

(Read 30 April 1897.)

## EXPERIMENTS UPON PROPELLER VENTILATING FANS, AND UPON THE ELECTRIC MOTOR DRIVING THEM.

BY MR. WILLIAM GEORGE WALKER, OF LONDON.

*Former Experiments.*—In 1892 the author read a paper to this Institution (Proceedings, page 514) upon experiments on the arrangement of the surface of a screw-propeller; and in the discussion he mentioned (page 561) some experiments which he had made with Air Propellers, in order to try the effect of the thickness of the blades. In 1893 at the Nottingham meeting of the British Association he read a note on some experiments which he had made with Ventilating Fans having blades of various cross sections. Those preliminary trials led him to undertake the further experiments described in the present paper, which have been carried out during 1895–6 in his laboratory in Westminster. The primary purpose was to ascertain:—

1. Whether this kind of fan follows the ordinary laws respecting the mutual relations of speed of fan, power absorbed, and amount of air discharged.

2. The general characteristics regarding the speed of fan, power absorbed, and quantity of air discharged, with different angles of the blades.

3. The effect of fans differing from one another only in the cross-section of their blades.

As he was not acquainted with any systematic experiments upon this kind of fan, the investigation of the relations of speed, power absorbed, and air discharged was a necessary preliminary to the modification arising from different angles and shapes and cross sections of the blades. The knowledge already acquired on the subject is confined principally to the centrifugal form of ventilator, and may be said to consist of a tolerably accurate and complete

determination of those characteristics of performance which are common to all centrifugal fans of ordinary pattern, together with a somewhat vague appreciation of the character of the difference in design upon which the efficiency of a centrifugal fan depends.

*Propeller Ventilating Fans.*—The experiments on the Propeller Ventilating Fans showed that, in fans tried under the same conditions, the relations of speed of fan, power absorbed, and air discharged, are in accordance with the ordinary laws. The propeller, or helical form of fan as it is sometimes called, is essentially a ventilator, its principal object being to move large volumes of air at atmospheric pressure. It is unable to maintain a static pressure of air, which would be absolutely necessary if the air were met by opposing resistances. The propeller ventilating fan is of use where comparatively large quantities of air have to be exhausted from a building; its volumetric efficiency is therefore of much greater importance than its pressure efficiency, and in many cases is also of greater importance than its mechanical efficiency. All the results given in the present paper are with fans having a free discharge, the outlet being practically equal in area to the inlet of the fan. The ordinary centrifugal blowing fan generally works with a contracted outlet, its static pressure is comparatively high, and its mechanical efficiency is a maximum at a certain contraction of the outlet which gives a certain static pressure, while its volumetric efficiency is correspondingly low. The converse is the case with the propeller ventilating fan; its volumetric efficiency is a maximum with free discharge, and rapidly falls off with any baffling of the outlet pipe. It is important to avoid giving any shock to the air; the outlet and inlet should be as free as possible; sharp turns or blocking in the delivery pipe may reduce the efficiency to a considerable extent, owing to the fact that the flow of air is not assisted by any static pressure, but is due only to its velocity.

*Experimental Apparatus.*—Seventeen three-bladed fans were tried, all  $23\frac{3}{4}$  inches diameter, specially constructed for these experiments; they are shown in Figs. 1 to 17, Plate 79. The fans were driven

by an electric motor, fixed centrally to a cast-iron frame in the rear of the fans, Figs. 18 and 19, Plate 80, which were keyed to the spindle of the armature, and were thus driven direct at the same speed as the motor. The latter was a continuous-current series-wound machine of about one-third of an electrical horse-power. The current was taken off the 100-volt mains of the Westminster Electric Supply Corporation. The air was delivered through a tube 24 inches bore and 4 feet long, Figs. 21 and 22, Plate 81, made of stiff sheet-iron and placed concentric with the fan axis and at the end of the frame. The speed of the fan was indicated by a tachometer, which was attached by Dr. Hook's joint to the spindle of the motor, and was read to two revolutions per minute. The usual meters were employed for registering the number of volts and amperes. The speed of the motor was varied by means of electrical resistances. In most of the experiments the fans were run at a speed of 600 revolutions per minute; the speed was kept constant during each experiment by the adjustment of a suitable form of resistance. The volts in the supply mains were found to vary somewhat, which had an effect on the speed of the motor; during certain times of the day an almost continuous process of adjustment of the electrical resistance was necessary, in order to keep the volts quite constant.

The velocity of the air was measured by an anemometer of  $2\frac{3}{4}$  inches diameter, placed at the outer end of the delivery tube, as shown in Figs. 21 and 22, Plate 81. The velocity varied greatly in different positions in the same cross section of the tube; a smooth brass rod  $5\text{--}16$ ths inch diameter was placed horizontally across the end of the tube, to which the anemometer was attached, so that it could be tried at different positions on the rod. The centre of the anemometer moved in the horizontal diameter of the tube for all positions on the rod, and the instrument always moved in the same plane across the current.

The brake horse-power of the motor was obtained by a dynamometric brake, of which the arrangement is shown in Fig. 20, Plate 80. The fan having been removed, the brake pulley was fixed in the same position upon the spindle. The brake was highly effective; it was sensitive, yet ran steady with no oscillation. The pulley

was of cast-iron, 9.4 inches diameter, with smooth circumference; it ran true, and a fine silk cord was wound once round it, the lower end being attached to the scale pan, and the upper end to a Salter's balance hung above the pulley and perpendicularly over the scale pan. The difference between the pull on the balance and the weight in the scale pan, multiplied by the circumference of the pulley and the number of revolutions per minute, and divided by 33,000, gave the brake horse-power. The pull and the weights in the scale pan were read in ounces. The silk cord was in direct contact with the surface of the pulley, and no cleats were employed to keep it on.

