, 4, 5 and 13 Area = 92.13 sq. ft. Other tests " = 130 sq. ft.

### Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb Temp F.</th>
<th>Wet Bulb Temp F.</th>
<th>Barometer Inches</th>
<th>Dew Point F.</th>
<th>Relative Humidity Per cent</th>
<th>Wt. of air 1 cu. ft.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>46.85</td>
<td>45.06</td>
<td>29.481</td>
<td>43.23</td>
<td>87.52</td>
<td>.076969</td>
<td>Surface</td>
</tr>
<tr>
<td>50.03</td>
<td>50.5</td>
<td>30.340</td>
<td>50.25</td>
<td>97.68</td>
<td>.078530</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>52.60</td>
<td>52.22</td>
<td>30.423</td>
<td>51.98</td>
<td>97.28</td>
<td>.079556</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>59.02</td>
<td>57.91</td>
<td>30.301</td>
<td>57.08</td>
<td>93.28</td>
<td>.077986</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.89 lbs/sq.ft.
Kinglassie Colliery, Fife,shire.
27th December, 1925.

Fan:- 18 ft. diameter Walker "Indestructible" - double inlet rope driven by steam engine.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Volume in Kiloseconds</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>98½</td>
<td>12.27</td>
<td>573</td>
<td>51.27</td>
<td>1.279</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-0.05</td>
<td>95</td>
<td>0.212</td>
<td>Normal speed</td>
</tr>
<tr>
<td>3</td>
<td>113½</td>
<td>15.65</td>
<td>656</td>
<td>1.468</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>84½</td>
<td>9.03</td>
<td>506</td>
<td>1.129</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>72.8</td>
<td>6.79</td>
<td>444</td>
<td>0.991</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>65½</td>
<td>5.51</td>
<td>406</td>
<td>0.906</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>55</td>
<td>3.92</td>
<td>345</td>
<td>0.780</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>45</td>
<td>2.62</td>
<td>287</td>
<td>0.640</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>37</td>
<td>1.61</td>
<td>235</td>
<td>0.524</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>27½</td>
<td>0.92</td>
<td>186</td>
<td>0.415</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>0</td>
<td>-0.04</td>
<td>79</td>
<td>0.176</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>91½</td>
<td>10.69</td>
<td>554</td>
<td>1.236</td>
<td></td>
</tr>
</tbody>
</table>

Area of Fan Drift = 133.86 sq. ft.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb F</th>
<th>Wet Bulb F</th>
<th>Barometer Inches</th>
<th>Dew Point F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>40.70</td>
<td>40.70</td>
<td>28.880</td>
<td>40.70</td>
<td>100.00</td>
<td>.076355</td>
<td>Surface</td>
</tr>
<tr>
<td>54.16</td>
<td>52.60</td>
<td>29.983</td>
<td>51.40</td>
<td>90.68</td>
<td>.077080</td>
<td>Bottom of D.C. Fan Drift</td>
</tr>
<tr>
<td>54.33</td>
<td>53.90</td>
<td>28.750</td>
<td>52.80</td>
<td>94.88</td>
<td>.073850</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>63.16</td>
<td>61.66</td>
<td>29.850</td>
<td>60.77</td>
<td>92.22</td>
<td>.075269</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 2.26 lbs/sq.ft.
### Kinglassie Colliery
28th March 1926.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements Velocity ft/min.</th>
<th>Volume in Kilolitres</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>93½</td>
<td>12.00</td>
<td>578</td>
<td>1.290</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>- 0.01</td>
<td>75</td>
<td>0.167</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>27</td>
<td>2.77</td>
<td>296</td>
<td>0.660</td>
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</tr>
<tr>
<td>4</td>
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<td>1.91</td>
<td>192</td>
<td>0.428</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>35</td>
<td>1.67</td>
<td>225</td>
<td>0.502</td>
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</tr>
<tr>
<td>6</td>
<td>24</td>
<td>3.69</td>
<td>331</td>
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</tr>
<tr>
<td>7</td>
<td>70</td>
<td>6.61</td>
<td>455</td>
<td>1.015</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>44/5</td>
<td>4.77</td>
<td>392</td>
<td>0.875</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>64</td>
<td>5.59</td>
<td>405</td>
<td>0.904</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>60</td>
<td>10.48</td>
<td>554</td>
<td>1.236</td>
<td></td>
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<tr>
<td>11</td>
<td>83</td>
<td>9.12</td>
<td>538</td>
<td>1.200</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>75</td>
<td>7.37</td>
<td>475</td>
<td>1.060</td>
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</tr>
</tbody>
</table>

#### Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer inches Hg</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>44.98</td>
<td>43.28</td>
<td>29.233</td>
<td>41.51</td>
<td>87.11</td>
<td>0.076628</td>
<td>Surface</td>
</tr>
<tr>
<td>55.30</td>
<td>54.63</td>
<td>30.475</td>
<td>53.76</td>
<td>92.44</td>
<td>0.078054</td>
<td>Bottom of D.C</td>
</tr>
<tr>
<td>54.87</td>
<td>54.67</td>
<td>29.147</td>
<td>54.55</td>
<td>98.72</td>
<td>0.074773</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>61.16</td>
<td>60.1</td>
<td>30.347</td>
<td>59.45</td>
<td>94.14</td>
<td>0.076847</td>
<td>Bottom of U.C</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 1.60 lbs./sq.ft
Valleyfield Colliery, Fife, Scotland.

25th July, 1925.

Fan: 18 ft. diameter Walker "Indestructible" - double inlet.
Rope driven by Steam Engine.

<table>
<thead>
<tr>
<th>Test</th>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Velocity ft/min.</td>
</tr>
<tr>
<td>1</td>
<td>142</td>
<td>19.82</td>
<td></td>
<td>974</td>
</tr>
<tr>
<td>2</td>
<td>183</td>
<td>16.96</td>
<td></td>
<td>916</td>
</tr>
<tr>
<td>3</td>
<td>135</td>
<td>16.86</td>
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<td>917</td>
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<td></td>
<td>749</td>
</tr>
<tr>
<td>5</td>
<td>0</td>
<td>0.02</td>
<td></td>
<td>37*</td>
</tr>
<tr>
<td>6</td>
<td>86.5</td>
<td>7.58</td>
<td></td>
<td>623</td>
</tr>
<tr>
<td>7</td>
<td>68</td>
<td>4.72</td>
<td></td>
<td>473</td>
</tr>
</tbody>
</table>

Area of Fan Drift = 170.33 sq. ft.

Atmospheric Observations

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer Inches of Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>63.25</td>
<td>60.15</td>
<td>29.825</td>
<td>58.30</td>
<td>84.10</td>
<td>0.75234</td>
<td>Surface</td>
</tr>
<tr>
<td>62.00</td>
<td>61.00</td>
<td>31.225</td>
<td>59.15</td>
<td>84.30</td>
<td>0.78660</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>61.60</td>
<td>60.75</td>
<td>29.630</td>
<td>60.23</td>
<td>95.52</td>
<td>0.74943</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>60.93</td>
<td>60.06</td>
<td>30.105</td>
<td>59.52</td>
<td>95.38</td>
<td>0.76264</td>
<td>600 feet down U.C.</td>
</tr>
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</table>

Calculated Natural Ventilating Pressure = 1.18 lbs/sq.ft.
Valleyfield Colliery.
26th December, 1925

