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THE FAN

INCLUDING THE

THEORY AND PRACTICE OF CENTRIFUGAL AND AXIAL FANS.

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THE THEORY AND PRACTICE OF CENTRIFUGAL AND AXIAL FANS

BY

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"CENTRIFUGAL PUMPS, TURBINES, AND WATER MOTORS," AND
"PROBLEMS IN MACHINE DESIGN"

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AUTHOR'S PREFACE.

In the following pages I give a theory of the fan which differs considerably from anything that I have seen in print, and which may therefore meet with some criticism. In all works in this or any other language that I have read I have found the equation

$$H = \frac{k v_2^2}{1 + \frac{o^2}{a^2}},$$

where H is the head of air against which the fan works, v_2 is the tip speed, a is a constant for any particular fan, and o is the equivalent orifice.

That this equation cannot be that of all fans is obvious when we remember that in many fans the manometric efficiency increases at first as the orifice increases from zero, while this equation states that it decreases; nor can it be maintained that the equation applies to fans whose manometric efficiency is greatest at zero orifice, for if the curve of manometric efficiency be drawn with this as ordinate and orifices as abscissæ, it will be found that the tangent at the point where the curve cuts the vertical axis is horizontal; and that this is not the case may be seen from fig. 49, which gives curves for eleven fans differing widely in construction. Having already studied the centrifugal pump, it occurred to me about ten years ago that its theory might be applied to that of the fan; and except that the fan does not actually lift air, as the pump lifts water, but acts like a centrifugal pump that pumps against the resistance of horizontal piping only, I consider that the same theory may be applied to both.

In Chapter VIII. I have endeavoured to show that my theory agrees with the results of experiment, as far as these may be trusted. The following pages commence with the theory of the centrifugal fan, following which are experiments with and descriptions of this type; and in Chapter X. is given a description of Prof. Rateau's high-pressure fans, in whose design it may be mentioned the variation of the density of the air must be taken into account. In Chapter XII. will be found an imperfect theory of propeller fans, imperfect because I cannot find all the information I require from published experiments. Following this will be found descriptions of this type and of Prof. Rateau's screw fans, the theory of which closes the book.

I hope this book will be of service to those who have to design, or who wish to understand the working of fans. There are many very inferior fans used which do their work wastefully, and it would be satisfactory to see these replaced by others of scientific design.

CHAS. H. INNES.

RUTHERFORD COLLEGE,
NEWCASTLE-ON-TYNE,
October, 1904.

NOTE.

IN this edition the work has been thoroughly revised, and various alterations and expansions of the text have been made. The calculations have been checked, and a revised and uniform system of notation has been adopted.

Almost all the figures have been redrawn. Chapter XIII., dealing with recent practice mainly in the construction of fans and centrifugal compressors, is quite new.

It is hoped that in its altered form the work may be increasingly useful.

W. M. WALLACE,
F. RABY JOLLEY.

September, 1916.

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CENTRIFUGAL FANS

CHAPTER I.

The Conservation of Energy.—Energy is indestructible, although it may appear in a number of forms, some of which are useful to man, whilst others are not. Thus steam at a high pressure contains energy in the form of heat, part of which can be converted into useful work, whilst the remainder is wasted in overcoming friction, or is rejected with the condensed steam at a low temperature, and is of no service to man. But the law of conservation of energy supplies us with equations which are of the utmost service in correctly designing machines in which a flow of fluids takes place, because we know that changes of pressure, volume, and velocity, are accompanied by alterations of the forms in which the initial energy existed, but that the quantity of energy is unaltered. Although we are now dealing with air, a compressible gas, and should therefore, in strict accuracy, take into account the alteration of volume that accompanies change of pressure, yet in all except the high-pressure type of fans, since this change of pressure is so small, it need not be considered, especially if we denote the volume of air passing through the fan as the volume occupied at a pressure which is the arithmetic mean between that at suction and discharge. Let P lb. per square foot be the difference between the suction and discharge pressures, and Q the mean volume in cubic feet per second of the air passing through the fan measured as above; then P Q foot lb. is the useful work done by the fan per second.

Application of the Law of Conservation of Energy to an Incompressible Fluid in Motion.—Suppose a liquid of density δ lb. per cubic foot is contained in a vessel (fig. 1), whose free horizontal surface is maintained at a constant height H above any assumed level, while the liquid is being discharged from a pipe connected to the vessel. Consider a point b in the surface of the liquid, and a point a in the pipe h feet above the given level. Let v feet per sec. be the velocity of a particle at a, where the pressure is p lb. per square foot, and let A be the sectional area of the pipe in square feet.

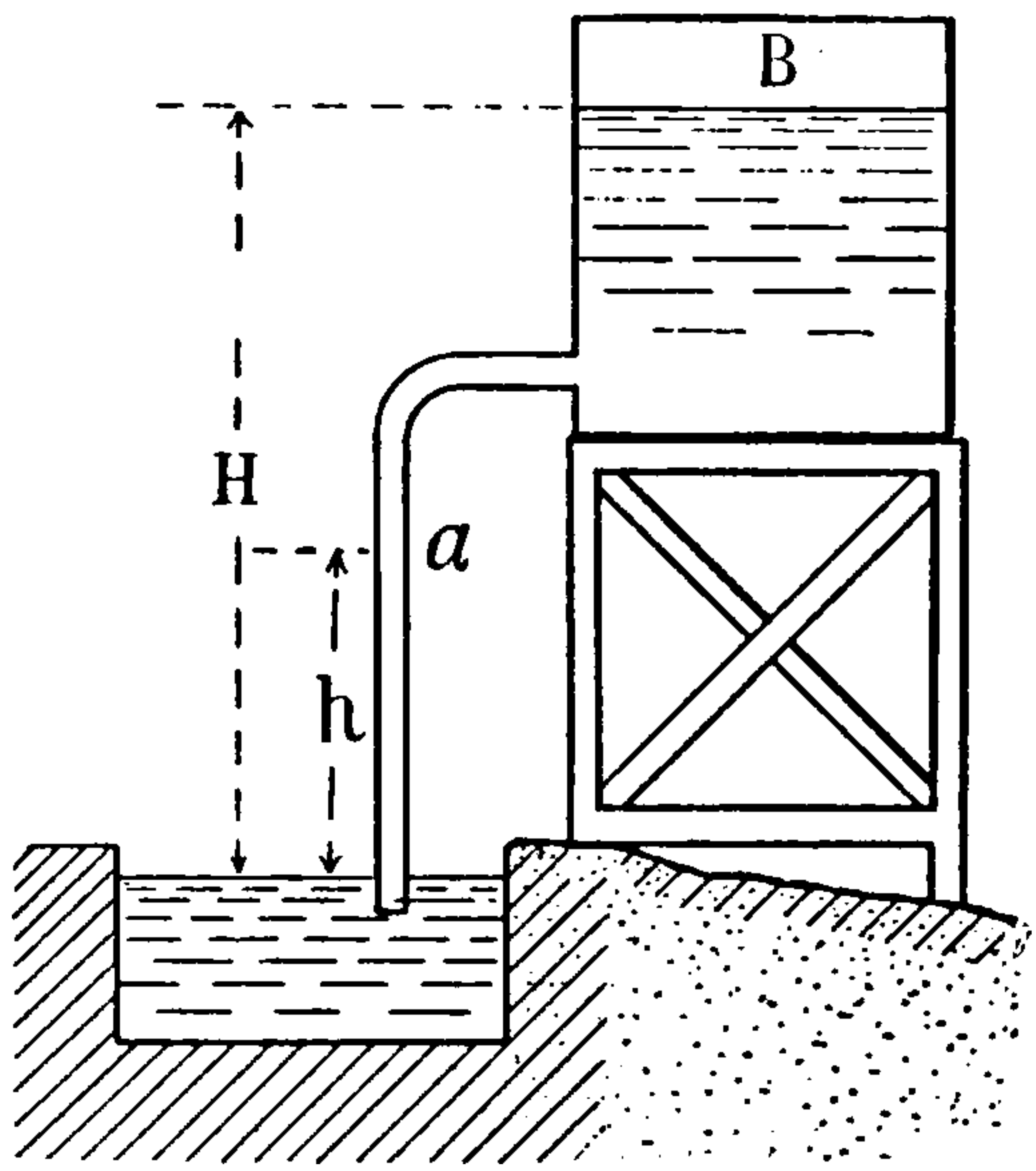


FIG. 1.

Then, assuming a piston to move with the water at a, a displacement of 1 foot would involve pA foot-pounds of work being done upon the piston, at the expense of A cubic feet of liquid, or p foot-pounds at the expense of each cubic foot; or, as 1 cubic foot contains δ lb., $p \div \delta$ foot-pounds of work are done at the expense of each pound of liquid. Taking atmospheric pressure as zero, the energy of the water at b is completely potential in nature, due to a level head H feet; whilst it follows that the energy at a exists in three different forms—viz., potential energy h, kinetic energy $v^2 \div 2g$, and energy due to pressure, or pressure energy, $p \div \delta$. Hence we say that a liquid under pressure p, velocity v, and at a height h, has a pressure head $p \div \delta$, velocity head $v^2 \div 2g$, and actual or level head h, while the equivalent head due to all three is H; or, more simply, we may state that—

$$H = h + \frac{p}{\delta} + \frac{v^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Reason for treating Air as if it had a Constant Volume.—In practice the highest water gauge against which a fan works is about 12 inches. As the water barometer is about 34 feet, this amount corresponds to a compression of $\frac{1}{34}$th of the original volume of the air. Thus, if the discharge of a fan is measured by the mean volume, the actual volume at any instant can only be $\frac{1}{68}$th part of the mean volume greater or less than this. Hence for practical purposes the air may be treated as an incompressible fluid obeying equation (1). In exact experiments on fans, the hygroscopic state of the air must be considered, as well as its temperature and pressure.

Application of Formula (1).—Suppose, for example, that air weighs 0·075 lb. per cubic foot, and moves with a velocity of 40 feet per second in a straight pipe which gradually enlarges in section so that the velocity is reduced to 20 feet per second, what will the change of pressure per square foot be if the pipe is horizontal? Let p_1, v_1 be the pressure and velocity when the latter is 40 feet per second, and p_2, v_2 similar quantities when it is 20. Then, putting σ for the density of air, we have—

$$H = h_1 + \frac{p_1}{\sigma} + \frac{v_1^2}{2g} = h_2 + \frac{p_2}{\sigma} + \frac{v_2^2}{2g},$$

and
$$h_1 = h_2;$$

then
$$\frac{p_2 - p_1}{\sigma} = \frac{v_1^2 - v_2^2}{2g}$$

—*i.e.*,
$$p_2 - p_1 = \frac{0{\cdot}075}{64}(1600 - 400)$$
$$= 1\tfrac{13}{32} \text{ lb. per sq. ft.}$$

This corresponds to a change of head $= 18\tfrac{3}{4}$ feet of air.

It will be seen later, by the application of the principle here involved, that the mechanical efficiency of a fan can be considerably increased by using various means for gradually reducing the velocity of the air, and thus increasing its pressure.

The Loss of Head by Surface Friction of a Fluid under Steady Flow in a Pipe or Passage of Uniform Section.—The following analysis, due to Dr. Nicolson, is offered as a solution of the problem :

The irregularities or obstacles on the surface of a pipe create an eddying motion in the fluid in the nature of vortices. These increase in size as they pass to the centre of the stream, where their velocity is made up out of that of the remaining uneddied fluid. Consider the flow in a

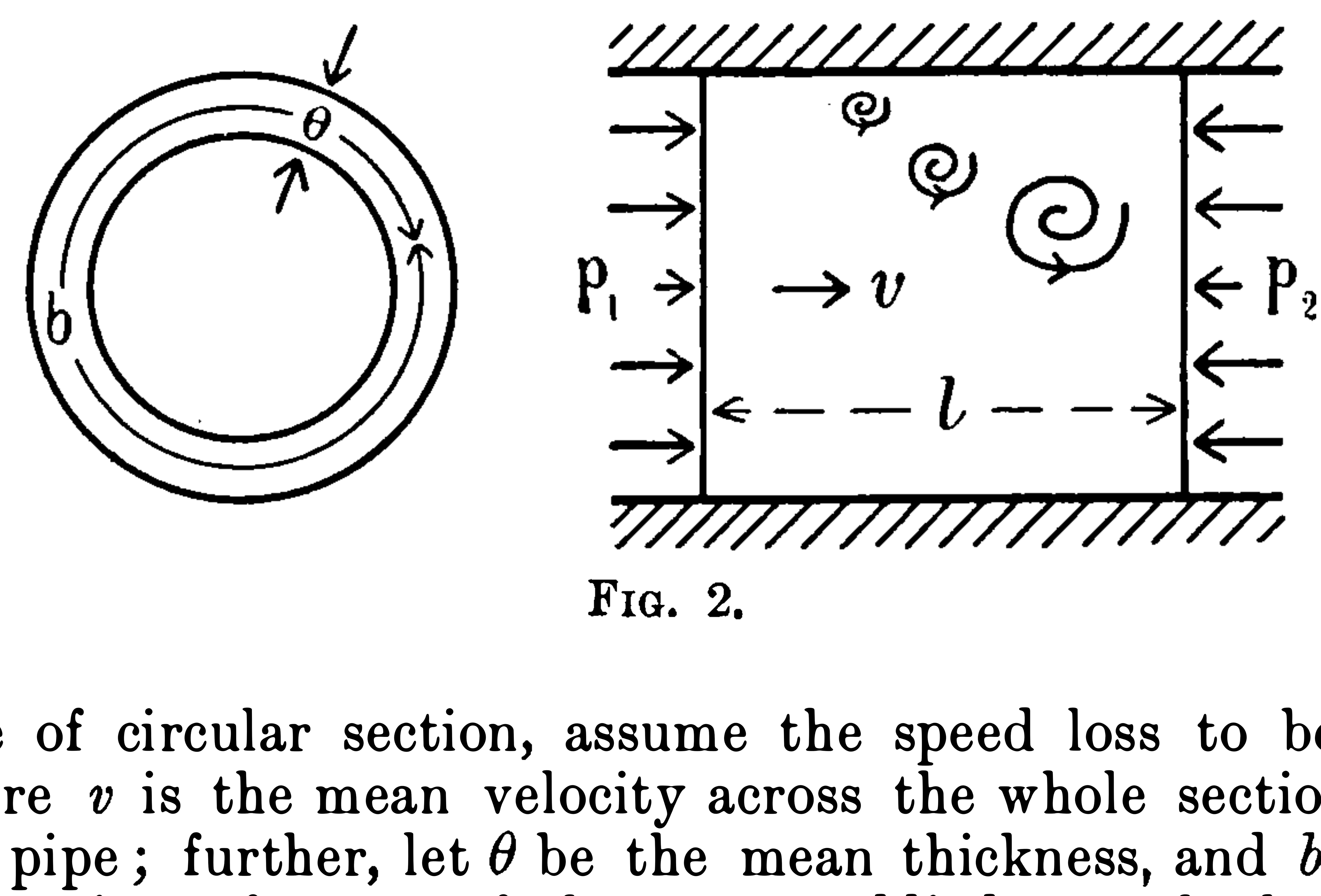


FIG. 2.

pipe of circular section, assume the speed loss to be cv, where v is the mean velocity across the whole section of the pipe; further, let θ be the mean thickness, and b the mean circumference, of the vortex eddied at each obstacle on the surface of the pipe, as represented in fig. 2.

Then the volume of fluid eddied per second at each obstacle $= b\theta v$, and the energy lost by eddy formation per obstacle per second $= \frac{\sigma b\theta v}{2g}(cv)^2$, where σ is the density of the fluid.

If there are n eddies per foot length of pipe, then the kinetic energy lost per foot length $= \frac{\sigma b\theta vn}{2g}(cv)^2$.

The work done per second on a block of fluid l feet long $= (p_1 - p_2)av$, where p_1 and p_2 are the pressures at points l feet apart and a is the sectional area of the pipe.

Now, it follows that the work done per foot length per second is equal to the kinetic energy lost per foot length, since the forward motion is constant and the lost energy is made good out of the pressure head, thus—

$$\frac{p_1 - p_2}{l} av = \frac{\sigma . nb\theta v}{2\,g}(cv)^2,$$

so that loss of head

$$h_1 = \frac{p_1 - p_2}{\sigma} = n\theta c^2 . \frac{bl}{a} . \frac{v^2}{2\,g} = \zeta \frac{l}{m}\frac{v^2}{2\,g}. \quad . \quad . \quad (2)$$

where m is the mean hydraulic depth of the pipe—*i.e.*, section divided by circumference. Thus it follows that ζ is proportional to the number of obstacles per foot length, to the square of the fractional speed loss, and to the thickness of water eddied which depends on the height of the obstacles; it is also evident that ζ in no way depends on the nature of the fluid.

Professor Unwin in "Development and Transmission of Power" gives—

$$\zeta = 0{\cdot}0027\left(1 + \frac{3}{10\,d}\right).$$

Examples from Weisbach show—

$$\zeta = 0{\cdot}005\left(1 + \frac{1}{12\,d}\right) \text{ for clean pipes,}$$

$$\zeta = 0{\cdot}01\left(1 + \frac{1}{12\,d}\right) \text{ for encrusted pipes,}$$

where d is the diameter of the pipe in feet.

For a pipe of circular section, $m = d \div 4$, so that equation (2) becomes

$$h_1 = \zeta\frac{4\,l}{d} . \frac{v^2}{2\,g} . = f . \frac{v^2}{2\,g}.$$

f is the coefficient of resistance of the pipe referred to the velocity v. And as the losses of head due to bends and elbows, and passages of any form, are proportional to the square of the velocity, in a given machine we can say that the loss of head is

$$F_1 \frac{v^2}{2g}$$

in addition to that part of the loss of head due to sudden changes of direction and velocity, where F_1 is the coefficient of resistance of the machine referred to the velocity v.

Dr. A. H. Gibson's Formula.—In the case of air Dr. Gibson has shown[1] that formula (2) only applies if the coefficient ζ is varied, not only with the physical condition of the interior surface of the pipe, but also with its diameter, with the mean velocity of flow, with the mean pressure, and with the temperature of the air. For either cast-iron or wrought-iron pipes laid under normal conditions as to jointing, etc., he gives the following relationship for the drop in pressure in lb. per sq. in. (δp) in a length l ft. of pipe of diameter d ft.—

$$\delta p = 0{\cdot}00000125 \frac{p_m^{n-1} v_m^n l}{6{\cdot}6^n d^{3-n}},$$

p_m being the mean pressure in lb. per sq. in., and v_m the mean velocity of the air in ft. per sec. The above relationship holds for a mean temperature of 66° F. The value of n depends upon the diameter of the pipe as follows:—

Dia. (in.):	3.	5.	7.	9.	12.
n	1·83	1·81	1·79	1·775	1·77

For a 3 in. pipe the formula becomes—

$$\delta p = 0{\cdot}0000002\, p_m^{0{\cdot}83} v_m^{1{\cdot}83} l,$$

and the drop in pressure in lb. per sq. in. on 100 ft. length of 3 in. pipe, when the mean pressure is atmospheric, say **14·7** lb. per sq. in., is for the following values of v (in ft. per sec.)—

v ...	10	20	30	40	60
δp ...	0·01258	0·04474	0·09393	0·1591	0·3345

For a mean pressure of 40 lb. per sq. in. the drop in pressure at these velocities becomes—

v ...	10	20	30	40	60
δp ...	0·02889	0·1027	0·2156	0·3651	0·7678

It is evident that the loss of head or pressure drop is affected very much by the mean pressure of the air.

The values of n in the formula were deduced from the experiments of Riedler and Gutermuth on the Paris air

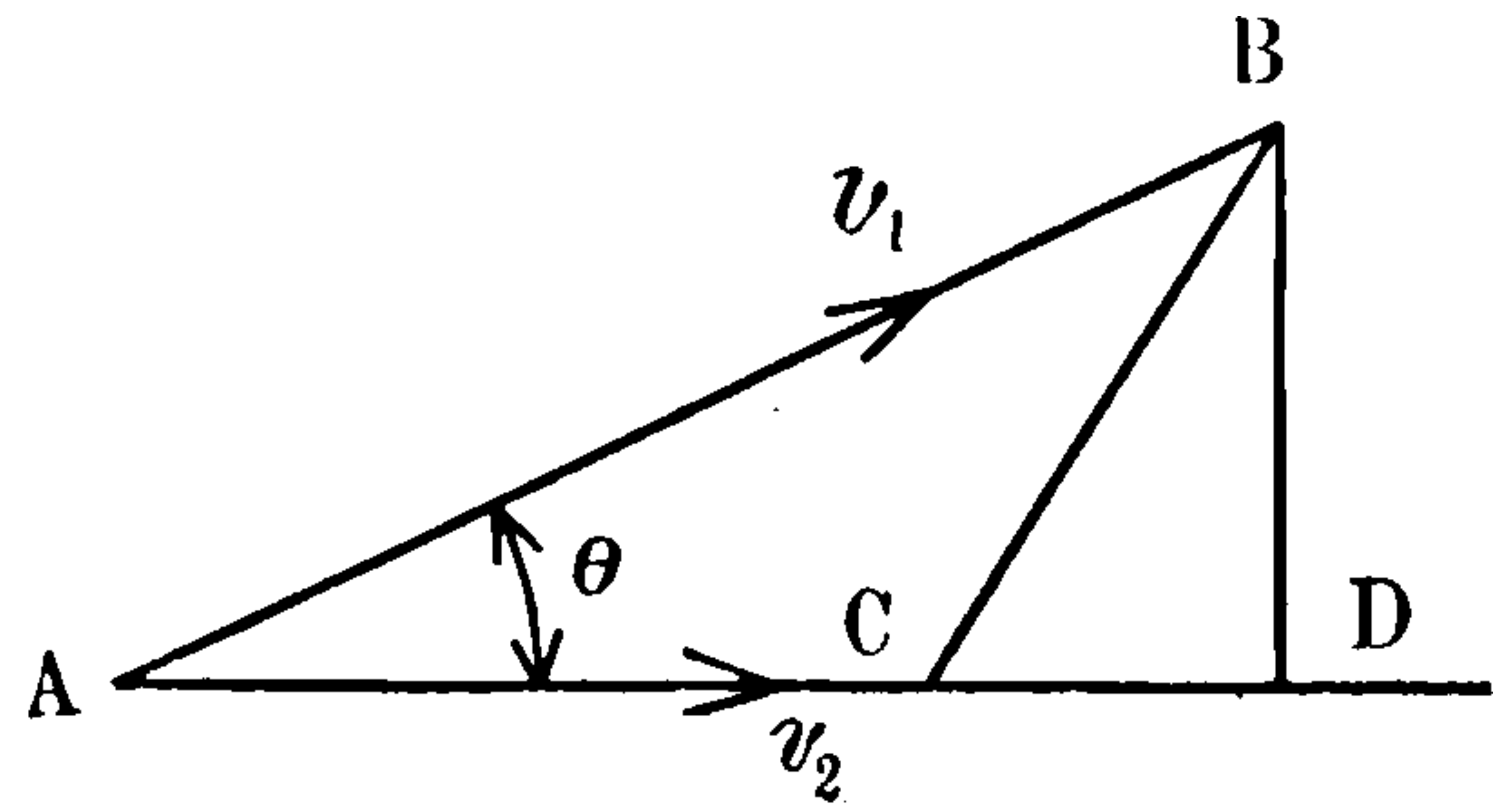


FIG. 3.

mains, the length of pipes ranging from 10,980 ft. to 54,270 ft., the mean pressures from 92 to 118 lb. per sq. in., and mean velocity from 8·5 to 28·6 ft. per sec. Unfortunately, the mean temperature which would affect the drop somewhat was unknown, but assumed at 65° F. Experimental results on smaller and shorter pipes by M. Stockalper, Dr. Brix, and the author, were also consulted, though in some of these the mean temperature had to be assumed.

Condition for Minimum Loss of Head due to a Sudden Change in the Direction of Motion.—When a sudden change of direction of motion and of velocity takes place, such as that represented in fig. 3 from A B to A C, where A B represents a velocity v_1, A C a velocity v_2, and the angle B A C is called θ, then the loss of head

$$h_2 = \frac{\text{B C}^2}{2g}$$

$$= \frac{v_1^2 + v_2^2 - 2 v_1 v_2 \cos \theta}{2g} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This expression obviously has its least value when B C coincides with B D, the perpendicular on A C, so that

$$v_2 = v_1 \cos \theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Thus the minimum loss of head

$$h_2 = \frac{v_1^2 - v_1^2 \cos^2 \theta}{2g}$$

$$= \frac{v_1^2 \sin^2 \theta}{2g}.$$

Condition for Maximum Gain of Pressure Head.—Let p_1 be the pressure before the change, and p_2 after, then

$$\frac{p_2}{\sigma} + \frac{v_2^2}{2g} + \text{loss of head} = \frac{p_1}{\sigma} + \frac{v_1^2}{2g},$$

$$\frac{p_2 - p_1}{\sigma} = \text{gain of pressure head}$$

$$= \frac{v_1^2 - v_2^2}{2g} - \frac{v_1^2 + v_2^2 - 2 v_1 v_2 \cos \theta}{2g}$$

$$= \frac{v_1 v_2 \cos \theta - v_2^2}{g} = \frac{\text{A C C D}}{g} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The maximum gain of pressure head occurs when

$$\frac{d}{dv_2}(v_1 v_2 \cos \theta - v_2^2) = 0\,;$$

i.e., when $$v_2 = \tfrac{1}{2} v_1 \cos \theta,$$

or when $$\text{A C} = \text{C D}.$$

CHAPTER II.

Manometer and Pitot Tube.—To measure the work done by a fan, we require to obtain the pressure difference between suction and discharge and the quantity of air passing through the fan in unit time. An instrument termed a manometer is used to obtain difference of pressure. If a bent tube (fig. 4) contains water, and the end *c* is exposed to greater pressure than the end *b*, the

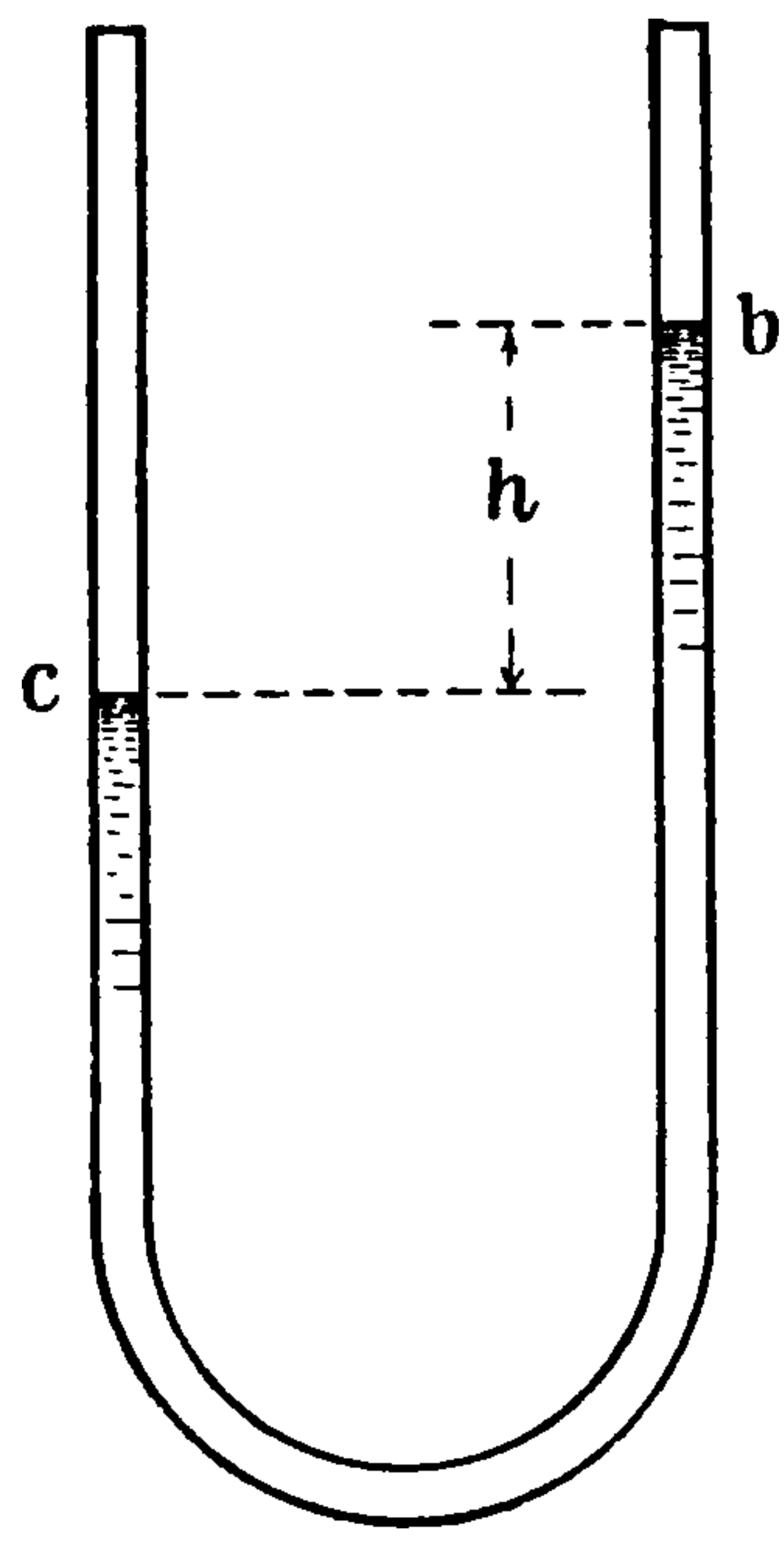


FIG. 4.

liquid will rise on the latter side to a height proportional to the difference of pressure. At the average temperature a cu. ft. of water weighs 62·3 lb., so that each inch of gauge registers a pressure of 5·192 lb. per sq. ft.

A distinction must be made between the dynamic and the static pressure of a gas. When the mouth of the tube is placed parallel to the direction of flow of the air, the static pressure is measured at that point; but if the tube is bent so that its mouth faces upstream, and is at right angles to the lines of flow, there is, in addition to the static pressure, a dynamic pressure due to the bombardment of the mouth by the gas molecules.[1] This dynamic pressure is taken advantage of in order to measure the velocity of air in accurate fan work. The amount by which the pressure is raised is

$$\frac{v^2}{2g} \sigma \text{ lb. per sq. ft.,}$$

so that if h is the difference of level of the water manometer (fig. 4) when the end c is exposed to the dynamic and b to the static pressure,

$$h = \frac{12 v^2}{2g} \cdot \frac{\sigma}{\delta} \text{ in.}$$

In order to measure the dynamic pressure or head accurately in practice, Pitot's tube is used, one form of

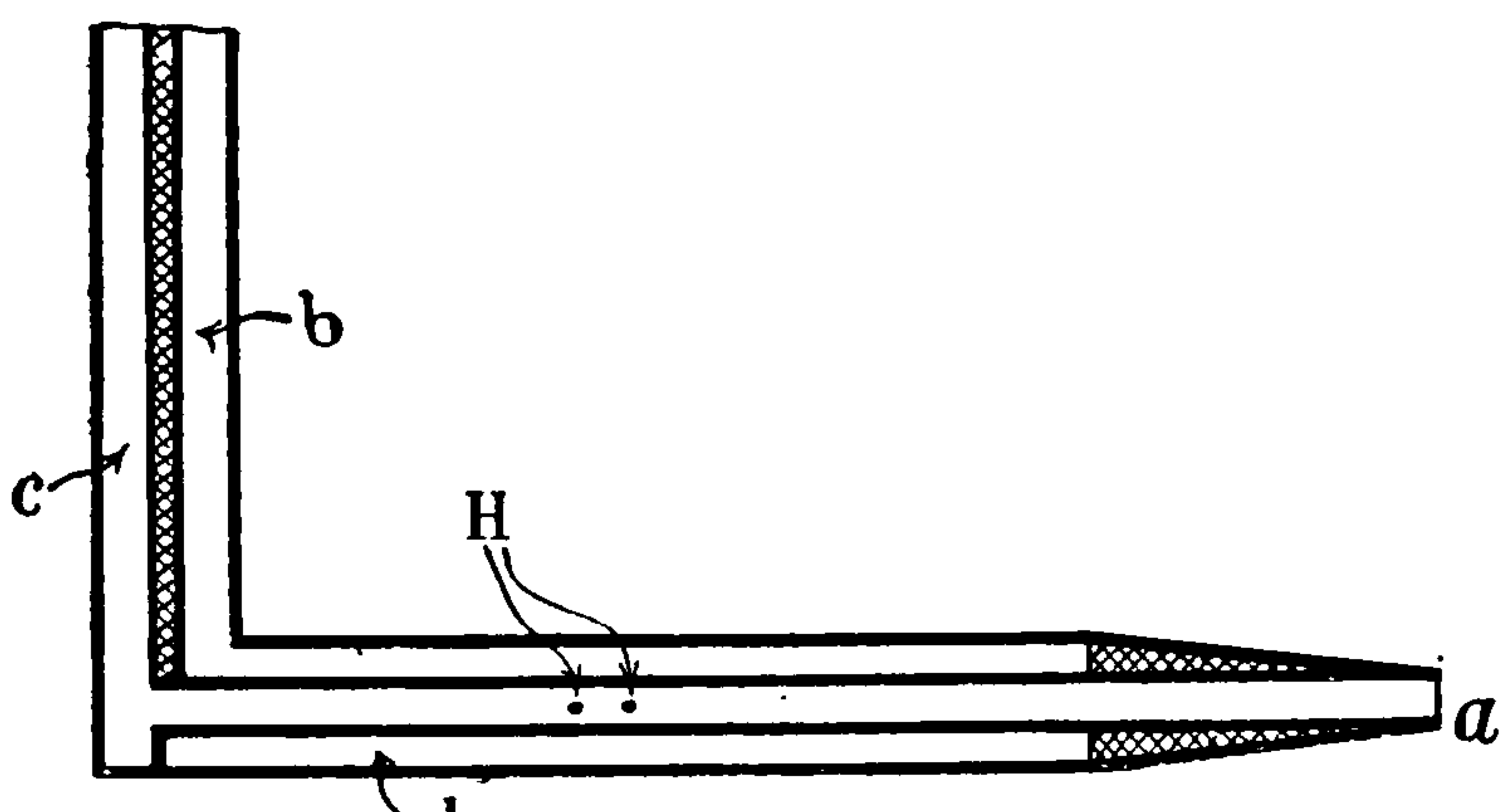


FIG. 5.

which is shown in fig. 5. It consists essentially of two concentric tubes containing air. The end a of the inner or dynamic tube faces upstream, and this tube transmits the dynamic pressure through c to one leg of the manometer. The static pressure is transmitted from the outer tube through tube b to the other leg of the manometer, small holes 0·02 in. dia. in the outer tube allowing it to communicate with the outside air.

After much experimenting, it has been established that the exact size or shape of the mouth *a* does not materially affect the result, but that the exact size, shape, and position, of the static opening are important considerations.[3] If the static opening is large or is in the shape of a slot in the side of the Pitot tube, the pressure registered is too large. Placed in positions where the flowing air produces a suction effect, the pressure registers too low; for instance, an ordinary manometer with an open mouth parallel to the flow might register too high or too low, according to the size of the mouth and the velocity of flow. For these reasons a Pitot tube of the form illustrated in fig. 5 is to be recommended.

For small pressure differences, in order to magnify the readings, the manometer is inclined to the horizontal, or a light liquid such as gasoline used.

Anemometer.—For rough work the quantity of air discharged by a fan is measured by an instrument called an anemometer. It consists of a small wheel carrying vanes set at an inclination to the plane of rotation, so that when a current of air passes through the wheel in the direction of its axis the wheel will rotate and communicate its motion to gearing which works a counter showing the number of revolutions. It was originally supposed that the number of revolutions was proportional to the velocity of the air, but this is not the case even when the current of air has a uniform velocity, because of the friction of the apparatus, and is very far from being the case when the velocity is variable. In the latter case the anemometer, which is usually graduated by rotating it in still air at the end of an arm driven at uniform velocity, greatly exaggerates the quantity of air passing through it. That this is so will be readily understood if we consider the case of an anemometer which is alternately placed during equal short periods in a current of air, and in still air; when in the former it attains a speed very nearly proportional to that of the air, and when in the latter it slows down very gradually, so that the total number of revolutions is considerably greater than would have been obtained had the instrument been placed in the current of air for half the time.

Determination of Head from the Water Gauge, and the Calculation of Air Density for a Given Humidity.—A water gauge of h in. corresponds to a pressure P lb. per sq. ft., where

$$P=\frac{\delta h}{12} \text{ lb. per sq. ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and δ = weight of 1 cu. ft. of water at the temperature at which the experiment was made. If σ is the weight of 1 cu. ft. of air at that same temperature (to find which the height of the barometer and the moisture contained by the air must be known), then the equivalent head of air against which the fan is working is

$$H=\frac{h\delta}{12\sigma} \text{ ft. of air} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

and as $\delta=62{\cdot}3$ and $\sigma=0{\cdot}075$ at 62 Fah., we may take

$$H=70\,h \text{ or } \frac{10{,}000}{144}\,h \text{ ft. of air} \quad . \quad . \quad . \quad (11)$$

where rough calculations only are necessary. Where greater accuracy is desired we require the barometer, thermometer, and hygrometer to obtain the correct value of σ. From Dalton's Law of Partial Pressures it is known that the total pressure exerted by a mixture of two gases filling a space is equal to the sum of the pressures that they would produce if they filled the space alone. In the case of unsaturated moist air we can find the pressure of the vapour, because by the hygrometer we can find the dew point or temperature at which the amount of moisture in the air would just saturate it.

The variation in the saturation pressure and volume of aqueous vapour with temperature has been carefully investigated by Regnault, but no simple physical law apparently exists between these quantities, so that they are most conveniently presented in tabular form.

Knowing the saturation volume for a given temperature, we can find the weight of moisture per cu. ft.

Let P $=$ Pressure due to the air and moisture combined, in inches of mercury.

P_s $=$ Pressure of aqueous vapour at the dew point, in inches of mercury.

[*Note.*—P_s is slightly greater than the pressure at the temperature considered, but this change in pressure may be neglected.]

σ_d $=$ Density of dry air at the standard pressure 29·92 in., and at any absolute temperate τ.

σ_s $=$ Density of aqueous vapour.

σ $=$ Density of air and vapour combined.

σ_x $=$ Density of dry air at pressure and temperature considered.

Then the weight of dry air in 1 cu. ft. of the atmosphere at the given temperature τ °F. is

$$\sigma_x = \sigma_d \frac{P - P_s}{29{\cdot}92};$$

further,

$$p\,v = 53{\cdot}2\,\tau,$$

where p is in lb. per sq. ft. and v is the volume of 1 lb. in cu. ft.

Thus $$\sigma_d = \frac{29{\cdot}92 \times 13{\cdot}6 \times \delta}{12 \times 53{\cdot}2 \times \tau} = \frac{39{\cdot}8}{\tau},$$

whence by substitution

$$\sigma_x = \frac{39{\cdot}8\,(P - P_s)}{29{\cdot}92\,\tau}.$$

This must be added to σ_s, the density of the water vapour, obtained from the saturation volume given in the table below. Thus the density of moist air at the given temperature and pressure is

$$\sigma = \sigma_x + \sigma_s.$$

If very great accuracy is desired account must be taken of the slight increase in length of the barometric column, due to the fact that the temperature τ at which it is read is not 32 deg. Fah., at which temperature only the standard barometer is 29·22. To allow for this we must write

$$\sigma_x = \frac{39{\cdot}8\,(P - P_s)}{29{\cdot}92\,(1 + 0{\cdot}0001\,\{F - 32\})\,\tau},$$

the constant 0·0001 being the specific heat of mercury.

The table of saturation pressures and volumes of aqueous vapour given on p. 16 has been taken from Dr. Jude's "Physics."[4]

The following example will make the method of obtaining the value of σ clear:—

The temperature of the atmosphere is 77 deg. Fah., and the dew point as obtained by the hygrometer is 41 Fah., what is the weight of 1 cu. ft. of air if the barometer is 29 in.? The pressure of the moisture is 0·2572 in., so that

$$P - P_s = 28{\cdot}7428 \text{ and } \sigma_x = \frac{39{\cdot}8\,(P - P_s)}{29{\cdot}92\,\tau}$$

$$= \frac{39{\cdot}8 \times 28{\cdot}74}{538 \times 29{\cdot}92} = 0{\cdot}0713 \text{ lb. per cu. ft.,}$$

because $\tau = 77 + 461 = 538$ deg. absolute Fah.

From the table we find that at 41 Fah. it requires 2,406 cu. ft. of vapour to form 1 lb.

$$\sigma_s = \frac{1}{2406} = 0{\cdot}00042 \text{ lb. per cu. ft.,}$$

so that the total weight of 1 cu. ft. of air is

$$\sigma = \sigma_x + \sigma_s - 0{\cdot}0717 \text{ lb. per cu. ft.}$$

SATURATION PRESSURES AND VOLUMES OF AQUEOUS VAPOUR.

Temperature Fah.	Saturation pressure. Inches of mercury. P_s.	Saturation volume. No. of cu. ft. per lb. $\frac{1}{\sigma_s}$.
32	0·1811	3390
41	0·2572	2406
50	0·3608	1732
59	0·5000	1264
68	0·6846	935
77	0·9279	699
86	1·2420	529
95	1·6470	405
104	2·1620	313
113	2·8110	244
122	3·6210	192
131	4·6260	152·4
140	5·8580	122
149	7·3580	98·45
158	9·1770	80·02
167	11·3600	65·47
176	13·9600	53·92

CHAPTER III.

Law of Change of Moment of Momentum.—One of the most important mechanical laws that applies to the 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 the weight of a body is W, and its velocity is v, its momentum is $\frac{W v}{g}$, and if r is the perpendicular from any point A, fig. 6, upon the line indicating the direction of motion,

corresponding radii in feet. Let c_2 be the absolute velocity of discharge, and c_1 that of inflow just before the vanes act upon each particle, while c_a is the absolute velocity immediately after; further, let u_1 and u_2 be the velocities of inflow and outflow relative to the wheel. The parallelograms of velocity of outflow and inflow are L M N B and A E F G. If a_1 and a_2 are the tangential components of c_1 and c_2 at entry and exit from the wheel, and δw is the weight of the element of air under consideration, it follows $\frac{\delta w}{g} a_2 r_2$ is the angular momentum of all forces acting on the element, since its moment of momentum is zero before reaching the wheel, and $\frac{\delta w}{g} a_2 r_2$ after leaving the wheel. If, then, ω is the angular velocity of the wheel in radians, and W is the total weight passing through the wheel per sec.,

$$\frac{W}{g} a_2 r_2 \omega = T \omega,$$

$$\frac{W}{g} a_2 v_2 = T \omega$$

= work per sec. transmitted to the wheel if T = total twisting moment in lb. ft. Hence,

$$a_2 v_2 \div g = \text{work done by wheel per lb. of air} \quad . \quad (12)$$

and neglecting the friction of the bearings, which is never a very great quantity, this is the work done per lb. of air on the fan shaft, however the air may approach the fan. For if no force acts on the air before it reaches the fan it can have no moment of momentum, and therefore must approach the fan radially, or, if inflow is axial, axially. Hence the work done by the wheel, and therefore that done on the shaft by the motor, must be $a_2 v_2 \div g$; but if the friction of shaft or arms acts on the air and gives it angular momentum before reaching the wheel, then $a_2 v_2 \div g$ is the work done by the wheel vanes, arms, and the friction of the shaft upon the air, or is the useful work done by the motor on the shaft. Thus equation (12) not only applies to fans in which the flow through the wheel is wholly in a plane perpendicular to the axis of the shaft, or *radial flow fans*, but also to those in which the flow is changed from an axial direction to a radial direction, or *mixed flow fans*.

Losses of Energy or Head while passing through the Fan.— In passing through a fan there are several losses of head,

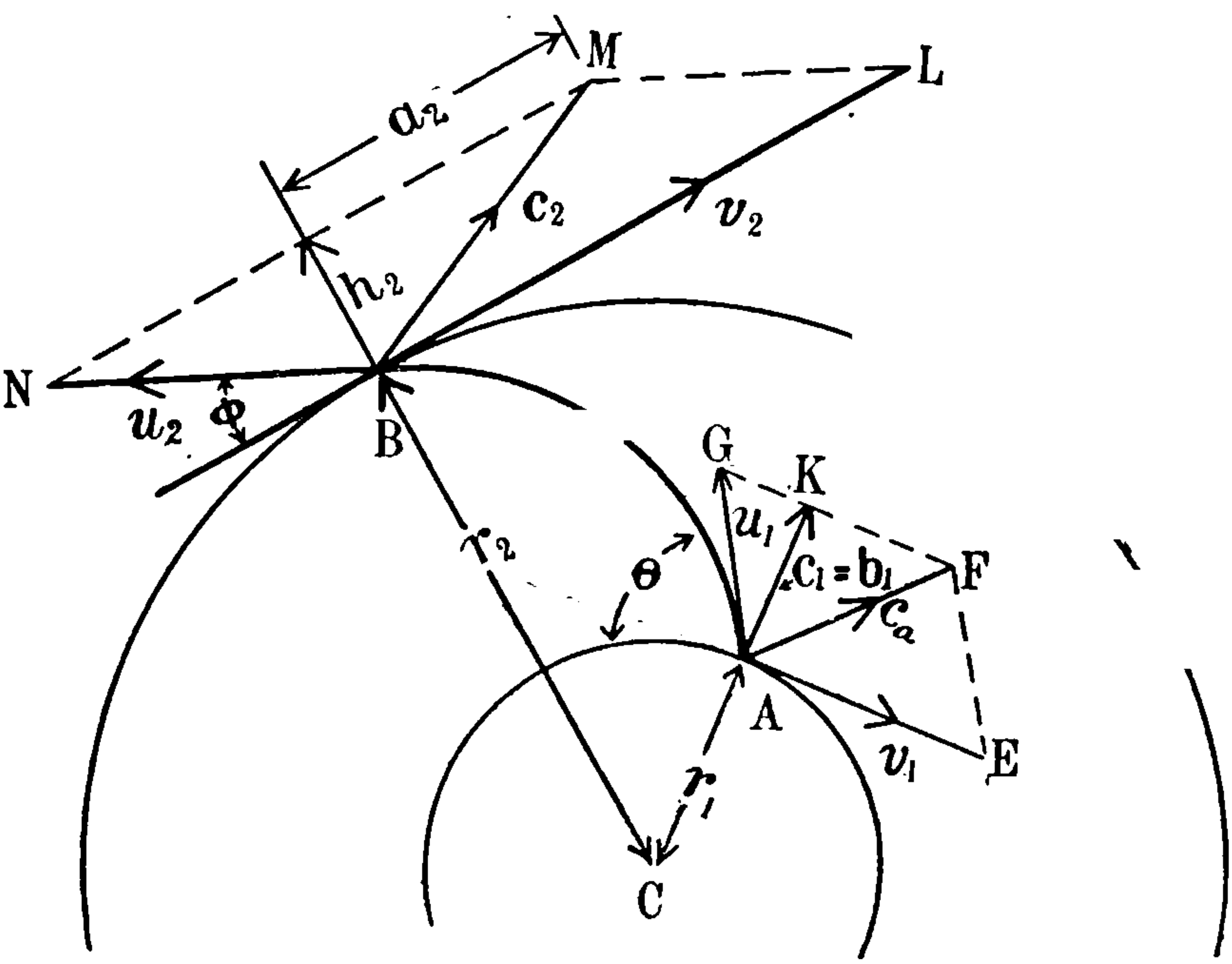


FIG. 7.

which by proper design may either be entirely avoided or reduced to a minimum when the orifice or

$$\frac{Q}{\sqrt{gH}}$$

has the value for which the fan has been designed, where Q = cu. ft. or metres per sec., H = head of air in ft. or metres given by formula (10) or (11), and $g = 32{\cdot}2$ for British units, or 9·81 for metric units. Fig. 8 is an outline drawing of a fan showing two sectional elevations. In some fans there are two eyes, A, at which the air enters; in others one eye only is employed. The air passes from the eye through the wheel B, which rotates clockways, into the diffuser C. This diffuser is sometimes made with parallel sides, but often with a slight taper of about 7 deg. Its inner surface is always cylindrical, but its outer surface is sometimes of a spiral form, as in the Rateau ventilator to be described later. It is, however, usually cylindrical. Most fans are constructed without any diffuser at all, and the wheel B discharges directly into the volute D, which is usually of rectangular or circular section increasing from the beak E, according to a formula to be dealt with later, and having its greatest section at the base of the chimney F. The latter gradually increases in section as shown, so as to reduce the velocity of discharge. Referring to fig. 7, we see that when the air enters the wheel at A its direction may be suddenly changed from c_1 to c_a, so that the loss of head is

$$L_1 = \frac{(K\,F)^2}{2\,g}$$

$$= \frac{(v_1 - b_1 \cot \theta)^2}{2\,g}$$

where b_1 is the radial component of the absolute speed immediately after entry to the wheel, and θ is the angle made by the vane A B at A, with a tangent to the circle through A. In order that this loss of head may be avoided, we must make θ such that

$$\cot \theta = \frac{v_1}{b_1},$$

which, we shall presently show, can only be the case for one value of

$$\frac{Q}{\sqrt{g\,H}}.$$

After passing through the wheel the air enters either the atmosphere if the fan has no casing, the diffuser, or in the case of a fan without any diffuser it passes directly into the volute; in the first case the head lost is $c_2{}^2 \div 2\,g$, as the kinetic energy at discharge is all lost; in the second case there is no loss when entering the diffuser, whilst in the third the loss is (fig. 9)

$$L_2 = \frac{PM^2}{2g}$$

$$= \frac{b_2^2 + (a_2 - c_v)^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where c_v is the velocity in the volute which has a direction very nearly tangential to the wheel. If the fan has no

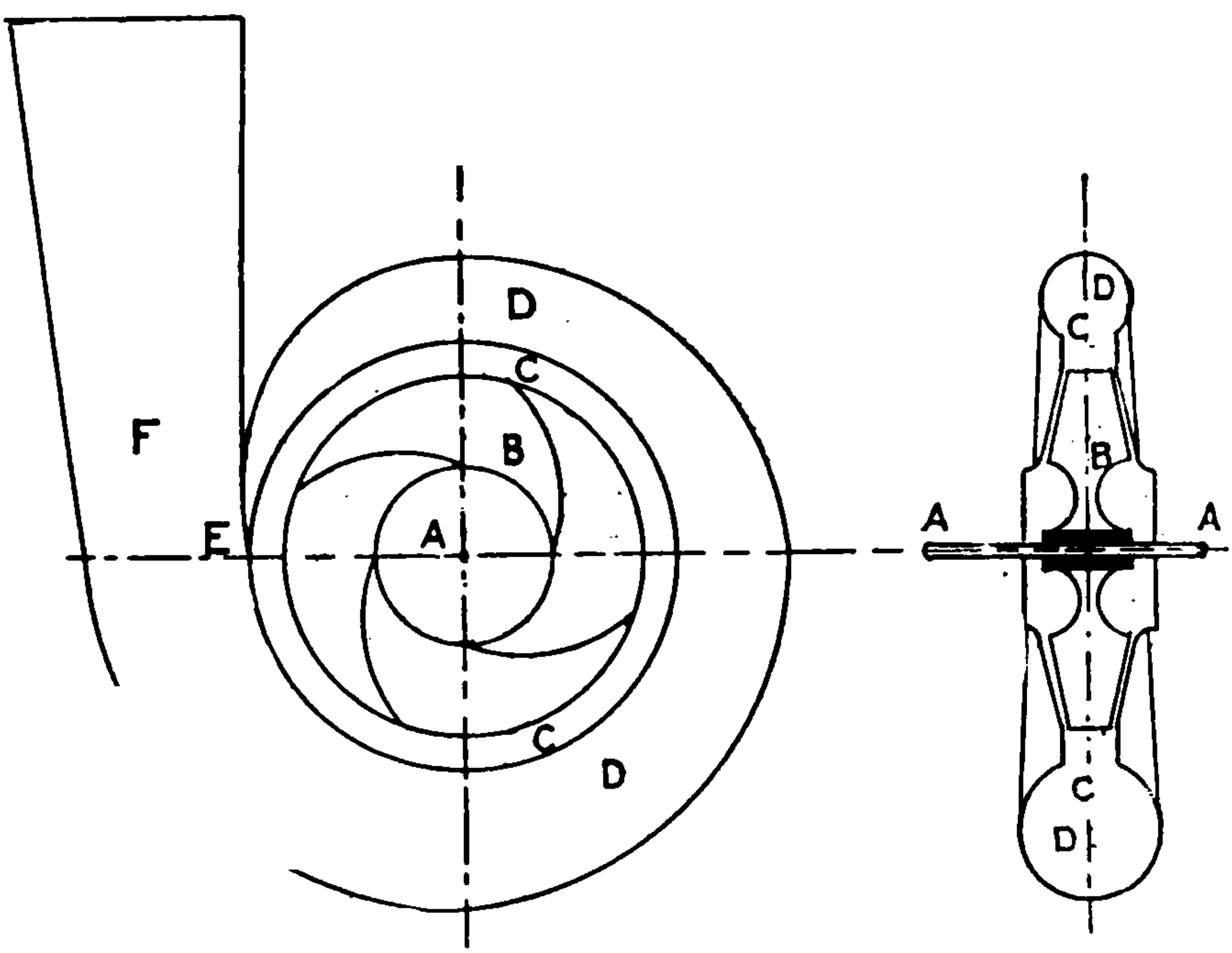


FIG. 8.

chimney or expanding discharge pipe c_v should $= \frac{1}{2} a_2$, so as to obtain a maximum gain of pressure head; but if it has, then c_v should $= a_2$, giving a minimum shock loss, for the reasons given in Chap. I, equations (4) and (6). If the fan has a diffuser, and BD is its inner and CE its outer circumference (fig. 10), the latter having a radius r_3, then the change of the angular momentum of each particle of air therein is nil, because no force acts on it during its passage through the diffuser. Further, if a_3, b_3 are the tangential and radial components at discharge, and s_2, s_3 the breadths of diffuser at inflow and discharge, then

$$a_2 r_2 = a_3 r_3$$

$$\frac{a_3}{a_2} = \frac{r_2}{r_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

and

$$2 \pi r_2 s_2 b_2 = 2 \pi r_3 s_3 b_3$$

$$\frac{b_3}{b_2} = \frac{r_2 s_2}{r_3 s_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

and if the sides of the diffuser are parallel, or $s_2 = s_3$,

$$\frac{b_3}{b_2} = \frac{r_2}{r_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

This, however, neglects the thickness of the vanes; the path B C is then an equi-angular spiral. But s_3 is usually made slightly greater than s_2, so that the sides are inclined to one another at an angle of about 7 deg.

The air next passes into the volute, and the loss of head is

$$L_3 = \frac{b_3^2 + (a_3 - c_v)^2}{2 g} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

and if there is an expanding discharge pipe we should have $c_v = a_3$, and if not, $c_v = \frac{1}{2} a_3$, according to Chap. I., equations (4) and (6). In addition to the above there is the loss due to surface friction and bends which may be written

$$L_4 = F_1 \frac{u_2^2}{2 g} + F_2 \frac{c_v^2}{2 g} \quad . \quad . \quad . \quad . \quad . \quad (18)$$

where $F_1 \frac{u_2^2}{2 g}$ accounts for friction and bends in the wheel, and $F_2 \frac{c_v^2}{2 g}$ the loss by friction in the diffuser and friction and bends in the volute. There is also the loss of head due to the kinetic energy contained by the air at discharge.

$$=\frac{c_d^2}{2g},$$

where c_d is the speed in the discharge pipe.

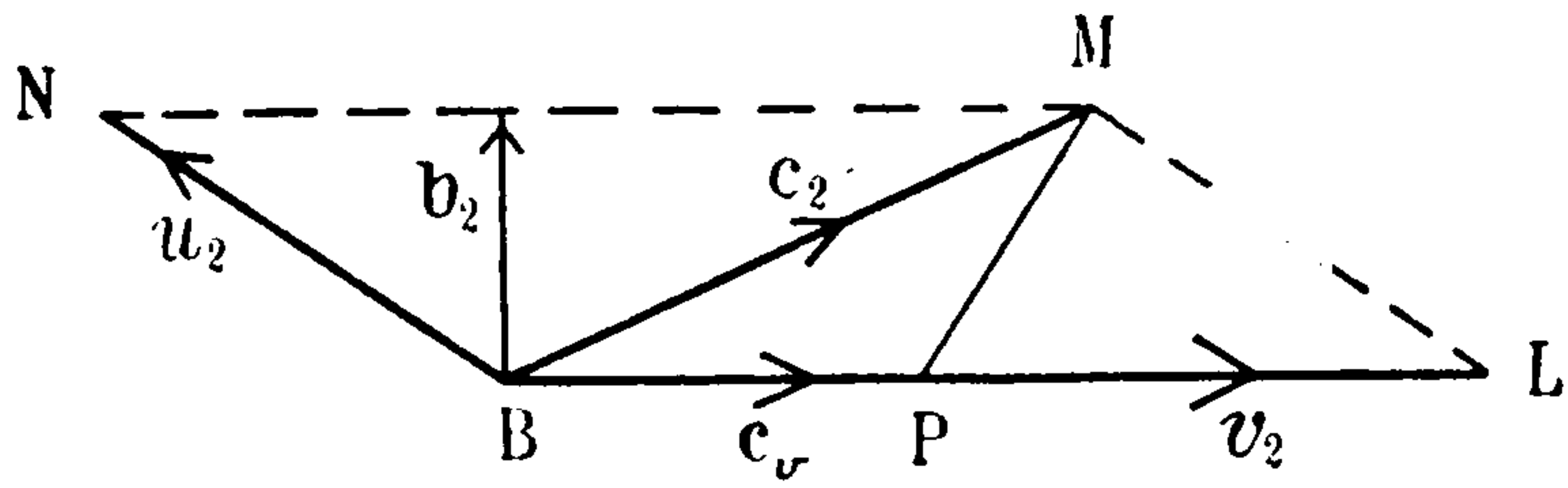


FIG. 9.

The values of c_v in terms of a_2 and a_3 can only be obtained at one orifice, and make

$$L_2=\frac{b_2^2}{2g}, \text{ or } \frac{b_2^2+\frac{a_2^2}{4}}{2g} \quad . \quad . \quad . \quad . \quad . \quad (19)$$

and
$$L_3=\frac{b_3^2}{2g}, \text{ or } \frac{b_3^2+\frac{a_3^2}{4}}{2g} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

Equation for finding the Manometric and Mechanical Efficiencies in Terms of ϕ.—The work done by the wheel per lb. of air delivered is equal to the head H given to the air, together with the work absorbed by losses of head; hence

$$\frac{a_2 v_2}{g} - \text{losses of head} = \text{H}.$$

Case A.—In the case of a fan without any casing these losses include—

(1) Shock loss at entry to the wheel.
(2) Losses by friction and bends in passing through the wheel.
(3) The leaving loss depending on the absolute speed from the wheel.

Thus we have

$$\frac{a_2 v_2}{g} - \frac{c_2^2}{2g} - F_1 \frac{u_2^2}{2g} - \frac{(v_1 - b_1 \cot \theta)^2}{2g} = H \quad . \quad . \quad (21)$$

where $a_2 = v_2 - b_2 \cot \phi$,

ϕ being the angle between tangent to the curve A B at B, and the circle through B, measured clockwise (fig. 7). In some fans ϕ is greater than 90 deg., so that cot ϕ is negative.

$$c_2^2 = a_2^2 + b_2^2 = b_2^2 + (v_2 - b_2 \cot \phi)^2$$

$$u_2^2 = b_2^2 \operatorname{cosec}^2 \phi = b_2^2 (1 + \cot^2 \phi)$$

and $$v_1 = v_2 \frac{r_1}{r_2},$$

so that it is evident that the equation for a given fan can be thrown into the form,—

$$v_2^2 + P v_2 Q - R Q^2 - S g H = 0 \quad . \quad . \quad . \quad (22)$$

where P, R, and S are constants containing F, ϕ, and θ, of which R and S are positive and P may be positive or negative.

Case B.—If the fan has a diffuser and no volute, then the losses comprise—

(1) Shock loss at entry to the wheel.
(2) Losses by friction and bends in passing through the wheel.
(3) Friction loss in the diffuser.
(4) Leaving loss depending on the absolute speed from the diffuser.

Thus we have

$$\frac{a_2 v_2}{g} - \frac{(v_1 - b_1 \cot \theta)^2}{2g} - F_1 \frac{u_2^2}{2g} - (1 + F_2) \frac{c_3^2}{2g} = H \quad (23)$$

angles as the discharge becomes small. The reader must remember that θ and ϕ are the angles made with the tangents to the inner and outer circumferences of the wheel, by the mean direction of flow to and from each passage (fig. 8). It is such points as these which account for the difference, usually of small magnitude, however, which is found between a series of experimental results and a theory, which is not absolutely correct, owing to the difficulty encountered in expressing mathematically the exact variation of the quantities involved.

We shall now consider the effect of varying ϕ in equations (21) to (25) on the mechanical efficiency of the fan, and on the manometric efficiency

$$N_m = \frac{g\,H}{v_2^2} \quad \ldots \ldots \quad (25a)$$

The mechanical efficiency does not differ much from the ratio of the useful work done per lb. H to the work done by the wheel per lb. $a_2\,v_2 \div g$. The difference is due to the friction of the wheel bearings, and to the friction of the outside casing of the wheel with the air, which we have not included in the above equations, and which is only noticeable at very small orifices. We shall therefore find the ratio, which we shall call the air efficiency,

$$\eta = \frac{g\,H}{a_2\,v_2} \quad \ldots \ldots \ldots \quad (26)$$

and we shall throughout take

$$b_2 = 0{\cdot}5\,\sqrt{g\,H} \quad \ldots \ldots \quad (27)$$

when $$v_1 = b_1 \cot\theta$$

—that is, when the air glides on to the wheel vanes without shock loss.

Equation (21) thus becomes

$$\frac{v_2\,(v_2 - b_2\cot\phi)}{g} - \frac{b_2^2 + (v_2 - b_2\cot\phi)^2}{2\,g} - F_1\frac{b_2^2\,\text{cosec}^2\,\phi}{2\,g} = H$$

$$2\,v_2^2 - 2\,v_2\,b_2\cot\phi - b_2^2 - v_2^2 + 2\,b_2\,v_2\cot\phi - b_2^2\cot^2\phi$$
$$- F_1\,b_2^2\,\text{cosec}^2\,\phi = 2\,g\,H\,;$$

from which

$$v_2^2 - b_2^2\,(1 + \cot^2\phi) - b_2^2\,F_1\,\text{cosec}^2\,\phi = 2\,g\,H,$$

and $$v_2^2 - b_2^2\,(1 + F_1)\,\text{cosec}^2\,\phi = 2\,g\,H\,;$$

whence by equation (27)

$$v_2^2 - \tfrac{1}{4}\,(1 + F)\,\text{cosec}^2\,\phi\,g\,H = 2\,g\,H.$$

Eliminating v_2

$$N_m = \frac{1}{m^2} = \frac{g\,H}{v_2^2} = \frac{4}{8 + (1 + F_1)\,\text{cosec}^2\,\phi} \quad . \quad . \quad (28)$$

also the air efficiency

$$\eta = \frac{g\,H}{v_2\,(v_2 - b_2\cot\phi)},$$

which from equation (27) becomes

$$m\,(m - \tfrac{1}{2}\cot\phi) \qquad ? \quad . \quad . \quad . \quad . \quad (29)$$

Considering next case, B, in which there is a diffuser but no volute, let r_3 the external radius of the diffuser be $k\,r_2$, and suppose the sides parallel to one another; then from equation (23)

$$\frac{v_2\,(v_2 - b_2\cot\phi)}{g} - F_1\,\frac{b_2^2\,\text{cosec}^2\,\phi}{2\,g} - (1 + F_2)\,.\,\frac{a_3^2 + b_3^2}{2\,g} = H,$$

since the shock loss at entry to the wheel

$$= \frac{v_1 - b_1\cot\theta}{2\,g} = 0.$$

Now, in the case of a parallel diffuser

$$a_3\,r_3 = a_2\,r_2 \text{ and } b_3\,r_3 = b_2\,r_2\,;$$
$$\therefore\ a_3 = \frac{1}{k}\,a_2 \text{ and } b_3 = \frac{1}{k}\,b_2.$$

Hence we get

$$\frac{v_2(v_2 - b_2 \cot \phi)}{g} - F_1 \frac{b_2^2 \operatorname{cosec}^2 \phi}{2g}$$
$$- (1 + F_2) \frac{b_2^2 + (v_2 - b_2 \cot \phi)^2}{2 g k^2} = H ;$$

$$2 v_2^2 - 2 v_2 b_2 \cot \phi - F_1 b_2^2 \operatorname{cosec}^2 \phi$$
$$- \frac{1}{k^2}(1 + F_2)[b_2^2 + v_2^2 - 2 v_2 b_2 \cot \phi + b_2^2 \cot^2 \phi] = H . 2g$$

$$v_2^2\left[2 - \frac{1+F_2}{k^2}\right] - 2 b_2 v_2 \cot \phi \left[1 - \frac{1+F_2}{k^2}\right]$$
$$- b_2^2 \operatorname{cosec}^2 \phi \left[F_1 + \frac{1+F_2}{k^2}\right] = 2 g H,$$

and putting $$b_2 = \tfrac{1}{2}\sqrt{g H}$$

$$v_2^2\left[2 - \frac{1+F_2}{k^2}\right] - v_2 \sqrt{g H} \cot \phi \left[1 - \frac{1+F_2}{k^2}\right]$$
$$- g H \left[2 + \tfrac{1}{4}\left(F_1 + \frac{1+F_2}{k^2}\right) \operatorname{cosec}^2 \phi\right] = 0 \quad . \quad (30)$$

Giving v_2 as a function of H, whence substituting for v_2^2 in $N_m = \frac{g H}{v_2^2}$, we obtain an expression for manometric efficiency in terms of the variable ϕ.

If the fan has a volute and chimney, but no diffuser, and we suppose that $c_v = a_2$, then equation (24) may be put in the form

$$2 v_2^2 - 2 v_2 b_2 \cot \phi - F_1 b_2^2 \operatorname{cosec}^2 \phi - b_2^2 - F_2 (v_2 - b_2 \cot \phi)^2$$
$$- c_d^2 = 2 g H.$$

Let $c_d = \frac{1}{8}\sqrt{g} H$. Then, since $b_2 = \frac{1}{2}\sqrt{g} H$,

$$v_2^2 (2 - F_2) - v_2 \sqrt{g} H \cot \phi (1 - F_2)$$
$$- g H \left[2\tfrac{17}{64} + \frac{F_1}{4} \operatorname{cosec}^2 \phi + \frac{F_2}{4} \cot^2 \phi\right] = 0 \quad . \quad (31)$$

Thus we can find v_2 in terms of H, and subsequently N_m in terms of ϕ.

If on the other hand $c_v = \frac{1}{2}\, a_2$, and there is no chimney, then equation (24) becomes

$$2\, v_2^2 - 2\, v_2\, b_2 \cot\phi - F_1\, b_2^2 \operatorname{cosec}^2\phi - b_2^2 - \tfrac{1}{4}(v_2 - b_2\cot\phi)^2$$
$$- \tfrac{1}{4} F_2 (v_2 - b_2 \cot\phi)^2 - \tfrac{1}{4}(v_2 - b_2\cot\phi)^2 = 2\, g\, H,$$

since c_d is now the velocity of discharge from the volute, and therefore

$$= \tfrac{1}{2}(v_2 - b_2\cot\phi).$$

$$v_2^2\left(\frac{3}{2} - \frac{F_2}{4}\right) - v_2\, b_2 \cot\phi\left(1 - \frac{F_2}{2}\right)$$
$$- b_2^2\left[1 + F_1 \operatorname{cosec}^2\phi + \cot^2\phi\left(\frac{1}{2} + \frac{F_2}{4}\right)\right] = 2\, g\, H,$$

and putting $$b_2 = \tfrac{1}{2}\sqrt{g\,H}$$

$$v_2^2\left(\frac{3}{2} - \frac{F_2}{4}\right) - v_2\sqrt{g\,H}\,\frac{\cot\phi}{2}\left(1 - \frac{F_2}{2}\right)$$
$$- \tfrac{1}{4}\, g\, H\left[9 + F_1 \operatorname{cosec}^2\phi + \cot^2\phi\left(\frac{1}{2} + \frac{F_2}{4}\right)\right] = 0 \qquad (32)$$

A quadratic in v_2 which ultimately gives N_m in terms of ϕ by the application of equation (25*a*).

If there is a diffuser, volute, and chimney, and if $r_3 = k\, r_2$, so that $a_3 = \frac{a_2}{k}$ and $b_3 = \frac{b_2}{k}$, then from equation (25)

$$2\, v_2^2 - 2\, v_2\, b_2 \cot\phi - F_1\, b_2^2 \operatorname{cosec}^2\phi$$
$$- \frac{b_2^2}{k^2} - F_2\,\frac{(v_2 - b_2\cot\phi)^2}{k^2} - \frac{g\,H}{64} = 2\, g\, H,$$

since $c_r = a_3 = \frac{a_2}{k}$ and $c_d = \frac{1}{8}\sqrt{g\,H}$

$$v_2^2\left(2 - \frac{F_2}{k^2}\right) - 2\, v_2\, b_2 \cot\phi\left(1 - \frac{F_2}{k^2}\right)$$
$$- b_2^2\left(-\frac{1}{k^2} + F_1 \operatorname{cosec}^2\phi + F_2\,\frac{\cot^2\phi}{k^2}\right) - \frac{g\,H}{64} - 2\, g\, H = 0,$$

and putting $$b_2 = \frac{1}{2}\sqrt{g\,\mathrm{H}}$$

we obtain $$v_2^2\left(2 - \frac{\mathrm{F}^2}{k^2}\right) - v_2\sqrt{g\,\mathrm{H}}\cot\phi\left(1 - \frac{\mathrm{F}_2}{k^2}\right)$$
$$- g\,\mathrm{H}\left(\frac{1}{4\,k^2} + \frac{\mathrm{F}_1\operatorname{cosec}^2\phi}{4} + \frac{\mathrm{F}_2\cot^2\phi}{4\,k^2} + 2\tfrac{1}{64}\right) = 0 \qquad (33)$$

Which gives v_2 in terms of ϕ and H. Hence we can find the manometric efficiency in terms of ϕ.

The values of F_1, F_2 that agree best with practice are both $\frac{1}{8}$. Substituting this in (28), we obtain

$$v_2^2 - 0{\cdot}281\,g\,\mathrm{H}\operatorname{cosec}^2\phi = 2\,g\,\mathrm{H} \quad . \quad . \quad . \quad (34)$$

from which the following table is calculated :—

$\phi = 15$	$v_2 = 2{\cdot}49\sqrt{g\,\mathrm{H}}$	$\mathrm{N}_m = 0{\cdot}16$	$\eta = 0{\cdot}65$
$= 30$	$1{\cdot}77$	$0{\cdot}32$	$0{\cdot}63$
$= 45$	$1{\cdot}60$	$0{\cdot}39$	$0{\cdot}57$
$= 90$	$1{\cdot}51$	$0{\cdot}44$	$0{\cdot}44$

so that the efficiency of an open-running fan without diffuser or volute is very low unless the vanes are curved backwards, and it must be remembered that to get the mechanical efficiency of engine and fan we must multiply η by 0·85 on the average, so that the greatest mechanical efficiency of the combination is 0·56 per cent. Taking next the case of the open-running fan with a diffuser whose sides are parallel and whose external radius is $1\frac{1}{4}$ that of the fan, that is

$$r_3 = 1\tfrac{1}{4}\,r_2,$$

equation (30) becomes

$$1{\cdot}28\,v_2^2 - 0{\cdot}28\,v_2\sqrt{g\,\mathrm{H}}\cot\phi$$
$$- g\,\mathrm{H}\,(2 + 0{\cdot}211\operatorname{cosec}\phi^2) = 0 \quad . \quad . \quad . \quad (35)$$

which gives the following table :—

$\phi = 15$	$v_2 = 2{\cdot}46\sqrt{g\,\mathrm{H}}$	$\mathrm{N}_m = 0{\cdot}17$	$\eta = 0{\cdot}68$
30	$1{\cdot}69$	$0{\cdot}35$	$0{\cdot}72$
45	$1{\cdot}49$	$0{\cdot}45$	$0{\cdot}68$
90	$1{\cdot}31$	$0{\cdot}58$	$0{\cdot}58$

so that the highest possible mechanical efficiency of engine and fan would be 61 per cent. It must clearly be understood that ϕ is not the vane angle, but the angle of flow, and this will probably be from 15 deg. to 30 deg. greater than the vane angle. When the fan has no diffuser but a volute and chimney, and the former of such a section that $c_v = a_2$, then (31) becomes

$$1{\cdot}875\, v_2^2 - 0{\cdot}875 \cot\phi\, v_2 \sqrt{g\mathrm{H}} - g\mathrm{H}\left(2{\cdot}297 + \frac{\cot^2\phi}{16}\right) = 0 \quad . \quad . \quad . \quad (36)$$

from which we obtain

$\phi =$ 15	$v_2 = 2{\cdot}43\sqrt{g\mathrm{H}}$	$\mathrm{N}_m = 0{\cdot}17$	$\eta = 0{\cdot}72$
30	1·62	0·38	0·81
45	1·38	0·53	0·83
90	1·11	0·81	0·82
120	0·98	1·03	0·80
135	0·91	1·20	0·78

giving a maximum efficiency of engine and fan of 70 per cent.

If, however, there is no chimney, but the air is discharged direct from the volute, and $c_v = \frac{1}{2} a_2$, then (32) becomes

$$1{\cdot}47\, v^2 - 0{\cdot}469\, v^2 \sqrt{g\mathrm{H}} \cot\phi - g\mathrm{H}(2{\cdot}28 + 0{\cdot}164 \cot^2\phi) = 0 \quad . \quad . \quad . \quad (37)$$

which gives us the following table :—

$\phi =$ 15	$c_1 = 2{\cdot}45\sqrt{g\mathrm{H}}$	$\mathrm{N}_m = 0{\cdot}17$	$\eta = 0{\cdot}69$
30	1·67	0·36	0·74
45	1·46	0·47	0·71
90	1·24	0·65	0·65

giving a maximum efficiency of engine and fan of 63 per cent.

Finally, if there are diffuser, volute, and chimney,

$$1{\cdot}92\, v_2^2 - 0{\cdot}92 \cot\phi\, v_2 \sqrt{g\mathrm{H}} - g\mathrm{H}(2{\cdot}2 + 0{\cdot}051 \cot^2\phi) = 0 \quad . \quad . \quad . \quad (38)$$

and we get the following table :—

$\phi = 15$	$v_2 = 2 \cdot 41 \sqrt{g\,\mathrm{H}}$	$\mathrm{N}_m = 0 \cdot 172$	$\eta = 0 \cdot 75$
30	1·59	0·39	0·86
45	1·35	0·55	0·87
60	1·22	0·67	0·87
90	1·07	0·87	0·87
120	0·94	1·13	0·87
135	0·87	1·32	0·84

giving a maximum efficiency of engine and fan of 74 per cent.

A fan designed in this manner would require a very long chimney, as the taper of a chimney cannot be very great, in order to reduce the velocity of the air to so low a value as $\frac{1}{8}\sqrt{g\mathrm{H}}$. We shall therefore consider the case in which the external radius of the diffuser is $1\frac{1}{2}$ that of the wheel, and the velocity c_v in the volute is $\frac{1}{2}\,a_3$. Equation (25) then becomes, when we put

$$v_1 = b_1 \cot \theta, \qquad a_3 = \frac{v_2 - b_2 \cot \phi}{1 \cdot 5},$$

and

$$b_2 = 0 \cdot 5 \sqrt{g\,\mathrm{H}},$$

$$1 \cdot 875\, v_2^2 - \tfrac{7}{8}\, v_2 \sqrt{g\,\mathrm{H}} \cot \phi$$
$$- g\,\mathrm{H}\,(2 \cdot 16 + 0\ 0625 \cot^2 \phi) = 0 \quad . \quad . \quad (38a)$$

from which we obtain the following table :—

$\phi = 15$	$v_2 = 2 \cdot 41 \sqrt{g\,\mathrm{H}}$	$\mathrm{N}_m = 0 \cdot 17$	$\eta = 0 \cdot 75$
= 30	1·60	0·39	0·85
= 45	1·35	0·55	0·87
= 90	1·08	0·86	0 86
= 120	0·96	1·09	0·84
= 135	0·88	1·29	0·82

because if A_1, A_2, etc., are the sections of passages in a machine through which a liquid flows, and V_1, V_2, etc., the corresponding mean velocities of flow, then

$$A_1 V_1 = A_2 V_2 = A_3 V_3 = \quad \ldots \ldots \quad (41)$$

Again, if $c_d = \frac{1}{8}\sqrt{g\,H}$ when $b_2 = \frac{1}{2}\sqrt{g\,H}$,

$$c_d = \tfrac{1}{4}\, b_2,$$

so that (24) becomes

$$2\,v_2^2 - 2\,v_2\,b_2 \cot\phi - \left(\frac{v_2^2}{9} - \tfrac{2}{3}\,v_2\,b_2 \cot\theta + b_2^2 \cot^2\theta\right)$$
$$- \tfrac{1}{8}\,b_2^2 \operatorname{cosec}^2\phi - [b_2^2 + (v_2 - b_2 \cot\phi - 1{\cdot}52\,b_2)^2]$$
$$- 0{\cdot}289\,b_2^2 - 0{\cdot}0625\,b_2^2 = 2\,g\,H.$$

$$\tfrac{8}{9}\,v_2^2 + (3\ 04 + \tfrac{2}{3}\cot\theta)\,v_2\,b_2$$
$$- (\cot^2\theta + 1\tfrac{1}{8}\cot^2\phi + 3{\cdot}04\cot\phi + 3{\cdot}79)\,b_2^2 = 2\,g\,H \quad . \ (42)$$

So that

$$\frac{1}{N_m} + \tfrac{9}{8}\,(3\ 04 + \tfrac{2}{3}\cot\theta)\,\frac{b_2}{\sqrt{g\,H}} \cdot \frac{1}{\sqrt{N_m}}$$
$$- \tfrac{9}{8}\,(\cot^2\theta + 1\tfrac{1}{8}\cot^2\phi + 3{\cdot}04\cot\phi + 3{\cdot}79)\,\frac{b_2^2}{g\,H} - \tfrac{9}{4} = o.$$

Putting $\phi = 30$, $\cot\theta = 1{\cdot}08$, this becomes

$$\frac{1}{N_m} + 4{\cdot}23\,\frac{b_2}{\sqrt{g\,H}} \cdot \frac{1}{\sqrt{N_m}} - 15{\cdot}3\,\frac{b_2}{g\,H} - \tfrac{9}{4} = o,$$

and thus

$$\frac{1}{\sqrt{N_m}} = \frac{-4{\cdot}23\,\dfrac{b_2}{\sqrt{g\,H}} + \sqrt{(17{\cdot}9 + 61{\cdot}2)\dfrac{b_2}{g\,H} + 9}}{2}$$

$$= \frac{-4{\cdot}23\,\dfrac{b_2}{\sqrt{g\,H}} + \sqrt{79{\cdot}1\,\dfrac{b_2^2}{g\,H} + 9}}{2} = \frac{v_2}{\sqrt{g\,H}} = m \quad . \ (43)$$

From which we can find m, and thence η, so that Table 2 is readily obtained; only it must be remembered that for small orifices the fan efficiency is really less than $g\,H \div a_2\,v_2$, owing to external friction between the outer surface of the wheel casing and the air, or, if the wheel is open at the sides, between the fan casing and the air, which friction $\propto v_2^2$ in any fan, and the work wasted $\propto v_2^3$. If the efficiency were equal to $g\,H \div a_2\,v_2$, then at zero orifice it would be equal to N_m, whereas its real value is zero.

TABLE 2.—VARIATION OF MANOMETRIC AND AIR EFFICIENCY WITH $v_2 \div \sqrt{g\,H}$ AND $b_2 \div \sqrt{g\,H}$.

$\frac{b_2}{\sqrt{g\,H}}$.	$m = \frac{v_2}{\sqrt{g\,H}}$.	N_m.	η.
0	1·50	0·44	0
0·1	1·35	0·55	0·63
0·2	1·32	0·57	0·78
0·3	1·37	0·53	0·86
0·4	1·48	0·46	0·86
0·5	1·62	0·38	0·82
0·6	1·79	0·31	0·74
0·8	2·17	0·21	0·59
1·0	2·58	0·15	0·46
1·5	3·67	0·074	0·25
2·0	4·80	0·043	0·16

If $\phi = 90$ deg., then, when there is no shock at inflow, $v_2 = 1{\cdot}11\sqrt{g\,H}$, and therefore

$$\cot\theta = \frac{v_1}{c_1} = \frac{1{\cdot}11}{1{\cdot}5} = 0{\cdot}74\,;$$

also at this orifice $c_v = a_2 = v_2 = 1{\cdot}11\sqrt{g\,H}$;

$$\therefore\ c_v = \frac{1{\cdot}11}{0{\cdot}5}\,b_2 = 2{\cdot}22\,b_2 \text{ at all orifices,}$$

and
$$c_d = \frac{b_2}{4},$$

so that (24) becomes

$$2\,v_2^2 - \left(\frac{v_2^2}{9} - \tfrac{2}{3}\,v_2\,b_2 \cot\theta + b_2^2 \cot^2\theta\right) - \tfrac{1}{8}\,b_2^2 \operatorname{cosec}^2\phi$$

$$-\left[b_2^2 + (v_2 - 2{\cdot}22\,b)^2\right] - 0{\cdot}616\,b_2^2 - \frac{b_2^2}{16} = 2\,g\,\mathrm{H}.$$

$$\tfrac{8}{9}\,v_2^2 + 4{\cdot}93\,v_2\,b_2 - 7{\cdot}28\,b_2^2 - 2\,g\,\mathrm{H} = o$$

$$v_2^2 + 5{\cdot}54\,v_2\,b_2 - 8{\cdot}19\,b_2^2 - \tfrac{9}{4}\,g\,\mathrm{H} = o$$

that is, $$\frac{1}{\mathrm{N}_m} + 5{\cdot}54\,\frac{1}{\sqrt{\mathrm{N}_m}}\frac{b_2}{\sqrt{g\,\mathrm{H}}} - 8{\cdot}19\,\frac{b_2^2}{g\,\mathrm{H}} - \tfrac{9}{4} = o$$

$$\therefore\ m = \frac{1}{\sqrt{\mathrm{N}_m}} = \frac{-5{\cdot}54\,\dfrac{b_2}{\sqrt{g\,\mathrm{H}}} + \sqrt{63{\cdot}5\,\dfrac{b_2^2}{g\,\mathrm{H}} + 9}}{2} \qquad (44)$$

from which we obtain Table 3.

TABLE 3.—VARIATION OF MANOMETRIC AND AIR EFFICIENCY WITH $b_2 \div \sqrt{g\mathrm{H}}$ AND $v_2 \div \sqrt{g\mathrm{H}}$.

$\frac{b_2}{\sqrt{g\mathrm{H}}}$.	$\frac{v_2}{\sqrt{g\mathrm{H}}}$.	$\mathrm{N}m$.	η.
0	1·50	0·44	0
0·1	1·27	0·62	0·62
0·2	1·14	0·76	0·76
0·3	1·09	0·85	0·85
0·4	1·08	0·86	0·86
0·5	1·11	0·82	0·82
0·6	1·16	0·75	0·75
0·8	1·30	0·59	0·59
1·0	1·48	0·45	0·45
1·5	2·00	0·25	0·25
2·0	2·57	0·15	0·15

If $\phi = 135$, then, when there is no shock at inflow, $v^2 = 0{\cdot}91\ \sqrt{g\text{H}}$, and therefore

$$\cot\theta = \frac{v_1}{b_2} = \frac{0{\cdot}91}{1{\cdot}5} = 0{\cdot}606.$$

Also at this orifice

$$c_v = a_2 = v_2 - b_2 \cot\phi = v_2 + b_2 = 1{\cdot}41\ \sqrt{g\text{H}},$$

so that $c_v = 2{\cdot}82\ b_2$ at all orifices, and $c_d = b_2 \div 4$, so that (24) becomes

$$2v_2^2 - 2\,v_2 b_2 \cot\phi - \left(\frac{v_2^2}{9} - \tfrac{2}{3}\,v_2 b_2 \cot\theta + b_2^2 \cot^2\theta\right)$$

$$-\tfrac{1}{8}b_2^2 \operatorname{cosec}^2\phi - \left[b_2^2 + (v_2 - 1{\cdot}82\,b_2)^2\right] - \frac{2{\cdot}82^2}{8} b_2^2 - \frac{b_2^2}{16} = 2g\text{H}$$

$$\tfrac{8}{9}\,v_2^2 + 6{\cdot}043\,v_2 b_2 - 5{\cdot}983\,b_2^2 - 2\,g\,\text{H} = o\,;$$

$$\therefore\ v_2^2 + 6{\cdot}80\,v_2 b_2 - 6{\cdot}73\,b_2^2 - \tfrac{9}{4}\,g\,\text{H} = o,$$

and thus

$$m = \frac{v_2}{\sqrt{g\text{H}}} = \frac{1}{\sqrt{\text{N}_m}} = \frac{-6{\cdot}80\dfrac{b_2}{\sqrt{g\text{H}}} + \sqrt{73{\cdot}1\dfrac{b_2}{g\text{H}} + 9}}{2} \qquad (45)$$

from which Table 4 is calculated.

TABLE 4.—VARIATION OF MANOMETRIC AND AIR EFFICIENCY WITH $b_2 \div \sqrt{g\text{H}}$ AND $v_2 \div \sqrt{g\text{H}}$

$\dfrac{b_2}{\sqrt{g\text{H}}}$.	$\dfrac{v_2}{\sqrt{g\text{H}}}$.	N_m.	η.
0	1·500	0·44	0
0·2	1·045	0·91	0·77
0·4	0 915	1·20	0·83
0·5	0·911	1·21	0 78
0·8	1 015	0·97	0·54
1·0	1·130	0·78	0·41
2·0	1·880	0·28	0·137

Hence in this type of fan there is a considerable advantage in making the vanes curve forward at discharge—that is, in making ϕ equal to or greater than 90 deg. It frequently happens that the orifice at which a fan has to work, during the time it is in use, is not constant; for example, owing to the enlargement of a mine, more air may be required at the same gauge. Now, with vanes having ϕ greater than 90 deg., the manometric efficiency between the orifice at which $b_2 = 0{\cdot}4\sqrt{g\,\mathrm{H}}$ and that at which $b_2 = 0{\cdot}8\sqrt{g\,\mathrm{H}}$ is more nearly constant than when $\phi = 30$ deg. Further, this type of fan must have an efficient chimney, in which the gain of pressure head is equal to the loss of velocity head. It will be seen later that this can only be obtained with a uniform flow and a proper inclination of the sides of the chimney.

If the fan has a diffuser, its efficiency at a given orifice will be practically the same when $\phi = 135$ deg. as when $\phi = 30$ deg., because the principal cause of difference is the loss of head in the volute, due to friction; the effect of the diffuser being to reduce the loss of head, owing to the reduced speed in the volute. Where there is not space to fit a good chimney (*e.g.*, on board ship), then a fan must be used in which $c_v = \frac{1}{2}a_2$, and $\phi = 30$ deg., so that a high efficiency may be obtained at a constant orifice; the vane angle at discharge might then be about 15 deg.