*Measurement of Air Discharge.*—It is difficult to conduct experiments with fans so as to obtain reliable results, especially in a long series of experiments extending over considerable time, when slight variations in the conditions may affect the results to a material degree. Changes due to slight variations in the position or condition of the apparatus, atmospheric changes of temperature and pressure and moisture, or alteration in condition of the room in which the experiments are made, all tend to affect the results. Having chosen the anemometer method for measuring the delivery of the air, the author made some experiments with a thin plate of small area, attached to the end of a thin well-balanced lever, and placed in the delivery tube so that the air impinged normally against its surface; it was adjusted at right angles to the stream of air by twisting by hand a wire, which formed the fulcrum of the lever, through an angle proportional to the pressure on the plate; the pressure being thus known, the velocity of the air can be calculated from it. This is somewhat similar to the Pitot tube method, which also measures the dynamic pressure of the air. With high velocity of air discharge through a delivery tube of small diameter, where it is necessary to explore the section of the tube to the very edges in order to get the mean velocity of the air, either the Pitot or the plate method is suitable. The anemometer was calibrated at Kew Observatory for speeds varying from 50 to 2,000 feet per minute; and the errors found were identical with those originally notified by the makers of the

instrument. The author also made some experiments himself with the anemometer, and allowing for calibration errors found it to agree fairly well with the Pitot tube method. Mr. Bryan Donkin, to whom his thanks are also due for several suggestions, lent him a carefully made brass Pitot tube,  $\frac{1}{4}$  inch internal diameter, which was fixed with its open end facing the current of air issuing from the delivery tube ; the other end of the Pitot tube was connected to a U shaped water-gauge, in which the height of the column of water was read by means of a strong light and powerful magnifying glass. The dynamic pressure of the air must not be confounded with the static pressure of its compression. The dynamic pressure is due to the impact of the moving air, transmitted from the face of the Pitot tube to the surface of the water column. In obtaining the static pressure it is necessary to eliminate the effect of the velocity of the air ; for in a U shaped water-gauge connected to a chamber of compressed air the static pressure is measured by the height of the column of water, which would be increased if velocity were imparted to the air so that it should impinge against the orifice of a tube connected to the gauge and facing the current. In the fans tested the static pressure was practically nil, because they were tried with free discharge and consequently at atmospheric pressure. If the face of the Pitot tube were reversed, to look in the direction towards which the current was flowing, so that the air would then act upon it by induction instead of by impact, an anomalous reading of the gauge might occur, showing zero pressure when compression was known to exist, the static pressure being neutralized by the induction of the moving air.

For each experiment anemometer readings were taken at each of the four following radii of the delivery tube :  $-1\frac{7}{8}$ ,  $5\frac{1}{8}$ ,  $7\frac{7}{8}$ ,  $10\frac{5}{8}$  inches. As shown in Fig. 23, Plate 81, the cross section of the delivery tube was divided into four imaginary concentric rings, and each of the above radii corresponded with the centre line of one of the rings ; each of the three outer rings was equal in breadth to the diameter of the anemometer. The velocity of the air in feet per minute, as ascertained at each of the four radii, was multiplied by the area of the corresponding rings in square feet ; and the products being added

together gave the number of cubic feet of air discharged per minute. The velocities obtained at each of the four radii are given in Tables 1 and 2 (pages 458-60) for all the fans tried. It will be noticed that the velocity varied considerably at the different radii with different fans; and no mean radius for the position of the anemometer could be obtained. The areas of the four imaginary rings, making up the total 3.141 square feet area of the tube of 2 feet diameter, were 1.275, 0.945, 0.614, and 0.307 square foot. The mean velocity of the air was obtained by dividing the air discharge in cubic feet by the 3.141 square feet area of the delivery tube. Readings of the anemometer were taken for two minutes at each of the four radii for each experiment, together with the volts, ampères, height of barometer in inches, and temperature of the air. The condition of the experimental room remained unchanged throughout the experiments, with all doors and windows shut.

*Measurement of Brake Horse-Power.*—The electrical motor was calibrated, so that by simply reading the ampères and noting the number of revolutions the brake horse-power was obtained. The torque or turning moment of an electrical motor is proportional to the current, and is nearly independent of the speed. A series of experiments were made with the motor running at 600 revolutions per minute, and the experimental readings are shown plotted in the diagram, Fig. 24, Plate 82. The weights on the brake in ounces are plotted as abscissae, and the ampères as ordinates; the line CC drawn through the plottings is a straight one, showing that the weights are proportional to the ampères. The readings range from the motor running light, until the resultant pull on the brake pulley was about 45 ounces. The majority of the fans were run at 600 revolutions per minute; the ampères being noted, the equivalent weight on the brake pulley was read directly off the chart, from which the brake horse-power was calculated. With a given torque the ampères were not quite constant for all speeds of the motor; they increased slightly and uniformly with increase in the number of revolutions. It was easy however to frame a formula which gave the torque at any speed the

motor might be running at, taking into account the small increase in the current due to speed. The small variation of current with speed for the same torque was probably due to the variation of the skin resistance of the armature. It required 0.5 ampère just to rotate the armature, and 0.6 ampère to drive it at 600 revolutions per minute with no external load on; this current was therefore spent in overcoming the friction of the motor; and it required 0.05 ampère to drive the tachometer at 600 revolutions per minute. In finding the brake horse-power, the skin resistance of the brake pulley was allowed for, which was obtained by running the motor light with the pulley off and on successively. The power required to overcome the friction of the motor was nearly constant throughout the whole series of experiments. The motor was frequently calibrated during the experiments, and any slight variation of ampères and torque was allowed for. The ampère meter was read to one-hundredth of an ampère.

It is sometimes convenient in testing different fans to run them so that the same horse-power shall be absorbed by each. In Fig. 25, Plate 82, is drawn a curve for one-tenth of a horse-power, in which the revolutions are plotted as abscissae and the ampères as ordinates. For a particular horse-power with fans of different kinds, the torque and revolutions are varying quantities in relation to each other, though their product will be constant. In using this curve, the speed of the fans must be adjusted by the electrical resistances until the co-ordinates of the revolutions and current meet on the curve; then the motor is transmitting one-tenth of a brake horse-power. The electric motor may therefore be used as a convenient form of transmission dynamometer for testing not only fans but also other machines, the ammeter giving after calibration the power transmitted to the machine that is being tested.