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements lbf./min</th>
<th>Volume in Kilocusecs</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>112.3</td>
<td>14.48</td>
<td>877</td>
<td>2.49</td>
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</tr>
<tr>
<td>2</td>
<td>152.5</td>
<td>23.94</td>
<td>1042</td>
<td>2.95</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>126.3</td>
<td>16.14</td>
<td>885</td>
<td>2.52</td>
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<tr>
<td>4</td>
<td>0</td>
<td>-0.12</td>
<td>227</td>
<td>0.64</td>
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</tr>
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<td>134.8</td>
<td>18.61</td>
<td>920</td>
<td>2.61</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>99.5</td>
<td>9.36</td>
<td>722</td>
<td>2.050</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>112.5</td>
<td>11.92</td>
<td>779</td>
<td>2.050</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>770</td>
<td>5.24</td>
<td>291</td>
<td>1.050</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>85.6</td>
<td>5.94</td>
<td>630</td>
<td>1.485</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>63.1</td>
<td>3.59</td>
<td>523</td>
<td>1.141</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>39.6</td>
<td>6.63</td>
<td>402</td>
<td></td>
<td>Normal speed</td>
</tr>
</tbody>
</table>

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb°F</th>
<th>Wet Bulb°F</th>
<th>Barometer Inches Mercury</th>
<th>Dew Point°F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>33.64</td>
<td>33.2</td>
<td>29.446</td>
<td>32.5</td>
<td>95.44</td>
<td>0.079042</td>
<td>Surface</td>
</tr>
<tr>
<td>40.58</td>
<td>38.93</td>
<td>30.853</td>
<td>38.11</td>
<td>91.12</td>
<td>0.081632</td>
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</tr>
<tr>
<td>50.45</td>
<td>50.30</td>
<td>29.912</td>
<td>50.22</td>
<td>99.28</td>
<td>0.075645</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>61.95</td>
<td>60.60</td>
<td>30.625</td>
<td>60.40</td>
<td>94.80</td>
<td>0.077322</td>
<td>Bottom of U.C.</td>
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</tbody>
</table>

Calculated Natural Ventilating Pressure = 4.48 lbs/sq.ft.
Valleyfield Colliery  
20th February 1926

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurement Velocity ft/min.</th>
<th>Volume in Kilocusec</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>131</td>
<td>21.22</td>
<td>993</td>
<td>2.819</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>+ 0.16</td>
<td>166</td>
<td>0.471</td>
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</tr>
<tr>
<td>3</td>
<td>140.4</td>
<td>25.13</td>
<td>1122</td>
<td>3.185</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>89</td>
<td>10.32</td>
<td>708</td>
<td>2.010</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>66</td>
<td>5.89</td>
<td>529</td>
<td>1.502</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>55.5</td>
<td>3.60</td>
<td>485</td>
<td>1.377</td>
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</tr>
<tr>
<td>7</td>
<td>35.6</td>
<td>1.50</td>
<td>354</td>
<td>1.005</td>
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</tr>
<tr>
<td>8</td>
<td>21</td>
<td>0.50</td>
<td>279</td>
<td>0.792</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>78.4</td>
<td>8.12</td>
<td>643</td>
<td>1.825</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>102</td>
<td>13.95</td>
<td>851</td>
<td>2.416</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>113</td>
<td>16.79</td>
<td>928</td>
<td>2.634</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>0</td>
<td>-0.12</td>
<td>163</td>
<td>0.463</td>
<td></td>
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</tbody>
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Atmospheric Observations:

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer Inches Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>49.80</td>
<td>50.00</td>
<td>29.88</td>
<td>47.68</td>
<td>92.32</td>
<td>0.077520</td>
<td>Surface</td>
</tr>
<tr>
<td>50.46</td>
<td>50.00</td>
<td>30.89</td>
<td>48.90</td>
<td>94.76</td>
<td>0.080023</td>
<td>Bottom of B.C.</td>
</tr>
<tr>
<td>54.86</td>
<td>54.13</td>
<td>29.555</td>
<td>53.57</td>
<td>95.42</td>
<td>0.075839</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>61.46</td>
<td>61.07</td>
<td>30.576</td>
<td>60.85</td>
<td>97.77</td>
<td>0.077362</td>
<td>600 ft. down U.C.</td>
</tr>
<tr>
<td>62.98</td>
<td>61.93</td>
<td>30.85</td>
<td>61.35</td>
<td>94.5</td>
<td>0.077837</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 2.05 lbs./sq.ft.
Wellsley Colliery, Fifeshire
19th July, 1925.