The reader may possibly imagine that Tables 3 and 4 might be altered by a change in the ratio $b_1 \div b_2$. This, however, is not the case, because $b_1 \cot\theta = v_1$ when $b_2 = 0{\cdot}5\sqrt{g\,\mathrm{H}}$; hence the term $b_1 \cot\theta$ will be unchanged whatever $b_1 \div b_2$ may be. In the above we have assumed that the external radius r_2 of the wheel is three times the internal r_1, and a change in this will slightly affect the coefficients of $v_2\,b_2$, b_2^2, and $g\,\mathrm{H}$. But whatever the ratio $r_2 \div r_1$ may be, the real manometric efficiency at zero orifice should not differ much from $\frac{1}{2}$, because the air within the eye is really rotating with the same angular velocity as that within the fan, and under these circumstances $v_2 = \sqrt{2\,g\,\mathrm{H}}$. When $b_2 = 0{\cdot}5\sqrt{g\,\mathrm{H}}$ the mechanical and manometric efficiencies are entirely independent of $r_2 \div r_1$, because $v_1 - b_1 \cot\theta = o$.

Volumetric Efficiency.—If a fan were made very large in proportion to its discharge, it would give a high mechanical efficiency, owing to the fact that the loss due to the friction of air in the passages would be extremely small; but the cost of such a fan would probably more than outweigh the advantage thus obtained. It is, therefore, necessary for designers to have another ratio, which is usually called the volumetric efficiency, which compares the discharge with the tip speed v_2 and the dimensions of the fan. Let Q be the discharge, v_2 the tip speed, and r_2 the external radius of the wheel; then, if the fan has one eye, the volumetric efficiency is

$$N_v = \frac{Q}{v_2 r_2^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

and if it has two eyes—*i.e.*, one on each side—

$$N_v = \frac{Q}{2 v_2 r_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (46a)$$

A radial or mixed flow fan having a small diameter and moderate breadth at outflow may have a higher volumetric efficiency than one which has a large diameter and small breadth, although the circumferential area through which the air flows may be the same in both, and they may be so designed that $Q \div v_2 r_2^2$ may be the same for both. For this reason we consider that a better measure of the volumetric efficiency is

$$N_{v1} = \frac{Q}{v_2 r_2 s_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

where s_2 is the breadth of the wheel internally at the external radius.

In the following pages we shall mean by volumetric efficiency the quantity N_v, and shall denote the more correct value by the letter N_{v1}. These efficiencies can be greater than unity.

CHAPTER V.

Design of Radial Flow Fans.—Fans may be divided into three classes, viz : *Radial Flow Fans* - those in which the direction of flow through the wheel is perpendicular to the axis.

Mixed Flow Fans—those which have an axial inflow and radial outflow.

Screw or Propeller Fans—those in which the flow is wholly axial.

We shall now consider the design of the Radial Flow Fan.

We shall first consider the value $\frac{b_2}{\sqrt{gH}}$.

Heenan and Gilbert[5] found that with a fan 17 in. diameter and 8 in. broad the discharge was about 3,400 cu. ft. of air per min. at the most suitable orifice; the water gauge was 9·3 in., and consequently

$$b_2 = \frac{Q}{2\pi r_2 s_2} = \frac{3400 \times 144}{\pi \times 17 \times 8 \times 60},$$

and

$$gH = \frac{32{\cdot}2 \times 9{\cdot}3}{12} \times 820,$$

$\frac{1}{820}$ being taken as the mean relative density of the air at an average temperature and pressure;

hence

$$\frac{b_2}{\sqrt{gH}} = 0{\cdot}134.$$

In Bryan Donkin's paper[6] a Rateau fan with an external radius of 9·8 in, has its maximum efficiency at an equivalent orifice of 0·4 sq. ft. The equivalent orifice

$$O_1 = \frac{Q}{0{\cdot}65\sqrt{2gH}} = 1{\cdot}088\frac{Q}{\sqrt{gH}} \quad . \quad . \quad (48)$$

Let r_0 be the radius of the eye, then

$$2\pi r_0^2 \times 33 \cdot 2 = 1000 = Q$$

$$r_0^2 = \frac{500}{33 \cdot 2\,\pi}$$

$$r_0 = 2 \cdot 19 \text{ ft.}$$

The internal radius of the vanes should be made slightly larger than this, say $2\frac{1}{2}$ ft. Let the velocity of inflow be $0 \cdot 5\ \sqrt{g\,H}$. The thickness of the vanes may be neglected, and we have

$$2\pi r_1 s_1 \times 0 \cdot 5 \sqrt{g\,H} = 1000;$$

where s_1 is the effective breadth of the wheel at radius r_1

$$s_1 = 1 \cdot 92 \text{ ft.}$$

If the radial component at outflow $b_2 = 0 \cdot 5 \sqrt{g\,H}$, and the vanes are made to touch the circumference at the outer radius, then we have already shown that $\eta = 0 \cdot 645$ and $v_2 = 2 \cdot 49 \sqrt{g\,H} = 165$ ft. per sec. If the revolutions N per min. are fixed, we can calculate r_2 from the formula

$$2\pi r_2 N = 60\ v_2,$$

but if not, let $r_2 = 2\ r_1 = 5$ ft.; then

$$N = \frac{60 \times 165}{10\,\pi} \doteq 315 \text{ rev. per min.,}$$

and the effective breadth at the outer radius

$$s_2 = s_1 \frac{r_1}{r_2} = 0 \cdot 96 \text{ ft.}$$

It must be remembered that if b_2 does not $= 0 \cdot 5 \sqrt{g\,H}$, another value of v_2 must be deduced from equation (21). In the case under consideration,

$$v_1 = v_2 \frac{r_1}{r_2} = 1 \cdot 245 \sqrt{g\,H};$$

$$\therefore \quad \cot\theta = \frac{v_1}{b_1} = \frac{1 \cdot 245}{0 \cdot 5} = 2 \cdot 49.$$

The brake horse power required to drive the fan is

$$\text{B.H.P.} = \frac{62 \cdot 3\, Q\, h}{12 \times 550 \times \eta} = \frac{62300 \times 2}{12 \times 550 \times 0\ 645} = 29 \cdot 3,$$

so that the indicated horse power of the engine is

$$\text{I.H.P.} = \frac{\text{B.H.P.}}{0 \cdot 85} = 34 \cdot 5.$$

Next suppose our fan is provided with a diffuser; the vane should be a tangent to the outer circumference, so that, allowing for the divergence of the air, $\phi = 15$ deg., $v_2 = 2 \cdot 46 \sqrt{g\, \mathrm{H}}$, N = 312, and

$$\cot \theta = \frac{v_1}{b_1} = \frac{\frac{1}{2} \times 2 \cdot 46 \sqrt{g\, \mathrm{H}}}{\frac{1}{2} \sqrt{g\, \mathrm{H}}} = 2 \cdot 46.$$

The diffuser may be made with its sides diverging from one another about 7 deg., corresponding to an inclination to the vertical of about $\frac{1}{16}$, and its external radius is

$$r_3 = 1\tfrac{1}{2}\, r_2 = 7\tfrac{1}{2} \text{ ft.}$$

The diffuser may either rotate with the wheel or form part of the casing; the sides of the wheel should appear concave on a radial section, so that the air may have no motion parallel to the axis when discharged from the wheel into the diffuser. The breadth of the diffuser at the external radius is

$$s_3 = s_2 + \tfrac{1}{8}\,(r_3 - r_2) = 1 \cdot 27 \text{ ft.}$$

The brake horse power is now

$$\text{B.H.P.} = \frac{62 \cdot 3 \times 1000 \times 2}{12 \times 550 \times 0 \cdot 685} = \frac{18 \cdot 88}{0\ 685} = 27 \cdot 6,$$

and $$\text{I.H.P.} = \frac{\text{B.H.P.}}{0 \cdot 85} = 32 \cdot 5.$$

If the fan has no diffuser, or chimney, but a volute in which $c_v = \frac{1}{2}\, a_2$, then, as the highest efficiency is obtained with

$\phi = 30$ deg., the vane angle at discharge should be 15 deg., $v_2 = 1{\cdot}67\sqrt{g\,\mathrm{H}}$, N = 212 rev. per min., and cot $\theta = 1{\cdot}67$, B.H.P. $= \frac{18{\cdot}88}{0{\cdot}74} = 25{\cdot}5$, I.H.P. $= 30$, $a_2 = v_2 - b_v \cot\phi =$ $\left(1{\cdot}67 - \frac{1{\cdot}732}{2}\right)\sqrt{g\mathrm{H}} = 0{\cdot}81\sqrt{g\mathrm{H}} = 53{\cdot}7$. The section of the volute is proportional to the angu'ar distance from its beak E, fig. 8, because the discharge from the fan is uniform all round the circumference, and if ψ is this angular distance in radians, s the section, and S is its section at discharge, then

$$s = \frac{\psi}{2\pi}\mathrm{S}$$

and $$\mathrm{S} = \frac{\mathrm{Q}}{c_v} = \frac{2\,\mathrm{Q}}{a_2} = \frac{2000}{53{\cdot}7} = 37{\cdot}3 \text{ sq. ft.},$$

corresponding to a diameter of 6·9 ft. The section, however, need not be circular; a rectangular form is frequently used. In some cases the wheel discharges into the centre of this section; in others, the volute is wholly outside the wheel. If the fan has a chimney, then $a_2 = c_v$, but in this case it is possible to obtain a high manometric efficiency. We shall consider the design in two cases: firstly, when $\phi = 90$ deg.; secondly, when it is 120 deg. In the former case, we believe that it is necessary to make the vane angle about 105 deg., so that the average flow may be radial, if the number of vanes are few, say six; but if twelve vanes are used, then the vanes may be radial at discharge. In either case, we may assume $v_2 = a_2 = 1{\cdot}11\sqrt{g\,\mathrm{H}}$, N = 140 rev. per min. The section of the volute at discharge is

$$\mathrm{S} = \frac{\mathrm{Q}}{a_2} = \frac{1000}{73{\cdot}6} = 13{\cdot}6 \text{ sq. ft.},$$

while the larger end of the chimney has a section

$$\mathrm{S}_1 = \frac{\mathrm{Q}}{c_d} = \frac{1000}{\frac{1}{8}\sqrt{g\,\mathrm{H}}} = 124 \text{ sq. ft.}$$

$$\text{B.H.P.} = \frac{18{\cdot}88}{0{\cdot}81} = 23{\cdot}3$$

and $$\text{I.H.P.} = \frac{\text{B.H.P.}}{0\cdot85} = 27\cdot4$$

$$\cot\theta = 1\cdot11.$$

If $\phi = 120$ deg. the vane angle should be 135 deg. if there are six vanes, and 120 deg. if there are twelve; then $v_2 = 0\cdot98\sqrt{g\text{H}}$, $\text{N} = 124$, and $\cot\theta = 0\cdot98$, $\text{B.H.P.} = \frac{18\cdot88}{0\cdot80} = 23\cdot6$, $\text{I.H.P.} = \frac{\text{B.H.P.}}{0\cdot85} = 27\cdot8$, $a_2 = \left(0\cdot98 + \frac{0\cdot578}{2}\right)$ $\sqrt{g\text{H}} = 1\cdot27\sqrt{g\text{H}}$. $\text{S} = \frac{\text{Q}}{a_2} = \frac{1000}{84\cdot3} = 11\cdot9$ sq. ft. $\text{S}_1 = 124$ sq. ft. as before.

We now come to the case where there is a diffuser, volute, and chimney, and shall assume $\phi = 120$ deg., making the vane angle 135 deg. if there are six vanes, and 120 deg. if there are twelve. The external radius r_3 of the diffuser is to be $1\frac{1}{4}$ that of the wheel, and therefore

$$r_2 = 9\tfrac{1}{4}\text{ ft.}$$

$v_2 = 0\cdot94\sqrt{g\text{H}}$, $\text{N} = 119$, $\cot\theta = 0\cdot94$, $\text{B.H.P.} = \frac{18\cdot88}{0\cdot87} = 21\cdot7$, $\text{I.H.P.} = \frac{21\cdot7}{0\cdot87} = 25\cdot5$. The breadth of the diffuser is as before $1\cdot11$ ft., and since

$$a_2 = v_2 - b_2\cot\phi = (0\cdot94 + 0\cdot288)\sqrt{g\text{H}} = 1\cdot228\sqrt{g\text{H}}.$$

$$a_3 = \frac{r_2}{r_3}a_2 = \tfrac{4}{5} \times 1\cdot228\sqrt{g\text{H}} = 68\cdot6.$$

Hence $\text{S} = \frac{1000}{68\cdot6} = 14\cdot5$ sq. ft., and the large end of the chimney 124 sq. ft. as before.

It is the practice of some fan-makers to make the sides of the wheel parallel, because no doubt it is very much cheaper; there must be, however, a considerable loss of head at inflow owing to eddy formation. On the other hand, there is less loss of head at outflow from the wheel owing to $b_2^2 \div 2g$ being very much less than $b_1^2 \div 2g$. Anyone who has read the above designs will have no difficulty in calculating the dimensions of such a fan; but to be strictly accurate he must not assume any of the tabulated values of $\frac{v_2}{\sqrt{g\,H}}$; but must find them from equations (21) to (25), substituting the correct value of $\frac{v_2}{\sqrt{g\,H}}$, and in these equations a term $F_3 \frac{b_1^2}{2g}$ should be added to the left hand to represent the additional loss of head at inflow and in the wheel.

Design of Mixed-flow Fans.—In the previous designs we have supposed that inflow takes place from both sides, and when a fan is forcing air into closed passages there is no reason why this should not be the case; but when the air is being drawn from passages it is more convenient to erect the fan with inflow from one side, when the area of the eye must be made twice, and consequently the radius 1·414 times, as great as that of a fan taking air from both sides. The ratio of the external to the internal radius of the wheel cannot be rigidly fixed. The vanes seize upon the

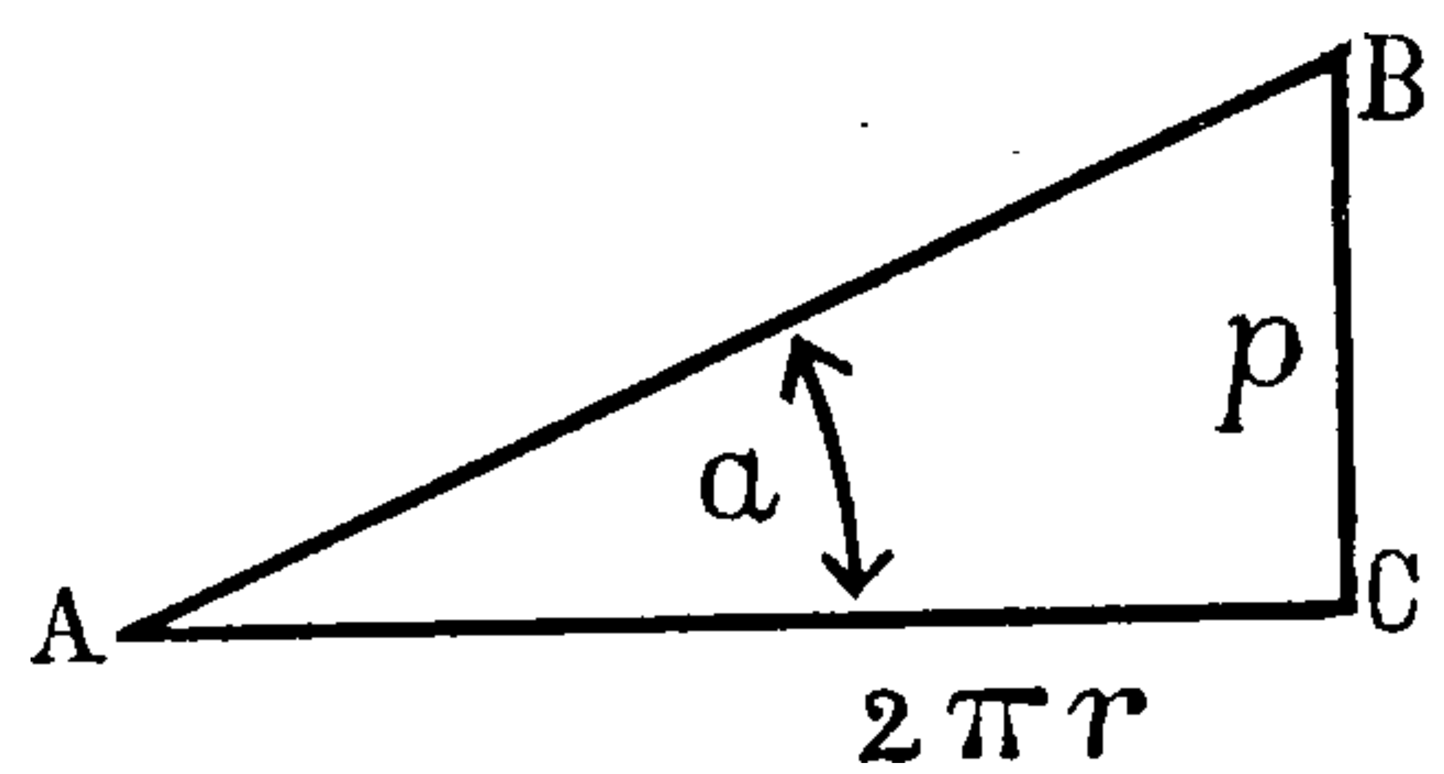


FIG. 11.—DEVELOPMENT OF A SCREW.

air immediately it enters the eye of the fan, although there is a mouthpiece before the eye whose object is to make the velocity of inflow as uniform as possible over the whole area. The angle θ must therefore vary with the radius, because

$$\cot\theta = \frac{v_1}{b_1} = \frac{r\,\omega}{b_1},$$

where ω = angular velocity of wheel in radians per sec. $= \frac{2\,\pi}{60} \times N$, N being the number of rev. per min.

But r varies from zero at the centre to r_1 at the outer radius of the eye. Now, if a spiral of pitch p, fig. 11, and radius $2 \pi r$, be traced round a cylinder and the cylinder developed, one turn of the spiral will give us the line A B, whose inclination a to a line perpendicular to the axis gives us

$$\cot a = \frac{2 \pi r}{p}.$$

Now, if a number of these spirals are traced with different values of r, it is clear that the cotangent of the angle increases as r, and the surface generated by joining all these spirals by lines perpendicular to the axis is called a helical or screw surface. A straight line which passes through and is perpendicular to the axis, and which rises a distance p with uniform velocity while it makes a complete turn with uniform velocity, will also trace a helical surface. A true helix is usually generated in this way, but a surface traced by a curve, whose plane contains the axis, and whose outer end is on a cylinder having this axis, this end tracing a spiral on the cylinder, will also trace a surface, any cylindrical section of which will give a spiral in which

$$\cot a = \frac{2 \pi r}{p},$$

and the end of each vane of a mixed-flow fan must be a surface of this description. It should then guide the air round, so that it may flow in planes perpendicular to the axis of the fan.

As the Rateau fan is the best illustration of a mixed-flow fan with which we are acquainted, we shall now describe it and give its proportions (figs. 12, 13). The fan centre consists of a east-iron wheel A, upon which the vanes are fixed. A is formed by the revolution of the arc of a circle about the axis. At the eye it is conical, and at the periphery it is normal to the axis. In small fans the vanes are placed in the mould when A is cast, and in larger sizes they are held to it by angle irons. The number of vanes is 20 to 24 for small fans, and 24 to 30 for large. The

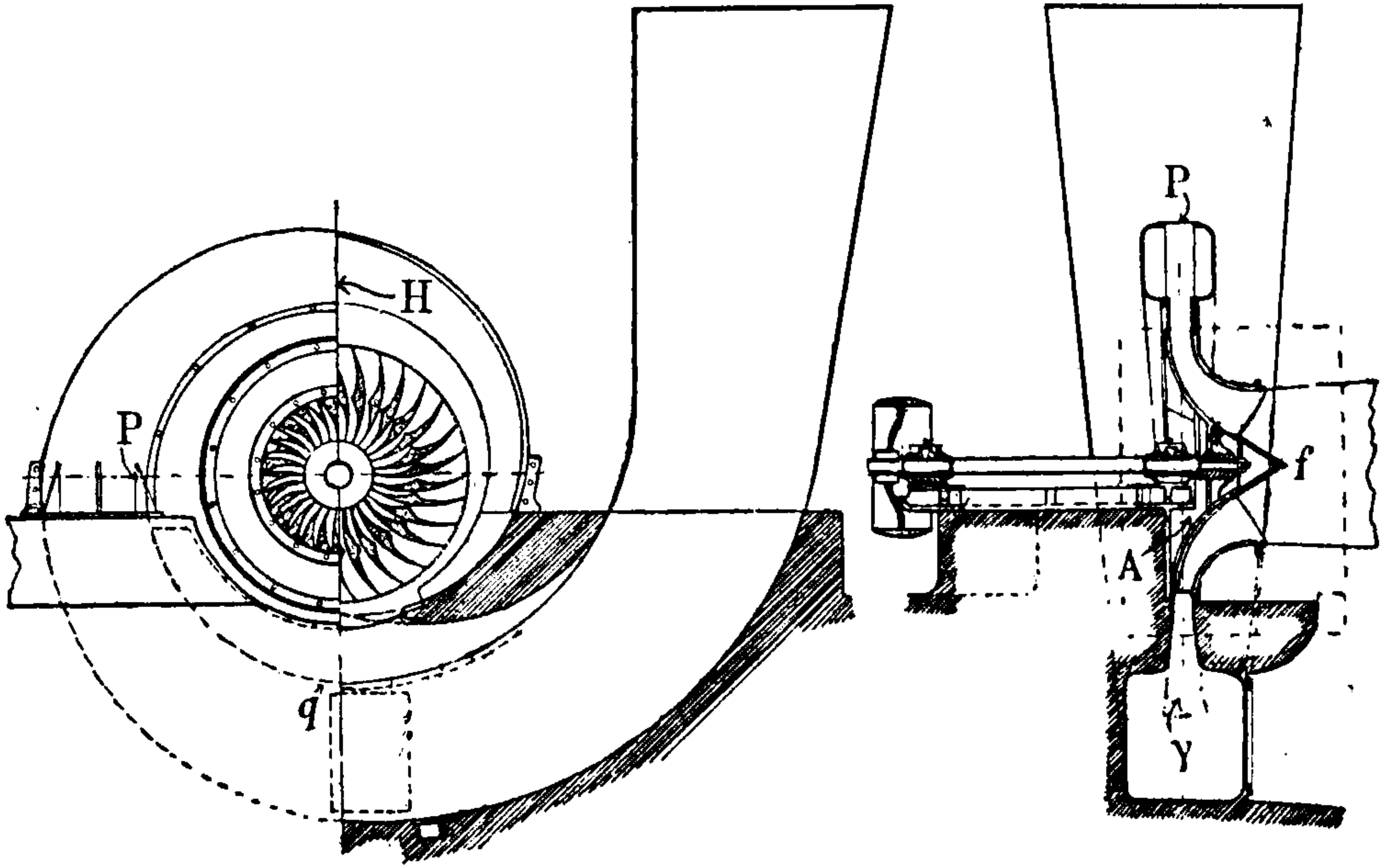


FIG. 12.—RATEAU FAN.

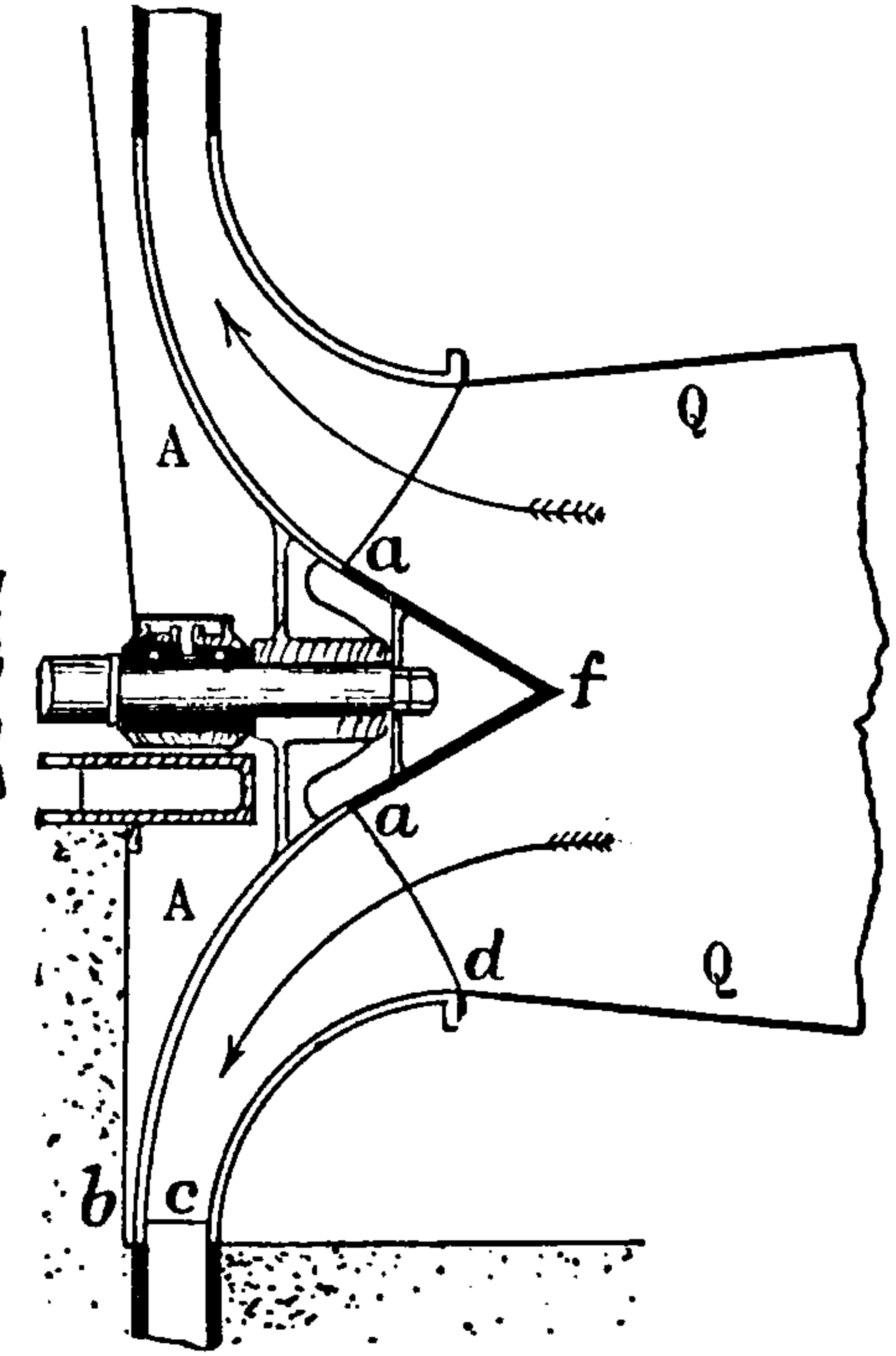


FIG. 13.—ENLARGED SECTION.

The vanes are of wrought iron, and are stamped in a mould. They are very rigid in consequence of their curvature in every direction, which is such that the trajectory of every particle of fluid has an almost constant curvature. This is of great importance in lessening eddies.

Fig. 15 shows the geometrical construction of the vanes: K O is the axis of the fan shaft, and circles passing through S, P, Q, R have radii equal to the external radius of the fan, while P R is the breadth of the vane at the tip. Take a point B on the circle S, and through it draw arcs B e,

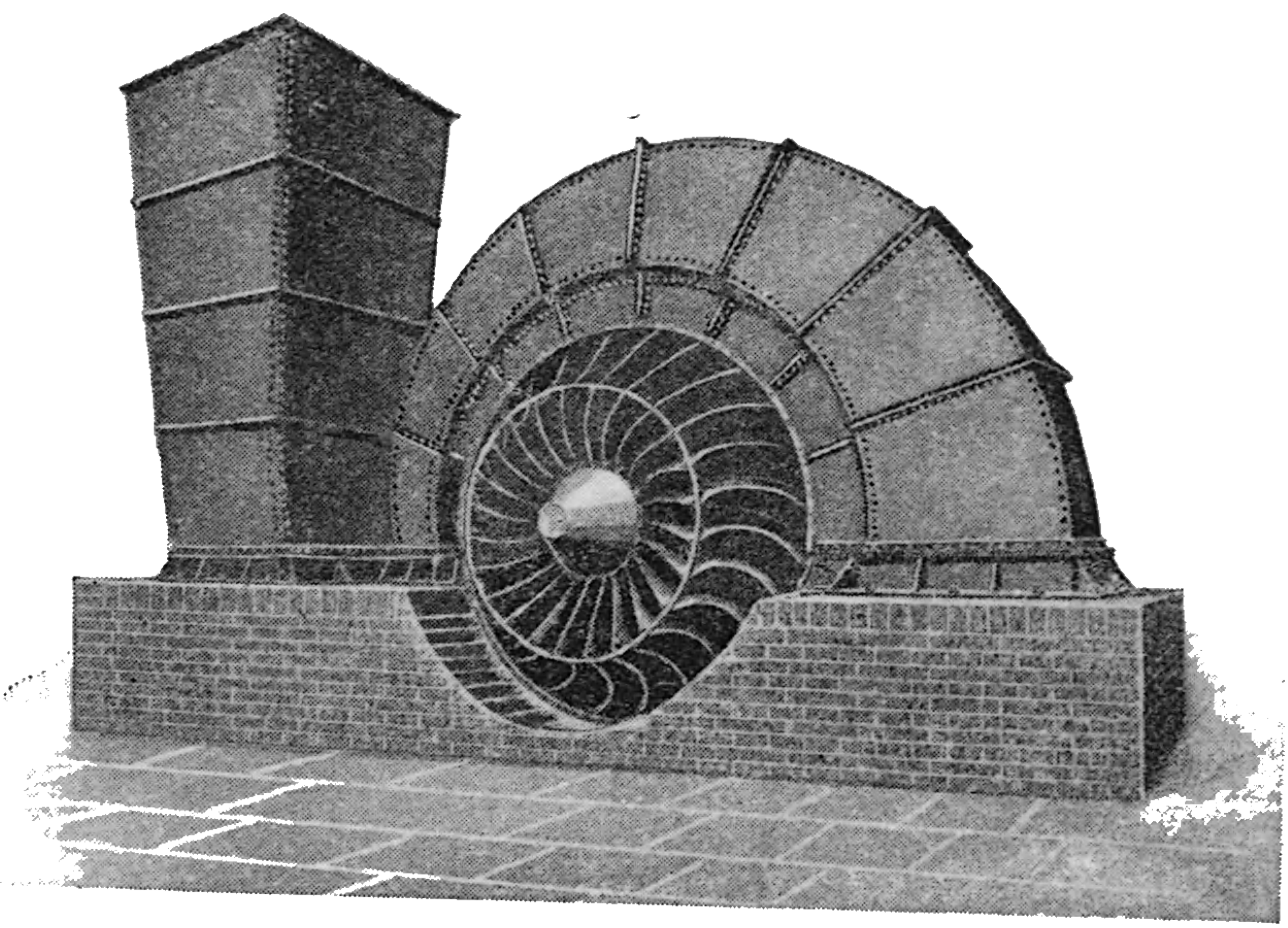


FIG. 14.—PERSPECTIVE VIEW OF RATEAU FAN.

B C; the angle made by the arc B e with the tangent at B is 45 deg. for a fan of type A (see Table 5), which is generally the angle made by the vane at the external radius r_2 with the tangent. The rotation in the figure is supposed to take place from B to the right, so that if vanes curved backwards are used the centre of B e would lie on the opposite side. The circular arc B C makes an angle C B D with the line B D parallel to the axis, such that, if b_1 is the intended velocity of the air at normal speed, and v_2 is the peripheral speed at the extreme radius, then tan

C B D = v_2/b_1, and this will ensure that at any point on the arc B *e* the air will meet the edge of the vane when the fan is running at the orifice for which it is designed. In tracing out the surface of the vane, the arc B *e* moves with B on the arc B C, and *e* on the cylindrical surface, with

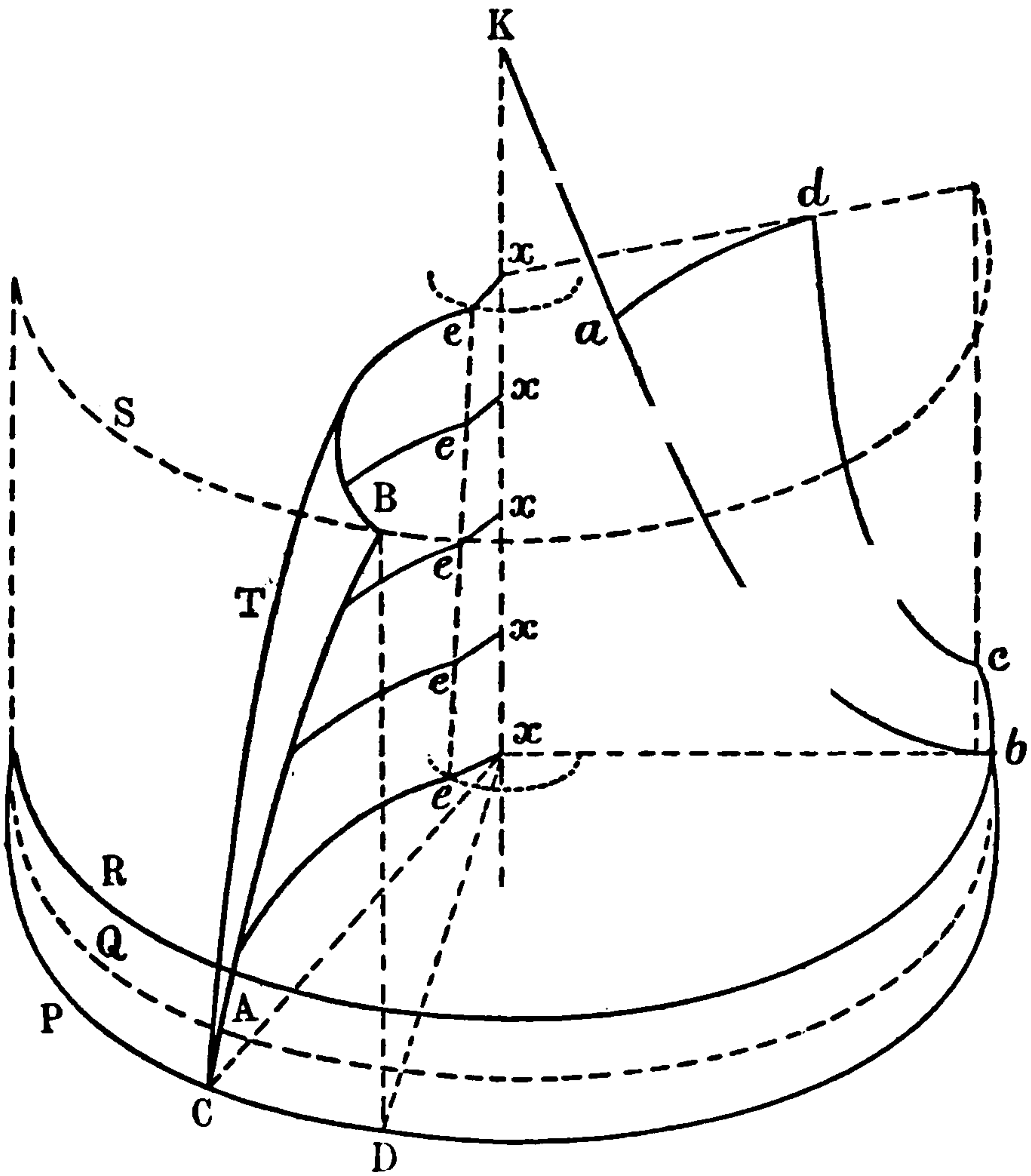


FIG. 15.—GEOMETRICAL CONSTRUCTION OF VANES.

e x as radius and K O as axis; the whole of this surface is, however, not required for the vane. Imagine *abcd* to sweep round the axis K O and to cut out a portion; this forms the vane, *a d* being the part near the eye and *b c* that near the periphery, while K a forms the cone at the eye. This construction may therefore be used for any type of mixed-flow vane, whatever the angles θ and ϕ have to be. In Table 5 the proportions of these fans are given, and in fig. 14 is shown a perspective view of a Rateau fan, with part of the casing removed to show the vanes.

To design a fan of this type for a water gauge of 8 in. and 180,000 cu. ft. per mm., we shall assume that the velocities of flow b_1, b_2 are $0{\cdot}5\sqrt{g\,\mathrm{H}}$—thus,

$$b_2 = b_1 = 0{\cdot}5\sqrt{32{\cdot}2 \times \tfrac{8}{12} \times 820} = 66{\cdot}3 \text{ ft. per sec.,}$$

so that the area O of the suction orifice (Table 1)

$$\mathrm{O} = r_2{}^2 = \frac{\mathrm{Q}}{b_1} = \frac{3000}{66{\cdot}3} = 45{\cdot}2 \text{ sq. ft.,}$$

$r_2 = 6{\cdot}73$ ft., $r_1 = 0{\cdot}6\ r_2 = 4{\cdot}04$ ft., and the radius of cone $p = 1{\cdot}68$, so that the external diameter of the fan is 13·46 ft. The vanes are curved forwards, so that $\phi = 135$ deg., and 12 vanes are used, so that we may assume the angle of flow practically coincides with the angle of the vane.

What is the best number of vanes for a given fan is a question that we are unable to answer, as no experiments have been published upon this subject, and designers differ widely on the point. Up to a certain limit, the greater the number of vanes the higher the water gauge, but a point must be reached at which additional surface friction not only reduces the efficiency, but reduces the gauge.

It will be noticed that the width of the vane at outflow, which we denote by s_2, is $0{\cdot}16\ r_2$, so that the cylindrical surface through which the air leaves the fan at radius r_2 is $2\pi \times 0{\cdot}16\, r_2{}^2$, which is very little greater than $r_2{}^2$. The final depth of the diffuser spiral D is 3·365 ft., so that the radius at that point is 10·09 ft. $= r_3$, and as the tip-speed of the wheel, from (38),

$$v_2 = 0{\cdot}87\sqrt{g\,\mathrm{H}}, \quad \mathrm{N} = 165 \text{ rev. per min.}$$

$$\therefore\ a_2 = (0{\cdot}87 + \tfrac{1}{2})\sqrt{g\,\mathrm{H}} = 182 \text{ ft. per sec.,}$$

$$a_3 = a_2\frac{r_2}{r_3} = \frac{182}{1{\cdot}5} = 121 \text{ ft. per sec.,}$$

TABLE 5.—PROPORTIONS OF RATEAU FANS.

(*The radius of the fan is denoted by r_2.*)

		Types. A. For large volumes and small pressures (see figs. 5, 5a).	B. For moderate volumes and pressures.	C. For small volumes and high pressures.
SUCTION ORIFICE—				
External radius ...	r	$0{\cdot}6\ r_2$	$0{\cdot}5\ r_2$	$0{\cdot}4\ r_2$
Radius of cone	ρ	$0{\cdot}25\ r_2$	$0{\cdot}25\ r_2$	$0{\cdot}25\ r_2$
Area of suction orifice ..	O	r_2^2	$0{\cdot}6\ r_2^2$	$0{\cdot}35\ r_2^2$
VANES—				
Initial inclination at the circumference of the suction orifice with the axis	β	45°	45°	45°
Final inclination at the circumference of the fan with the rods ...	a	45°	60°	60°
DEPTH OF VANE—				
At inflow	s_1	$0{\cdot}40\ r_2$	$0{\cdot}28\ r_2$	$0{\cdot}19\ r_2$
At outflow	s_2	$0{\cdot}16\ r_2$	$0{\cdot}08\ r_2$	$0{\cdot}054\ r_2$
DIFFUSER—				
Width	s_3	$0{\cdot}175\ r_2$	$0{\cdot}09\ r_2$	$0\ 05\ r_2$
DEPTH OF SPIRAL—				
Initial	d	$0{\cdot}10\ r_2$	$0{\cdot}10\ r_2$	$0{\cdot}10\ r_2$
Final	D	$0{\cdot}50\ r_2$	$0{\cdot}50\ r_2$	$0{\cdot}50\ r_2$
Inclination of one face to another	γ	$4° + 3°$	$4° + 3°$	$3° + 2°$
Approximate length of the spiral*	λ	$9{\cdot}2\ r_2$	$9{\cdot}2\ r_2$	$9{\cdot}2\ r_2$
VOLUTE—				
Formula giving the sections as functions of the arc of the spiral x † ...	s	$0{\cdot}5\,x(1+0\ 8x)\mathrm{O}$	$0{\cdot}45\,x(1+x)\ \mathrm{O}$	$0{\cdot}40\,x(1+1{\cdot}2x)\mathrm{O}$
CHIMNEY—				
Height ‡	L	5 to 7 r_2	5 to 7 r_2	5 to 7 r_2
Angle of chimney ...	θ	7°	6°	5°

* The end of the spiral is at a distance r_2 from the point which is on a radius passing through the origin of the spiral.

† x is the ratio of the length of the arc of the spiral from the origin to the point at which the section is to be calculated to the total length of the spiral λ.

x may be calculated as a function of the angle ω between two radii passing through the origin of the spiral, and the point of the spiral where the section is to be calculated by the following formula: $x=\frac{\omega}{480}\left(1+\frac{\omega}{1980}\right)$, the angle ω being expressed in degrees.

‡ Calculated from the end of the spiral.

and, according to our theory, this is the velocity in the volute for a fan with a very long chimney which will reduce the velocity to a low value. Hence, following our theory, the section of the volute at discharge

$$S = \frac{3000}{121} = 24{\cdot}8 \text{ sq. ft.};$$

while, according to Table 5,

$$s = 0{\cdot}5\, x\,(1 + 0{\cdot}8\, x)\, r_2^2,$$

and $x = 1$ when $s = S = 0{\cdot}5 \times 1{\cdot}8 \times 45{\cdot}3 = 40{\cdot}7$ sq. ft. According to our theory the section of the chimney would be

$$S_1 = \frac{Q}{\frac{1}{8}\sqrt{g\,H}} = \frac{3000}{16{\cdot}6} = 180 \text{ sq. ft.},$$

and, assuming the section of volute and chimney square, the chimney would require to be long enough to alter the side of the square from 4·98 ft. to 13·42 ft., which, with a total taper of 7 deg., would require a chimney whose height was

$$L = \frac{13{\cdot}42 - 4{\cdot}98}{\tan 7^\circ} = \frac{8{\cdot}44}{0{\cdot}1228} = 69 \text{ ft., nearly}$$

$$= 10{\cdot}2\, r_2,$$

instead of 5 to 7 times r_2 (Table 5). In this case the brake horse power

$$\text{B.H.P.} = \frac{62{\cdot}3 \times 8 \times 180000}{12 \times 33000 \times 0{\cdot}84} = 270;$$

$$\text{I.H.P.} = \frac{270}{0{\cdot}85} = 318.$$

If, however, we assume a diffuser in which $r_3 = 1{\cdot}5\, r_2$, and that $c_v = \frac{1}{2}\, a_3$, the chimney will be greatly reduced in height. The values of r_1, r_2 are the same as before, but, from (38*a*),

$$v_2 = 0{\cdot}88\sqrt{g\,H}, \quad N = 167 \text{ rev. per min.}$$

$$a_2 = 1{\cdot}38\sqrt{g\,H} = 183 \text{ ft. per sec.},$$

$$c_v = \tfrac{1}{2}\, a_3 = \tfrac{1}{3}\, a_2 = 0{\cdot}46\sqrt{g\,H} = 61 \text{ ft. per sec.},$$

hence the section of the volute at discharge

$$=\frac{3000}{61}=49\cdot2 \text{ sq. ft.},$$

which would form a square whose side is 7·01 ft., so that the length of the chimney would require to be

$$L=\frac{13\cdot42-7\cdot01}{\tan 7^\circ}=\frac{6\cdot4}{0\cdot1228}=52\cdot2 \text{ ft.}$$

$$=7\cdot8\,r_2.$$

Here $\text{B.H.P.}=\dfrac{180000\times8\times62\cdot3}{33000\times12\times0\cdot82}=277,\quad \text{I.H.P.}=327.$

In the first case, the angle θ that the vanes at the outer radius of the eye make with a tangent to the direction of motion—*i.e.*, with a line perpendicular to both radius and axis—is given by

$$\cot\theta=\frac{v_1}{b_1}=\frac{0\cdot6\,v_2}{0\cdot5\sqrt{g\mathrm{H}}}=\frac{0\cdot6\times0\cdot87}{0\cdot5},$$

$$=1\cdot04,$$

and in the second,

$$\cot\theta=\frac{0\cdot6\times0\cdot88}{0\cdot5}=1\cdot06,$$

so that θ is as nearly as possible 45 deg.

Theory of the Spiral Diffuser and Volute.—Let r_3 be the radius of the diffuser at an angle ψ_d from its commencement, and let

$$r_3=r_2+p\,\psi_d+q,$$

where p and q are constants.

Then $$\frac{a_3}{a_2}=\frac{r_2}{r_2+c\,\theta+a},$$

Since the moment of momentum of a particle of air in its passage through the diffuser is constant—that is, the velocity at any point varies inversely as the radius, so that the curve of flow is an equiangular spiral.

Let A, fig. 16, be the point where the particle leaves the wheel, and let the radius r_2 at this point make an angle ψ_w with the radius passing through the point of commencement of the diffuser. Further, let B be the point where the particle enters the volute, so that A B is an equiangular spiral. Let A B subtend an angle a at C; then if r is the radius at any point X on A B, and $r+\delta r$ the radius after the particle has moved a distance corresponding to δs along a circular arc, subtending δa at C, it follows that—

$$\delta s = r\,\delta a,$$

$$\text{and} \therefore\ \delta a = \frac{a_2}{b_2}\cdot\frac{\delta r}{r}. \quad \text{Thus,} \int_{\psi_w}^{\psi_d} d\,a = \frac{a_2}{b_2}\int_{r}^{r_3}\frac{d\,r}{r}.$$

$$\therefore\ a = \frac{a_2}{b_2}\log\frac{r_3}{r_2} = \frac{a_2}{b_2}\log\frac{r_2 + p\,\psi_d + q}{r_2}.$$

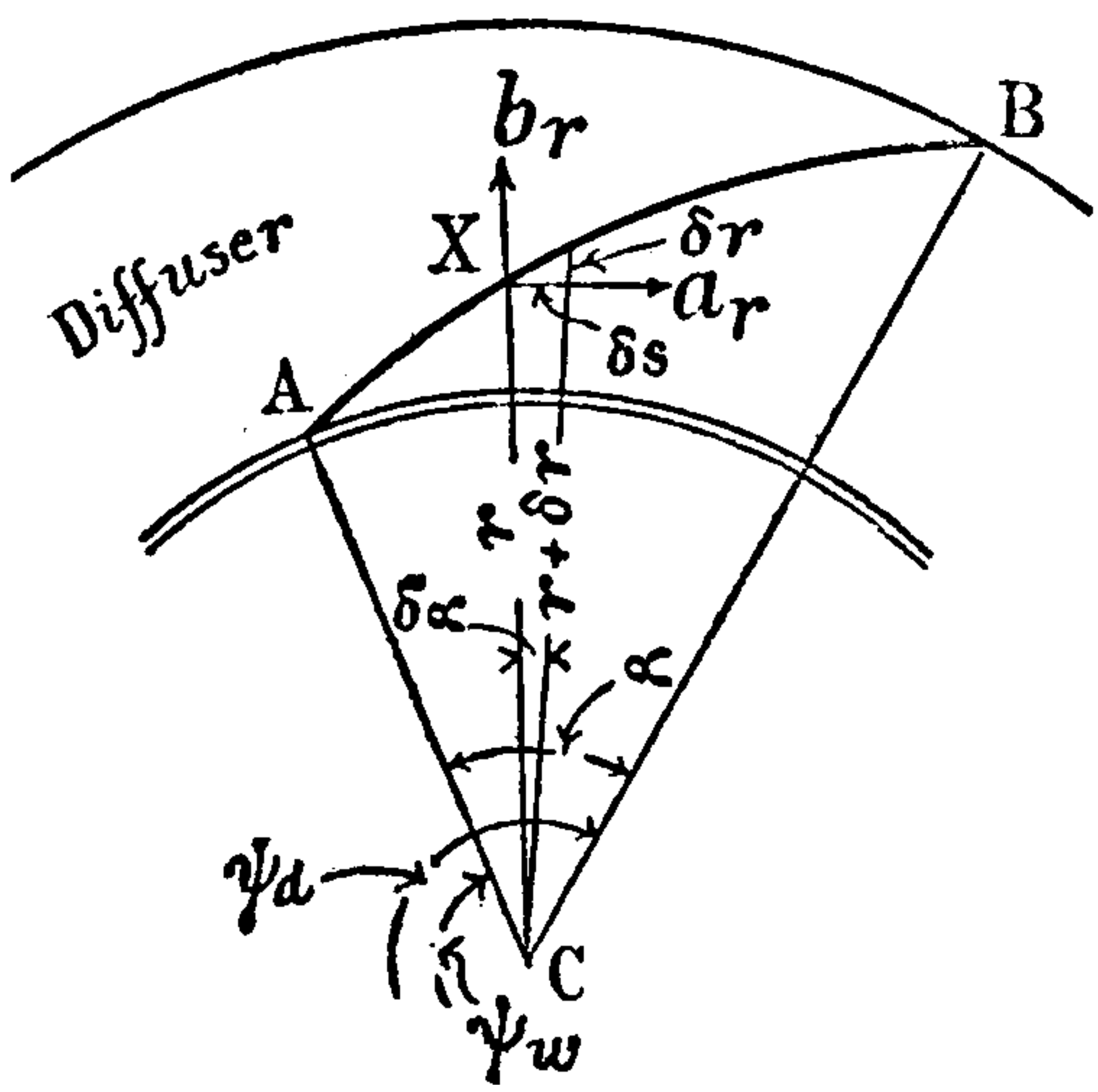


FIG. 16.—VELOCITY IN THE VOLUTE.

Let s be the section of the volute made by a plane passing through the axis and C B. Then the section s has a quantity of air passing through it per sec., represented by the quantity

$$Q\,\frac{\psi_d - a}{2\,\pi},$$

If β is the inclination of the relative direction of flow in the wheel with the direction of motion of the wheel, at the radius r, then the above head

$$=\frac{v^2 - v\,b \cot\beta}{g} - \frac{a^2 + b^2}{2\,g} - \mathrm{H}$$

$$=\frac{1}{2\,g}(v^2 - b^2 \operatorname{cosec}^2\beta - 2\,g\,\mathrm{H}),$$

where b and c are the radial component and absolute velocity of flow at radius r, corresponding to b_2 and c_2 (fig. 7) at radius r_2.

In the above consideration friction is neglected, and inflow is supposed to take place without shock. At the outer radius r_2 the pressure head becomes

$$\frac{1}{2\,g}({v_2}^2 - {b_2}^2 \operatorname{cosec}^2\phi - 2\,g\,\mathrm{H}).$$

The actual pressure in lb. per sq. ft. may be obtained from the head by multiplying it by 0·075. The total pressure on the wheel in feet of air is therefore

$$\int_{r_1}^{r_2} \frac{\pi\,.\,r\,.}{g}(v^2 - b^2 \operatorname{cosec}^2\beta - 2\,g\,\mathrm{H})\,d\,r$$

in radial flow fans, and since v, b, and β all involve r, the total pressure is best found by dividing the wheel into a number of small rings so that $\pi\ r\ d\ r$ can be obtained for each, and

$$\frac{1}{g}(v^2 - b^2 \operatorname{cosec}^2\beta - 2\,g\,\mathrm{H})$$

can be calculated at the middle of the ring whose mean radius is r and thickness $d\ r$. This will give a result which will be practically correct.

The above assumes that the wheel, although there is only one eye through which air enters, has an opening of equal diameter at the other side. It also assumes that the disc of the wheel is on one side so that there is axial thrust; by making the wheel with a disc on each side, end thrust can be entirely avoided, and the end thrust due to suction at the eye comes on the fan casing and not on the wheel. If, however, we are dealing with a mixed-flow fan we must proceed somewhat differently. Our explanation will be clearly understood by reference to fig. 13. We want to find the axial component of the pressures on the cone *a f a* and the curved surfaces *a b*, *a b* (neglecting vane thicknesses).

Let $r_1 = r_0$ be the radius of the eye, then at the point *f* the head is

$$-\mathrm{H} - \frac{1}{2g}\left(\frac{\mathrm{Q}}{\pi r_1^2}\right)^2;$$

but as we get further to the left on the surface of the cone the velocity of the air increases, and the section of passage should be calculated perpendicular to the mean direction of the flow shown by the arrows.

Let A be this section, then the head is

$$-\mathrm{H} - \frac{1}{2g}\left(\frac{\mathrm{Q}}{\mathrm{A}}\right)^2.$$

Further, we may suppose the cone divided up into a number of rings, and a central circle and areas projected on to a plane perpendicular to the axis. The total axial pressure on the wheel is then given by the summation of these areas and their corresponding pressures, which is

$$= \int_{r_p}^{r_2} \frac{\pi r}{g}\left(v^2 - b^2 \operatorname{cosec}^2 \beta - 2g\mathrm{H}\right) d r,$$

where r_p is the radius of the cone, and b cosec β is the velocity of flow of the air relative to the wheel. The latter is most easily estimated by drawing lines perpendicular to the arrow, representing the mean direction of flow, and calculating the areas of the frustra of cones swept out by these. Then, if a is the area swept out by one of these lines,

$$\therefore \quad b = \frac{\mathrm{Q}}{a},$$

and β must be measured from a drawing of the vanes if it cannot be calculated.

This pressure can be balanced by putting straight radial vanes at the back of the wheel; the effect of the vanes is to set the air in motion and thereby reduce the pressure. If there are no vanes at the back, a pressure corresponding to that exerted at the circumference will act over the whole of the back of the fan. We know of no case in which fans have been thus balanced, but show in fig. 17 a sectional elevation of a centrifugal pump balanced in this manner.[7] The holes shown at 0 are unnecessary.

Imagine a space, such as that at the back of the wheel, in which a number of radial vanes cause every particle of

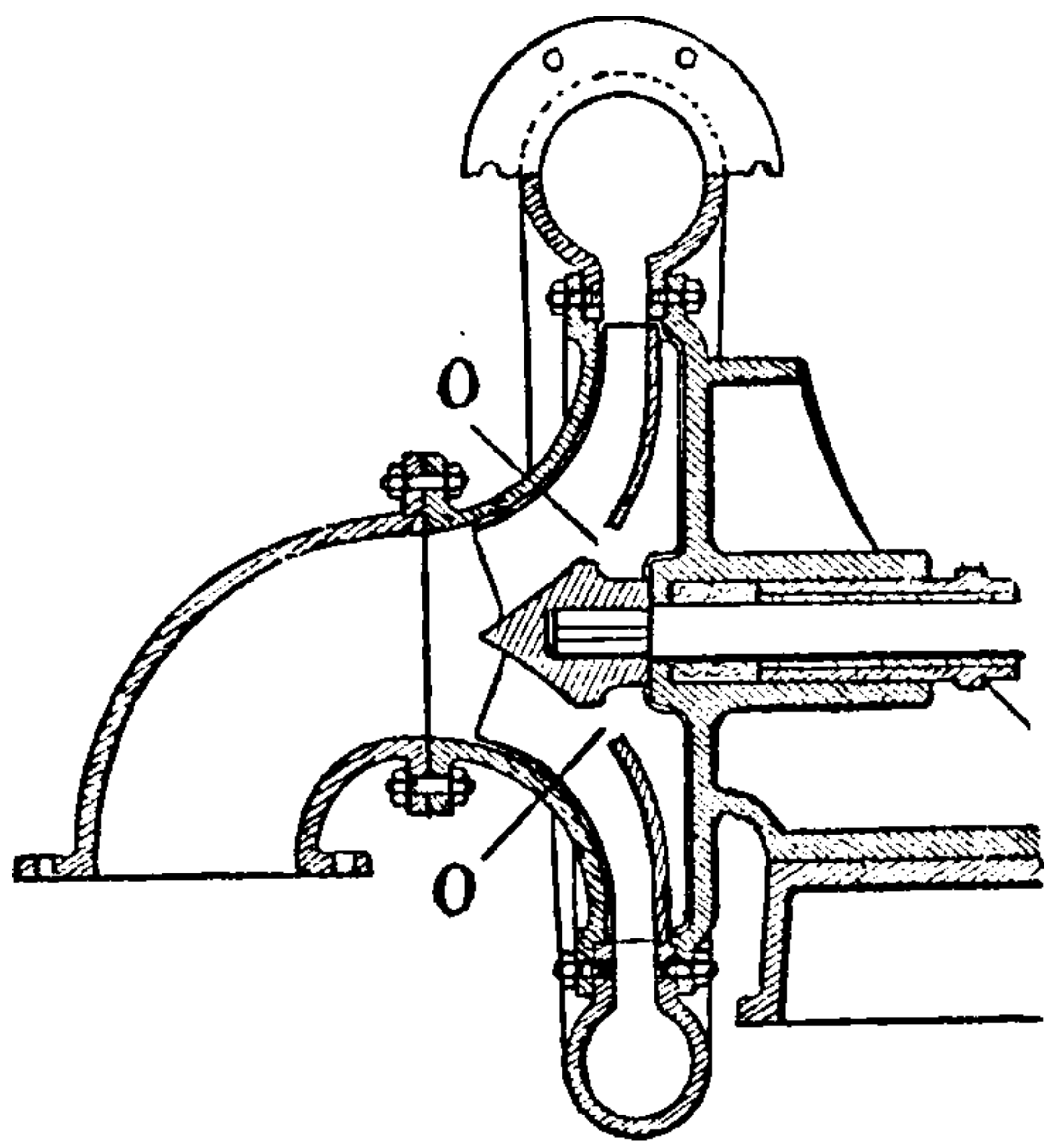


FIG. 17.—BALANCED CENTRIFUGAL PUMP.

air to rotate at the same angular velocity. Consider a ring of air of mean radius r, and radial thickness $d\,r$, and let the pressure outside be $p+\delta p$, and that inside the ring p and take the axial width as unity. If σ is the weight of 1 cu. ft. of air, the weight of a strip of the ring of length l is $\sigma\, l\, \delta\, r$, and the centrifugal force due to its rotation is

$$=\sigma\, l\, \delta\, r\, .\, \frac{v^2}{g\, r},$$

and the component normal to the diameter X Y is

$$\delta\, \mathrm{F}=\sigma\, l\, \delta\, r \frac{v^2}{g\, r} \sin\theta.$$

The resultant force due to half the ring is perpendicular to X Y, and

$$=\mathrm{F}=\int_0^{180}\sin\theta\,.\,d\,\theta\,.\,\sigma\,r\,\delta\,r\,\frac{v^2}{g\,r}=2\,\sigma\,\frac{\omega^2\,r^2}{g}\,\delta\,r.$$

The resultant air pressure inwards is also normal to X Y

$$=\mathrm{P}=\int_0^{180}\delta\,p\,l\sin\theta=\delta\,p\,.\,r\int_0^{180}\sin\theta\,.\,d\,\theta\,.=2\,r\,.\,\delta\,p.$$

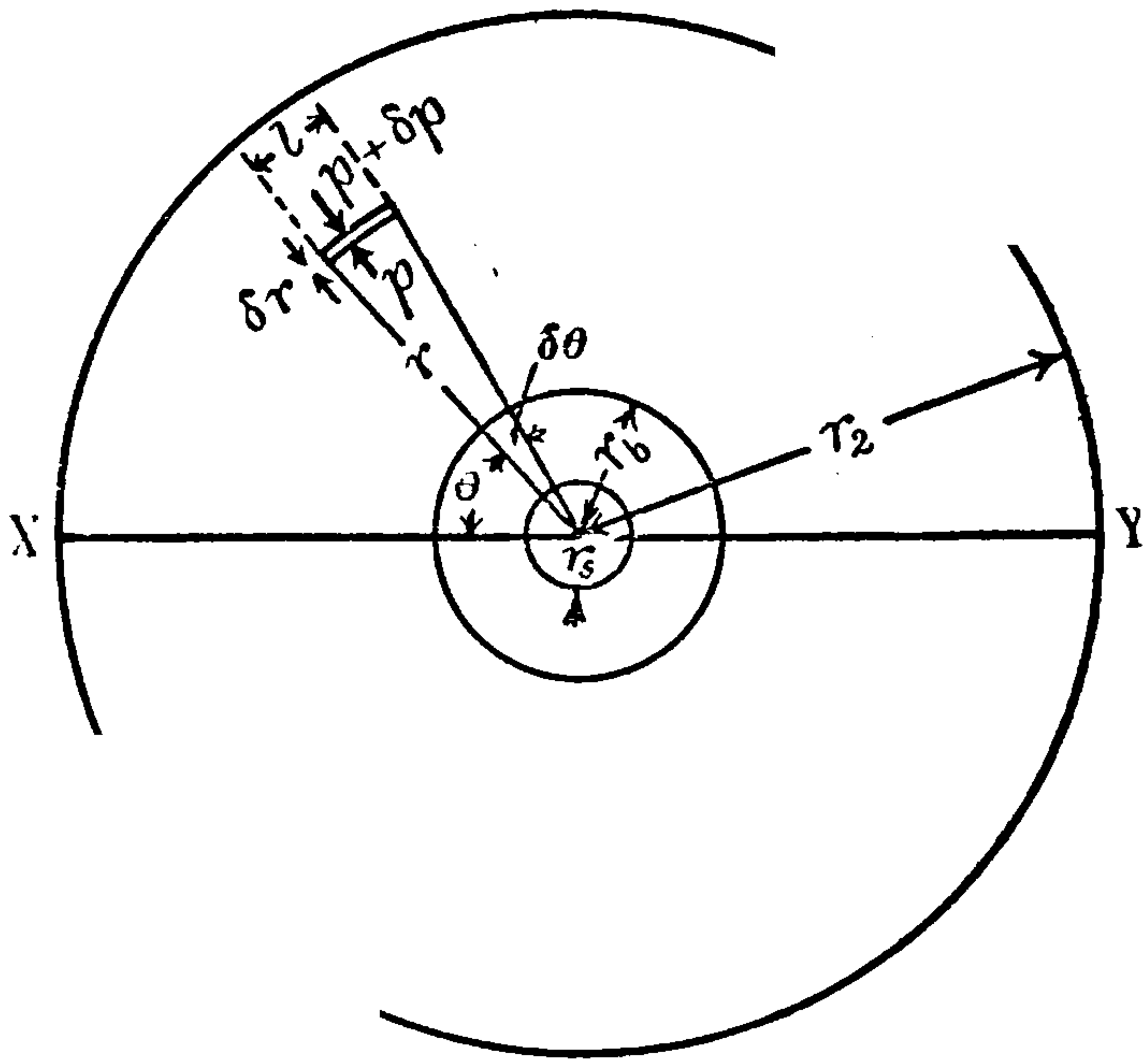


FIG. 18.—PRESSURE AT BACK OF WHEEL.

But P = F for equilibrium, so that

$$2\,\sigma\,\frac{r^2\,\omega^2}{g}\,d\,r=2\,r\,d\,p$$

$$\frac{r\,\omega^2}{g}=\frac{1}{\sigma}\,\frac{d\,p}{d\,r}.$$

Integrating $$\frac{r^2\,\omega^2}{2\,g}=\frac{p}{\sigma}+\mathrm{C},$$

where C is the constant of integration, so that, if r_b is the radius of the boss—that is, the internal radius at which rotation ceases—it follows that

$$\frac{(r_2^2 - r_b^2)\,\omega^2}{2\,g} = \frac{p_1 - p_b}{\sigma} = h_1 - h_b,$$

where p_b, h_b are the pressure per sq. ft., and head at radius r_b; and at any radius r if p, h are pressure and head,

$$\frac{p}{\sigma} = h_1 = h_2 - \frac{(r_2^2 - r^2)\,\omega^2}{2\,g},$$

where h_2, in the case of the fan, is

$$h_2 = v_2^2 - b_2^2 \operatorname{cosec}^2 \phi - 2\,g\,\mathrm{H}.$$

To calculate the whole pressure on the back of the wheel we must proceed as follows: The pressure head on a ring of mean radius r, and radial thickness $d\,r$, is

$$2\,\pi\,r\,d\,r\left(h_2 - \frac{(r_2^2 - r^2)\,\omega^2}{2\,g}\right),$$

so that the total pressure head between the radii r_b and r_2 is

$$\int_{r_b}^{r_2} 2\,\pi\,h_2\,r\,d\,r - \int_{r_b}^{r_2} \frac{2\,\pi\,\omega^2}{2\,g}\,r_2^2\,r\,d\,r + \int_{r_b}^{r^2} \frac{2\,\pi\,\omega^2}{2\,g}\,r^3\,d\,r$$

$$= (r_2^2 - r_b^2)\left[\pi\,h_2 - \frac{\pi\,\omega^2}{4\,g}(r_2^2 - r_b^2)\right]$$

To this we should have to add $h_b\,\pi\,(r_b^2 - r_s^2)$ the pressure on the flat annular end of the boss, r_s being the radius of the shaft, and h_b being $= h_2 - \frac{\omega^2}{2\,g}(r_2^2 - r_b^2)$. If the vanes at the back do not extend to a radius r_2, but to a smaller radius r_a, then the total pressure head on the back of the disc is

$$\pi(r_2^2 - r_a^2)h_2 + \pi(r_a^2 - r_b^2)\left\{h_2 - \frac{\omega^2}{4g}(r_a^2 - r_b^2)\right\} + h_b \pi(r_p^2 - r_s^2),$$

where $$h_b = h_2 - \frac{\omega^2}{2g}(r_a^2 - r_b^2).$$

Similar Fans.—Suppose two fans to be made from the same drawing but to a different scale, so that the dimensions of the one are n times that of the other, and suppose them to be driven so that the velocity of flow through corresponding parts of them is the same. Then the quantity delivered by the one will obviously be n^2 times that by the other, and if the water gauge produced by each is the same, then the orifices will be as n^2 is to 1, and the losses of head due to surface friction are the same because the areas of surface divided by the sections of the fan passages are the same, the roughness of the surfaces of course being supposed to be the same. If the manometric efficiencies are the same, the work per lb. done by the wheels and the losses due to shock at inflow to them and outflow from these are the same, and therefore the air efficiencies or

$$\eta = \frac{gH}{a_2 v_2} = \frac{\frac{a_2 v_2}{g} - L}{\frac{a_2 v_2}{g}}$$

where L is the losses of head, are the same. And we may reasonably suppose the manometric efficiency of the one to equal that of the other because the equation connecting v_2, H and internal velocities depends on the *proportions* of a fan and not on its absolute dimensions, and the proportions are the same in both; hence we have manometric efficiency and mechanical efficiency are the same for orifices in the ratio of n^2 to 1. Now if we call

$$\frac{Q}{r_2^2\sqrt{gH}} = O_R$$

the reduced orifice, then a diagram can be drawn with these as abscissæ, and the manometric and mechanical efficiencies as ordinates, and we shall get two curves which will apply to fans of a given type but of different sizes. The volumetric efficiencies $Q \div v_2 r_2^2$ is obviously the same at equal reduced orifices. In fig. 19 are shown characteristic curves of this nature for Rateau fans. We need not enlarge on the

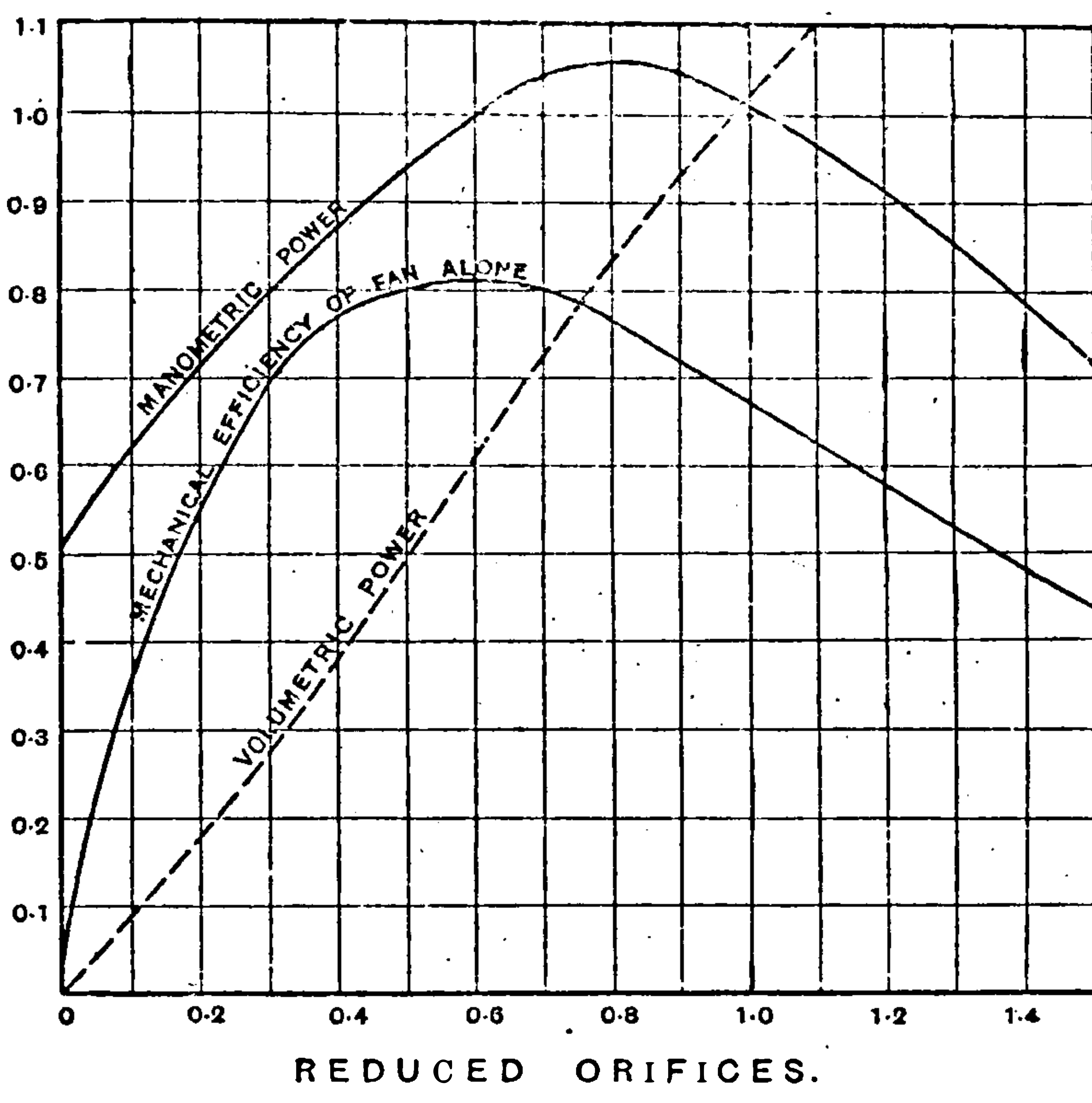


FIG. 19.—CHARACTERISTIC CURVES FOR RATEAU FANS.

assistance such curves should give to the intelligent designer, but he must remember that engine friction must not be included in the mechanical efficiency, and that if there are sharp angles before the eye of the fan which disturb the inflow both mechanical and manometric efficiencies will be much reduced.

being connected to a pipe soldered into the side of the delivery tube, while the other branch is attached to a pipe passing to the centre of the delivery tube, the end being suitably bent to face the stream. The pressure in the pipe C (inductive action being prevented) will be that due to the compression of the air only, but the pressure in D will be that duc to the pressure and velocity combined. Hence the gauge A, which indicates the difference between the pressures in the pipes C and D, will record the pressure due to velocity only. The compression of the air can be

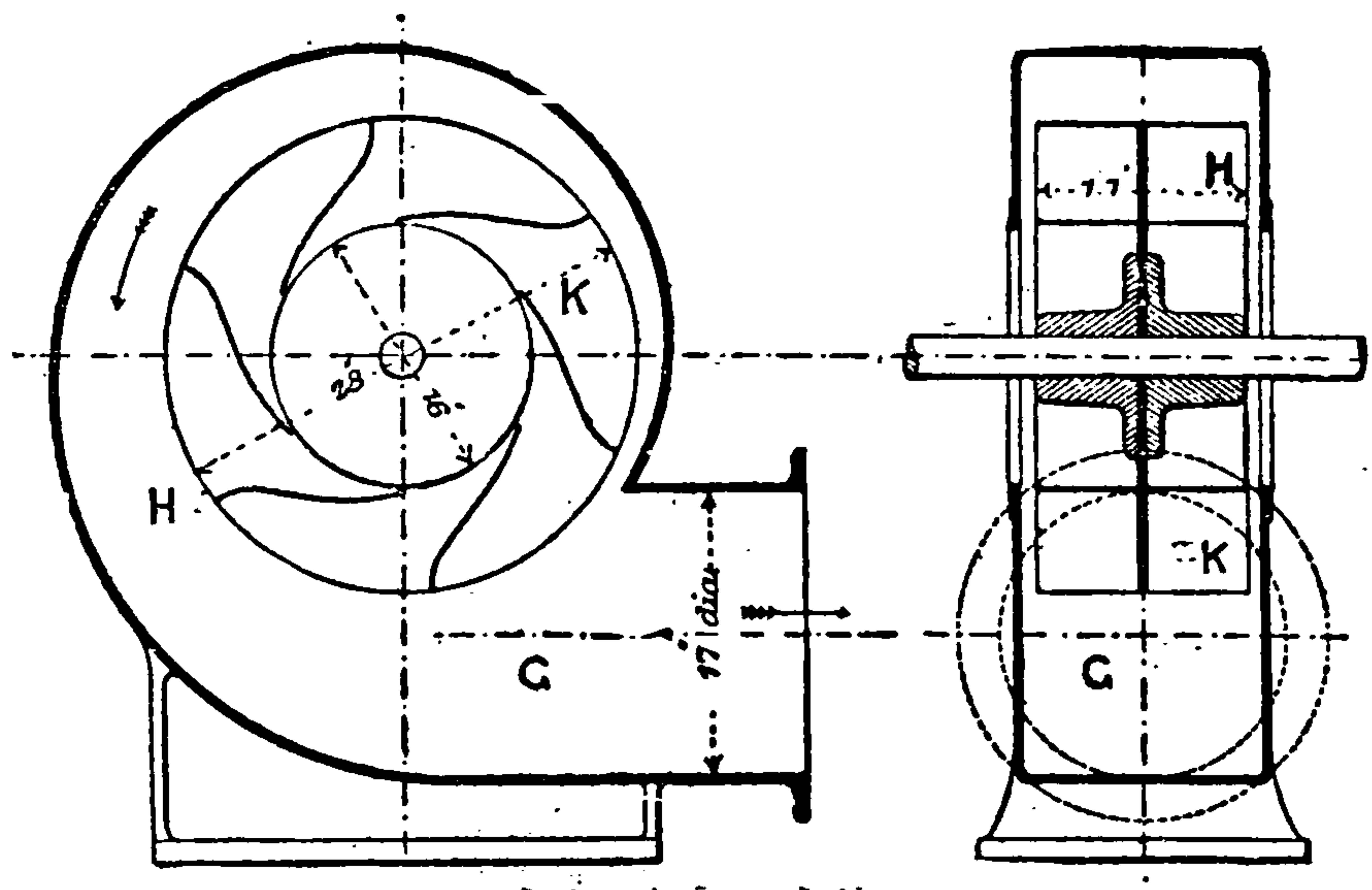


FIG. 20.—28″ FAN TESTED BY HEENAN AND GILBERT.

measured by the second gauge B. First, if the outlet be closed so that no air is delivered, the gauge A will remain at zero, since there is no flow of air through the tube, but the second gauge B will indicate a considerable compression, about $11\frac{1}{2}$ in. of water, if the tip speed of the fan be 12,000 ft. per min. Next let the end of the delivery tube be opened to give, say, half the area of the outlet K. The fan now passes a considerable quantity of air, about 8,000 cu. ft. per min., and on account of this flow the velocity gauge will indicate nearly $1\frac{3}{4}$ in. of water. The compression, as shown by the gauge B, will have fallen to

8 in. During the time the outlet of the delivery tube was closed, with no air being delivered, the efficiency was of course zero. But when the fan is passing 8,000 cu. ft. of air, under a compression of 8 in. of water, the efficiency reckoned on the compression alone will be about 66 per cent., 15 horse power being required to drive the fan. When the outlet of the delivery tube is fully opened, the

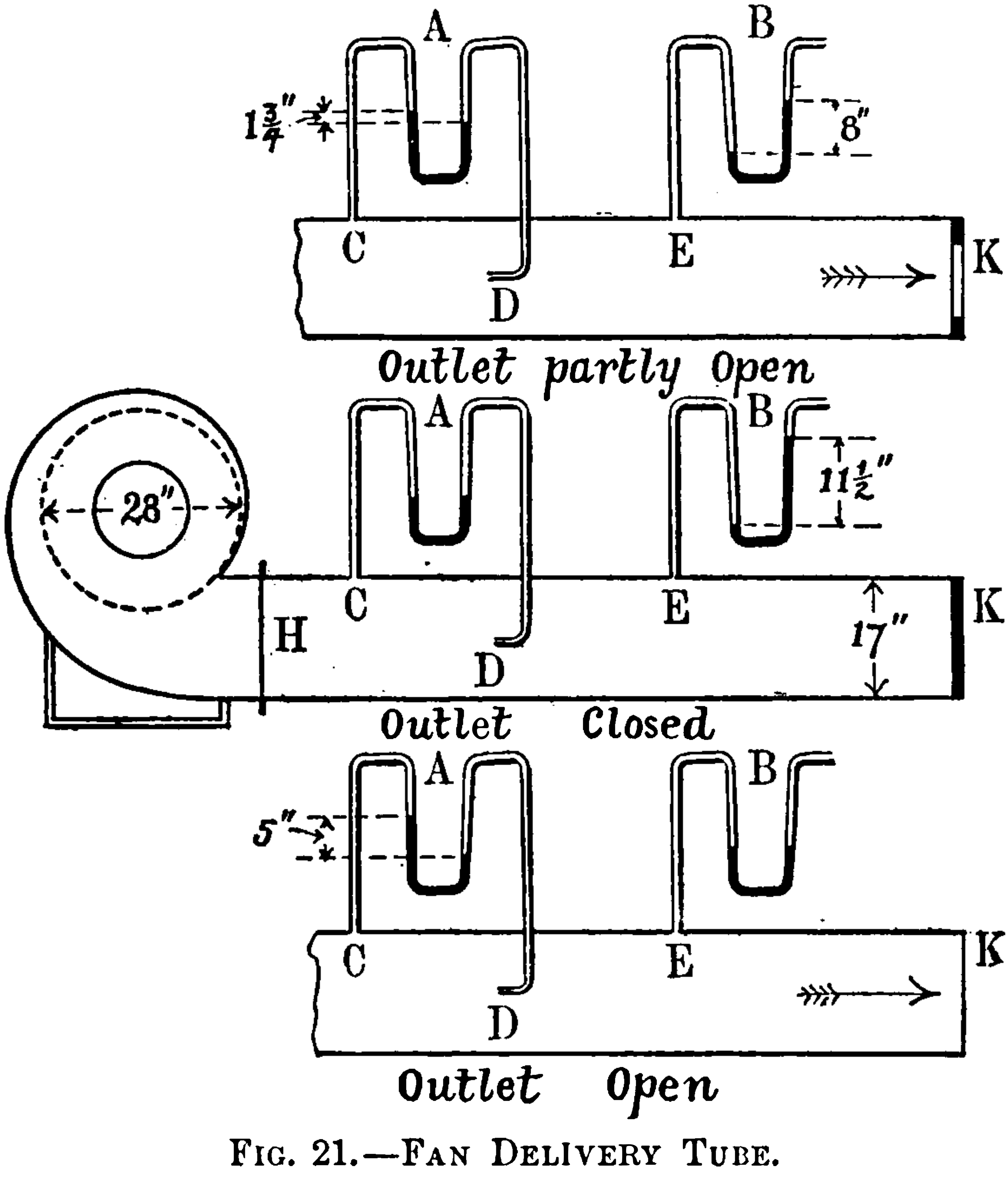


FIG. 21.—FAN DELIVERY TUBE.

fan delivers freely to the atmosphere, and gauge B shows that the air is under no compression whatever, but the amount of air has increased to about 13,700 cu. ft. per min., and the passage of this air through the delivery tube shows a velocity head on A of nearly 5 in. of water. Since the air is not compressed, but merely expelled at atmospheric pressure, the efficiency reckoned on the compression is now zero. There is necessarily a delivery intermediate between no delivery and no compression, at this tip speed of 12,000 ft. per min., for which the mechanical efficiency, reckoned on the compression produced on the air, is a maximum.

Heenan and Gilbert presented their results on diagrams in which the abscissæ were cu. ft. of air per min. for

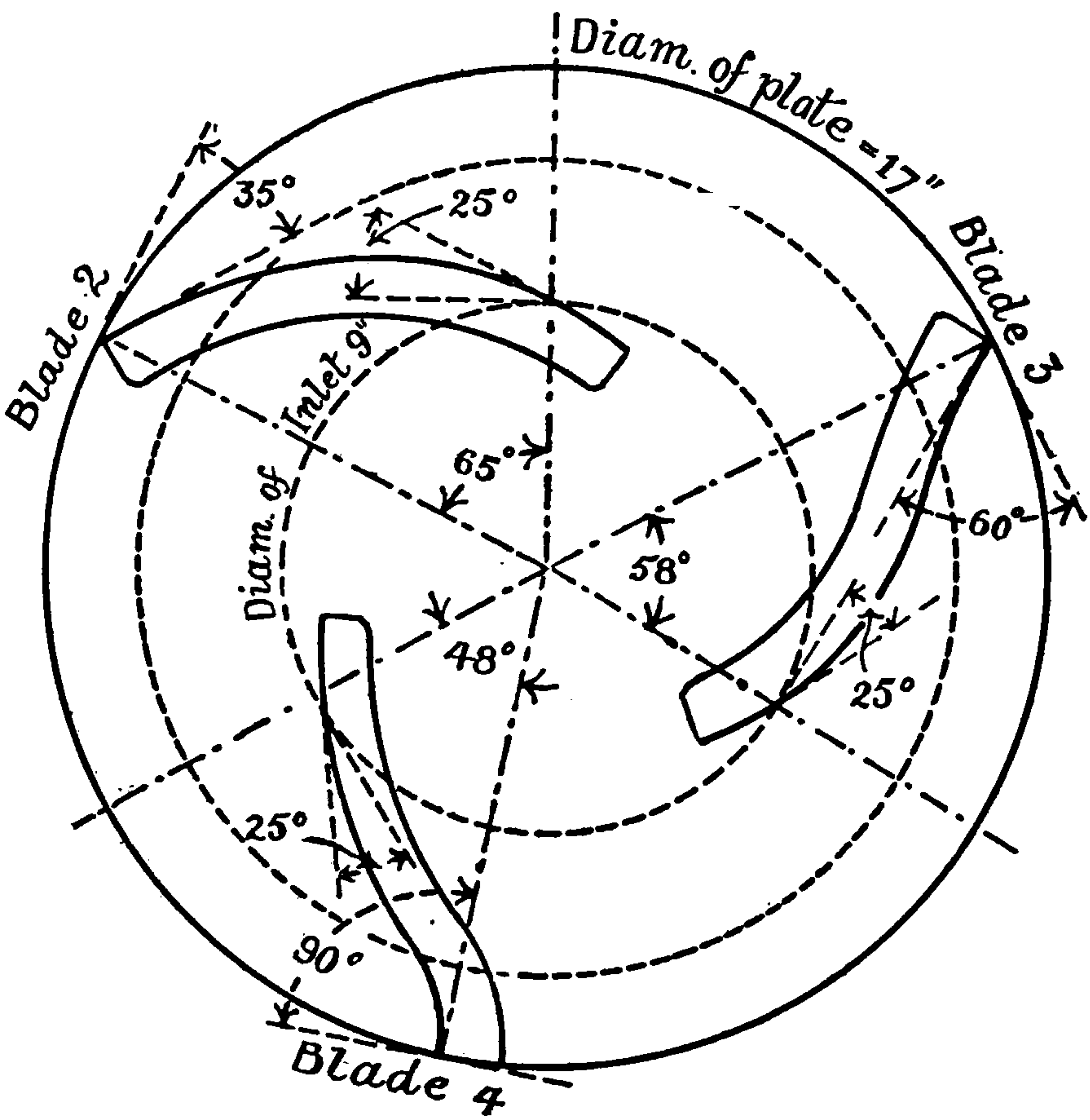


FIG. 22.—FORMS OF BLADE TESTED.

a fixed tip speed and ordinates were efficiencies, and compressions as shown by the gauge at that speed ; other curves were drawn showing the brake horse power required by the fan, and the total efficiency, the kinetic energy in the air discharged being added to the work done in compressing the air. This is sometimes called dynamic efficiency, and is misleading, except in cases in which the fan can be fitted with an expanding chimney, and even then the kinetic energy left in the air at its mouth must be deducted. A fan that has a high dynamic efficiency may be of very low real efficiency, and if quantity of air is of more consequence than the smallness of brake horse power, then the volumetric efficiency should be given, and not an imaginary mechanical efficiency which is not attainable. The horse power of the fan considering compression alone was calculated by the formula,

$$\text{Fan horse power} = \frac{\delta\, h\, Q_1}{33000 \times 12} = \frac{h\, Q_1}{6352},$$

where h = water gauge in inches and Q = discharge in cu. ft. per min., and δ weight in lb. of 1 cu. ft. of water. Three forms of blade were tested (shown in fig. 22, numbered 2, 3, and 4), making angles of 35 deg., 60 deg., and 90 deg., with tangents to the outer circumference, and 25 deg. with tangents to the inner circumference, the fan centre being 17 in. diameter and 8 in. wide. Figs. 23, 24, 25, give curves of efficiency, water gauge, and brake horse power, for each of these blades for a constant tip speed of 12,000 ft. per min. Blade No. 4 gives the best results, although not much superior to those of blade No. 3, whilst blade No. 2, having a tip angle of only 35 deg., gives a low efficiency, owing to the rapid drop of the compression curve as the discharge of air increases. The arrangement of apparatus for measuring the brake horse power, volume of air, compression, and speed, are shown in fig. 26. The fan is driven from the counter-shafting B, which derives its motion from a spherical steam-engine at C. The outlet of the fan is connected by a short circular iron delivery tube with a boiler flue E E, 30 in. diameter and 18 ft. long. At the centre of the flue a partition F is fitted, to which can be attached a series of plates having circular orifices of various sizes, varying between 4¾ in. and 18 in. diameter. A well-cut outlet of known diameter is placed at the end of the flue G, where the velocity of the air was measured by an anemometer. This opening was made much larger in diameter than the outlet of the fan, in order to avoid injuring the anemometer by a violent current of air. In fig. 26 it is shown in end elevation together with the levers whereby the anemometer was moved over all portions of the outlet, the instrument being kept truly perpendicular to the axis of the flue. The pulley driving the fan was not keyed to the shaft, but was driven by it through the Emerson power scale, a form of transmission dynamometer in which the moment of the driving effort is balanced, through a system of levers, by a pendulum moving over a graduated scale. A speed counter records the revolutions of the shaft. A tachometer coupled to the counter-shafting B by a spiral spring, enabled the speed of that shafting to be regulated, and the proper tip speed to be approximately maintained by the man in charge of the engine. A small band counter, carried in a sliding frame, and applied when necessary to the fan spindle, enabled the exact number of rev. per min. of the fan to be obtained. Measurements of the pressure and velocity of the air-stream were taken at the section M of the delivery tube. The velocity varied greatly in different positions on the same cross-section of the tube. Readings were taken at several points in a cross-section by means of two gauges, each of which could be traversed over a diameter at right angles to the other. It was afterwards found unnecessary to measure the velocity, since it can be readily calculated when the discharge, as measured by the anemometer, is known; but at the same time the velocity, as measured by the velocity gauge, gives a check on the anemometer readings. If the cross-section of the delivery tube be divided into a number of imaginary areas and the square root of the mean gauge reading for each area be multiplied by that area and by a suitable constant, then these results added together give the total discharge of the fan. The annexed table illustrates the variation of velocity referred to; it gives the gauge reading due to the velocity in four positions along a diameter of the cross-section of the delivery tube, where the gauge was fitted, as taken simultaneously.