*Speed Characteristics of Electric Motor.*—Although not essential for the present experiments, it is interesting to determine at what speeds the motor should be run so as to give maximum brake horse-power and maximum efficiency severally. Its speed characteristics are shown in Fig. 26, Plate 82, for constant electromotive force; the revolutions.

are plotted as abscissae, and the ordinates of electrical horse-power represent the volts multiplied by the ampères and divided by 746. The three curves show the relation between the speed, the electrical and brake horse-power, and the efficiency of the motor. They were found by the brake, the scale pan being loaded with different weights so as to produce different speeds. The brake horse-power is also plotted as ordinates at the several speeds at which the motor was run; and the ratio of the brake horse-power to the electrical is a measure of the efficiency of the motor; this ratio was calculated for all speeds at which the motor was run, and being plotted as ordinates it gives the efficiency curve. Referring to Fig. 26, it is seen that the electrical horse-power was a maximum at zero speed of the motor, and fell off with increase in the speed; the voltage or potential difference was taken at the terminals of the motor. The brake horse-power increased from zero at zero speed to a maximum of 0.14 B.H.P. at 800 revolutions per minute, after which it fell off with further increase in speed; at 1,500 revolutions per minute it had fallen to 0.07 B.H.P. The efficiency, or ratio of brake horse-power to electrical, rose from zero at zero speed to a maximum of 48.5 per cent. at about 1,200 revolutions per minute, after which it fell off rapidly with further increase in speed; at the speed of 800 revolutions per minute, which gave maximum brake horse-power, the efficiency was about 40 per cent.

*Mutual Relations of Revolutions, Brake Horse-Power, and Air Discharge.*—Fans 1 to 6 were tried at progressive speeds ranging from 300 to 1,000 revolutions per minute, and the following relations were verified for constant angle of blades and position of fan:—(1) air discharge varies as speed of revolution; (2) horse-power varies as (air discharge)<sup>3</sup>. Whence follow—(3) horse-power varies as (speed of revolution)<sup>3</sup>; (4) torque varies as (speed of revolution)<sup>2</sup>; and also (5) torque varies as electric current. Therefore (6) electric current varies as (air discharge)<sup>2</sup>. This last relation (6) applies to the particular motor used and to the limits of the experiments; it does not mean that all electric motors will invariably agree with it. Hence for all speeds of the same fan the ratio of air discharge to speed of revolution, and that of (air discharge)<sup>3</sup> to



brake horse-power, are both of them constant; and putting relation (6) into a different form, the ratio of (air discharge)<sup>2</sup> to ampère current is also constant. The mechanical efficiency of the fan is therefore constant at all speeds within the limits of the range tried, because it varies as the cube of the air discharge and inversely as the brake horse-power. These relations apply only within the limits of speed tried, and to the particular form of fans tested; at higher speeds the air discharge will probably fall off. The experimental results of several of the fans at different speeds have been plotted, the revolutions and air discharge in Fig. 27, Plate 83, and the brake horse-power and air discharge in Fig. 28. The results for each fan have been given in the tables for one speed only, but can be found for any other speed by the above relations.

*Calculation of Horse-power in Air discharged, and of Efficiencies.*—

Weight of one cubic foot of air at temperature  $t^{\circ}$  Fahr. =  $\frac{1.3304 \times B}{T}$  lb.,

where  $B$  = barometer height in inches of mercury, and  $T$  = absolute temperature =  $t^{\circ} + 461^{\circ}$  Fahr. Taking into account the moisture

in the air, weight of one cubic foot of air =  $\frac{1.3304}{T} (B - \frac{3}{8}b)$ , where  $b$  is the pressure due to the moisture in inch of mercury.

If  $W$  = weight of air discharged per second in lbs., and  $V$  = velocity of air in feet per second, then kinetic energy of air

discharged =  $\frac{WV}{2g}$ ; and horse-power of air discharged =  $\frac{WV^2}{2g \times 550} =$

$V^3 \times \text{constant}$ , for same fan under same conditions, which follows from relation (2) in page 446. Therefore if  $Q$  be the quantity of air discharged in cubic feet per second,

$$\begin{aligned} \text{horse-power of air discharged} &= \frac{V^2 Q}{550 \times 64.4} \times \frac{1.3304 \times B}{T} \\ &= \frac{V^2 Q B}{T} \times 0.00003756 \quad . \quad . \quad (7) \end{aligned}$$

Mechanical efficiency =  $\frac{\text{horse-power in air discharged}}{\text{brake horse-power}}$ . Suppose the air to be flowing against the mouth of a Pitot tube connected with a water gauge, and let  $H$  = height of column of air in feet, and  $h$  = equivalent column of water in inches.

$$\text{Then } H = \frac{V^2}{2g} \quad . \quad . \quad . \quad (8)$$

Taking the weight of a cubic foot of water at  $62^{\circ}$  F. to be 62.35 lbs.,

2 R

$$12 H \div h = 62 \cdot 35 \div \frac{1 \cdot 3304 B}{T}; \text{ whence } H = \frac{62 \cdot 35 T h}{1 \cdot 3304 B} = \frac{3 \cdot 9 T h}{B}.$$

The substitution of this value for  $H$  in (8) gives  $h = \frac{V^2 B}{251 T}$ . Whence substituting in (7), horse-power in air discharged  $= h \times 251 \times Q \times 0 \cdot 00003756 = h Q \times 0 \cdot 00943$ , from which the mechanical efficiency can then be determined.

Volumetric efficiency  $= \frac{Q}{U r^2}$ , where  $U$  = velocity of tips of blades in feet per second, and  $r$  = radius of fan in feet.

Pressure efficiency  $= \frac{V^2}{2 U^2}$ , the pressure varying as the square of the velocity.

The three efficiencies—mechanical, volumetric, and pressure—are each of them constant at all speeds with the same fan; this follows from the fact that the air discharge varies as the speed of revolution. There is great want of a recognised standard for comparing fans. Not being acquainted with any reliable experiments on the class of fans here tested, the author felt some hesitation in deciding upon the best method to employ. It may appear paradoxical that the volumetric efficiency can be greater than unity; but that the formula is based on correct principles is shown by the fact that it gives a constant efficiency for the same fan at all speeds. It was first employed, the author understands, by Professor Rateau of St. Etienne. The dynamic pressure varies as the velocity squared. The pressure efficiency varies as the square of the volumetric efficiency, both being a maximum at the same time. The pressure efficiency is not of essential importance in the fans here tested, because it is calculated only from pressure due to velocity of air, and not from compression as it would generally have to be calculated in a centrifugal fan.