Fan: 21 feet diameter Waddle - steam driven.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
<th>Volume in Kilograms</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>0.01</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>64</td>
<td>4.00</td>
<td>340</td>
<td>0.879</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>60.4</td>
<td>4.99</td>
<td>388</td>
<td>1.004</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>69.6</td>
<td>6.45</td>
<td>428</td>
<td>1.102</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>78.8</td>
<td>8.16</td>
<td>485</td>
<td>1.255</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>92.4</td>
<td>11.50</td>
<td>574</td>
<td>1.487</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>108.6</td>
<td>15.54</td>
<td>672</td>
<td>1.738</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>117.0</td>
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<td>721</td>
<td>1.863</td>
<td>Normal speed</td>
</tr>
<tr>
<td>9</td>
<td>128.0</td>
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<td>780</td>
<td>2.018</td>
<td></td>
</tr>
<tr>
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<td>24.48</td>
<td>845</td>
<td>2.185</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>0</td>
<td>0.073</td>
<td>60*</td>
<td>-0.155</td>
<td>*By smoke</td>
</tr>
</tbody>
</table>

Note: - Reversed Flow in Test No. 11
Area of Fan Drift = 155.25 sq. ft.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer Inches Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>65.00</td>
<td>61.25</td>
<td>29.182</td>
<td>59.00</td>
<td>81.00</td>
<td>0.074958</td>
<td>Surface</td>
</tr>
<tr>
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<td>65.00</td>
<td>31.13</td>
<td>64.22</td>
<td>95.88</td>
<td>0.077929</td>
<td>1305 feet down D.C.</td>
</tr>
<tr>
<td>69.25</td>
<td>65.80</td>
<td>31.305</td>
<td>64.00</td>
<td>95.06</td>
<td>0.077991</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>66.50</td>
<td>65.50</td>
<td>29.635</td>
<td>65.0</td>
<td>95.00</td>
<td>0.074168</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>66.70</td>
<td>65.80</td>
<td>30.960</td>
<td>65.35</td>
<td>95.60</td>
<td>0.077472</td>
<td>1313 feet down U.C.</td>
</tr>
<tr>
<td>75.50</td>
<td>73.00</td>
<td>31.030</td>
<td>71.95</td>
<td>88.80</td>
<td>0.076224</td>
<td>1325 feet down U.C.</td>
</tr>
<tr>
<td>70.0</td>
<td>68.35</td>
<td>31.255</td>
<td>67.6</td>
<td>92.20</td>
<td>0.077664</td>
<td>Bottom of D.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.85 lbs/sq.ft.
Wellesley Colliery
13th December 1925

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements Velocity ft/min.</th>
<th>Volume in Kiloliters</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>-0.107</td>
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<td>0.593</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>140</td>
<td>25.71</td>
<td>872</td>
<td>2.256</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>130.7</td>
<td>23.32</td>
<td>821</td>
<td>2.124</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>122</td>
<td>20.58</td>
<td>777</td>
<td>2.01</td>
<td></td>
</tr>
<tr>
<td>5</td>
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<td>17.88</td>
<td>730</td>
<td>1.889</td>
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</tr>
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<td>685</td>
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<td>608</td>
<td>1.573</td>
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<td>8.17</td>
<td>526</td>
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<td>66</td>
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<td>1.224</td>
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<td>300</td>
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<td>0.569</td>
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</tr>
</tbody>
</table>