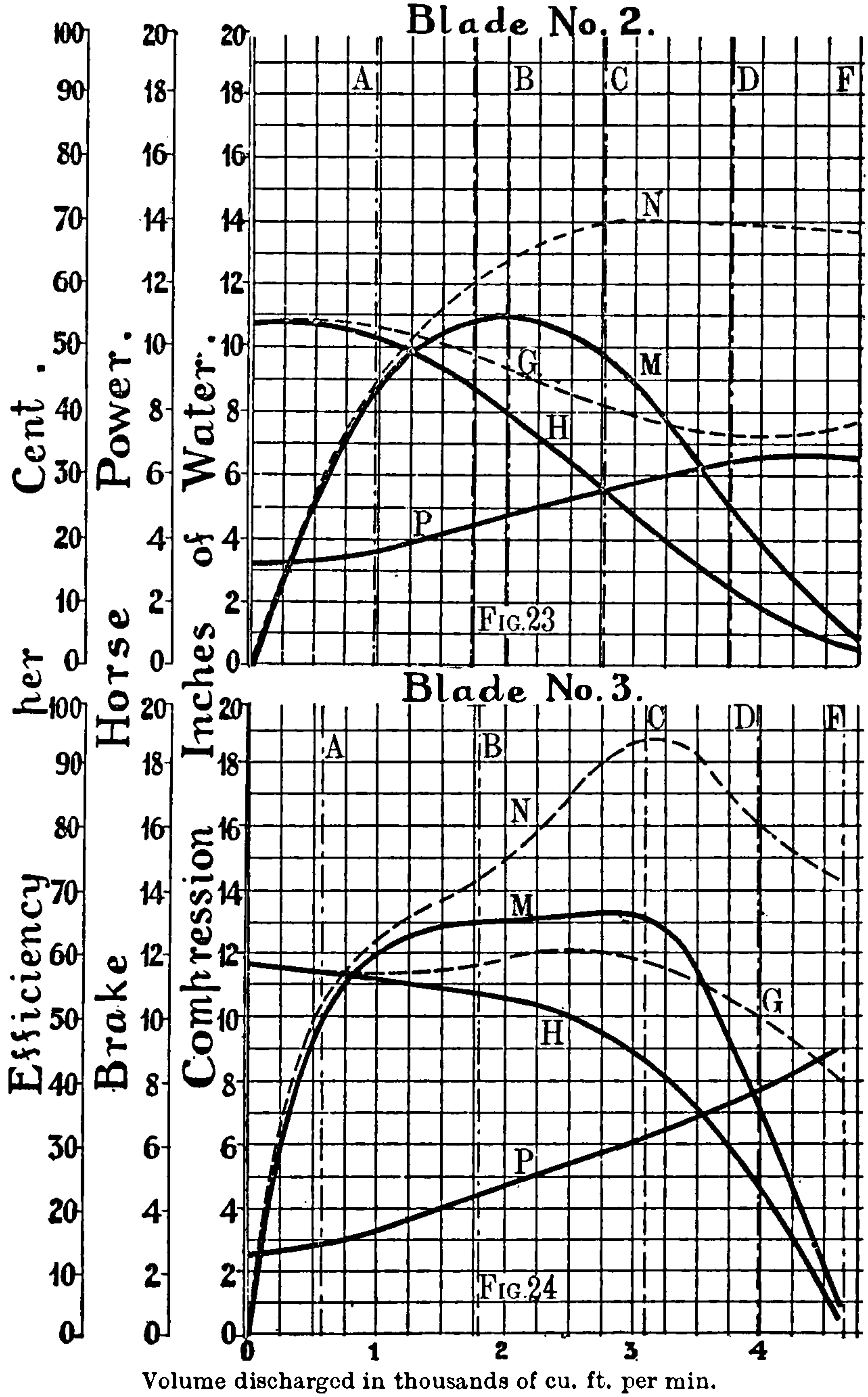
TABLE 6.—VARIATION OF VELOCITY.

Position.	Gauge reading due to velocity.	Air velocity, ft. per sec.	Distance from centre of tube.	Remarks.
			Inches.	
1	3·20	118	+ 3·1	Diameter of delivery tube, 8·9 in.
2	2·88	112	+ 1·1	
3	1·85	90	- 1·1	
4	1·70	86	- 3·1	

This, we think, is a proof of imperfect design of the fan, because if well designed there is no reason why the velocity should not be almost uniform over the whole section of the tube; if anything, it should be greater at the centre than the sides. It will be noticed in fig. 20 that some of the discharge from the fan is thrown against the upper surface of the discharge pipe, instead of the whole stream flowing out parallel to the axis of the tube. The measurement of the compression presented some difficulty, owing to the fact that the air flowing across the end of the side gauge caused a large amount of induction, a vacuum being often recorded where a pressure was known to exist. Professor W. C. Unwin found that a plate placed across the end of the tube prevents this inductive action. The form of side gauge used for measuring the static pressure is shown in fig. 42. It was tested by Heenan and Gilbert, and found to give very good results. To draw the characteristic curves from the experimental results, seven resistance plates, A, B, C, D, E, F, with graded circular orifices, were arranged to fasten on to the centre F of the boiler flue. Two observations were taken with each plate of the discharge, the compression and the horse power supplied to the fan at or near tip speeds of 5,000 ft., 6,000 ft., and 12,000 ft. per min. The authors verified the following laws:

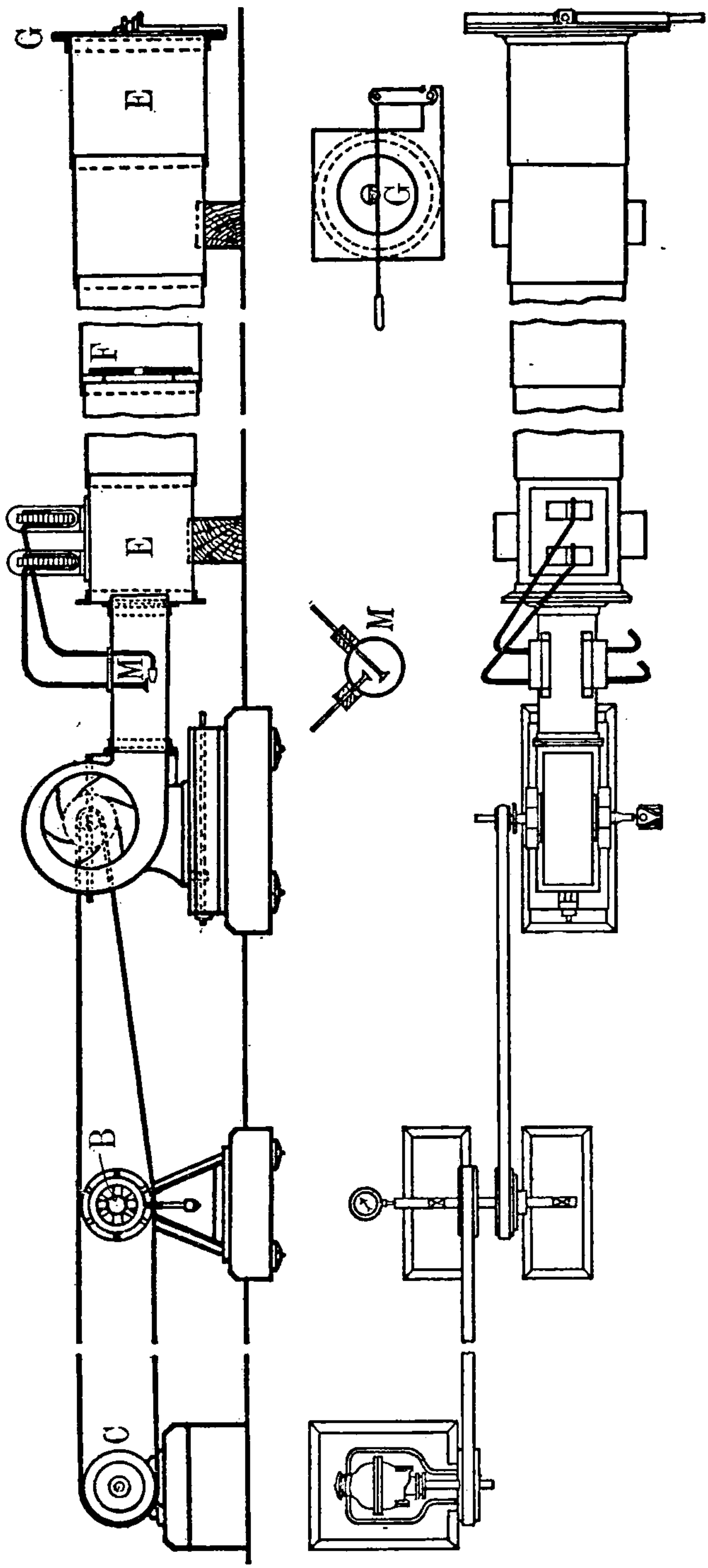
1. The air discharge varies as the speed.
2. The gauge reading varies as $(\text{speed})^2$.
3. The B.H.P. varies as the $(\text{speed})^3$.

} For a constant resistance.



FIGS. 23 AND 24.—EFFICIENCY GAUGE AND POWER DIAGRAMS.

Curves: *M* shows efficiency; *P*, brake horse power; *H*, compression; *G*, total gauge; *N*, total efficiency.



17″ Heenan Fan. (Tip speed, 12,000′ per min.)

FIG. 26.—HEENAN AND GILBERT'S APPARATUS FOR MEASURING POWER, VOLUME, AND SPEED.

tapering side plates. The diameter of the inlet was $5\frac{7}{8}$ in., the widths of the fan centre at inlet and outlet being $7\frac{1}{4}$ in. and $1\frac{1}{8}$ in. respectively. This fan ran in a concentric case, the clearance being $1\frac{1}{8}$ in. (which in our opinion made the comparison of tapering *versus* parallel sides worthless, as the preceding theory very clearly shows that a well-designed volute is necessary to obtain high efficiency and a suitable water gauge). The compression obtained with closed outlet was 9·4 in., against 11·2 in. with blade No. 4, fig. 25. Further, the compression fell off very rapidly with increase

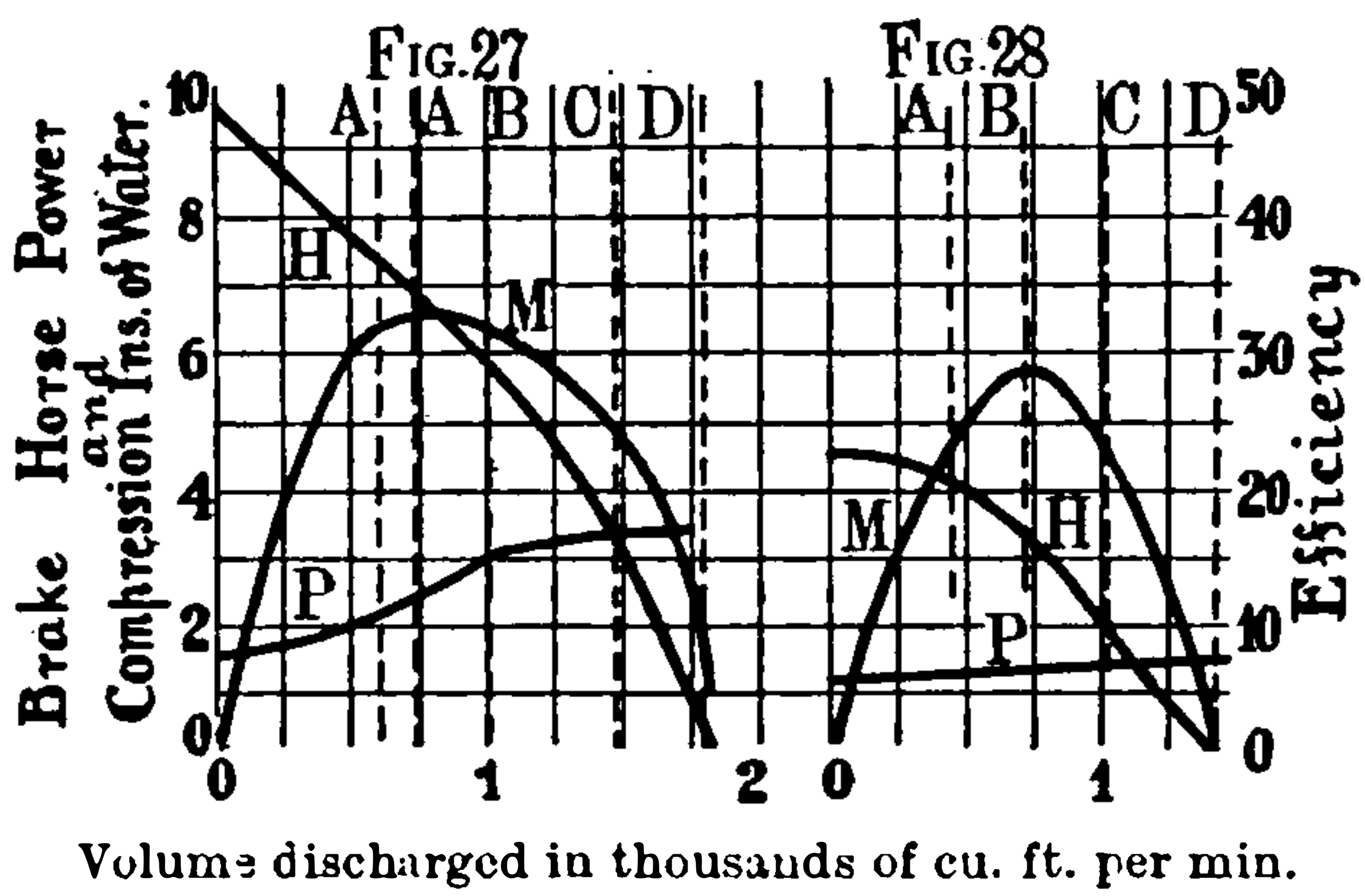


Fig. 27.—Characteristic Curve for Fan with Tapering Sides.

Fig. 28.—Characteristic Curve for Fan with 12-Inch Centre Diameter.

Curves: *M* shows efficiency; *P*, brake horse power; *H*, compression.

of output, so that the working compression would not be more than 8 in. The efficiency was also low (as might reasonably be expected). Fig. 28 shows the characteristic obtained at 12,000 ft. per min. from a fan centre 12 in. diameter, the maximum width being $2\frac{3}{4}$ in. The centre rotated in a whirlpool chamber of 23 in. diameter. The centre was of cast iron, and the tip angle of the blades was 30 deg. Of course the efficiency measured, which was less than 30 per cent., would not be representative for so small a fan, but the maximum compression did not exceed 4·4 in.

The effect of a whirlpool chamber (usually called a diffuser) is to produce a very quiet running fan.

Test of a Mine Ventilating Fan.—The fan selected for illustration (fig. 29) was supplied to the Parkend Colliery Co., South Wales, by Heenan and Gilbert, and works in connection with the approach tunnel and ventilating shaft of the mine. The wheel is 7 ft. diameter and 2 ft. wide. The upper portion of the case and evasé chimney is built up of wrought-iron plates, the lower portion being formed in brickwork $4\frac{1}{2}$ in. thick. The fan was driven by a horizontal, non-condensing engine, the cylinder being $12\frac{1}{8}$ in. diameter and $17\frac{3}{4}$ in. stroke. To provide a variable resistance for the

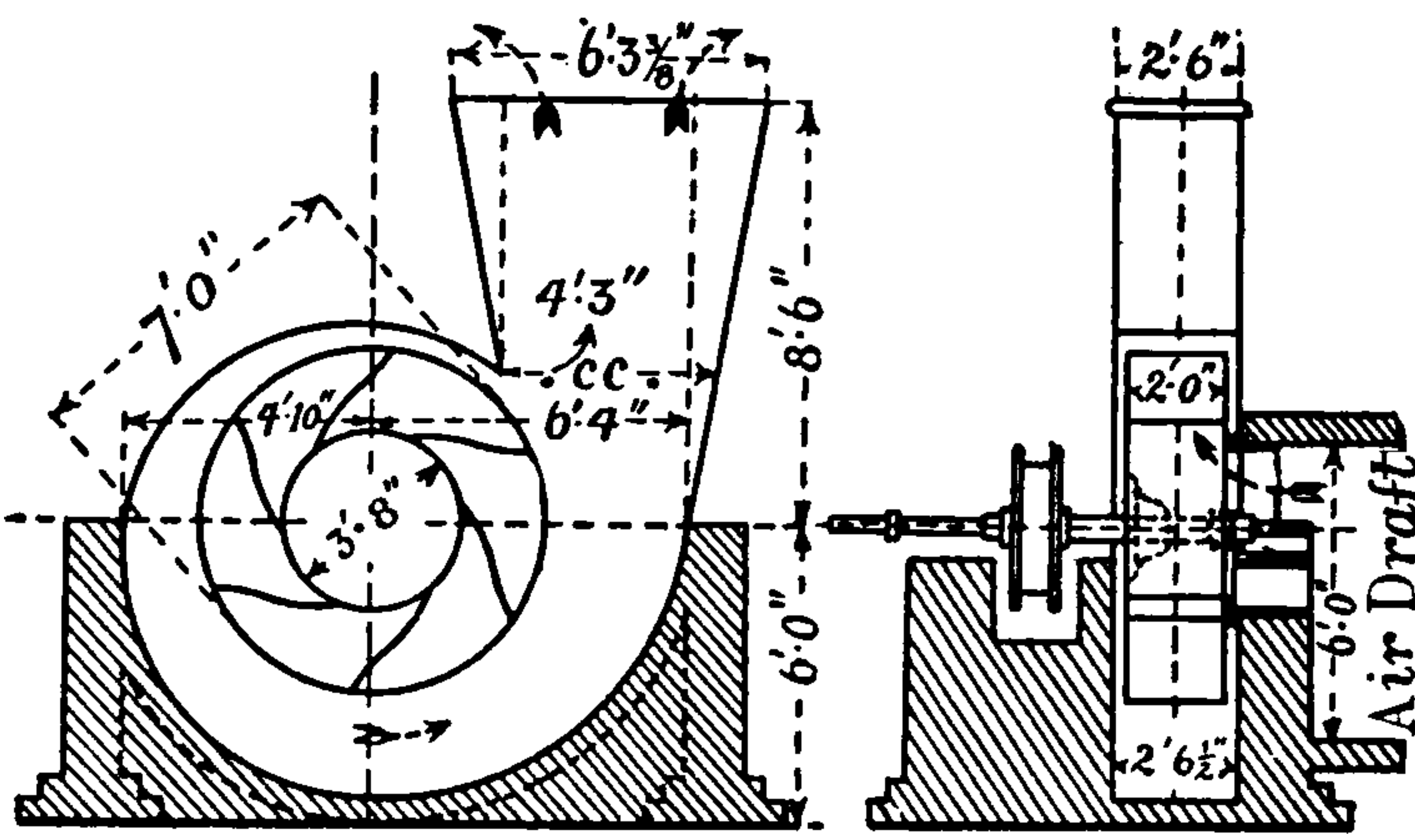


FIG. 29.—MINE VENTILATING FAN.

fan, three 9 in. by 3 in. planks were placed across the mouth of the air drift, and boards nailed to these planks restricted the flow of air to the fan more or less as required. The folding doors at the top of the ventilating shaft were open during the whole of the test. A tachometer, driven by a belt from the fan shaft, enabled the approximate speed of the fan to be judged and regulated by the man in charge of the engine; the number of revolutions in a two-minute reading being obtained by a hand-counter held to the fan shaft. The engine speed was obtained by a counter applied to the shaft in the same manner. The air discharge was measured by an anemometer at the top of the fan chimney. The area of the top of the chimney was divided into eight equal rectangles by tightly stretched wires, and the anemometer, attached to a small iron tube, was held for a quarter of a min. in each division. In this fan the flow of air was fairly uniform over the whole of the outlet area of the chimney, but in some cases, where the fan was run slowly for experimental purposes, guide vanes had to be fitted in the base of the chimney to secure the result mentioned. Four degrees of opening were arranged at the adjustable orifice, and, for each of these, readings were taken with tip speeds of 4,000 ft., 5,000 ft., 6,000 ft., 8,000 ft, and 9,000 ft. per min. The vacuum produced by the fan was measured by a side gauge, placed in the air drift close to the fan inlet, and a pipe led from this tip to a water gauge placed on a table outside. (In our opinion this is one of the reasons for the apparently high efficiency obtained by the fan, which at a tip speed of 9,000 ft. per min. reached 70 per cent.) We have already stated that if the manometer is placed in a strong current of air the water gauge will be increased, and that at inflow to the fan this will not be the correct gauge. When the fan was running at a tip speed of 9,000 ft. per min. the discharge from the fan at the efficiency of 70 per cent. was 14 cu. ft. per sq. in of diametral wheel section per min., so that the flow into the eye, assuming it to be uniform and neglecting the obstruction of the bearing, was at a velocity of

$$\frac{14}{60} \times \frac{24 \times 84}{\frac{\pi}{4} \times (3\frac{2}{3})^2} = 44{\cdot}6$$

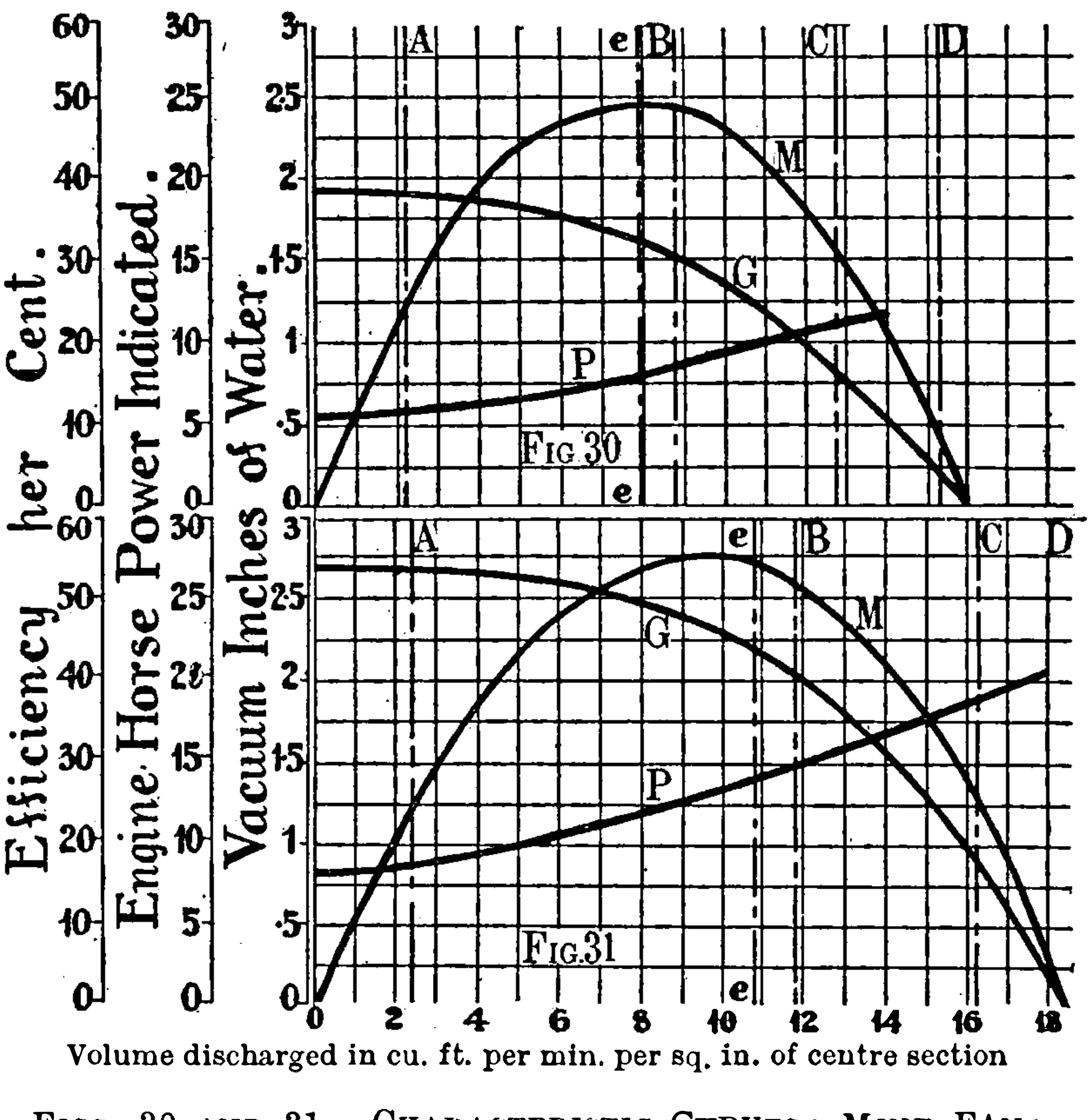
because the eye was $3\frac{2}{3}$ ft. diameter and the centre section 24×84 in. This velocity would increase the water gauge by

$$\frac{(44{\cdot}6)^2}{64{\cdot}4} \times \frac{12}{820} = 0{\cdot}52 \text{ in.}$$

It was 5·6 in., and should therefore be reduced to at least 5·15 in., so that the efficiency could not be more than

$$\frac{5{\cdot}15}{5{\cdot}6} \times 70 = 64{\cdot}4 \text{ per cent.}$$

(assuming that the anemometer did not exaggerate the discharge, which it invariably does). Three side gauges C C, fig. 29, were also placed at the root of the chimney, so that the vacuum produced, and consequently the efficiency



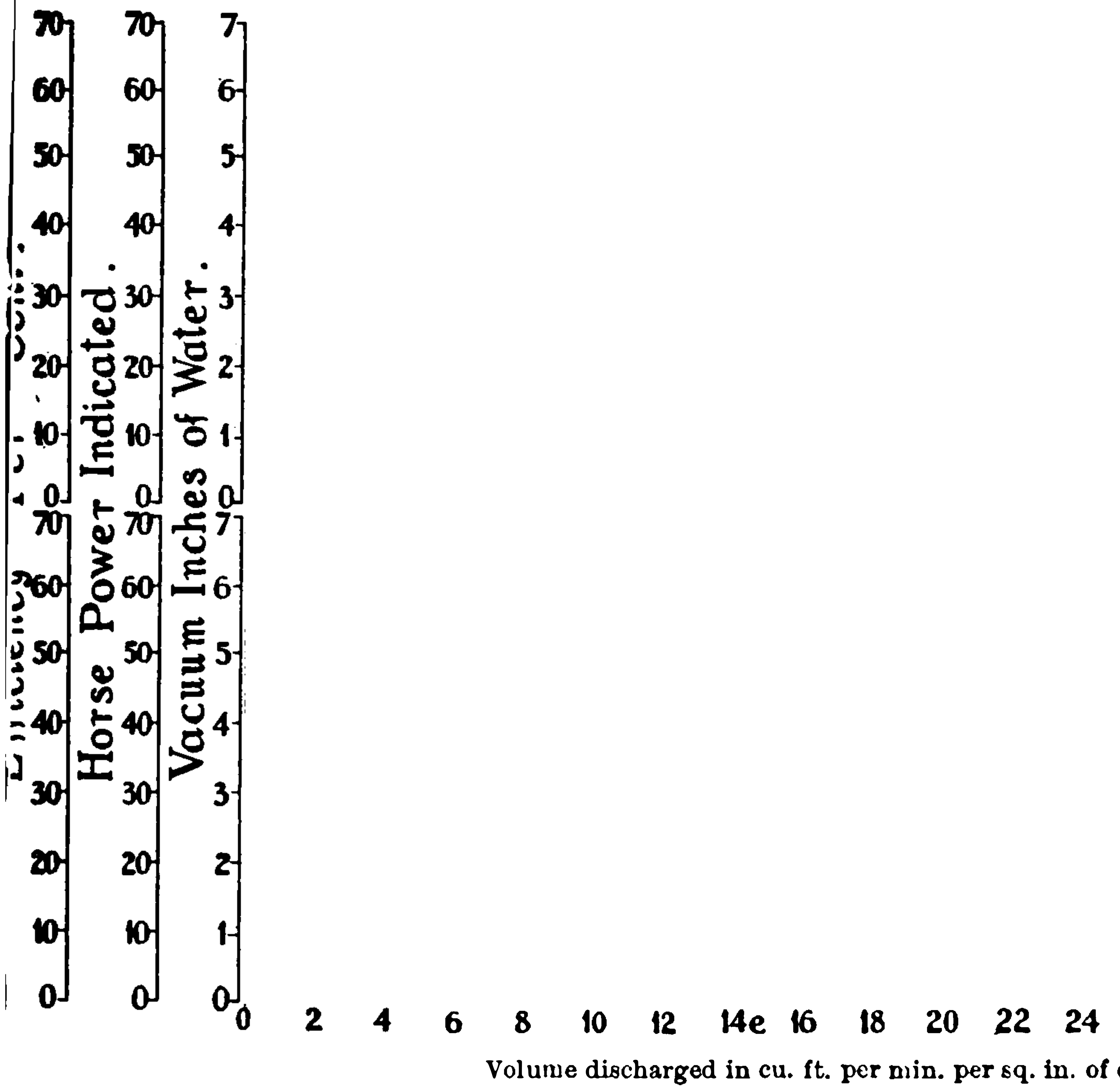
FIGS. 30 AND 31.—CHARACTERISTIC CURVES: MINE FANS.

30.—7-ft. Heenan Fan. Tip speed, 5,000 ft. per min.
31.—7-ft. Heenan Fan. Tip speed, 6,000 ft. per min.

Curves: *M* shows efficiency; *G*, water gauge; *P*, horse power (indicated).

of the chimney, could be determined. It was found that the vacuum was practically the same at all three, so that only one was read. Figs. 30 to 33 give the characteristic curves obtained from this test. The dotted lines *e e* correspond with the resistance offered when the choking boards were

removed and the fan took air from the mine only. Th
maximum efficiency increases with the speed, and is 67·2 pe

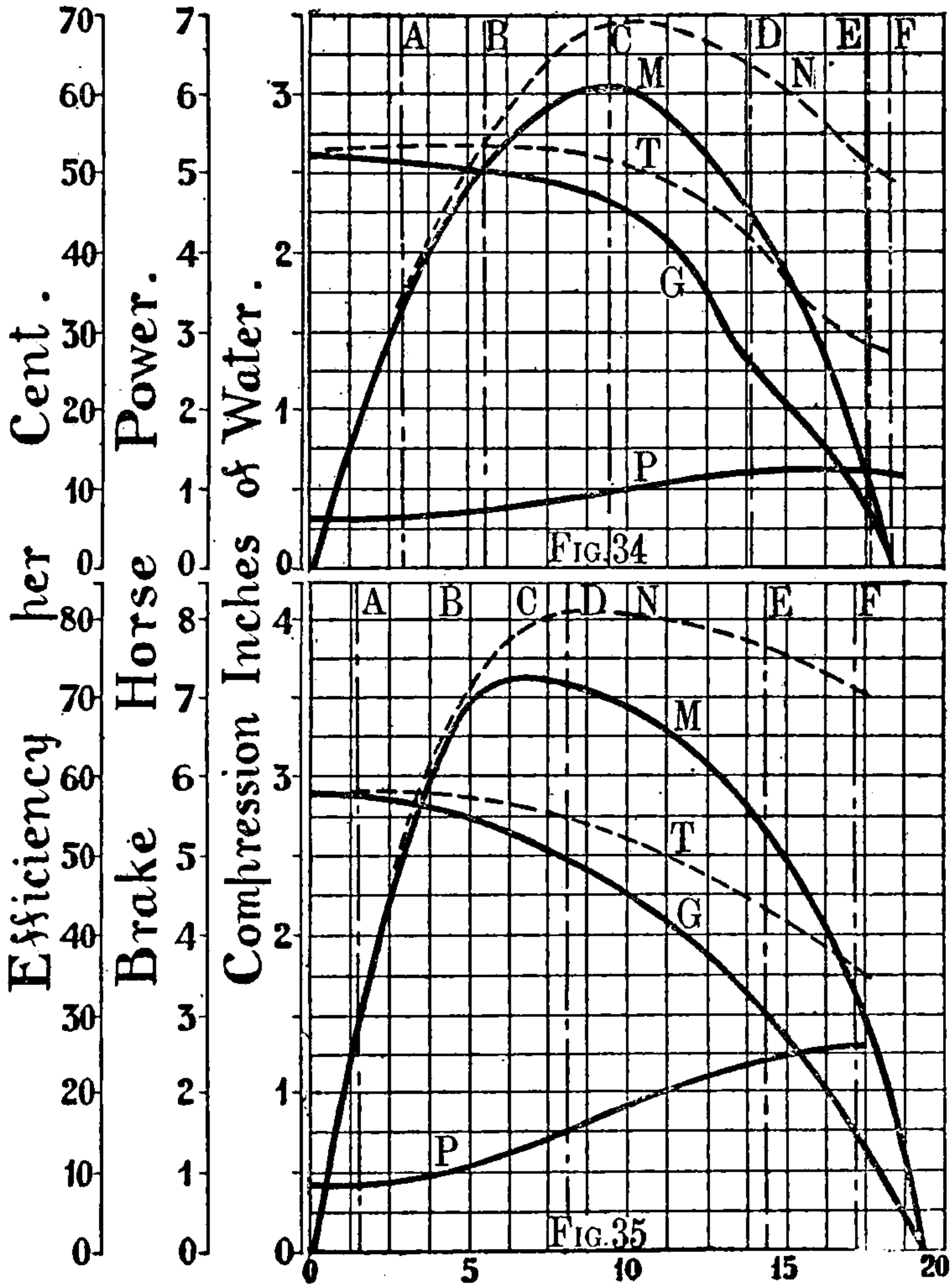


FIGS. 32 AND 33.—CHARACTERISTIC CURVES: MINE FANS.

32.—7-ft. Fan. Tip speed, 8,000 ft. per min.
33.—9,000 ft. per min.

Curves: *M*, efficiency; *G*, water gauge; *P*, horse power (indicated).

cent. at a tip speed of 8.000 ft. per min., and 70·3 per cent
at 9,000 ft. per min. The fan was designed to pass 20,00
cu. ft. per min. with a water gauge of 3½ in., at a spec



Volume discharged in cu. ft. per min. per sq. in. of centre section.

FIGS. 34 AND 35.—CHARACTERISTIC CURVES: MINE FANS.

34.—18·5″ Heenan Fan. 35.—28″ Heenan Fan.
Tip speed, 6,000 ft. per min. in each case.

Curves: *P* shows brake horse power; *G*, compression; *T*, total gauge; *M*, efficiency; *N*, total efficiency.

of 300 revolutions. At 7,000 ft. per min. and 318 rev, the water gauge was 3·45 and the discharge 23,150 cu. ft., so that the fan is amply large enough for the work.

and if A is the section of the jet

$$\sigma \frac{A v^2}{g} = P, \text{ or } \frac{P}{\sigma A} = \frac{v^2}{g} = \frac{p}{\sigma} = H,$$

where p is the pressure per sq. ft., σ is the density, and H is the head equivalent to the pressure p. Hence we might at first imagine that the reading of the Pitot tube would be $v^2 \div g$, instead of $v^2 \div 2g$ in feet of air; but the air deviated by the shock upon the end of the Pitot tube does not lose the whole of its momentum parallel to the axis of the tube, so that in reality

$$H = m \frac{v^2}{g}$$

where m is a coefficient smaller than unity, depending probably upon the section of the orifice and of that of the conduit in which it is situated. When the orifice is small compared with the conduit, experiment shows that $m = \frac{1}{2}$ very nearly. If h is the water gauge in inches,

$$h = \frac{12 v^2}{2g} \frac{\sigma}{\delta} = \frac{12 \times 39{\cdot}8 \, B v^2}{64{\cdot}4 \times 62{\cdot}3 \times 29{\cdot}92 \tau}$$

where δ = weight of 1 cu. ft. of water,

$$= \frac{B v^2}{251 \tau} \text{ for dry air} \quad . \quad . \quad . \quad . \quad . \qquad (49)$$

where B is the height of the barometer in inches of mercury, and τ is the absolute temperature in Fahrenheit degrees, or

$$\tau = F + 461;$$

so that if we put $B = 29{\cdot}9$ and $F = 62$ deg.,

$$h = \frac{v^2}{4390} \text{ in. of water.}$$

To test a tip as described in Heenan and Gilbert's paper, the water gauge connected with a revolving tube is observed, and the reading compared with that calculated from the equation

$$h = \frac{12\, v^2 \sigma}{2 g \delta},$$

the correct values of σ and δ being used; a correction must, however, be introduced for the vacuum produced by the centrifugal force of the air in the revolving horizontal tube carrying the tip. This vacuum we have already shown to be $\sigma v^2 \div 2 g$ at the centre; for, in a mass of air rotating, we proved that

$$\frac{r^2 \omega^2}{2 g} = \frac{h}{\sigma} + C,$$

so that if r_2 be the external radius, and

$$v = r_2 \omega,$$

$$\frac{v^2}{2 g} = \frac{p_2 - p_1}{\sigma}$$

where $p_2 - p_1$ is the difference of pressure between the centre and radius r_2. Hence if the tip measures the pressure due to the air velocity correctly, there should be no reading on the water gauge for any speed, as the vacuum due to the centrifugal force just balances the pressure due to the velocity of the moving tip against the air. As a matter of fact, small readings were observed on the manometer, but these were accounted for by supposing that the air in the tank was drawn round by the rotation of the tip, and that the maximum velocities of the air were—for gauge, fig. 37, 2·7 ft. per sec.; for gauge, fig. 38, 5·9 ft. per sec.; and for gauge, fig. 39, 5·4 ft. per sec.; so that the velocity of the air may be calculated by formula (49).[9]

Experiments on Centrifugal Fans by Bryan Donkin.[10]—Although these experiments were made on small fans, considerable value is attached to them, as it is extremely probable that the quantities of air recorded are correct. In accordance with Heenan and Gilbert experiments, Donkin found that when the passages remained the same the discharge varied as the speed. In each set of experiments upon a given type of fan the quantity of air passing was varied, and a change in the pressure was produced by throttling the flow in the delivery pipe at some distance

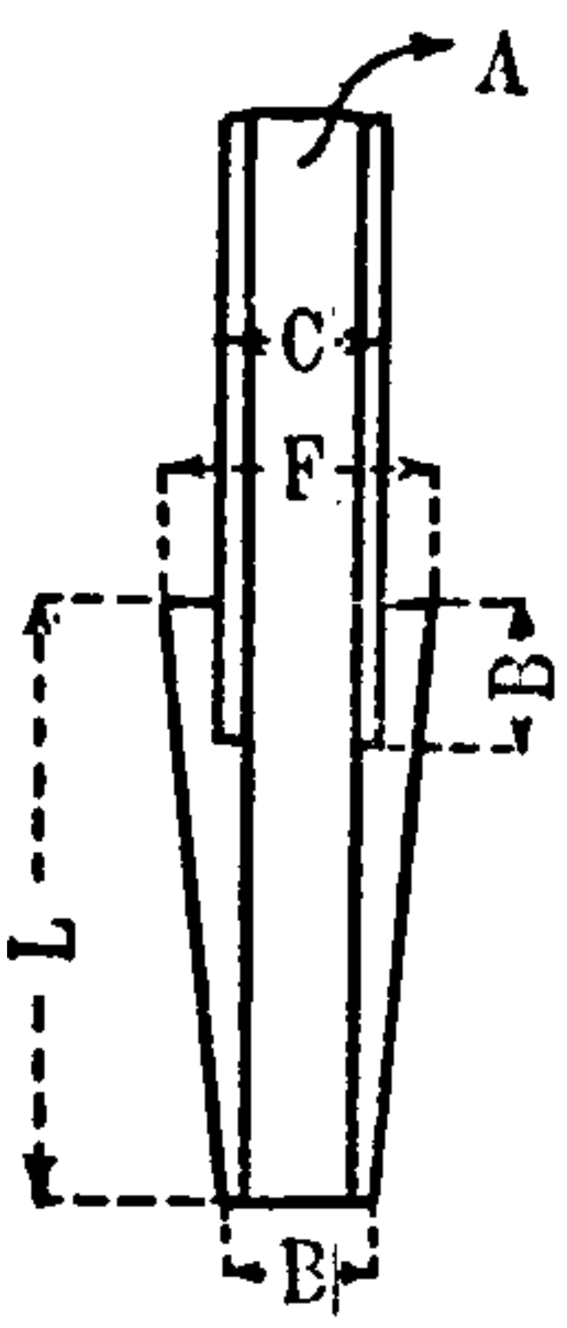


FIG. 37.

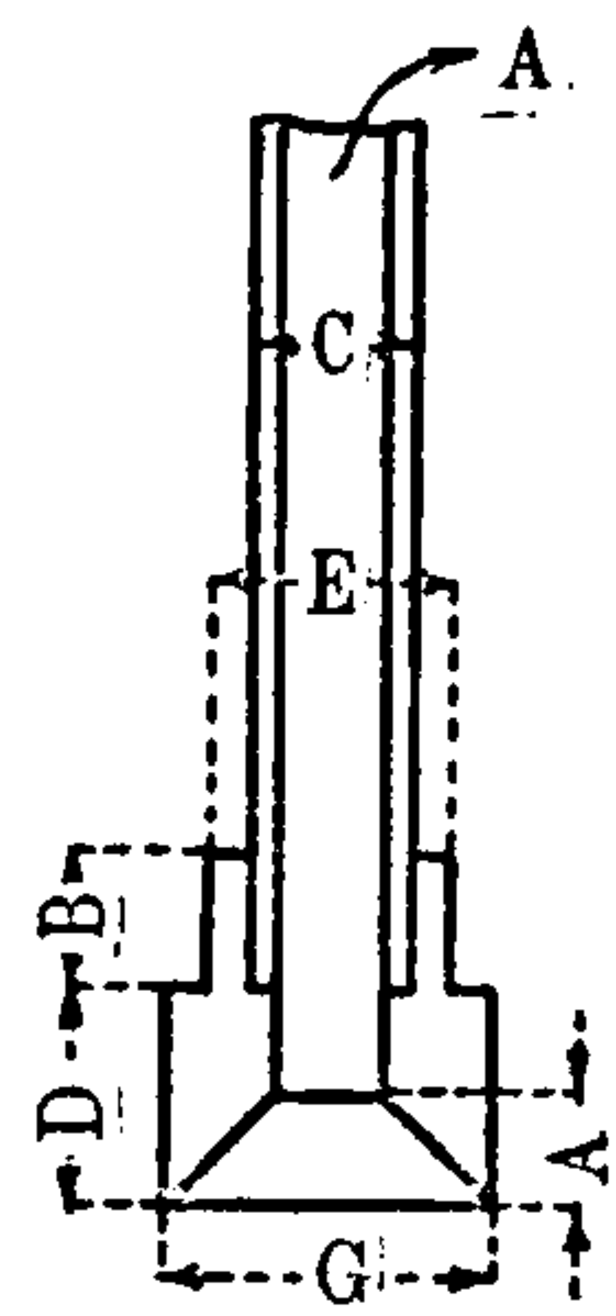


FIG. 38.

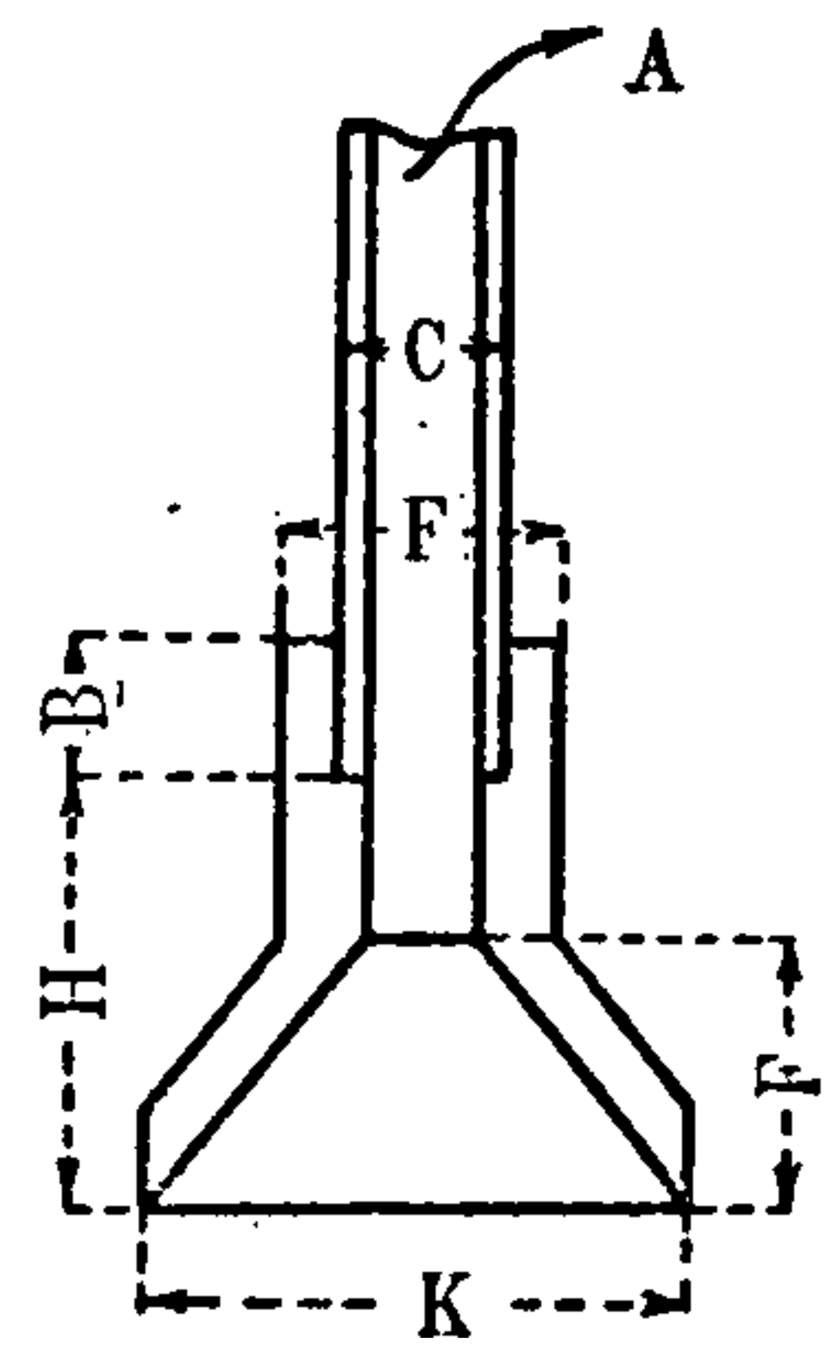


FIG. 39.

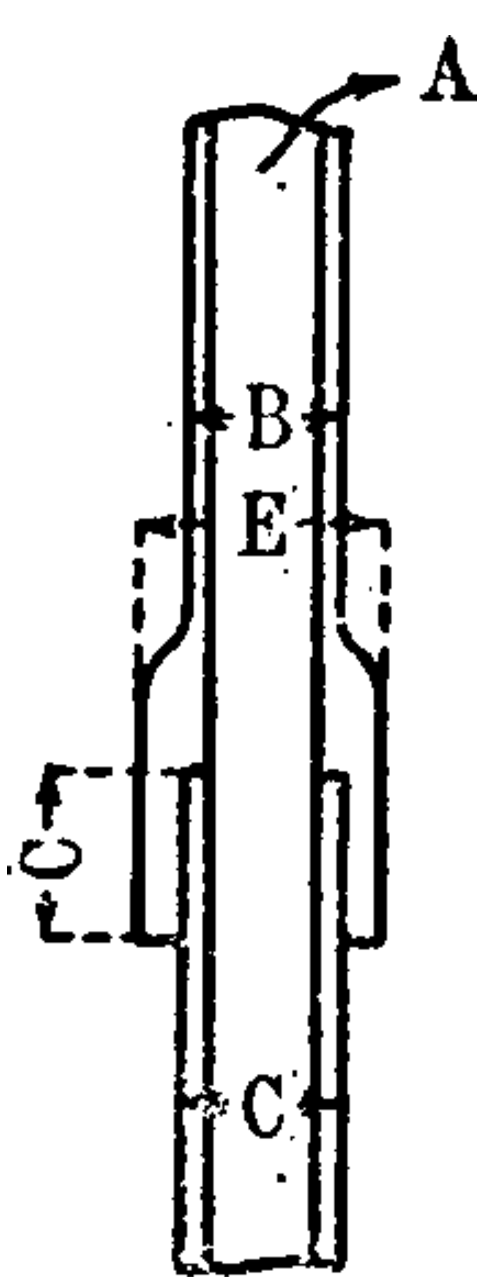


FIG. 40.

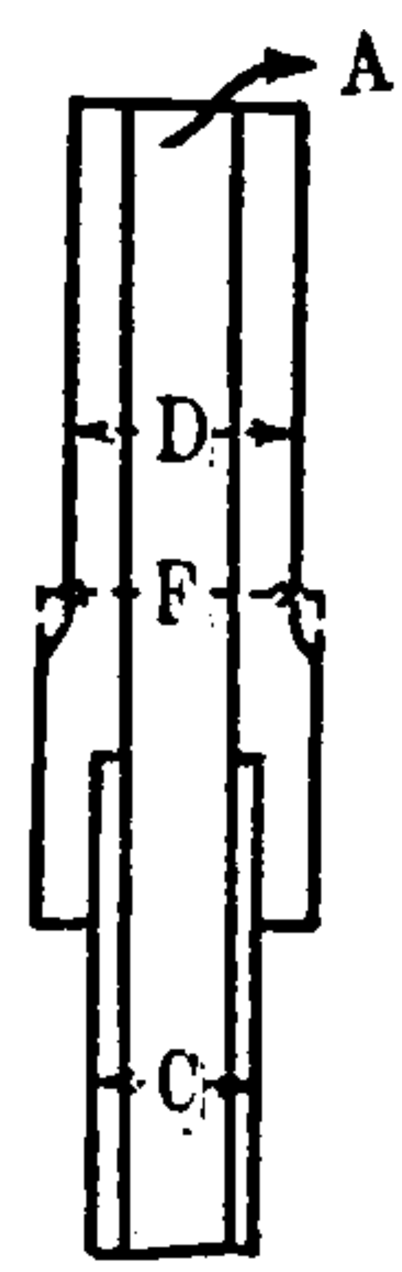


FIG. 41.

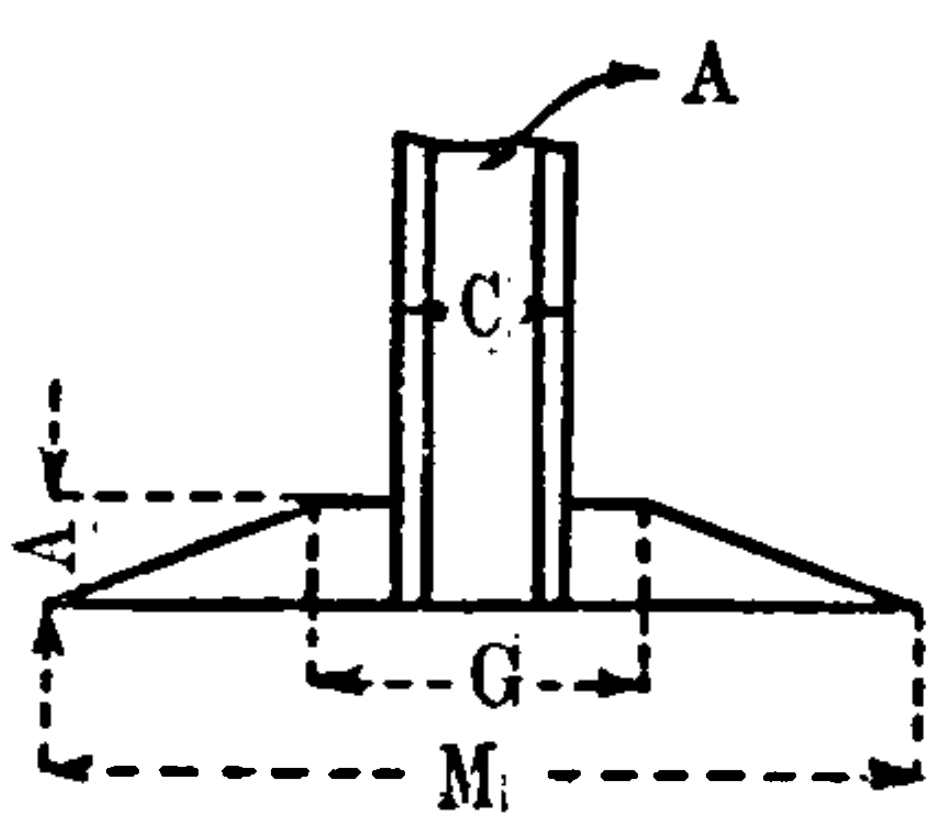


FIG. 42.

TYPES OF GAUGE TIPS.

A, $\frac{1}{4}''$; B, $\frac{5}{16}''$; C, $\frac{3}{8}''$; D, $\frac{1}{2}''$; E, $\frac{9}{16}''$; F, $\frac{5}{8}''$; G, $\frac{3}{4}''$; H, $1''$; K, $1\frac{1}{4}''$; L, $1\frac{3}{8}''$; M, $2''$.

from the fan, and allowing the air to pass successively in each experiment through wove wire of 3, 8, 30, and 50 meshes to an inch. The "equivalent orifice" was also varied by the insertion of one to four pieces of perforated zinc superposed. The wove wire of 3 and 8 meshes to an inch gave respectively by calculation an effective area of 80 per cent., and 56 per cent. of the area of the pipe. One piece of perforated zinc gave an area of 40 per cent. Experiments under these several conditions, as well as with no baffle, were made upon each fan, the end of the delivery pipe being open to the atmosphere in all cases. This end was, in a final experiment, completely stopped, and the air, instead of passing through the fan, was churned up inside it. The I.H.P. was taken in each case, as well as the pressure of the air and the speed of the fan. About ten experiments were made on each fan, each occupying about 15 min. after all conditions had become constant, and as a similar series of experiments were made upon each, with the same pipe and apparatus, the results are comparable.

Eleven different types of fan, illustrated in Fig. 45, were tested, of diameters varying between 16 in. and $25\frac{1}{2}$ in. The number and shape of the vanes differed considerably. Each fan was driven by a strap from the same steam-engine, which was indicated to give the power absorbed. The I.H.P. required to drive the engine at different speeds was accurately known, and was in each case deducted from the total I.H.P. A large quantity of air is required at low pressure in some cases, and in others a small quantity at high pressure. The volume of air passing at the maximum pressure, with a given speed of the fan and equivalent orifice, was determined in the experiments. The terms volumetric and manometric (or pressure) efficiencies, already explained, are—

$$N_v = \frac{Q}{v_2 r_2^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

$$N_m = \frac{g\,H}{v_2^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (25a)$$

and the mechanical efficiency of the fan itself is

$$N = \frac{62 \cdot 3\, Q\, h}{12 \times 550 \times \text{B.H.P.}}$$

where B.H.P. is the horse-power supplied to the fan shaft. This is approximately the same as the expression we

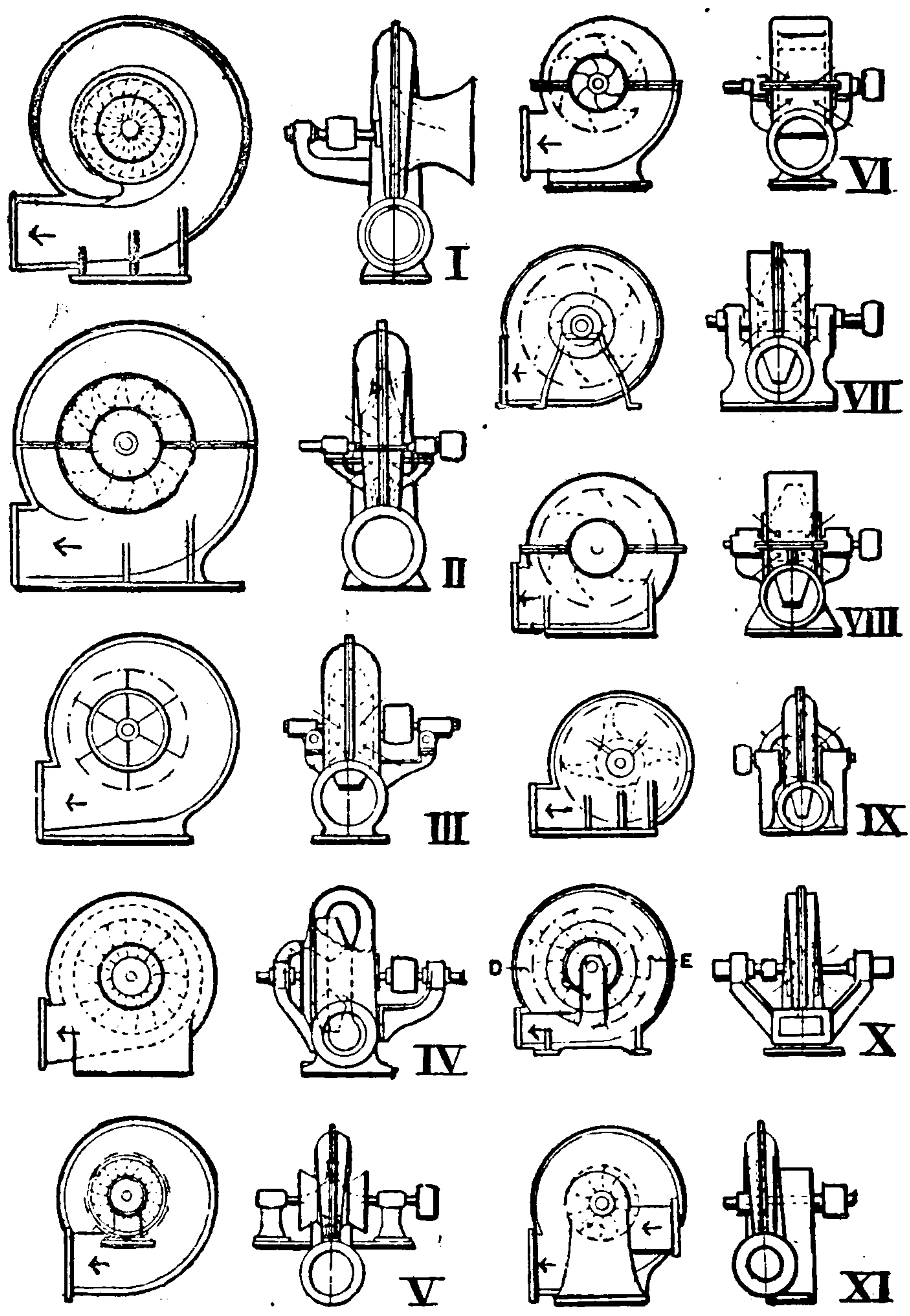
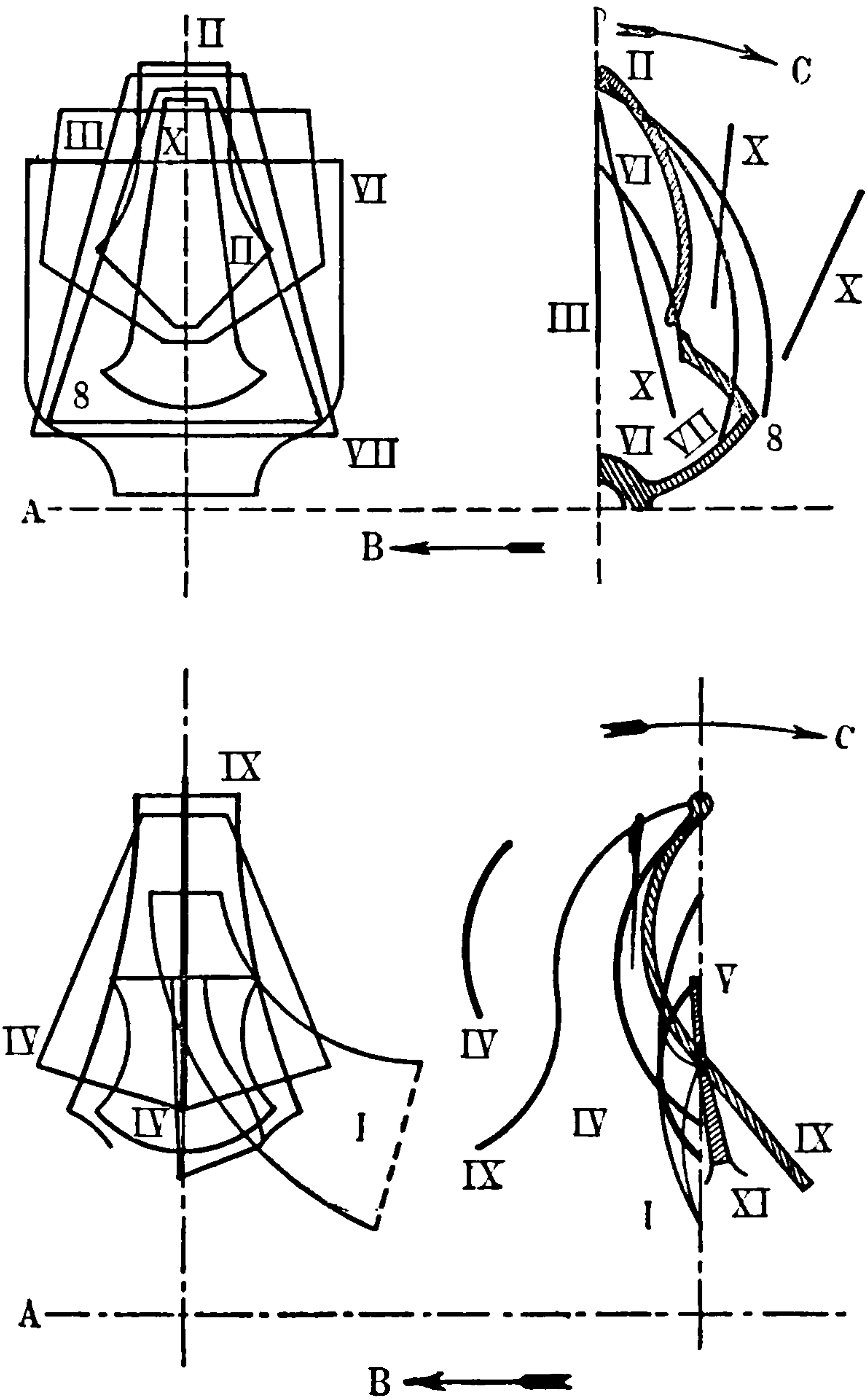


FIG. 45.—FANS TESTED BY BRYAN DONKIN.



Figs. 43 and 44.—Elevations of Fan Blades.

Blades of fans shown in fig. 45. *A*, centre line of fan spindle; arrow *B* shows direction of outlet; arrow *C* shows direction of rotation.

termed the air efficiency $\frac{g\,H}{a_2 v_2}$, and which we also denoted by N.

The number of vanes of each fan, together with their shape and direction of curvature, are given in Table 8. The casing in which the vanes revolved differed considerably in shape, but was always of cast iron. The vanes revolved in some cases with the concave, in some with the convex, and in others with the flat side to the outlet. They were set both radially and inclined to the centre of the shaft. Tests were made to determine the effect of driving the fan with the blades revolving in the opposite direction to that indicated by the makers, and an increase in delivery sometimes resulted, although other conditions remained the same. A drawing of each fan is given in fig. 45. The maximum efficiency or the best experiment on each is given in Table 8, and the volumetric, manometric (or pressure), and mechanical efficiencies are represented graphically in figs. 48, 49, and 50. As regards possible errors, the speeds, pressures, and quantities of air are probably correct to within 3 and 4 per cent. Table 9 gives the results of the experiments on the different fans with maximum and minimum equivalent orifices, together with particulars as to speeds, etc., observed in the tests.

Donkin's Experimental Apparatus.—A conical piece of pipe X Y (fig. 46), fitting the outlet of each fan, was bolted, as shown, to the fan under test on one side, and to a wrought-iron pipe, $14\frac{1}{32}$ in. in diameter, at the other. This latter was used for all experiments. Each fan was driven from a small engine by a strap from a rigger fixed on the crank shaft. Two assistants started the counters by signal at the same instant, and at the end of the experiment they were similarly thrown out of gear. By this means the speeds and slip of the strap were ascertained. At B and B two pipes of $\frac{1}{4}$ in. diameter were fixed to the air-pipe Y Z, and to these were attached two U water gauges, by which the static pressures of air were observed. The dynamic pressures were obtained by means of a dial gauge of special design. The circular pieces of wove wire or perforated

TWENTY WROUGHT-IRON VANES RUNNING CONCAVE TO THE OUTLET. ONE OUTLET ONLY.

Rev. of fan per min.	Slip of strap per cent.	Static press. at A in inches of water.	Dynamic press. at B in inches of water.	Velocity of the air at Z in ft. per min.	Cu. ft. of air per min.	Temperature of air at the end of the pipe by wet and dry thermometers in deg. Fah.	Barometric press. in inches of mercury.	Velocity of vane tips in ft. per sec.	Horse power absorbed by the fan itself.	Theoretical horse power required	Mechanical efficiency per cent.	Press. efficiency per cent.	Volumetric efficiency per cent.	Orifice $\frac{Q}{\sqrt{gH}}$ in sq. ft.	Equivalent orifice in sq. ft.	Baffles used.
847	4·8	$\frac{3}{4}$ vac.	$\frac{5}{8}$	3341	3560	61 wet, 76 dry	29·8	72·6	2·98	0·36	12·0	26·9	121·3	1·57	1·71	Free discharge.
857	4·6	$\frac{1}{2}$ vac.	$1\frac{3}{16}$	3319	3536	61 wet, 75 dry	30·2	73·6	2·67	0·46	17·1	33·6	119·0	1·39	1·52	One piece of 3-hole per inch woven wire.
887	3·4	$\frac{1}{8}$ vac.	$1\frac{1}{8}$	3100	3303	62 wet, $76\frac{1}{2}$ dry	30·2	76·1	2·73	0·58	21·3	42·8	107·5	1·11	1·20	One piece of 8-hole per inch woven wire.
1032	3·0	$1\frac{1}{8}$ press.	$2\frac{1}{8}$	3186	3395	53 wet, 62 dry	30·2	88·6	3·37	1·14	33·9	60·4	94·9	0·82	0·89	One piece of 30-hole per inch woven wire.
1173	5·0	$2\frac{1}{2}$	$3\frac{1}{2}$	2875	3064	64 wet, 76 dry	30·2	100·7	3·63	1·71	47·0	77·4	75·4	0·58	0·63	One piece of 54-hole per inch woven wire.
1291	3·7	$3\frac{7}{8}$	$4\frac{3}{4}$	2534	2700	44 wet, 54 dry	30·2	110·8	3·44	2·04	59·4	87·0	60·4	0·43	0·47	One sheet of perforated zinc. Best experiment.
1485	2·7	$5\frac{5}{8}$	6	1888	2012	63 wet, 78 dry	30·1	127·4	3·22	1·93	59·9	83·2	39·1	0·29	0·31	Two sheets of perforated zinc, superposed.
1553	2·7	$5\frac{3}{4}$	$5\frac{7}{8}$	1417	1510	54 wet, 65 dry	30·2	133·3	2·33	1·41	60·6	74·4	28·1	0·22	0·24	Three sheets of perforated zinc, superposed,
1525	1·3	$5\frac{1}{4}$	$5\frac{1}{2}$	726	773	46 wet, 52 dry	30 2	130·9	1·36	0·68	49·9	72·3	14·6	0·12	0·13	Four sheets of perforated zinc, superposed.
1527	5·0	$3\frac{13}{16}$	$3\frac{7}{8}$	—	—	—	30·1	131·1	1·41	—	—	50·0	—	—	—	Entirely blocked.

Radius of vanes 9·8 in. Area at the end of the pipe at Z less that of the wires, 1·066 sq. ft.

TABLE 8.—DONKIN'S BEST EXPERIMENT ON EACH OF THE ELEVEN TYPES OF FAN.

No. of fan.	Type of fan.	Dia. over vanes in inches.	Rev. of fan per min.	Static press. at outlet of fan A in inches of water.	Dynamic press. before baffle B, inches of water.	Cu. ft. of air per min.	I.H.P. of fan only.	Theoretical H.P.	Mechanical efficiency per cent.	Press. efficiency per cent.	Volumetric efficiency per cent.	Equiv. orifice in sq. ft.	Temperature of air at end of pipe F.	Barometric press. in mercury.	Direction of vanes.
I.	20 vanes, W.I., $1\frac{1}{2}$ in. wide at tip	$19\frac{5}{8}$	1291	$3\frac{7}{8}$	$4\frac{3}{4}$	2700	3·44	2·04	59·40	87·00	60·38	0·44	44 wet 54 dry	30·2	Concave
II.	12 vanes, C.I., $2\frac{1}{2}$ in. wide at tip	$25\frac{1}{2}$	1354	$4\frac{1}{2}$	$4\frac{7}{8}$	1905	3·31	1·47	44·49	48·68	18·65	0·33	75 wet $77\frac{1}{2}$ dry	30·08	Concave
III.	6 vanes, W.I., 7 in. wide at tip	$23\frac{1}{4}$	1248	$4\frac{1}{4}$	$4\frac{7}{16}$	1657	1·93	1·18	60·92	63·20	23·34	0·30	74 wet $82\frac{1}{2}$ dry	30·36	Radial
IV.	8 vanes, W.I., and 8 half vanes, $4\frac{1}{8}$ in. wide at tip	$23\frac{5}{8}$	1014	$4\frac{3}{16}$	$5\frac{3}{16}$	1261	3·16	1·05	33·13	52·65	20·75	0·21	55 wet 68 dry	29·8	Concave
V.	24 vanes, W.I., $3\frac{1}{2}$ in. wide at tip	$15\frac{3}{4}$	1500	$3\frac{3}{8}$	$3\frac{11}{16}$	2286	2·4	1·35	56·35	79·80	85·90	0·45	88 wet 91 dry	30·28	Concave
VI.	6 vanes, W.I.	20	1589	$3\frac{7}{8}$	$4\frac{1}{4}$	1636	2·39	1·11	46·59	50·49	28·33	0·30	69 wet 74 dry	29·70	Convex
VII.	6 vanes, W.I., $3\frac{1}{4}$ in. wide at tip	$24\frac{15}{16}$	1355	$3\frac{7}{16}$	$3\frac{3}{4}$	1526	2·47	0·91	36·95	39·02	15·93	0·30	61 wet 69 dry	30·20	Convex
VIII.	vanes, W.I., $1\frac{5}{8}$ in. wide at tip	24	1352	—	$3\frac{9}{16}$	1163	1·66	·66	39·57	39·59	13·82	0·24	56 wet 61 dry	30·50	Convex
IX.	4 vanes, C.I., $2\frac{1}{8}$ in. wide at tip	$24\frac{5}{8}$	1669	5	$5\frac{11}{16}$	1074	2·11	0·97	46·10	38·88	9·48	0·16	55 wet 63 dry	30·34	Concave
X.	18 vanes, 12 small, 6 large, $1\frac{3}{16}$ in. wide at tip	$23\frac{5}{8}$	1747	$9\frac{1}{8}$	$8\frac{7}{8}$	1280	3·22	1·82	56·57	59·25	12·22	0·17	57 wet 61 dry	30·00	Straight, but not radial
XI.	10 vanes, C.I., $\frac{3}{4}$ in. wide at tip	$15\frac{3}{4}$	2097	$1\frac{5}{16}$	$1\frac{5}{16}$	913	0·66	0·19	29·34	14·31	25·17	0·31	57 wet 63 dry	30·52	Concave

TABLE 9.—RESULTS OF EXPERIMENTS WITH MAXIMUM AND MINIMUM EQUIVALENT ORIFICES ON EACH FAN.

No. of Fan.	Baffle in pipe =hole wire, =sheets zinc.	R. p. M.	Max. and min. equiv-alent orifices.	Static press. at A in inches of water.	Dynamic press. at B in inches of water.	Velocity of air at end of pipe in ft. per min.	Quantity of air in cu. ft. per min.	Slip of strap per cent.	Mechanical efficiency per cent.	Press. efficiency per cent.	Volumetric efficiency per cent.
I.	3 wire	857	1·52	0·5 vac.	0·7	3,319	3,536	1·2	17·0	34·0	119·0
	4 zinc	1,525	0·13	5·75 press.	6·0	726	773	5·0	50·0	72·0	15·0
II.	3 wire	876	1·30	0·30 press.	1·2	3,523	3,755	4·4	25·0	29·0	57·0
	4 zinc	1,499	0·13	5·75 press.	6·0	1,063	1,134	2·8	45·0	49·0	10·0
III.	3 wire	980	1·43	0·16 press.	1·0	3,827	4,077	5·5	32·0	27·0	73·0
	4 zinc	1,088	0·17	5·10 press.	5·2	795	847	0·5	61·0	68·0	14·0
IV.	3 wire	706	1·44	1·00 vac.	0·2	1,842	1,962	5·0	3·0	5·0	46·0
	3 zinc	1,014	0·21	4·25 press.	5·0	1,183	1,262	0·7	33·0	53·0	21·0
V.	3 wire	1,074	1·26	0·05 press.	0·8	3,009	3,206	2·7	22·0	38·0	167·0
	4 zinc	1,056	0·22	3·30 press.	3·7	638	678	0·5	56·0	80·0	36·0
VI.	3 wire	1,289	1·42	0	0·5	2,409	2,566	3·5	10·0	8·5	55·0
	5 zinc	1,500	0·16	4·40 press.	4·7	785	837	5·5	46·0	55·0	15·0
VII.	3 wire	1,276	1·49	0	0·5	2,522	2,687	3·8	9·0	5·5	30·0
	4 zinc	1,185	0·21	4·00 press.	4·1	945	1,007	4·9	37·0	45·5	12·0
VIII.	3 wire	1,300	1·27	1·00 vac.	0·4	1,856	1,977	2·6	6·0	4·0	24·0
	4 zinc	1,186	0·16	3·00 press.	3·5	715	761	5·2	49·0	47·0	0·0
IX.	3 wire	1,184	1·23	0·67 press.	0·2	1,583	1,687	0·5	4·0	2·6	17·6
	4 zinc	1,669	0·17	5·00 press.	5·6	1,008	1,076	3·0	46·0	39·0	9·5
X.	3 wire	1,192	1·48	0 04 press.	0·5	2,663	2,840	7·0	10 5	8·0	40·0
	7 zinc	2,099	0·096	12¼ press.	12·3	815	867	10·0	56·6	61·0	7·0
XI.	3 wire	1,565	1·45	0	0	1,008	1,159	4·5	1·0	1·0	30·0
	3 zinc	2,541	0·24	2·67 press.	2·6	756	814	5·2	24·0	19·0	23·0

mean dynamic pressure at Z having been thus obtained, the mean velocity of the air was deduced from the formula (2) given in the list of formulæ used in the experiments. The mean velocity having been determined, the quantity of air in cu. ft. per sec. was obtained by multiplying the velocity by the area of the pipe. The latter was gauged inside the end of the pipe Z, allowance being made for the wire template. The temperature of the air at Z was noted in each experiment with a wet and dry bulb thermometer, and the barometric pressure was also observed.

The eleven types of fan used in Donkin's experiments were—

No. I., figs. 45, 44, was made with twenty specially-shaped wrought-iron vanes curved in two directions, and revolving with the concave side to the outlet. The casing, of volute form, gradually increased in cross-sectional area towards the outlet, and there was only one inlet for air of special bell-mouthed form. The driving shaft was fixed in the centre of the inlet, with a cone on its end to guide the air to the revolving blades; there was no bearing or obstruction to prevent the air from entering freely. The two brass bearings for the shaft, one with three thrust collars and one plain, measured $1\frac{3}{8}$ in. diameter by $5\frac{1}{4}$ in. long. This fan worked very quietly. Details of the ten experiments performed on this fan are given in Table 7. Little attention was paid to ensuring any particular speed in each experiment, as it was found in the fans tested that, all other conditions being the same, the quantity of air delivered was proportional to the speed. The three efficiencies of this fan are all high. Two special experiments were made with a sheet of perforated zinc in the pipe—one with the vanes varnished and covered with coal dust to represent dirty vanes in a coal mine, and the other with vanes clean and bright. Corrected for speed, the results of the comparison showed that $10\frac{1}{2}$ per cent. more air was delivered by the clean vanes, but the mechanical efficiency was about the same. The pressure efficiency was 11 per cent., and the volumetric efficiency 6 per cent. higher with clean than with dirty vanes. Two experiments were made, one with the large bell-mouthed inlet fixed in place as designed, and one with it removed. Corrected for speed, the result showed that 31 per cent. more air was passed when the inlet was used, and the mechanical efficiency was 9 per cent. higher. The pressure efficiency was 33 per cent., and the volumetric 15 per cent. higher. This proves the advantage of admitting the air without shock and in the right direction. In fig. 47 is given the variation of the pressures at the end of the pipe, taken with a Pitot tube connected with the dial water gauge, in an experiment on this fan. The pressure was taken in each of the eight equal areas. In this experiment, with two sheets of perforated zinc in the pipe, the mean of the eight readings

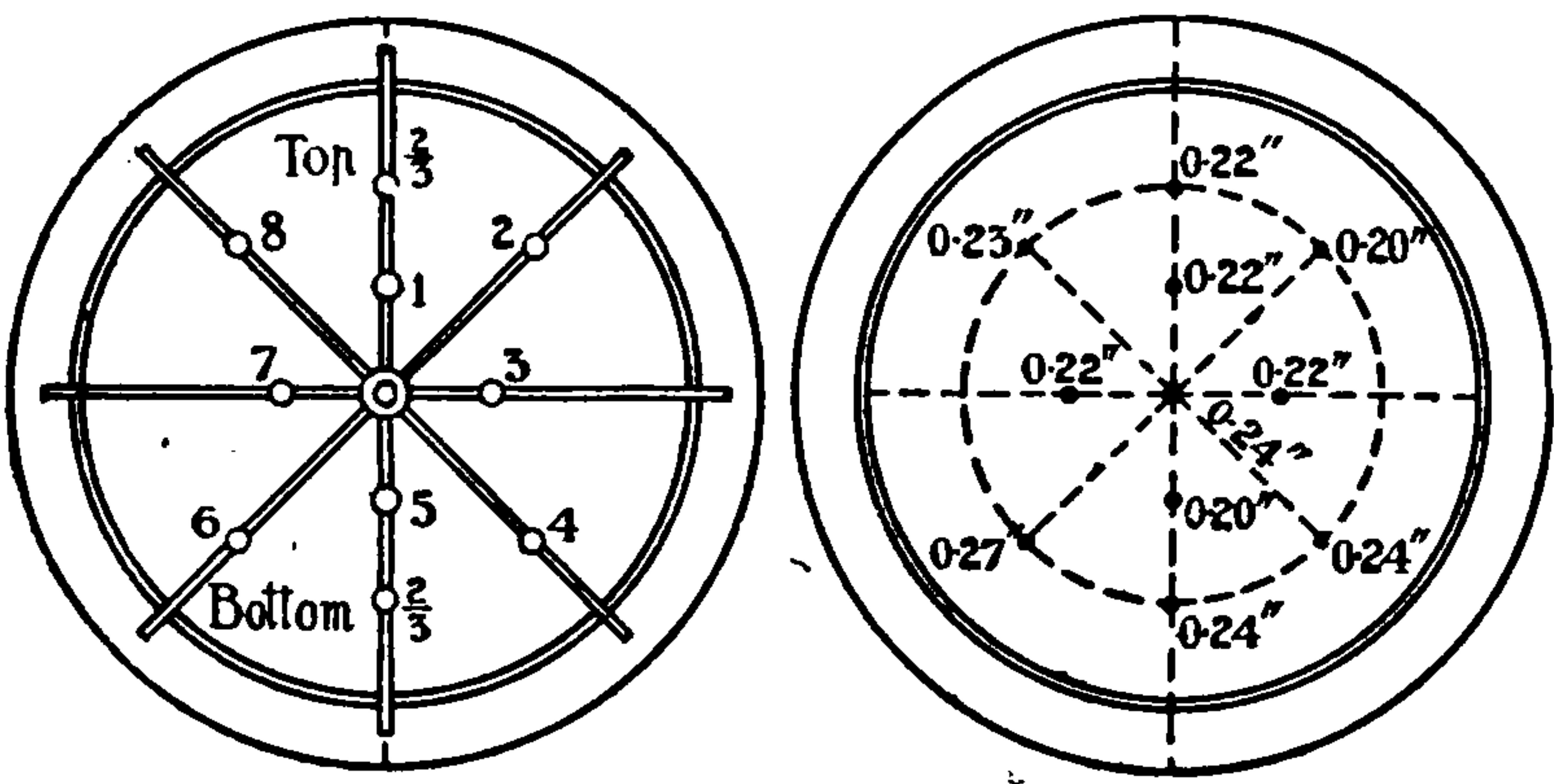


FIG. 47.—SUBDIVISION OF PIPE FREE-END.

End view of pipe at 3, showing points at which pressures were taken.

Cross-section of pipe, showing pressures at the eleven positions in inches of water.

gave a velocity of 1,888 ft. per min. (Table 7). The velocity obtained from the pressure at the centre of the pipe taken at the same time was 1,929 ft. per min., and that taken from the mean of the pressures at points two-thirds of the radius from the centre was 1,896 ft. per min.

No. II., figs. 45, 44, had twelve short curved vanes cast in one piece, revolving concave to the outlet. Some experiments, however, were made with the vanes running in the opposite direction to ascertain the effect on the pressures and quantities of air. The outer cast-iron casing was of volute form gradually increasing in area towards the outlet.

It was provided with two central air inlets. Two experiments were made with the vanes reversed and revolving convex to the outlet. The increase in the quantity of air delivered due to the vanes revolving concave to the outlet was found to be 11 per cent. The mechanical efficiency was 5 per cent. less in the latter case, but in the pressure efficiency a gain of $5\frac{1}{2}$ per cent. and in volumetric efficiency a gain of $2\frac{1}{2}$ per cent. were effected.

No. III., figs. 45, 43, was the simplest tested, and had only six short, straight radial wrought-iron vanes with two inlets for air ; the cast-iron casing was eccentric to the shaft, so that the cross-sectional area gradually increased towards the outlet. The fan worked quietly and the three efficiencies occupied a relatively good position.

No. IV., figs. 45, 44, was of a special duplex type, the air passing from an outer to an inner casing, and from thence to the outlet. The vanes were mounted on a central plate of wrought iron, so that one-half of them were in the outer and the other half in the inner casing. There were eight large and eight small vanes on each side of the centre plate revolving concave to the outlet. The bearings of white metal were three in number, $1\frac{1}{8}$ in. diameter and 5 in. long.

No. V., figs. 45, 44, was made with twenty-four short-curved wrought-iron vanes, intended to revolve with the concave side to the outlet. It was of the double flow type, being provided with two cones mounted on the shaft, and bell-mouthed inlets to direct the entering air. The outer casing was of cast-iron of volute form, the area increasing gradually to the outlet. The two bearings, arranged to allow for any deflection of the shaft, were $1\frac{1}{16}$ in. in diameter by $4\frac{5}{8}$ in. long, and were placed well away from the inlets. It will be seen that the three efficiencies in the best experiment are all relatively high. One experiment was made with the two external bell-mouthed inlets removed, the directions of the vanes remaining the same. The gain—due to these inlets—was $3\frac{1}{2}$ per cent. in the quantity of air delivered, $4\frac{1}{4}$ per cent. in mechanical efficiency, and $5\frac{3}{4}$ per cent. in pressure efficiency.

No. VI., figs. 45, 43, had six vanes, but their curvature was not continuous ; at about the middle they were set back and

No. IX., figs. 45, 44, represents a type used in considerable numbers on the Continent. Four vanes, an unusually small number, of thin cast iron are provided, and are intended to revolve concave to the outlet. There are two inlets for the air, placed centrally to the shaft and also to the outer casing, which was of cast iron. The two bearings, with white metal linings, were 1 in. in diameter by 5 in. long. These were situated close to the inlets. The fan was always noisy, especially at small orifices.

No. X., figs. 45, 43, was designed especially to deliver a small quantity of air at a high pressure. There were in all eighteen vanes, six of which were whole vanes, with twelve half vanes interposed. They were all cast in one piece and straight, but not set radially to the centre. There were two inlets centrally with the shaft, coned to guide the entering air. The boss of the revolving part was also curved to assist the entrance of the air. The outer casing was of cast iron, set concentric with the shaft and of rectangular section, gradually increasing in area to the outlet. The bearings were $1\frac{3}{16}$ in. diameter and 6 in. long, placed well away from the air inlets. Thirteen experiments were made with a different number of baffles. It will be seen that this fan gave fairly good results compared with others, but it must be remembered that it was designed to produce pressure. One experiment was made with the vanes running in a contrary direction to that shown in the figure. The results corrected for speed showed that the quantity of air delivered was 31 per cent. more, the mechanical efficiency 4 per cent. less, the pressure efficiency $1\frac{1}{2}$ per cent. less, and the volumetric efficiency 3 per cent. higher. (This last is difficult to understand, for if the discharge was 31 per cent. greater, the volumetric efficiency, which varies as the discharge and inversely as the speed for a given fan at a given orifice, should also be 31 per cent. more.) The fan worked quietly. An attempt was made to obtain the pressure and velocity of the air between the casing and the revolving blades when running at 1,758 rev. per min., and with three sheets of perforated zinc superposed in the pipe. A Pitot tube was held against the current close to the edge of the revolving blades, and also as near the inside of the casing as possible. The pressure was found to be higher when close to the vanes. At D, No. X., fig. 45, the dynamic pressure when the tube was held close up to the vanes was $11\frac{1}{8}$ in. of water, corresponding with a velocity of 13,080 ft. per min., and with the tube as near the outer casing as possible the pressure was $9\frac{15}{16}$ in. of water, or 12,500 ft. per min. At E, the opposite point on the circumference, the dynamic pressure when the tube was held close to the vanes was $11\frac{7}{8}$ in. head of water, or 13,690 ft. per min., and with the tube as near the outer casing as possible $11\frac{1}{2}$ in., or 13,480 ft. per min. Thus it will be seen that the velocity of the air increased round the inside of the casing with the direction of rotation for the same speed of fan. (If we can judge from the figure, the increase of section round the casing was not proportional to the angle, so that the ratio of the section to the quantity of air passing through it gradually decreased towards the outlet, and this would account for the increase of velocity mentioned. The less velocity near the outside of the casing was probably due to skin friction.) The static pressure at A in the experimental apparatus, fig. 46, at the same time was $9\frac{1}{8}$ in. of water, and the quantity of air delivered was 1,291 cu. ft. per min.