*Experiments with Fan Blades at different Angles.*—The angles are those which the plane of the blade makes with the plane of rotation. The experimental results of fan 17 for different angles varying from  $15^\circ$  to  $60^\circ$  are given in Table 2 (page 460), and are plotted in Fig. 29, Plate 84. These may be termed the characteristic curves of the fan for varying angles, showing at a glance its general performance. The angles are plotted as abscissae; the air discharge, brake horse-power, and efficiencies as ordinates. In this example it will be noticed that

the air discharge increases nearly with the angle from  $15^{\circ}$  to  $30^{\circ}$ ; after which the curve rises more and more slowly until  $45^{\circ}$  is reached, and then with further increase of the angle the air discharge rapidly falls off. The brake horse-power increases with fair uniformity until an angle of  $50^{\circ}$  is reached, after which it commences to fall off, showing that the resistance of the fan is not increased by increasing the angle beyond about this inclination. The mechanical efficiency increases with the angle, and reaches the maximum of 42.8 per cent. at an angle of  $27^{\circ}$ , after which it falls off with further increase of the angle. The volumetric and pressure efficiencies increase with the angle, until they reach the maximum of 76.7 and 2.87 per cent. respectively at an angle of  $45^{\circ}$ , after which they fall off with further increase in the angle. A difficulty arises from the mechanical and volumetric efficiencies not being the maximum at the same angle. If the blades were fixed at  $27^{\circ}$ , so as to give the maximum mechanical efficiency, the volumetric and pressure efficiencies would fall to 64.3 and 2.10 per cent. respectively; if placed at  $45^{\circ}$ , so as to give the maximum volumetric and pressure efficiencies, the mechanical efficiency falls to 26.4 per cent., falling somewhat rapidly after  $27^{\circ}$  is reached. The angle of the fan blades must therefore be governed to a certain extent by the nature of the work which the fan is required to perform, and also by the kind of motor employed, whether electric, steam, or gas. In electrically driven fans, supplied with current through a meter, the mechanical efficiency may be of more importance than the volumetric, the latter varying directly as the air discharge and inversely as the revolutions and the cube of the diameter. Taking into account the combined efficiency of motor and fan, it may be advisable to modify the application of the formula for the volumetric efficiency, for the reason that it may be more economical to run the motor at a comparatively high speed, and that the required amount of air discharge would then be obtained with blades set so as to give the maximum mechanical efficiency but not the highest volumetric efficiency.

In calculating the mechanical efficiency, the power due to the friction of the motor has been excluded, so that the actual performance of the fans could be better compared, and that the results should not be affected by a factor which depends only on the

state of the bearings. In Table 3 (page 461) the initial friction of the motor has been included with the brake horse-power, so that the mechanical efficiency of fans 16 and 17 may also be calculated including the friction. The mechanical, volumetric, and pressure efficiencies of fans 16 and 17 have been plotted with the friction of the motor included, in Figs. 30, 31, and 32, Plate 85. By including the friction of the motor, not only is the mechanical efficiency reduced, but also the angle of maximum mechanical efficiency is altered; for instance, the angle which gives the maximum mechanical efficiency is  $30^{\circ}$  in Fig. 30 when the friction is included, instead of  $27^{\circ}$  in Fig. 29 when no friction was included. At very small angles of the blades the horse-power of the air discharged is small, but the comparatively large friction of motor being included in the gross horse-power gives a low mechanical efficiency, which is increased as the air discharge becomes greater.

*Effect of Cross Section of Fan Blades.*—Seventeen three-bladed fans were tried, having blades of the sections shown in Figs. 1 to 17, Plate 79. They may be divided into four groups:—the first comprises the four fans 1 to 4; the second group the six fans 5 to 10; the third group the five fans 11 to 15; and the remaining two are the fans 16 and 17. The blades are of sheet iron 1-16th inch thick; with the exception of fan 10, their cross-sectional lines are all composed of straight lines or arcs of circles. The fans in each group differed from one another only in the cross section of their blades, which were flat, plano-convex, concavo-convex, of different degrees of curvature. Fan 1 had flat blades. Fan 2 was formed by fixing a circular face upon the non-propelling surface or back of the blades of fan 1, giving a plano-convex section to the blades. Fan 3 was formed by curving the blades of fan 1, so that the cross section became concavo-convex, the propelling face being concave. Fan 4 was formed by fixing upon the back or non-propelling face of the blades of fan 3 a still more convex surface, so that the cross section remained concavo-convex with the propelling face concave, but the section was not of uniform thickness. The blades of the other groups were similar in form, but of different area and thickness. In Figs. 19 and 21, Plates 80 and 81, is shown a fan of the second group in

position on the spindle; the same boss and spokes were used throughout the experiments.

From Table 1 (pages 458-9) it will be seen that variation in the cross section of the blades of the fan, while retaining the same shape and area and the same conditions of trial, produced a considerable effect on the results. The first group, fans 1 to 4, were all tried with their blades at an angle of  $17^{\circ}$  to the plane of rotation, and under identical conditions. The mechanical, volumetric, and pressure efficiencies of fan 1 were 21.2 and 38.2 and 0.7 per cent. respectively. The effect of putting the curved surface upon the back of the blades in fan 2 was to increase the efficiencies to 28.0 and 54.0 and 1.4 per cent. respectively. In fan 3 with the curved blades of uniform thickness the efficiencies were all greater than those of fan 1, being 23.9 and 53.0 and 1.4 per cent. respectively; the volumetric and pressure efficiencies were about the same as for fan 2, but the mechanical efficiency was 4.1 per cent. less. Fan 4 was the most efficient of the group, the section of its blades being concavo-convex with a hollow space between the faces; the thickness of the blades on the centre line was 7-16ths inch; and the three efficiencies were respectively 28.0 and 65.0 and 2.1 per cent. The efficiencies of these fans were thus increased by making the blades thicker in the middle of their breadth. These results are in accord with those obtained in the trials mentioned in the author's former paper in 1892, which were made in order to test the effect of the thickness of the blades of model screw-propellers 14 inches diameter, rotating in a fixed position, and in air instead of in water. As then stated (page 561) "the screws were of two kinds: in one the transverse section of the blades was similar to that of an ordinary propeller, only much thicker in proportion; in the other kind the thickness was reduced to that of a thin plate, the other dimensions being identical. Screws of two, three, and six blades were tried at progressive revolutions ranging from 800 to 1,800 per minute; and it was found that the screws with the thicker blades were more efficient than those with the thinner blades." In his note to the British Association in 1893 (page 884) the author dealt with fans differing from one another only in the cross section of their blades; and the results then obtained corroborate as far as they went the experiments now described. The