Area of Fan Drift = 155.25 sq. ft.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb of F</th>
<th>Wet Bulb of F</th>
<th>Barometer of Hg</th>
<th>Dew Point of °F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>35.75</td>
<td>31.90</td>
<td>29.90</td>
<td>31.30</td>
<td>85.40</td>
<td>.079926</td>
<td>Surface</td>
</tr>
<tr>
<td>53.50</td>
<td>47.90</td>
<td>31.46</td>
<td>47.90</td>
<td>82.00</td>
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<td>Bottom of D.C.</td>
</tr>
<tr>
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<td>Fan Drift</td>
</tr>
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<td>69.16</td>
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<td>31.25</td>
<td>65.30</td>
<td>88.28</td>
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</tr>
<tr>
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<td>67.55</td>
<td>98.00</td>
<td>.078269</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 5.1 lbs/sq. ft.
FAN:- 119 inch Diameter Sirocco - double inlet - steam driven.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.F.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurement Volume in Kilomessos</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>214</td>
<td>29.60</td>
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</tr>
<tr>
<td>2</td>
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<td>264</td>
</tr>
<tr>
<td>3</td>
<td>177</td>
<td>21.85</td>
<td>828</td>
</tr>
<tr>
<td>4</td>
<td>166¾</td>
<td>18.58</td>
<td>761</td>
</tr>
<tr>
<td>5</td>
<td>181.6</td>
<td>11.20</td>
<td>598</td>
</tr>
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<td>6</td>
<td>108</td>
<td>7.85</td>
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</tr>
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<td>90</td>
<td>5.36</td>
<td>449</td>
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<tr>
<td>8</td>
<td>33</td>
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<td>62.6</td>
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<tr>
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<td>157</td>
<td>15.64</td>
<td>721</td>
</tr>
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<td>12</td>
<td>0</td>
<td>-0.02</td>
<td>178</td>
</tr>
<tr>
<td>13</td>
<td>0</td>
<td>-0.18</td>
<td>223</td>
</tr>
</tbody>
</table>

Note: In Test No. 13 the air was passing through the Waddle fan - Sirocco being sealed off and standing.

Atmospheric Observations:

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer Inches of Mercury</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>44.25</td>
<td>43.50</td>
<td>29.750</td>
<td>42.76</td>
<td>94.00</td>
<td>.078066</td>
<td>Surface</td>
</tr>
<tr>
<td>55.87</td>
<td>53.75</td>
<td>30.897</td>
<td>54.20</td>
<td>94.30</td>
<td>.079137</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>61.96</td>
<td>61.96</td>
<td>29.366</td>
<td>61.96</td>
<td>100.00</td>
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<td>Fan Drift</td>
</tr>
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<td>69.45</td>
<td>68.47</td>
<td>30.088</td>
<td>68.35</td>
<td>96.6</td>
<td>.076823</td>
<td>1325 ft. down U.C.</td>
</tr>
<tr>
<td>67.66</td>
<td>67.23</td>
<td>30.89</td>
<td>67.00</td>
<td>97.68</td>
<td>.077120</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 2.95 lbs/sq ft
Fan:- 91 inch diameter Sirocco - double inlet - steam driven.

<table>
<thead>
<tr>
<th>No. of Feet</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>231</td>
<td>15.67</td>
<td>1346</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-1.23</td>
<td>538</td>
</tr>
<tr>
<td>3</td>
<td>211</td>
<td>13.53</td>
<td>1302</td>
</tr>
<tr>
<td>4</td>
<td>193</td>
<td>10.67</td>
<td>1166</td>
</tr>
<tr>
<td>5</td>
<td>170</td>
<td>8.25</td>
<td>1100</td>
</tr>
<tr>
<td>6</td>
<td>150.7</td>
<td>6.42</td>
<td>1027</td>
</tr>
<tr>
<td>7</td>
<td>131</td>
<td>4.50</td>
<td>938</td>
</tr>
<tr>
<td>8</td>
<td>117</td>
<td>2.69</td>
<td>858</td>
</tr>
<tr>
<td>9</td>
<td>107</td>
<td>1.67</td>
<td>774</td>
</tr>
<tr>
<td>10</td>
<td>93</td>
<td>0.39</td>
<td>725</td>
</tr>
<tr>
<td>11</td>
<td>0</td>
<td>-1.35</td>
<td>561</td>
</tr>
</tbody>
</table>