No. XI, figs. 45, 44, was made with blades fixed on one side only of a disc, having ten cast-iron slightly-curved thick vanes revolving concave to the outlet. It had only one inlet, which was so constructed that the air entered, not parallel with the fan shaft, as in all the other fans tested, but at right angles to it. In this way a rotary motion was given to the air before it came in contact with the revolving vanes. The object of this arrangement was to minimise friction. The experiments, however, show that it was of little use. The outer casing was of cast iron of volute form. There were two cast-iron bearings, $1\frac{1}{4}$ in. diameter, about 8 in. long. Nine experiments were made with the usual baffles. It will be seen that the three efficiencies occupy a low position in the plotted results shown in figs. 48, 49, 50.

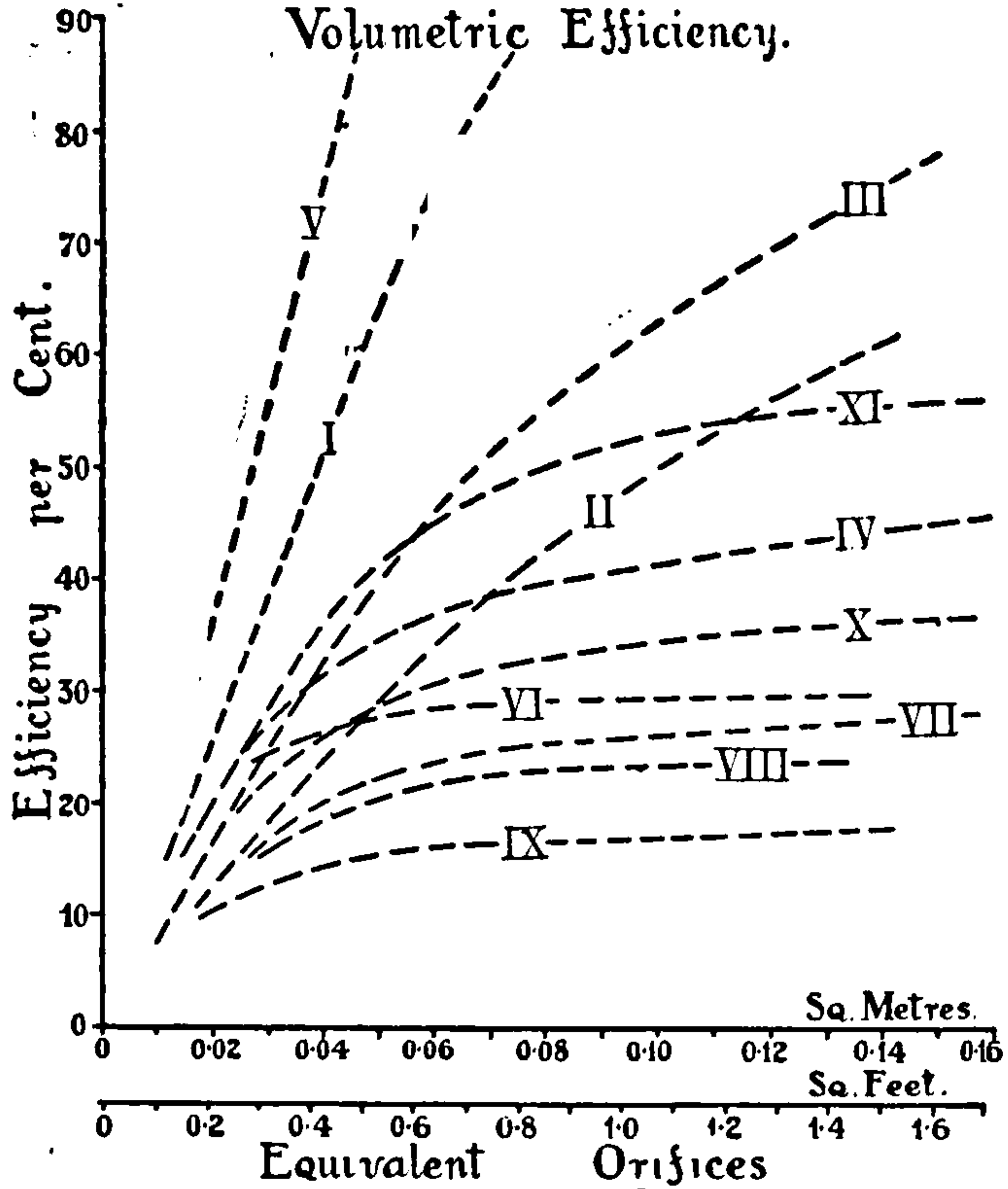


FIG. 48.

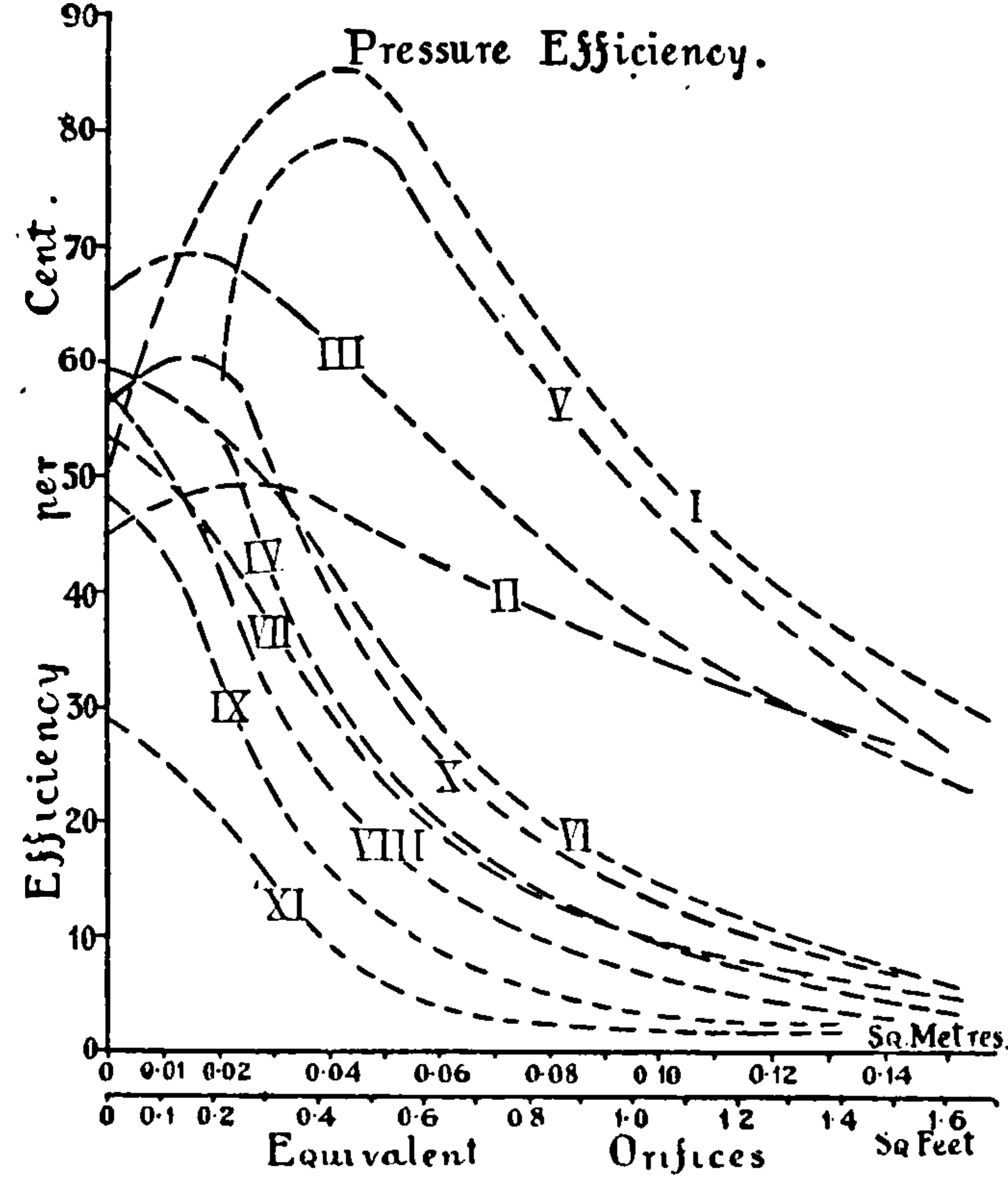


FIG. 49.

EFFICIENCIES OF FANS: DONKIN'S TESTS.

The roman numerals refer to the types of fan illustrated in fig. 45, p. 86.

space between them and the outer casing exercise a considerable influence on the various efficiencies. The final inclination or angle of the vanes at their circumference has more effect on the pressure of the air, and less on the mechanical and volumetric efficiencies. The revolving portion of the fan should always be accurately balanced. Donkin recommends continuous lubrication at high speeds to reduce journal friction, and allowance should be made for the deflection of the spindle. The pulley should not be too small or narrow, so as to reduce the slip of the strap. The friction of the air inside the casing is often excessive, and care should be taken to allow its entrance and passage through the vanes and out of the fan with a minimum of skin friction. Changes of direction and shocks should be avoided as much as possible.

Formulæ used in Donkin's Experiments.—These formulæ are expressed in metric measures, which were used by Donkin. The results are tabulated in English measures to accord with the text.

$$H = \frac{h}{w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

where H = metres of air, h = pressure in millimetres of water, w = weight of 1 cubic metre of air in kilograms at the atmospheric pressure and temperature.

$$V = 4\sqrt{h} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

where V = velocity of air in metres per sec. at the end of the pipe Z, and h = pressure in millimetres of water.

$$\left.\begin{array}{l}\text{Theoretical H.P.*}\\ \text{in chevaux-vapeur}\end{array}\right\} = \frac{Q \times w \times H}{75} \quad . \quad . \quad . \quad (3a)$$

* The writer does not make it clear whether or no the H.P. required to drive the fan shaft is also in chevaux-vapeur. One cheval-vapeur = 0·986 English H.P.

where Q = quantity of air delivered in cubic metres per sec.

$$\left.\begin{array}{r}\text{Mechanical}\\ \text{efficiency}\end{array}\right\} = \frac{\text{theoretical H.P.}}{\text{H.P. required to drive fan shaft}} \quad (4a)$$

$$\text{Pressure efficiency} = \frac{g\,H}{{v_2}^2} \quad . \quad . \quad . \quad . \quad (5a)$$

where $g = 9{\cdot}81$ metres, per sec., per sec., and v_2 = peripheral speed.

$$\text{Volumetric efficiency} = \frac{Q}{v_2 r_2^2} \quad . \quad . \quad . \quad (6a)$$

where r_2 = external radius in metres.

$$\text{Orifice in square metres} = \frac{Q}{\sqrt{g\,H}} \quad . \quad . \quad . \quad (7a)$$

$$\text{Equivalent orifice (in square metres)} = \frac{Q}{0{\cdot}65\sqrt{2\,g\,H}} \quad (8a)$$

$$= \text{Orifice} \times 1{\cdot}088.$$

Appended to this was a short account of the experiments made by the Prussian Mining Commission in 1884 on the measurement of air in a pipe by different methods, from a large air-holder at the Breslau gasworks.

Prussian Mining Commission Tests (1884).—A spare gas-holder was used for measuring the volume of the air. It was 85·3 ft. in dia., contained 70,634 cu. ft., and served to check the other methods adopted by the Commission. The tests were probably the best that have been published on the measurement of air by the following methods: (1) by anemometers; (2) by Pitot tubes; (3) through circular and square orifices. The practical questions the Commission endeavoured to solve by using this holder and causing the air to pass through a pipe were the following:—

1. Do the formulæ generally used for standardising anemometers in a circular path in still air give correct results?

2. Can the instrument known as the Pitot tube be applied practically for measuring the speed of air; and, if so, what formula should be used for calculating the speed and quantity of air?

3. May the fall in pressure between one side and the other of a thin orifice interposed in a pipe be used for calculating the quantity of air; and, if so, what formula should be applied?

4. What is the loss of head due to friction in regard to length and diameter of pipe used?

About eighty careful experiments were made, and the results and calculations appear to have been well checked. The cast-iron pipe was 14·3 in. dia. and 33 ft. long. For stopping and starting the anemometers quickly and accurately an electrical arrangement was adopted. The vertical fall of the air-holder in several places was carefully determined electrically. The first series of experiments in 1884 were made with the air-holder at a water pressure of $2\frac{7}{8}$ in., but in the 1885 tests the holder was loaded and the pressure was increased to $4\frac{1}{2}$ in. of water. The density and temperature of the air were noted, and the experiments were made during the autumn to avoid the heating effect of the sun upon the wrought-iron holder. U-gauges were fixed at different parts of the pipe. A Pitot tube was used for measuring the dynamic pressures of air, not only at the centre and at two-thirds of the radius distant from it, but also round the inner circumference of the pipe. The circular orifices used in these experiments measured 7·03 in. and 9·96 in. dia. The square orifice measured 6·26 in. along the side. The rectangular orifice was 9·17 in. by 4·45 in. The experimental coefficient determined for the circular orifice was 0·64, and for the square and rectangular orifices 0 61. Four Casella anemometers and one Robinson anemometer were tested. The Casella anemometers, previously tested in the usual way at the end of a radius bar and compared with direct measurement of air from the holder, showed variable errors, the excess ranging between 7 and 13 per cent. Anemometer readings should therefore be accepted with caution. In the 14·3 in. cast-iron pipe a considerable difference in speed was found at different parts and in the same vertical plane. The centre gave the maximum speed and pressure and the inner circumference the minimum; the mean speed of the air was found to be at two-thirds of the radius from the centre of the pipe. With regard to the resistance to the movement of the air in the pipe used, the following are the con elusions deduced from these experiments and given in the report: (1) that the resistance of the air increases as the square of its speed in the pipe; (2) the resistance to the air in the pipe decreases as the diameter of the pipe to the

although the earliest used, may be found ventilating many mines at the present day, both in this country and abroad. Its wheel is generally of considerable diameter, carrying a number of vanes, and enveloped over most of the circumference, allowing the air to escape by a single opening, regulated by a shutter to suit the orifice of the mine. The air enters the eye, and by its centrifugal action it reaches the circumference and passes out at the chimney. The vanes of Guibals are sometimes plane, and inclined in the opposite direction to that of rotation, but are usually

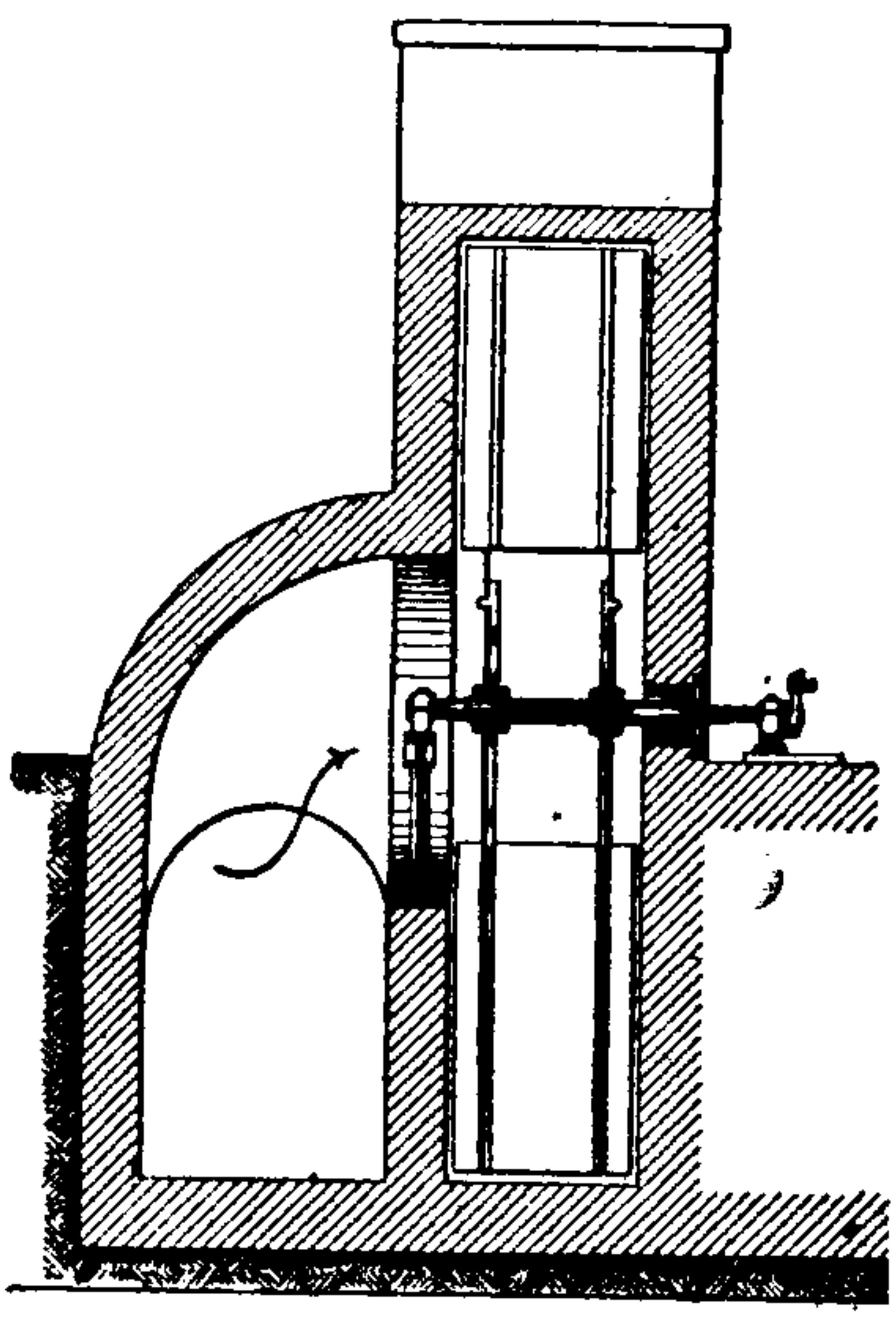


FIG. 51.

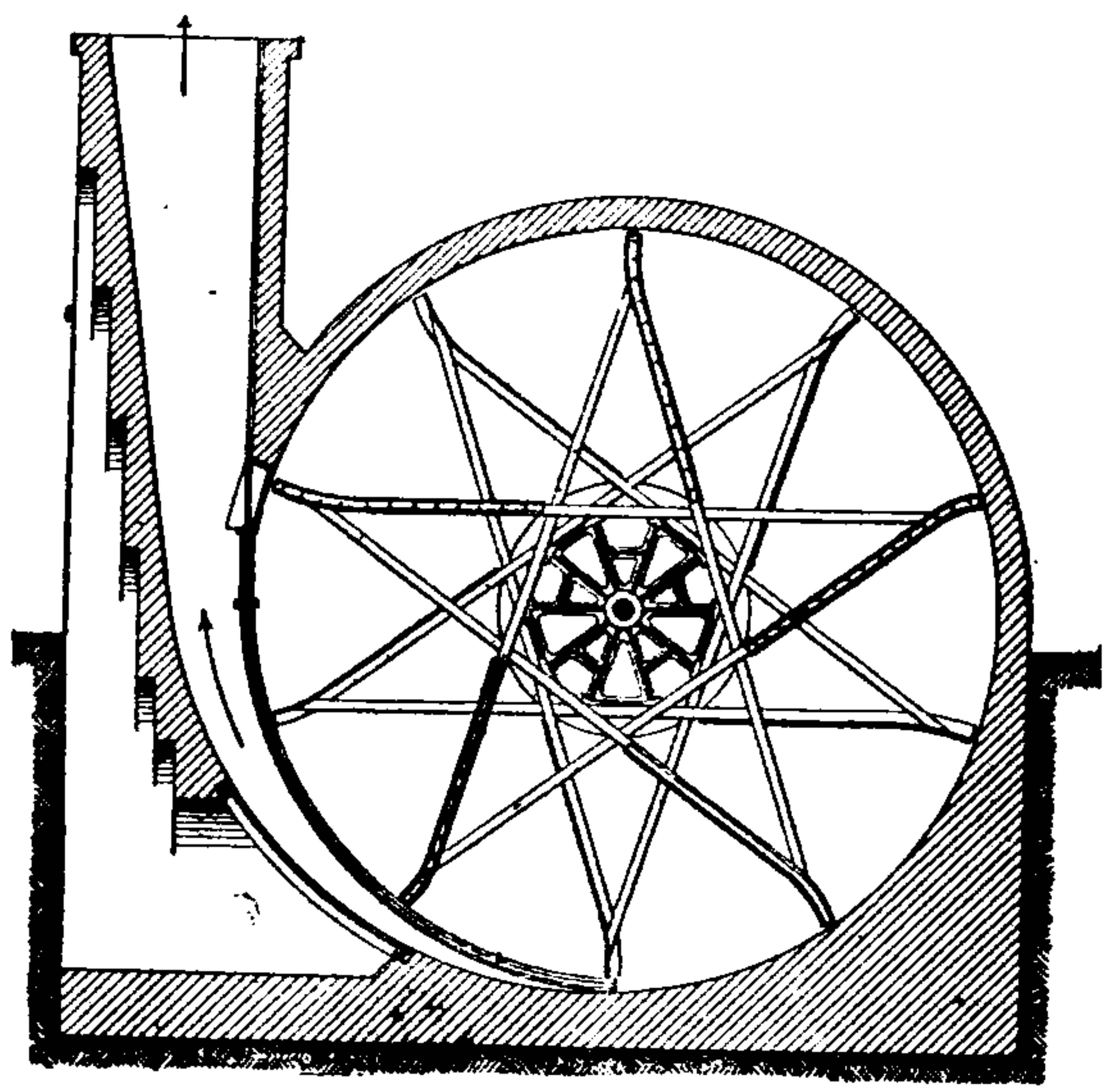


FIG. 52.

GUIBAL FAN.

curved near the outer extremity until they become radial. The number of vanes lies between six and ten for sizes varying between 19 ft. and 40 ft. The breadth of wheel for these diameters lies between $4\frac{1}{2}$ ft. and 10 ft. The chimney expands from the wheel to the mouth in order to reduce the velocity and increase the pressure of the air. Figs. 51, 52, 53, 54 show two examples of Guibals, the first of which is $39\frac{1}{3}$ ft. in diameter, and the latter 19 ft. The breadth of wheel of the first is $8\frac{1}{4}$ ft., and of the latter 6·4 ft. The Ser fan, figs. 55, 56, was designed in 1878 by

Professor Ser, of the Ecole Centrale of Paris. The theory of this fan is published in the "Mémoires de la Société des Ingenieurs Civils for 1878." It consists of a circular plate fixed to the shaft and carrying on each side 32 curved vanes, each of which forms part of a cylindrical surface whose generatrices are parallel to the shaft, and whose transverse section is circular. Their width is constant, and it is arranged that inflow shall take place without shock.

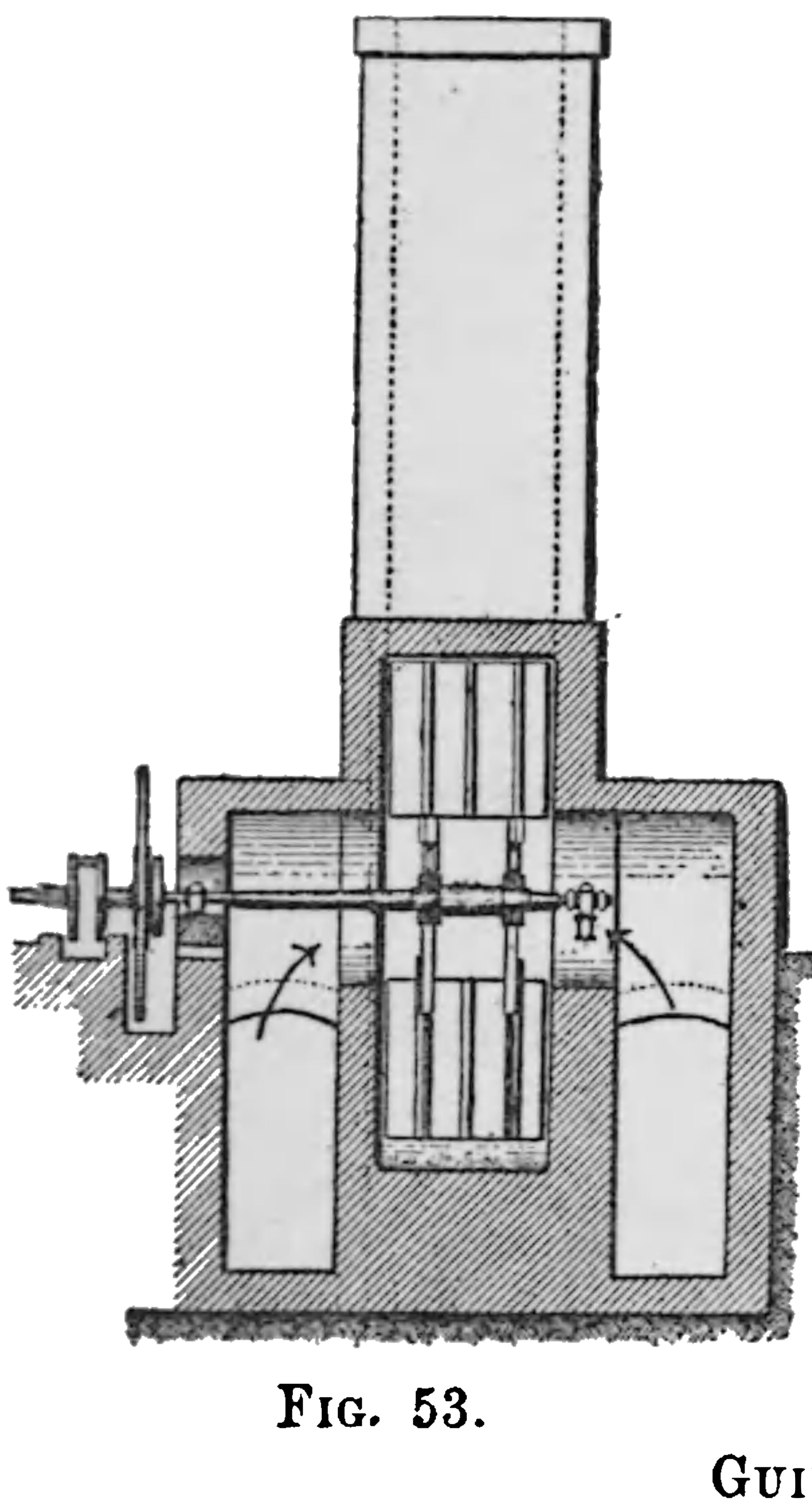


FIG. 53.

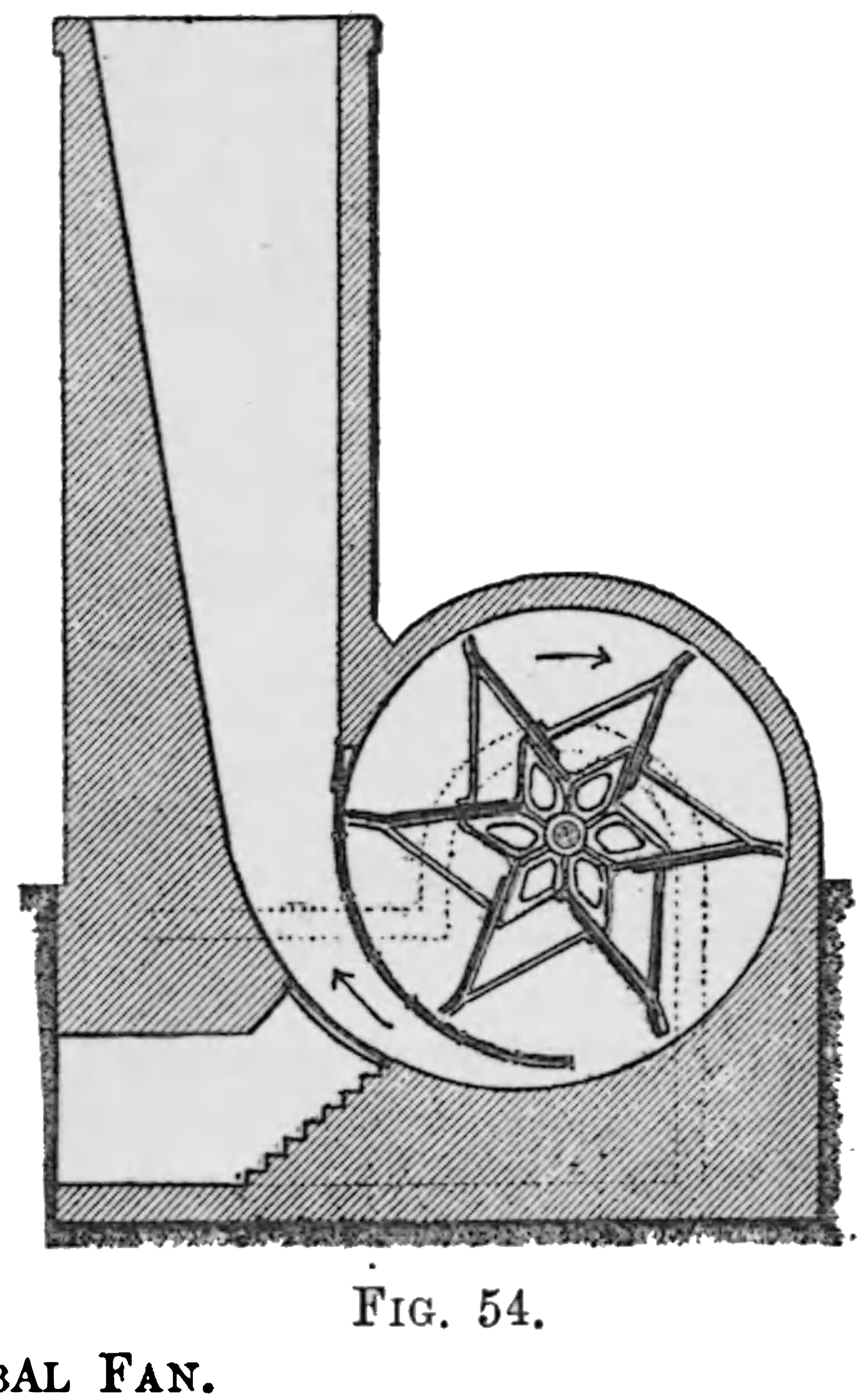
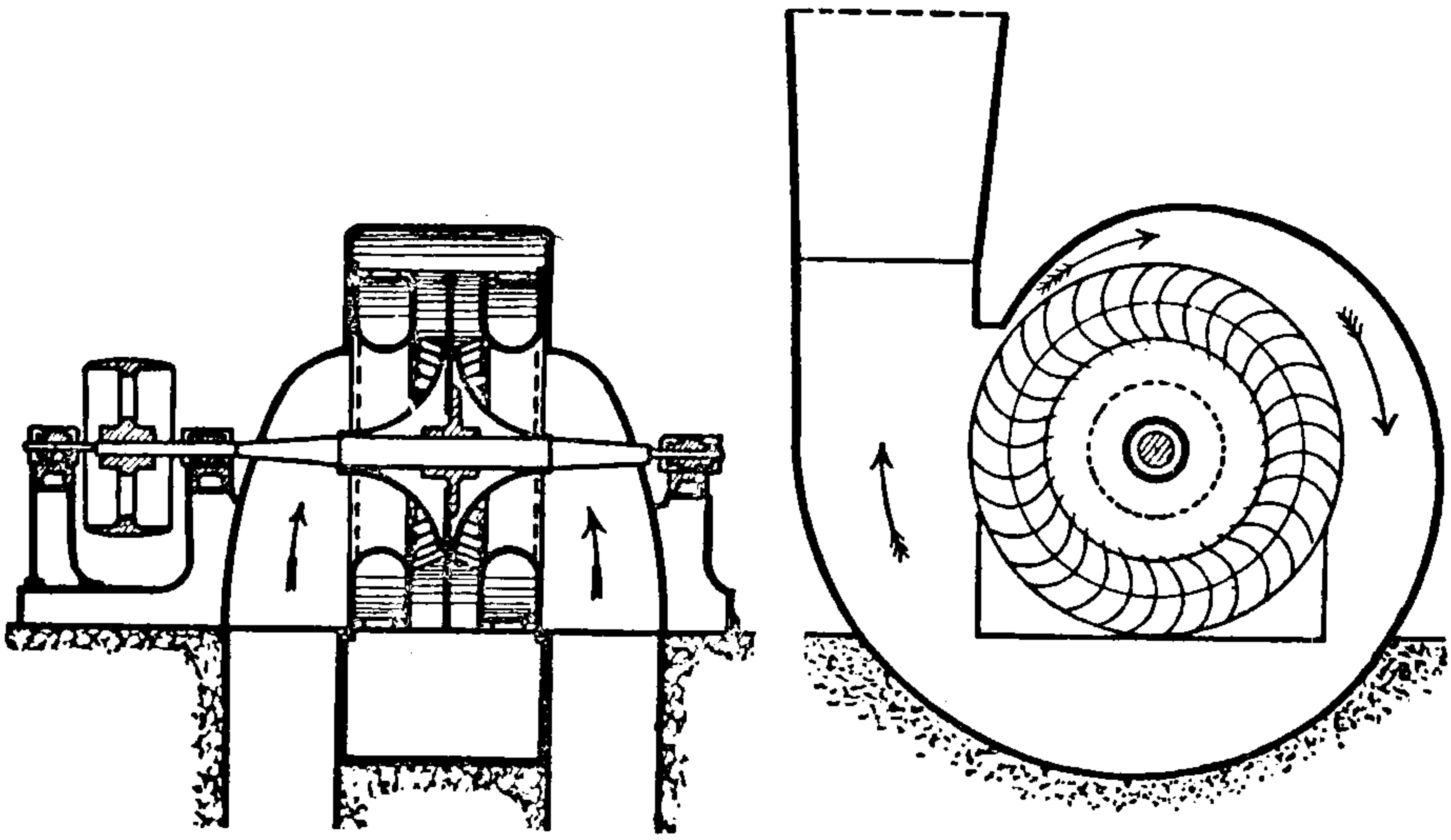


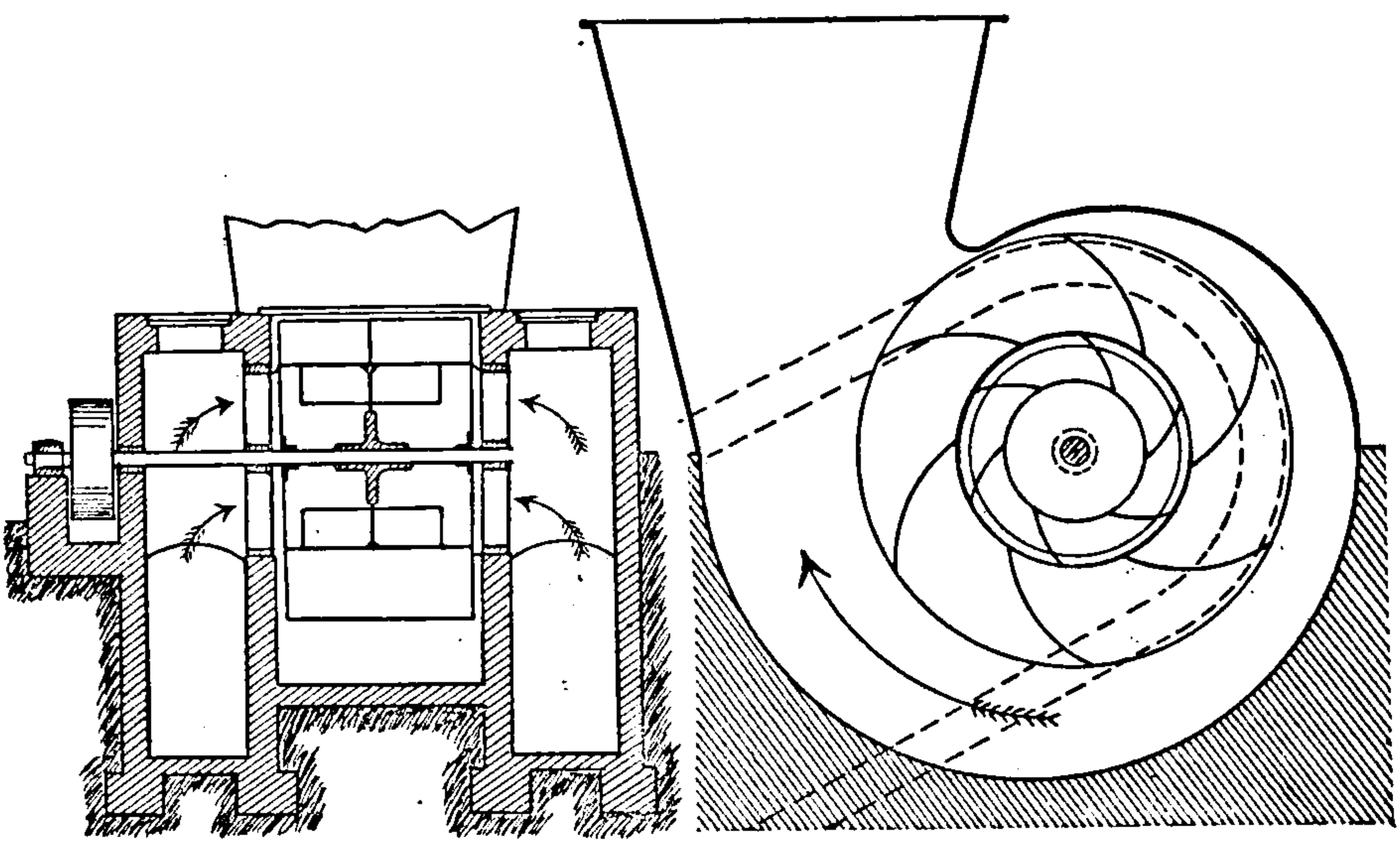
FIG. 54.

GUIBAL FAN.

The relative direction of outflow is at an angle of 45° with the radius. Inflow takes place at both sides, and the fan is provided with a volute and expanding chimney. The volute is so designed that the loss of energy at entrance from the circumference of the fan is a minimum, and the sides of the chimney are inclined at not more than 1 in 8 to avoid the loss due to sudden enlargement of passage. This type of fan is made in sizes varying from 4·6 ft. to 8·2 ft. The Capell fan (figs. 57, 58) is formed of two fans, one outside the other. The inner fan consists of a drum of steel plate, closed on one side if there is a single



FIGS. 55 AND 56.—SER FAN.



FIGS. 57 AND 58.—CAPELL FAN.

eye. Its diameter is that of the eye. The cylindrical surface contains six openings, usually rectangular, spaced equally round the circumference, having an area not less

TABLE 10.—EXPERIMENTS WITH GUIBALS.

No. of fan.	Rev. per min. of engine.	Rev. per min. of fan.	Tip speed in ft. per sec. v_2.	Engine H.P. in chevaux-vapeur. T.	Velocity of air in ft. per sec.	Cu. ft. of air per sec. Q.	Water gauge in inches. h.	Useful H P. done on air in chevaux-vapeur. t.	Equiv. orifice in sq. ft.	Mechanical efficiency per cent. $\frac{t}{T}$.	Mean manometric efficiency per cent.	Mean mechanical efficiency per cent.	Volumetric efficiency per cent. $\frac{Q}{v_2 r_2^2}$.
1	46	46	94·75	77·6	40·6	1960	2·16	42·18	31·7	54·3	—	—	—
	56	56	101·5	140·5	45·2	2180	3·31	71·82	28·6	51·1	53·9	52·5	—
	65	65	118·0	205·5	53·0	2560	4·21	107·34	29·6	52·2	—	—	5·6
2	50	50	103·0	59·9	20·2	837·5	2·87	23·94	11·7	39·9	—	—	—
	60	60	109·0	89·7	23·8	987·5	4·10	40·21	11·6	44·8	60·7	42·4	—
	70	70	127·0	152·1	27·2	1130	5·79	64·99	11·2	42·7	—	—	2·3
3	40	64	98·75	58·5	14·6	961	3·07	30·40	13·5	52·0	—	—	—
	47	75	115·5	89·5	16·2	1100	4·01	46·78	13·1	52·2	66·7	50·4	—
	53	85	131·0	118·1	17·0	1155	4·84	55·66	12·5	47·1	—	—	4·05
4	100	100	99·5	82·0	19·5	1565	3·15	49·06	20·9	59·8	—	—	—
	106	106	105·0	100·4	20·9	1670	3·54	59·78	21·4	59·5	69·4	60·00	—
	120	120	119·0	125·2	22·4	1800	4·26	76·14	20·9	60·8	—	—	16·75

TABLE 11.—EXPERIMENTS WITH SER FANS.

No. of fan.	Rev. per min. of engine.	Rev. per min. of fan.	Tip speed in ft. per sec. v_2.	Engine H.P. in chevaux-vapeur. T.	Velocity of air in ft. per sec.	Cu. ft. of air per sec. Q.	Water gauge in inches. h.	Useful H.P. done on air in chevaux-vapeur. t.	Equiv. orifice in sq. ft.	Mechanical efficiency per cent. $\frac{t}{T}$.	Mean manometric efficiency per cent.	Mean mechanical efficiency per cent.	Reduced equiv. orifices. $\frac{Q}{0{\cdot}65\sqrt{2gH}\times r_2^2}$.	Volumetric efficiency per cent. $\frac{Q}{v_2 r_2^2}$.
1	65	336	80·5	10·65	8·91	304·5	1·45	4·27	5·81	40·1	—	—	—	—
	75	391	93·6	16·86	10·64	364·0	1·85	6 47	6·12	38·4	47·7	40·5	—	—
	85	435	115·2	22·13	12·90	441 5	2·24	9·52	10·22	43·0	—	—	0·63	35·7
2	63	302	83·0	25·43	10·40	689	1·93	12·78	11·39	50·3	—	—	—	—
	73	345	94·7	37·63	12·64	840	2·68	21·60	11·81	57·4	64·3	54·6	—	—
	80	377	103·2	46·96	13·45	894	3·07	26·37	11·70	56·2	—	—	1·704	126·0
3	57	302	103·5	45·68	23·7 5·15	1260	1·61	19·60	22·9	42·9	—	—	—	—
	67	365	121·5	75·70	27·2 5·95	1455	2·36	33·16	21·7	43·8	34·5	44·9	—	—
	72	382	131·1	90·12	31·5 6·85	1685	2·68	43·35	23·6	48·1	—	—	4·48	244·0
4	36	122	52·3	17·36	10·13	875	0·94	7·94	20·6	45·7	—	—	—	—
	46	154	66·0	28·55	12·60	1090	1·38	14·43	21·4	50·5	73·4	48·0	—	—
	56	186	79·6	60·71	16·00	1385	2·09	29·08	22·1	47·9	—	—	1·312	103·5

TABLE 12.—EXPERIMENTS WITH CAPELL FANS.

No. of fan.	Rev. per min. of engine.	Rev. per min. of fan.	Tip speed in ft. per sec. v_2.	Engine H.P. in chevaux-vapeur. T.	Velocity of air in ft. per sec.	Cu. ft. of air per sec. Q.	Water gauge in inches. h.	Useful H.P. done on air in chevaux-vapeur. t.	Equiv. orifice in sq. ft.	Mechanical efficiency per cent. $\frac{t}{T}$.	Mean mano-metric efficiency per cent.	Mean mechanical efficiency per cent.	Volumetric efficiency per cent. $\frac{Q}{v_2 r_2^2}$.
1	46	184	118	64·42	20·5	1198	3·39	39·05	15·00	60·6	—	—	—
	56	224	144	95·08	25·4	1485	4·33	61·83	16·50	65·0	49·1	63·4	—
	66	264	170	150·14	28·3	1656	6·11	97·14	15·35	64·7	—	—	25·8
2	40	160	69·7	6·69	11·7	469	1·06	4·69	10·2	70·1	—	—	—
	50	200	85·7	14·84	15·7	62	1·89	10·97	10·3	73·9	53·4	69·8	—
	60	240	103	24·96	17·3	76	2·52	16·39	9·4	65·6	—	—	39·1
3	36	160	99	30·72	14·7	56	2·32	15·12	10·2	49·2	—	—	—
	40	178	110	41·11	16·6	67	2 83	20·88	10·4	50·8	50 9	48·2	—
	44	196	121	53·41	17·3	80	3·11	23·86	10·2	44·7	—	—	19·0

TABLE 14.—DIMENSIONS, COST, AND SERVICE OF GUIBAL FANS.

	No. of Fan.			
	1.	2.	3.	4.
Date installed	Jan. '77	Aug. '82	Sep. '86	Apl. '91
Period of service to January 1, 1892	15 yrs.	9 yrs. 4 mths.	5 yrs. 4 mths.	9 mths.
Total cost of plant in £	1,240	1,200	1,236	1,120
Cost of fan alone in £...	310	256	270	180
Dia. of wheel in ft. ...	39·3	39·3	29·5	19
Breadth of wheel in ft.	8·2	8·2	6·89	6·39
Dia. of eye	13·1	13·1	9·82	6·88 2 eyes
Height of chimney above centre of shaft	35·7	32	22·9	34·4
Length of mouth of chimney	8·8	11·2	7·7	10·65
Breadth of mouth of chimney	8·2	8·5	6·9	6·56
Dia. of shaft in inches ...	12·5	10·2	11·0	11·4 and 6·3
Length of shaft between bearings in ft.	14·5	12·5	12·3	15·5
Pulley dia. in ft. ...	None	None	8·2	None
Weight of moving parts in lb.	44,000	39,600	20,650	14,050
Number of steam cylinders	1	1	1	1
Dia. of piston in inches	26·7	24·4	24·4	16·5
Stroke in inches ...	33·4	39·4	33·4	29·5
Dia. of driving pulley in ft.	None	None	13·1	None
Distance between shaft centres in ft.	None	None	26·2	None
Size of engine house in ft.	42·6 × 13·1	—	55·7 × 16·4	10·6 × 32·8

TABLE 15.—DIMENSIONS, COST, AND SERVICE OF SER FANS.

	No. of Fan.			
	1.	2.	3.	4.
Date installed	June '84	July '87	June '88	Jan. '91
Period of service to January 1, 1892	7 yrs. 6 mths.	4 yrs. 5 mths.	3 yrs. 6 mths.	1 yr.
Total cost of plant in £	1,280	820	720	1,400
Cost of fan alone in £ ...	480	300	240	414
Dia. of wheel in ft. ...	6·56	5·25	4·59	8·2
Breadth of wheel in ft.	1·18	0·917	0·786	1·47
Dia. of eye	3·84	3·11	2·79	4·91
Height of chimney above centre of shaft	29·5	19·65	22·6	36·9
Length of mouth of chimney	7·87	5·25	5·15	7·87
Breadth of mouth of chimney	7·71	5·25	5·15	7·71
Dia. of shaft in inches ...	6·7	5·1	4·32	7·46
Length of shaft between bearings in ft.	9·7	6·46	5·85	9·25
Pulley dia. in ft. ...	3·28	2·13	1·64	3·93
Weight of moving parts in lb.	1,192	—	—	1,480
Number of steam cylinders	1	1	1	2 tandem
Dia. of piston in inches	19·6	13·8	11·8	15·1 and 19·7
Stroke in inches ...	27·5	29·5	25·6	31·5
Dia. of driving pulley in ft.	17·3	10·5	8·85	13·1
Distance between shaft centres in ft.	24·6	23	19	19·7
Size of engine house in ft.	26·2 × 23	26·2 × 23	26·2 × 19·6	26·2 × 19·6

TABLE 16.—DIMENSIONS, COST, AND SERVICE OF CAPELL FANS.

	No. of Fan.		
	1.	2.	3.
Date installed	Mar. '89	July '89	July '91
Period of service to January 1, 1892	2 yrs. 10 mths.	2 yrs. 6 mths.	6 mths.
Total cost of plant in £ ...	1,440	700	1,380
Cost of fan alone in £ ...	360	210	300
Dia. of wheel in ft. ...	12·3	8·2	11·8
Breadth of wheel in ft. ...	6·56	5·9	5·25
Dia. of eye	6·87 2 eyes	4·59	7·2 2 eyes
Height of chimney above centre of shaft	12·45	7·2	14·25
Length of mouth of chimney	10·5	9·2	11·9
Breadth of mouth of chimney	8·2	7·87	7·2
Dia. of shaft in inches ...	10·4 and 7·86	7·86	7·86 and 7·06
Length of shaft between bearings in ft.	21·5	18	18
Pulley dia. in ft.	3·6	2·46	2·95
Weight of moving parts in lb.	26,400	15,400	22,000
Number of steam cylinders	2 (cranks at right angles)	1	1
Dia. of piston in inches ...	20·5	15·7	17·7
Stroke in inches	31·5	23·6	29·5
Dia. of driving pulley in ft.	14·4	9·8	13·1
Distance between shaft centres in ft.	32·8	29·5	24·6
Size of engine house in ft.	23 × 29·5	36·1 × 12·1	26·2 × 22 9

Experiments with Rateau Fans.—The only experiments that we know of, made with a variable number of vanes, are those by M. Rateau, described in his work "Considerations sur les Turbo-Machines," in which a small fan 0·82 ft. in diameter was tested for water gauge with 18, 24, and 30 vanes. The number of vanes that gave the best result was 18, but the difference was trifling. The highest water gauges were 53·4, 53, and 50·8 millimetres, and the lowest 46·5, 48·9, and 44, with 18, 24, and 30 vanes respectively. Unfortunately, no tests were made of mechanical efficiency. It is extremely probable that a fewer number of vanes would give a better mechanical efficiency, although the water gauge would probably fall. The experiments in Table 18 were made with a Rateau fan having a diffuser, but no volute. Inflow took place from

TABLE 18.—RATEAU FAN TEST WITH DIFFUSER ONLY.

DIAMETER, 4·59 FT.

No. of experiment.	Rev. per min.	Cu. ft. of air per sec. Q.	Work done on air in H.P. (French).	Work done by engine in H.P. (French).	Mechanical efficiency per cent.	Reduced orifices. $Q \div r_2^2\sqrt{g\,H}$.	Manometric efficiency per cent.	Water gauge. In.
1	405	317	9·5	19	0·50	0·71	0·75	3·15
2	405	454	13·9	25·3	0·55	1·02	0·76	3·19
3	405	549	16·0	26·6	0·60	1·26	0·72	3·13
4	510	690	31·3	45	0·70	1·27	0·69	4·72
5	411	641	18·2	37	0·49	1·49	0·64	2·95
6	518	802	35·0	55·8	0·63	1·50	0·63	4·52
7	414	682	17·1	42·4	0·40	1·70	0·57	2·6
8	481	791	27·0	61·9	0·44	1·68	0·57	3·54
9	405	752	15·6	48·2	0·36	2·03	0·48	2·16
10	458	850	23·1	65	0·36	2·02	0·49	2·84
11	402	827	14·1	43·6	0·32	2·50	0·43	1·77
12	425	872	16·8	51·9	0·32	2·47	0·43	2·01

one side, and the wheel diameter was 4·59 ft. (fig. 59). The trials were very carefully made, as we see that, although experiments 3 and 4, 5 and 6, 7 and 8, 9 and 10, 11 and 12 were made at different revolutions, but each pair with the same baffle, the reduced orifices, for 3 and 4, are very nearly the same, as also their manometric efficiencies;

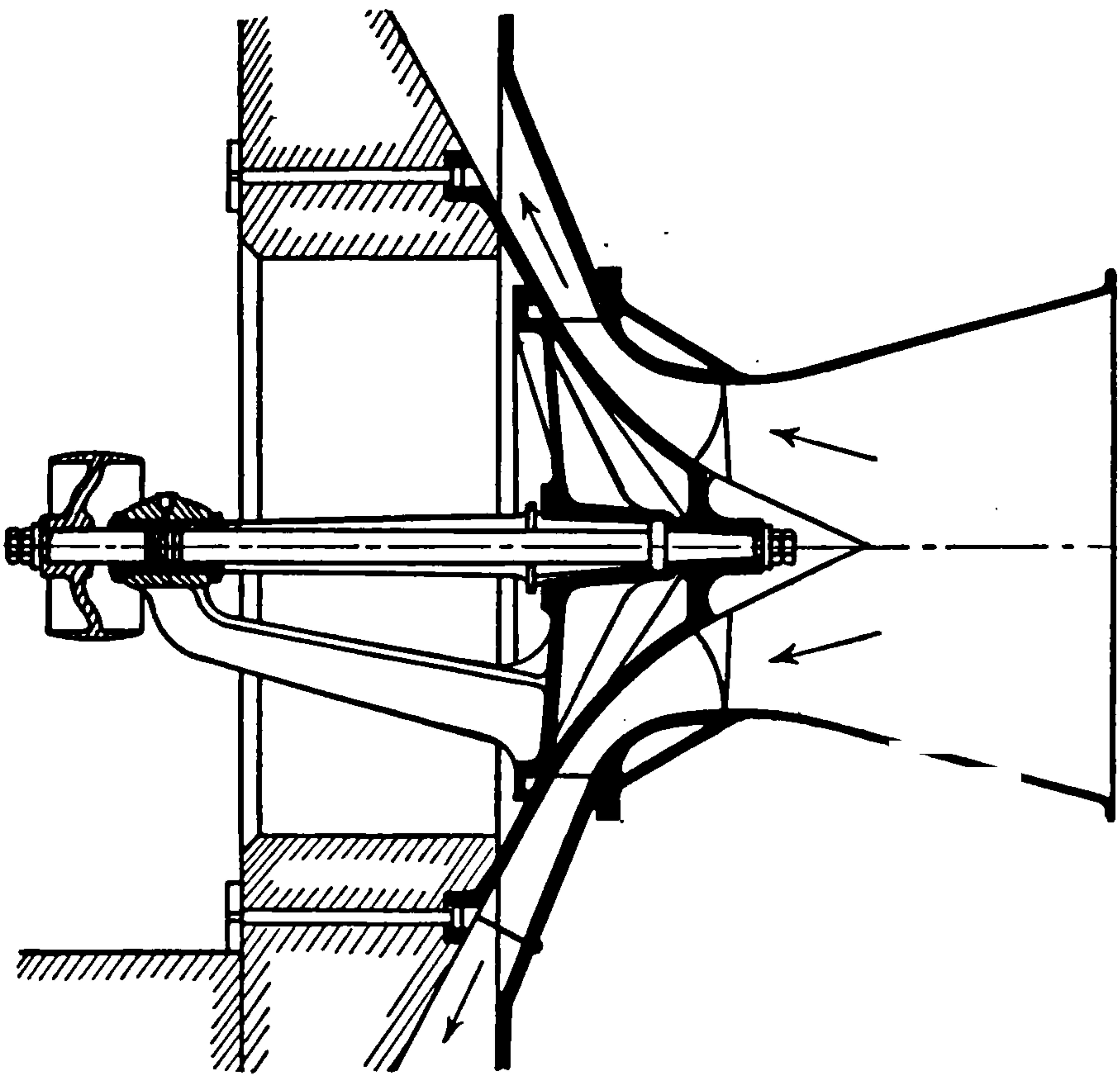


FIG. 59.—RATEAU FAN.

and the same may be said of each of the other pairs. The mechanical, which is the indicated, efficiency increases with the revolutions and power in the three first pairs, and is apparently the same in the last two. The increase is just what we might expect, as the efficiency of the fan alone is constant, and that of the engine increases with the power, so that at a constant orifice the efficiency of the two should increase with the power.

Table 19 shows the results of a series of experiments made with a Rateau fan of 4·59 ft. diameter, of type A, Table 5, tested October 18th, 1891. Each experiment lasted three minutes. The water gauge was taken in an outlet from the fan drift sheltered from the current. The discharge was measured at the top of the chimney, which was divided into 36 equal areas. Tests were also made to

TABLE 19.—RATEAU FAN TEST: TYPE A, TABLE 1.

4·59 FT. DIAMETER.

No. of test.	Rev. per min. of fan.	Cu. ft. of air per min. Q.	Corrected water gauge in inches. h.	Equivalent head of air in feet. H.	Equivalent orifice. Sq. ft.	Reduced orifice. $\frac{Q}{r_2^2\sqrt{gH}}$.	Manometric efficiency per cent.
1	274	546	0·99	71·4	12·69	2·19	53
2	359	614	2·05	146·8	9·99	1·70	64
3	380	635	2·38	170·5	9·45	1·63	67
4	385	592	2·82	201·5	8·16	1·41	77
5	372	519	3·11	222·7	6·76	1·17	91
6	263	324	1·66	118·7	5·80	1·00	96
6a	393	480	3·72	265·5	5·69	0·99	98
7	425	448	4·49	321	4·83	0·84	100
8	438	353	4·62	331	3·76	0·65	97
9	444	254	4·31	310	2·79	0·48	88
10	444	141	3·86	277	1·61	0 28	78
11	340	42	1·93	138·5	0·65	0·11	67

find the density of the air, so that the correct value H might be found from the water gauge. It will be seen in the table that the water gauge has been "corrected." This means that the gauge due to the velocity of discharge has been added, and that due to the velocity of inflow subtracted; while the latter was correct, the former was wrong, because the velocity of discharge, being of no value, ought not to be reckoned to the credit of the

TABLE 20.—RATEAU FAN TESTS: TYPE A, TABLE 1.

6·56 FT. DIAMETER.

No. of test.	Hole in baffle plate in sq. ft.	Rev. per min. of engine.	Rev. per min. of fan.	Observed water gauge in inches.	Corrected water gauge in inches. h.	Cu. ft. of air per sec. Section of chimney in 84·9 sq. ft. Q.	Mean velocity of air from the chimney in ft. per sec.	Indicated H.P. in chevaux-vapeur.	Useful work done on air.	Depression in feet of air. H.	Equivalent orifice in sq. ft.	Reduced orifice. $\frac{Q}{r_2^2\sqrt{gH}}$.	Mechanical efficiency per cent.	Manometric efficiency per cent.
1	18·9	127	281	2·56	2·84	1243	35·4	103	34	206·5	16·6	1·42	33	71
2	16·5	127	281	2·68	2·95	1232	35·1	94	35	213	16·2	1·38	37	74
3	14·2	130	290	3·07	3·35	1205	34·4	91	39	242·5	14·9	1·27	43	79
4	11·8	138	306	3·74	3·97	1152	33·1	96	44	288	13·1	1·12	46	85
5	9·4	135	300	3·94	4·14	972	27·9	74	39	302	10·7	0·92	53	91
6	7·1	147	326	5·00	5·07	825	23·6	67·5	40	371	8·2	0·70	60	95
7	4·72	147	326	4·92	4·96	582	16·7	45	28	361	5·9	0·50	61	93
8	2·36	141	314	3·74	3·74	275	7·86	23	10	275	3·2	0·27	43	75
9	0	126	279	1·97	1·97	45·9	1·3	15	0·7	144	0·75	0·06	5	52
10	*	148	329	4·92	5·00	652	18·7	53	31	371	5·9	0·55	59	94

* On the mine.

TABLE 21.—RATEAU FAN TESTS.

9·17 FT. DIAMETER.

No. of test.	Rev. per min. of fan.	Rev. per min. of engine.	Water gauge in inches (corrected). h.	Equivalent head of air in feet. H.	Mean velocity of discharge from the chimney.	Discharge in cu. ft. per sec. Q.	Equivalent orifice in sq. ft.	Ratio of equivalent orifice to hole in baffle plate.	Useful work done on air, H.P.	Indicated horse power.	Mechanical efficiency per cent.	Manometric efficiency per cent.	Reduced Orifice. $\frac{Q}{r_2^2\sqrt{gH}}$.
1	231	144	4·1	298	6·62	493	5·59	1·26	19·4	—	—	78	0·24
2	235	147	4·32	315	8·46	631	6·98	1·26	26·2	—	—	80	0·30
3	227	142	4·56	334	11·96	895	9·56	1·21	39·3	60	66	90	0·41
4	218	136	4·52	331	15·85	1181	12·80	1·25	51·5	78	66	97	0·55
5	218	136	4·92	361	20·1	1491	15·15	1·22	70·7	90	79	106	0·65
6	193	120	4·1	298	21·3	1591	18·05	1·21	62·7	98	64	112	0·77
7	174	109	3·35	243	22·6	1690	21·25	1·23	54·4	94	58	112	0·91
8	160	100	2·84	206	24·6	1830	25·1	1·20	50·4	90	56	112	1·08

TABLE 22.—RATEAU FAN TESTS.

9·17 FT. DIAMETER.

No. of test.	Rev. per min. of fan.	Watergauge in inches (corrected). h.	Equivalent head of air in feet. H.	Velocity of air at the chimney.	Discharge in cu. ft. per sec. Q.	Equivalent orifice in sq. ft.	Useful work done on air in horse power.	Indicated horse power.	Mechanical efficiency per cent.	Reduced orifice. $\frac{Q}{r_2^2\sqrt{gH}}$.	Manometric efficiency per cent.
1	180	2·93	213	8·91	635	8·37	18	32	56	0·36	92
2	192	3·87	280	15·25	1,085	12·25	40	—	—	0·55	107
3	187	3·91	283	19·95	1,415	16·32	53	—	—	0·71	114
4	148	2·53	183	20·25	1,435	20·50	35	—	—	0·90	117
5	154	2·29	167	25·5	1,831	27·40	40·5	84	48	1·19	99
6	148	1·60	116	26·7	1,899	34·00	29·3	85	35	1·48	73
7	180	3·49	252	23·6	1,680	20·40	56	99	57	0·89	110
8	192	3·73	271	24·5	1,735	20·40	62	110	57	0·89	103
9	136	1·98	144	18·45	1,305	21·10	25	47	53	0·92	104
10	86	0·83	60·6	13·10	928	23·20	7·3	—	—	102	113

plate. It will be noticed that the equivalent orifice is always greater than the opening in the baffle plate in the ratio of about 1·2 to 1. This is partly due to the passages behind the orifice, but also to the fact that the coefficient of contraction may be greater that 0·65. It probably increases with the size of the orifice.

Table 22 gives experiments on a fan of the same size at Montrambert, made on December 6th, 1891. The engine had a cylinder 14·95 in. diameter, and a stroke of 23·6. All the horse powers given with fans of this type are chevaux-vapeur, and consequently 0·985 of 1 English horse power.

Table 23 contains experiments on a Rateau fan, 13 ft. 1½ in. diameter, at the Consolidation Mines, Westphalia. The fan was guaranteed to give 175,000 cu. ft. of air per min., with a water gauge of 6 in. and an equivalent orifice of 28 sq. ft., which corresponds to a reduced orifice of 0·6 very nearly—*i.e.*,

$$\frac{Q}{r_2^2\sqrt{gH}} = 0\cdot 6.$$

In making this calculation we must remember that the equivalent orifice is 1·088 the orifice $Q \div \sqrt{gH}$.

Centrifugal Pump Experiments by Charles H. Innes.[12]—These experiments were made by the present writer on March 13th, 1897, and January 15th, 1898, at Wallsend Slipway. Two pumps are provided to empty the company's dry dock, but one only was used in the experiments. The diameters of suction and discharge are 36 in., that of the wheel 66 in., while its internal breadth at outflow is 5¾ in.; the internal diameter is 39 in., and the vanes are radial at the inner circumference, and curve back in the arc of a circle until they become tangents to the outer circumference. In both experiments special care was taken in closing the gates to minimise leakage, which was found to be about 30 cu. ft. per min.; this quantity is small in comparison with the total quantity delivered, hence it is neglected in the calculations. The suction pipe is 15 ft. 6 in. long and 36 in. diameter, and the discharge pipe enlarges with a bend to 54 in. diameter at the junction with the discharge pipe of the second pump. The remainder of the discharge pipe is 95 ft. in all, 54 in. diameter, with a right-angle bend. We do not think that the introduction of these experiments needs any apology, as, except for the fact that in a centrifugal pump ϕ should not be greater than 90 deg., the rules for designing pumps and fans are precisely the same, and deductions from experiments with the former apply to the latter. The weight of water was assumed as $62\frac{1}{2}$ lb. per cu. ft. in the first experiment and 64 in the second. In the former the effect of t'e tide on the density of the water was forgotten, and in the latter a sample of the water in the dock whose density it was intended to test did not reach the Rutherford College owing to an accident, and the author desired to make every allowance for the pump whose efficiency was low; but it must be remembered that the efficiency takes into account the friction of the pipes, so that that of the pump alone is greater. The losses of head in passing through the pump are—

(1) At inflow,

$$\frac{(v_1 - b_1 \cot \theta)^2}{2g} = \frac{v_1^2}{2g},$$

since $\theta = 90$ deg.

(2) At entrance into the volute,

$$\frac{b_2^2 + (a_2 - c_v)^2}{2g}.$$

(3) Loss of head, other than shock loss and losses due to bends and pipe friction.

The coefficient of resistance of the pump alone is here taken as $F = 3$, referred to the velocity of discharge c_v, so that

$$F\frac{c_v^2}{2g} = 0{\cdot}0466\ c_v^2.$$

The losses due to bends and pipe friction, together with the loss due to the radial speed, (2), namely $b_2^2 \div 2g = 0{\cdot}0233\ c_v^2$.

TABLE 24.—CENTRIFUGAL PUMP TRIA

1	Fall of water inside dock in ft.	0	1	2	3	4	5
2	Head outside dock in ft. and in.	19 8	19 8	19 8	20 0	20 0	20 0
3	Head inside dock at end of interval in ft. and in.	19 0	18 0	17 0	16 0	15 0	14 0
4	Boiler pressure	120	113	$107\frac{1}{2}$	105	110	110
5	Interval in min. ...	—	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$
6	Mean head	—	$1\frac{1}{6}$	$2\frac{1}{6}$	$3\frac{1}{3}$	$4\frac{1}{2}$	$5\frac{1}{2}$
7	I.H.P.	—	220·7	239·6	233·25	240·6	248
8	W.H.P.	—	12·12	22·1	34·1	46·1	53·9
9	F.H.P.	—	9·14	10·1	10·15	10·4	11·4
10	S.H.P.	—	211·56	229·5	223·1	230·2	236·6
11	Hydraulic efficiency, $\eta = \frac{\text{W.H.P.}}{\text{S.H.P.}}$ per cent.	—	5·74	9·62	15·27	20·4	22·4
12	Calculated efficiency, $\eta c = \frac{H}{H+L}$ per cent.	—	6·15	9·7	14·15	17·9	21·1
13	$\eta - \eta c$	—	– 0·41	– 0·8	0·57	2·5	1·3
14	Rev. per min.	—	125·5	136·5	137·5	140·3	144
15	Velocity of discharge from pump	—	12·7	12·7	12·72	12·75	12·2
16	Orifice $\frac{Q}{\sqrt{gH}}$ sq. ft. ...	—	16 25	10·8	8·66	7·5	6·5
17	Time	9 45	9 $52\frac{1}{2}$	10 0	10 $7\frac{1}{2}$	10 15	10 $22\frac{1}{2}$
18	Quantity discharged during interval in cu. ft.	—	41,209	40,428	40,408	40,383	38,784
19	$\frac{gH}{a_2 v_2}$* $\times 100$	—	9·5	10·9	17·85	21·7	23·7

* The angle of relative discharge from the fan is assumed to be at 26° 26′ varies between

TABLE 25.—CENTRIFUGAL PUMP T

1	Fall of water inside dock in ft.	0	1	2	3	4
2	Head outside dock in ft. and in.	21 0½	20 9	20 2	19 9	19 6
3	Head inside dock at end of interval in ft. and in.	19 0	18 0	17 0	16 10	15 0
4	Boiler pressure ...	110	110	110	110	110
5	Interval in min. and sec.	—	11·45	11·45	11·0	10·30
6	Mean head in ft. ...	—	2·395	2·957	3·457	4·125
7	I.H.P.	—	83·75	81·4	105·1	123·6
8	W.H.P.*	—	16·3	19·7	24·66	30·7
9	F.H.P.	—	7·06	7·08	7·5	8·26
10	S.H.P.	—	76·69	74·32	97·5	
11	Hydraulic efficiency, $\eta = \frac{\text{W.H.P.}}{\text{S.H.P.}}$ per cent.	—	21·3	26·5	25·3	
12	Calculated efficiency, $\eta c = \frac{H}{H+L}$ per cent.	—	22·5	26·1	26·4	
13	$\eta - \eta c$	—	−1·2	+0·4	+1·1	
14	Rev. per min. ...	—	97·1	98·3	103·5	
15	Velocity of discharge from pump	—	8·25	8·1	8·65	
16	Orifice $\frac{Q}{\sqrt{gH}}$ in sq. ft.	—	6·65	5·85	5·82	
17	Time in hours and minutes	9 41½	9 53¼	10 5	10 16	
18	Quantity discharged during interval in cu. ft.†	—	41,209	40,428	40,408	

* In the above the weight of water per cu.
† This does not include leakage, which at the

BY CHARLES H. INNES (1898).

1				9	10	12	14	16	
2	18 10			17 8	17 5	16 8	15 11	15 5	14
3				10 0	9 0	7 0	5 0	3 0	1
4				110	110	110	110	110	11
5	9·15	9·15		9·0	9·0	8·30	8·45	8·20	
6	5·457		6·707	7·375	8·041	9·375	10·582	11·9575	
7	150·5	176·0	188·0	188·25	217·0	262·6	276·0	274·1	
	44·4	49·9	54·1	59·6	65·0	79·6	85·25	98·0	117·
			9·72	10·0	10·4	13·13	14·0	15·1	
	141·4	166·6	178·28	178·25	206·6	249·47	262·0	259·0	273·
	31·5		30·3	33·5	31·5	31·9	32·6	37·9	
	29·55	30·6	32·0	33·2	33·6	33·9	35·6	38·5	
	+1·95	−0·6	−1·7	+0·3	−2·1	−2·0	−3·0	−0·6	
	124·8	128·0	132·5	135·7	140·1	150·2	154·0	157·8	158·
				9·83	9·83	10·35	9·8	9·95	
		4·99	4·72	4·50	4·32	4·21	3·77	3·57	
	10 45¼	10 54½	11 3½	11 12½	11 21½	11 38½	11 55½	12 12½	
	38,761	38,738	37,525	37,503	37,480	37,255	36,341	35,095	

taken at 64 lb.
of the trial was found to be 30·6 cu. ft. per min.

Combining these, we have—loss (3) together with the loss due to radial speed $= 0\cdot07\ c_v^2$. In both sets of trials only one pump and engine were used. The friction of the engine was taken by running the engine unloaded at speeds varying between 122 and 185 rev. The H.P. absorbed by the engine under these conditions is termed the Friction H.P., and the horse power transferred to the pump shaft is

$$\therefore \text{ S.H.P.} = \text{I.H.P.} - \text{F.H.P.}$$

This is not exactly correct, because the friction of a loaded engine is a little more than that of one running light, but the error is not very great. See Tables 24, 25, pp. 128-131.

The following is the method of comparing the efficiencies obtained from experiment and calculation: In the first trial at the 14th foot we find I.H.P. $= 255\cdot8$; the useful work done by the pump in raising Q cu. ft. of water in the given interval (namely, W.H.P.) $= \dfrac{62\cdot5 \times 36{,}341 \times 13\cdot875}{33{,}000 \times 10\cdot5} = 91\cdot1$; rev. per min. $= 154\cdot1$ (these were taken by a counter read every minute); mean head, 13·875; this was measured by two posts outside and inside, giving heights in feet above the sill of the dock, and each interval commenced at a foot on the inner scale. On the outer scale inches were measured by a pole which had alternate inches at its end painted in black and white for a foot length, and this held against the post enabled very accurate readings to be taken, as the motion of the water was not more than 4 in. on March 13th and not more than 2 in. on January 15th. We mention these details, as we believe there have been no experiments on pumps made with greater care than these. The quantity of water discharged was 36,341 cu. ft. during an interval of $10\frac{1}{2}$ min. This gives a velocity of discharge of 8·17 ft. per sec. from the pump, and as F.H.P. at 140·5 rev. was 10·4 and at 160 rev. was 16, by interpolation at 154·1 it was 14·3.

Hence $\qquad \text{S.H.P.} = \text{I.H.P.} - \text{F.H.P.} = 241\cdot5.$

$$\text{Pump efficiency} = \frac{91\cdot1}{241\cdot5} = 0\cdot3775.$$

TABLE 26.—CENTRIFUGAL PUMP EXPERIMENTS BY R. C. PARSONS.

No. of experiment.	Gallons per min.	Lift in feet.	Ft.-lb. raised per min.	Ft.-lb. indicated per min.	Rev. per min.	Efficiency per cent.	Corrected efficiency per cent. η_3.	$\frac{g\,H}{a_2\,v_2}$. η_1.	$\frac{H}{H+L}$. η_2.
1	1,012	14·67	148,461	208,438	392	49·74	58·57	57·5	57·7
4	1,280	14·7	188,160	343,754	398	54·74	62·99	58·7	60·1
6	1,431	14·75	211,073	374,954	400	56·20	63·95	60·0	62·1
8	1,568	14·75	231,280	404,737	403	57·01	64·29	61·2	61·25
10	1,695	14·75	250,012	419,790	405	59·60	66·70	62·1	65 0
11	1,753	14·8	259,450	435,630	406	59·42	66·39	63·25	65·9
12	1,012	17·4	176,088	370,458	424	47·53	54·06	56·2	53·75
15	1,280	17·3	221,440	417,214	428	53·08	59·51	58·7	60·0
17	1,431	17·4	248,994	447,552	431	53·63	61·86	60·0	60·75
19	1,568	17·4	272,832	471,552	433	57·86	63·95	61·2	63·7
21	1,695	17·6	298,310	486,050	435	61·37	67 64	62·6	64·7
22	1,753	17·6	308,528	494,210	436	62·43	68·68	63·5	64·9

at 410 rev. (the mean of the number during the experiments), while the pump was not discharging. It was found that 45,000 ft.-lb. per min. were required, and this amount was deducted from the work done by the engine, and the work done by the pump being divided by this was called the corrected efficiency.

The corrected efficiency η_3 is more than the hydraulic efficiency, because the work required to drive a pump when not discharging is very much greater than the work expended in overcoming the friction of the fan shaft and the surface friction of the disc. The real hydraulic efficiency obviously is between this "corrected efficiency" and the ratio of the work done by the pump to that done by the engine. It will also be seen that the calculated efficiencies do lie between the latter and Parsons' corrected efficiencies, and are therefore close to the real efficiences of the pump. These experiments give us reasonable confidence in asserting that the efficiency of a pump is—

$$\eta_1 = \frac{gH}{a_2 v_2}$$

$$\text{or } \eta_2 = \frac{H}{H+L},$$

where L are the losses of head in ft.

The method of calculating η_1 and η_2 is as follows: The dimensions of the pump are given in a paper on Centrifugal Pumps by Prof. Unwin.[14] The external radius of the fan is $r_2 = 9\cdot25$ in. $= 2\,r_1$, the internal radius. The breadths, s_2 and s_1, at the external and internal radii, are both 5·75 in. There were eight vanes, and as their thickness is not given, they are assumed to be $\frac{1}{4}$ in. at their ends. The velocity c_v in the volute is given by Prof. Unwin as $3\,b_2$, but as he evidently neglects the vanes, this must be modified. Assuming a coefficient of contraction of $\frac{9}{10}$ at discharge from the fan, which is also the custom in radial-flow turbines,[15] we get

$$c_v = K\left[1 - \frac{n\,s_2\,t_2 \operatorname{cosec}\phi}{2\,\pi\,r_2\,s_2}\right] \times 3\,b_2 = 2\cdot35\,b_2\,;$$

and since

$$b_2 (2 \pi r_2 s_2 - n s_2 t_2 \operatorname{cosec} \phi) \mathrm{K} = b_1 (2 \pi r_1 s_1 - n s_1 t_1 \operatorname{cosec} \theta) \mathrm{K},$$

$$b_1 = \frac{(2 \pi r_2 - n t_2 \operatorname{cosec} \phi)}{(2 \pi r_1 - n t_1 \operatorname{cosec} \theta)} b_2 = 1{\cdot}94\, b_2,$$

where $\mathrm{K} = \frac{9}{10}$,
$n = 8$, the number of vanes,
$\theta = 40°$,
$\phi = 15°$.

Let G = gallons per min.,

then $$b_1 = \frac{\mathrm{G}}{60 \times 6{\cdot}25\, (2 \pi r_1 s_1 - n s_1 t_1 \operatorname{cosec} \theta) \mathrm{K}}$$

and $$b_2 = \frac{\mathrm{G}}{60 \times 6{\cdot}25\, (2 \pi r_2 s_2 - n s_2 t_2 \operatorname{cosec} \phi) \mathrm{K}}$$
$$= 0{\cdot}001475\ \mathrm{G}.$$

Taking the first experiment as an example, $\mathrm{G} = 1012$, $b_2 = 1{\cdot}49$, $c_v = 3{\cdot}50$, $v_2 = 31{\cdot}6$, $a_2 = v_2 - b_2 \cot \phi$, and assuming the relative angle of flow coincides with the angle of vane,

$$a_2 = 26{\cdot}04,$$

$$100\, \eta_1 = \frac{g \mathrm{H}}{a_2 v_2} \times 100 = \frac{32{\cdot}2 \times 14{\cdot}67}{31{\cdot}6 \times 26{\cdot}04} = 57{\cdot}5 \text{ per cent.}$$

The losses of head L in passing through the pump are—
The loss at entry to the wheel

$$= \frac{(v_1 - b_1 \cot \theta)^2}{2 g} = 2{\cdot}37\,;$$

the loss at entry to the volute

$$= \frac{b_2^2 + (a_2 - c_v)^2}{2 g} = 7{\cdot}92\,;$$

and the surface friction $= \mathrm{F} \frac{c_v^2}{2 g}$. The most suitable value of F is 2·5, so that

$$\mathrm{F} \frac{c_v^2}{2 g} = 0{\cdot}475.$$

having an expanding rim (fig. 61). In other respects it is a simple open-running fan, with the vanes curved so as to be convex to the outlet. The following are the principal dimensions of the fan :—

Diameter to periphery of divergent outlet	36 ft. 4 in.
Diameter to the extremities of the blades	35 ft.
Diameter of inlet ring	13 ft. 6 in.
Width at outlet	1 ft. $1\frac{1}{2}$ in.
Width at periphery of fan...	2 ft. $2\frac{3}{4}$ in.
Capacity of fan	2,583 cu. ft.

The experiments in Table 28 have been selected from those given.

TABLE 28.—TESTS RESULTS: WADDLE FAN.

	No. of experiment.				
	1	2	4*	6	V.
Rev. per min. of fan ...	50	57·2	59·73	37·2	67·47
Cu. ft. of air per min.	140,158	169,953	231,306	108,777	174,008
Water gauge	2·07	2·41	2·671	1·082	3·274
Useful work done by fan, H.P.	45·68	64·5	97·33	18·51	89·68
I.H.P.	69·91	100·64	144·5	27·72	139·94
Efficiency	0·653	0·641	0·673	0·668	0·641

* In this experiment the separation doors were open ; the others were made on the mine.

Marine Ventilating Fan Experiments by M. Lelong.[17]— M. Lelong made a number of very interesting experiments in order to obtain data for the calculation of the dimensions of ventilators for warships. The first quantity requiring formulæ was the resistance of the circuit through which the fan discharged its air. This is made up of (1) surface friction, (2) changes of section, (3) changes of direction.

(1) Surface Friction.—The loss of head due to friction may be expressed by the formula (put in our own notation),

$$H_f = \zeta \frac{l}{m} \frac{v^2}{2g},$$

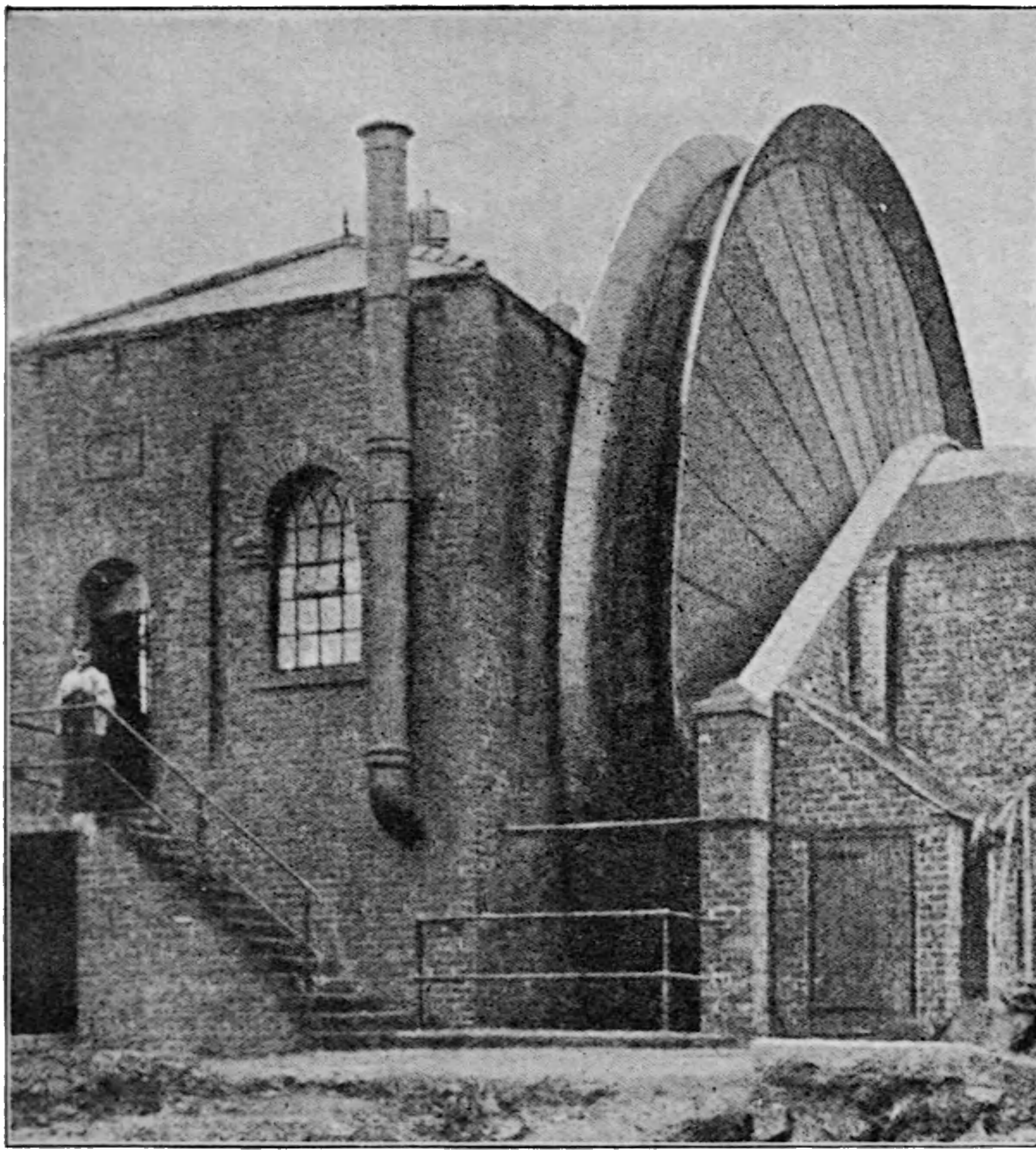


FIG. 61.—WADDLE FAN.

where l is the length in ft. m the mean hydraulic depth equal to the area of section, a, divided by the perimeter, s, and v the velocity in ft. per sec. ζ is a coefficient whose mean value is about 0·006. Calling

$$\frac{2\,g\,H_f}{Q^2}$$

the resistance due to this, we get

$$\frac{2\,g\,H_f}{Q^2}=\frac{\zeta l}{m\,a^2}=0{\cdot}006\,\frac{l\,s}{a^3}.$$

(2) Changes of Section.—When a passage ends in a very large space, the kinetic energy of the current is completely lost, and the corresponding loss of head is

$$\frac{v^2}{2\,g},$$

while the resistance due to this

$$\frac{2\,g\,H_k}{Q^2}=\frac{1}{a^2}.$$

Inversely, if the current of air flows from a large space into a cylindrical pipe, we usually allow a coefficient of contraction $c_c=0{\cdot}83$, for the vein entering the pipe, so that the loss of head here becomes.

$$H_c=\left(\frac{1}{c_c^2}-1\right)\frac{v^2}{2\,g}=\left(\frac{1}{(0{\cdot}83)^2}-1\right)\frac{v^2}{2\,g},$$

and the corresponding resistance

$$\frac{2\,g\,H_c}{Q^2}=\frac{0{\cdot}45}{a^2}.$$

M. Lelong made several experiments in order to find the value of the coefficient applicable to rectangular passages of large dimensions, such as one finds on board ship. The experiments were made with passages of two sizes. The smaller was made by dividing one 1·31 ft. by 2·62 ft. section into two parts by a longitudinal partition. They were 9·84 ft. in length, and were made of carefully planed wood, one end being connected with the atmosphere, whilst the other was enclosed in a chamber receiving the air delivered by the fan. The static pressure in this chamber was given by a manometer; the discharge was given by an anemometer. The total resistance includes not only that at inflow, but also that due to friction and the loss of the kinetic energy at discharge; we have therefore deducted these two last, using 0·004 for the coefficient of friction. The results obtained are given in Tables 29, 30.

For the second passage the results given are the means between those obtained for each of the two passages.

By connecting the exit or entry of a pipe to the larger space by means of a cone, the loss of head is much reduced, and can become zero. It is negligible if the angle at the vertex of the cone does not exceed 30 deg. when the passage is to be reduced, and 7 deg. when the passage must increase.

(3) Resistances due to Change of Direction.—The loss of head at elbows is probably due to a contraction formed by the stream. Taking the loss as that given by Péclet,—

$$H_\epsilon = \zeta_\epsilon \frac{v^2}{2g}$$

where $\zeta_\epsilon = \sin\phi$, ϕ being the angle made by one pipe with the prolongation of the other.

For bends the loss is much less. Weisbach gives the following formula:—

$$H_b = \zeta_b \frac{v^2}{2g},$$

where

$$\zeta_b = \left\{0{\cdot}131 + 1{\cdot}847\left(\frac{d}{2\rho}\right)^{\frac{7}{2}}\right\}^*$$

for pipes of circular section, diameter d and mean radius of curvature ρ, and

$$\zeta_b = \left\{0{\cdot}124 + 3{\cdot}104\left(\frac{s}{2\rho}\right)^{\frac{7}{2}}\right\}^*$$

for rectangular sections, where s is the length of the side of the section parallel to the radius of curvature ρ. According to these formulæ, the loss of head does not depend on the total angle of the bend, but on the ratio of s to ρ.

Several experiments were made by M. Lelong to see if Weisbach's formula could be applied to large rectangular-sectioned ventilation passages, such as are found in warships.

* We have usually seen these formulæ multiplied by α, the fraction of two right angles of the bend, but even this modification does not bring about an agreement between Weisbach's formula and the results in the table.

The passages used were similar to those described above. At the ends of the bends it was necessary to add a length of passage of 3 metres, so as to obtain a uniform outflow of air, which appeared to be extremely irregular after leaving the bends. Table 31 gives the results of experiments and compares the coefficients ζ_b thus obtained with those deduced from Weisbach's formula. Before the air entered the bend it had to pass through 3 metres of passage, and after leaving the bend through the same distance.

The experimental values of ζ_b are generally less than those given by Weisbach. The greatest discrepancies are to be found in experiments (4) and (8). These may be explained by the small angle of the bend, of which Weisbach's formula as given by M. Lelong does not take account. The figures show, however, that the total resistance is very nearly independent of the discharge, and that the loss due to bends is less than that due to sharp corners, if we accept the formula

$$H_e = \sin\phi \frac{v^2}{2g}.$$

In experiments (5) to (8) the passages were divided by vertical longitudinal partitions, and experiments (5A) and (7A) refer to those having the greater radii of curvature ρ. The principal difficulty in making such experiments as these seems to be that the resistance of the bend is only a small part of the whole, and therefore, as it is obtained by difference, a large percentage error is possible.