amount of curvature of the blades for the best effect appears to be governed somewhat by the width of the blade and the nature of the feed. With a very wide blade working in a tube, its efficiency may be reduced by thickening it, in consequence of the feed becoming thereby contracted. In the majority of cases the angles of maximum efficiency have been given in the tables. Other shapes were tried, and the general results were that the highest efficiency was obtained by making the blades as symmetrical and simple as possible. The experiments show that the backs or non-propelling faces of the blades exercise an important influence on the working of the fan, and that a curved back is more efficient than a flat one.

*Fans feeding from Tips of Blades.*—From all points behind the fan the air is sucked into the rarefied spaces left by the revolving blades; and is discharged in an axial direction in the shape of a cylindrical column of air having a spiral motion, the pitch of the spiral depending upon the angle at which the blades are set. Some of the fans were tried with the perimeter exposed, by moving the delivery tube  $4\frac{1}{2}$  inches forward, as shown dotted in Fig. 21, Plate 81, so as to increase the area of feed, the air being then drawn radially inwards from the tips of the blades, in addition to the axial feed through the spaces in the fan disc. The revolutions and air discharge of fan 9 have been plotted in Fig. 27, Plate 83, when the fan was tried in and out of the delivery tube. When out of the tube, the mechanical, volumetric, and pressure efficiencies were increased respectively from 16.9 up to 29.4, from 62.0 up to 78.0, and from 2.0 up to 3.1 per cent. In the fans arranged to feed from the tips, a much wider form of blade may be employed. It will be noticed that the general efficiency of the fans is increased by arranging them to feed from the tips in addition to the feed through the spaces in the fan disc. It is therefore important that, where possible, the fan should be fixed with its circumference exposed, which is not always done in practice.

*Velocity of Air on entering and on leaving fans.*—Some further experiments have recently been carried out by the author, in order to ascertain the velocity of the air at different points on entering

as well as on leaving the fan. They were made with two fans of 24 inches and 48 inches diameter. The 24-inch fan, shown in Figs. 33 and 34, Plate 86, is a three-bladed fan with the blades set at an angle of  $35^{\circ}$  to the plane of rotation; it is a kind which has been designed and employed by the author for the ordinary ventilation of buildings, factories, and ships, and for drying. It was tested at 600 revolutions per minute, and was driven by a belt from a shunt-wound electric motor. Anemometer readings were taken at a distance of 18 inches in front of the fan, as well as behind. In Fig. 35, Plate 87, is shown the velocity of air in feet per minute at the different points indicated. No delivery tube was employed, the fan being entirely open in front as well as behind. The air on the delivery side, at the distance of 18 inches at which the readings were taken, had no tendency to spread beyond a radius equal to the outside radius of the fan, the velocity being zero with the anemometer placed at a radius of 13 inches, that is, one inch greater radius than the radius of the fan. The maximum axial velocity on the delivery side at 18 inches distance in front of the fan without delivery tube was 1,230 feet per minute at a radius of about 5 inches; the minimum velocity was 175 feet per minute at 12 inches radius. The velocity of the entering air was measured at points on the elliptical curve shown in Fig. 35, the axis of the anemometer being normal to the curve in all positions. The inward or radial velocity at the tips of the blades, at right angles to the fan spindle, was 250 feet per minute. The axial velocity, parallel to the fan spindle, was 303 feet per minute at a radius of  $1\frac{3}{4}$  inches. The velocity of suction behind the fan depends on the distance of the anemometer from the fan disc. The stream of entering air converging towards the fan appeared to be rapidly accelerated in velocity as it approached the fan from all points behind.

The 48-inch fan, shown in Figs. 37 to 39, Plate 88, has four blades, set at an angle of  $32\frac{1}{2}^{\circ}$  to the plane of rotation; it is designed specially for tea-drying in Ceylon and India. In order to reduce the shipping charges, it is made as light as possible; the arms carrying the fan spindle are of mild steel, 1 inch diameter, screwed into cast-iron bosses and to a cast-iron ring. The blades are hollow, of plano-convex section, made of sheet steel 1-32nd inch

thick, brazed together, and riveted upon the lugs of a cast-iron boss. Though so light, the blades are so rigid as to be practically incapable of vibration; consequently the fans are nearly silent at all speeds. In Fig. 36, Plate 87, is shown the velocity of the air at different points on entering and on leaving the fan, when driven at 350 revolutions per minute. No delivery tube was employed. Anemometer readings were taken at a distance of 3 feet in front of the fan, and at the points shown on the elliptical curve behind. The inward or radial velocity at the tips of the blades was 900 feet per minute, while the axial entering velocity at a radius of 10 inches was only 540 feet per minute. The fan delivered 17,000 cubic feet of air per minute at 350 revolutions per minute, and about 34,000 cubic feet at 700 revolutions per minute.

*Fan combined with Electric Motor.*—In Figs. 40 and 41, Plate 89, is shown a fan combined with an electric motor specially designed for driving it. The field magnets are cylindrical, as shown in section in Fig. 42, and are made as small as possible in order to offer as little obstruction as possible to the passage of the air. In order to find the best position for the central electric motor, trials were made by fixing it both behind and in front of the fan, at different distances along the spindle. The position which offered the least obstruction to the air and gave the highest efficiency depended upon whether the fan was working with a contracted or a free orifice, and upon the amount of contraction; the best position was generally found to be at a little distance in front of the fan. There appears to be a central region immediately in front of the fan, where only a little stream of air is delivered, owing probably, in fans working with free discharge, to the centrifugal action on the front face of the blades, which is apparent near the centre.