Area of Fan Drift = 128.5 sq. feet.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb °F</th>
<th>Wet Bulb °F</th>
<th>Barometer Inches</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>35.40</td>
<td>34.98</td>
<td>29.543</td>
<td>34.40</td>
<td>90.00</td>
<td>0.009055</td>
<td>Surface</td>
</tr>
<tr>
<td>48.6</td>
<td>44.6</td>
<td>30.875</td>
<td>40.3</td>
<td>72.80</td>
<td>0.006030</td>
<td>Bottom of No. 1 D.C.</td>
</tr>
<tr>
<td>47.25</td>
<td>44.75</td>
<td>31.66</td>
<td>42.15</td>
<td>82.6</td>
<td>0.002593</td>
<td>Bottom of No. 2 D.C.</td>
</tr>
<tr>
<td>60.16</td>
<td>57.62</td>
<td>29.345</td>
<td>55.95</td>
<td>86.16</td>
<td>0.004496</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>72.30</td>
<td>64.20</td>
<td>30.90</td>
<td>59.50</td>
<td>64.4</td>
<td>0.006615</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 8.37 lbs. / sq. ft.
Silksworth Colliery, Co. Durham.
20th December, 1921.

Fan: 12 ft. diameter - double inlet - steam or electrically driven.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurements:</th>
<th>Volume in Kilocusecs</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>246</td>
<td>33.144</td>
<td>968</td>
<td>3.270</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>-0.33</td>
<td>287</td>
<td>0.970</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>205</td>
<td>24.03</td>
<td>831</td>
<td>2.808</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>190</td>
<td>19.58</td>
<td>770</td>
<td>2.601</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>181</td>
<td>18.30</td>
<td>746</td>
<td>2.517</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>173</td>
<td>16.22</td>
<td>706</td>
<td>2.385</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>97</td>
<td>3.53</td>
<td>486</td>
<td>1.642</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>108</td>
<td>5.11</td>
<td>529</td>
<td>1.787</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>122.5</td>
<td>2.64</td>
<td>560</td>
<td>1.642</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>146</td>
<td>10.41</td>
<td>615</td>
<td>2.078</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>160</td>
<td>13.54</td>
<td>668</td>
<td>2.257</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>0</td>
<td>-0.06</td>
<td>302</td>
<td>1.020</td>
<td></td>
</tr>
</tbody>
</table>