Fan Experiments by M. Lelong.—The first fan tested was one designed for the *Du Chayla.* Its dimensions were as follows: External diameter of wheel, 5·25 ft.; diameter of eye, 3·28 ft.; number of vanes, 24; width of vanes at the outer circumference, 0·492 ft. The casing was a volute whose sections were calculated by the formula

$$s = \frac{\psi}{2\pi} \frac{Q}{c_v},$$

where Q = discharge in cu. ft. per sec., ψ = angle measured in radians from the commencement of the volute to the

No. of experiment.	Description of bend or bends.	Head in metres of air. H.	Cubic metres of air per sec. Q.	Total resistance. $\frac{2gH}{Q^2}$.	Mean resistance.	Resistance due to bends.	Experimental. ζ_b.	Weisbach's. ζ_b.
1	Right-angle bend ...	12·8	3·51	20·2	20·7	4	0·43	0·874
		20	4·29	21·2	20·7	4	0·43	0·874
2	Two right-angle bends in opposite directions	12·8	3·21	24·1	24·3	7·31	0·79	1·748
		20	3·99	24·5	24·3	7·31	0·79	1·748
3	One bend at 45 deg. ...	12·8	3·56	19·7	20·25	3·68	0·398	0·874
		20·8	4·42	20·8	20·25	3·68	0·398	0·874
4	Two bends at 45 deg. in opposite directions	12·8	3·55	19·95	20·1	3·39	0·368	1·748
		20·8	4·46	20·3	20·1	3·39	0·368	1·748
5	One right-angle bend ...	15·2	1·76	96·5	100	12·14	0·31	0·398
		24·8	2·17	104	100	12·14	0·31	0·398
5*a*	One right-angle bend ...	15·2	1·73	100	100·5	11·14	0·26	0·148
		24·8	2·20	101	100·5	11·14	0·26	0·148
6	Two right-angle bends in opposite directions	15·2	1·66	109	108·5	17·5	0·45	0·546
		24	2·10	108	108·5	17·5	0·45	0·546
6*a*	Two right-angle bends in opposite directions	15·2	1·70	104	105·5	14·5	0·37	0·546
		24	2·11	107	105·5	14·5	0·37	0·546
7	One bend at 45 deg. ...	14·4	1·74	94	94	7·70	0·20	0·398
		22·4	2·16	94	94	7·70	0·20	0·398
7*a*	One bend at 45 deg. ...	14·4	1·81	87	87·5	0·50	0·013	0·148
		22·4	2·23	88·5	87·5	0·50	0·013	0·148
8	Two bends at 45 deg. in opposite directions	15·2	1·85	88	90·5	1·80	0·04	0·546
		24	2·26	93	90·5	1·80	0·04	0·546
8*a*	Two bends at 45 deg. in opposite directions	15·2	1·76	97	96	7·30	0·18	0·546
		24	2·24	95	96	7·30	0·18	0·546

fig. 63, represents the characteristic of this fan. Curve (4) shows the effect of doing away with the diffuser. In both these fans M. Lelong considers that the manometric efficiency is the same as the mechanical efficiency on account of the outlet angles of their vanes being 90 deg. This,

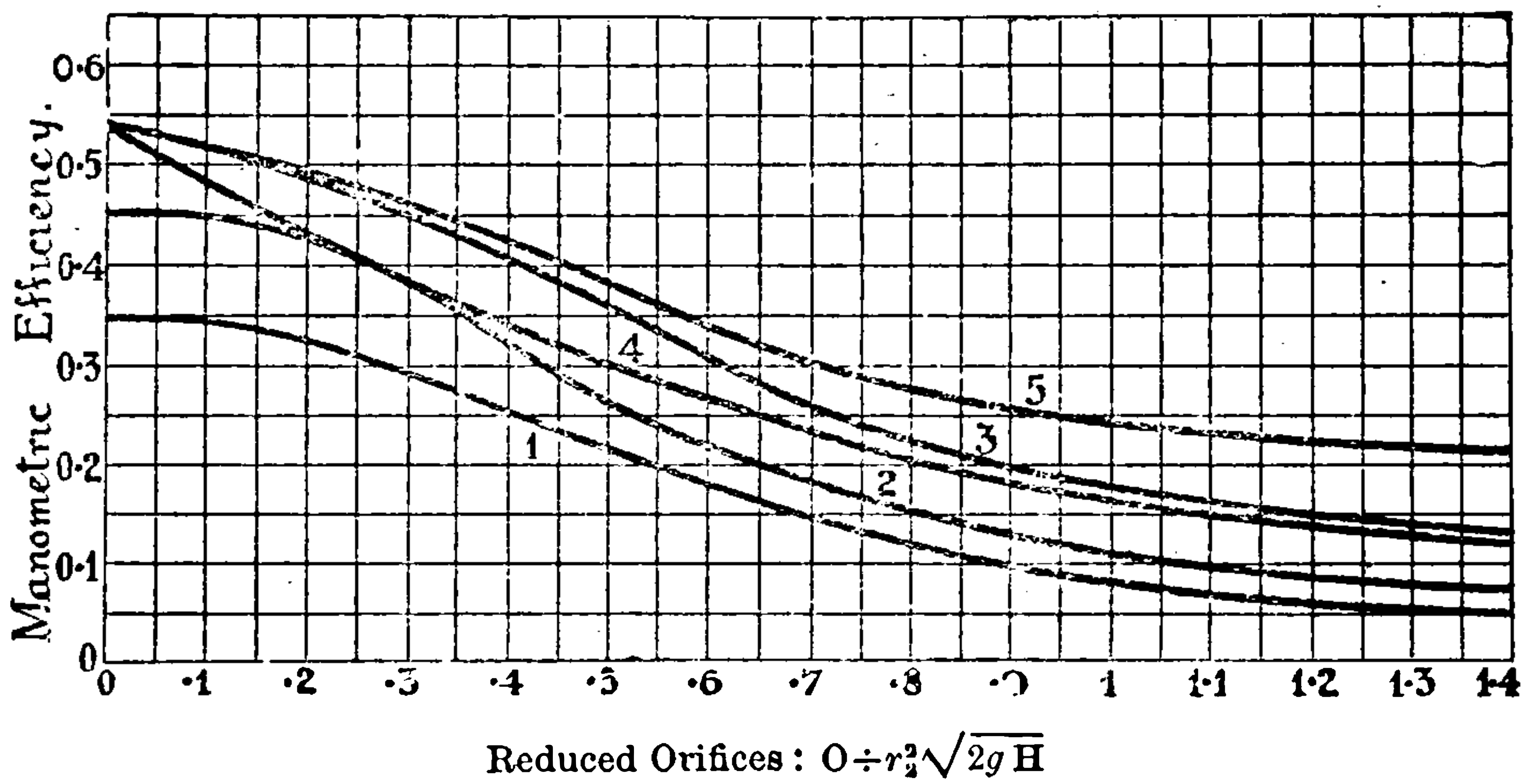


FIG. 63.—CHARACTERISTIC OF FANS TESTED BY M. LELONG.

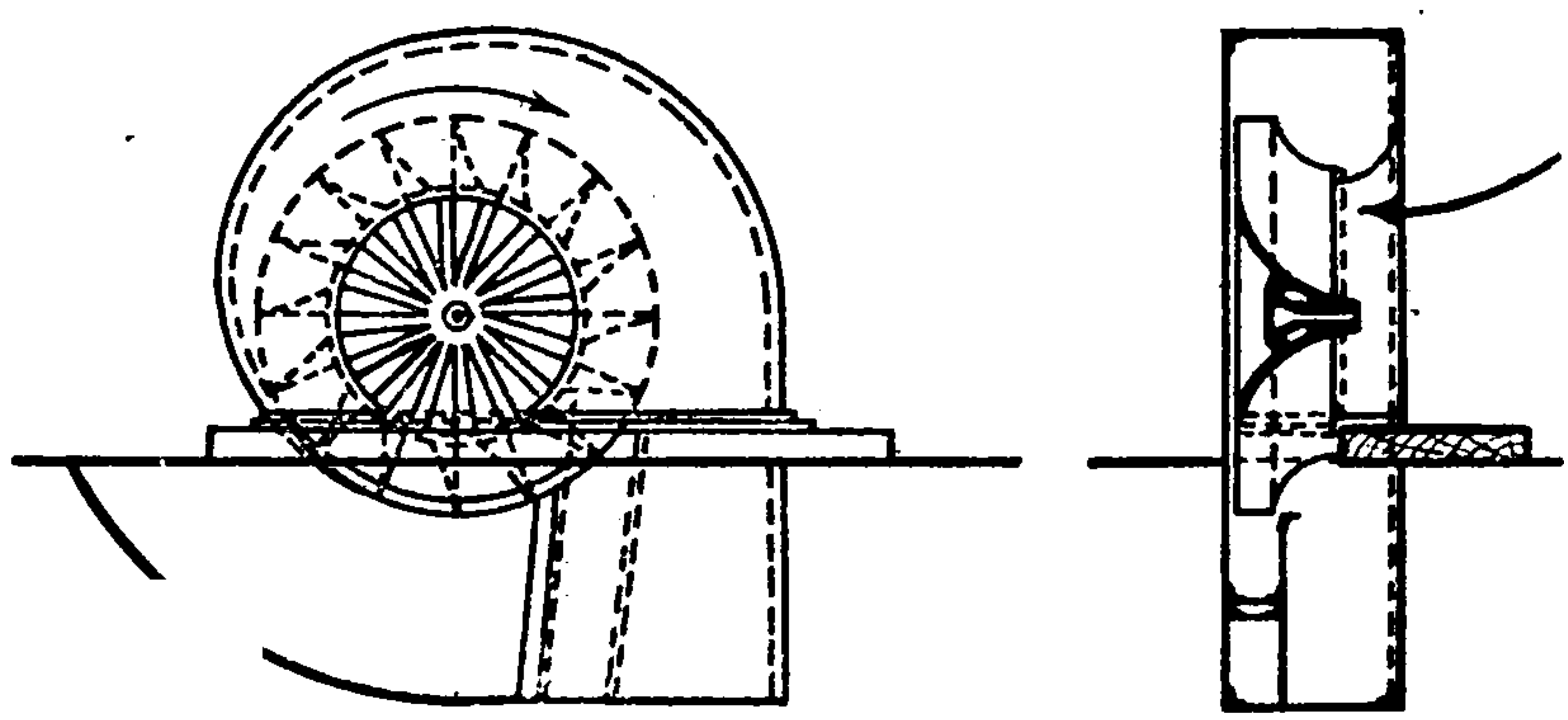


FIG. 64 —SECOND FAN TESTED BY M. LELONG.

however, is doubtful. The latter is probably more, because it is quite possible that $a_2 < v_2$ as the angle of flow is not always the same as the angle of vane.

The third fan tested, shown in fig. 65, had two eyes. Its vanes were inclined forward at 45 deg. to the radius.

The sections of the volute were the same as in the last case, but there was no diffuser, and this we believe is the reason that the maximum manometric efficiency was only about 65 per cent. The external diameter of the vanes was 4·59 ft., the diameter of each eye 0·295 ft.; the number of vanes was 16, and their width at the outer diameter of the wheel 0·459 ft. The characteristic curve of this fan (6), fig. 66, is much higher than the last, but its mechanical efficiency is not any greater than that of the first or second. If we assume that the angle of relative outflow from the wheel is the angle of the vane, the efficiency of the fan alone in this case should be very nearly

$$\eta = \frac{g\mathrm{H}}{a_2 v_2} = \frac{\sqrt{\eta_m}\ \sqrt{g\mathrm{H}}}{v_2 + b_2} = \frac{\sqrt{\eta_m}}{\dfrac{1}{\sqrt{\eta_m}} + \dfrac{b_2}{\sqrt{g\mathrm{H}}}}$$

$$= \frac{\eta_m}{1 + \dfrac{\mathrm{Q}\sqrt{\eta_m}}{2\pi r_2 s_2 \sqrt{g\mathrm{H}}}} = \frac{\eta_m}{1 + \dfrac{\mathrm{O_R}\, r_2 \sqrt{\eta_m}}{\pi\sqrt{2}\,.\,s_2}}$$

$$= \frac{\eta_m}{1 + 1{\cdot}126\,\mathrm{O_R}\sqrt{\eta_m}},$$

where $\mathrm{O_R} = \dfrac{\mathrm{Q}}{r_2^2\sqrt{2g\mathrm{H}}}$ and $s_2 = 0{\cdot}2\, r_2$.

This gives us from curve (6) the following table:—

Reduced orifice $\mathrm{O_R}$	0·2	0·3	0·4
η	0·54	0·50	0·45

which shows very clearly the mistake of not having a proper diffuser to receive the air discharged from the wheel.

Curve 7 shows the dynamic manometric efficiency of the fan. Fig. 67 represents an open running fan with radial vanes, tested with a casing. Its dimensions were: Diameter of wheel, 4·42 ft.; diameter of eye, 2·62 ft.; number of vanes, 34; breadth of wheel at discharge, 0·354 ft. It gave the characteristic curve 8. Curves 9 and 10 (fig. 68) represent the characteristics, with and without an inflow mouthpiece, of a fan constructed for experimental purposes, and only differing from that shown in fig. 67 in the in-

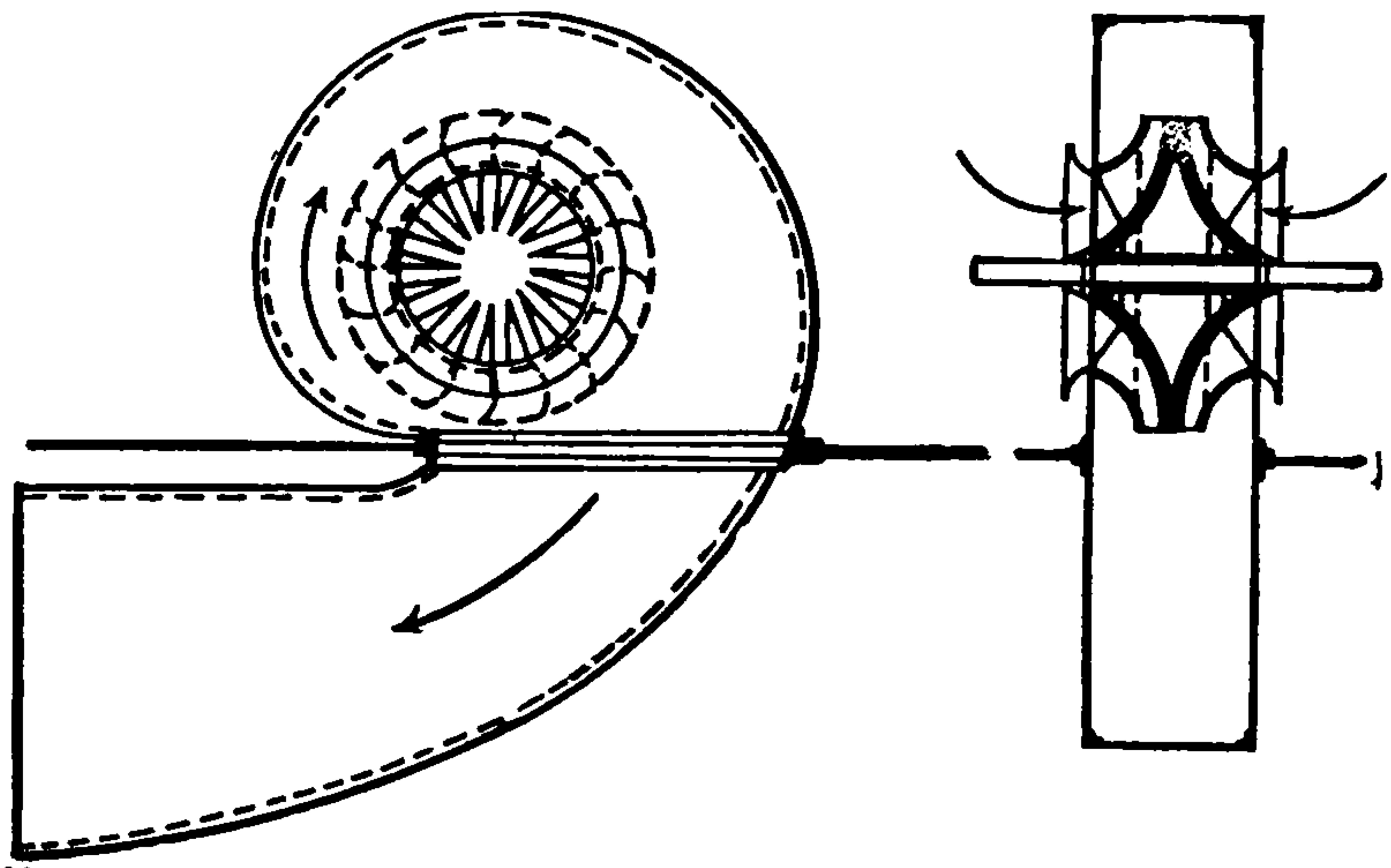


FIG. 65.—TWO-EYED FAN TESTED BY M. LELONG.

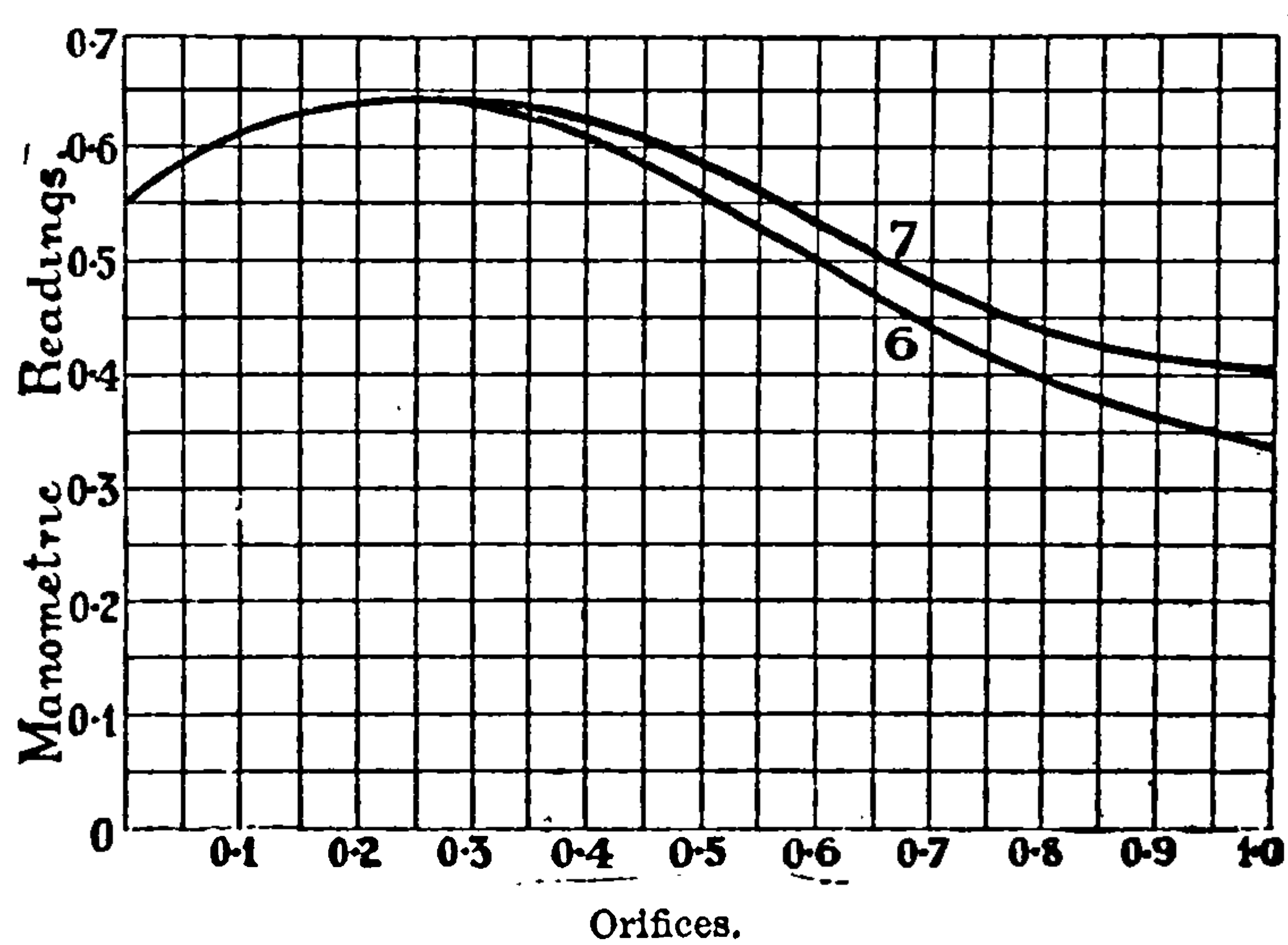


FIG. 66.—CHARACTERISTIC OF TWO-EYED FAN.

clination of its vanes at the outer radius (fig. 69). Its manometric efficiency was less than that of the fan pre-

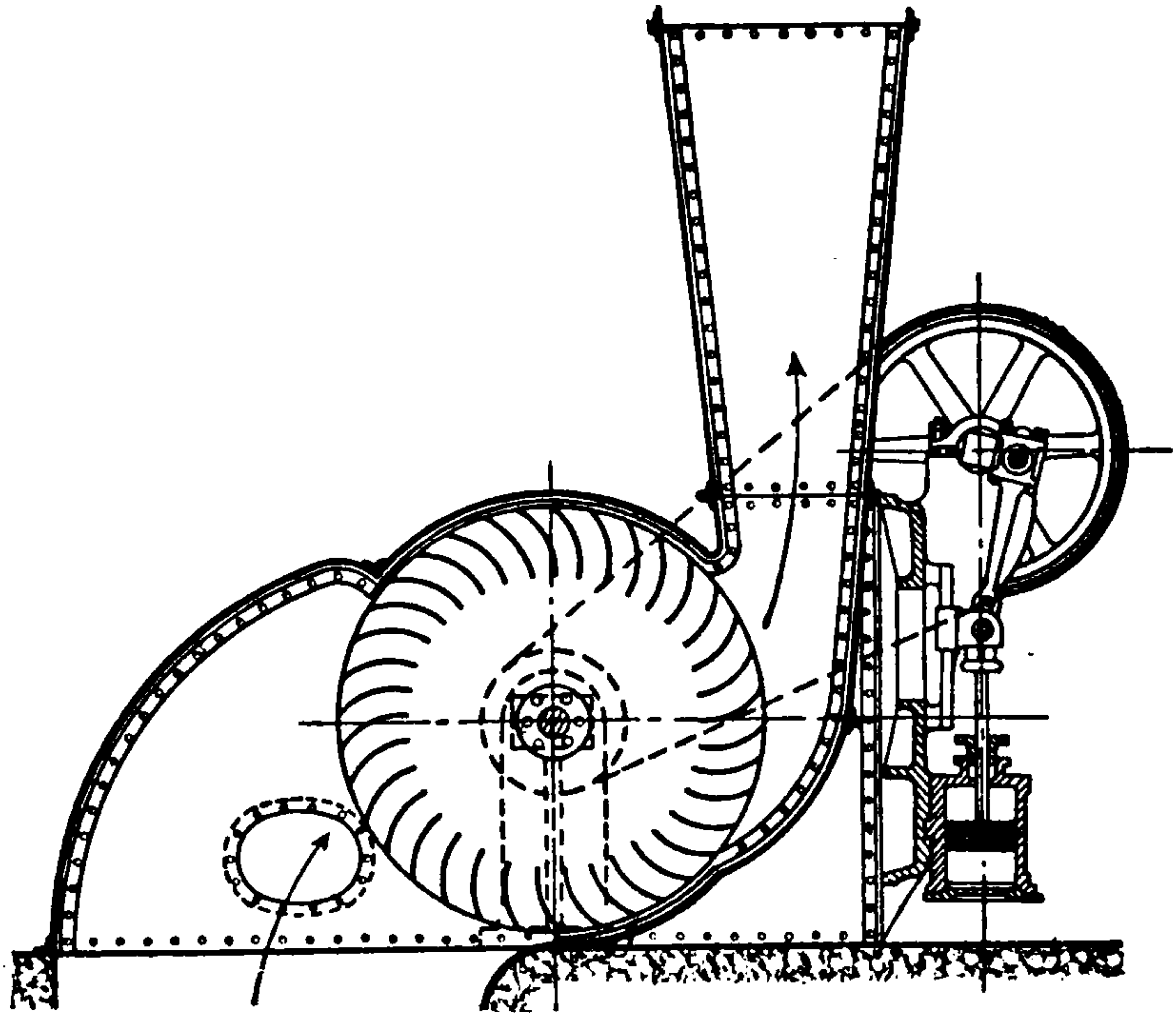


FIG. 70.—MORTIER DIAMETRAL FAN.

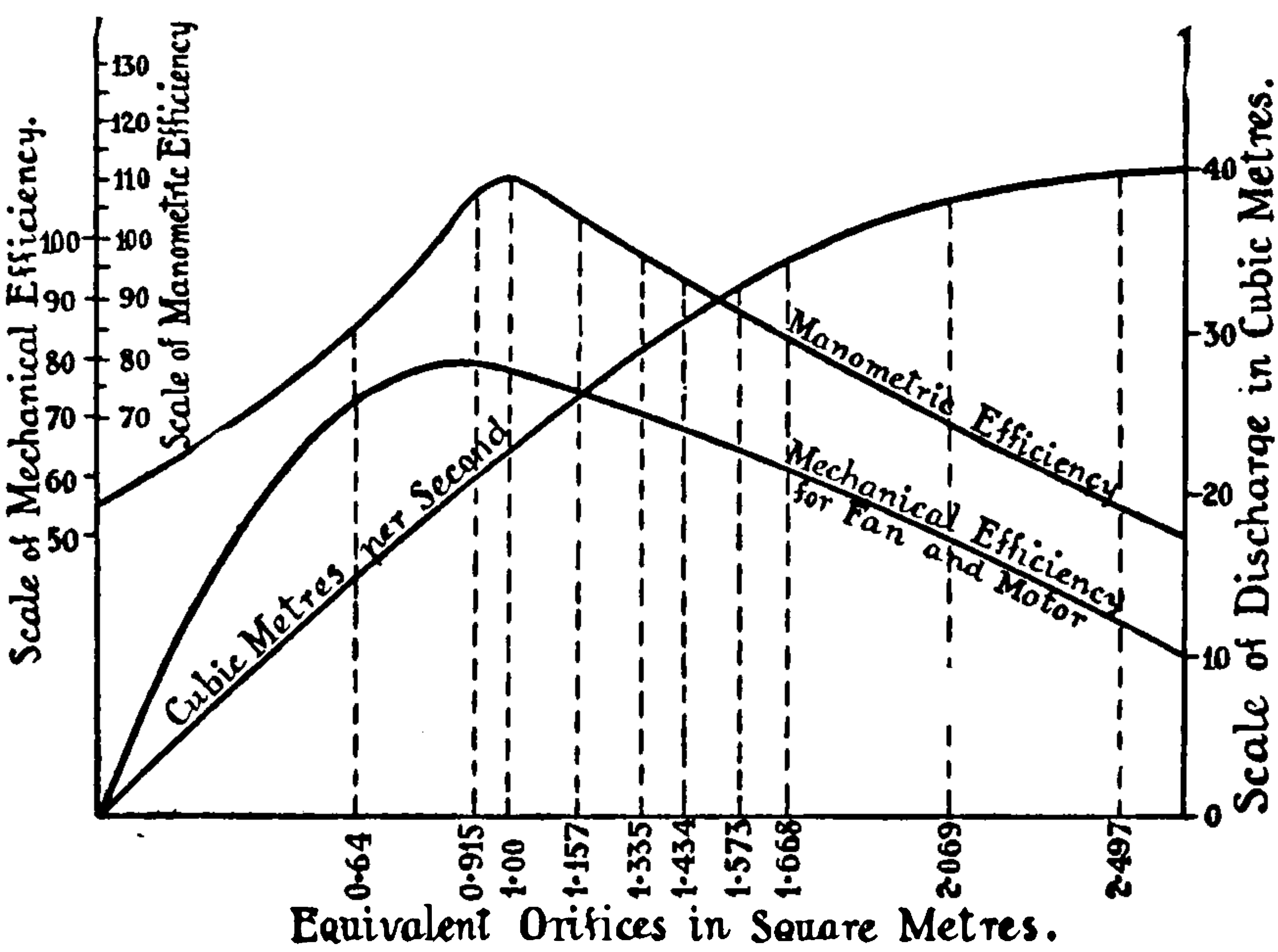


FIG. 71.—RESULTS OF EXPERIMENTS WITH MORTIER FAN.

viously discussed, as it had bent-back vanes. If we increase the height of curve 8 in the same proportion as 9 is above 10, we get curve 11, which M. Lelong considers would have been the characteristic of the fan (fig. 67) if an inflow mouthpiece had been added.

Mortier Diametral Fan Experiments.—In fig. 70 is shown a Mortier diametral fan. The direction of rotation is counter clockwise, and the air enters and leaves as shown

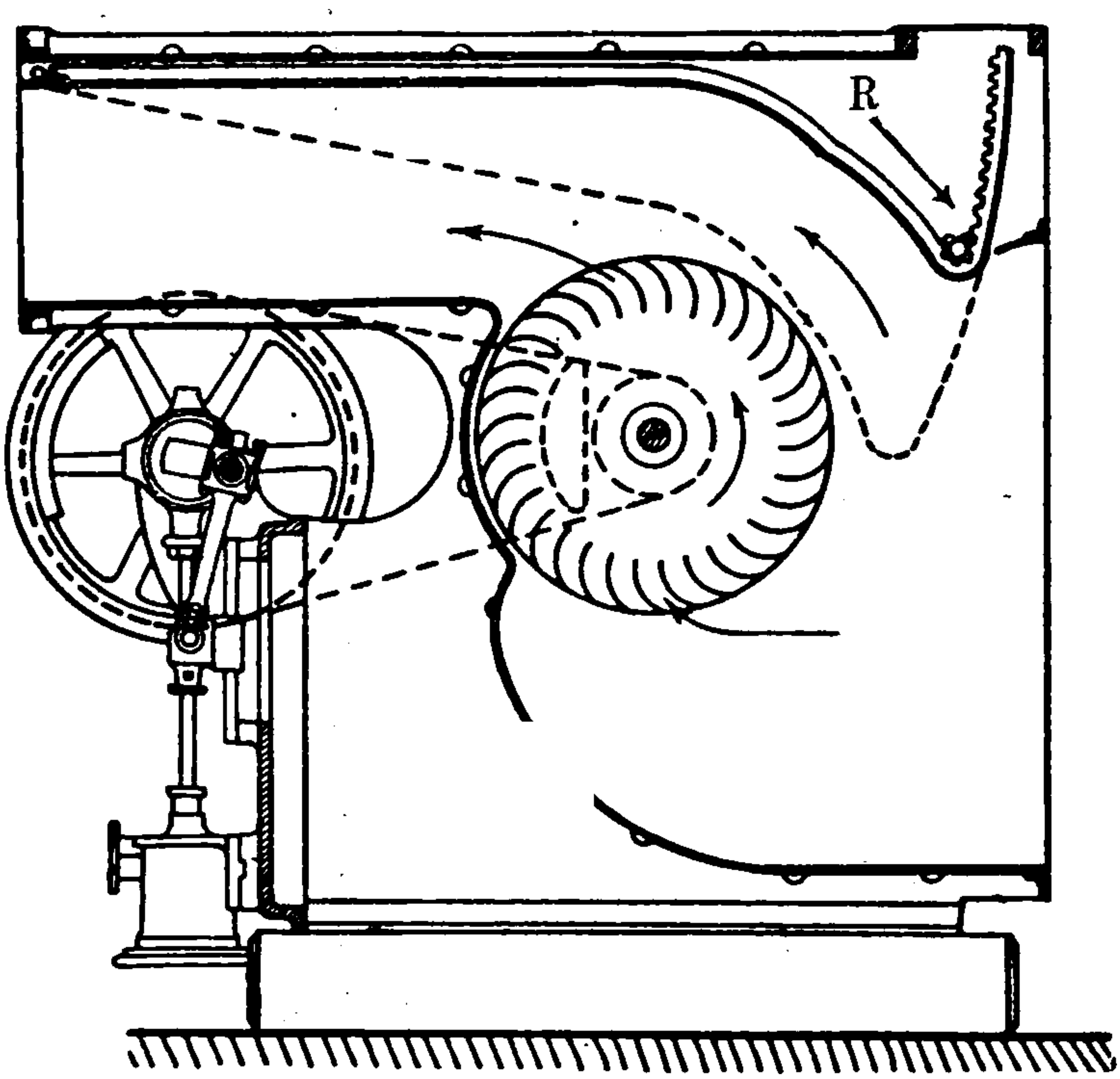


FIG. 72.—ILLUSTRATING MOVING OF CASING OF MORTIER FAN TO SUIT THE MINE.

Orifice: *R*, Rack and pinion for moving casing.

by the arrows. The vanes are bent forward so that inflow takes place without shock, and the air at outflow is thrown forwards as well as outwards. Its velocity head at discharge must be considerable; this, however, is partly converted into pressure head by the chimney. These fans are made by Louis Galland, of Chalon-sur-Saône. The results of experiments with a fan of this type, 6·56 ft. diameter by 3·94 ft. broad, are shown in fig. 71, in which the mechanical efficiency of fan and engine and the manometric

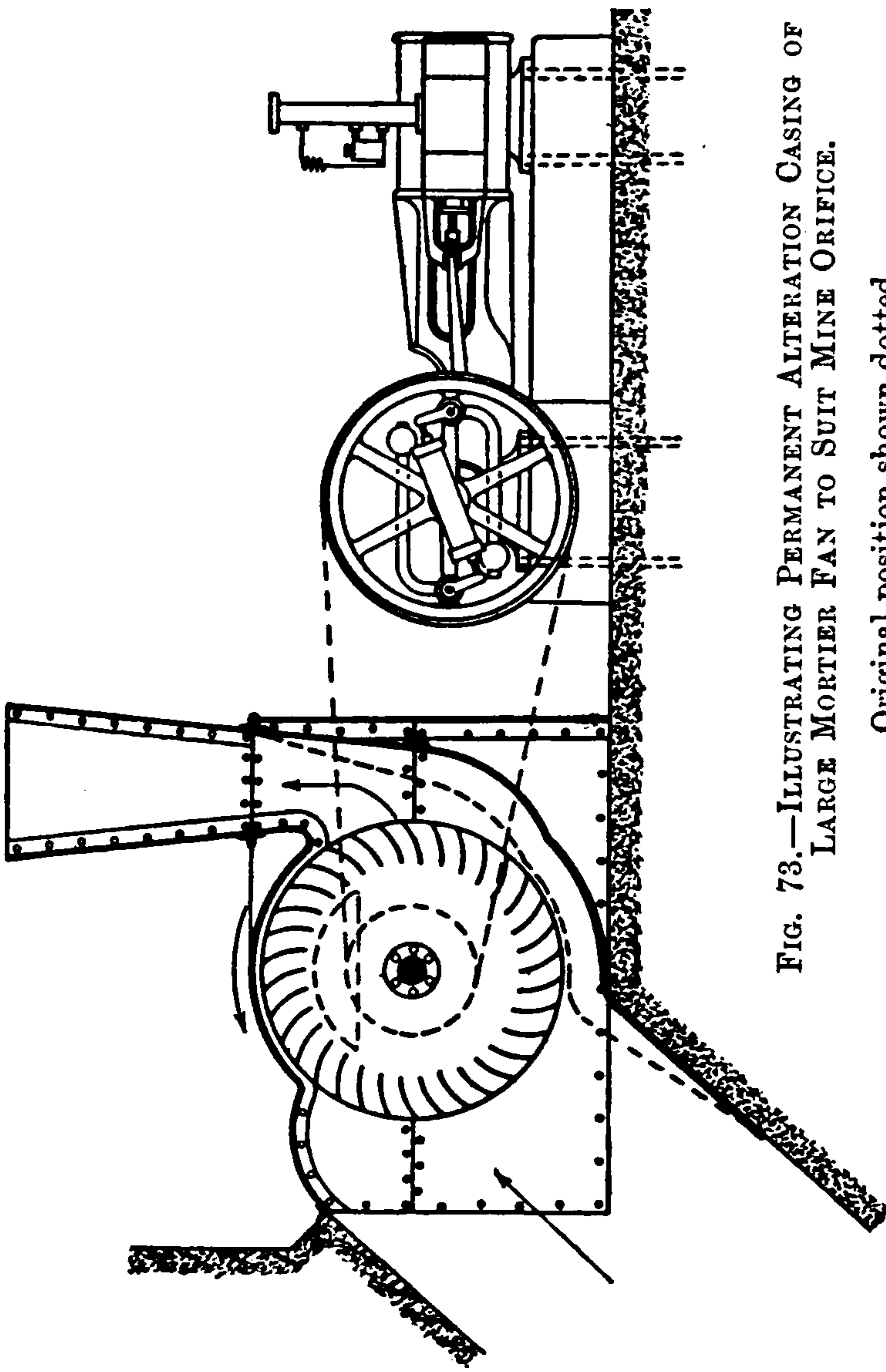


FIG. 73.—ILLUSTRATING PERMANENT ALTERATION CASING OF LARGE MORTIER FAN TO SUIT MINE ORIFICE.

Original position shown dotted.

efficiency are given. The quantity of air that would be discharged per sec. in cu. metres at 225 rev. per min., or a peripheral velocity of 77·4 ft. per sec., is also shown.

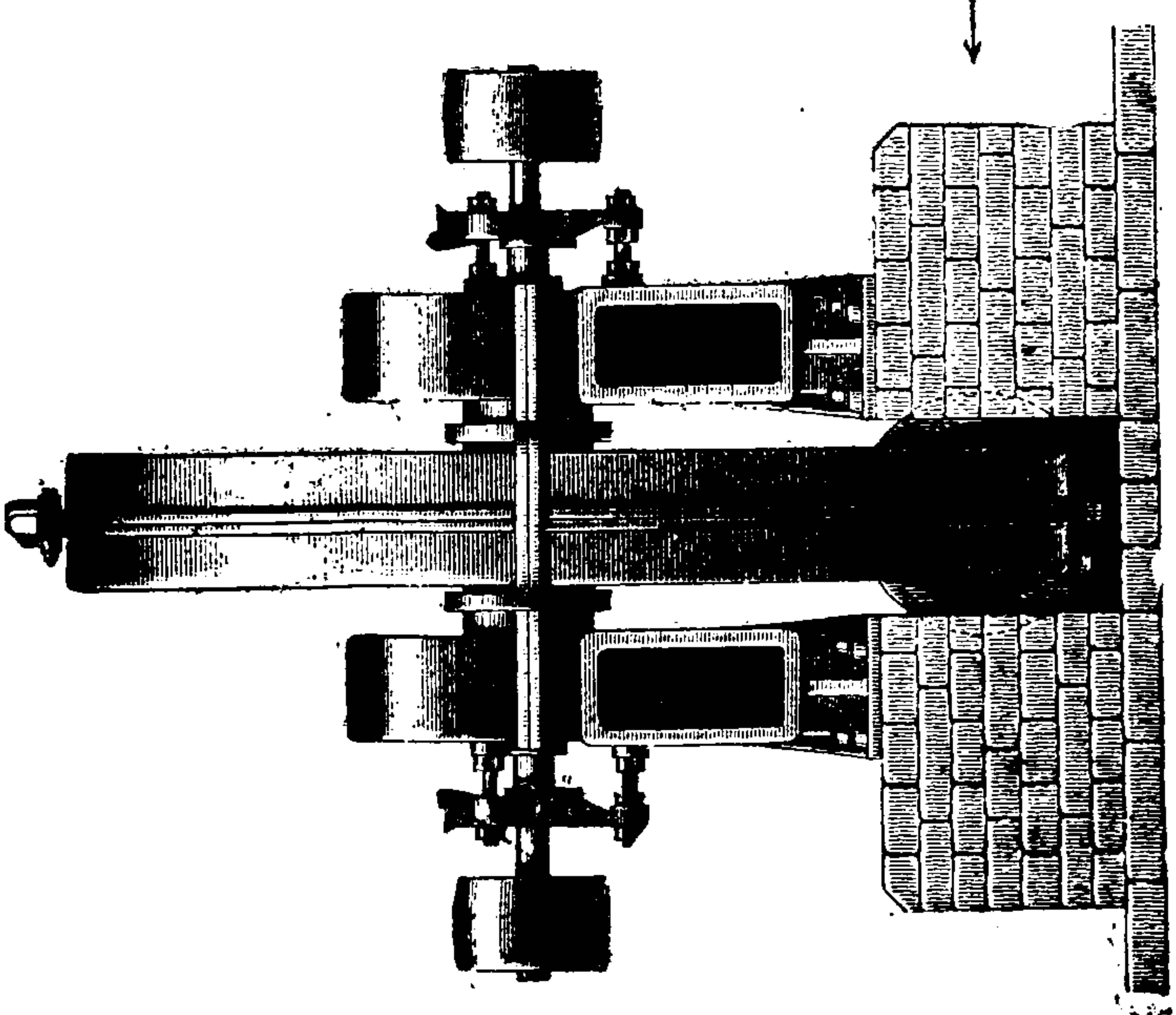


Fig. 74.

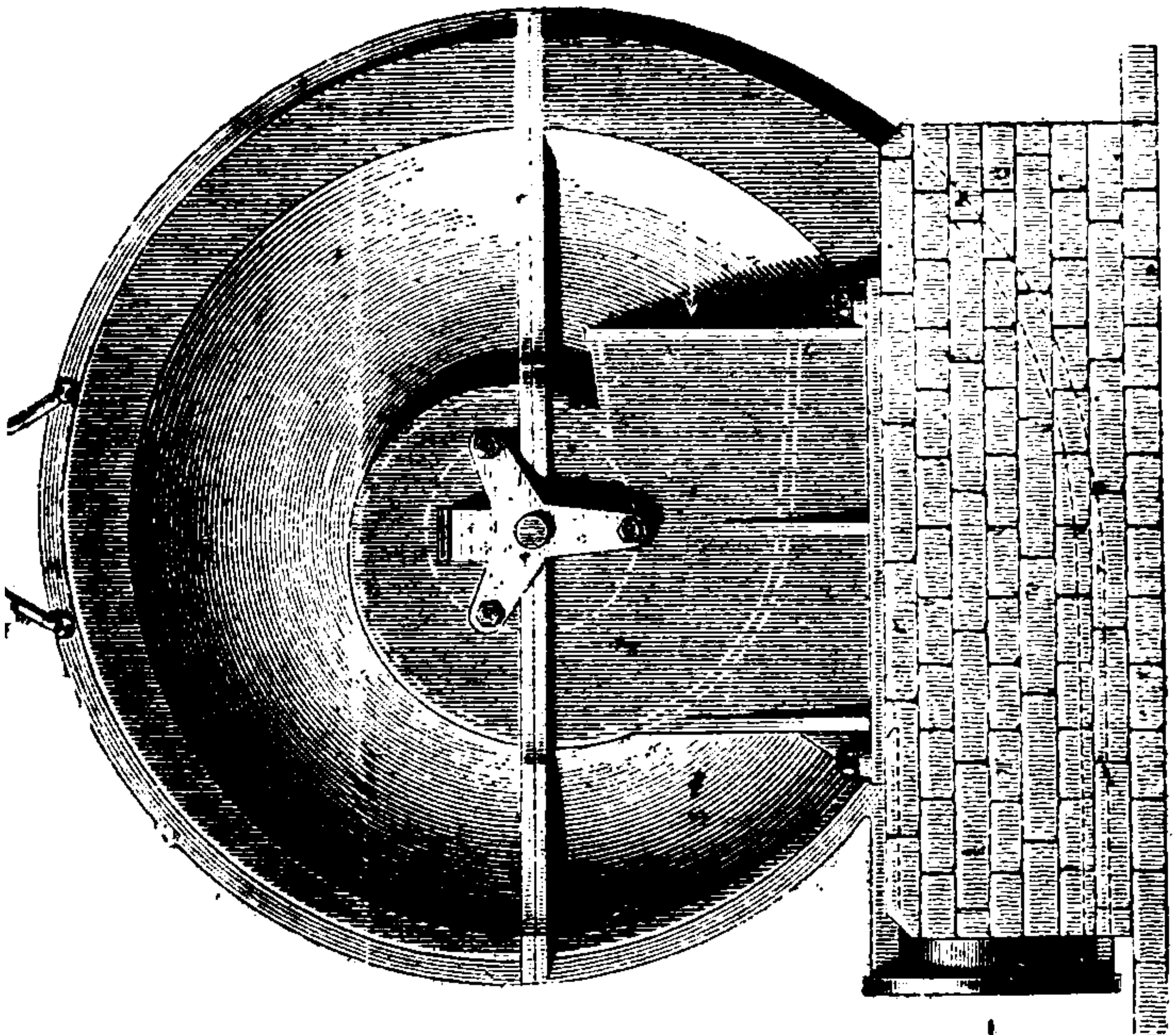


Fig. 75.

Kley Ventilator.

TABLE 32.—EFFICIENCY OF A KLEY FAN.

Rev. of fan per min.	Water gauge in the suction pipe in inches. h.	Cu. ft. of air per sec. Q.	Approximate manometric efficiency per cent., assuming $H=\frac{10000}{144}h$. η_m.	Mechanical efficiency of engine and fan per cent.
30	0·86	493	90	
40	1·35	622	79	54
50	1·97	751	74	
60	2·85	881	74	56
70	3·84	1,020	74	
72	4·10	1,055	74	58

TABLE 33.—TESTS OF KLEY FANS (SMALLER TYPE).

No. of fan.	Discharge in cu. ft. per min. Corre- sponding revs.	Water gauges in inches.					Dia. of wheel in inches.	Dia. of discharge pipe in inches.
		7·89	11·8	15·75	19·70	23·60		
1	Discharge	705	880	1,055	1,230	1,410	29·6	8·87
1	Revs. ...	1,289	1,569	1,810	2,026	2,216		
2	Discharge	1,230	1,410	1,760	1,930	2,110	39·4	11·8
2	Revs. ...	960	1,177	1,358	1,520	1,662		
3	Discharge	1,760	2,110	2,640	3,000	3,170	49·25	14 8
3	Revs. ...	768	942	1,086	1,216	1,330		
4	Discharge	2,810	3,520	4,230	4,760	5,290	61·5	18·4
4	Revs. ...	615	755	870	974	1,065		
5	Discharge	5,640	6,700	8,450	9,510	10,550	87·5	26·2
5	Revs. ...	—	—	—	—	—		

These fans are constructed with inflow at one or both sides; for forges and foundries from 11·8 in. to 7·2 ft. diameter, and for mine ventilation from 16·4 to 39·3 ft. diameter. Table 32 gives a series of experiments upon a ventilator of this type of the following dimensions: External diameter, $29\frac{1}{2}$ ft.; internal diameter, 19·65 ft.; external breadth, 2·62 ft.; internal breadth, 3·94 ft.

Table 33, with figs. 74 and 75 (for which we have to thank C. Mehler, of Aachen), gives particulars of the smaller fans of this type.

Pelzer Dortmund Fan.—This is shown in figs. 76 to 79. It is largely used on the Continent, and has in late years been greatly improved. Fig. 79 shows the wheel fitted with twelve curved vanes, which receive the air without shock. After entering the wheel axially the air is received by vanes which are plane and radial, and with this arrangement the manometric efficiency is usually about 50 per cent, but by alterations in the construction a considerably higher value can be obtained.

Figs. 76, 77, and 78 show the diffuser, volute, and chimney, together with the plant for driving the fan. The aerial pressure on the wheel is balanced by allowing the air from the diffuser to flow into the conical spaces surrounding the circumference of the wheel on the suction side. A thrust to the right is thus obtained to balance the thrust that naturally acts towards the left or suction side.

These fans are made with diameters between 11·8 in. and 19·7 in., while the breadths of the wheel are—

0·8	of the dia.	for a water	gauge of	0·4	inch.
0·8	„	„	„	0·2	„
0 7	„			1·8	„
75	„			2	inches.
7				3	„
7				4	,
7				6	,
7				8	,
6				10	,
6				12	
0·6				14	

Bumstead and Chandler Fan Experiments.—An example of this type of fan is shown in sectional plan and elevation, figs. 80, 81. It is fitted with a pair of tandem engines, one on each side of the fan, which can be run separately if desired; they have high-pressure cylinders, 16 in. diameter, low-pressure, 24 in., and a 16-in. stroke. These engines when running at 220 rev. indicate 320 horse power. The fan is 15 ft. in diameter and 6 ft. 6 in. wide, and is capable of discharging 250,000 cu. ft. of air per min., but as it was

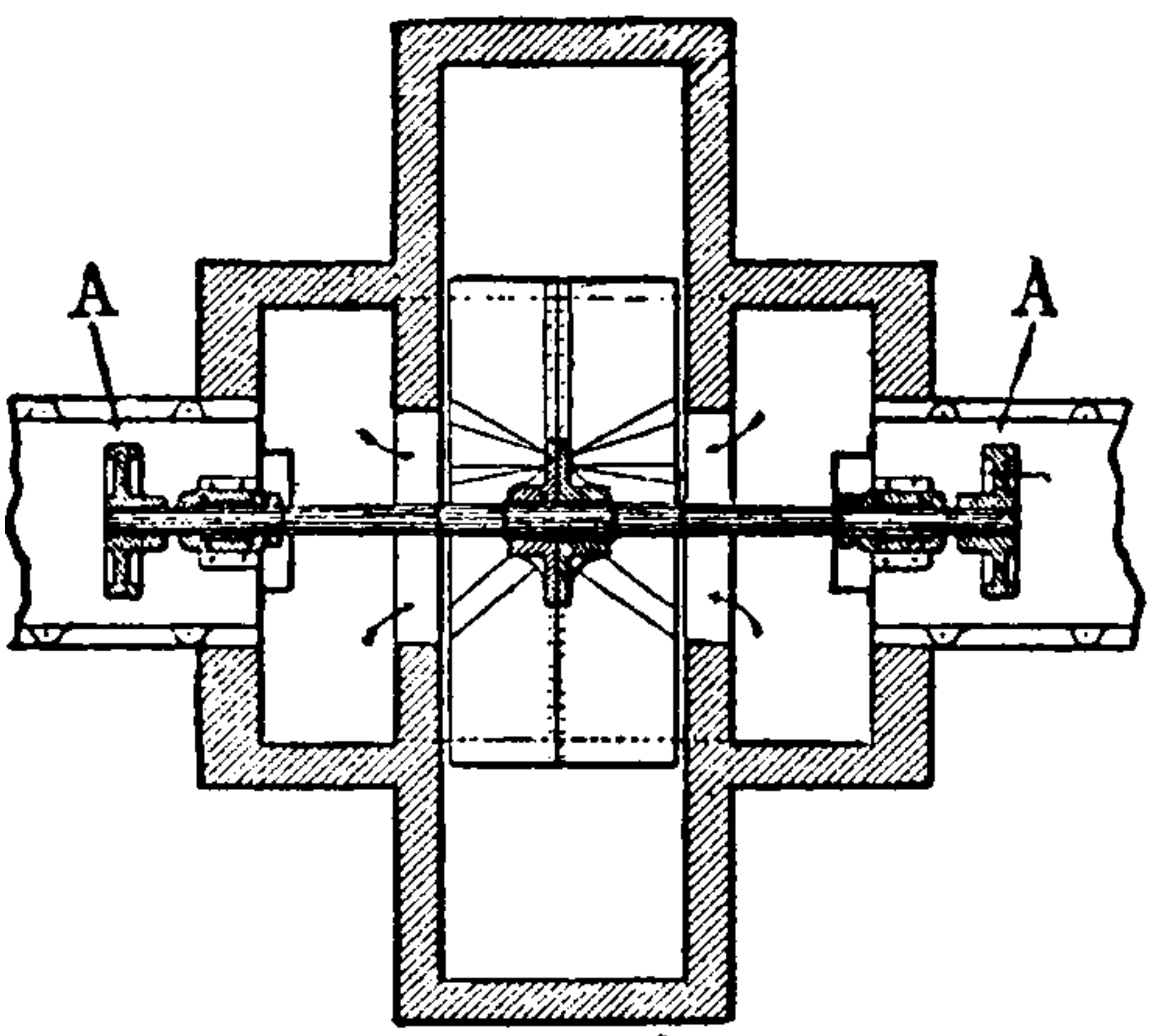


FIG. 80.—BUMSTEAD AND CHANDLER FAN: TOP SECTIONAL VIEW.

A A, Couplings.

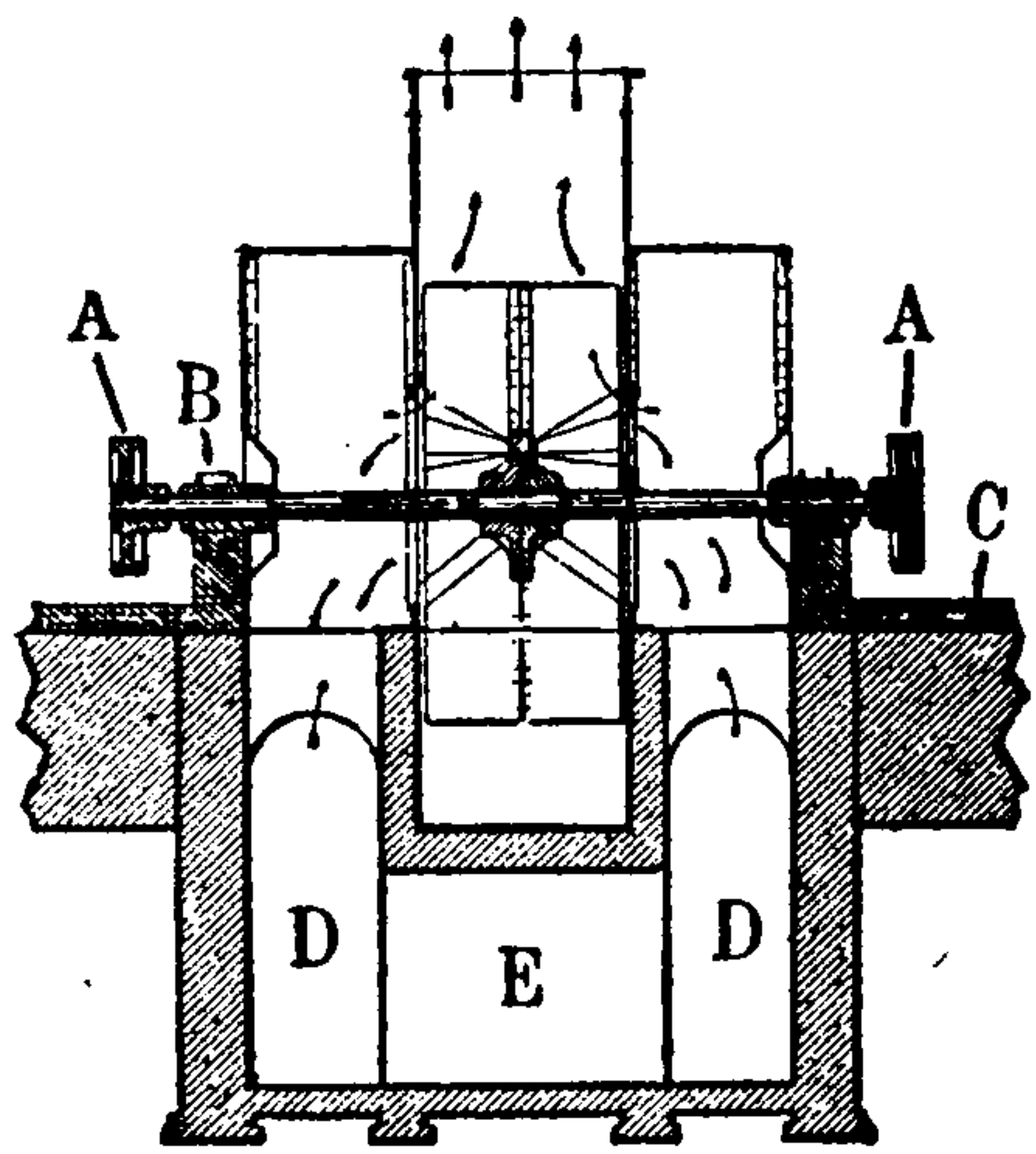


FIG. 81.—BUMSTEAD AND CHANDLER FAN: END SECTIONAL VIEW.

A A, Couplings; *B*, adjustable bearing; *C*, engine base; *D D*, drift; *E*, arch-connecting drifts.

found impossible to get this quantity through the mine at the stipulated water gauge in the fan drift, it was decided to permit some air to enter the top of the upcast shaft, and then to measure the total volume in the fan drift, which was of ample area and length to obtain accurate measurements. The air from the mine enters the fan at both sides, and is delivered upwards to the atmosphere. The chimney is very large, in order that the air may be discharged at low velocity. The casing is a volute. The blades of the fan are mounted on a steel disc, $10\frac{1}{2}$ ft. diameter.

Experiments to find the best form of blade proved to

TABLE 34.—BUMSTEAD AND CHANDLER FAN EXPERIMENTS.

No. of test.	Rev. per min.	Water gauge in the fan drift 30 ft. from the fan.	Volume passed through the fan drift, cu. ft. per sec.	Useful work done by fan in H.P.	I.H.P. of the engines.	Efficiency of engine and fan.	Fan blade tip speed in ft. per sec.	Manometric efficiency per cent.
1	150	2·8	2,860	75·9	106·5	71·28	117·8	45·0
2	151	3·1	2,885	84·5	117·2	72·1	118·6	49·1
3	204	5·2	3,470	170·6	260·2	65·5	160·2	45·3
4	205	4·75	4,100	184·3	261·4	70·5	161·0	41·0
5	204	5·5	[illegible]	186·4	263·5	70·74	160·2	48·0
6	200	4·5	3,885	165·1	230·0	71·5	157·0	[illegible]
7	202	2·8	4,650	—	—	—	158·6	24·8
8	203	3·1	4,650	—	—	—	159·4	27·4

the satisfaction of the makers that a modified **S** form is the best, with the inner end of the blade curved forward in the direction of rotation, so as to cut into the air, and gradually raise its velocity as it passes outwards.

The fan shaft bearings are provided with a vertical adjustment; they are not placed in the air drift in the usual way, but are isolated therefrom by a sheet-steel cover. They are thus free from all dirt and dust that passes through the fan. The results of a series of tests made by M. Strick, manager of the Cossall Colliery, are given in Table 34.

The manometric efficiency has been added, the values of H being calculated by the formula

$$H = \frac{10000}{144} h.$$

Ser Fan Experiments.—This fan has been described (see p. 107) and illustrated (see p. 108 and figs. 55 and 56). We add some experiments with a small blowing fan :—[18]

External diameter	19·7 in.
Internal diameter	11·8 in.
Length of vanes radially	3·94 in.
Width of vanes parallel to the axis	3·55 in.
Cross-section of the discharge pipe	10·25 × 9·84 in. = 0·7 sq. ft.

TABLE 35.—SMALL SER FAN EXPERIMENTS.

Rev. of fan per min.	Water gauge in inches. h.	Manometric efficiency per cent.	H.P. measured at the dynamometer.	Useful H.P. of fan.	Mech. efficiency of fan alone.	Cu. ft. of air per sec. Q.	Equivalent orifice in sq. ft. $\frac{Q}{43 \cdot 5\sqrt{h}}$
1,292	5·26	93·5	7·8	4·96	63·6	99·5	1·0
1,094	3·69	91·0	4·65	2·91	62·6	83·2	0·975
1,002	3·16	93·0	3·45	2·31	67·0	77·0	0·996
830	2·22	95·0	2·37	1·35	57·0	64·5	0·995

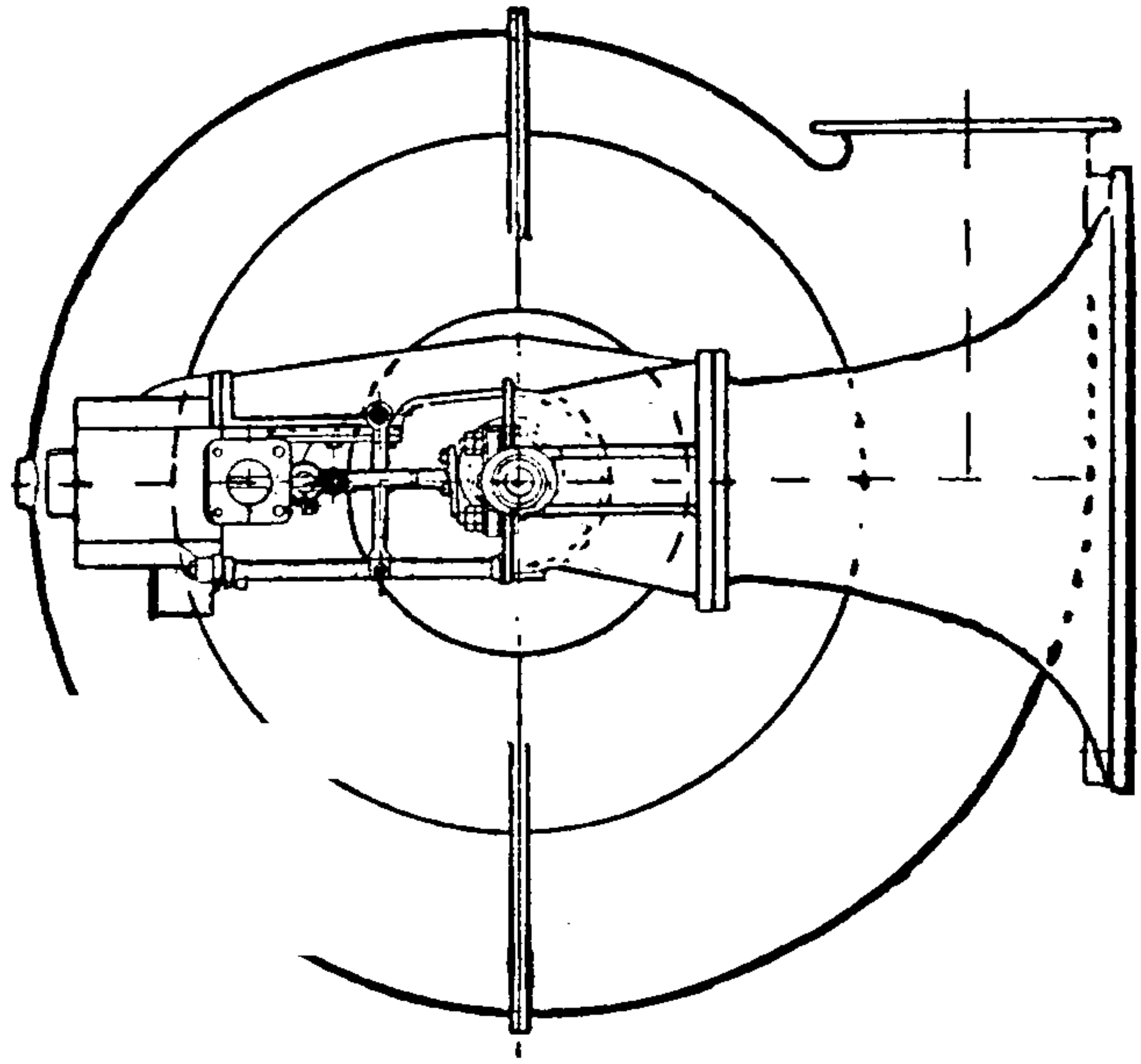


Fig. 83.

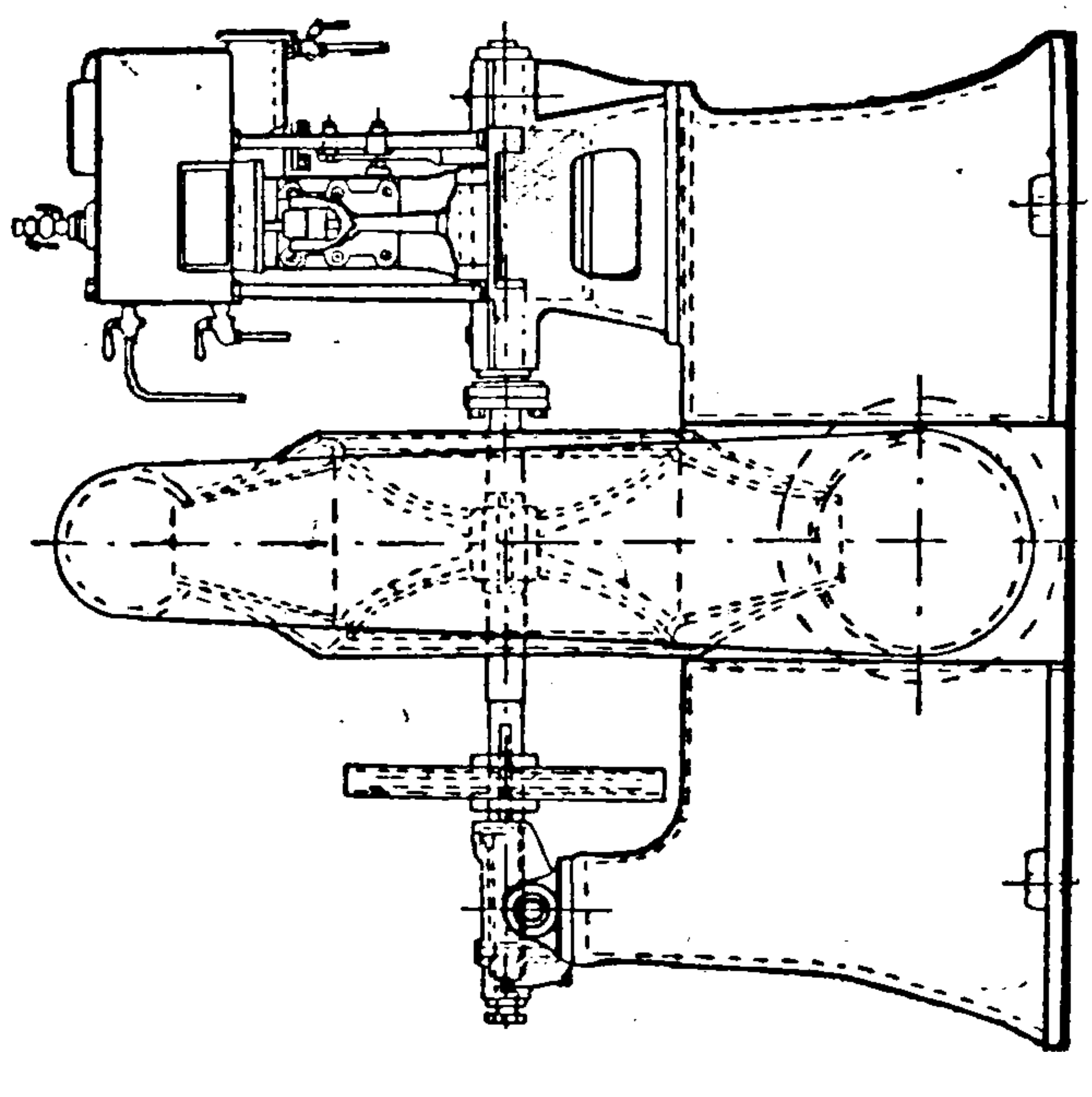


Fig. 82.

Beck and Henkel Fan.

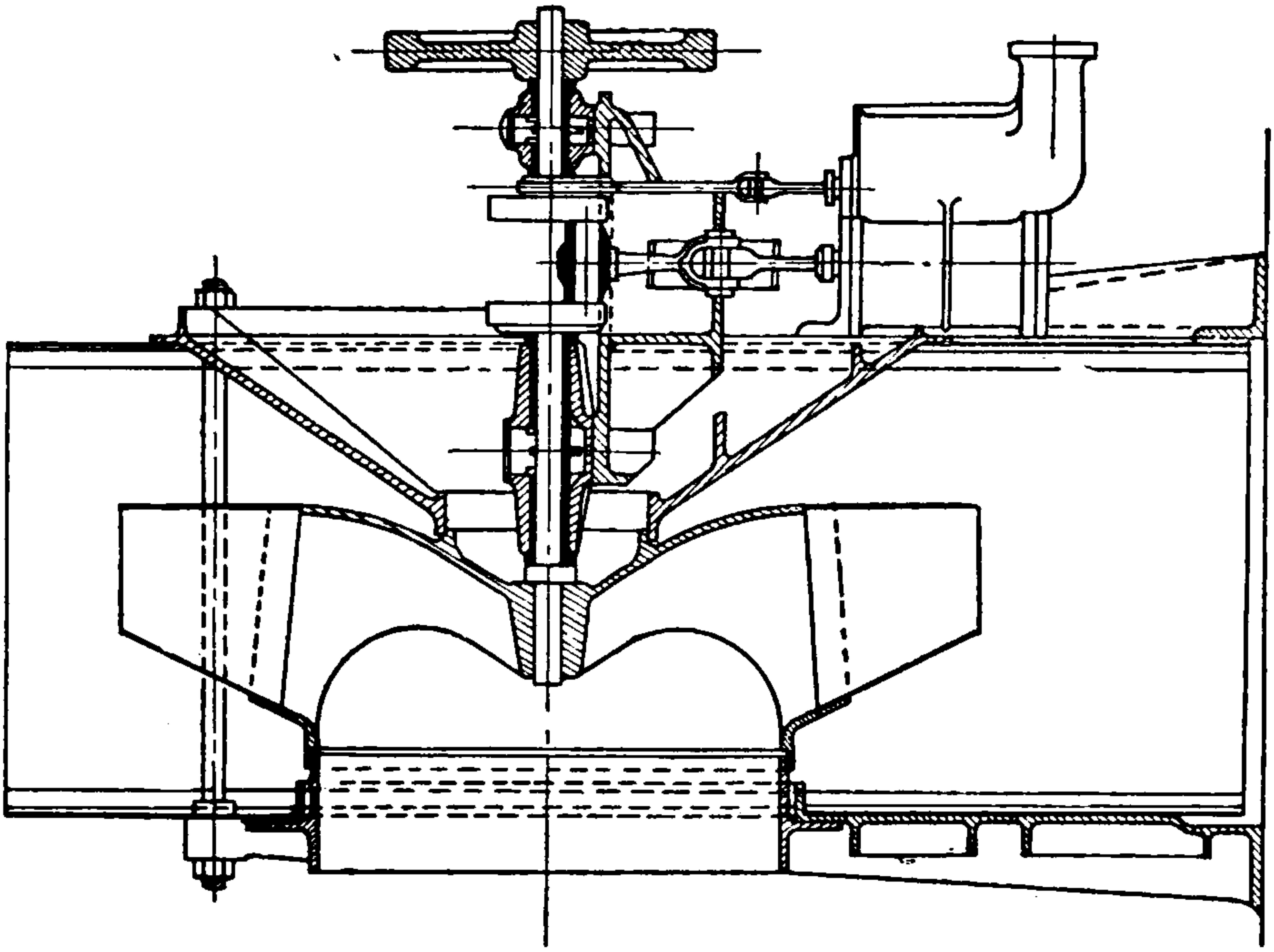


FIG. 85.

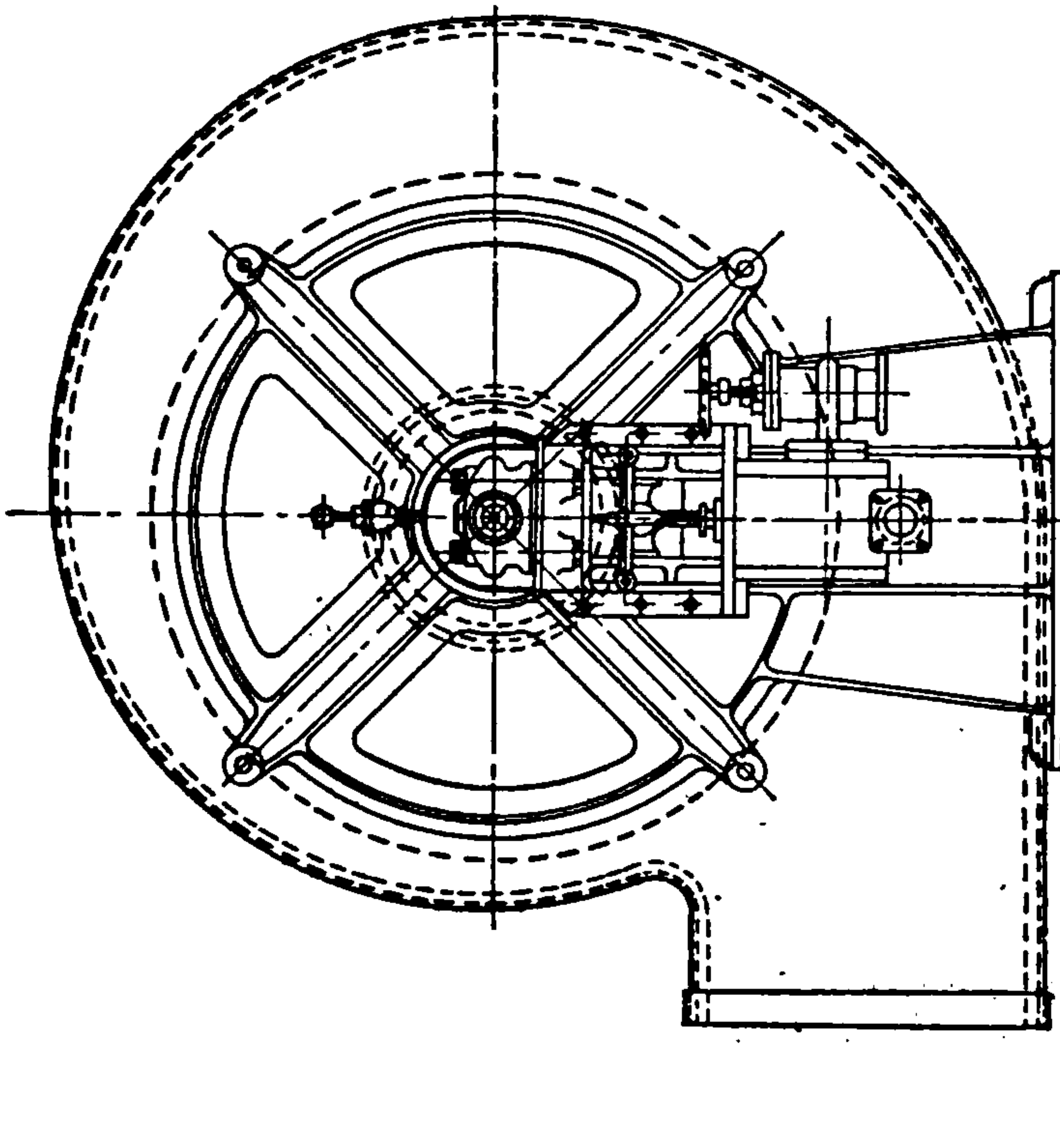


FIG. 84.

BECK AND HENKEL FAN.

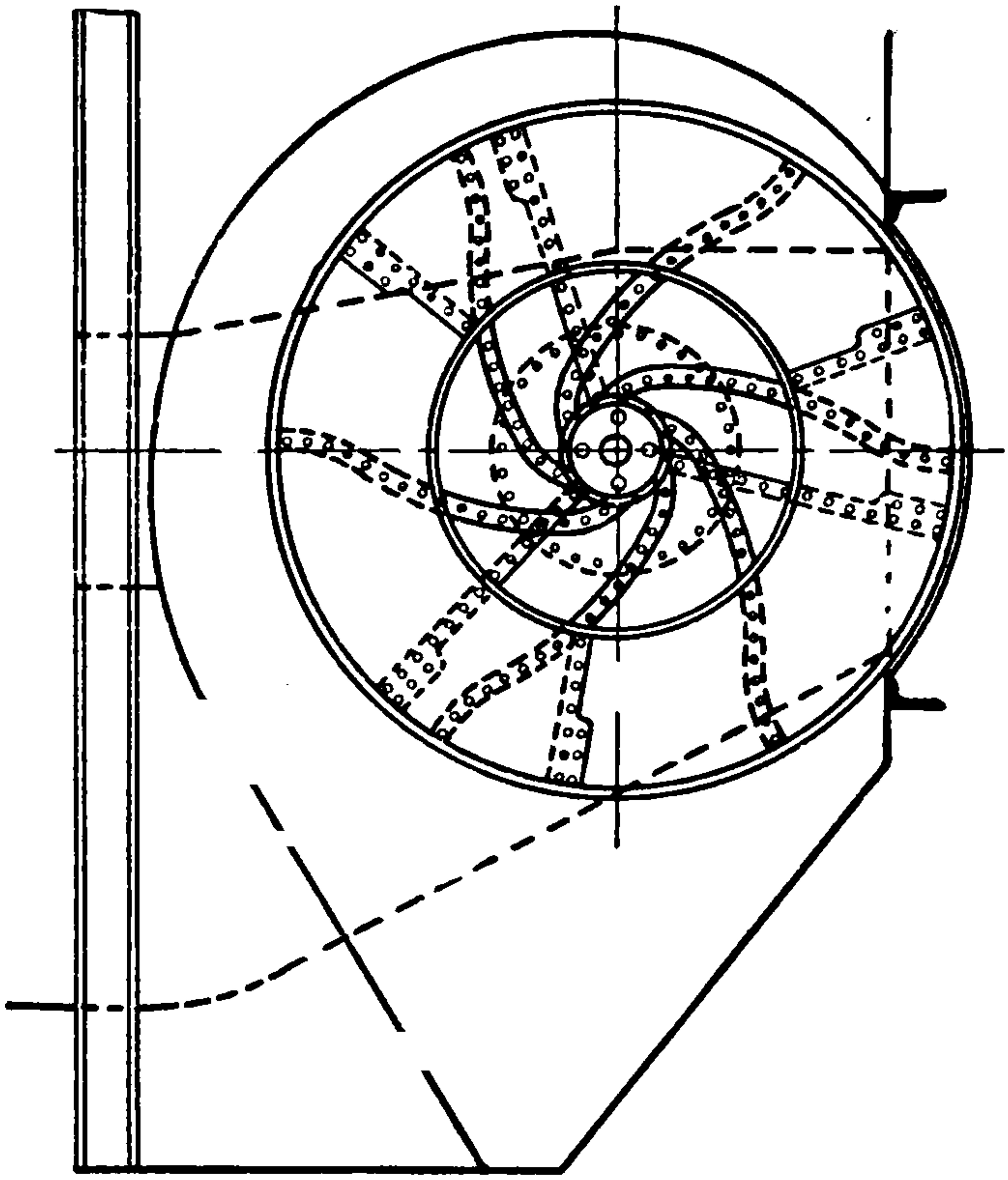


FIG. 87.

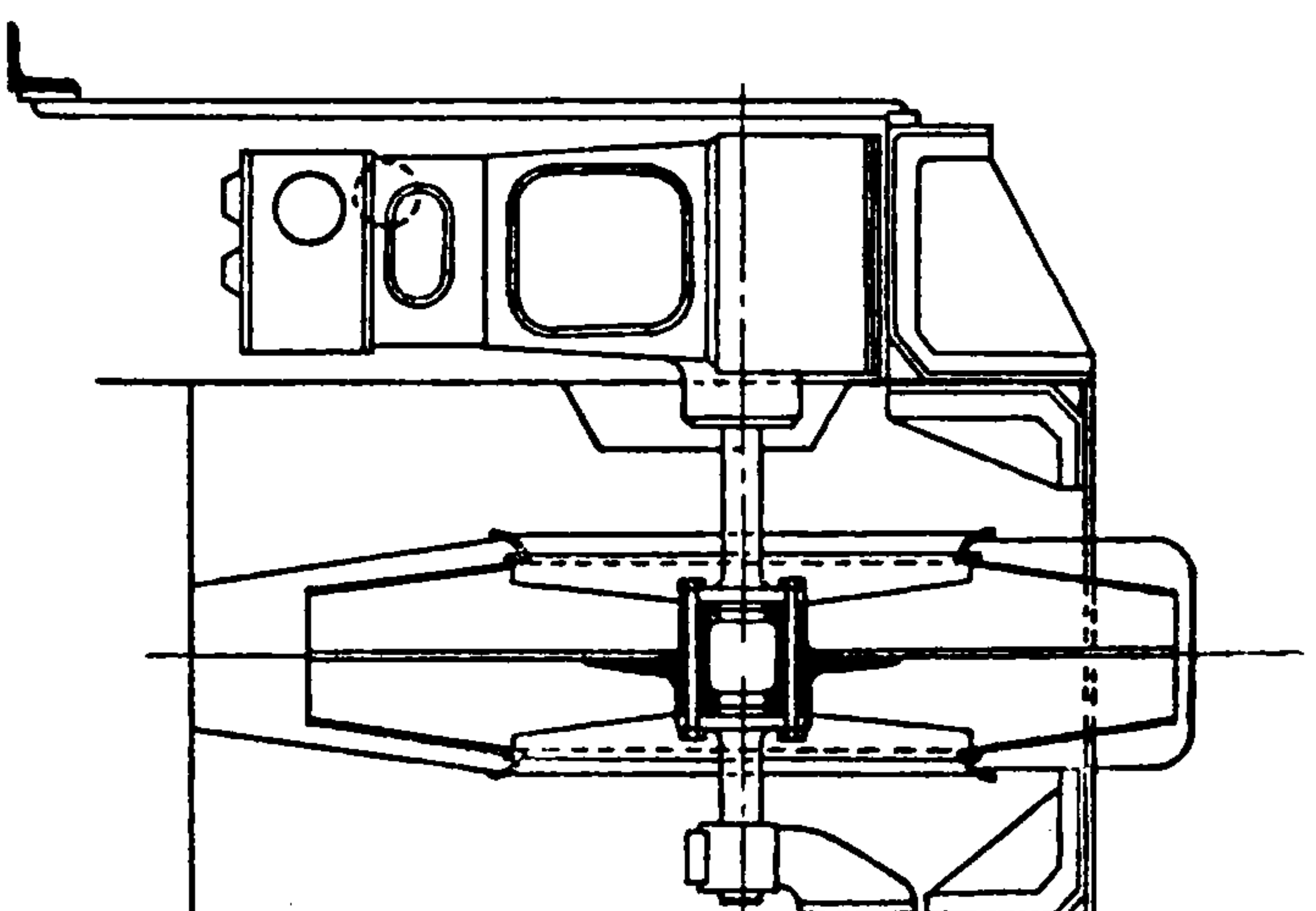
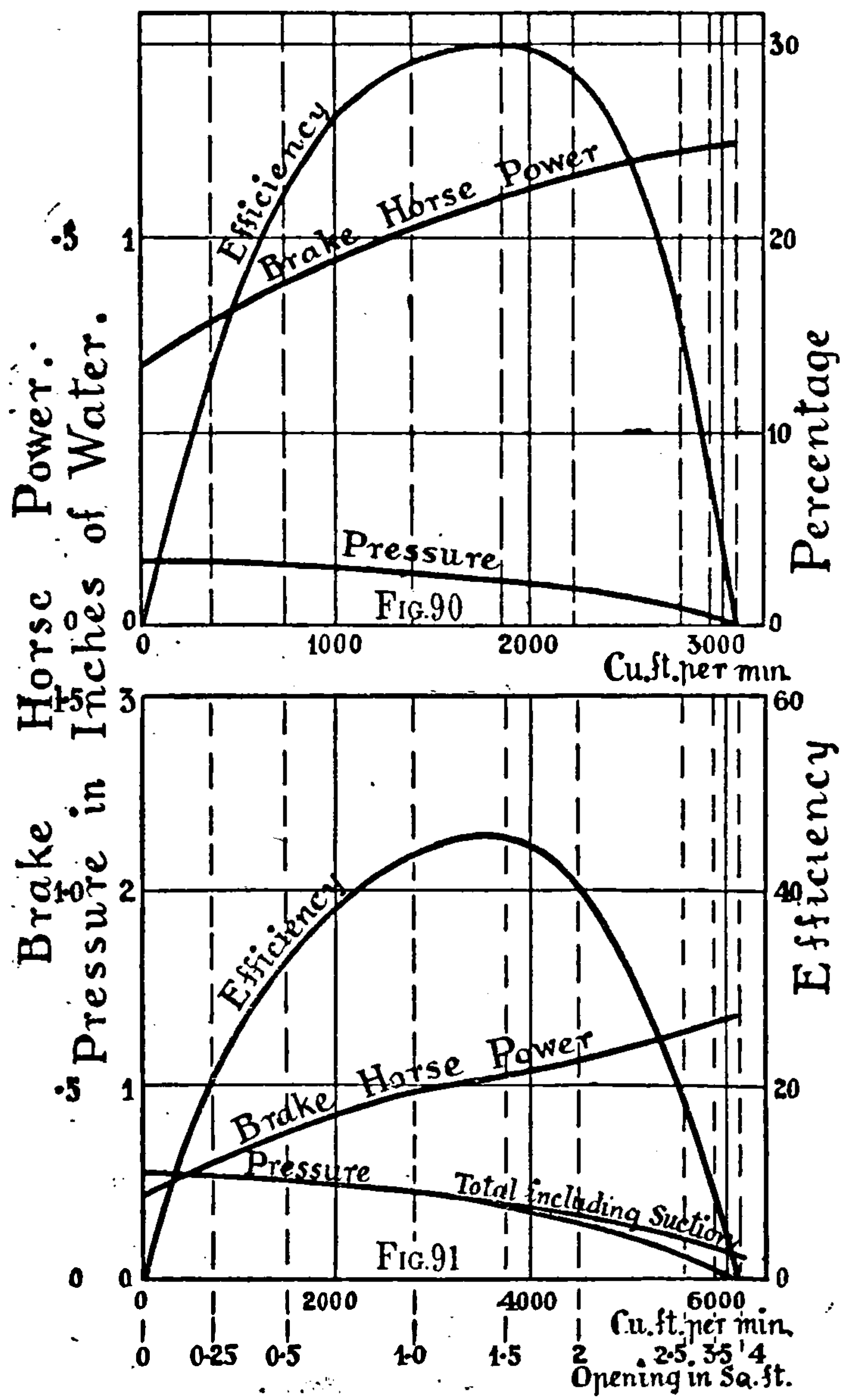


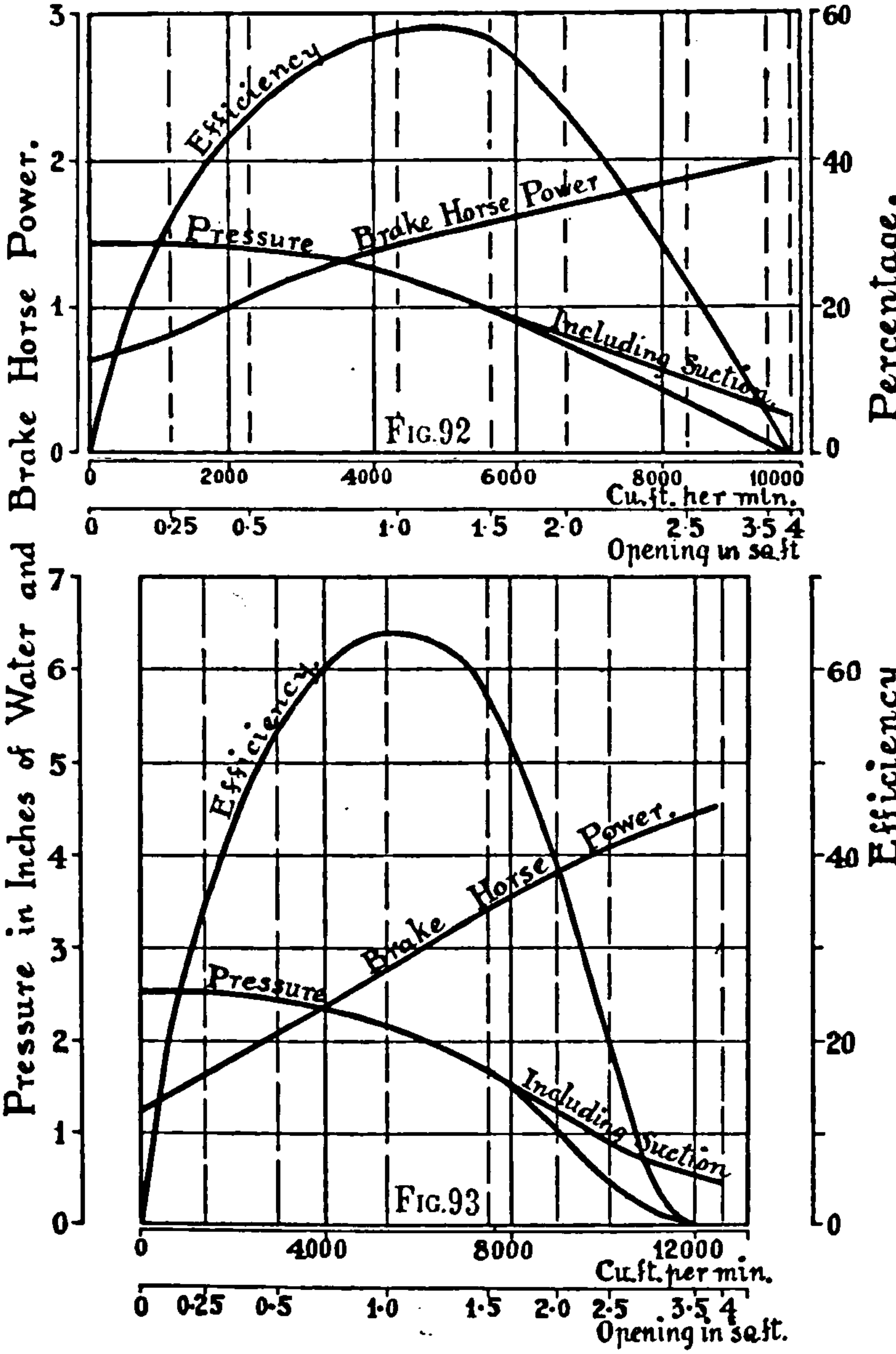
FIG. 86.