*Experiments with Contracted Outlet and Inlet.*—These experiments were made with the three-bladed fan of  $23\frac{3}{4}$  inches diameter, Figs. 33 and 34, Plate 86, with blades set at an angle of  $35^\circ$  to the plane of rotation, driven at 800 revolutions per minute by belt from a shunt-wound motor. In Figs. 43 and 44, Plate 90, the fan is shown working partly in the 24-inch delivery tube. The outer end of the tube was



contracted by closing it successively with plates containing central holes of 6 and 12 and 18 inches diameter. The fan was tried both propelling and exhausting air, and its efficiency in either action was found to be much reduced by the contraction. One of the reasons that this kind of fan was unable to maintain static pressure in the air is probably the comparatively slow speed of the blades near the centre, in consequence of which the air tended to pass back again through the centre of the fan. The effect was therefore tried of fixing a circular disc in front of the fan on the delivery side, so as to prevent the air from returning through it; this had the effect of increasing the efficiency to a great extent when working against resistance, whereby a static pressure was obtained in the air delivered. Experiments were made with discs of different diameters, and it was found that the size of disc to give the best result depended upon the contraction of the delivery orifice, or upon the resistance offered to the passage of the air. Discs of 7, 10, 14, 17, and 22 inches diameter were tried for the three contractions of the orifice. The trials showed that, the more the air passage was baffled, the larger became the disc required for maximum efficiency; with free discharge the presence of the disc only reduced the efficiency. The results are given in Tables 4 and 5 (page 462). In Table 4 and Fig. 43 the fan was propelling air into the delivery tube, and the disc was placed in front of the fan in the delivery tube. In Table 5 and Fig. 44 the fan was exhausting air through the tube, and the disc was placed outside the tube but in front of the fan, which had been reversed upon its spindle. With the fan propelling air through the 6-inch orifice, Table 4, only 31 cubic feet per minute were discharged when no disc was used; and with discs of 10 and 14 and 17 inches diameter the discharge rose to 129 and 169 and 188 cubic feet respectively, the 17-inch disc giving maximum efficiency. If the volumetric efficiency of the fan without the disc be taken as 100, then with the 17-inch disc it becomes 606; while the mechanical efficiency is increased 223 times, from 100 up to 22,300. With the larger orifice of 12 inches diameter the effect was not so great: the 14-inch disc increased the volumetric efficiency 39 per cent., and the mechanical efficiency 169 per cent. With the still larger orifice of 18 inches diameter, no difference was noticed when the 7-inch disc

was added; and the larger discs reduce the efficiency. Somewhat similar effects, but of less magnitude, were obtained when exhausting air instead of propelling. With the 6-inch orifice in Table 5, the 14-inch and 17-inch discs increased the volumetric efficiency 17 and 19 per cent., and the mechanical 58 and 67 per cent. The fan was also tried with the disc on the back or inlet side, and also with the central part of the blades filled up to the same extent; but the result of either plan was that the efficiency was reduced. The disc was also tried at various distances in front of the fan, and the best position was found to be when touching the blades; it then allows the fan to feed near the centre at the back, and the air in entering the fan takes a radial direction due to centrifugal action, and then passes through over the disc. In Fig. 45 is shown the probable direction of the currents of air when the fan is propelling through a contracted orifice without a disc, showing that most of the air propelled by the tips takes a short circuit and enters the fan again at the centre, being thus only circulated in the fan. In view of the fact that this kind of fan is being employed in many cases with a contracted orifice, as in Fig. 44—as when drawing air through a material to be dried, like wool, or through tortuous flues, as in refrigerating apparatus—the adoption of the central disc becomes important, inasmuch as the efficiency of the fan can thereby be increased to such a great extent.

*Forced and Induced Draught.*—It is interesting to notice that with a contracted orifice the efficiency of the fan is higher in exhausting than in propelling. Referring to the experiments without circular discs in Tables 4 and 5 (page 462), while 31 cubic feet of air per minute were propelled through the 6-inch orifice at 800 revolutions per minute, the delivery was increased to 451 cubic feet when exhausting, the other conditions being identical; the volumetric efficiency was increased fifteen times. With the 12-inch orifice 497 cubic feet of air per minute were discharged by propulsion, and 1,374 when exhausting, the volumetric efficiency being increased nearly three times. With still less contraction the difference was not so great, the volumetric efficiency with the 18-inch orifice being increased only

from 58½ per cent. when propelling to 67 per cent. when exhausting. It thus appears that the freedom of the discharge or outlet side of the fan is of greater importance than the freedom of the suction or inlet side. As far as they go, and under the conditions under which they were made, the experiments seem to present an argument in favour of induced draught over forced.

*Negative Slip, effect of thickness of blades.*—It is interesting to notice the great effect of the thickness of the blades, caused by fixing the curved surface to the backs of the blades. Comparing fans 1 and 2 in Table 1 (page 458), the air discharge was increased from 2,128 cubic feet per minute in fan 1 to 2,888 cubic feet in fan 2 at 30 revolutions less per minute, merely by making the blades thicker. This is a substantial reason for the “apparent negative slip,” which is sometimes noticed in a screw-propeller, when the pitch is calculated from the front face, without taking into account the round back whereby the virtual pitch is increased; and negative slip has generally been noticed in those propellers that have thick blades and round backs.

Without entering into any theoretical views as to the action of the blades, which is a complicated question affected by many conditions, it may be said that, having regard to the stream-line principle, the section of the blades should be as ship-shape as possible, and that rotary motion of the air should be as small as possible. The two losses in an air propeller are rotary motion imparted to the air, and skin friction of the blades; and the problem is to make the sum of these two losses as small as possible, and at the same time the delivery of air as large as possible. The loss due to the skin friction of the blades is comparatively small, as was ascertained by rotating the flat thin blades when their plane was set to coincide with the plane of rotation.

NOTE.—For convenience in making the calculations the fans have been taken at 24 inches diameter, the same as the bore of the tube. This slight difference from the exact dimensions affects the results to an extremely small extent only.

TABLE 1 (continued on opposite page).