Area of Fan Drift = 202.71 sq. ft.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb F</th>
<th>Wet Bulb F</th>
<th>Barometer Inches</th>
<th>Dew Point °F</th>
<th>Relative Humidity per cent.</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>36.16</td>
<td>35.80</td>
<td>28.76</td>
<td>35.30</td>
<td>96.56</td>
<td>.076782</td>
<td>Surface</td>
</tr>
<tr>
<td>45.00</td>
<td>41.60</td>
<td>29.84</td>
<td>37.55</td>
<td>75.10</td>
<td>.078125</td>
<td>1125 feet down D.C.</td>
</tr>
<tr>
<td>48.25</td>
<td>44.10</td>
<td>30.505</td>
<td>39.40</td>
<td>71.30</td>
<td>.079476</td>
<td>1621 feet down D.C.</td>
</tr>
<tr>
<td>48.63</td>
<td>44.23</td>
<td>30.649</td>
<td>39.30</td>
<td>70.35</td>
<td>.079795</td>
<td>1740 feet down D.C.</td>
</tr>
<tr>
<td>60.43</td>
<td>59.93</td>
<td>28.30</td>
<td>59.63</td>
<td>97.39</td>
<td>.071728</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>65.10</td>
<td>62.00</td>
<td>29.595</td>
<td>60.20</td>
<td>84.20</td>
<td>.074354</td>
<td>1121 feet down U.C.</td>
</tr>
<tr>
<td>64.80</td>
<td>58.95</td>
<td>30.200</td>
<td>55.07</td>
<td>70.55</td>
<td>.076011</td>
<td>1616 feet down U.C.</td>
</tr>
<tr>
<td>74.30</td>
<td>66.05</td>
<td>30.130</td>
<td>61.65</td>
<td>64.70</td>
<td>.074364</td>
<td>1734 feet down U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 7.76 lbs/ sq. ft.
Fan:- 175 inch diameter Sirocco; single inlet - rope driven by steam engine.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure in lbs. per sq. ft.</th>
<th>Air Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Velocity in ft/min.</td>
</tr>
<tr>
<td>1</td>
<td>119</td>
<td>17.67</td>
<td>306</td>
</tr>
<tr>
<td>2</td>
<td>128</td>
<td>19.84</td>
<td>324</td>
</tr>
<tr>
<td>3</td>
<td>110.6</td>
<td>15.40</td>
<td>272</td>
</tr>
<tr>
<td>4</td>
<td>92.8</td>
<td>11.12</td>
<td>263</td>
</tr>
<tr>
<td>5</td>
<td>76</td>
<td>7.82</td>
<td>243</td>
</tr>
<tr>
<td>6</td>
<td>59.7</td>
<td>4.96</td>
<td>212</td>
</tr>
<tr>
<td>7</td>
<td>43.3</td>
<td>2.66</td>
<td>199</td>
</tr>
<tr>
<td>8</td>
<td>30.4</td>
<td>1.38</td>
<td>183</td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>-0.06</td>
<td>154</td>
</tr>
</tbody>
</table>

Area of Fan Drift = 227.5 sq. ft.

Atmospheric Observations

<table>
<thead>
<tr>
<th>Dry Bulb Temp</th>
<th>Wet Bulb Temp</th>
<th>Barometer</th>
<th>Dew Point</th>
<th>Relative Humidity</th>
<th>Wt. of 1 cu. ft. air</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>41.30</td>
<td>38.98</td>
<td>29.62</td>
<td>36.05</td>
<td>81.50</td>
<td>0.078265</td>
<td>Surface</td>
</tr>
<tr>
<td>52.38</td>
<td>50.60</td>
<td>30.913</td>
<td>49.14</td>
<td>88.51</td>
<td>0.079787</td>
<td>Bottom of D.C.</td>
</tr>
<tr>
<td>57.90</td>
<td>57.33</td>
<td>29.407</td>
<td>56.95</td>
<td>97.10</td>
<td>0.074964</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>69.00</td>
<td>66.62</td>
<td>30.933</td>
<td>65.35</td>
<td>91.40</td>
<td>0.077965</td>
<td>Bottom of U.C.</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 6.45 lbs./sq.ft.
Craven Colliery, Warwickshire.

13th February, 1926.

Fan: 49 inch diameter Sirocco Double Inlet - Direct Driven by D.C. motor.

<table>
<thead>
<tr>
<th>No. of Test</th>
<th>Fan Speed R.P.M.</th>
<th>Fan Drift Pressure lbs. per sq. ft.</th>
<th>Air Measurement Velocity ft/min</th>
<th>Volume in Kilolitres</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>269</td>
<td>3.72</td>
<td>646</td>
<td>0.517</td>
<td>Normal speed</td>
</tr>
<tr>
<td>2</td>
<td>388</td>
<td>8.34</td>
<td>861</td>
<td>0.689</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>316.2</td>
<td>5.54</td>
<td>729</td>
<td>0.583</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>243.2</td>
<td>7.37</td>
<td>509</td>
<td>0.487</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>216.4</td>
<td>5.66</td>
<td>578</td>
<td>0.462</td>
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</tr>
<tr>
<td>6</td>
<td>182.2</td>
<td>1.99</td>
<td>508</td>
<td>0.406</td>
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</tr>
<tr>
<td>7</td>
<td>190</td>
<td>1.78</td>
<td>488</td>
<td>0.390</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>0</td>
<td>0.23</td>
<td>240</td>
<td>0.192</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>282</td>
<td>4.55</td>
<td>663</td>
<td>0.536</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>336.4</td>
<td>6.82</td>
<td>769</td>
<td>0.615</td>
<td></td>
</tr>
</tbody>
</table>