66" DOUBLE INLET FORCED DRAUGHT FAN.



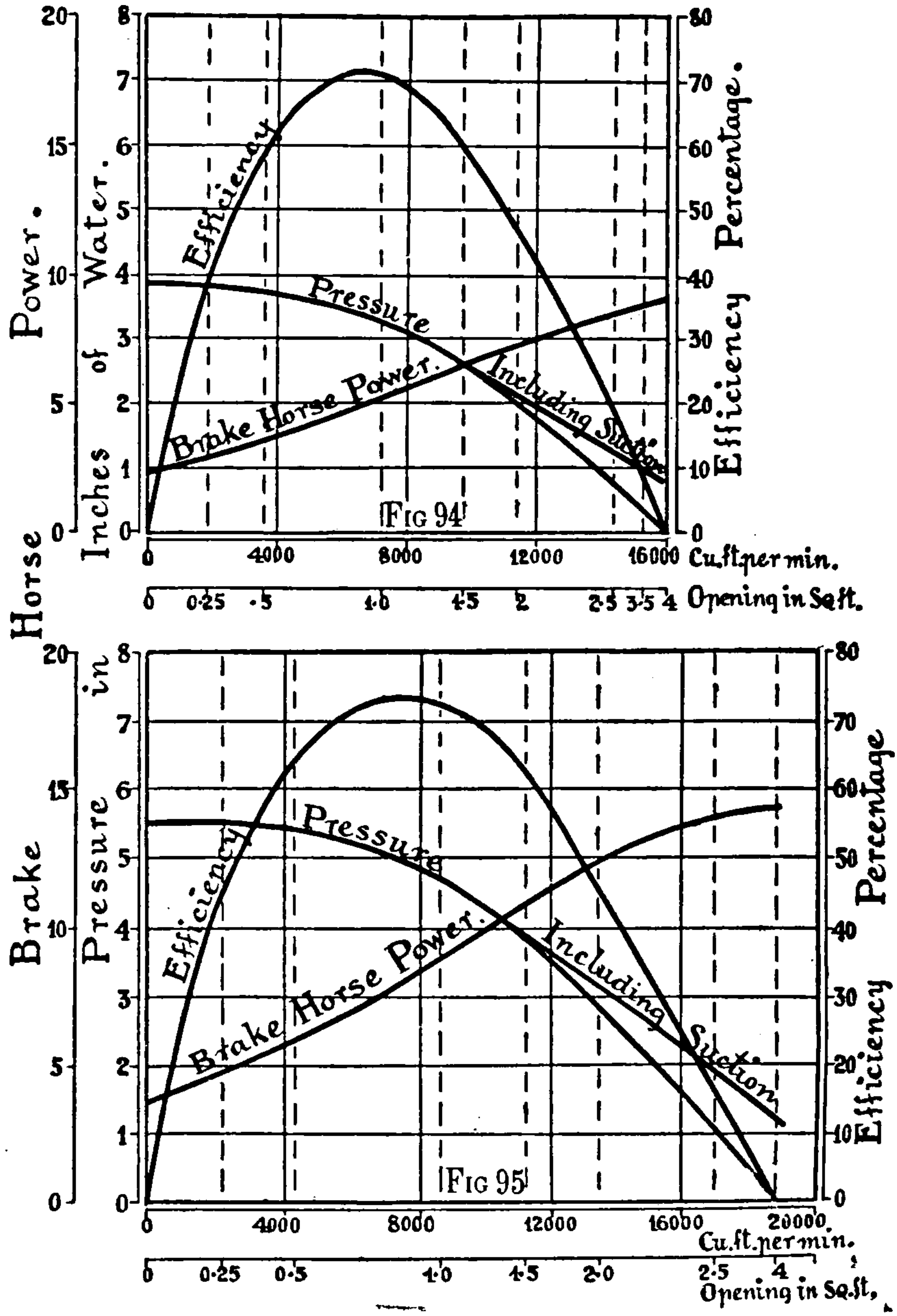
FIGS. 90 AND 91.—TESTS OF 4′ 6″ SINGLE INLET FAN. MOTOR DRIVEN.

90.—100 rev. per min. 91.—200 rev. per min.



FIGS. 92 AND 93.—TESTS OF 4′ 6″ SINGLE INLET FAN. MOTOR DRIVEN.

92.—300 rev. per min. 93.—400 rev. per min.



FIGS. 94 AND 95.—TESTS OF 4′ 6″ SINGLE INLET FAN. MOTOR DRIVEN.

94.—500 rev. per min. 95.—600 rev. per. min.

pages. In the first place, it will be noticed that the discharge when the water gauge is zero is a little more than thirty times the number of rev. per min., and if the equivalent orifices are calculated when the discharge in cu. ft. is fifteen times the rev. per min., the following table results :—

Rev. per min. ...	200	300	400	600	700	800
Cu. ft. per min. ...	3,000	4,500	6,000	9,000	10,500	12,000
Water gauge in inches	0·5	1·2	2·0	4·6	6·1	8·3
Equivalent orifice ...	1·63	1·58	1·63	1·61	1·64	1·60

The equivalent orifice is $\dfrac{Q}{0{\cdot}65\sqrt{2gH}}$, where Q is cu. ft. per sec. and H is head of air in ft. Taking

$$H = h \times \frac{10000}{144},$$

which gives a correct value for the average densities of air and water, we obtain the equivalent orifice

$$O = \frac{Q}{0{\cdot}92 \times 60 \times \frac{100}{12}\sqrt{gh}} = \frac{Q}{2600\sqrt{h}},$$

Q being here taken in cu. ft. per min., from which the values in the above table are calculated. This shows that when $Q \propto v_2$ tip speed, $Q \propto \sqrt{h}$; thus it follows that the manometric efficiency at this equivalent orifice is constant. It can be calculated by the formula

$$\eta_m = \frac{gH}{v_2^2} = \frac{g\,\frac{10000}{144}\,h}{(2\pi r_2 \frac{N}{60})^2} = \frac{40300\,h}{N^2},$$

where N = number of rev. per min. Hence the manometric efficiency at this orifice is approximately 50 per cent. The mechanical efficiency decreases with the speed, as the brake horse power of the motor that drives the fan is partly absorbed by the work done on the air by the wheel, and by the surface friction of the outside of the wheel, two quantities of work which vary as the cube of the revolutions; but in addition to this there is the bearing friction, which, although not strictly accurate, we will take proportional to the revolutions.

If we assume

$$\text{B.H.P.} = A\left(\frac{N}{1000}\right)^3 + C\,\frac{N}{1000},$$

$$\text{or } \frac{\text{B H.P.}}{\frac{N}{1000}} = A\left(\frac{N}{1000}\right)^2 + C,$$

then for 800 and 400 rev. we get

$$\frac{20}{0{\cdot}8} = 0{\cdot}64\,A + C$$

$$\text{and } \frac{6}{0{\cdot}8} = 0{\cdot}16\,A + C,$$

so that $A = 36{\cdot}5$ and $C = 1{\cdot}67$.

This gives the following table:—

B.H.P. actual	13·7	8·50	1·50	0·500
B.H.P. calculated ...	13·7	8·88	1·48	0·625
Rev. per min.	700	600	300	200

which shows a very close agreement, and justifies our assumptions. The useful work varies as the cube of the revolutions for $Q \propto \sqrt{h} \propto N$ at a given orifice, and since the useful work $\propto Q\,h$, it also $\propto N^3$. The air efficiency

$$\eta = \frac{\text{useful work done}}{\text{work transmitted to wheel}}$$

is therefore probably a constant quantity.

Work expended in Overcoming Disc Friction.—In the foregoing analysis we asserted that that portion of the brake horse power required to overcome the surface friction between the air and the outside of the wheel varied as the cube of the revolutions. A proof of this statement may be conveniently introduced here. We will assume that the surface of the disc is of such a nature that a film of air adheres to the surface, so that if a certain area s sq. ft. is moving with a linear speed v ft. per sec , the retarding force

$$F = c\,\sigma\,s\,v^2 \text{ lb.},$$

where c is a constant and σ is the density of the air in cu. ft. per lb. Applying this to an elementary annulus, a radius r ft. and thickness dr, fig. 98, we have

$$s = 2\pi r\,dr$$

$$\text{and } u = \omega r,$$

$$\text{so that } dF = c\sigma \,.\, 2\pi r\,dr\,\omega^2 r^2$$
$$= c\,2\pi\sigma\omega^2 r^3\,dr \text{ lb.}$$

dF is the resistance opposing the motion of the annulus.

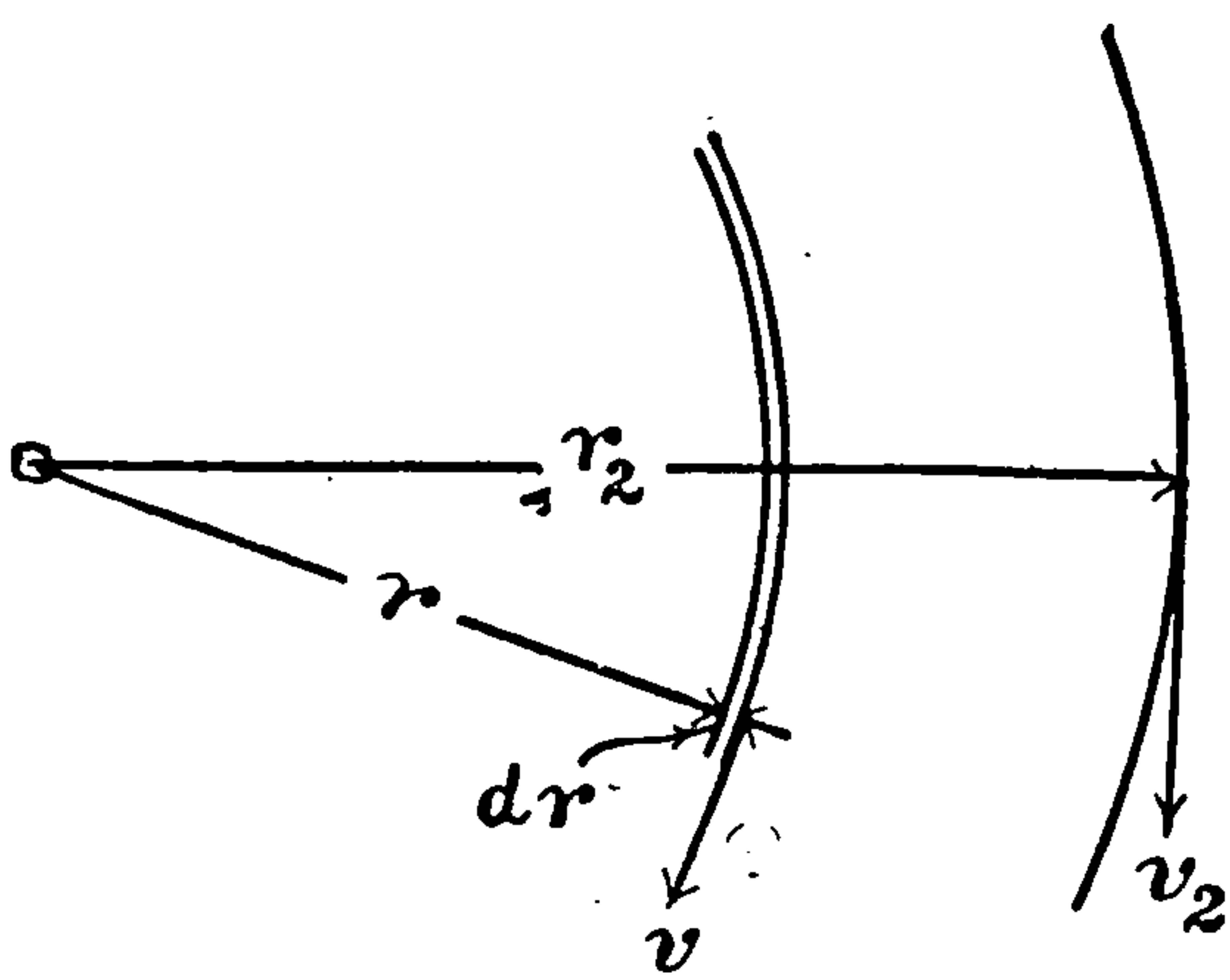


FIG. 98.—DISC FRICTION.

The resisting couple $r\,dF = c\,2\,\pi\,\sigma\,\omega^2\,r^4\,dr$ lb.-ft., so that the total couple due to the whole disc

$$c = \int_0^{r_2} r\,dF$$
$$= c\,2\pi\sigma\omega^2 \frac{r_2^5}{5}.$$

The work done per sec. by the impeller in overcoming this resistance

$$= c\,\omega = c\,2\pi\sigma\omega^3 \frac{r_2^5}{5}$$

$$\therefore \text{ D.H.P.} = \frac{c\,\omega}{550} = c\,\sigma\,\omega^3 r_2^5$$
$$= c\,\sigma\,v_2^3\,r_2^2,$$

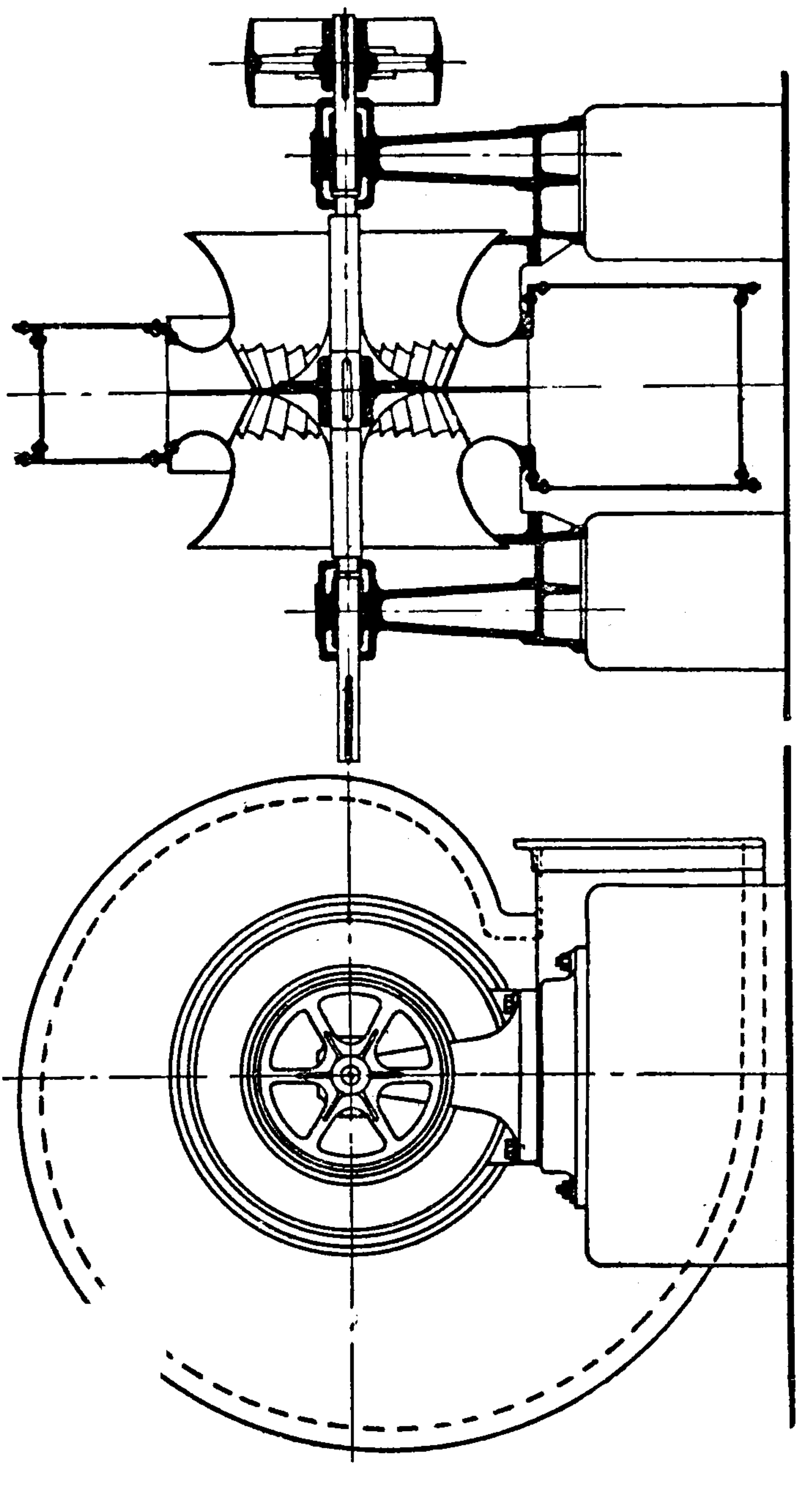


Fig. 99. Fig. 100.

Geneste-Herscher Fan.

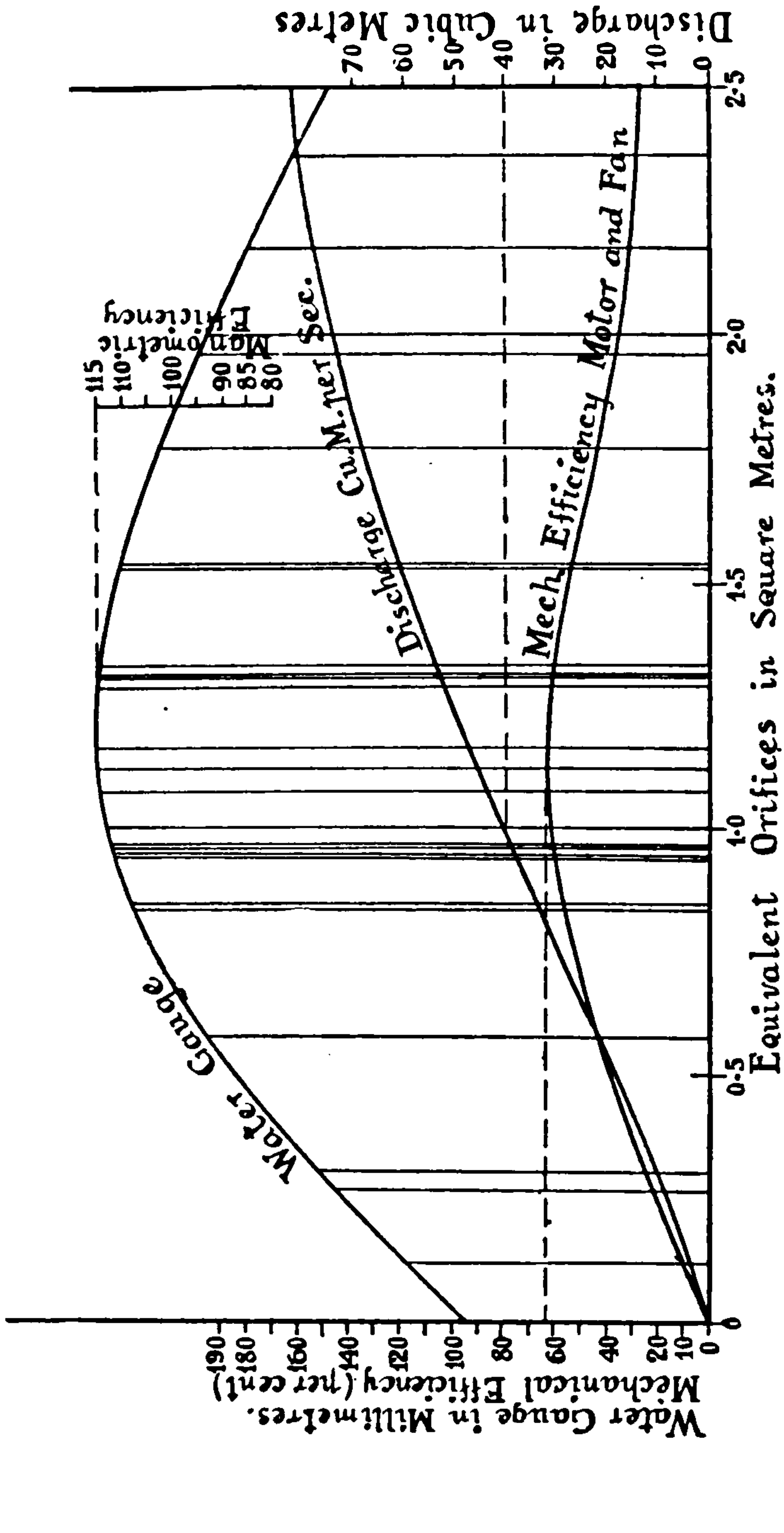


FIG. 101.—DIAGRAM OF GENESTE-HERSCHER FAN TESTS.

which gives the same reduced orifice as that of the Blanzy fan, multiplied by $\frac{9}{10}$ to allow for exaggeration by the anemometer. Then

$$\frac{gH}{a_2 v_2} = 0{\cdot}70$$

$$\sqrt{M}\frac{\sqrt{gH}}{v_2 - b_2 \cot\phi} = 0{\cdot}70$$

$$\frac{v_2}{\sqrt{gH}} - \frac{1{\cdot}05}{0{\cdot}70} = \frac{b_2}{\sqrt{gH}}\cot\phi$$

$$\frac{0{\cdot}95 - 1{\cdot}50}{\frac{b_2}{\sqrt{gH}}} = \cot\phi.$$

But
$$b_2 = \frac{Q}{2\pi r_2 s_2} = \frac{O \times 0{\cdot}92\sqrt{gH}}{\pi \times 2{\cdot}62 \times 0{\cdot}584}$$
$$= 0{\cdot}647\sqrt{gH}.$$
$$\therefore \cot\phi = -\frac{0{\cdot}55}{0{\cdot}647} = -0{\cdot}85$$
$$\phi = 130° - 22'$$

the angle of outflow.

The velocity of whirl $a_2 = v_2 - b_2 \cot\phi$

$$= (0{\cdot}95 + 0{\cdot}55)\sqrt{gH} = 1{\cdot}50\sqrt{gH}.$$

That in the volute

$$c_v = \frac{144\,Q}{17{\cdot}6 \times 17{\cdot}1}$$

$$= \frac{144 \times 3{\cdot}38 \times 0{\cdot}92}{17{\cdot}6 \times 17{\cdot}1}\sqrt{gH} = 1{\cdot}49\sqrt{gH}.$$

It is clear that a chimney is necessary to reduce c_v, which would cause a loss of head, if not reduced, of

$$\frac{c_v^2}{2g} = 1{\cdot}11\,H,$$

so that the efficiency would be below 30 per cent., since

$$\frac{c_v^2}{2a_2 v_2} = \frac{1{\cdot}11}{0{\cdot}95 \times 1{\cdot}50} = 0{\cdot}78.$$

Table 36 gives the calculated and experimental values for H for the Rateau fan of Table 20. The quantities are in this case in the metric system. The last two columns show how very closely the fan obeys the law.

Table 37 gives a comparison of the manometric efficiency obtained by experiment and that from

$$\frac{1}{\eta_m} - 0{\cdot}136 \frac{O}{\sqrt{\eta_m}} - 14{\cdot}18\, O^2 - 0{\cdot}69 = 0,$$

which is equation (39) for fan No. VIII.

TABLE 37.—FAN NO. VIII. : BRYAN DONKIN'S EXPERIMENTS.

Equivalent orifice in sq. ft.	Actual manometric efficiency.	Calculated manometric efficiency.
	Per cent.	Per cent.
0	59·0	59·0
0·1	52·5	54·5
0·2	43·0	43·5
0·3	32·0	33·1
0·4	24·0	24·7
0·5	19·0	18·6
0·8	9·5	9·0
1·0	6·6	6·1
1·5	3·0	2·87

Fans VI., X., and XI. have been treated in a similar manner in Tables 38, 39, and 40 respectively.

TABLE 38.—FAN NO. VI.: DONKIN'S EXPERIMENTS.

Equivalent orifice in sq. ft.	Calculated manometric efficiency.	Actual manometric efficiency.
	Per cent.	Per cent.
0	60·0	60·0
0·1	57·5	58·0
0·2	54·1	54·0
0·3	50·0	50·0
0·4	44·4	43·0
0·5	36·25	36·25
0·8	20·7	22·0
1·0	14·25	15·0

$$\frac{1}{\eta_m} + 0·192 \frac{O}{\sqrt{\eta_m}} - 5·85\, O^2 - 1·666 = 0.$$

TABLE 39.—FAN NO. X.: DONKIN'S EXPERIMENTS.

Equivalent orifice in sq. ft.	Calculated manometric efficiency.	Actual manometric efficiency.
	Per cent.	Per cent.
0	57·0	57·0
0·1	63·0	60·0
0·2	59·8	59·8
0·3	51·0	52·0
0·4	41·6	41·6
0·5	33·0	33·0
0·8	17·7	17·8
1·0	12·1	13·5
1·5	6·34	7·5

$$\frac{1}{\eta_m} + 2·43 \frac{O}{\sqrt{\eta_m}} - 13·55\, O^2 - 1·755 = 0.$$

TABLE 40.—FAN NO. XI.: DONKIN'S EXPERIMENTS.

Equivalent orifice in sq. ft.	Calculated manometric efficiency.	Actual manometric efficiency.
	Per cent.	Per cent.
0	28·5	28·5
0·1*	32·8	26·0
0·24	19·0	19·0
0·3	14·5	15·5
0·4	9·47	10·5
0·5	6·5	6·5
0·8	2·78	2·8
1·0	1·8	2·0
1·5	0·08	1·0

* The smallest equivalent orifice at which a test was made, except zero orifice, was 0·24 sq. ft.

The equation to the above fan is—

$$\frac{1}{\eta_m} + 10{\cdot}05 \frac{O}{\sqrt{\eta_m}} - 126{\cdot}4\,O^2 - 3{\cdot}51 = 0.$$

Fan No. 1, in the same paper, has an equation

$$v_2{}^2 + 0{\cdot}0649\, v_2\, Q - 4440\, h - 0{\cdot}00196\, Q^2 = 0,$$

where v_2 is the peripheral velocity in ft. per sec., and Q is the number of cu. ft. of air per min., h being the static water gauge. The observed head cannot be expected to agree closely with that calculated from the above equation, as the static water gauges were often lower than they should have been owing to induction, in certain cases a vacuum being shown where there was undoubtedly pressure, for how could the air flowing in a pipe of uniform section pass through baffle plates, and yet with unchanged velocity reach a space of greater pressure, a manifest contradiction of the law of the conservation of energy? The reason for the apparent vacuum was that induction took place, as

TABLE 42.—MANOMETRIC EFFICIENCIES OF THE PARKEND MINE FAN.

Volume in cu. ft. per min. per sq. in. of diametrical section, $\div \sqrt{\text{water gauge}}$.	Manometric efficiency.	Tip speed in ft. per min.
Zero	0·604	9,000
Zero	0·602	5,000
Zero	0·600	8,000
Zero	0·603	6,000
0·91	0·6 nearly	8,000
1·48	0·603	6,000
2·76	0·587	8,000
3·68	0·594	9,000
4·51	0·563	5,000
5·06	0·559	6,000
5·68	0·562	8,000
5·89	0·565	9,000
5·91	0·554	9,000
7·34	0·483	5,000
7·62	0·495	9,000
8·48	0·447	6,000
10·39	0·375	8,000
12·0	0·322	5,000
12·10	0·327	9,000
15·5	0·237	9,000
16·0	0·224	6,000
23·5	0·125	8,000
26·8	0·100	8,000
27·2	0·099	9,000

We now come to a formula much harder to prove, viz., that the air efficiency

$$\eta = \frac{g\,H}{a_2\,v_2}.$$

The difficulty is due to the exaggeration of the discharge and possibly also of the water gauge. It is also in many cases difficult to determine ϕ, and therefore a_2; further, ϕ is often a variable quantity. In Donkin's paper, however, where the records of the discharge can be taken as being accurate, experiments are submitted in which the agreement between the air efficiency η is almost perfect except at very small orifices, where the formula does not hold good, owing to the fact that losses of energy which we neglect in the above formula become of consequence. At these small orifices the friction between the wheel disc and the air, and that of the air against the fan casing during its passage through the wheel, become of importance. The former varies as v_2^2, whilst the corresponding work wasted varies as v_2^3. The latter is a quantity of the second degree in v_2 and Q—*i.e.*, it may be expressed by the formula

$$v_2^2 + m\, v_2\, Q + n\, Q^2.$$

The importance of these terms becomes manifest when we point out that when Q is zero

$$\frac{gH}{a_2 v_2} = \frac{gH}{v_2^2} = \eta_m = \tfrac{1}{2} \text{ on the average,}$$

whilst at zero orifice the actual air efficiency is zero.

Fan No. 1 in Donkin's experiments is a Rateau fan, and ten experiments with this are given in Table 7. There are twenty wrought-iron vanes which are inclined forwards at 45 deg. to the tangent to the outer circumference. Also from Table 5, for fans discharging against a small head,

$$b_2 = \frac{Q}{2\pi r_2 s_2} = \frac{Q}{r_2^2} \text{ approximately.}$$

$$\text{Also } a_2 = v_2 - b_2,\ \cot\phi = v_2 + b_2,$$

as $\phi = 135$ deg.; assuming that the angle of discharge coincides with the vane angle, and that there is no coefficient of contraction at discharge from the wheel. These assumptions are justifiable, as there are twenty vanes (a large number for a diameter of 19·6 in.), and the vanes are so designed that uniform outflow probably takes place. It has been proved by experiment that the inflow to the fan is uniform. Then

$$\frac{gH}{a_2 v_2} = \frac{32\cdot2 \times 10000\,h}{144\,v_2\,(v_2 + b_2)},$$

where h is the dynamic water gauge. As an example in experiment 6,

$$Q = 2700,\ v_2 = 110\cdot8,$$
$$b_2 = 67\cdot2,\ a_2 = 178,$$
$$h = 4\cdot75\ ;\ \therefore\ \frac{gH}{a_2 v_2} = 0\cdot54,$$

as compared with 0·594 in the experiment.

Table 43 gives the experimental efficiencies and the corresponding values of η:—

TABLE 43.—FAN NO. I.: DONKIN'S EXPERIMENTS.

Experimental efficiency.	Calculated air efficiency.	Air efficiency difference.
12·0	11·9	0·1
17·1	15·3	1·8
21·3	20·8	0·5
33·9	30·9	3·0
47·0	43·8	3·2
59·4	53·8	5·6
59·9	59·3	0·6
60·6	57·7	2·9
49·9	62 6	−12·7
0	50·0	−50·0

and shows that there is a very close agreement between the two, except in the last two cases; this difference, however, was predicted.

Fan No. 3 had plane radial vanes, so that the outflow must have been radial, and whether uniform or no is of no consequence, because $a_2 = v_2$. It will be clear to the reader that the outflow must have been radial, because we are justified in supposing the wheel fixed and a centrifugal accelerative force acting on each particle. It must not be

From Table 13, taking the third experiment with each fan, we get Table 47.

In all the above except 7, 8, 10 of Table 20, 1 and 9 of Table 22, and 3 of Table 13, the values of η are less than the mechanical efficiencies, although these include engine friction. If we multiply η by $\frac{9}{10}$, which is the highest engine mechanical efficiency we are justified in assuming, all experiments except 3 of 13, 8 of 20, and 1 of Table 22, which are at very small reduced orifices, give values of η below the corresponding mechanical efficiencies. The greatest discrepancies are those in Table 13, where the experimenters acknowledge that their discharges are too high. It is reasonable to suppose that the discrepancies are therefore mainly due to exaggerated discharge, because in this case it would reduce η, for by an increase of b_2 we increase a_2. In Table 48 we have supposed the actual discharge in Table 20 to be $\frac{9}{10}$ths of that given by the anemometer, so that η is increased and the mechanical efficiency reduced. We get the following results:—

Now, except at small orifices such as 8, and 9, which is omitted, as it is practically zero, efficiency of fan $=\eta$ very nearly

$$=\frac{\text{efficiency of engine and fan}}{\text{efficiency of engine}}.$$

TABLE 45.—RATEAU FAN: TYPE A, TABLE 1.

No. of experiment.	η per cent.	Mechanical efficiency per cent.
1	32·3	33
2	33·8	37
3	37·1	43
4	41·7	46
5	48·5	53
6	56·4	60
7	62·7	61
8	60·7	43
10	61·2	59

TABLE 46.—RATEAU FAN: 9·17′ DIAMETER.

No. of experiment.	η per cent.	Mechanical efficiency per cent.
1	68	56
5	45	48
6	32	35
7	57	57
8	54	57
9	54	53

TABLE 47.—RATEAU FANS: BELGIAN COMMISSION.

No. of fan.	η per cent.	Mechanical efficiency per cent.
1	29·8	48·3
2	44·4	82·6
3	54·5	47·0
4	50 5	77·5

Hence efficiency of engine

$$= \frac{\text{efficiency of engine and fan}}{\eta},$$

and should be about 0·85 to 0·90. We infer from the above that the exaggeration of the discharge is greater than what we have assumed, viz., the ratio $\frac{10}{9}$, but not very much, and this agrees fairly well with the results of the Prussian Commission if we allow for the additional exaggeration due to the variable velocity of the air which is always found in mines.

TABLE 48.—RATEAU FAN : TYPE A, TABLE 1.

No. of experiment.	η per cent.	Mechanical efficiency per cent.	$\dfrac{\text{Mechanical efficiency}}{\eta}$.
1	34·2	29·7	0·87
2	35·8	33 3	0·93
3	39·2	38·7	0·99
4	44·1	41·4	0·94
5	50·9	47·6	0·94
6	58·9	54·0	0·92
7	64·9	54·9	0·85
8	62·0	38·7	0·62
10	63·5	53·1	0·84

Let us next consider Heenan and Gilbert's experiment with a fan 17 in. diameter and 8 in. wide. The efficiency here given is the ratio of the useful work done by the fan to the work done on a shaft which drives the fan by a belt, so that it should not be much less than the efficiency of the fan alone, or η at orifices of moderate size, if we can assume the truth of the statement that the work done per lb. by the wheel is $a_2 v_2 \div g$. In fig. 23 we have the results of experiments with a blade terminating at 35 deg. to the circumference, so that if the angle of relative outflow were the same as the vane angle we should have $\phi = 35$ deg. Consider the experiment in which the discharge was 2,000 cu. ft. per min., and the tip speed 12,000 ft. per min. The total water gauge is 9·6 in., so that we have

$$v_2 = \frac{12000}{60} = 200, \; Q = \frac{2000}{60}, \text{ and } b_2 = \frac{Q}{2 \pi r_2 s_2}$$

$$= \frac{2000 \times 144}{60 \times \pi \times 17 \times 8} = 11{\cdot}22 \text{ ft. per sec.}$$

If we consider the discharge of 3,500 cu. ft. per min.

$$Q=\frac{3500}{60},\ b_2=\frac{3500\times 144}{60\times\pi\times 17\times 8}=19\cdot 7,$$

$v_2=200$, as before, and the dynamic or total efficiency, from fig. $23=0\ 7$; also

$$H=7\cdot 6\times\frac{10000}{144}\text{ ft, so that } a_2=\frac{32\cdot 2\times 76000}{200\times 144\times 0\cdot 7}=121,$$

$$\cot\phi=\frac{200-121}{19\cdot 7}=\frac{79}{19\cdot 7}=4\cdot 01,\ \phi=14\text{ deg.},$$

and even if we suppose a coefficient of contraction of 0 8,

$$\phi=17\text{ deg. }18\text{ min.}$$

Fig. 25 shows similar curves for blade No. 4, fig. 22, which has double curvature, and terminates radially at the outer circumference, so that if the angle of flow was the same as the angle of vanes, ϕ would be 90 deg. The dynamic efficiency, when the flow is 2,500 cu. ft. per min., is 83 per cent., and the dynamic gauge is 12·8 in.

$$b_2=\frac{2500\times 144}{60\times\pi\times 17\times 8}=14\cdot 05,$$

$$H=\frac{128000}{144}=890,\quad a_2=\frac{32\cdot 2\times 890}{200\times 0\cdot 83}=172\cdot 7,$$

$$\cot\phi=\frac{200-172\cdot 7}{14\cdot 05}=\frac{27\cdot 3}{14\cdot 05}=1\cdot 942,$$

$$\phi=27\text{ deg. }14\text{ min.},$$

which we consider extremely improbable, as there are six vanes. In our opinion the discharge is very much exaggerated. The manometric efficiency in this case is

$$\eta_m=\frac{gH}{v_2{}^2}=\frac{32\cdot 2\times 890}{200\times 200}=71\cdot 6\text{ per cent.},$$

which is probably very much nearer the true value of the mechanical efficiency. At the same discharge the compression or static gauge is 11 in., and this gives

$$a_2 = \frac{32 \cdot 2 \times 110000}{200 \times 144 \times 0 \cdot 7} = 175 \cdot 8,$$

since the mechanical efficiency is 70 per cent.;

$$\cot \phi = \frac{200 - 175 \cdot 8}{14 \cdot 05} = \frac{24 \cdot 2}{14 \cdot 05} = 1 \cdot 723,$$

$$\phi = 30 \text{ deg. } 8 \text{ min.},$$

while the manometric efficiency is

$$\eta_m = \frac{32 \cdot 2 \times 110000}{200^2 \times 144} = 0\ 615.$$

The discharge of 4,000 cu. ft. per min. gives the following results: $b_2 = 22 \cdot 5$, $a_2 = 152 \cdot 7$, $\cot \phi = 2 \cdot 10$, $\phi = 25° \ 24'$, the dynamic efficiency being 0·85 and the dynamic water gauge 11·6 in., while the manometric efficiency is 64·7 per cent., and the mechanical efficiency 50 per cent We do not think that the mechanical efficiency is as low as the manometric efficiency, because it is quite possible that ϕ may be less than 90 deg., and we believe that it is. In support of this statement we mention some experiments with a Farcot centrifugal pump at Khatatbeh, Egypt,[19] which were made with very great care, and are probably accurate. In these the mechanical efficiency of engine and pumps was 65 per cent., corresponding to a probable efficiency of pump alone of between 72·2 and 76·5 per cent., while the manometric efficiency was 65 9 per cent.

We shall now discuss some experiments[20] with an open-running fan at the Seghill Colliery. Before doing so we may state that we believe the efficiency is exaggerated. Not knowing the section of the fan drift, we cannot find the correct reduction of the water gauge due to the velocity of the air therein; but, neglecting this, let us consider the total energy of air rejected from the outer circumference of the fan. See Table 49.

TABLE 49.—OPEN RUNNING FAN: SEGHILL COLLIERY.

Average of experiments.	2 to 9.	10 to 17.	18.*
Revs. of fan per min. ...	39·81	60·44	60
Discharge in cu. ft. per min.	87,940	135,700	247,500
Water gauge in drift ...	1·09	2·46	2·14
H.P. in the air	15·10	52·65	83·47
I.H.P.	29·61	96·1	153·7
Mechanical efficiency per cent.	51·02	54·78	54·32

* Separation doors open.

Taking the average of experiments 10 to 17,

$2\,r_2 = 35{\cdot}08$ ft., $s_2 = 1{\cdot}292$,
$N = 60{\cdot}44$ rev. per min., so that $v_2 = 111$ ft. per sec.
the water gauge in the drift $2{\cdot}46''$,

$$\text{the discharge } Q = \frac{135700}{60} = 2262 \text{ cu. ft. per sec.,}$$

$$b_2 = \frac{2265}{\pi \times 35{\cdot}08 \times 1{\cdot}292}, \text{ neglecting contraction.}$$

$$a_2 = \frac{g\,.\,H}{v_2\,.\,\eta}, \text{ where } \eta = 54{\cdot}78,$$

and a_2 will be least if we give η its largest possible value. Assume that the mechanical efficiency of the engine was only 85 per cent., and we get

$$a_2 = \frac{32{\cdot}2 \times 24600 \times 85}{144 \times 111 \times 54{\cdot}78} = 77,$$

and assuming the density of the air as 0·0761, the energy rejected at outflow in H.P. is

$$\frac{b_2^2 + a_2^2}{2g} \times Q \times \frac{0{\cdot}0761}{550} = \frac{77^2 + 15{\cdot}9^2}{64{\cdot}4} \times \frac{2265 \times 0{\cdot}0761}{550} = 30.$$

Adding to this the impeller H.P. 52·6, we account for 82·6 H.P. of the 96·1 I.H.P., giving a total dynamic

Adding to this useful H.P. 97·33, we find the dynamic efficiency is

$$\frac{38{\cdot}6+97{\cdot}3}{144{\cdot}5}=94{\cdot}1 \text{ per cent.},$$

which is, of course, impossible. If we assume that the mechanical efficiency of the engine alone is 90 per cent., the dynamic efficiency becomes

$$\frac{42{\cdot}9+97{\cdot}3}{144{\cdot}5}=97 \text{ per cent.}$$

The only conclusion is that the anemometer very greatly exaggerates the discharge of air.

CHAPTER IX.

HIGH-PRESSURE FANS.

THIS chapter is a summary of a paper by Prof. A. Rateau.[21]

Hitherto fans have not been required to give a water gauge of more than 24 in., while it is only lately that centrifugal pumps have been used for heads over 50 ft. By means of steam turbines and a single pump a head of nearly 1,000 ft. has been obtained. These turbines, pumps, and fans are the design of Prof. Rateau, and those mentioned herein were constructed by Sautter-Harlé, of Paris.

In the theory of centrifugal pumps and fans there are four quantities of importance:

The mechanical efficiency:

$$\eta=\frac{\sigma Q H}{W_s}.$$

The volumetric efficiency:

$$\eta_v=\frac{Q}{v_2 r_2{}^2}.$$

The manometric efficiency:

$$\eta_m=\frac{gH}{v_2}.$$

The coefficient of power transmitted to the shaft of the pump :

$$\tau = \frac{W_s g}{v_2^3 r_2^2 \sigma} = \frac{\eta_m \eta_v}{\eta}.$$

The above quantities are numbers independent of all units. In the formulæ Q is the volume in cu. ft. or cu. metres per sec., H is the head in metres or ft., σ is the weight in lb. or kilogrammes of 1 cu. ft. or metre of the fluid pumped, r_2 is the external radius of the wheel, W_s is the work done per sec. in ft.-lb. or kilogrammetres, and v_2 is the velocity of the wheel at the outer radius. Prof. Rateau

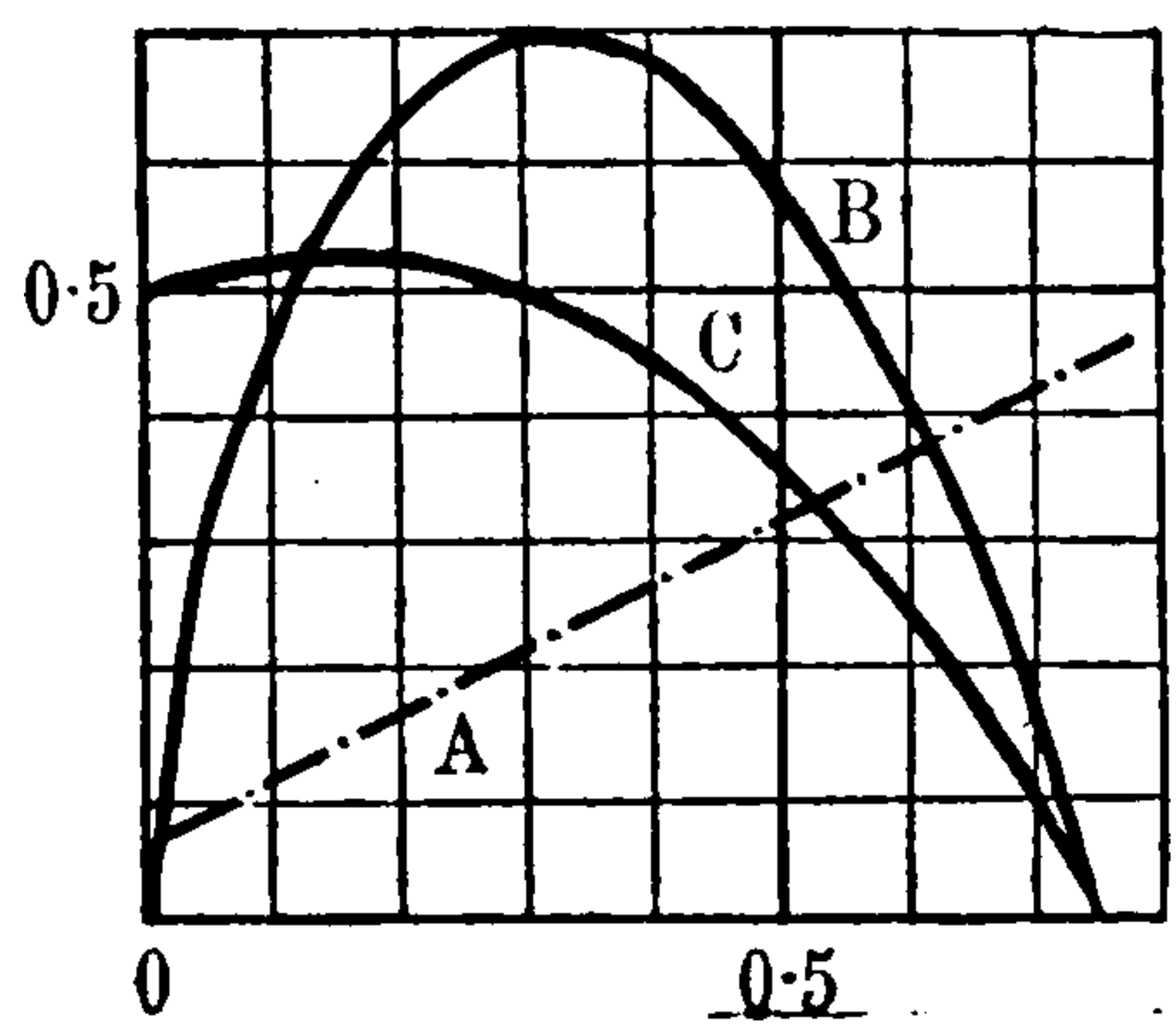


FIG. 102.—CURVES.

$A=\tau$; $B=\eta$; $C=\eta_m$.

used volumetric efficiencies as abscissæ instead of orifices, and drew three curves of η_m, η, and τ, as in figs. 102 and 103. Fig. 102 are those of a centrifugal pump, and 103 of a Rateau fan. For a given pump at a fixed number of rev. per min., the curves τ and η_m are those of W_s and H to suitable scales.

Figs. 104 and 105 show a centrifugal fan driven by a steam turbine and intended to produce a considerable pressure. The fan is made of steel of very good quality, capable of running at a peripheral velocity of over 800 ft. per sec.; it turns in a cast-iron casing having two openings for suction and forming a diffuser and volute. The turbine, which is a steam Pelton wheel, is 11·8 in. in dia., while the fan is 10 in.; the method of raising the oil from a lower to a higher reservoir is shown in fig. 106, in which the pipe A is connected to the lower reservoir and B to the higher. The small tube M brings a small amount of air from the fan, which, mixing in small bubbles with the column of oil B, lowers its specific gravity to an extent sufficient to enable the column A to raise it to the higher reservoir. The discharge or pressure of the fan can be controlled by a pneumatic governor. It consists of a cylinder D, fig. 104, containing a piston P, the details of which are shown in

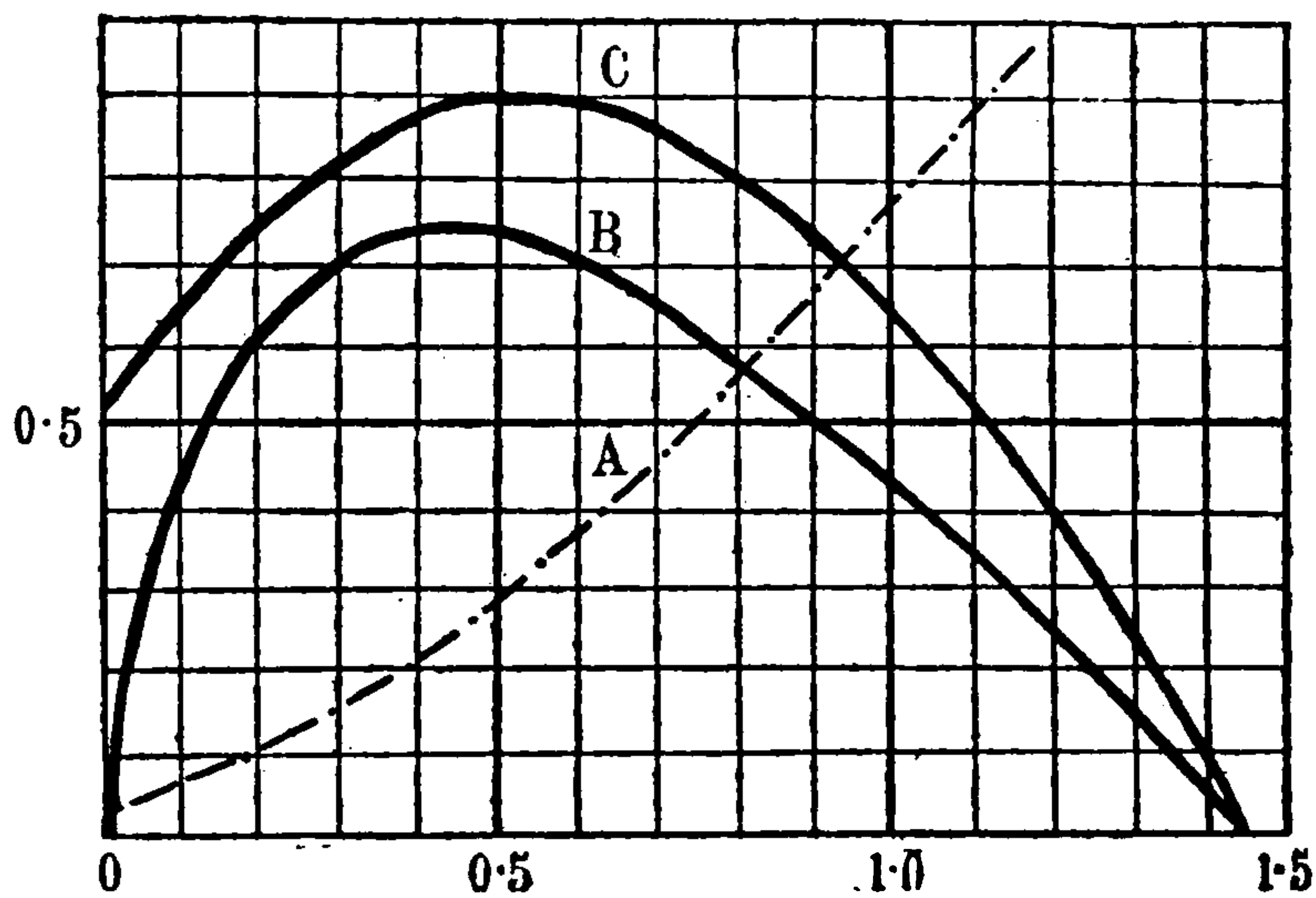


FIG. 103.—CURVES.

$A=\tau$; $B=\eta$; $C=\eta_m$.

fig. 106. The rod of this piston is connected by a short rod to the point C of the lever A B C, which oscillates about the point B, and is connected to the rod of the steam throttle valve at A. The cylinder D has its two ends connected by the pipes 1, 2, figs. 104 and 105, with a straight tube (1) and Pitot tube (2), both placed in the discharge pipe of the fan; see also 3 and 4, fig. 107. Thus the difference of the pressures on the two sides of the piston P is proportional to the square of the velocity of discharge, and exerts an upward pressure which is balanced by the weight of the piston, assisted by an additional weight if necessary and a spring S,

friction a rubber diaphragm is sometimes used. The experiments were made at Sautter-Harlé's Paris works, and the measurements by M. Rateau and M. Chatelain. The

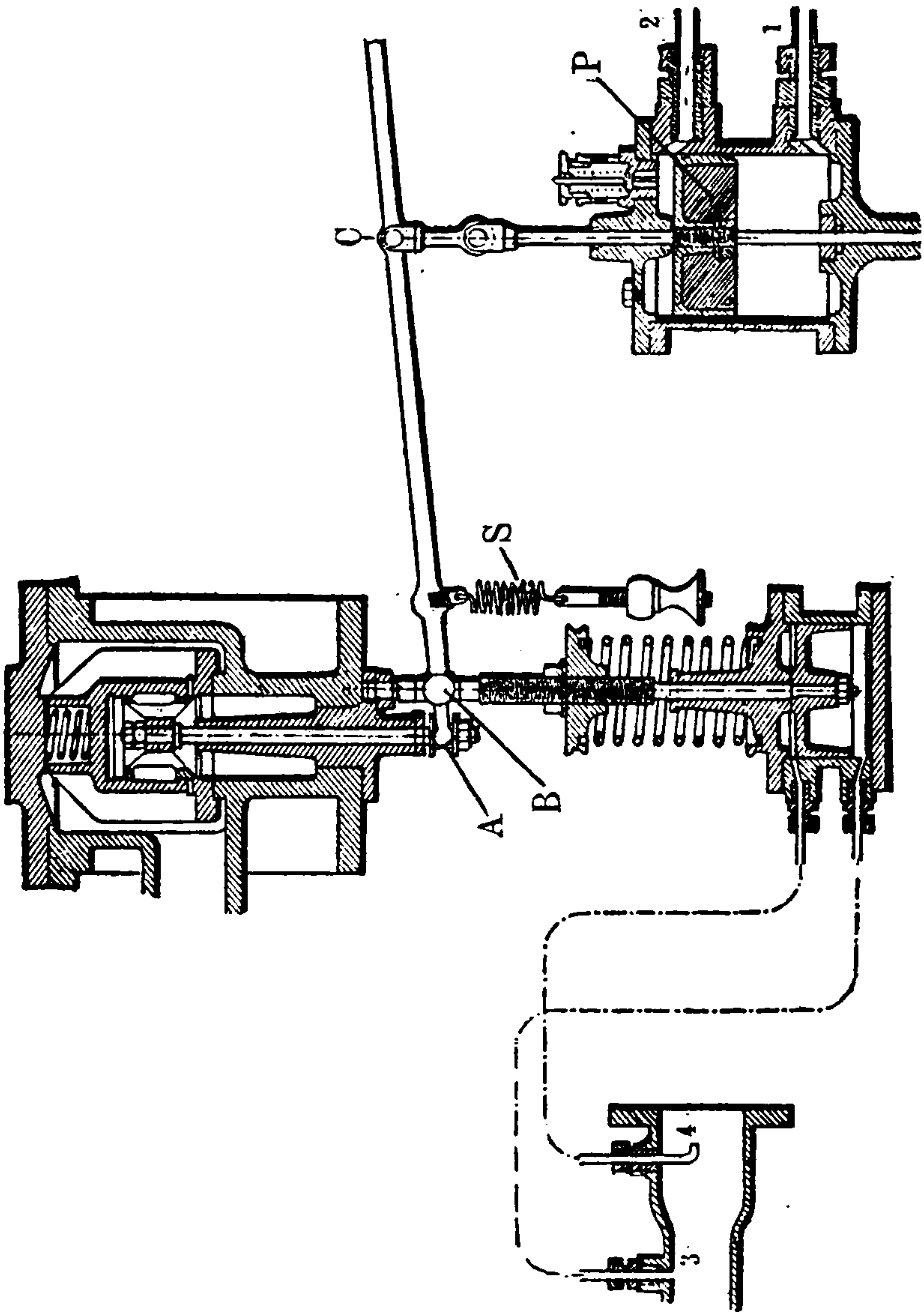


Fig. 106.

discharge was measured by a convergent discharge pipe, two of whose faces were fixed and parallel, while the other two were movable. At its larger end this was fixed to the discharge pipe of the fan; a mercury manometer was used to measure the pressure in this convergent mouthpiece, and the discharge could be calculated from this. The speed was changed from 8,000 to 20,200 rev. per min. The peripheral speed of the fan reached nearly 870 ft. per sec., while the discharge pressure amounted to 16·75 in. of mercury, or 228 of water—more than half an atmosphere. The revolutions were measured by a counter driven by worm gear, which reduced the speed one-thirtieth. One, two, or three of the steam turbine nozzles were opened to give the power required by the fan, and the steam pressure in the steam chest was noted, as by this means the power of the turbine

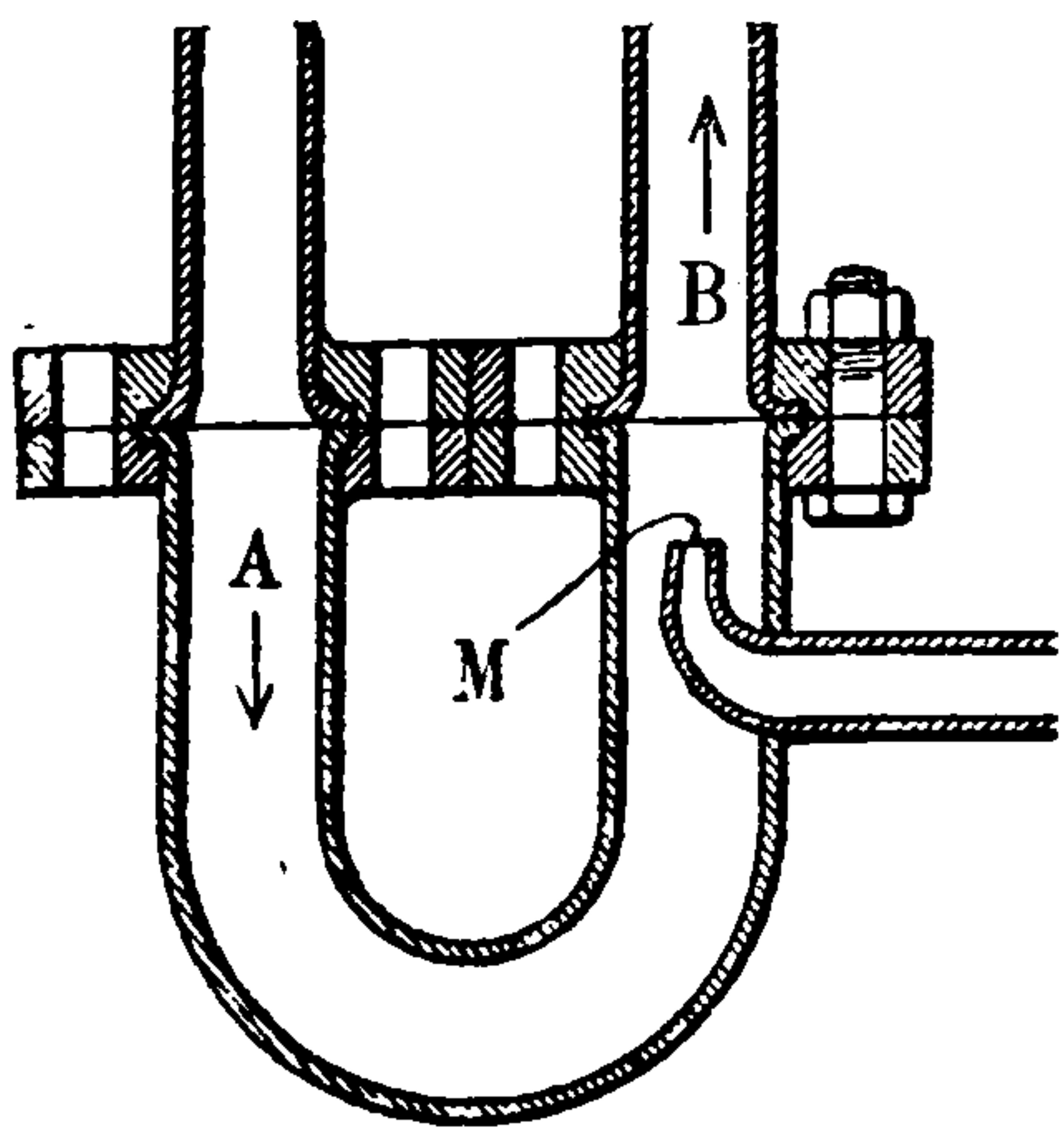


FIG. 107.

could be deduced from previous experiments, so that the total efficiency of engine and fan was thus obtained.

By formulæ[22] obtained by M. Rateau the consumption of steam, and consequently the power theoretically available, could be calculated. The governor was not used during these experiments, so that the pressure in the steam chest was not affected by it. The discharge Q, the useful power in the air T_u, the theoretical power of the steam T_t, and the coefficients η_m, η, and η_v were calculated in the following manner: the discharge Q, in cu. ft. per sec., estimated at atmospheric pressure, is equal to the product of S, the section of the convergent discharge pipe in sq. ft., and the velocity of flow c_d in ft. per sec. Hirn's experiments prove that the coefficient of discharge of a convergent pipe does not differ more than 1 or 2 per cent. from unity. M. Rateau proves in his paper that without serious error

$$\frac{c_d^2}{2g} = \frac{\Delta p}{\sigma},$$

where Δp is the difference of pressure per sq. ft. between suction and discharge, and σ is the mean density of the air. The figures in the seventh column of Table 50 are obtained in this manner. The sixth column gives H in ft. of water, and in the calculation of σ the temperature must be assumed to be 38 deg. Cen. The reduction of volume during compression must be taken into account in calculating the useful power T_u. To obtain the figures in the ninth column the variation of pressure per sq. ft. must be multiplied by the volume in cu. ft. at the mean pressure, so that

$$T_u = \frac{\Delta p \, . \, Q \dfrac{32{\cdot}8}{32{\cdot}8 + 0{\cdot}5\,H}}{550}.$$

The work theoretically obtainable from the steam T_t is calculated as follows: The discharge of steam is obtained from the formula—

$$I = s\,P\,(15{\cdot}20 - 0{\cdot}96 \log P),$$

where s is the total section of the nozzles in sq. centimetres, and P is the pressure in kilogrammes per sq. centimetre, while I is the quantity of steam discharged per sec. in grammes; if this is multiplied by 3·6, it gives kilogrammes per hour. If K is the number of kilogrammes per horse power hour, when the steam chest pressure is P and the exhaust p, in this case the atmospheric pressure, then

$$K = 0{\cdot}85 + \frac{6{\cdot}95 - 0{\cdot}92 \log P}{\log P - \log p}.$$

Then the theoretical horse power obtainable from the steam is

$$T_t = \frac{3{\cdot}6\,I}{K}.$$

The mechanical efficiency

$$\eta = \frac{T_u}{T_t},$$

and is given in the last column of Table 50 on p. 201. As the density of the air varies at different speeds, owing to the great compression, the manometric efficiency is not constant for a given opening of the convergent discharge pipe. Although the fan is only 10 in. dia. it develops

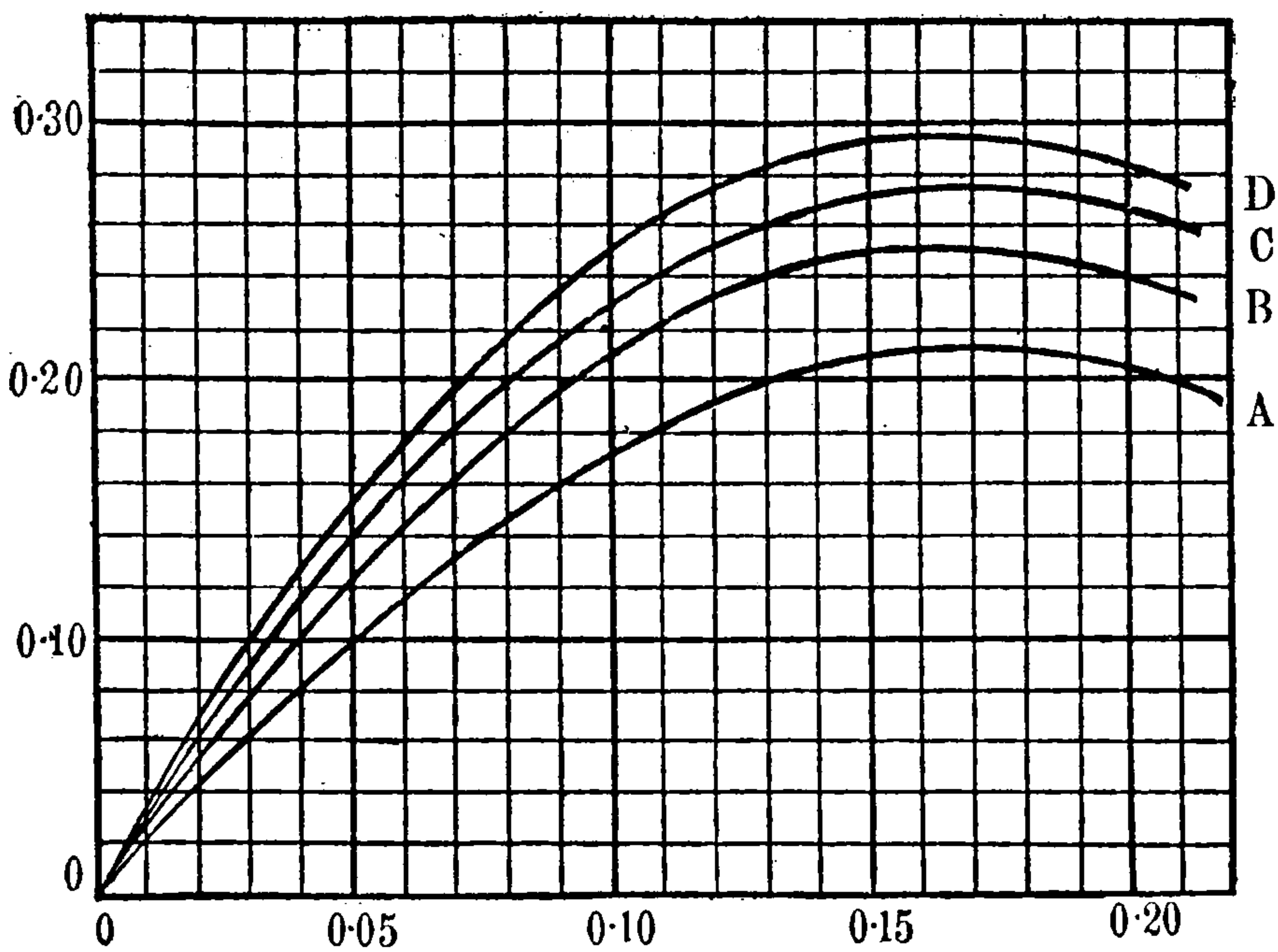


FIG. 108.

a maximum of 45·55 H.P., and a maximum pressure of 19 ft. of water. Fig. 108 gives characteristic curves, a separate characteristic being given for each speed as the efficiency of the turbine increases with the speed. The ordinates are values of η, while the abscissæ are values of η_v. *It must be remembered that the efficiency is the ratio of the useful work done by the fan to the ideal amount obtainable from the steam in the Rankine or Clausius cycle.* At normal speed the efficiency passes 28 per cent., showing that the fan alone has an efficiency of 56 per cent., as that of the turbine is about 50 per cent., according to experiments previously made.

High-pressure fans driven by steam turbines can be used for cupolas, blast furnaces, and Bessemer converters, and wherever a water-gauge of more than 36 in. is required. They can even be employed to compress air to 70 lb. per sq. in. or over. A single wheel can increase the pressure in the ratio of 1·5 to 1, so that two wheels working in series would give a pressure of 2·25, a third 3·4, and a fourth 5 atmospheres. Their mechanical efficiency is slightly inferior to that of ordinary piston compressors, but superior to Roots' blowing machines, whose efficiencies are not more than 35 to 40 per cent. For the supply of air to a blast furnace whose capacity is 160 tons of cast iron per day, and requiring 19,200 cu. ft. of air per min. at atmospheric pressure, and compressed to half an atmosphere, the fan would be 2 ft. 7½ in. dia., would run at 6,000 rev. per min., and the steam turbine would be about the same size. The efficiency of the turbine fan for this high power—500 useful H.P.—would reach 10 per cent., corresponding to a steam consumption of 49·8 lb. of steam per useful H.P. hour if the turbine worked with condensation.

CHAPTER X.

The Theory of Propeller Ventilating Fans.—The propeller is the simplest form of fan; it requires neither diffuser nor volute, although frequently provided with a chimney. Propeller fans are used when a large volume of air is required at a very low pressure, as, for example, in the ventilation of buildings, in which case the volumetric efficiency becomes far more important than the manometric or mechanical efficiency. 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. 109 is shown the section of a blade, A B, θ is the relative angle of inflow, and ϕ that of outflow, if we assume that the angle of flow coincides with the angle of vane. Let b be the axial component of the velocities of inflow and outflow, v the peripheral velocity at a radius r, v_2 that at the extreme radius r_2, a the component of the absolute velocity of the air at outflow at radius r, perpendicular to both radius and axis, or, in other words, the velocity of the whirl. The motion

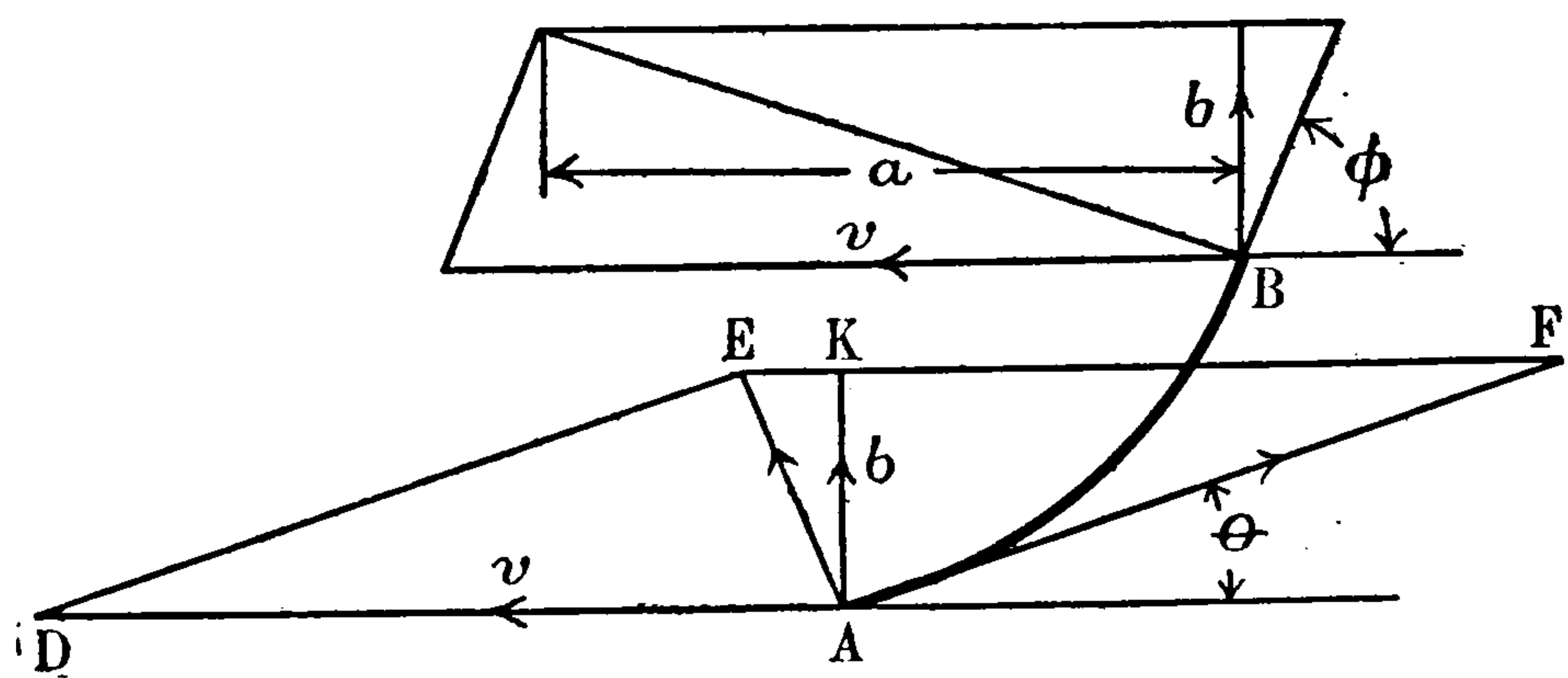


FIG. 109.—VELOCITY DIAGRAMS FOR PROPELLER FAN.

of the blade is to the left. Then from the triangle of velocities at outflow

$$a = v - b \cot \phi$$

also $\dfrac{av}{g}$ = work done by the blade at that radius per pound of air, and if the air enters the wheel without sudden change of direction, it follows, from the triangle of velocities at inflow, that

$$\cot \theta = \frac{v}{b}.$$

is proportional to a^2, and as the motion is a, the total work wasted is proportional to a^3, and may be represented by

$$F_4 \frac{a^3}{2g},$$

which is independent of the quantity passing through the fan; hence the loss of energy per pound, or the loss of head, is represented by

$$\frac{F_3}{2g} \frac{a^3}{b}.$$

because b is proportional to the weight passing through the fan.

Let us now suppose that the axial velocity is the same at any radius; then

$$Q = b \pi r_2^2,$$

and the total loss of head at inflow is

$$L_1 = \frac{\int_{r_1}^{r_2} \frac{(v - b \cot \theta)^2}{2g} 2 \pi r b \sigma . dr}{b \pi (r_2^2 - r_1^2) \sigma};$$

where r_1 is the internal radius. It must be remembered that θ is a function of r, and not necessarily a constant. The loss of head at outflow is

$$L_{2a} = \frac{\int_{r_1}^{r_2} \frac{(v - b \cot \phi)^2}{2g} 2 \pi r b \sigma dr}{b \pi (r_2^2 - r_1^2) \sigma} + \frac{b^2}{2g} \cdot \frac{\int_{r_1}^{r_2} 2 \pi r b \sigma . dr}{b \pi (r_2^2 - r_1^2) \sigma};$$

where ϕ is a function of r, and $v = r \omega$ where ω is the angular velocity in radians. If there is a chimney, the loss of head at outflow becomes

$$L_{2b} = \frac{\int_{r_1}^{r_2} \frac{(v - b \cot \phi)^2}{2g} \frac{r_2^2}{R^2} 2 \pi r b \sigma dr}{b \pi (r_2^2 - r_1^2) \sigma} + \frac{b^2 r_2^4}{2 g R^4} \cdot \frac{\int_{r_1}^{r_2} 2 \pi r \sigma b dr}{\sigma b \pi (r_2^2 - r_1^2)}.$$

The total loss of head by friction in the fan becomes

$$L_3 = \frac{\int_{r_1}^{r_2} \frac{1}{2g} F_1 b^2 \operatorname{cosec}^2 \phi \, 2\pi r b \sigma \, dr}{b\pi (r_2^2 - r_1^2)\sigma},$$

and the total loss of head in the chimney is

$$L_4 = F_2 \frac{b^2}{2g} \frac{\int_{r_1}^{r_2} 2\pi r b \sigma \, dr}{b\pi (r_2^2 - r_1^2)\sigma} + F_3 \frac{\int_{r_1}^{r_2} \frac{(v - b\cot\phi)^3}{2g} . 2\pi r b \sigma \, dr}{b^2 \pi (r_2^2 - r_1^2)\sigma}.$$

Let us first suppose that the blade of the fan is a plane surface, and that $\theta = \phi$ constant. Then L_1 becomes

$$L_1 = \frac{1}{2g}\left[\frac{\omega^2 (r_2^2 + r_1^2)}{2} + b^2 \cot^2 \phi - \tfrac{4}{3} \frac{(r_2^3 - r_1^3)}{r_2^2 - r_1^2} b \omega \cot \phi\right]$$

To simplify this, put $r_1 = 0$.

$$L_1 = \frac{1}{2g}\left[\frac{r_2^2 \omega^2}{2} + b^2 \cot^2 \phi - \tfrac{4}{3} r_2 b \omega \cot \phi\right]$$

$$= \frac{1}{2g}\left[\frac{v_2^2}{2} + b^2 \cot^2 \phi - \tfrac{4}{3} v_2 b \cot \phi\right]$$

The loss at outflow, if no chimney is used, is

$$L_{2a} = \frac{1}{2g}\left(\frac{v_2^2}{2} + b^2 \cot^2 \phi - \tfrac{4}{3} v_2 b \cot \phi\right) + \frac{b^2}{2g};$$

but if a chimney is used, it becomes

$$L_{2b} = \frac{r_2^2}{2g R^2}\left(\frac{v_2^2}{2} + b^2 \cot^2 \phi - \tfrac{4}{3} v_2 b \cot \phi\right) + \frac{b^2 r_2^4}{2g R^4},$$

and the loss of head by friction in the fan becomes

$$L_3 = \frac{F_1}{2g} b^2 \operatorname{cosec}^2 \phi.$$

The work done by the fan per lb. is

$$\frac{\displaystyle\int_{r_1}^{r_2} \frac{v(v-b\cot\phi)}{g} 2\pi b r \sigma\, dr}{b\pi(r_2^2-r_1^2)\sigma} = \frac{\displaystyle\frac{2}{g}\int_{r_1}^{r_2}(r^3\omega^2 - br^2\omega\cot\phi)\,dr}{r_2^2-r_1^2}.$$

Integrating and putting $r_1=0$, the expression becomes

$$\frac{2}{g}\left(\frac{r_2^2\omega^2}{4}-\frac{b\omega r_2}{3}\cot\phi\right)=\frac{\frac{1}{2}v_2^2-\frac{2}{3}v_2 b\cot\phi}{g}.$$

Hence, since the work done by the fan per lb. of air delivered is equal to the head, together with the work absorbed by the losses of head, it follows, in the case of a fan with no chimney, that

$$2gH = v_2^2 - \tfrac{4}{3}v_2 b\cot\phi - (v_2^2 + 2b^2\cot^2\phi - \tfrac{8}{3}v^2 b\cot\phi).$$
$$- b^2 - F_1 b^2\operatorname{cosec}^2\phi - F_2 b^2,$$

neglecting the loss by friction on the casing, due to the whirl speed. Collecting and rewriting, the equation becomes

$$2gH = \tfrac{4}{3}v_2 b\cot\phi - b^2(1 + 2\cot^2\phi + F_1\operatorname{cosec}^2\phi + F_2).$$

Walker's Experiments.—As this type of fan usually discharges against very little pressure, let us first suppose $H=0$, and find suitable values of F_1, F_2, from experiments made by Walker[23] with a propeller fan whose blades were plane, and set at various angles. When the blades were set at 40 deg. to a plane perpendicular to the axis, the volumetric efficiency was 69·7, and when at 25 deg. it was 53·8 per cent. Substituting these values in the above equation—*i.e.*, putting

$$H=0,\ \phi=40,\ \frac{Q}{v_2 r_2^2}=\frac{b\pi r_2^2}{v_2 r_2^2}=\frac{b\pi}{v_2}=0{\cdot}697,$$

$$\text{and}\quad \phi=25,\ \frac{b\pi}{v_2}=0{\cdot}538\text{—we get}$$

$$2{\cdot}42\,F_1+F_2=3{\cdot}31,\ \text{for}\ \phi=40°,$$
$$\text{and}\ 5{\cdot}60\,F_1+F_2=6{\cdot}49,\ \text{for}\ \phi=25°;$$
$$\text{so that}\ F_1=1{\cdot}00\ \text{and}\ F_2=0{\cdot}90,$$

friction caused by the whirling motion of the air becomes of importance, and, as we have neglected it, theory cannot be expected to agree with practice.

The only efficiency that we can consider in this case is the dynamic efficiency, so that

$$H=\left[\frac{\frac{1}{2}v^2-\frac{2}{3}v_2 b\cot\phi}{g}\right]$$

$$\therefore \eta_d=\frac{b^2}{v^2_2\left(1-\frac{4}{3}\frac{b}{v_2}\cot\phi\right)}$$

which can be transformed to

$$\eta_d=\frac{\eta_v^2}{\pi^2\left(1-\frac{4}{3}\frac{\eta_v}{\pi}\cot\phi\right)}$$

where η_v is the volumetric efficiency.

The results obtained from this formula differ entirely from those given by experiment, because the angle of flow does not coincide with the vane angle. The above equation gives us

$$\cot\phi=\frac{1-\frac{\eta_v^2}{\eta_d\pi^2}}{\frac{4}{3}\frac{\eta_v}{\pi}}.$$

The following gives the calculated values of ϕ, and shows that the direction of flow does not follow the curved back of the vanes, since the angle of flow relative to the wheel differs considerably from the vane angle :—

Vane angle ...	15°	30°	45°	50°
ϕ	10° 30′	18° 18′	22° 49′	23° 45′
Experimental efficiency per cent.	33	41·6	26·4	20·9

In the experiments with fan No. 16, with plane blades, the mechanical efficiency of the fan alone is not given, but it can be calculated by deducting 0·0338 horse power from that of the motor, which gives the shaft or brake horse power, and the horse power in the air divided by this gives us the efficiency of the fan alone. Applying the above formula for the mean relative angle of outflow ϕ, we obtain—

Angle of vane ...	15°	20°	25°	27°	30°	35°	40°
Experimental efficiency per cent. of fan alone	30·8	46·1	46·0	42·4	40·2	33·7	29·3
Volumetric efficiency per cent.	28·6	43·0	53·8	58·0	64·7	68·4	69·7
Cot ϕ	8 01	5·26	4·06	3·72	3·26	2·95	2·81
ϕ	7° – 7′	10° – 46′	13° – 50′	15° – 3′	17° – 3′	18° – 44′	19° – 35′

It will be seen from the above that ϕ is approximately half the vane angle, and also that the volumetric efficiency of fan No. 17, with rounded back vanes, is little better than that of fan No. 15, with plane blades, for angles of 30 deg., 35 deg., and 40 deg.

Considering next fans with curved vanes in which θ and ϕ are constant but unequal, we find that in the equation for volumetric efficiency it is best to take θ as the angle of the face, and ϕ as the mean between the angles of the face and back. This is probably due to the fact that the direction of outflow depends on the change in direction which takes place between inflow and outflow; the latter is determined by both the back and front angles of the vanes; further, at inflow a sudden change takes place from an axial to a forward motion, and this is produced by the front of the blade. There are three fans in Walker's paper with curved blades, Nos. 9, 14, and 15. The first and third have curved faces and backs, while the second has a plane face, but a curved back. If a is the mean angle of inclination of a blade,

in the first $\phi = a + 30\frac{1}{4}$, $\theta = a - 20$;

in the second $\phi = a + 20$, $\theta = a$;

in the third $\phi = a + 31\frac{3}{4}$, $\theta = a - 20$.

Now, in this type of fan we have the equation
$$2\,g\,H = v_2^2 - \tfrac{4}{3}\,v_2\,b\,\cot\phi - \left(\frac{v_2^2}{2} + b^2\cot^2\theta - \tfrac{4}{3}\,v_2\,b\cot\theta\right)$$
$$-\left(\frac{v_2^2}{2} + b\cot^2\phi - \tfrac{4}{3}\,v_2\,b\cot\phi\right) - (1+F_2)\,b^2 - F_1 b^2\operatorname{cosec}^2\phi;$$
$$2gH = \tfrac{4}{3}\,v_2\,b\cot\theta - b^2(\cot^2\theta + \cot^2\phi + 1 + F_2 + F_1\operatorname{cosec}^2\phi).$$

When $H=0$, this becomes
$$\tfrac{4}{3}\,v_2\cot\theta = b\,(\cot^2\theta + \cot^2\phi + 1 + F_2 + F_1\operatorname{cosec}^2\phi),$$
and the volumetric efficiency
$$\frac{b\pi}{v_2} = \frac{4\,\pi\cot\theta}{3\,(\cot^2\theta + \cot^2\phi + 1 + F_2 + F_1\operatorname{cosec}^2\phi)},$$
but the above will not give results agreeing with practice for a constant value of F_1 because, the vane being curved, the relative velocity of the air over its surface is variable. Hence, for the last term in the denominator we substitute $\tfrac{1}{2}\,F_1\,(\operatorname{cosec}^2\phi + \operatorname{cosec}^2\theta)$, and obtain very good results. These three fans (9, 14, 16) were tested with mean angles of 17 deg., 27 deg., and 40 deg., and we think that the results obtained with the two first angles are of very little use except to show that the last is better, both for mechanical and volumetric efficiencies, because the air at inflow was struck by the back of the blade; in fact, with a decrease in the angle of outflow, the angle of inflow is decreased too rapidly by turning round the blade. Calculating the volumetric efficiency from the equation
$$\frac{b\,\pi}{v_2} = \frac{4\,\pi\cot\theta}{3\,\{\cot^2\theta + \cot^2\phi + 1 + F_2 + \tfrac{1}{2}F_1(\operatorname{cosec}^2\theta + \operatorname{cosec}^2\phi)\}},$$
and writing $F_1 = 1{\cdot}00$, $F_2 = 0{\cdot}90$, as in the first case,
$$\frac{b\,\pi}{v_2} = \frac{4\,\pi\cot\theta}{3\,\{(\operatorname{cosec}^2\theta + \operatorname{cosec}^2\phi)\times 1{\cdot}5 - 0{\cdot}1\}}$$
$$= \frac{2{\cdot}79\cot\theta}{\operatorname{cosec}^2\theta + \operatorname{cosec}^2\phi - 0{\cdot}067},$$

η for $\phi-30°$, per cent.	—	7·6	12·0	14·8	14·0
$\dfrac{v_2}{\sqrt{g\,H}}$ for $\phi=15°$	4·91	3·8	4·025	5·28	—
$\dfrac{v_2}{\sqrt{g\,H}}$ for $\phi=30°$	—	5 36	4·44	4·30	4·54

which shows that without a chimney these fans cannot be used, efficiently, to produce pressure. It will also be noticed that the efficiency decreases as ϕ increases.