*Experiments on Propeller Ventilating Fans Nos. 1 to 10, Plate 79, of 23 $\frac{3}{4}$  inches diameter,  
all revolving inside delivery air tube of 24 inches bore, except those marked \* revolving 4 $\frac{1}{2}$  inches out of tube.  
Blades set at angle to plane of revolution. Anemometer placed at radius of 1 $\frac{7}{8}$ —5 $\frac{1}{8}$ —7 $\frac{7}{8}$ —10 $\frac{5}{8}$  inches from axis of tube.*

Number of Fan.	Angle of Blades.	Revolutions per minute.	Motor.		Velocity of Air per minute at radius				Air discharged per minute.	Weight on Brake Pulley.	Horse-Power.		Efficiency.		
			Volts.	Am- pères.	1 $\frac{7}{8}$	5 $\frac{1}{8}$	7 $\frac{7}{8}$	10 $\frac{5}{8}$			Brake.	Air.	Mech- anical.	Volu- metric.	Pressure.
No.		Revs.	V.	A.	Feet.	Feet.	Feet.	Feet.	Cubic Ft.	Ounces.	H.P.	H.P.	P. c.	P. c.	P. c.
1	17°	890	69.7	1.50	762	767	688	607	2128	11.0	0.0458	0.0097	21.2	38.2	0.7
2	17°	860	81.5	1.92	880	917	920	930	2888	21.5	0.0866	0.0241	28.0	54.0	1.4
3	17°	645	60.0	1.58	690	730	690	675	2172	14.2	0.0430	0.0103	23.9	53.0	1.4
4	17°	525	54.5	1.58	617	713	730	645	2139	14.2	0.0350	0.0098	28.0	65.0	2.1
5	27°	610	80.0	2.33	723	804	913	951	2790	29.0	0.0829	0.0218	26.3	73.0	2.7
6	27°	490	78.3	2.53	423	690	880	903	2536	33.5	0.0770	0.0164	21.3	82.0	3.4
7	17°	758	80.8	2.07	703	785	805	760	2427	18.5	0.0650	0.0144	22.2	51.0	1.3
8	17°	650	78	2.28	645	780	805	700	2330	22.2	0.0677	0.0127	18.8	57.0	1.6
	27°	*600*	85	2.45	910	1115	1145	1050	*3385*	31.7	0.0894	0.0390	43.0	89.8	4.1
	27°	595	77	2.20	670	780	875	990	2773	26.6	0.0742	0.0214	29.9	74.0	2.8
	40°	*495*	100	2.70	735	970	1147	1070	*3265*	39.4	0.0916	0.0350	38.2	105.0	5.6
9	17°	600	69	1.91	320	365	530	635	1633	20.6	0.0590	0.0043	7.4	43.3	0.9
	27°	600	79	2.23	600	650	755	825	2348	27.3	0.0768	0.0130	16.9	62.0	2.0
	27°	*600*	86	2.47	740	890	1030	960	*2965*	32.3	0.0892	0.0262	29.4	78.0	3.1
	40°	570	88	2.75	745	810	910	960	2870	37.7	0.1075	0.0237	22.0	80.0	3.2
10	27°	*600*	81	2.31	805	930	950	930	*2891*	28.7	0.0809	0.0243	30.0	76.7	3.0
	17°	*600*	68	1.85	260	405	555	645	*1675*	19.7	0.0550	0.0047	8.5	44.4	1.0

TABLE 1 (concluded from opposite page).

*Experiments on Propeller Ventilating Fans Nos. 11 to 17, Plate 79, of  $23\frac{3}{4}$  inches diameter, all revolving inside delivery air tube of 24 inches bore, except those marked \* revolving  $4\frac{1}{2}$  inches out of tube. Blades set at angle to plane of revolution. Anemometer placed at radius of  $1\frac{7}{8}$ — $5\frac{1}{8}$ — $7\frac{7}{8}$ — $10\frac{5}{8}$  inches from axis of tube.*

Number of Fan.	Angle of Blades.	Revolutions per minute.	Motor.		Velocity of Air per minute at radius				Air discharged per minute.	Weight on Brake Pulley.	Horse-Power.		Efficiency.		
			Volts.	An- pers.	$1\frac{7}{8}$	$5\frac{1}{8}$	$7\frac{7}{8}$	$10\frac{5}{8}$			Brake.	Air.	Mechanical.	Volu- metric.	Pressure.
No.		Revs.	V.	A.	Feet.	Feet.	Feet.	Feet.	Cubic Ft.	Ounces.	H.P.	H.P.	P. c.	P. c.	P. c.
11	17°	850	74	1.70	640	745	850	990	2720	15.5	0.0620	0.0202	32.6	50.9	1.3
	27°	610	63	1.66	645	725	860	930	2641	16.0	0.0458	0.0185	40.4	68.9	2.4
	27°	*600*	64	1.75	765	895	870	850	*2699*	17.6	0.0489	0.0190	40.4	71.6	2.6
12	27°	600	63	1.70	590	635	740	745	2222	16.8	0.0473	0.0110	23.3	58.9	1.7
13	27°	600	74	2.06	655	745	860	995	2737	23.8	0.0671	0.0200	30.0	72.6	2.6
	17°	600	61	1.63	430	525	640	680	1926	15.3	0.0431	0.0070	16.7	51.0	1.3
14	17°	604	53	1.40	415	540	605	640	1846	10.7	0.0300	0.0063	21.0	48.6	1.2
	40°	605	85	2.40	780	940	1110	1105	3272	31.0	0.0874	0.0348	39.8	86.0	3.7
15	40°	600	84	2.40	775	965	1120	1085	3270	30.6	0.0862	0.0350	40.9	86.7	3.8
16	35°	600	65	1.77	720	780	860	830	2580	18.1	0.0510	0.0172	33.7	68.4	2.4
17	35°	600	67	1.87	685	830	920	875	2705	19.1	0.0538	0.0199	37.0	71.7	2.6

TABLE 2.