Area of Fan Drift = 48 sq. ft.

Atmospheric Observations.

<table>
<thead>
<tr>
<th>Dry Bulb F</th>
<th>Wet Bulb F</th>
<th>Barometer Inches Mercury</th>
<th>Dew Point F</th>
<th>Relative Humidity per cent</th>
<th>Wt. of 1 cu. ft. air lbs.</th>
<th>Place of Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>37.22</td>
<td>35.67</td>
<td>29.760</td>
<td>33.50</td>
<td>86.10</td>
<td>0.079303</td>
<td>Surface</td>
</tr>
<tr>
<td>38.93</td>
<td>37.79</td>
<td>30.096</td>
<td>36.30</td>
<td>90.48</td>
<td>0.079993</td>
<td>Bottom of D.O</td>
</tr>
<tr>
<td>50.60</td>
<td>50.60</td>
<td>29.69</td>
<td>50.68</td>
<td>100.00</td>
<td>0.076866</td>
<td>Fan Drift</td>
</tr>
<tr>
<td>53.50</td>
<td>52.64</td>
<td>29.95</td>
<td>52.00</td>
<td>95.00</td>
<td>0.077086</td>
<td>Bottom of U.O</td>
</tr>
</tbody>
</table>

Calculated Natural Ventilating Pressure = 0.66 lbs/sq.ft.
TRANSACTIONS of the INSTITUTION of MINING ENGINEERS.


Briggs & Williamson: An Experimental Study of Fan Evases, Vol. LXVIII, p. 323.


Hare: Presidential Address, Vol. LXVI, p. 63.


133.

TRANSACTIONS of OTHER INSTITUTIONS.

Bouvat-Martin: - Considerations sur la Resistance Interieure des Ventilateurs,

Fairweather: - The Resistance of the Air to the Motion of Fans,

Flugge-de-Smidt: - The Effect on the Ventilating Current of Stopping a Mine Fan,


Lacey: - Flow of Gas In Pipes,

Morris: - The Magregor-Morris Anemometer
Trans. Brit. Assoc., 1922, Section A.

Penman and Wetherall: - Experiments on the Flow of Air in Mines,
Min. and Geol. Inst. India, 1925.

Stanton and Pannel: - Similarity of Motion in Relation to the Surface Friction of Fluids
Phil. Trans. 1914, Vol. 214 (X) p. 199.

Stott: - The Characteristics of the Propeller Fan,

Telland and Ransom: - Rock and Air Temperatures in Deep Lavel Mines,

Walker: - Propeller Ventilating Fans,

PUBLICATIONS of H.M. STATIONARY OFFICE.

Experiments with a Family of Airscrews
Reports and Memoranda, No. 829.

Dependence of the Efficiency of an Airscrew on the Speed and the Diameter
Reports and Memoranda, No. 442.

MINING TECHNICAL JOURNALS.


*Practical Points for Electrical and Mechanical Men.*  
Vol. 11th March, 1926, p. 368.

Coll. Engineering: - *Fan Problems* (Briggs)  
Vol. 2, No. 16, June 1925.

TEXT BOOKS.

(Griffin and Co., 1919)

Griffiths: - *Engineering Instruments and Meters.*  
(Routledge, 1920)


Watts: - *The Design of Screw Propellers for Aircraft.*  
(Longmans, Green & Co. 1920)