We may next consider the case of the above types of propeller with a chimney whose outlet has a diameter three times that of the fan. The general equation of the fan is then

$$2\,g\,H=r_2^2-\tfrac{4}{3}\,v_2\,b\cot\phi-\left(\frac{v_2^2}{2}+b\cot^2\theta-\tfrac{4}{3}\,v_2\,b\cot\theta\right)$$
$$-\frac{r_2^2}{R^2}\left(\frac{v_2^2}{2}+b^2\cot_2\phi-\tfrac{4}{3}\,v_2\,b\cot\phi\right)-\frac{b^2\,r_2^4}{R_4}$$
$$-b^2\,F_2-\tfrac{1}{2}\,F_1\,b^2\,(\operatorname{cosec}^2\phi+\operatorname{cosec}^2\theta),$$

which merely states that the head produced is equal to the work per pound done by the wheel, less the losses at inflow to the fan, outflow from the chimney, and those due to friction. This becomes, when simplified,

$$2\,g\,H=\frac{v_2^2}{2}\left(1-\frac{r_2^2}{R^2}\right)+v_2 b\left[\tfrac{4}{3}\cot\theta-\tfrac{4}{3}\cot\phi\left(1-\frac{r_2^2}{R^2}\right)\right]$$
$$-b^2\left[\cot^2\theta+\frac{r_2^2}{R^2}\cot^2\phi+\frac{r_2^4}{R^4}+F_2+\frac{F_1}{2}(\operatorname{cosec}^2\phi+\operatorname{cosec}^2\theta)\right]$$

and when $r\div R=\frac{1}{3}$ this gives us

$$2\,g\,H=\tfrac{4}{9}\,v_2^2+\tfrac{4}{3}\,r_2\,b\,[\cot\theta-\tfrac{8}{9}\cot\phi]$$
$$-b^2\left[\cot^2\theta+\tfrac{1}{9}\cot^2\phi+\tfrac{1}{81}+F_2+\frac{F_1}{2}(\operatorname{cosec}^2\theta+\operatorname{cosec}^2\phi)\right]$$

First consider the case of plane blades when $H=0$; we then get, putting $F_1=0\ 90$ and $F_2=1{\cdot}00$,

$$\tfrac{4}{9}\, v_2{}^2+\tfrac{4}{27}\, v_2\, b \cot \phi$$
$$-b^2\left[\tfrac{10}{9} \cot^2 \phi+0{\cdot}912+\operatorname{cosec}^2 \phi\right]=0$$
$$\tfrac{4}{9}\, v_2{}^2+0{\cdot}148\, v_2\, b \cot \phi-b^2\left[2{\cdot}11 \cot^2 \phi+1{\cdot}912=0\right.;$$

so that the formula for the volumetric efficiency is

$$\frac{b\, \pi}{v_2}=\frac{6\, \pi}{13{\cdot}11 \sqrt{\cot^2 \phi+0{\cdot}90}-\cot \phi}.$$

which gives

	$\phi=30$ deg.	45 deg.	60 deg.	90 deg.
η_v per cent. =	78	110	135	151

The dynamic efficiency $\eta_d=\dfrac{\eta_v{}^2}{\pi^2\left(1-\frac{4}{3}\frac{\eta_v}{\pi} \cot \phi\right)}$

and we shall here suppose ϕ to be the vane angle.

	$\phi=30$ deg.	45 deg.	60 deg.	90 deg.
η_d per cent. =	14·5	23·0	27·1	23·1

We should probably get in practice a very much higher efficiency than this, as ϕ would actually be about half the above values; in the third case, for a vane angle of 60 deg., supposing the angle of outflow is actually 45 deg., we should have an efficiency of 43·3 per cent.

When H is not zero, we get

$$2\, g\, H=\tfrac{4}{9}\, v_2{}^2+0{\cdot}148\, v_2\, b \cot \phi-b^2\left[2{\cdot}11 \cot^2 \phi+1{\cdot}912\right],$$

from which

$$\frac{v^2}{\sqrt{g\, H}}=\frac{b}{\sqrt{g\, H}}\left[\sqrt{4{\cdot}78 \cot^2 \phi+4{\cdot}30+\frac{4{\cdot}5\, g\, H}{b^2}}-\frac{\cot \phi}{6}\right].$$

This gives the peripheral speed for various values of $b \div \sqrt{g H}$, and the corresponding values of air efficiency can be found from the equation

$$\eta=\frac{2\, g\, H}{v_2{}^2\left(1-\frac{4}{3}\, \frac{b}{v_2} \cot \phi\right)}.$$

The velocities and efficiencies are given in Table 51.

TABLE 51.

$\frac{b}{\sqrt{g\,H}}$	$\frac{v_2}{\sqrt{g\,H}}$ for $\phi=15°$.	$\phi=30°$.	$\phi=45°$.	$\phi=60°$.
0·1	2·22	2·14	2·13	2·13
0·2	2·58	2·23	2·17	2·15
0·3	3·11	2·40	2·26	2·20
0·4	3·73	2·62	2·37	2·29
0·5	4·40	2·88	2·52	2·40

$\frac{b}{\sqrt{g\,H}}$	η for $\phi=15°$.	$\phi=30°$.	$\phi=45°$.	$\phi=60°$.
0·1	0·523	0·490	0·470	0·457
0·2	0·490	0·508	0·485	0·467
0·3	0·398	0·488	0·476	0·462
0·4	0·308	0·451	0·459	0·441
0·5	0·239	0·403	0·429	0·415

which shows that the best angle of inclination ϕ of the vanes, if we consider volumetric efficiency as well as mechanical, is about 45 deg., but that this type of fan is very unsuitable as a means of producing pressure, because of its low manometric and mechanical efficiencies.

Next, consider the case of vanes in which ϕ is constant, but θ varies, so as to do away with the loss of shock at inflow; in this case the general equation becomes

$$2\,g\,H = v_2^2 - \tfrac{4}{3}\,r_2\,b\cot\phi - \frac{r_2^2}{R^2}\left(\frac{v_2^2}{2} + b^2\cot^2\phi - \tfrac{4}{3}\,v_2\,b\cot\phi\right)$$
$$- \frac{b^2 r_2^4}{R^4} - b^2\,F_2 - \tfrac{1}{2}\,F_1\,b^2\,(\operatorname{cosec}^2\phi + \operatorname{cosec}^2\theta),$$

TABLE 53.

$\frac{b}{\sqrt{g\mathrm{H}}}$	η for				
	$\phi=30°$.	$\phi=45°$.	$\phi=60°$.	$\phi=90°$.	$\phi=135°$.
0·1	0·685	0·700	0·708	0·717	0·722
0·2	0·630	0·660	0·675	0·701	0·712
0·3	0·557	0·612	0·639	0·669	0·678
0·5	0·415	0·500	0·544	0·585	0·589

Now, the volumetric efficiency is $b\pi \div v_2$ for these fans, and in a Rateau fan we have

$$\text{volumetric efficiency} = \frac{\mathrm{Q}}{v_2 r_2^2} = \frac{2\pi r_2 s_2 b_2}{v_2 r_2^2} = \frac{b_2}{v_2},$$

since $\qquad 2\pi r_2 s_2 = r_2^2,$ for fans of type A.

Hence, even if we assume for the Rateau fan a manometric efficiency of 90 per cent., the volumetric efficiencies for $b=0{\cdot}5\sqrt{g\mathrm{H}}$ of the two fans are 102 and 47·6 per cent., so that the propeller fan is the superior of the two in this respect, but, of course, much inferior in mechanical and manometric efficiencies. The above assumes $\phi=135$ deg. in both cases, and even if $\phi=90$ for the propeller, the volumetric efficiency is 85·2 per cent. No propeller fan with $\phi=135$ has yet been constructed as far as we know, and it is quite possible that such a fan might discharge the air in the wrong direction.

CHAPTER XI.

Helical Propellers.—Next consider the case of helical blades, in which, however, the pitch at inflow is not the same as at outflow, but that θ at every radius is so arranged that inflow takes place without shock, and therefore L_1 is zero. Let P be the pitch at outflow, then

$$\cot \phi = \frac{2 \pi r}{P};$$

so that $\dfrac{2 \pi r}{\cot \phi} = \dfrac{2 \pi r_2}{\cot \phi_2}$, and $\therefore \cot \phi = \dfrac{r}{r_2} \cot \phi_2 = A r.$

For a fan with a chimney the loss at outflow is,

$$L_{2b} = \frac{\displaystyle\int_{r_1}^{r_2} \frac{(v - b \cot \phi)^2}{2 g} \frac{r_2^2}{R^2} 2 \pi r b \sigma d r}{b \pi (r_2^2 - r_1^2) \sigma} + \frac{b^2 r_2^4}{2 g R^4}$$

$$\frac{\displaystyle\int_{r_1}^{r_2} 2 \pi r b \sigma d r}{b \pi (r_2^2 - r_1^2) \sigma}$$

$$= \frac{\displaystyle\int_{r_1}^{r_2} \frac{(\omega r - b A r)^2}{g} \frac{r_2^2}{R^2} r d r}{(r_2^2 - r_1^2)} + \frac{b^2 r_2^4}{2 g R^4}$$

$$= \frac{(\omega - A b)^2 r_2^2}{g R^2} \cdot \frac{\displaystyle\int_{r_1}^{r_2} r^3 d r}{(r_2^2 - r_1^2)} + \frac{b^2 r_2^4}{2 g R^4}$$

$$= \frac{(\omega - A b)^2}{4 g} \frac{r_2^2}{R^2} (r_1^2 + r_2^2) + \frac{b^2 r_2^4}{2 g R^4}.$$

The loss by friction between the air and the surface of the vanes is,

$$L_3 = \frac{\int_{r_1}^{r_2} \frac{1}{2g} F_1 b^2 \tfrac{1}{2} (\operatorname{cosec}^2 \phi + \operatorname{cosec}^2 \theta)\, 2\pi r b \sigma\, dr}{b \pi (r_2^2 - r_1^2) \sigma}$$

$$= \frac{F_1 b^2}{2g} \frac{\int_{r_1}^{r_2} (\operatorname{cosec}^2 \phi + \operatorname{cosec}^2 \theta)\, r\, dr}{(r_2^2 - r_1^2)}$$

$$= \frac{F_1 b^2}{2g} \frac{\int_{r_1}^{r_2} \left(2 + A^2 r^2 + \frac{v^2}{b^2}\right) r\, dr}{(r_2^2 - r_1^2)}$$

$$= \frac{F_1 b^2}{2g} \frac{\int_{r_1}^{r_2} \left(2r + A^2 r^3 + \frac{r^3 \omega^2}{b^2}\right) dr}{(r_2^2 - r_1^2)}$$

$$= \frac{F_1 b^2}{2g} \frac{\left\{ (r_2^2 - r_1^2) + A^2 \frac{(r_2^4 - r_1^4}{4} - \frac{\omega^2}{b^2} \frac{r_2^4 - r_1^4}{4} \right\}}{(r_2^2 - r_1^2)}$$

$$= \frac{F_1 b^2}{2g} \left\{ 1 + \left(\frac{A^2}{4} + \frac{\omega^2}{4 b^2}\right) \left(r_1^2 + r_2^2\right) \right\}.$$

The loss of head by friction in the chimney is,

$$L_4 = F_2 \frac{b^2}{2g} \frac{\int_{r_1}^{r_2} 2\pi r b \sigma\, dr}{b \pi (r_2^2 - r_1^2) \sigma} + F_3 \frac{\int_{r_1}^{r_2} \frac{(\omega r - b A r)^3}{2g} 2\pi b r \sigma\, dr}{b^2 \pi (r_2^2 - r_1^2) \sigma}$$

$$= F_2 \frac{b^2}{2g} + F_3 \frac{(\omega - A b)^3}{g b} \frac{\int_{r_1}^{r_2} r^4\, dr}{(r_2^2 - r_1^2)}$$

$$= F_2 \frac{b^2}{2g} + F_3 \frac{(\omega - A b)^3}{5 g b} \frac{(r_2^5 - r_1^5)}{(r_2^2 - r_1^2)}.$$

Putting $F_1 = 1{\cdot}00$ and $F_2 = 0{\cdot}90$, $\frac{r_2}{R} = \frac{1}{3}$, the equation becomes

$$2\,g\,H = v_2^2\left[(1+m^2)\ (0\ 6944)\right] - v_2\,b\cot\phi_2\left[0{\cdot}889\ (1+m^2)\right] - b^2\left[\cot^2\phi_2\ (1+m^2)\ 0{\cdot}3056 + 1{\cdot}91\right],$$

and if $m = \frac{1}{3}$

$$2\,g\,H = 0{\cdot}772\ v_2^2 - 0\ 988\cot\phi_2\ v_2\,b - (0{\cdot}34\cot^2\phi_2 + 1{\cdot}91)\,b^2;$$

so that

$$v_2^2 - 1{\cdot}28\cot\phi_2\ v_2\,b - (0{\cdot}44\cot^2\phi + 2{\cdot}475)\,b^2 - 2{\cdot}59\,g\,H = O.$$

Further, the air efficiency

$$\eta = \frac{H}{\frac{(\omega^2 - A\,b\,\omega)}{2\,g}\,(r_1 + r_2^2)} = \frac{1{\cdot}8\,g\,H}{v_2^2 - v_2\,b\cot\phi_2}.$$

This we can now determine with an assumed value of ϕ_2 for various values of $b \div \sqrt{g\,H}$; since for $\phi_2 = 45°$

$$\frac{v_2}{\sqrt{g\,H}} = \frac{b}{\sqrt{g\,H}}\left[0{\cdot}64 + \sqrt{3{\cdot}325 + \frac{2{\cdot}59\,g\,H}{b^2}}\right],$$

and for $\phi_2 = 90°$

$$\frac{v_2}{\sqrt{g\,H}} = \frac{b}{\sqrt{g\,H}}\left[\sqrt{2\ 475 + \frac{2{\cdot}59\,g\,H}{b^2}}\right],$$

from which the values of η are obtained as follows:

$\frac{b}{\sqrt{g\,H}}$.	$\phi_2 = 45°$.		$\phi_2 = 90°$.	
	$\frac{v_2}{\sqrt{g\,H}}$.	η.	$\frac{v_2}{\sqrt{g\,H}}$.	η.
0·2	1·78	0·640	1·64	0·670
0·5	2·17	0·497	1·79	0·562

The above shows that the efficiency decreases as the discharge increases, and increases with the angle ϕ_2. It is probably even greater for $\phi_2 = 135$, but as the air might be discharged in the wrong direction with this arrangement, we have not considered it.

The volumetric efficiency

$$= \frac{Q}{v_2 r_2^2} = \frac{\pi b}{v_2} \frac{(r_2 - r_1^2)}{r_2^2}.$$

Supposing $H = 0$, then this becomes

$$= \frac{\frac{8}{9}\pi}{0{\cdot}64 \cot \phi_2 + \sqrt{0{\cdot}85 \cot^2 \phi_2 + 2{\cdot}475}},$$

which obviously increases with ϕ_2, and is 113 per cent. for $\phi_2 = 45$ deg., and 177·5 per cent. for $\phi_2 = 90$ deg. The dynamic efficiency

$$\eta_d = \frac{\frac{b^2}{2g}}{\frac{(v_2^2 - v_2 b \cot \phi_2)}{2g}(1 + m^2)}.$$

$$= \frac{\eta_v^2}{(1+m^2)(1-m^2)^2 \pi^2 \left[1 - \frac{b}{v_2} \cot \phi_2\right]}$$

$$= \frac{1{\cdot}139\, \eta_v^2}{\pi^2 \left[1 - \frac{9}{8}\frac{\eta_v}{\pi} \cot \phi_2\right]};$$

so that for $\phi_2 = 45°$, $\eta_d = 24{\cdot}8$,

and for $\phi_2 = 90°$, $\eta_d = 36{\cdot}4$.

Hence we may conclude that, if an efficient chimney is used, ϕ_2 should be 90 deg., and the inflow edge of the blade should be helical, and of such form that if θ_2 is the value of θ when $r = r_2$,

$$\cot \theta_2 = \frac{v_2}{b},$$

so that inflow may take place without shock, and a fair mechanical and a high volumetric efficiency can be obtained.

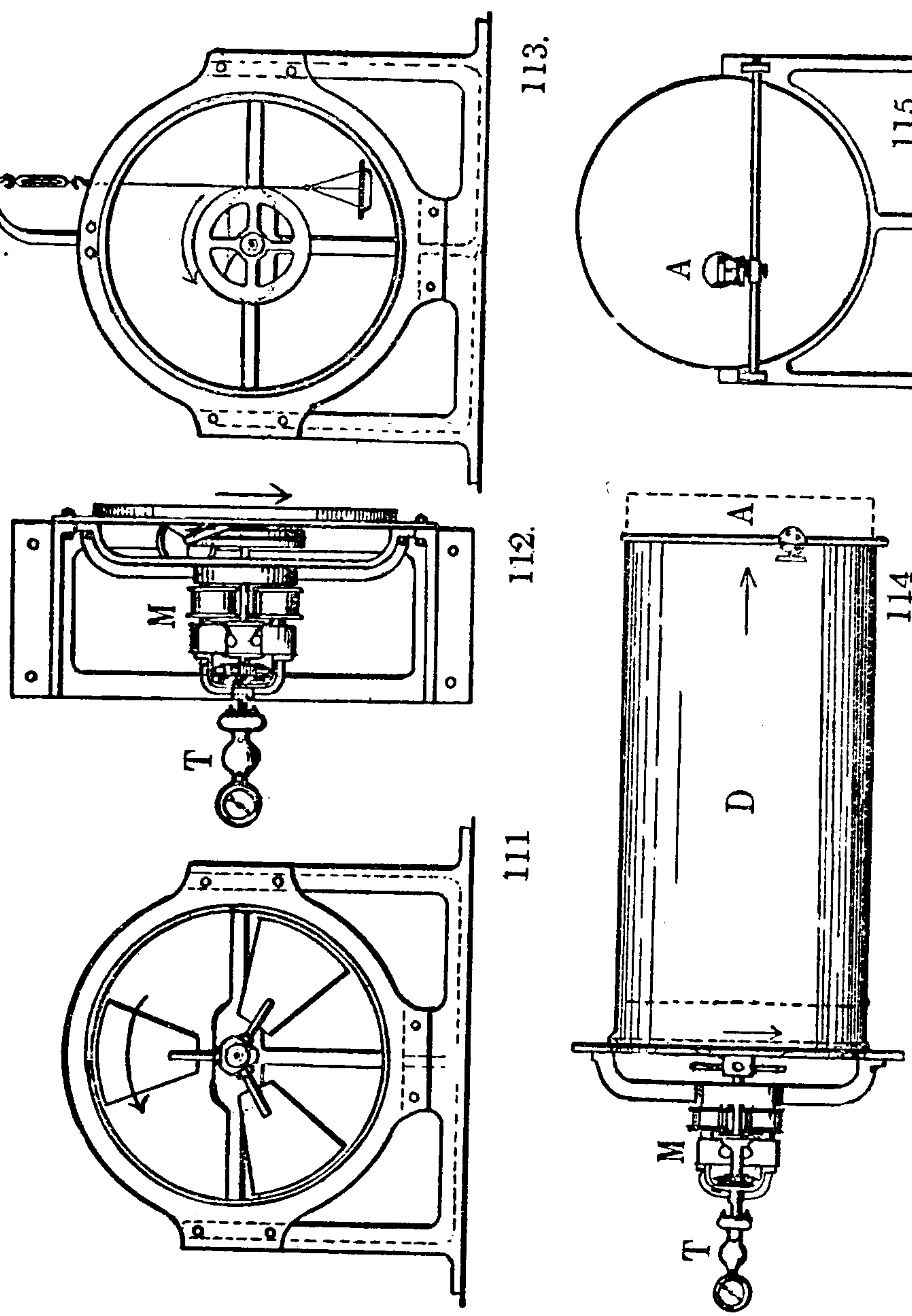
Experiments with Propeller Ventilating Fans.—These experiments (24) were made by Mr. Walker during 1895-6 at Westminster. The primary purpose was to ascertain :

(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.

The experiments showed that the ordinary laws hold good, and that the propeller fan is adapted to the discharge of large volumes of air at small pressures, and further that volumetric efficiency is more important than even mechanical efficiency. All the experiments were made with fans having a free discharge (except a few at the end), the outlet being the same as the inlet. The volumetric efficiency of the propeller fan is greatest with free discharge, and falls off rapidly if the discharge pipe is baffled. Seventeen three-bladed fans were tested, all $23\frac{3}{4}$ in. dia. They are shown in fig. 110. The fans were driven direct from the spindle of an electro-motor fixed centrally to a cast-iron frame in the rear of the fans (figs. 111, 112). The motor was a continuous-current series-wound machine of about one-third of an electrical horse power. The air was delivered through a tube 24 in. bore and 4 ft. long (figs. 114, 115), 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, attached by a Hooke's joint to the motor spindle, and was read to two rev. per min. In most of the experiments the fans were run at 600 rev. per min. ; the speed being kept constant by the adjustment of a suitable form of resistance. The velocity of the air was measured by an anemometer of $2\frac{3}{4}$ in. dia., placed at the outer end of the delivery tube; the instrument was calibrated at Kew Observatory for speeds from 500 to 2,000 ft. per min. The velocity varied greatly in different positions of the same cross-section of the tube. A smooth brass rod $\frac{5}{16}$ in. dia. was placed



FIGS. 111 AND 112.—ARRANGEMENT OF ELECTRIC DRIVE. FIG. 113.—DYNAMOMETRIC BRAKE. FIGS. 114 AND 115.—DELIVERY AIR TUBE.

A, Anemometer; *D*, delivery air tube; *M*, motor; *T*, tachometer.

horizontally across the end of the tube to which the anemometer was attached, so that the centre of the latter 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 B.H.P. of the motor was obtained by a dynamometric brake, fig. 113. 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, and ran without oscillation. The pulley was of cast iron, 9·4 in. dia., with smooth circumference; round the pulley was wound a fine silk cord, the upper end of which was attached to a Salter's balance, while the lower end, supporting a scale pan, hung vertically beneath. If W lb. = load in the scale pan and w = reading of the balance, then the B.H.P. absorbed by the brake = $\frac{(W - w)\, 2\, \pi\, R\, N}{33{,}000}$ where R is the radius of the pulley in feet, and N is the number of rev. per min.

For each experiment anemometer readings were taken at each of the four following radii of the delivery tube: $1\frac{7}{8}$, $5\frac{1}{8}$, $7\frac{7}{8}$, $10\frac{5}{8}$ in. 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 these rings; each of the three outer rings was equal in breadth to the diameter of the anemometer. The velocity of the air in ft. per min. as ascertained at each of the four radii was multiplied by the area of the corresponding rings in sq. ft., and the products being added together gave the number of cu. ft. of air discharged per minute. The velocities given at each of the four radii are given in Tables 54 and 55 for all the fans tried. The areas of the four imaginary rings were 1·275, 0·945, 0·614, 0·307 sq. ft. The mean velocity of the air was obtained by dividing the air discharge in cu. ft. by 3·141 sq. ft. Readings of the anemometer were taken for two minutes at each of the four radii for each experiment, together with volts, ampères, height of barometer, and temperature of air. A series of experiments were made with the motor running at 600 rev. per min., and the experimental readings are

TABLE 54.—

No. of fan.	Angle of blade in degs.	Revs. per min.	Volts of motor.	Ampères of motor.	Velocity of air per min. at radius.			
					$1\frac{7}{8}$ in.	$5\frac{1}{8}$ in.	$7\frac{7}{8}$ in.	$10\frac{5}{8}$ in.
1	17	890	69·7	1·50	762	767	688	607
2	17	860	81·5	1·92	880	917	920	930
3	17	645	60·0	1·58	690	730	690	675
4	17	525	54·5	1·58	617	713	730	645
5	27	610	80·0	2·33	723	804	913	951
6	27	490	78·3	2·53	423	690	880	903
7	17	758	80·8	2·07	703	785	805	760
8	17	650	78	2·28	645	780	805	700
	27	*600	85	2·45	910	1,115	1,145	1,050
	27	595	77	2·20	670	780	875	990
	40	*495	100	2·70	735	970	1,147	1,070
9	17	600	69	1·91	320	365	530	635
	27	600	79	2·23	600	650	755	825
	27	*600	86	2·47	740	890	1,030	960
	40	570	88	2·75	745	810	910	960
10	27	*600	81	2·31	805	930	950	930
	17	*600	68	1·85	260	405	555	645
11	17	850	74	1·70	640	745	850	990
	27	610	63	1·66	645	725	860	930
	27	*600	64	1·75	765	895	870	850
12	27	600	63	1·70	590	635	740	745
13	27	600	74	2·06	655	745	860	995
	17	600	61	1·63	430	525	640	680
14	17	604	53	1·40	415	540	605	640
	40	605	85	2·40	780	940	1,110	1,105
15	40	600	84	2·40	775	965	1,120	1,085
16	35	600	65	1·77	720	780	860	830
17	35	600	67	1·87	685	830	920	875

Experiments on Propeller Ventilating Fans, Figs. 110 and 115, of those marked * revolving $4\frac{1}{2}$ in. out of the tube. Blades set at an $5\frac{1}{8}$ in., $7\frac{7}{8}$ in., $10\frac{5}{8}$ in. from the axis of the tube.

TABLE 55.—W. G. WALKER'S EXPERIMENTS.

Angle of blades.	Motor.		Velocity of air in ft. per min. at radius.				Cu. ft. of air per min.	Horse power.		Efficiencies.	
	Volts.	Amps.	$1\frac{7}{8}$.	$5\frac{1}{8}$.	$7\frac{7}{8}$.	$10\frac{5}{8}$.		Brake.	Air.	Mechanical.	Volumetric.
Deg.											
15	40	05	460	470	510	510	1,562	0·0115	0·0038	33·0	41·4
20	45	1·22	570	590	610	630	1,916	0·0181	0·0070	38·7	50·8
25	52	1·36	630	630	730	770	2,291	0·0282	0·0120	42·6	60·8
27	55	1·43	640	710	790	820	2,423	0·0334	0·0143	42·8	64·3
30	65	1·66	665	770	875	860	2,598	0·0423	0·0176	41·6	68·9
35	67	1·87	685	830	920	875	2,705	0·0538	0·0199	37·0	71·7
40	76	2·12	705	855	970	900	2,805	0·0707	0·0222	31·4	74·4
45	84	2·40	780	910	960	880	2,827	0·0858	0·0227	26·4	76·7
50	91	2·59	790	880	930	870	2,771	0·1024	0·0214	20·9	75·2
60	86	2·48	210	290	450	680	1,534	0·0911	0·0035	3·8	40·7

Experiments on a Propeller Ventilating Fan (fig. 110), No. 17, with blades set at different angles to the plane of revolution. Fan diameter, $23\frac{3}{4}$ in.; revolving in tube, 24 in. dia.; speed, 600 rev. per min.; anemometer placed at radii of $1\frac{7}{8}$, $5\frac{1}{8}$, $7\frac{7}{8}$, $10\frac{5}{8}$ in.

TABLE 56.—W. G. WALKER'S EXPERIMENTS.

	Fan 16.						Fan 17.					
		Horse power.		Efficiency per cent.				Horse power.		Efficiency per cent.		
Angle of blades.	Air discharged in cu. ft. per min.	of motor = B.H.P. + H.P. required to overcome friction.	Useful horse power in the air.	Mechanical.	Volumetric.	Pressure.	Air discharged in cu. ft. per min	of motor = B.H.P. + H.P. required to overcome friction.	Useful horse power in air.	Mechanical.	Volumetric.	Pressure.
Deg.												
15	1,079	0·0377	0·0012	3·1	28·6	0·41	1,562	0·0453	0·0038	8·3	41·4	0·87
20	1,622	0·0429	0·0042	9·7	43·0	0·94	1,916	0·0519	0·0070	13·4	50·8	1·31
25	2,030	0·0521	0·0084	16·1	53·8	1·48	2,291	0·0620	0·0120	19·3	60·8	1·88
27	2,185	0·0588	0·0106	18·0	58·0	1·73	2,423	0·0672	0·0143	21·2	64·3	2·10
30	2,439	0·0698	0·0145	20·7	64·7	2·13	2,598	00 761	0·0176	23·1	68·9	2·42
35	2,580	0·0848	0·0172	20·2	68·4	2·39	2,705	00 876	0·0199	22·7	71·7	2·62
40	2,627	0·0958	0·0182	18·9	69·7	2·47	2,805	01 045	0·0222	21·2	74·4	2·82

Comparison of Propeller Ventilating Fans Nos. 16 and 17 (fig. 110) with blades set at different angles to the plane of revolution. Both fans are 23¾ in. dia., revolving inside a delivery air tube of 24-in. bore; speed 600 rev. per min. Friction of motor included.

shown in fig. 116. It appears from this diagram that the current varies directly as the difference of tensions in the cord, hence the relation between the current and torque can be expressed by a linear equation, thus—

$$T = aC - b,$$

where T = torque, C = current, and a and b are constants.

Hence the B.H.P. of the motor at 600 rev. per min.

$$= \frac{T\omega}{33{,}000} = \frac{(aC - b)\,\omega}{33{,}000},$$

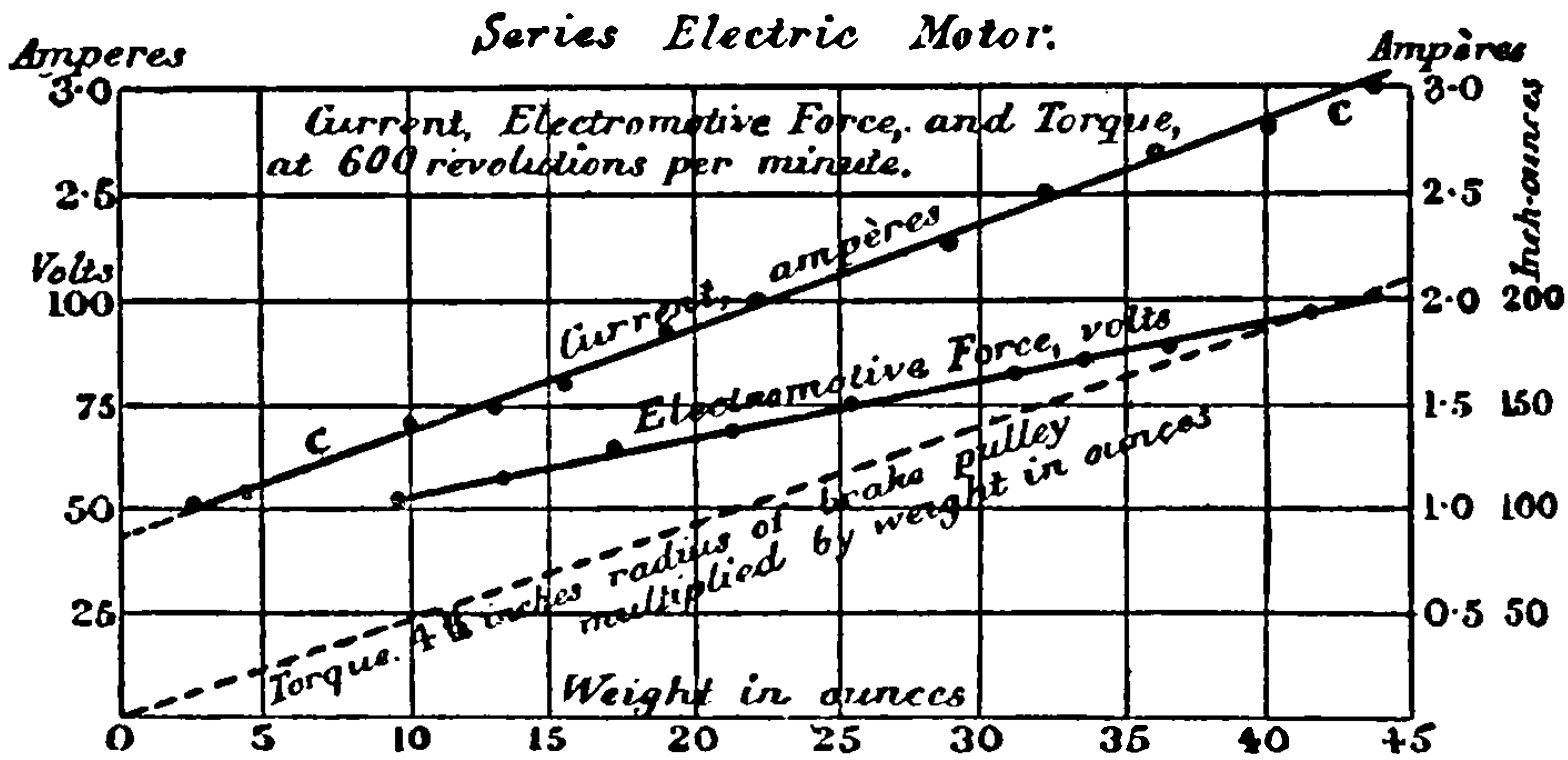


FIG. 116.—EXPERIMENTAL READINGS AT 600 REV. PER MIN.

(See Table 54, p. 228; Series Electric Motor.)

C, Current in ampères; *E*, electromotive force; *T*, torque; radius of pulley in inches × weight in ounces.

where T is in lb. feet, and ω is in radians per min. With a given torque the ampères were not quite constant for all speeds of the motor; they increased slightly and uniformly with the increase in the number of rev. per min. 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 of the current due to speed. Although not essential for the present experiments, it was interesting to determine at what speeds the motor should

into account the moisture in the air, the weight of 1 cu. ft. of air

$$=\frac{1\cdot3304}{F+461}\left(B-\tfrac{3}{8}\,b\right) \text{ approximately,}$$

where b is the pressure due to the moisture in in. of mercury; see section 6.

If W = weight of air discharged in lb. per sec., and V = velocity of air in ft. per sec., then the kinetic energy of the air discharged is $\frac{W\,V^2}{2\,g}$, and the horse power of air discharged is

$$\frac{W\,V^2}{2\,g\times550}=V^3\times\text{constant,}$$

for the same fan under same conditions. Hence if Q = cu. ft of air per sec., the horse power of the air discharged

$$=\frac{V^2\,Q}{550\times64\cdot4}\times\frac{1\cdot3304\ B}{F+461}=\frac{V^2\,Q\,B}{F+461}\times0\cdot00003756.$$

The mechanical efficiency $=\dfrac{\text{horse power in air discharged}}{\text{B.H.P.}}$, volumetric efficiency is as usual $\dfrac{Q}{v_2\,r_2{}^2}$, and dynamic pressure efficiencies are evidently $\dfrac{V^2}{2\,v_2{}^2}$, the static pressure efficiency being zero.

Experiments were made with fan blades at different angles to the plane of rotation. The results with fan 17, having plane surfaces and rounded backs to the blades, are given in Table 55, and plotted in fig. 118. These may be termed the characteristic curves of the fan for varying angles. It should be noted that maximum volumetric efficiency is not obtained with the same angle as maximum mechanical efficiency. In Table 56 fans 16 and 17 have been compared, and the latter certainly has the better volumetric efficiency by a very small amount, and the mechanical efficiency of fan and motor, which, of course, includes motor friction, is better for the latter; but when we deduct

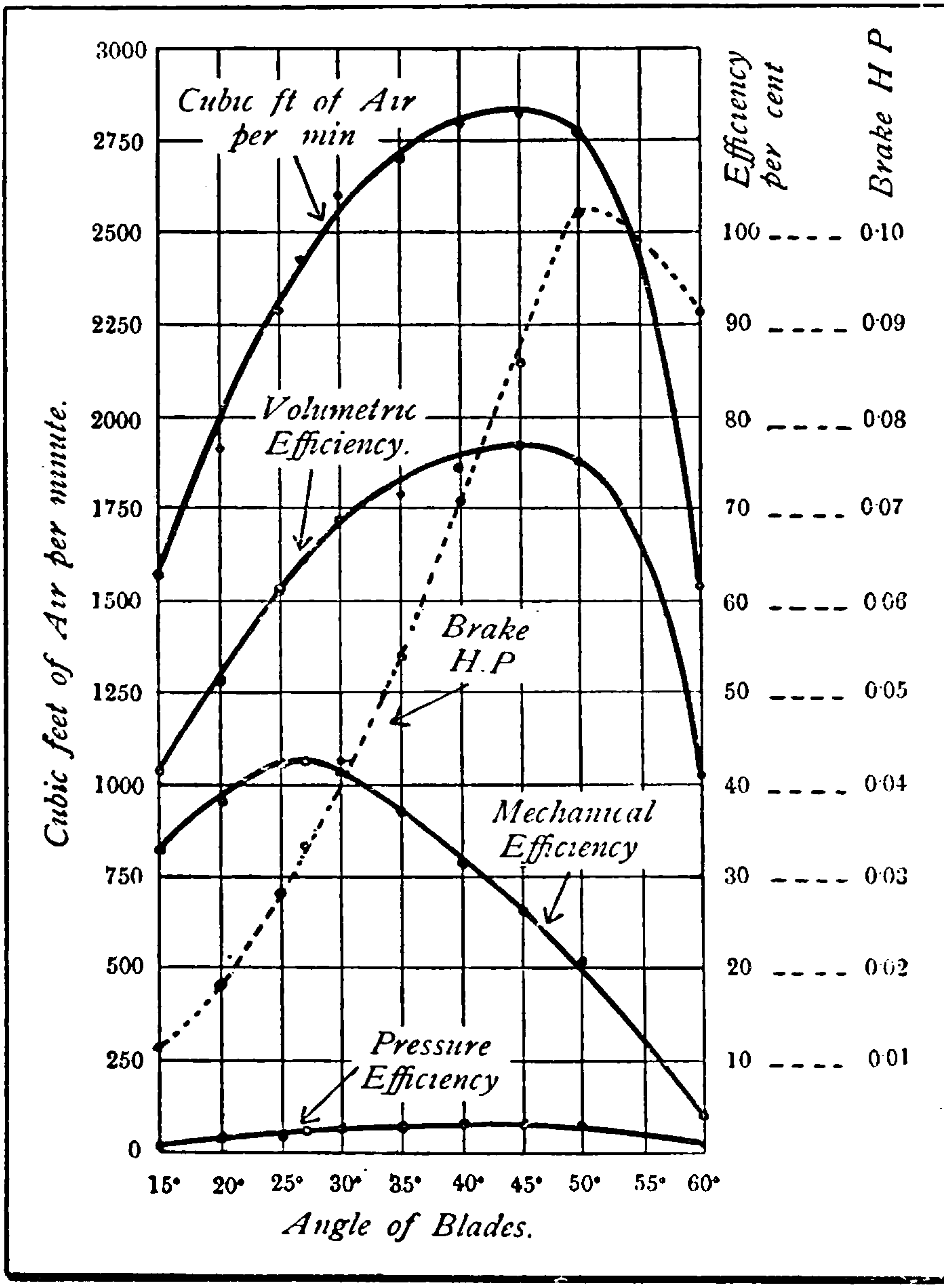


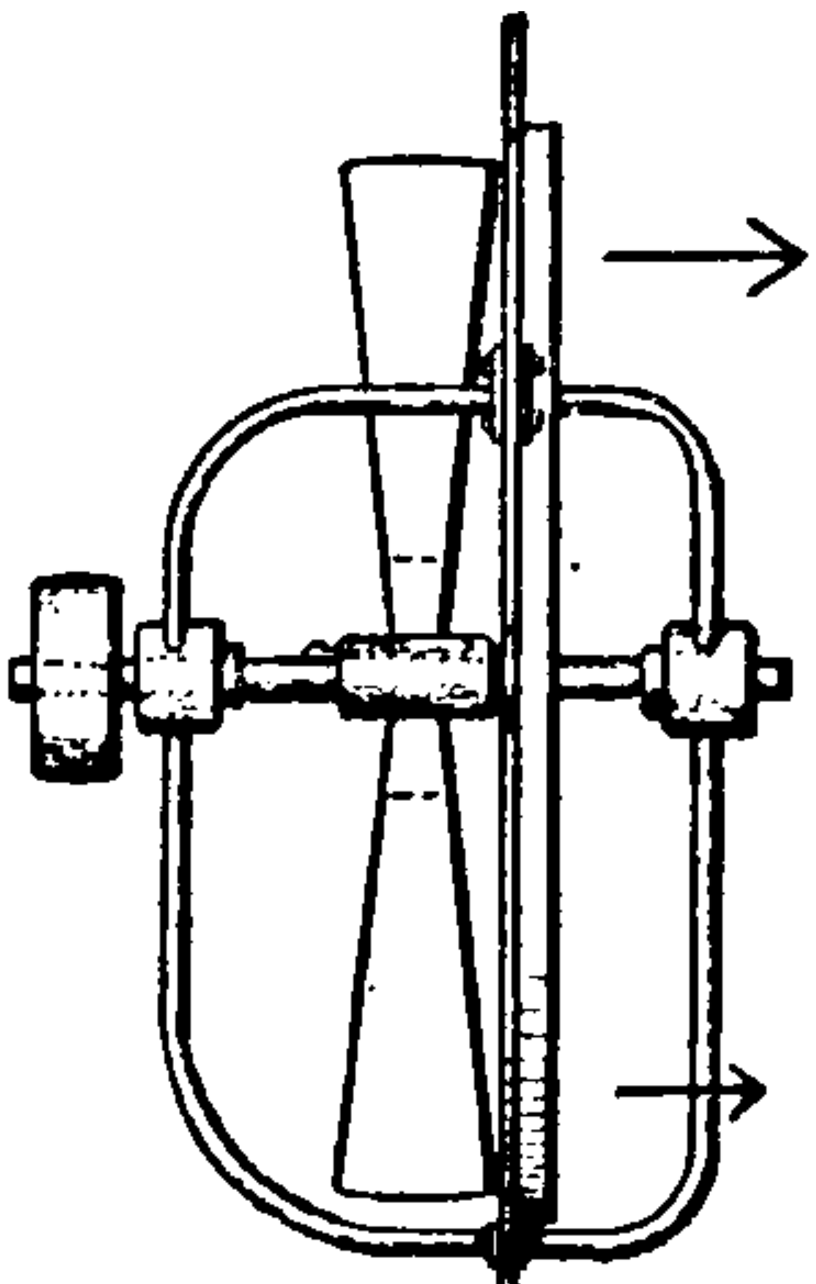
FIG. 118.—PLOTTINGS OF TABLE 55 (p. 230).

0·0338 horse power for the bearing friction of the motor, the efficiencies of fan alone are as below:

Angle of vane ...	15°	20°	25°	27°	30°	35°	40°
Mechanical efficiency of fan, No. 16	30·8	46·1	46·0	42·4	40·2	33·7	29·3
Mechanical efficiency of fan, No. 17	33·0	38·7	42·6	42·8	41·6	37·0	31·4

So that one is about as good as the other. Seventeen three-bladed fans were tried in order to test the effect of the cross-section of the fan blades, fig. 110. They may be divided into four groups. The first comprises fans 1 to 4, the second 5 to 10, the third 11 to 15, and the fourth 16 and 17. The blades were of sheet iron $\frac{1}{16}$ in. thick, and excepting 10 their cross-sections are either lines or arcs of circles. The fans in each group differed from one another only in the cross-section of their blades, which were linear, plano-convex, concavo-convex, of different curvatures. Fan 1 had flat blades. Fan 2 was formed by fixing a circular back to fan 1. Fan 3 was formed by curving the blades of fan 1. Fan 4 was formed by fixing to the back of fan 3 a still more convex surface. The blades of the other groups were similar in form, but of different area and thickness. These changes in shape produced considerable effect. Fan 1 to fan 4 were all tried with their blades at 17 deg. inclination. The superiority in mechanical and volumetric efficiencies of the last should be noticed.

It appears that on exposing the perimeter of the fan, air is sucked into the outer circumference, so that the volumetric efficiency is largely increased—see fan 8 at 40 deg. Some of the fans were tried with exposed perimeter by moving the delivery tube $4\frac{1}{2}$ in. forward, as shown in dotted line in fig. 114. Thus in the case of fan 9 the mechanical and volumetric efficiencies were increased from 16·9 to 29·4 and 62 to 78 per cent. respectively. A much wider form of blade may be used in fans arranged to feed from the tips. This type of fan should therefore, where possible, be fixed with its circumference exposed.



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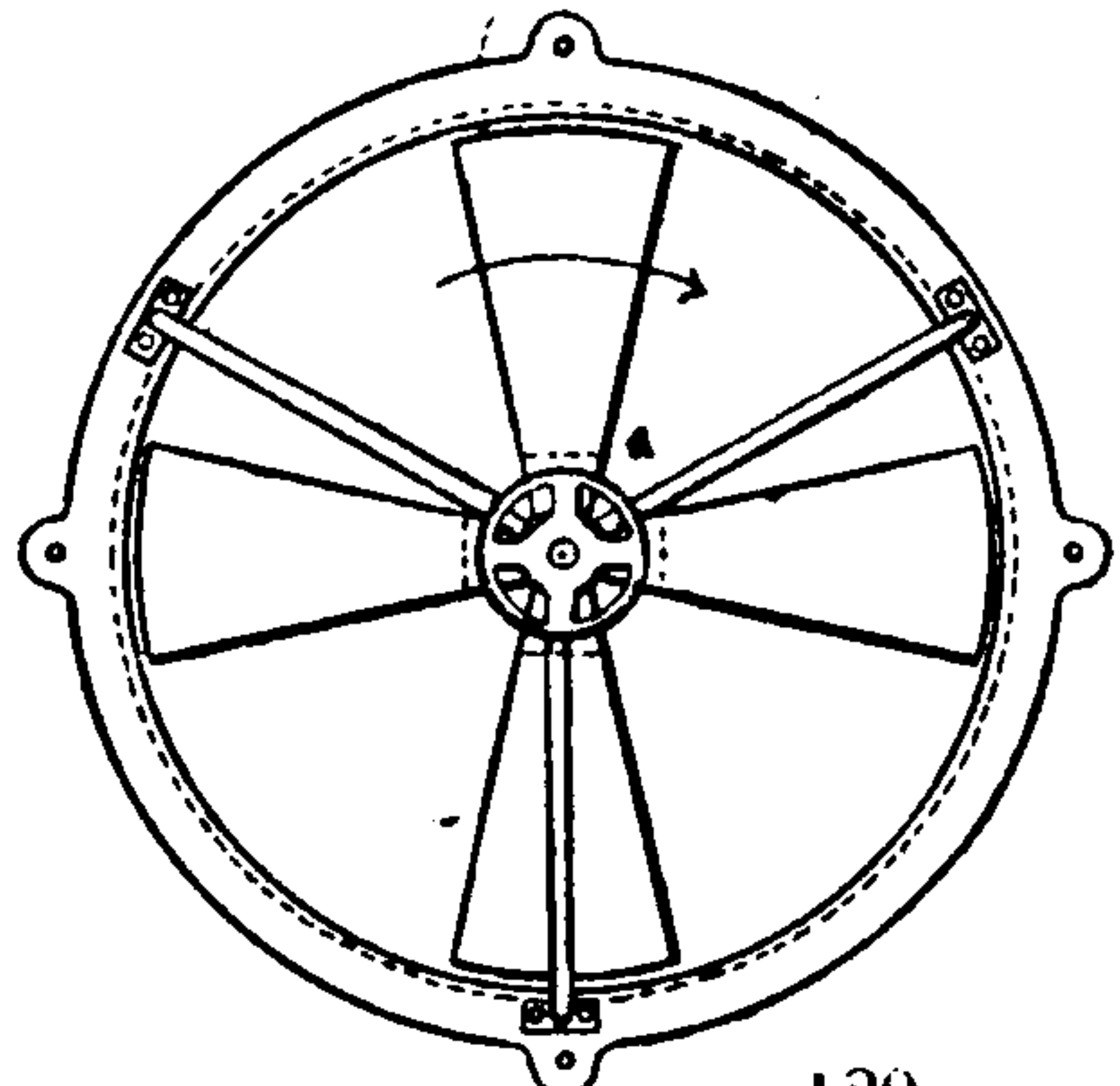

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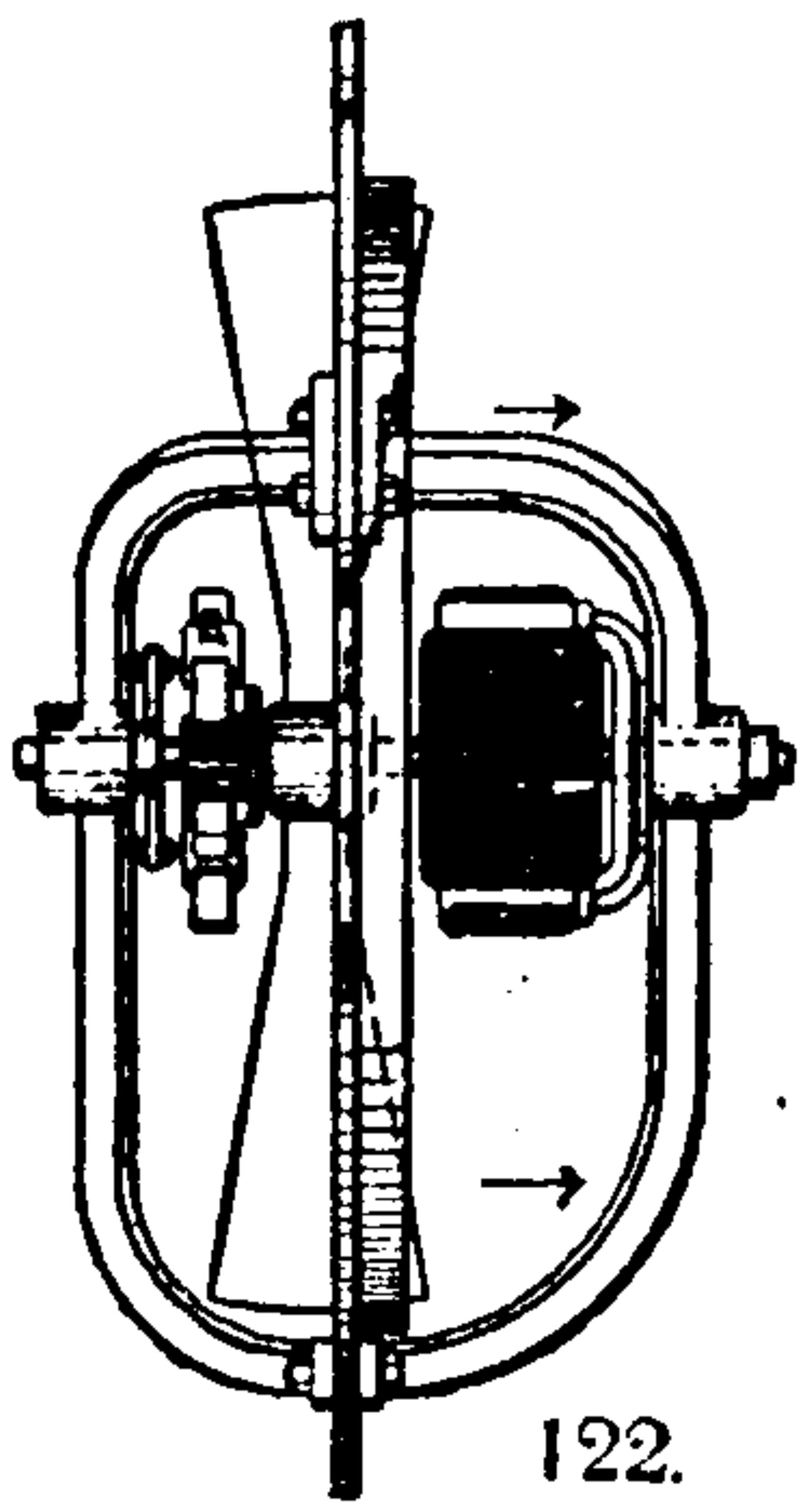

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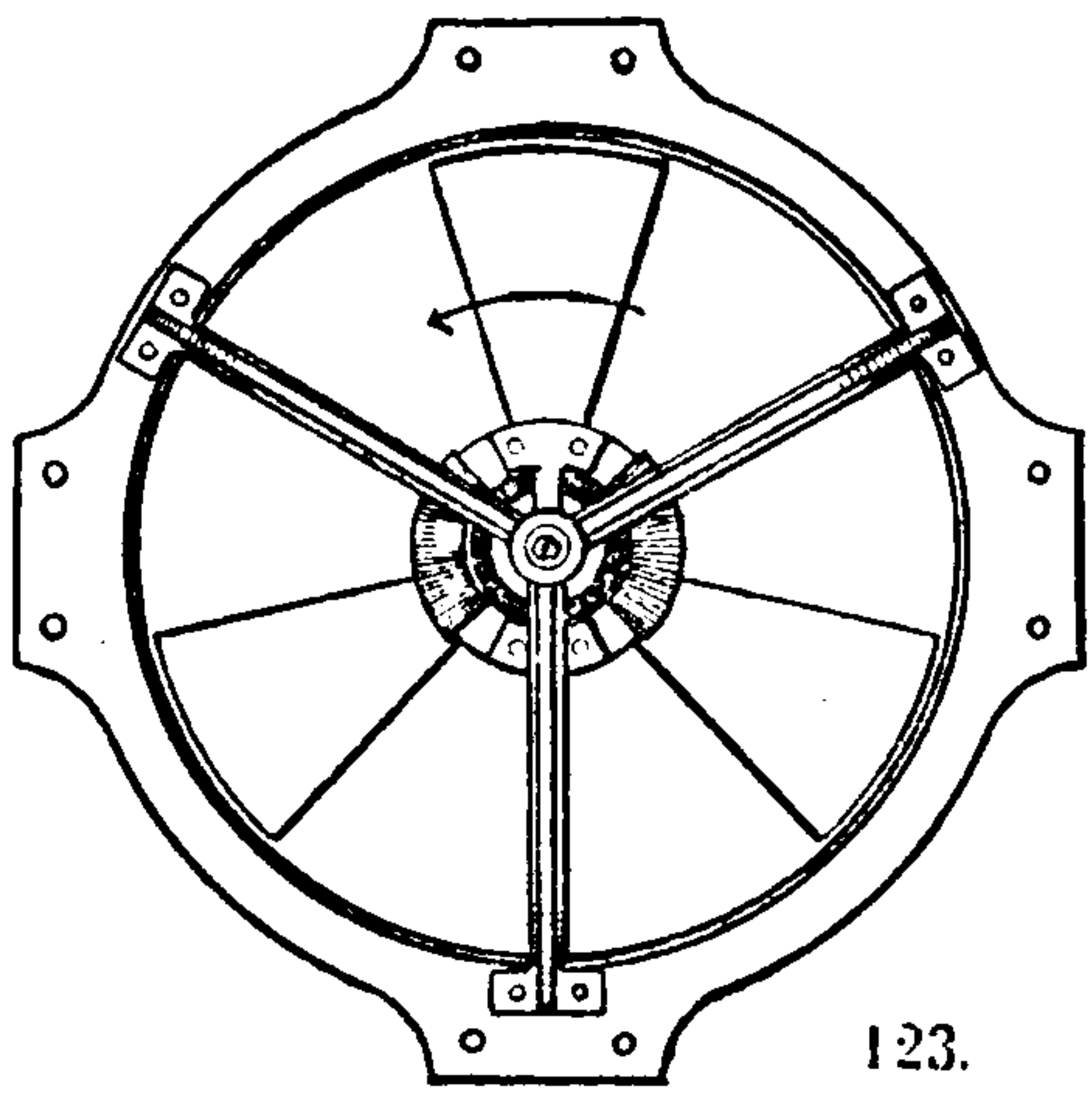

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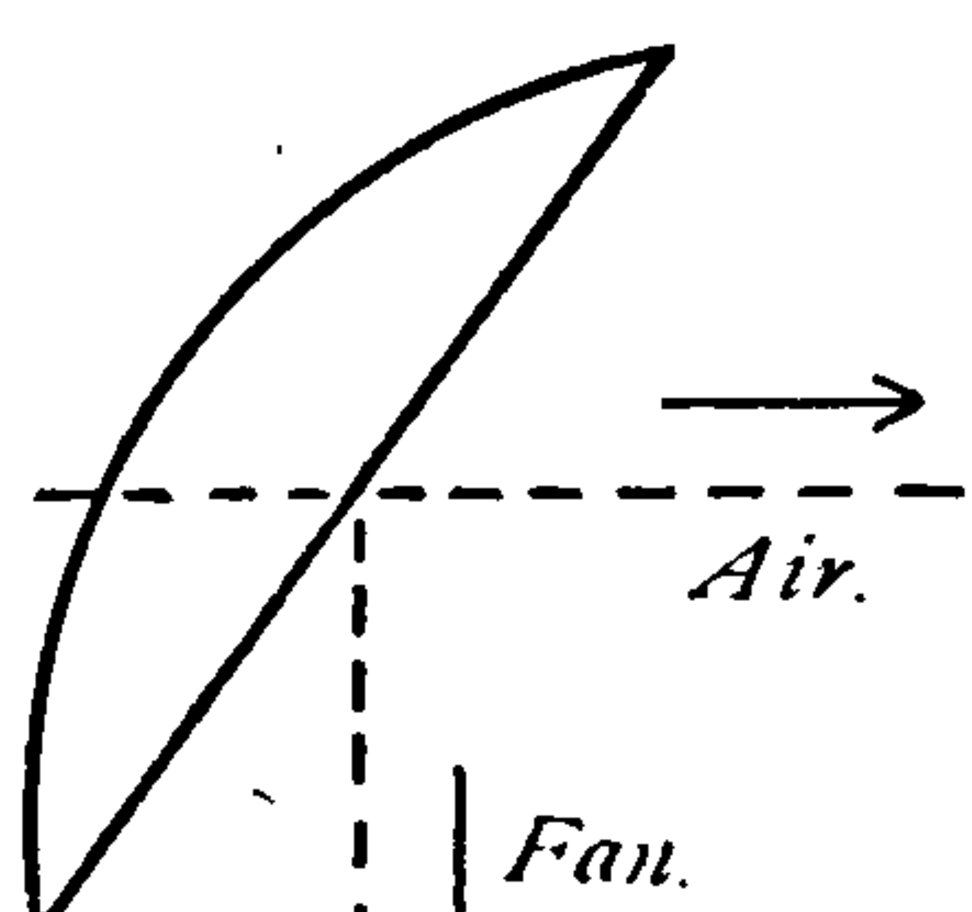

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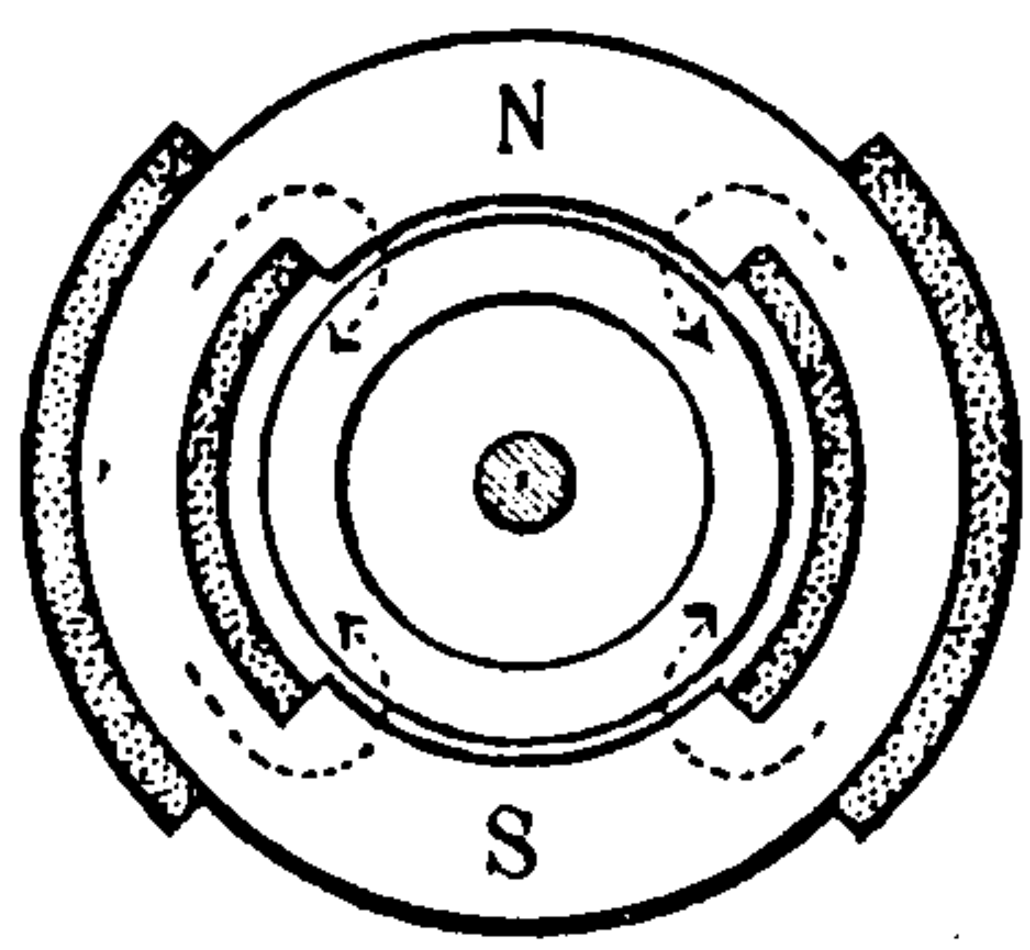

124.

FIGS. 119 AND 120.—24-IN. AND 48-IN. FANS. FIG. 121.—CURVATURE OF BLADES IN FAN, FIGS. 119 AND 120. FIGS. 122 AND 123.—FAN WITH SPECIALLY DESIGNED ELECTRIC MOTOR. FIG. 124.—FIELD MAGNETS (IN SECTION) OF FAN, FIGS. 122 AND 123.

EFFECT OF DISC ON FANS IN FIGS. 122 AND 123.

In figs. 122, 123 is shown a fan with a specially designed electric motor. The field magnets are cylindrical, and are shown in section in fig. 124; they are made as small as possible in order to reduce the resistance offered to the passage of the air. The best position of the motor, from this point of view, was generally found to be a little in front of the fan, the exact position depending on the size of the orifice. 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 a free discharge, to the centrifugal action on the front face of the blades, which is apparent near the centre.[23]

Several experiments were also made with contracted outlet and inlet with a fan $23\frac{3}{4}$ in. dia., the blades being set at 35 deg. to the plane of rotation. The fan was driven at 800 rev. per min. from a shunt-wound motor; it discharged into a 2 ft. dia. delivery tube 4 ft. long with partially closed outlet, having central holes 6 in., 12 in., and 18 in. dia. Under these conditions the fan was tried both for propelling and exhausting air, and its efficiency was in both cases much reduced. The fact that this type of fan is unable to maintain static pressure, is probably due to the comparatively slow speed of the blades near the centre, in consequence of which the air tends to pass back again through the centre of the fan. The effect of fixing a circular disc on the delivery side, so as to prevent the air from returning through the fan, was to increase 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 in order to obtain a good efficiency, the size of disc should increase with the contraction of orifice. The reason for this is evident, because a and v are greater nearer the outer circumference, and $a\,v$ must always exceed g H.

Figs. 125, 126, 127 show the effect of a disc with fan, fig. 122, 123, on the circulation of the air; in the case in which the disc is omitted the air to a great extent returns through the centre of the fan. Considering that this type of fan is used for 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 a necessity. The fan is more efficient when exhausting than when producing pressure. Without a circular disc 31 cu. ft.

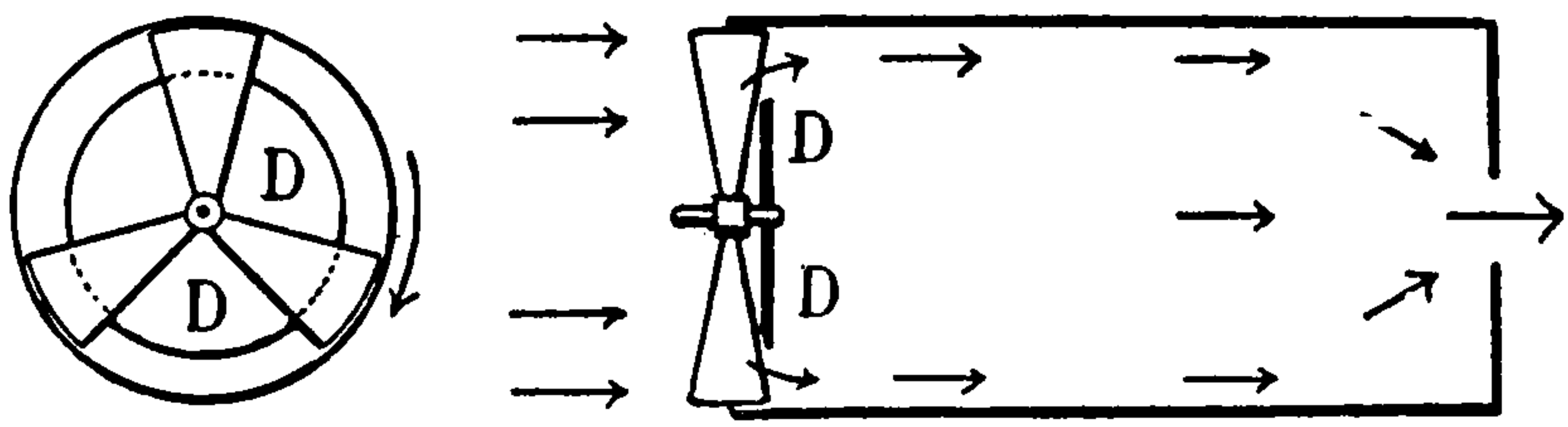

FIG. 125.—WITH DISC (*DD*), CONTRACTED OUTLET, PROPELLING.

of air were driven through the 6 in. orifice per min., when running at a speed of 800 rev. per min. The delivery was increased to 451 cu. ft. when exhausting, the

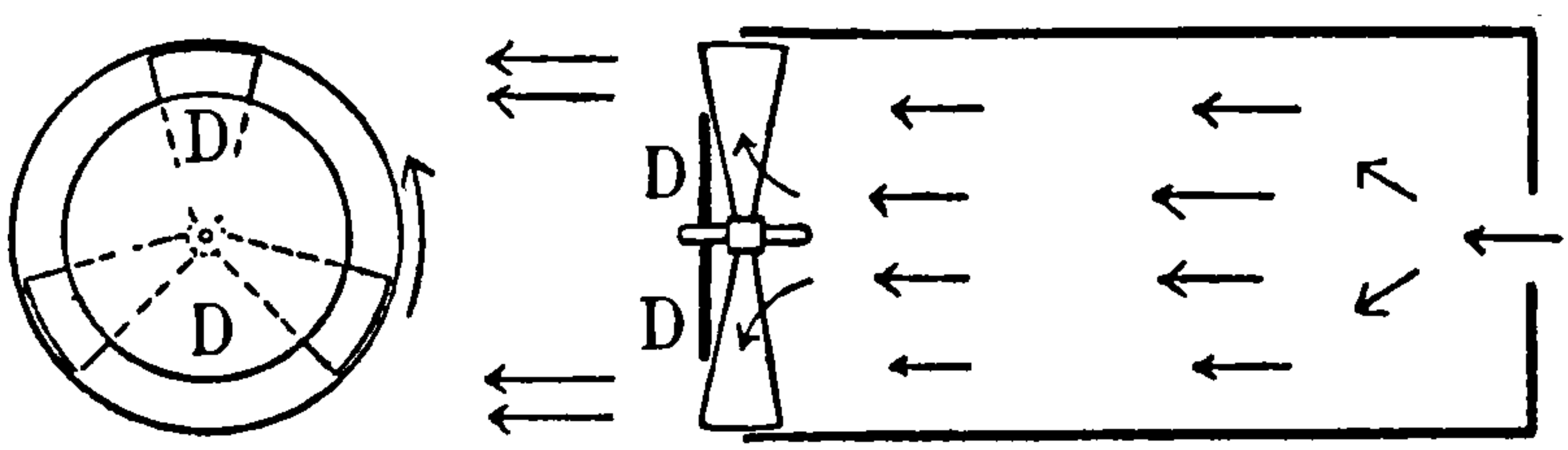

FIG. 126.—WITH DISC, CONTRACTED INLET, EXHAUSTING.

other conditions being identical; the volumetric efficiency was thus increased nearly 15 times. With the 12 in. orifice the discharges were 497 when blowing and 1,374 when

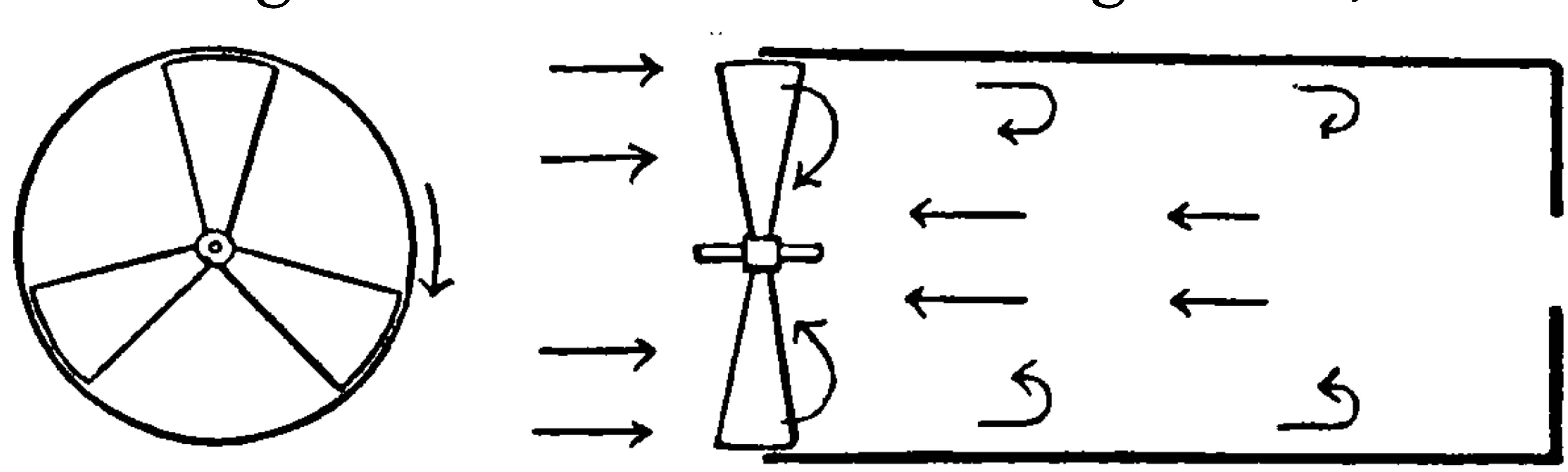

FIG. 127.—WITHOUT DISC, CONTRACTED OUTLET, CIRCULATING.

exhausting. With the 18 in. orifice the volumetric efficiency was increased only from 58½ per cent. when blowing to 67 when exhausting. The advantage of suction over blowing is largely due to the fact that the dynamic head is very

noticed in the axial discharge of the air. The angle at the tips was 27 deg. with the plane of revolution. The helical blades were made of $\frac{1}{16}$ in. sheet brass pressed on a wood mould. The fan was run at 600 rev., and the anemometer was placed 18 in. in front of the fan. No delivery tube was employed, and the axial velocities at radii of 2 in., 5 in., and 12 in. were 770, 1,340, and 230, while at the centre the velocity was 665. The rotary velocities were also measured by placing the anemometer wheel in a plane passing through the axis. The velocities at radii of 2 in., 5 in., 8 in., and 12 in. were 234 ft., 562 ft., 530 ft., and 185 ft. per min. The efficiency was increased by putting rounded backs to the blades, but the experiments showed that helical blades did not possess any advantages over the ordinary non-helical blades. With fans of later design, having plano-convex blades, Walker obtained volumetric efficiencies of 86 to 90 per cent., but he does not give the angle of blade.

The horse power necessary for driving the fan to produce a given discharge of air is as follows: Taking the barometer at 30 in., the temperature of the air at 60 deg Fah., and the mechanical efficiency at 30 per cent., let d be the diameter of the fan in feet, a the area of the fan disc in square feet, V the velocity of the air in ft. per sec, and Q the quantity of air discharged in cu. ft. per sec., then H.P. required to drive the fan

$$=\frac{\text{H.P in d scharged air}}{\text{mechanical efficiency}}=\frac{V^2 Q B}{\tau}\times\frac{0{\cdot}00003756}{0{\cdot}3},$$

and substituting $\tau = 60 + 461$, and $V = \frac{4Q}{\pi d^2}$

$$\text{H.P. required to drive the fan} = 1{\cdot}17 \times 10^{-5}\frac{Q^3}{d^4}.$$

Further $Q = \eta_r \cdot v_2 v_2^2$, and taking the volumetric efficiency at 90 per cent.

$$Q = \frac{N d^3}{85}.$$

where N = rev. per min. Thus a fan 2 in. dia., running at 600 rev. per min., discharges 3,400 cu. ft. per min.

It is evident from these two formulæ that d should be as large as possible. Of course these formulæ only apply to free discharge. The effect of increase of diameter is shown by the fact that to discharge 6,000 cu. ft. of air per min. the 2 ft. fan would require 0·73 horse power, while a 4 ft. fan would need only 0·045.

CHAPTER XII.

Other Propeller Ventilating Fans and Rateau Screw Fans.

There are many types of ventilating fans in use at the present day. The Hattersley-Pickard fan is designed not only to deliver a large volume of air, but to discharge against a considerable static head. It is fitted with a boss of large diameter; this is because the central position of the propeller is not only useless for moving air, but is absolutely harmful when working against pressure. The blades are inclined towards the intake so as to enable the air to cross them at right angles, and therefore with the least possible friction. They are of helical construction, with a longitudinally or axially increasing pitch—*i.e.*, ϕ is greater than θ, and at the outer circumference they appear excellently adapted for drawing in the air radially, as well as axially. This appears to be a most carefully designed fan. The following is a list of particulars of standard sizes :

Diameter of blade in inches.	Revolutions per minute.	Cu. ft. of air per min.	Actual H.P. required.	Diameter of pulley (inches).	Width of belt (inches).	Volumetric efficiency at max. discharge and revolutions.
18	700 to 1,200	2,150 to 3,600	$\frac{1}{8}$ to $\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{1}{2}$	113 %
24	500 to 900	3,500 to 6,600	$\frac{1}{6}$ to $\frac{5}{8}$	4	$1\frac{3}{4}$	117 %
30	450 to 750	5,700 to 9,800	$\frac{1}{4}$ to 1	5	2	117 %
36	400 to 650	9,500 to 16,000	$\frac{1}{2}$ to $1\frac{1}{4}$	6	$2\frac{1}{2}$	117 %
48	300 to 350	17,500 to 31,500	1 to $2\frac{1}{4}$	8	$3\frac{1}{2}$	118 %

The Blackman fan is very largely used in this country. Figs. 128 and 129 show a propeller fan made by Beck and Henkel, of Cassel, designed to discharge large quantities of air at low pressure. The vanes are made of steel, and revolve in a conical casing. These fans are made with

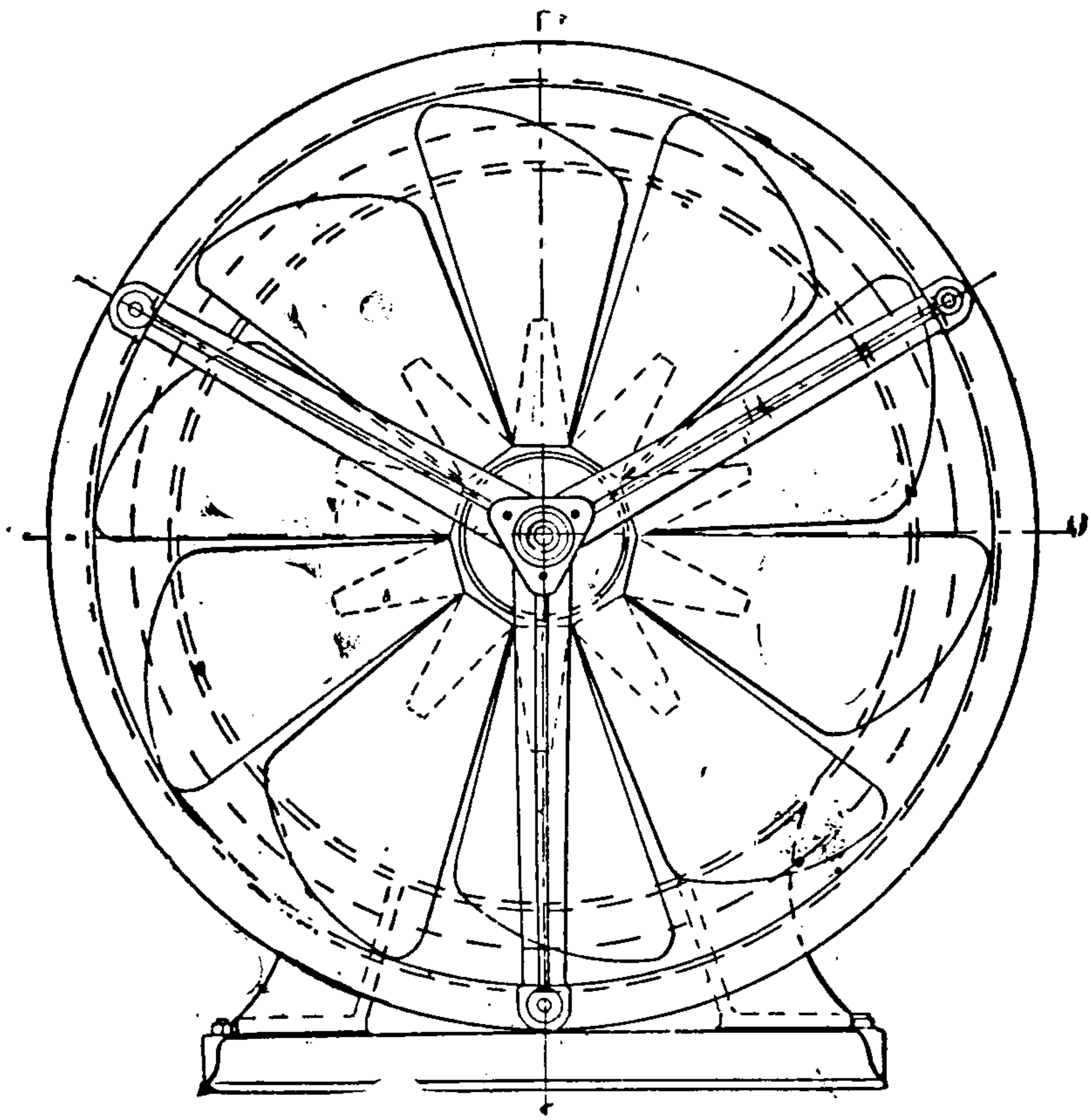

FIG. 128.

diameters from 10 in. to 118 in.; they can be driven direct or by transmitted power.

Rateau Screw Fans.—The propeller type of fan is certainly that best suited for delivering a large amount of air at an exceedingly small pressure; but where a large amount has to be discharged against some considerable head, a modification of the propeller fan, in which only the outer portions of the blades (*i.e.*, the part where v is great)

cylindrical section through the guide and wheel blades of a Rateau screw fan. The direction of the air's motion is shown by the arrows. The guide vanes are *m*, *m* and the wheel vanes *a*, *b*. The motion of the wheel is upwards, and the wheel vanes are so designed that inflow takes place at the normal orifice without shock. The sectional elevation of fig. 131 needs no comment, except that the chamber D has its inner side partly conical in order to

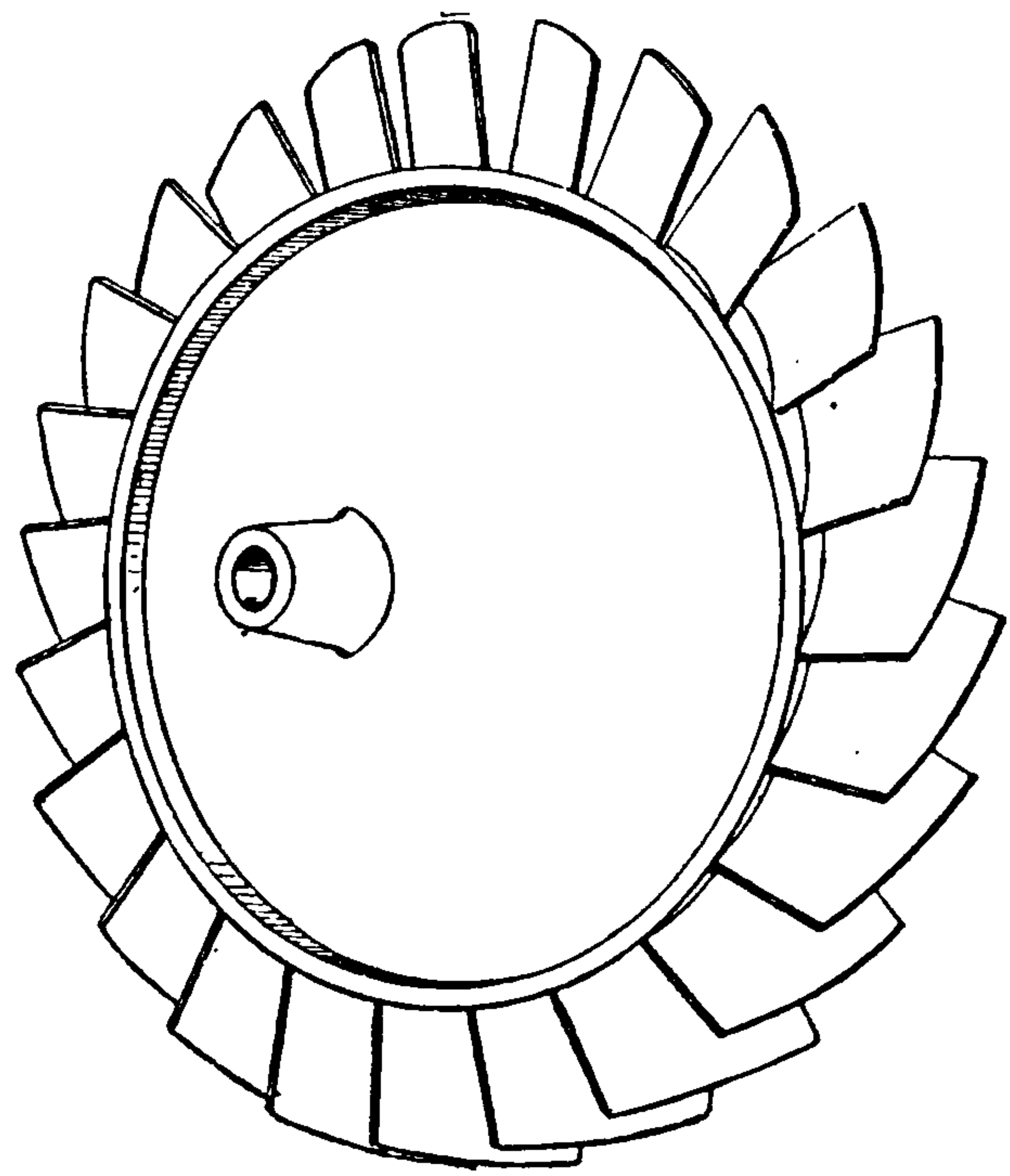

FIG. 130.

reduce the velocity of the air by increasing the section of discharge. The wheel, fig. 130, has its vanes formed of steel plate fixed to the rim of a slightly conical wheel of cast iron or bronze, by means of angle irons in the case of large fans, or by embedding them into the rim in smaller sizes. A general view is seen in fig. 132. An alternative arrangement is shown in figs. 133, 134. The spiral admission chamber gives the entering air velocity in the opposite direction to that of rotation, and after leaving the fan with a velocity wholly axial, the air is discharged through a passage whose section is increased by making its inner surface the frustrum of a cone. A third method is shown in fig. 135; here the air enters parallel to the axis, and

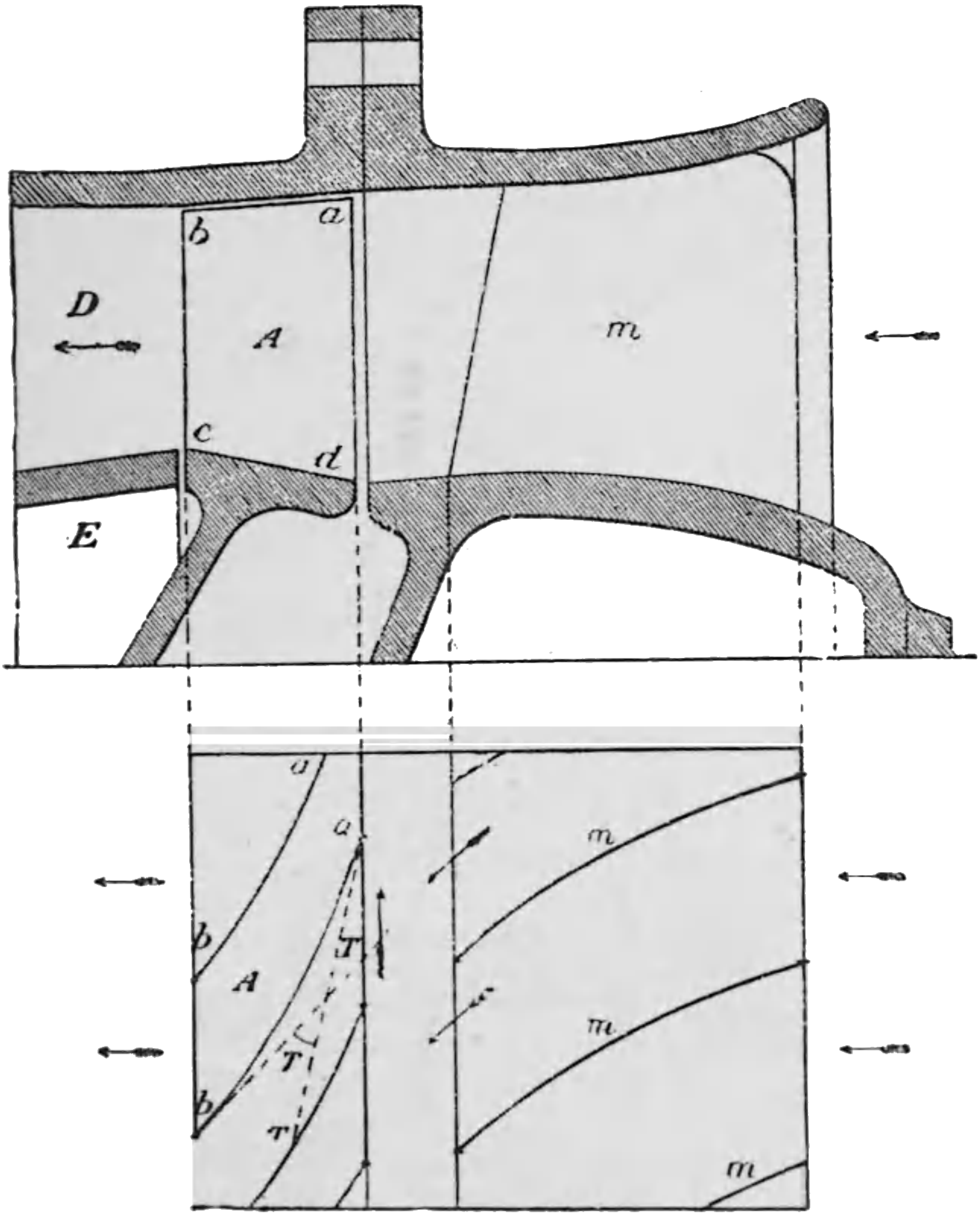

FIG. 131.

the vanes at inflow are so inclined as to receive it without shock. The change of the moment of momentum is effected by providing a volute to reduce the tangential motion at discharge, and so convert the kinetic energy of

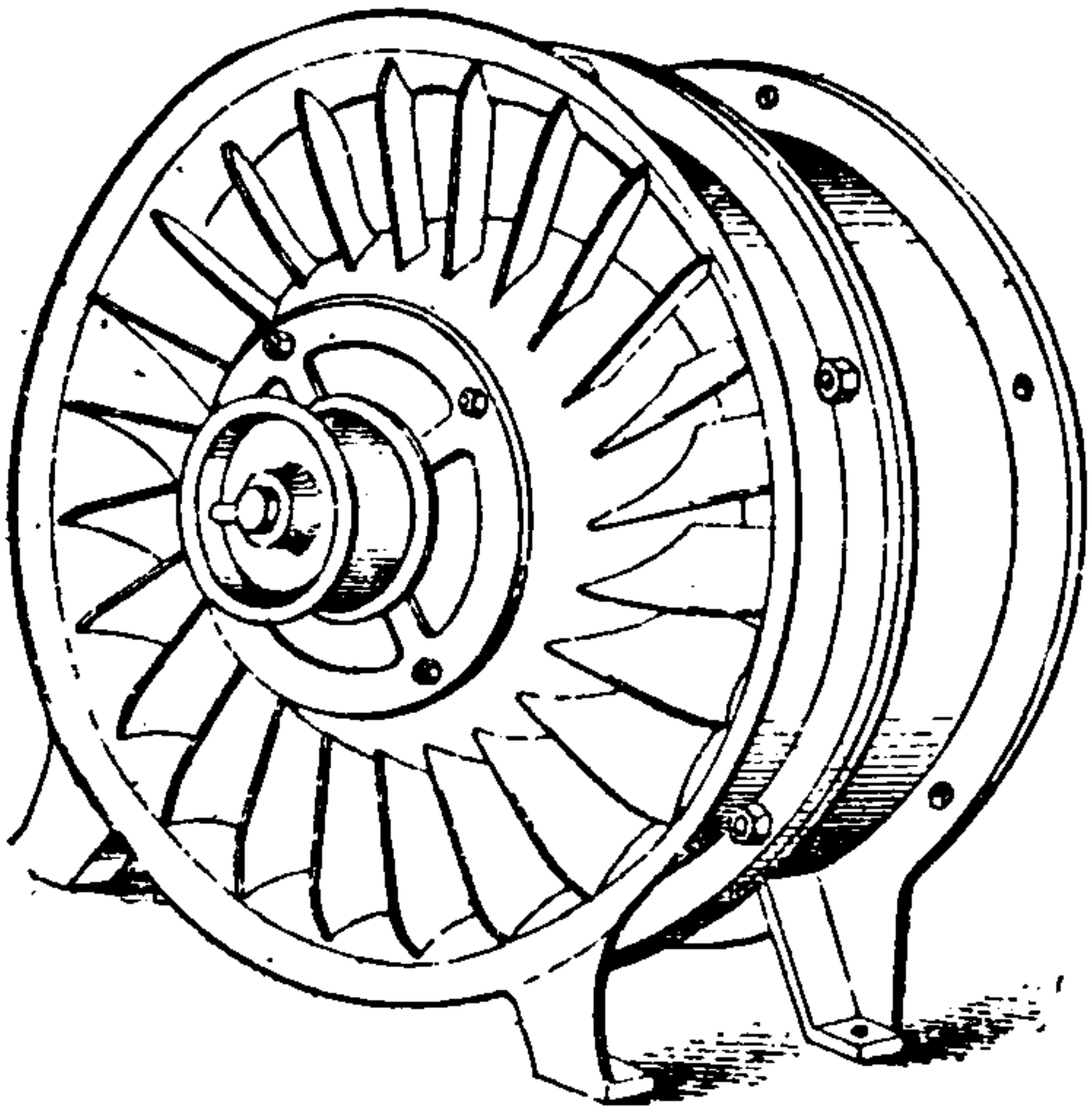

FIG. 132.