*Experiments on Propeller Ventilating Fan No. 17 with Blades set at different Angles to plane of revolution. Fan 23 $\frac{1}{4}$  inches diameter, revolving inside delivery air tube of 24 inches bore; speed 600 revolutions per minute. Anemometer placed at radius of 1 $\frac{1}{8}$ —5 $\frac{1}{8}$ —7 $\frac{1}{8}$ —10 $\frac{5}{8}$  inches from axis of tube. See Plates 79 and 81.*

Angle of Blades.	Motor.		Velocity of Air per minute at radius				Air discharged per minute.	Weight on Brake Pulley.	Horse-Power.		Efficiency.		
	Volts.	Am-pere.	1 $\frac{1}{8}$	5 $\frac{1}{8}$	7 $\frac{1}{8}$	10 $\frac{5}{8}$			Brake.	Air.	Mechanical.	Volu-metric.	Pressure.
	V.	A.	Feet.	Feet.	Feet.	Feet.	Cubic Ft.	Ounces.	H.P.	H.P.	P. c.	P. c.	P. c.
15°	40	1·05	460	470	510	510	1562	4·08	0·0115	0·0038	33·0	41·4	0·87
20°	45	1·22	570	590	610	630	1916	6·41	0·0181	0·0070	38·5	50·8	1·31
25°	52	1·36	630	680	730	770	2291	10·00	0·0282	0·0120	42·2	60·8	1·88
27°	55	1·43	640	710	790	820	2423	11·84	0·0334	0·0143	42·8	64·3	2·10
30°	60	1·66	665	770	875	860	2598	15·00	0·0423	0·0176	41·6	68·9	2·42
35°	67	1·87	685	830	920	875	2705	19·07	0·0538	0·0199	37·0	71·7	2·62
40°	76	2·12	705	855	970	900	2805	25·00	0·0707	0·0222	31·4	74·4	2·82
45°	84	2·40	780	910	960	880	2827	30·60	0·0858	0·0227	26·4	76·7	2·87
50°	91	2·59	790	880	930	870	2771	36·31	0·1024	0·0214	20·9	75·2	2·75
60°	86	2·48	210	290	450	680	1534	32·30	0·0911	0·0035	3·8	40·7	0·84

TABLE 3.

*Comparison of Propeller Ventilating Fans Nos. 16 and 17 with Blades set at different Angles to plane of revolution. Both Fans 23 $\frac{3}{4}$  inches diameter, revolving inside delivery air tube of 24 inches bore ;  
| speed 600 revolutions per minute. Friction of motor included. See Plate 79.*

Angle of Blades.	FAN No. 16.						FAN No. 17.					
	Air discharged per minute.	Horse-Power.		Efficiency.			Air discharged per minute.	Horse-power.		Efficiency.		
		Gross.*	Air.	Mechanical. †	Volu- metric.	Pressure.		Gross.*	Air.	Mechanical. †	Volu- metric.	Pressure.
	Cubic Ft.	H.P.	H.P.	P. c.	P. c.	P. c.	Cubic Ft.	H.P.	H.P.	P. c.	P. c.	P. c.
15°	1079	0·0377	0·0012	3·1	28·6	0·41	1562	0·0453	0·0038	8·3	41·4	0·87
20°	1622	0·0429	0·0042	9·7	43·0	0·94	1916	0·0519	0·0070	13·4	50·8	1·31
25°	2030	0·0521	0·0084	16·1	53·8	1·48	2291	0·0620	0·0120	19·3	60·8	1·88
27°	2185	0·0588	0·0106	18·0	58·0	1·73	2423	0·0672	0·0143	21·2	64·3	2·10
30°	2439	0·0698	0·0145	20·7	64·7	2·13	2598	0·0761	0·0176	23·1	68·9	2·42
35°	2580	0·0848	0·0172	20·2	68·4	2·39	2705	0·0876	0·0199	22·7	71·7	2·62
40°	2627	0·0958	0·0182	18·9	69·7	2·47	2805	0·1045	0·0222	21·2	74·4	2·82

\* Gross Horse-Power = brake horse-power + horse-power due to friction of motor.

† Mechanical Efficiency = ratio of horse-power in air discharged to gross horse-power.

*Experiments on Ventilating Fan of 23 $\frac{3}{4}$  inches diameter, revolving partly inside air tube of 24 inches bore.*

*Circular Disc fixed directly in front of fan blades, parallel to plane of rotation,  
except in trials marked "none."*

*Outer orifice of tube contracted to 6-12-18 inches diameter. Speed 800 revolutions per minute.*

TABLE 4.—*Fan Propelling air. Fig. 43, Plate 90.*

Diameter of Outlet.	Diameter of Disc.	Mean Velocity of Air per min.	Air discharged per minute.	Volumetric Efficiency. P. c.	Relative Efficiency.	
					Volumetric.	Mechanical.
Ins.	Ins.	Feet.	Cub. Ft.	P. c.		
6	none	160	31	0.6	100	100
6	10	660	129	2.6	416	7,190
6	14	860	169	3.4	545	16,200
6	17	960	188	3.7	606	22,300
12	none	632	497	9.8	100	100
12	7	752	591	11.8	119	168
12	10	780	612	12.2	123	187
12	14	880	691	13.7	139	269
12	17	760	597	11.9	120	173
18	none	1660	2938	58.5	100	100
18	7	1660	2938	58.5	100	100
18	10	1540	2726	54.3	93	79
18	14	1140	2018	40.2	68	32

TABLE 5.—*Fan Exhausting air. Fig. 44, Plate 90.*

Diameter of Inlet.	Diameter of Disc.	Mean Velocity of Air per min.	Air exhausted per minute.	Volumetric Efficiency. P. c.	Relative Efficiency.	
					Volumetric.	Mechanical.
Ins.	Ins.	Feet.	Cub. Ft.	P. c.		
6	none	2300	451	9.0	100	100
6	10	2500	490	9.7	109	128
6	14	2680	525	10.4	117	158
6	17	2730	535	10.6	119	167
6	22	2640	517	10.3	115	151
12	none	1750	1374	27.3	100	100
12	10	2000	1570	31.2	114	149
12	14	2040	1601	31.9	117	158
12	17	2040	1601	31.9	117	158
18	none	1900	3363	67.0	100	100
18	7	1875	3319	66.0	99	96
18	10	1850	3275	65.0	97	92
18	14	1830	3240	64.0	96	89