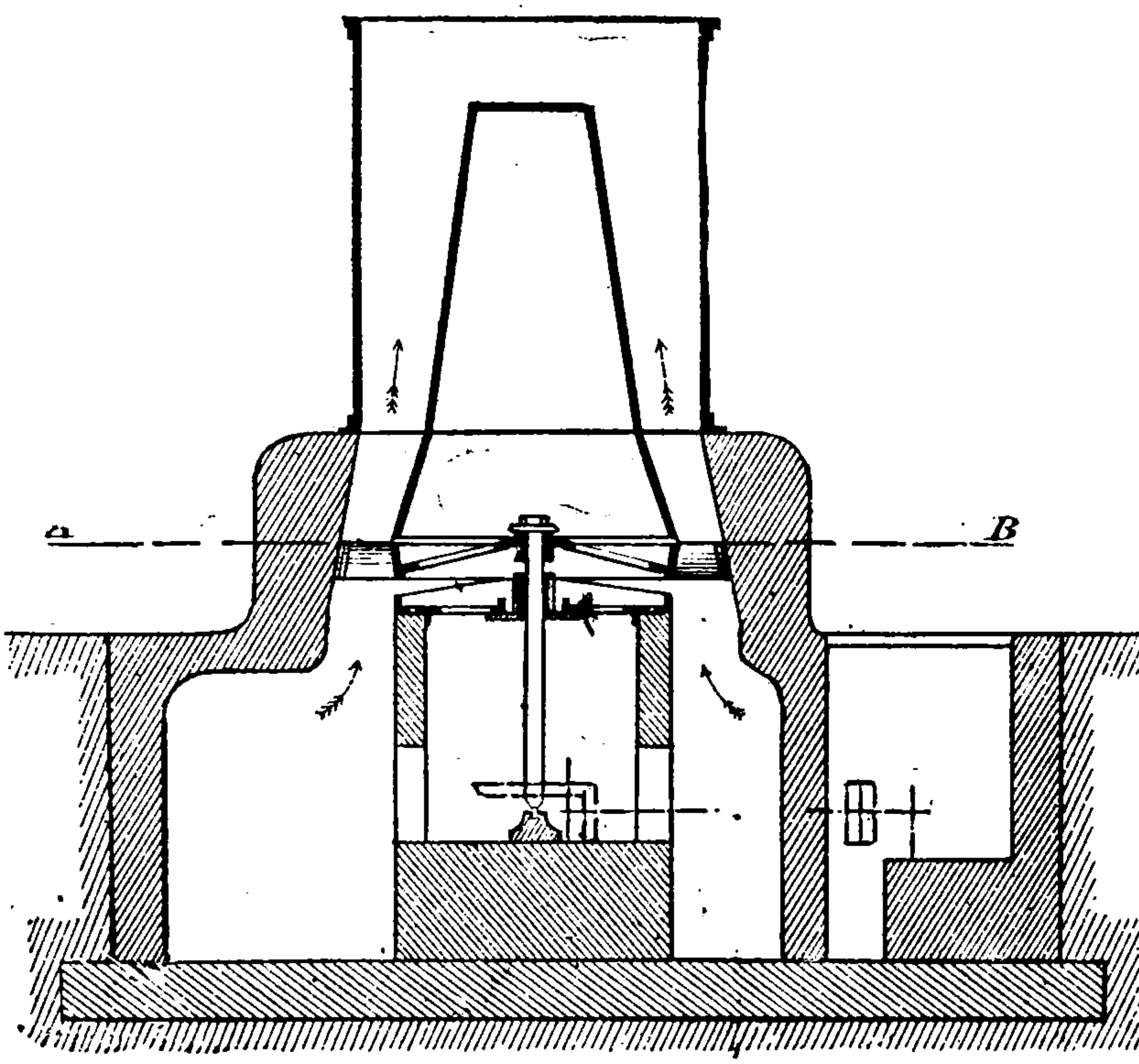

FIG. 133.

where Q = cubic feet of air per second,
A = area of section in square feet,
c = coefficient of contraction,
H = pressure head in feet of air,
h = water gauge in inches,
δ = density of water,
σ = density of air.

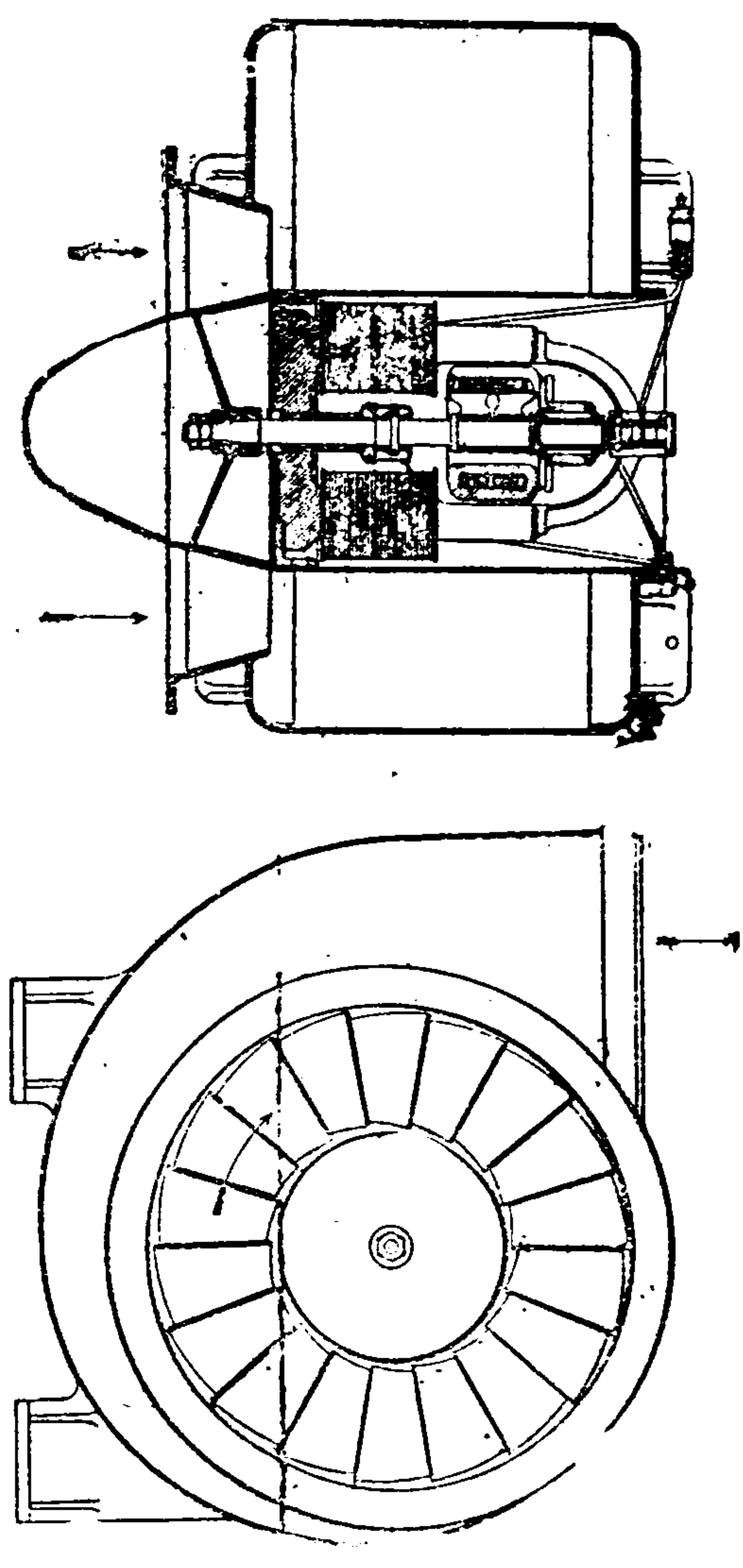

FIG 135

Test of an Axial or Screw Fan of a Type similar to that shown in Fig. 135, of 4·69 ft. Extreme Diameter of Vanes.

No. of revolutions per minute	526	545	535	532	552	556	556	564	573
Water gauge in ins.	1·84	1·61	1·53	1·53	1·53	1·53	1·34	1·29	1·22
Kilowatts of motor...	10·7	9·9	9·65	9·85	10·1	10·0	9·5	9·6	9·9
Foot-pounds of work by fan per second	0	1156	1705	2390	3030	3170	3180	3180	3461
Discharge in cu. ft. per second... ...	0	137	215	296	386	396	452	474	554
Mechanical efficiency of fan and motor ..	0	0·16	0·24	0·32	0·41	0·43	0·45	0·45	0·47*
Manometric efficiency	0·26	0·21	0·205	0·21	0·19	0·19	0·18	0·155	0·14
Volumetric efficiency	0	0·185	0·30	0·42	0·52	0·53	0·61	0·62	0·71*

* It should be noted that maximum volumetric and mechanical efficiencies occur at the same orifice.

The Theory of Rateau Screw Fans.—We shall first consider the type shown in figs. 130, 131 and 132. Referring to the sectional view of guide and wheel vanes, fig. 131, let a be the outlet angle made by the guide vanes m at the mean radius of the wheel, with a plane perpendicular to the axis, θ ϕ the angles made with such a plane by the wheel vanes at inflow and outflow. Let v_m be the speed of the wheel at the mean radius r_m and b_1 the axial component of the air at discharge from the guide wheel. Then from the velocity triangle (fig. 136), if the air enters the moving wheel without shock,

$$\cot \theta = \frac{v_m}{b_1} + \cot \alpha.$$

Again, at outflow, which is axial—*i.e.*, $a_2 = o$—if b_2 is the axial component of discharge, which in fig. 136 is greater than b_1, then

$$v_m = b_2 \cot \phi.$$

The work done per pound of air by the wheel is—

$$\frac{a_1 v_m}{g} = \frac{v_m\, b_1 \cot \alpha}{g}$$

The loss by friction in the guides is—

$$= F_1 \frac{b_1^{\,2}}{2\,g} \operatorname{cosec}^2 \alpha,$$

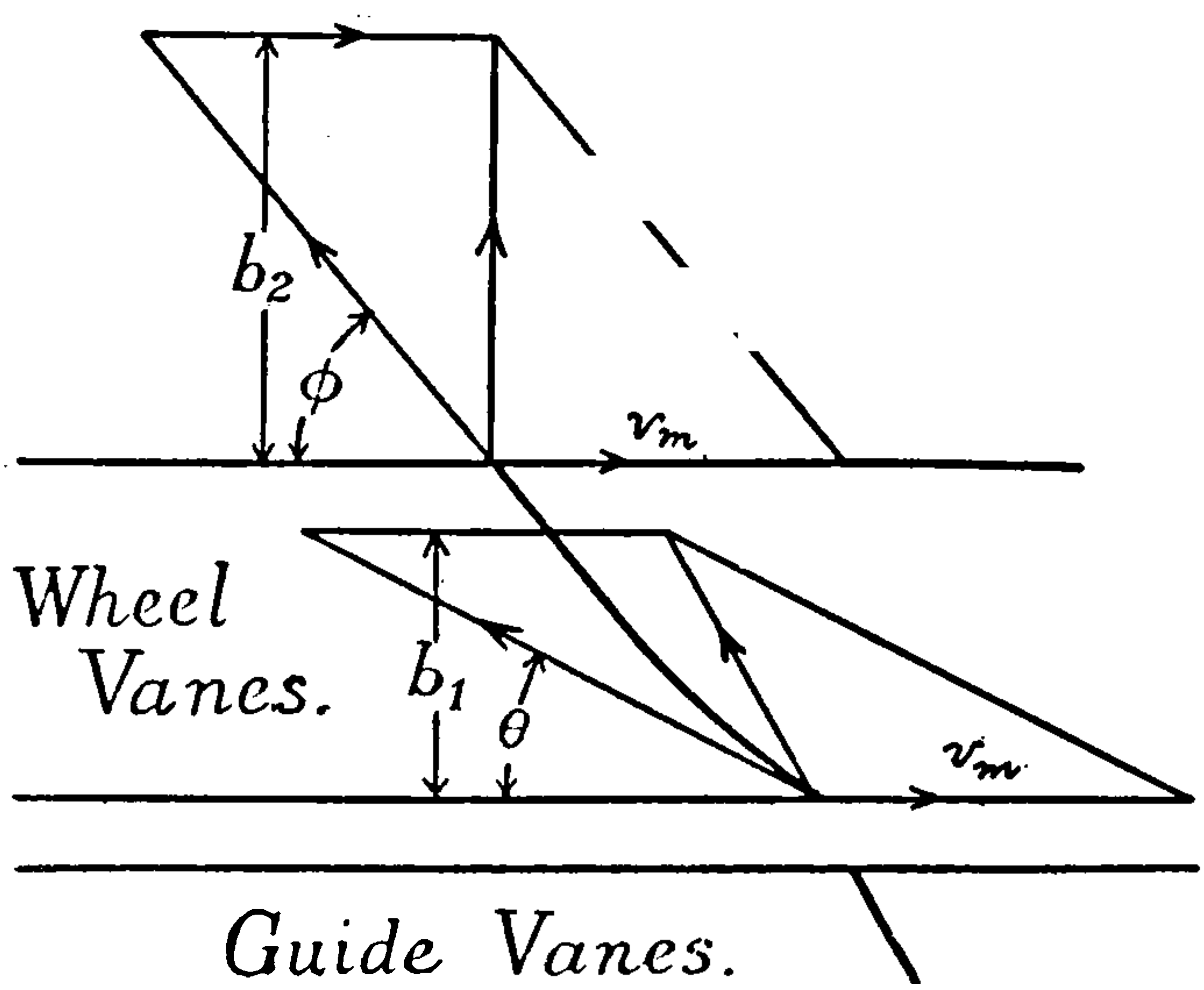

FIG. 136.

the frictional loss in the wheel—

$$= \frac{F_2}{4g}(b_1^{\,2} \operatorname{cosec}^2 \theta + b_2^{\,2} \operatorname{cosec}^2 \phi),$$

and the leaving losses are—

$$= (1 + F_3)\, \frac{b_3^{\,2}}{2\,g},$$

where b_3 = velocity of discharge from the fan casing to the left of D.

we obtain an equation between H, v_m, b_2, s_1, s_2, s_3, and a in the form of a quadratic in v_m and b_2—

$$F_2\, v_m{}^2 - \left[\left(1 - \frac{F_2}{2}\right) 2\,\frac{s_2}{s_1} \cot\ a\right] v_m\, b_2$$
$$+\left[\left(F_1 + \frac{F_2}{2}\right) \operatorname{cosec}^2 a\,\frac{s_2{}^2}{s_1{}^2} + \frac{F_2}{2} + (1 + F_3)\frac{s_2{}^2}{s^3{}_2}\right] b_2{}^2 + 2gH = O.$$

In designing this type of fan, however, it is best to commence by making certain assumptions We can safely assume that the mechanical efficiency of the fan alone is about 60 per cent., and as this type of fan is intended to give a high volumetric efficiency, we should take this as 75 per cent.; then

$$\eta_v = \text{volumetric efficiency} = \frac{Q}{v_2\, r_2{}^2}$$

where v_2 = peripheral speed,

r_2 = radius of the tips of the blades,

$$\eta_v = \frac{2\pi\, r_m\, s_1\, b_1}{v_m \frac{r_2}{r_m}\, r_2{}^2} = \frac{2\,\pi\, r_m{}^2\, s_1\, b_1}{v_m\left(r_m + \frac{s_1}{2}\right)^3}.$$

Now, we shall suppose that $s_1 = \frac{r_m}{3}$,

so that $$\eta_v = \frac{2}{3}\,\frac{\pi\, b_1}{1{\cdot}166 \times 1{\cdot}36\, v_m} = 1{\cdot}32\,\frac{b_1}{v_m},$$

and putting $\eta_v = 0{\cdot}75$, $b_1 = \dfrac{0{\cdot}75\, v_m}{1{\cdot}32} = 0{\cdot}568\, v_m$

$$\eta = \frac{gH}{a_1\, v_m} = \frac{gH}{v_m\, b_1 \cot a}$$

$$\therefore \cot a = \frac{1{\cdot}32\, gH}{0{\cdot}6 \times 0{\cdot}75\, v_m{}^2}$$
$$= 2{\cdot}93\, \eta_m$$

$$b_2 = \frac{s_1\, b_1}{s_2} = \frac{7}{6} b_1 \text{ let us say,}$$
$$= 0{\cdot}663\, v_m,$$

so that $$\cot \phi = \frac{v_m}{b_2} = 1{\cdot}51$$

$$\therefore \phi = 33 \text{ deg. } 40 \text{ min.},$$

independent of the manometric efficiency, and dependent only on η_v, r_m, s_1, and s_2. Again,

$$\cot \theta = \frac{v_m}{b_1} + \cot a = 1{\cdot}76 + 2{\cdot}93\,\eta_m.$$

The following is a list of values of a and θ for various values of η_m, calculated from the above equations:

η_m per cent. =	10	20	40	60
a = ...	73° 40′	59° 38′	40° 29′	29° 38′
θ = ...	25° 58′	23° 5′	18° 50′	15° 52′

In the next type of fan with a spiral inflow passage but no guide vanes, the manometric efficiency will increase the greater velocity of the inflowing air; the speed of the air in this passage is the tangential component of the velocity of inflow, a. Let A be the sectional area of the passage, and Q the discharge in cu. ft. per second; then

$$a_1 = \frac{Q}{A}, \qquad b_1 = \frac{Q}{2 . \pi . r_m . s_1},$$

$$b_2 = \frac{Q}{2\,\pi\, r_m\, s_2}, \text{ and } v_m = b_2 \cot \phi,$$

$$\cot \theta = \frac{v_m + a_1}{b_1}.$$

Then we may proceed as follows:

$$\eta = \frac{g\,H}{a_1\, v_m}; \qquad a_1 = \frac{g\,H}{v_m\, \eta}.$$

$$\frac{a_1}{\sqrt{g\,H}} = \frac{\sqrt{\eta_m}}{\eta} = k \text{ say},$$

so that if η and η_m are assumed, a_1 and v_m are known in terms of H. Assuming $\eta = 60$ and $\eta_v = 75$ per cent., and $s_1 = \frac{v_m}{3}$ as before, then—

$$\cot\theta = \frac{a_1 + v_m}{0{\cdot}568\, v_m}$$

and is now known, and $b_2 = 0{\cdot}663\, v_m$ as before.

Further, $\cot\phi = \frac{c_1}{u_1} = 1{\cdot}51$ and $\phi = 33^\circ\ 40'$ as before.

We need not deal with the type of fan in fig. 135, as its theory is precisely the same as that of the radial-flow fan with volute.

In designing the type, fig. 137, we shall suppose the volumetric efficiency 75 per cent. as before, and that

$$s_1 = \frac{2}{3} r_m; \qquad s_2 = \frac{4}{11} r_m$$

then
$$\eta_v = \frac{2\,\pi\, r_m^2\, s_2\, b_1}{v_m\left(r_m + \frac{s_1}{2}\right)^3} = \frac{\pi\, b_1}{(1\tfrac{1}{3})^2\, v_m},$$

$$\frac{b_1}{v_m} = 0{\cdot}566\, \eta_v = 0{\cdot}425,$$

$$\frac{b_2}{v_m} = \frac{s_1}{s_2} \cdot \frac{b_1}{v_m} = 0{\cdot}778,$$

v_m being, of course, the velocity at the mean radius.

We can now find the vane angle at inflow, since

$$\cot\theta = \frac{v_m}{b_1} = 2{\cdot}35$$

and
$$\therefore \theta = 23^\circ\ 3'.$$

If ϕ is the angle made by the mean direction of motion of the air relative to the wheel at outflow, then

$$a_2 = v_m - b_2 \cot\phi$$

$$\cot\phi = \frac{v_m - a_2}{b_2} = 1{\cdot}286\left(1 - \frac{a_2}{v_m}\right).$$

The air leaving the wheel has an axial component b_2 and a tangential component a_2, and if the guide vanes are to receive the air without shock they must be inclined at angle α such that—

$$\tan \alpha = \frac{b_2}{a_2} = 0{\cdot}778 \frac{v_m}{a_2}$$
$$= \frac{0{\cdot}466}{\eta_m}.$$

The following is a list of values of ϕ and α for various values of M:

M per cent. =	10	20	30	40	60	80
ϕ = ...	43° 4′	49° 21′	57° 16′	66° 48′	90°	113° 12′
α = ...	77° 52′	66° 46′	57° 15′	49° 21′	37° 49′	30° 12′

CHAPTER XIII.

FANS AND COMPRESSORS: RECENT PRACTICE.

IMPROVEMENTS IN FAN DESIGN.

IN centrifugal fans, especially of the drum type, in which the blades are long in an axial direction compared with the radial depth, there is a tendency for the reduction in pressure at the inlet to cause the air on this side of the fan to be actually drawn from the discharge to the suction through the blades. This was first noted by S. C. Davidson when experimenting with a fan wheel open to the air at one end (fig. 138, I.) and rotating without an enclosing casing of any sort. The outward discharge of air was strongest at the back or closed end A, but the strength of this discharge gradually diminished and became nil at a point O nearly midway therefrom, and from this point onwards it assumed a centripetal flow. For this reason he termed A the positive and B the negative end of the fan. When

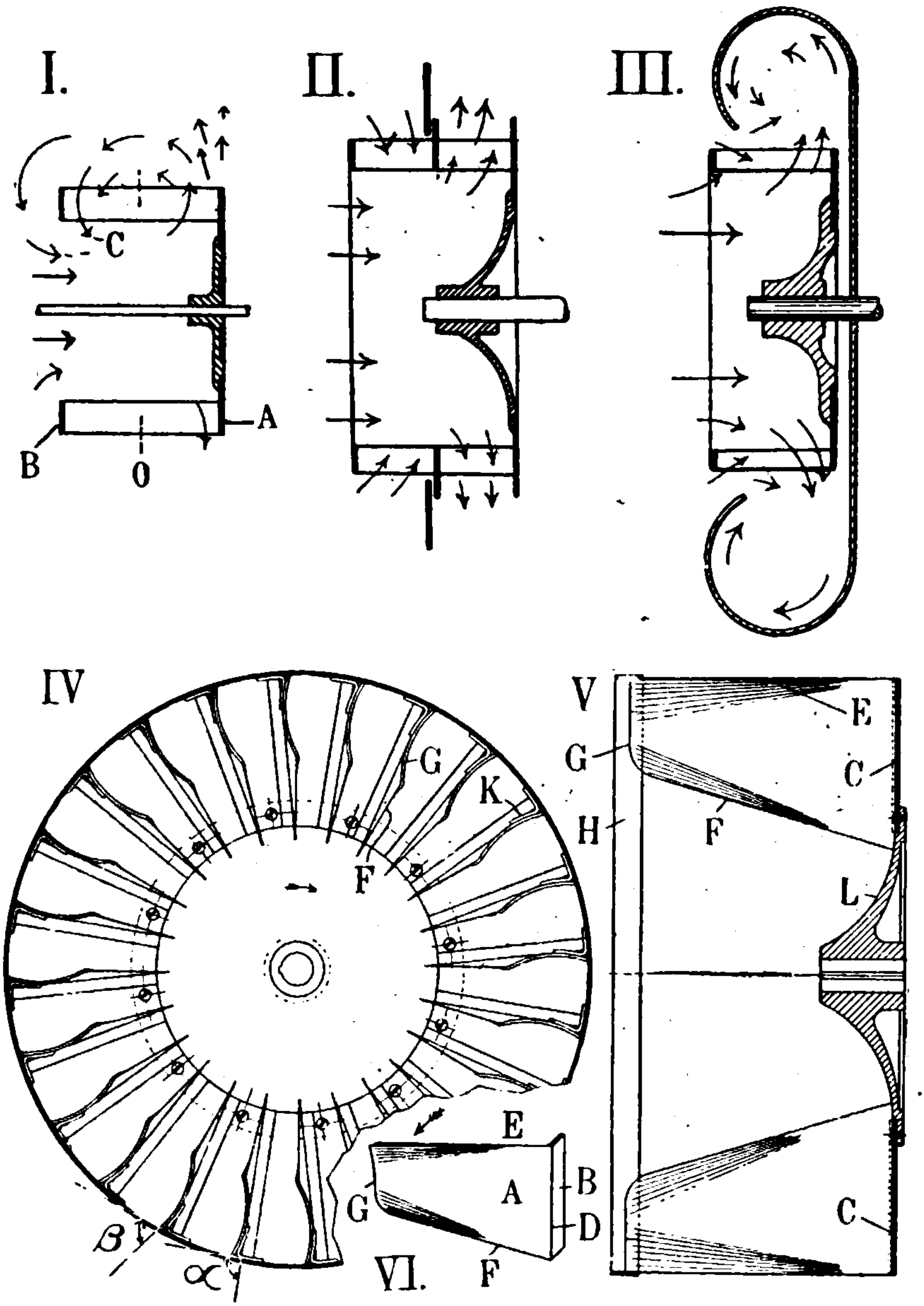

FIG. 138.—IMPROVED FANS.

I., Illustrating the peculiar cyclonic movement of air in an open drum type of fan. II., Partitions placed in and outside a fan to prevent the air movements shown in I. III., Design of casing for encased fan to compel the negative end of the wheel to draw from the suction region. IV., V., VI., Drum type of centrifugal fan in which the angle of blade at the inlet end is made less than at the disc end to cause a more evenly distributed discharge.

such inflow takes place it must be due to the reduction of pressure at *C* being so great that the pressure difference between the discharge and suction is large enough to force air against the centrifugal action of the blades. To counteract this he proposed to place a partition in the fan wheel to separate the positive from the negative end, and also to so place the fan with respect to the wall separating the suction and discharge regions of the outside air that the negative end would operate in the suction region as shown in fig. 138, II. He even goes so far as to propose such a design as shown in fig. 138, III., for a fan of the encased type. The curved part of the casing approaches the fan wheel in such a manner that it admits air from the suction side to the outside of the negative end of the wheel, whilst leaving only the positive end to the discharge. These difficulties show the fruits of departing—for the sake of cheapness of construction—from the more rational design of the Rateau fan, in which the discharge area of the fan wheel is limited to the positive end and is much reduced. It is evident that many designers have been well aware, if not of an actual inward flow, of a very unequal distribution of discharge along the wheel. For example, the blades of the Keith fan are deeper in a radial direction at the negative than at the positive end of the wheel, and it is claimed that the added centrifugal effect due to this increases the discharge at the so-called negative end. Again, Siemens and D. A. Hackett have patented a form making the angle at which the fluid is discharged from the vanes less at the intake than at the disc or positive end of the fan. This angle of discharge is defined as the angle between the vane and the tangent to the circumference of the wheel. Thus the angle β (fig. 138, IV.) at the intake end being less than α at the positive end, the air at the intake end is given a greater velocity of discharge. The blade consists of two parts, namely, *A* the blade proper and a lip *B* (fig. 138, VI.). The lip *B* is attached to the disc *C* (fig 138, V.) by point welding. The edge *D* of the part *A* (fig. 138, VI.) is straight, the outer and inner edges *E* and *F* are contained in one plane; the edge *G* appears concave when looking at the front side. The direction of rotation is indicated by

is riveted to it. When dealing with the rivets placed close to the periphery of the wheel a suitable tool, such as a holding-up hammer *E*, for holding the heads of the rivets is passed through the outer opening of the wheel. When dealing with the rivets situated towards the inlet of the wheel, however, the holding-up hammer cannot be conveniently held fast during the riveting operation, and it is therefore necessary to pass the hammer through the inlet, as shown in the first fig. But in building up a small delivery wheel the two discs are so close to one another as to render it impossible to pass the holding-up hammer through the inlet. This difficulty is overcome in a method of construction recently patented by Rateau, by forming the rear disc of a flat or slightly conical member *F* having a rearward extension *G* to form a hub. When the vanes have been riveted to the disc *F* and the outer disc *A* has been placed in position as before, access to the front of the disc *F* remains quite free. The holding-up hammer can then be introduced straight within the wheel for enabling the rivet heads to be held. For finishing the wheel after it has been mounted on the shaft a distance-piece *H* is added, in the form of a sleeve, having the required profile to constitute the bottom of the inlet. This sleeve also serves to maintain adjacent discs of successive wheels at the required distance apart in cases where a number of wheels are used on the same machine.

STEPPED VANES FOR CENTRIFUGAL FANS.

In order to give greater rigidity to stepped vanes or blades of centrifugal fans Alldays and Onions and G. F. Jenks, Birmingham, have formed on the back of the blades a rib or short longitudinal corrugation at the part where one plain part passes by a step into another plain part or rib. *A* is a ring or disc to which the inner ends of the blades *B* are fixed, as is usual, and *C* is the ring to which the outer ends of the blades are fixed. Each of the blades is provided with two shoulders or steps *D*, and rigidity is given to the blades at the stepped parts by ribs or short longitudinal corrugations. The vanes *B* may be arranged inclined to the disc *A* instead of at right angles to them as shown.

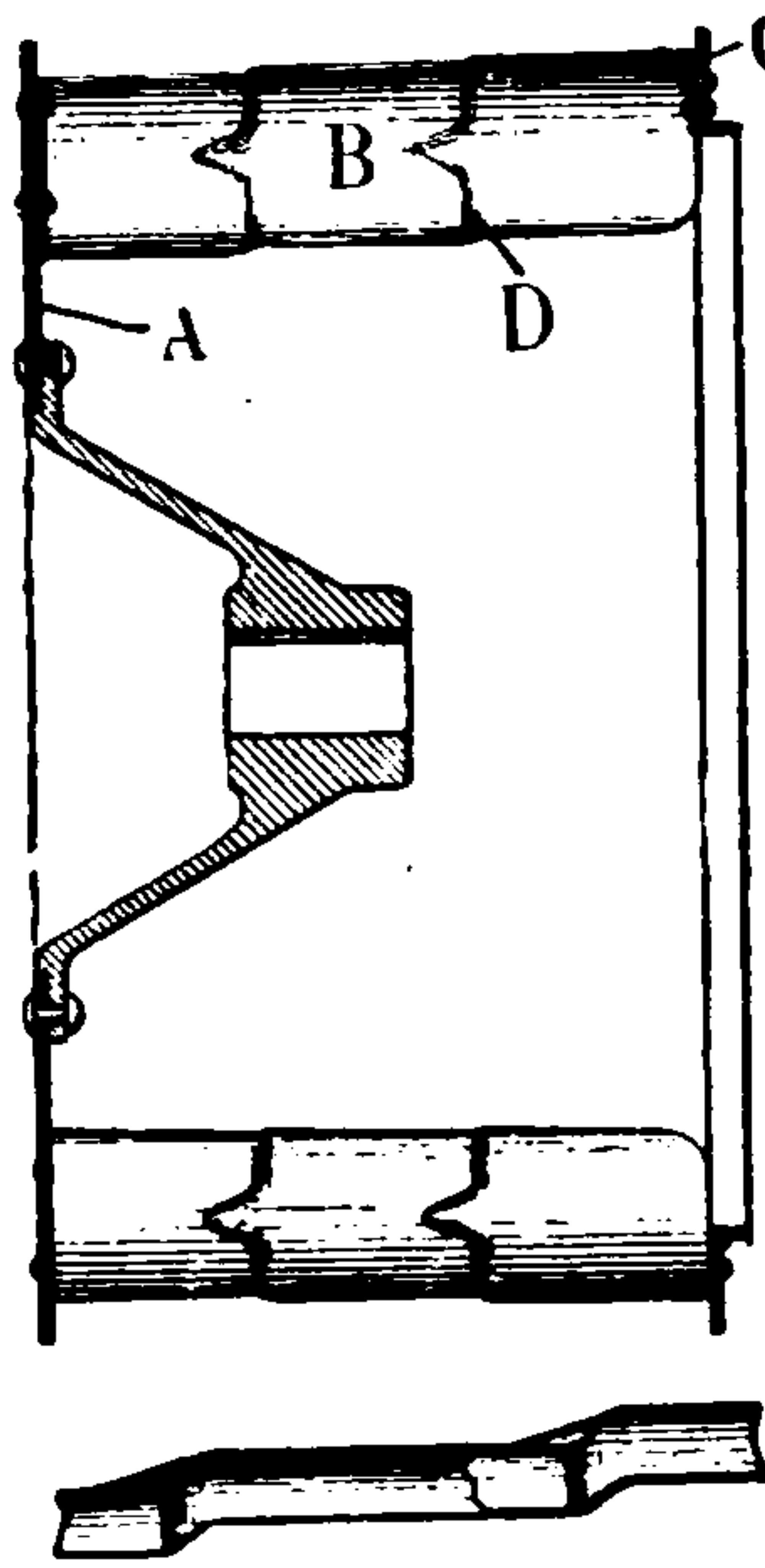

FIG. 140.—STEPPED BLADES FOR A CENTRIFUGAL FAN, WITH RIBS FOR STIFFENING THEM AT THE JOINTS.

ARRANGEMENT OF FANS.

It is extremely important in arranging fans that they should be fixed in such positions that the flow at discharge is interfered with as little as possible even at some distance from the periphery of the fan. In this work it is useful to bear in mind the molecular theory of gases and regard the air molecules as solid particles travelling in straight paths, until they impinge on some surface, when they rebound, following more or less the ordinary laws of impact. If they meet a surface normally they will be driven directly back again; of course, no surface is perfectly regular, so that a certain amount of scattering is caused by this and by the returning molecules colliding with each other. A molecule returning to the fan at high velocity will require more work to be done upon it to induce it to discharge again than was needed to discharge it in the first instance. The commotion caused by a multitude of such erratic molecules disturbs what may be called the "stream line" flow of the fluid. The proof that this is not mere theorizing may be found in the recently developed molecular air pump and in recently protected methods of arranging fans. For example, in torpedo-boat destroyers the fan for producing forced draught is usually uncased and is placed in one of the upper corners of the stokehold close to the deck and sides. The result is that the air, which is forced out radially all round the fan, strikes against the deck and sides and the beams supporting them. The fan should be partly or wholly encased, with two or more curved delivery outlets so arranged as to guide the outgoing air directly into the stokehold and away from any obstructions. Fig. 141, I., shows a partly-encased Allen fan with two delivery outlets O, arranged so as to avoid the overhead girders shown in section. Again, in the case of fans for the stokeholds of steamships, where several fans have to be placed side by side, it is found that some of the air delivered from one fan obstructs the delivery of air from an adjacent fan. To overcome this, W. H. and R. W. Allen have patented deflectors to guide the air in the desired path. Fig. 141, II., shows the deflector *D* for two adjacent fans rotating in opposite directions and the arrows indicate the paths taken by the air at discharge, the intake being, as in the previous case, at the eye of the fan. Without this deflector there is a tendency for the delivery of one fan near the thinnest part of the volute to enter the other fan. Fig. 141, III., shows the deflectors *D* used where two adjacent fans rotate in the same direction, the left-hand deflector preventing the molecules from bombarding the awkward faces of the casing and the right-hand deflecting the stream of its fan gradually into the delivery of the left-hand fan.

CONSTRUCTION OF CENTRIFUGAL FANS.

Much attention is apparently being paid by designers of multi-wheel centrifugal fans to the principle of construction in efforts to obtain good balance, easy and rapid repair, efficiency and strength. Fig. 142, I, and II, show a two-stage fan designed by P. W. Allday and T. H. Plummer.

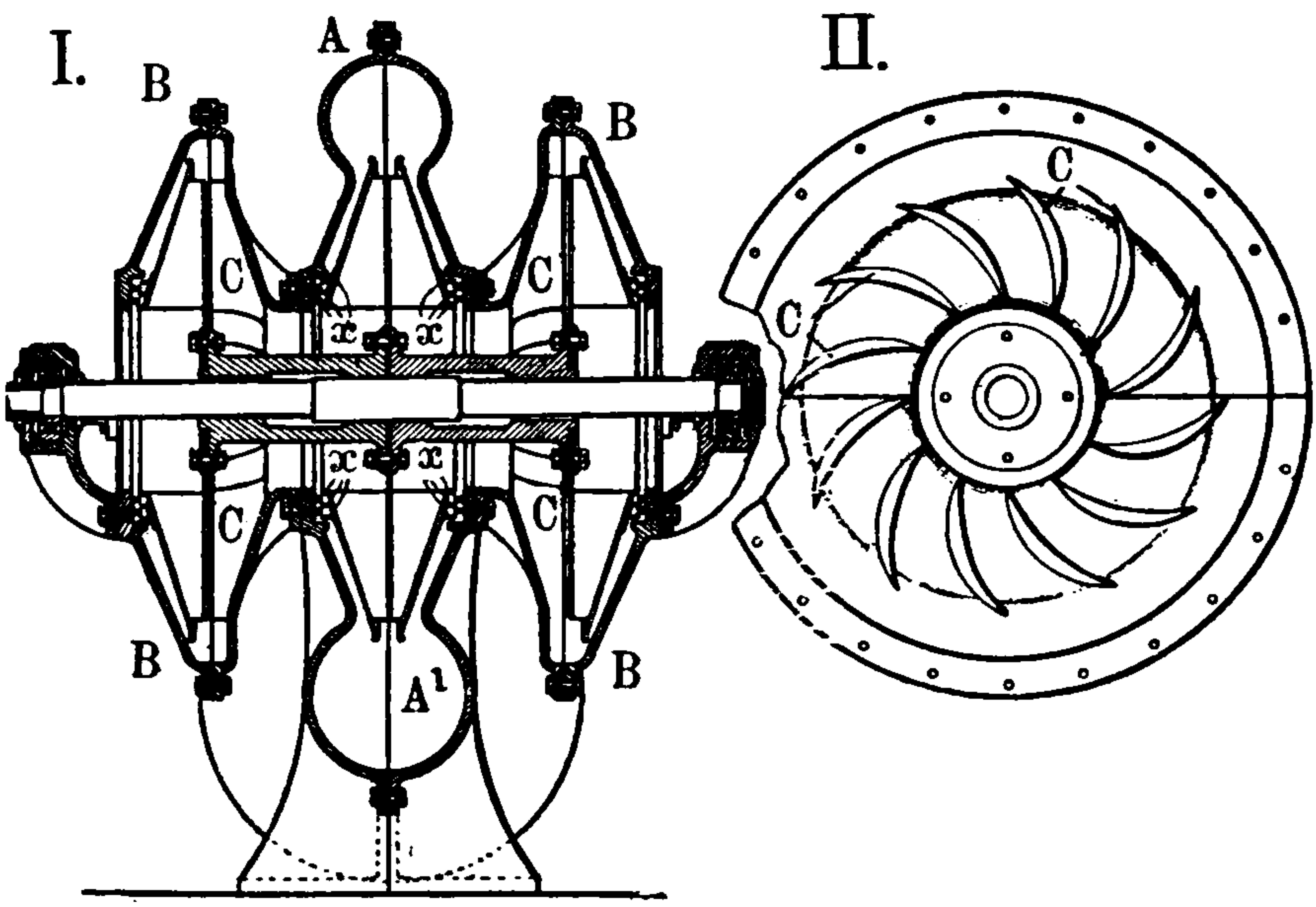

FIG. 142 —TWO-STAGE CENTRIFUGAL FAN.

I., Longway section. II., Elevation of inner half of end casing, showing guide vanes.

The middle casing *A* has axial inlets at opposite sides and resembles generally the casing of an ordinary double-inlet single fan; the end casings *B B* are connected by set screws or studs to the opposite halves of the casing *A*. The two end fans *B B* force the air into the axial inlets of *A*. The casing of the middle fan and base of the compound fan integral with it is divided into two symmetrical halves in a vertical plane perpendicular to the shaft, and the two halves flanged and bolted together. The runner of the middle fan casing receives axially the air compressed to a certain pressure by the end fans and discharges it into the tangential discharge opening A^1, opening preferably in the base. The casings of the two outer fans are made in two symmetrical halves, meeting together in a horizontal plane containing the axis of the shaft. The inner walls of the end casings have curved webs or vanes C (fig. 142, II.), which guide the air forced into the circumferential space into the axial eyes of the middle fan. The bearings for the runner shaft are carried by the lower halves of the outer casings, and the common hub of the three runners is made in two halves meeting together end to end at the middle of the inner fan. By this arrangement a convenient means of balancing the runner is obtained, as the one half can be partially rotated with respect to the other half when balance of the runners is being tested. When the best balance is ascertained, the three runners are secured together by the bolting of the two halves end to end. To prevent the return of the compressed air to the inlet openings of any fan, concentric flanges (shown at x of middle fan, fig. 142, I), which work between annular concentric flanges on the casings and constitute obstructions, are fixed to the runners. These flanges have no doubt been suggested by the labyrinth packing method of steam turbine practice. The fan is patented.

CENTRIFUGAL GAS PURIFIER.

Gases containing tar or dust in suspension may be purified by being passed through a centrifugal fan of special construction. By centrifugal action the solid particles are flung against the interior surface of the volute and collect there, whilst the purified gas passes to the discharge duct. When the fluid to be purified holds in suspension substances liable to liquefy by the action of heat, such as, for example, when treating tar-laden coal gas, a heating jacket for de-tarring is provided around the volute. Since the fluid leaves the impeller with a considerable velocity of whirl, it is evident that there is a loss of energy here unless some useful purpose can be served. In an ingenious arrangement by Rateau, illustrated, the gases, after passing over the fan blades B and through the volute or whirling chamber C, are made to pass over the blades E of an inward radial flow turbine before being discharged, the fixed guide blades of this turbine being shown at D. The turbine may be fixed on the same shaft as the fan, and its action reduces the power necessary to drive the fan, or it may be fixed on a separate shaft running at a different speed and used for some external purpose. The jacket G, through which steam or suitable hot gas or liquid is passed, is for de-tarring, the water of condensation in the casing being with-

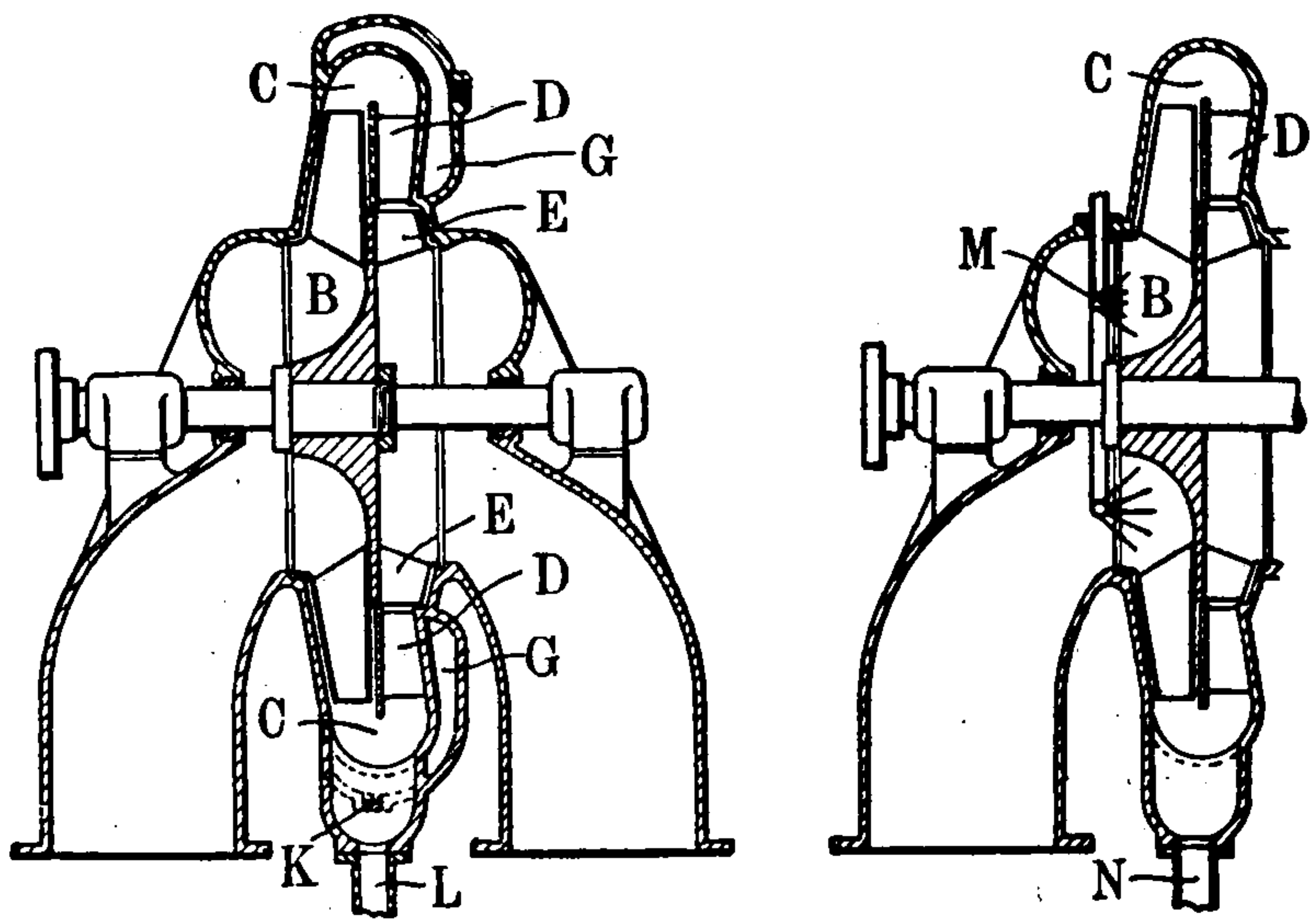

FIG. 143.—CENTRIFUGAL GAS PURIFIER, CONSISTING OF A FAN AND A TURBINE IN SERIES.

drawn through the outlets K and the melted tar passing out through an outlet L, which should be immersed in a water seal in order that the extraction shall take place automatically. When dust is to be extracted from the gas, water under pressure is injected into the fan inlet through tuyeres M, the particles of dust being thus caused to agglomerate and then to be separated from the gas in chamber C by centrifugal action, the dust-laden water passing out through the opening N. In some cases, instead of placing fan and recuperation turbine back to back as shown in fig. 143, they are separated and the whirling

B into the conduit *C*, the suction being produced by the fan *D*, the eye of which is connected through pipe *E* to *C*. After passing through fan *D* these gases are led by an enveloping flange *F* and curved webs and vanes *G* to the eye or inlet opening of the pressure fan *H*. The fumes or dust-laden air under pressure in the casing of the fan *H*, passes into the main *K*, from which they may be taken by branch pipes *L* to the hearths of the blacksmiths' forges *A*, or to furnaces or other appliances in which they may be used. The patentees are E. Allday and T. H. Plummer.

Induced Draught for Furnaces.

As an alternative to natural draught, the use of a fan in a duct forming a by-pass to the chimney is now common

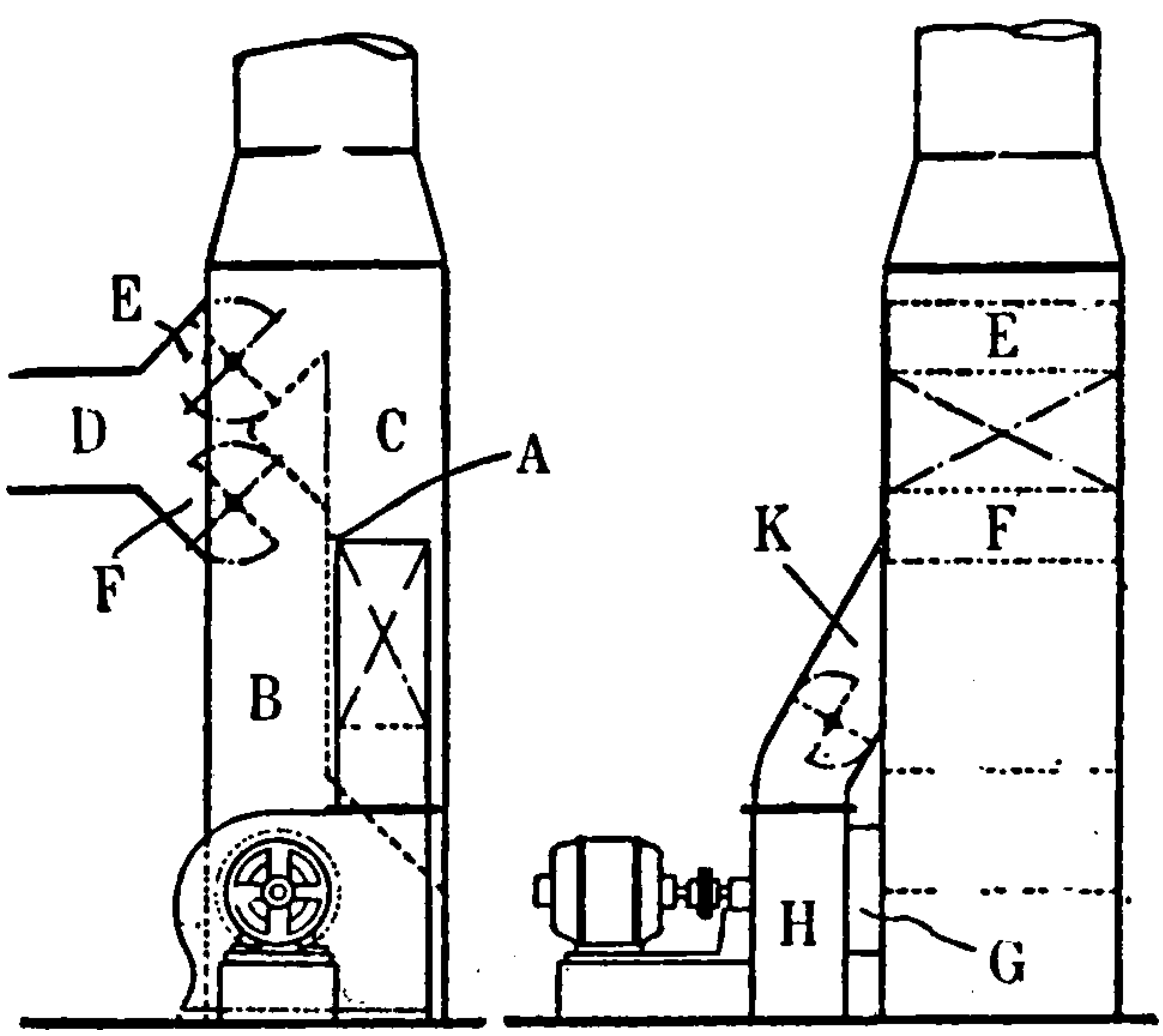

Fig. 145.—Combined Furnace Draught.

Combined induced and natural chimney draught, in which the furnace outlet is at some distance above ground level Dampers allow the gases to pass either directly up the chimney or be drawn by a fan.

enough, but where the outlet flue is at some distance above ground level the arrangement is unsuitable, because the length of the requisite ducts becomes excessive, and it is undesirable to mount the fan at an elevation. To meet these difficulties Babcock and Wilcox and R. Agar have patented the arrangement shown in fig. 145, which are elevations at right angles to each other. The outlet flue *D* from the furnace diverges as it meets the chimney, and two dampers *E* and *F* are placed here in branch ducts formed by the outlet flue walls in conjunction with angularly disposed walls connected to partition *A*. This partition *A* divides the chimney into two longitudinal chambers *B* and *C*. Chamber *B* is connected by a short duct *G* to the fan (*H*) inlet, the discharge from which passes through duct *K* to compartment *C* and thence up the chimney. When natural draught is desired damper *F* is closed and *E* opened, thus passing the gases directly up the chimney.

TURBINE-DRIVEN VENTILATING FAN.

Various combinations of fans and turbines have been proposed in which a single vane-wheel is used which draws air into a casing in the direction of its axis and is itself driven by an approximately tangential stream of actuating fluid. Both flat vanes and curved vanes of the bucket type have been used, and it is usual to arrange the nozzle supplying the actuating fluid and the conduit through which the fan delivers the stream of air in a straight line with each other. The association of the fan and turbine to form a single unit makes the apparatus simple, cheap, and easily portable. Fig. 146 illustrates a form with curved vanes. It will be noticed that the buckets for the actuating fluid are more numerous than the fan blades proper. The buckets are bent at their tips like those of a Pelton wheel. The air sucked into the casing is mixed with the actuating fluid, and the two fluids enter the conduit. A guide *G* is arranged in front of the nozzle to retain the jet in its path toward the vanes. The function of the jet is twofold : (1) to drive the wheel; (2) to create suction in the conduit. The Pelton buckets may extend across the entire width of the wheel, or across only part of it. An alternative proposal, which appears to be a good one, fixes them to a separate ring member, which is attached to the wheel; the latter may be a double wheel consisting of two parts side by side with the turbine ring between them. The inventor, however, has missed the one thing which undoubtedly would have made his invention an improvement on previons arrangements. To have separated the turbine ring of blades from the fan blades by a disc would have improved the working considerably. With Pelton buckets arranged in the manner proposed, the actuating fluid will, to a certain extent, be driven radially into the fan opposing the outward motion of the air. It will set up eddies in the fan wheel and generally disturb the natural lines of flow; and this will be very marked with a guide vane such as *G* directing the fluid partly in a radial direction. The

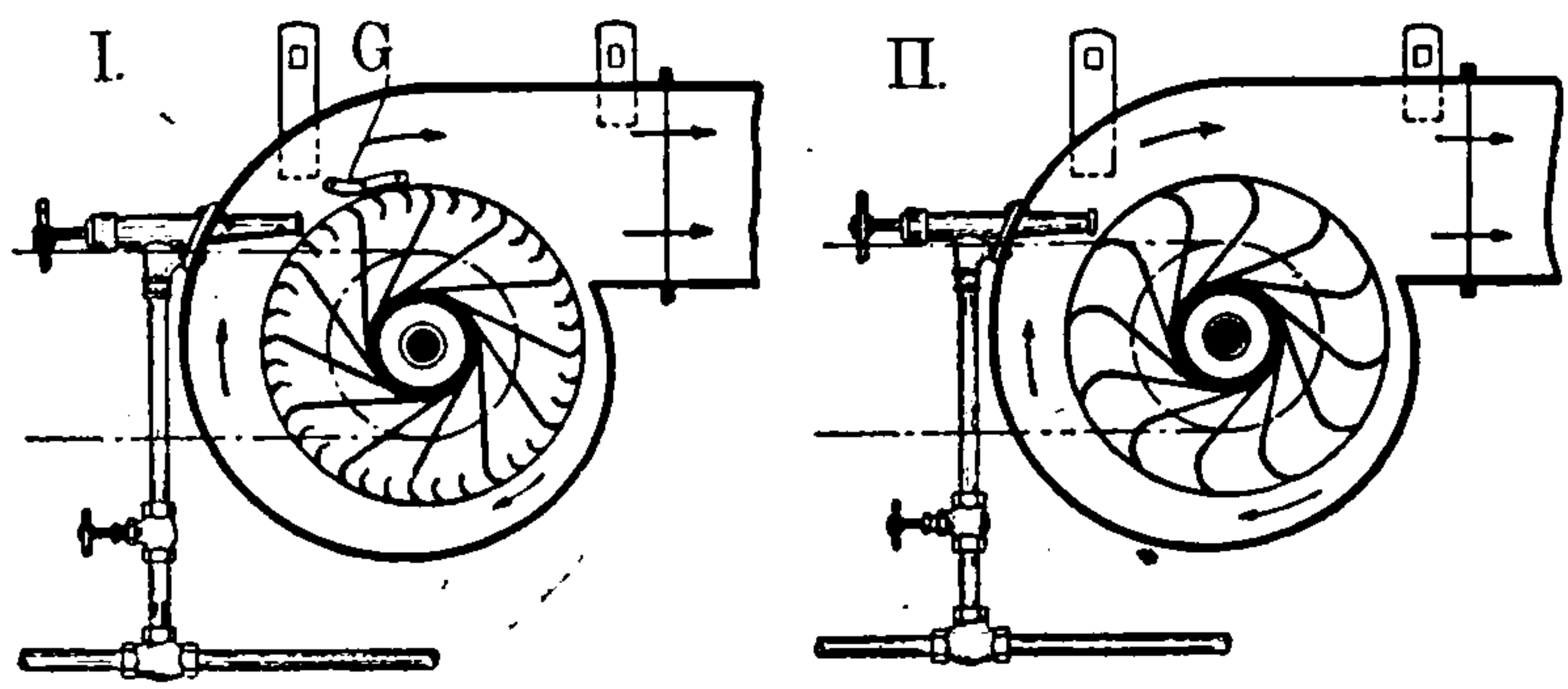

FIG. 146.—VENTILATING FAN DRIVEN BY A STREAM OF FLUID IMPINGING ON BUCKETS AT THE PERIPHERY OF THE FAN

efficiency would be much improved by attaching the turbine blades to the outside of the fan wheel disc so that the lines of flow from the fan would be undisturbed and the suction effect of the actuating jet very materially improved. Moreover, it will be noticed that the inventor, who describes himself as a foreman, writes of the vanes "being bent at their tips to form buckets like those of a Pelton wheel." The buckets are not fixed on the wheel in the manner usual in Pelton wheels. In the latter the stream of actuating fluid travels round the bucket in a plane more or less parallel to the axis of the wheel, and this arrangement would interfere less with the air streams of the fan than the one proposed in which the stream is directed in a radial direction round the bucket and inwards.

owing to failure of some part of the system it becomes necessary to pump gas of high temperature through the compressor. This temperature may be sufficiently great to melt babbitt out of the shaft bearings, thus necessitating a shut down. Often it is imperative to keep going at any cost, so that the compressor requires to be rugged in construction and capable of withstanding high temperatures. If repairs must be made, the compressor should be of a character to permit of them being easily and quickly made. Again, the various conduits carrying steam or hot gases expand unequally, and sometimes impose great stresses on the casings of the machine. To meet these difficulties the General Electric Co., of New York, has patented the form of construction illustrated in fig. 147. It is a radial vaned compressor driven by a steam turbine of the Curtis type, all completely enclosed in a casing. The compressor portion 1 of the casing (I.) is divided in a vertical plane into right- and left-hand sections to facilitate casting and machining, and in particular the surfacing of the annular walls of the gas-conveying passage immediately surrounding the impeller 2. The joint between the casing parts is carefully machined, and after the machine is completed will not ordinarily be disturbed. This portion of the casing is also divided horizontally in the plane of the shaft into upper and lower sections. The various sections are flanged and bolted together. The lower half of the casing instead of being supported on a foundation is mounted directly on the gas inlet conduit 3 and the gas discharge conduit 4 (III.), both of which lie in a plane perpendicular to the shaft, are inclined to each other, and are capable of yielding laterally by a slight amount when for any reason the steam inlet pipe or the steam exhaust conduit 5 expands and contracts. The conduits 3, 4, 5, should be long enough to permit these movements to take place without unduly straining their supports or the joints between them and the casing.

Impeller.—In some cases the body of the impeller 6 is made integral with the shaft in order to insure rigidity. The vanes are radial and the outer portions extend between the side plates 7, which project into the annular discharge chamber 8. Gas is admitted to both sides of the impeller.

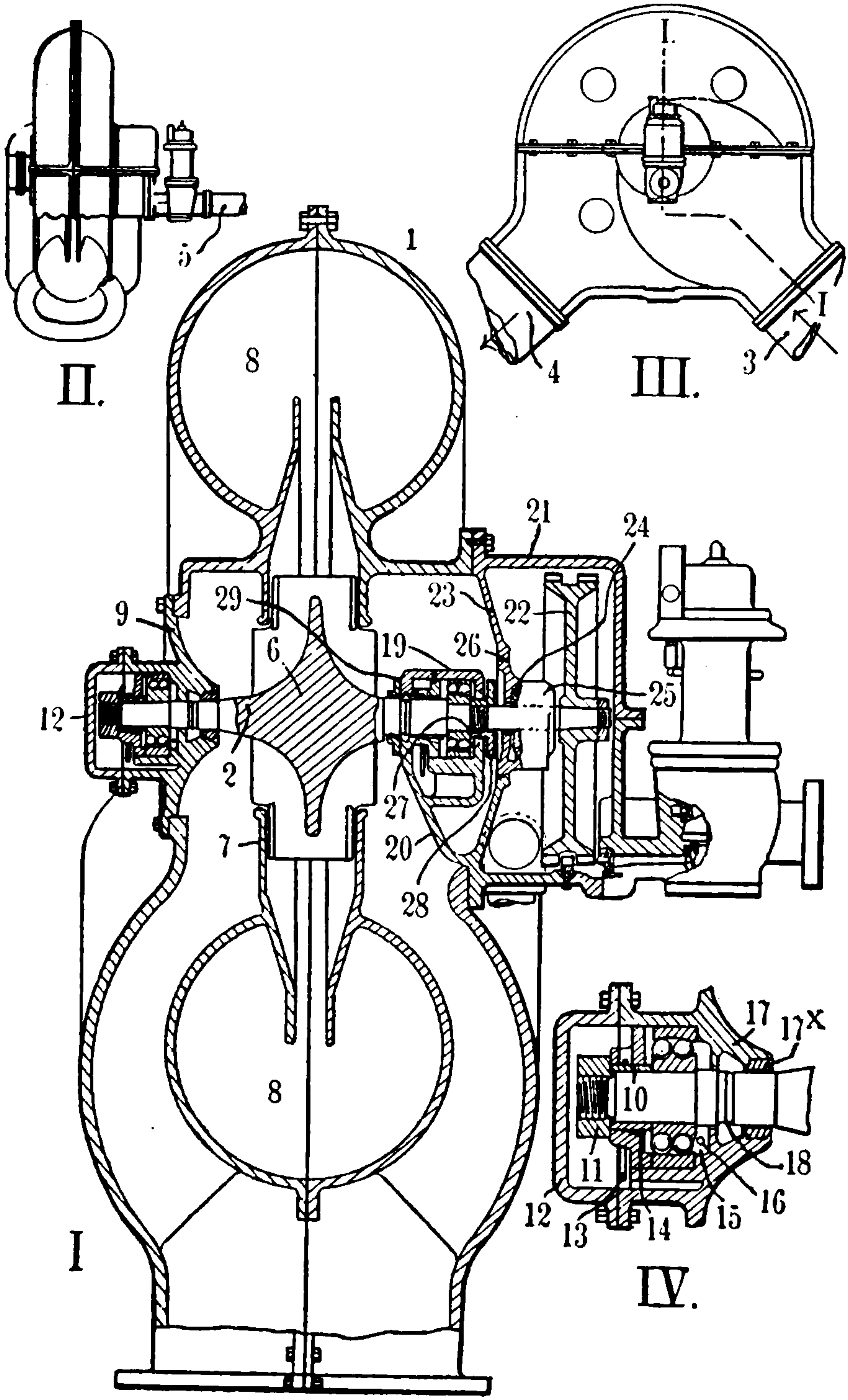

FIG. 147.—TURBO COMPRESSOR FOR GAS.

I. Vertical sec io

Bearings.—The left-hand side of the compressor portion of the casing is provided with an opening of suitable size carefully finished and a shouldered and chambered head 9 secured to it. The outer member of the bearing is seated on a shoulder formed in the head. Part 10 (IV.) indicates a spacing device which engages a shoulder on the shaft and is clamped by the nut 11. This serves to position the rotating element of the machine. Access may be had to the nut by removing the cap 12, which is secured pressure-tight on the head. Surrounding the spacer is an oil ring 13, which conveys oil from the chamber below to the spacer, from which it flows through the downwardly inclined passage 14 to the bearing. The lubricant is maintained in the bearing at a suitable level by the dam 15, the excess being returned to the oil chamber by the passage 16. Between the chamber in the head and the impeller chamber is a partition 17, and supported by this is a grooved metal packing 17^{x} of such character that it will not be injured by relatively high temperatures. This packing does not normally make contact with the shaft, so there is no wear at this point. It acts to prevent hot gases from freely entering the bearing chamber, where it would soon destroy the lubricant. In a small chamber between the packing and bearing is an oil thrower 18 of ordinary construction. The right-hand end of the shaft is supported by a ball or roller bearing 19 of the same character as the one on the left. It is supported by a web 20, which forms an integral part of the turbine portion 21 of the casing. This portion encloses the rotor 22 of the steam turbine and is seated pressure-tight on the compressor portion of the casing. The shaft does not extend through the outer walls of the casing at either end, so that there is no opportunity for gas or steam to escape from the casing, or for air to enter. In order to prevent steam from the turbine portion entering the compressor portion a packing 24 is provided composed of a number of rings. Carbon is satisfactory for this purpose, since it will take a high polish as the shaft rotates, will not cut the shaft and will stand relatively high temperatures. This packing is carried by a housing 25 that has a peripheral tongue 26 fitting pressure-tight in a groove in

The blades are usually made radial in order to avoid the bending stresses due to curvature. In order to allow the shaft to "whip" slightly without weakening the attachments, special means have to be taken to insure a small amount of free sliding action at certain points. The various rotating parts must be accurately balanced.

As the air leaves the impellers at high velocity it is necessary to gradually convert the kinetic energy to pressure energy—as first suggested by Professor Osborne Reynolds—by passing the air through gradually expanding passages Vanes that are radial at outlet are not quite as mechanically efficient as backward-turned vanes, and this supplies a further reason for the use of divergent passages at outlet. Radial vanes which are continued straight to the inlet would be very inefficient, as it is necessary to curve them at the inlet to suit the relative velocity of the air to the vane at this point. Some makers divide the vane into two parts, making the larger external part straight and radial—a form that can be readily machined —and attaching to this a curved but radial portion at the inlet, which, being small, can also be readily machined. An example of recent construction in high-pressure compressors is illustrated in figs. 148, 148A, I.-VII. Referring to I. and II., the air suction is shown at 2; after passing over the first two impellers 13 and 14 (I.), the air is discharged through the divergent passages 11 (II.) to the space 17, where it divides, part passing along channel 18 to the right eye of the next wheel, and the remainder along passage 19 to the left eye of the same wheel.

Construction of Casing.—The casing is divided into an upper and lower part, bolted together in an axial plane. The head 1 (I.) is integral with the inlet conduit 2. Closely surrounding the shaft is an annular member 3 that forms a continuation of the inlet conduit. This member is divided into upper and lower parts, and permits the removal of the head 1 without disturbing the shaft coupling which is enclosed in the guard or housing 4. The diameter of the bore of the head is greater than the diameter of the coupling. The guard is made of sheet metal formed into a cylinder and fastened by strips 5. Within the casing are diaphragms 6 seated in suitable shoulders 7 formed on the inside of the casing. The diaphragms are provided with shoulders 8 to receive the plates 9 and 10, between which are secured the discharge vanes 11.

Construction of Impellers.—The impeller is provided with vanes 13 and 14 (I.), which are strictly radial and integral with the body 12. The impeller is divided into two parts (see V.), which are finished on their opposite faces and shouldered (VI.) to insure proper fitting. In assembling these parts, the female member of the impeller is expanded by heating and then allowed to shrink on the shoulder projection of the other member. The parts are then united by set screws. In order to direct the air into the channels or passages between the impeller vanes at the proper angle, curved inlet or guide vanes 15 and 16 are provided, which rotate with the impeller. These vanes are carried by a cast metal support 21 (III.), which is slipped over the hub portion of the impeller body. Each of these vanes is provided with a pair of lugs 22, which engage opposite sides of an impeller (II. and III.) and prevent the outer ends from moving out of their proper position. These small inlet vanes, though curved, are strictly radial, so that there is no tendency to fold over under centrifugal action.

Method of Fixing Impeller to Shaft.—The support 21 carrying the inlet vanes is slipped over the hub 23 of the impeller, and is secured in place by a locking ring 24 (III.). This ring is provided with an internal shoulder 25 that is seated in a groove formed in the hub. The ring is shrunk on; the shoulder not being deeper than $\frac{6}{1000}$ in. or $\frac{7}{1000}$ in. Formed in the bore of the impeller are grooves 26 (V.), each containing a split ring, preferably of hard non-rusting metal of high crushing strength. The grooves are slightly dovetailed and the rings are rolled into the grooves under high pressure which forces the metal under the overhanging walls of the groove. After the rings are forced in place, they are machined to a diameter about 0·012 in. smaller than the diameter of the shaft. The impellers are then forced endwise on the shaft under pressure and prevented from rotating by the key 27 and rings in 26. By these means the metal of the impeller is

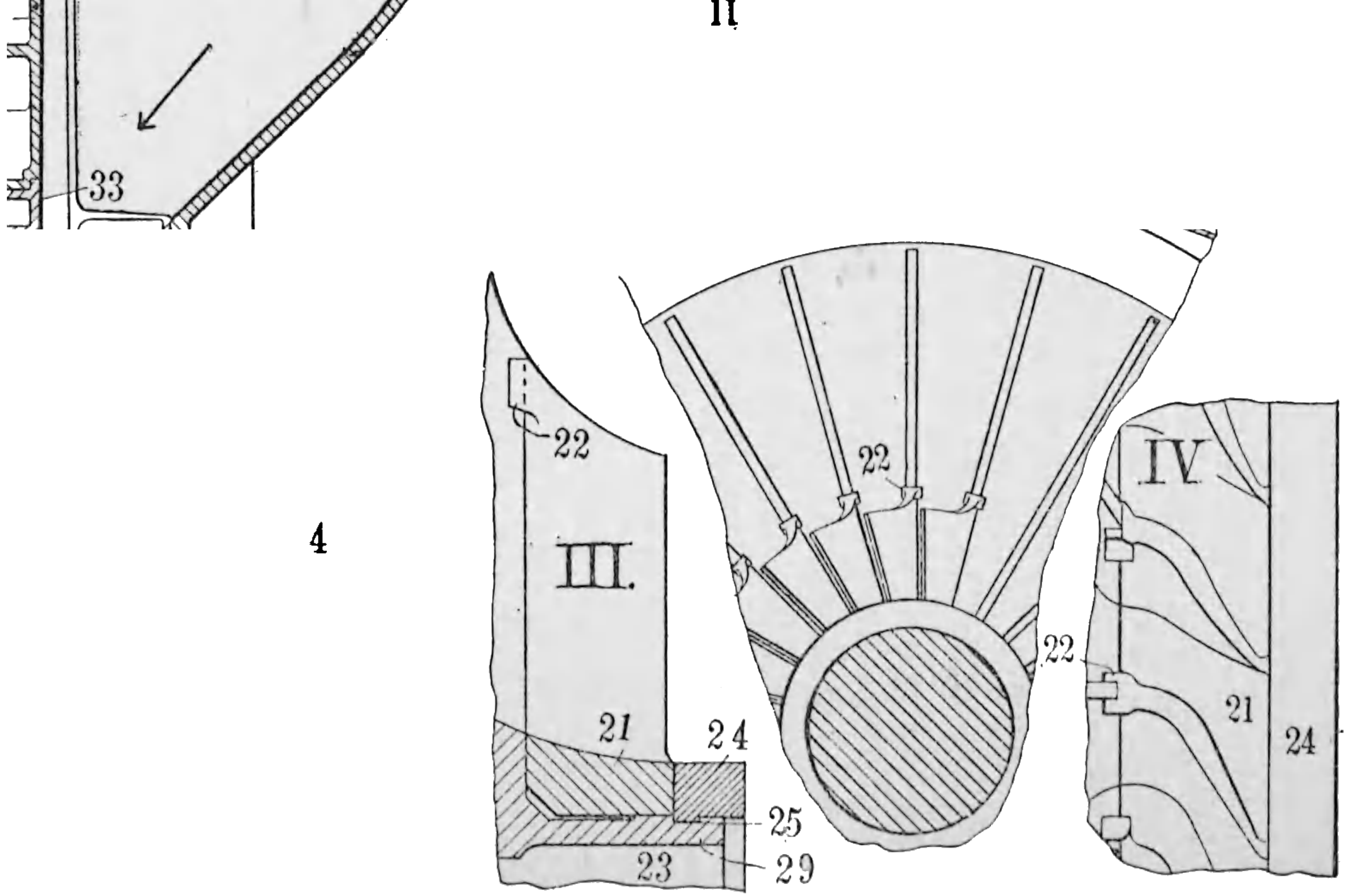

FIG. 148.—HIGH-PRESSURE CENTRIFUGAL COMPRESSOR.

I., Partial axial section of a multi-stage centrifugal air compressor. II., Partial transverse section, showing relation of impeller vanes to the stationary discharge vanes. III., Rotary guide vane, side elevation. IV., Rotary guide vane, plan view.

put under radial compression, which varies from the centre outwards, the idea being that when the centrifugal tensile stresses are induced the resultant stress will be so small that very little increase in diameter ensues. Mounted on the shaft are sleeves 28—made of some non-rusting metal—which fit inside the ends of the hubs of the impeller (V.). The bore of the hub is 0·002 in. in diameter less than the external diameter of the sleeve, and so forms a compression fit. Surrounding the end of the hub is the shrunk ring 24 (III.), which, in addition to preventing the inlet vanes from moving axially, serves to hold down the stretching effect of the end of the hub—a $\frac{2}{1000}$ in. to $\frac{4}{1000}$ in. fit is found best for this. In order to permit a little axial movement between the impeller and its shaft, and so avoid the impactive stresses which would otherwise come on the impeller when the shaft whipped, it is necessary to have approximately a sliding fit between the impeller and shaft. To this end the hub is slotted at 29 (V.), so that, in effect, a ring 29× is formed, which is connected to the hub proper. By reason of the slot 29 (VII.) the ring 29× is relieved of tangential stresses, so that its diameter will not be perceptibly increased by centrifugal forces in the hub, but will retain as nearly as possible its original diameter and thereby form a support with a constant degree of tightness on the shaft.

Construction of Discharge Vanes.—The discharge vanes 11 (I. and II.) are made of heavy sheet stock and are U shaped in cross-section. The down-turned sides of each vane are united by bolts, screws, or rivets to the side plates 9 and 10. The side portions of one vane follow at their edges the shape of the adjacent vane, and preferably rest on that vane so as to form good passages and also give support. The vanes are stiffened at their inlet ends by rivets 20. The spaces 8 between the walls of the impeller may be used for circulating cooling water.

Packing.—In order to space the impellers on the shaft, cylindrical members 30 are provided that have internal shoulders which engage the sleeves 28 (V.) and so prevent axial movement. The periphery of the cylindrical member 30 is provided with projections 31 that extend between corresponding projections on cylinder 32—bolted to the diaphragm—and thus form a labyrinth packing to prevent the passage of air from one chamber to another along the shaft.

Balancing.—Although each of the impellers is carefully balanced either statically or dynamically, it sometimes happens that when they are all assembled on the shaft the machine is out of balance as a whole, and in order to permit of balancing without taking off the top part of the casing, openings filled by removable plugs as 34 (V.) and 33 (I.) are provided in the ends of the casing. By taking out these plugs, access can be had to the body of the impeller and metal added or taken away as desired. This method of construction is patented by the General Electric Co., of New York.

CENTRIFUGAL AIR COMPRESSORS.

Surging Effects.—The surging or pulsating of the air in a centrifugal air compressor may be due to two quite distinct causes. The cause which is more easy to comprehend and to guard against is that of working against a fluctuating head, such as a blast furnace, in which the resistance occasioned by the charges of fuel, ore, and flux varies considerably from time to time. In such cases, when the flow is retarded, the air is liable to accumulate in the delivery conduit of the compressor until its pressure rises so high as to cause a backward rush in the conduit, to be succeeded by a forward rush to restore the lost pressure. The surging sometimes becomes so violent as to endanger the apparatus. In order to reduce these surging effects the New York General Electric Co.[7] place a sheet metal nozzle *A* (fig. 149, I.) in the conduit, preferably the intake pipe of the compressor. This permits the air to flow freely into the machine, but any reverse flow is throttled at the nozzle. The smaller end of the nozzle points toward the compressor, and its diameter is from one-third to one-quarter that of the conduit. In the flange *B* are several ports *C*, which are controlled by an annular damper *D* having ports which are adapted to be brought in or out of registration

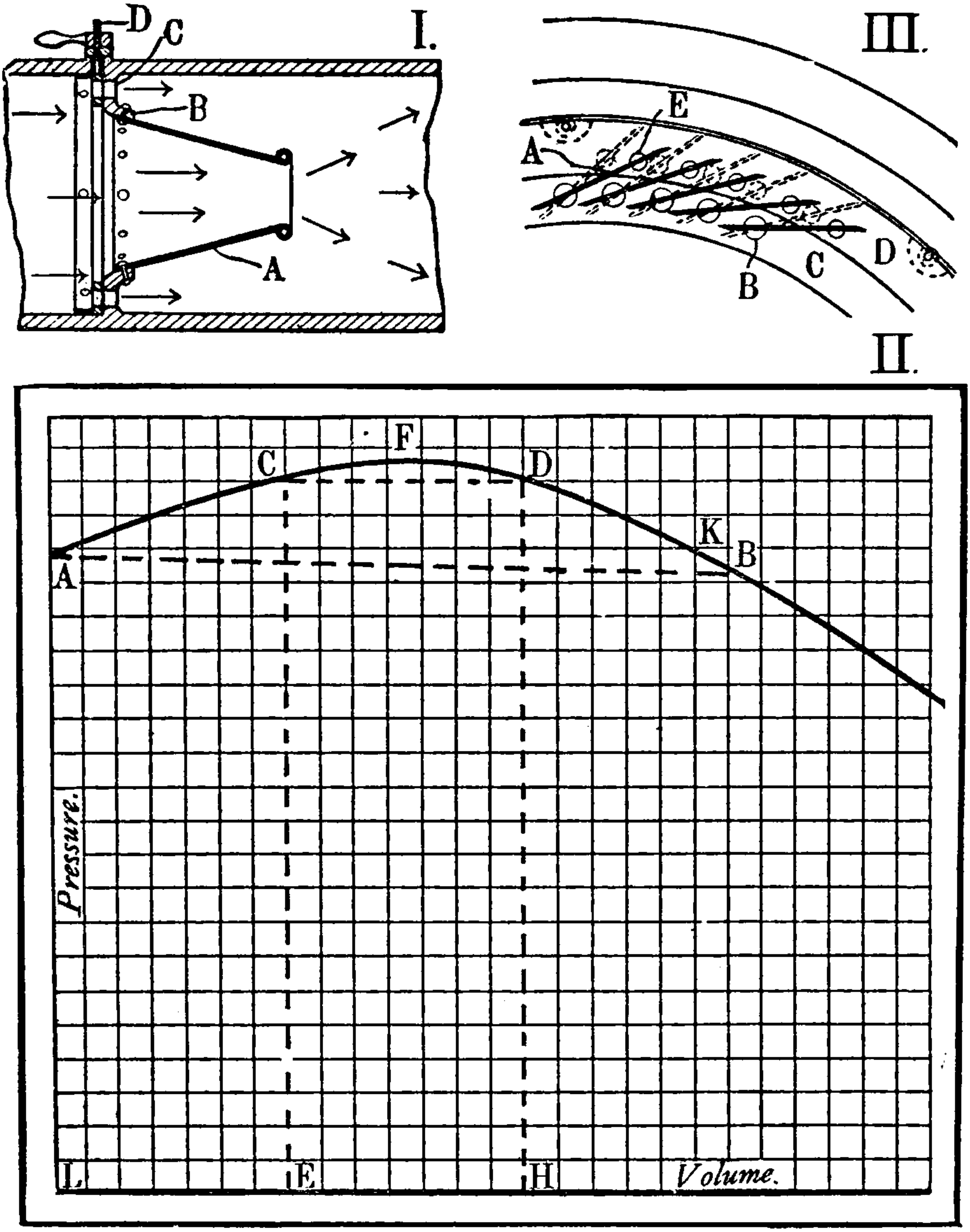

FIG. 149.—SURGING IN AIR COMPRESSORS.

I., Sheet metal nozzle in intake to prevent surging. II., Diagram of pressure and volume relation. III., Weathercock guide vanes self-adjusting to the direction of flow.

with the ports in the flange. These ports serve to regulate the flow somewhat.

The second cause is neither so easy to understand nor to

will cut the curve in two points *C* and *D* corresponding to two quite different flows *L E* and *L H* respectively. When the compressor is working, therefore, at a certain speed against this particular pressure the flow of fluid through it may be either *L E* or *L H*. The flow is ambiguous, and consequently unstable. It may change from one to the other at its own pleasure, and this fickleness leads to surgings or pulsations when working on this part of the load. In seeking for a remedy it is best to study the reason of this up-and-down variation of the flow. At a certain speed of running the blade of a compressor or turbine pump gives a definite amount of energy to the working fluid, partly as pressure and partly as kinetic energy. Now, the kinetic energy to be of use must be largely converted into pressure energy by passing through the diverging discharge vanes. The vanes are set at an angle to suit a given direction of discharge from the impeller—that is, to suit a given flow, so that, if the flow is not suitable, a certain amount of kinetic energy is lost in eddies. The fluid molecules are thrown hither and thither and career around each other in little circular paths. This frittering away of energy may be as serious at too low as at too high a discharge, hence the shape of the pressure volume curve. Obviously the correct remedy would be to vary either the outlet angle of the impeller or the inlet angle of the discharge vanes to suit each flow. The maximum pressure would then occur at or near the zero discharge line; it would probably be a little distance to the left unless the inlet angle of the impeller were made to suit a low discharge. T. H. C. Homersham has proposed[25] floating discharge vanes which automatically adjust themselves to the direction of flow as a weathercock takes the direction of the wind. In the illustration (fig. 149, III.) the straight guide blades *A* are mounted by means of a pin *B* in an annulus *C*, which annulus is mounted in the frame. About this annulus a floating ring or annulus *D* is mounted, which bears by means of antifriction devices in the framework. This floating annulus carries small pivotal pins *E* having saw cuts in their heads, into which the edges of the guide blades enter. The floating annulus

D is free to move in a circumferential direction. A special device is arranged so that the attendant may hold the guide vanes in any desired position or may assure himself of the freedom of the floating ring.

A method[26] proposed by the General Electric Co., of New York, to abolish this instability is to throttle the flow of air preferably at the intake to the compressor. The instability is due to the upward trend of the pressure curve on light loads, and the idea is to introduce such resistance —and, therefore, loss of pressure—at the inlet that the increase in pressure due to the better conditions in the compressor on increased flow is slightly more than neutralised by this loss. The curve then takes a downward direction, as indicated by the dotted line *A B* (fig. 149, II.). At no pressure under these conditions are there two possible flows, and therefore there is no instability. It is plain, however, that the remedy is a somewhat wasteful one. The method of throttling the intake is shown in fig. 149A, IV. The intake *A* of the compressor *B* is provided with a throttle valve *C*, mounted on a stem, which has at its lower end a dashpot. The valve is a disc located in an inverted frusto-conical section of the intake, so that as it rises and falls it enlarges and diminishes the annular opening between its periphery and the wall of the section. The stem passes up through the top of the elbow in the intake pipe and carries a cam *D*, whose back is parallel with the stem and bears against light guide rollers. A lever having two arms at right angles is pivoted to the upper end of the standard. The vertical arm carries a roller bearing on the cam. A weight *W* is adjustable along the horizontal arm. With an increase in volume of air passing through the intake the valve rises and the cam tilts the lever, throwing the arm with the load upward, and thereby shortening the effective distance of the weight *W* from the pivot. It is hoped by some such arrangement to be able to produce just the amount of throttling necessary for any given flow. The loss of pressure necessary varies from zero at the outset to a maximum, and then reduces again to zero. The vertical part of the acting face of the cam allows the valve to lift against practically no load at the commencement, and the throttling action is therefore small. With increased flow the curved part of the cam face comes into action gradually, increasing the resistance to the motion of the valve under the bombardment of the inrushing air molecules. The throttling action is thus increased, for it must be remembered that the increased flow requires a larger valve opening, and the loss of pressure is proportional to the square of the increased velocity due to any deficiency in valve opening. As the flow increases further the gradually diminishing leverage of the weight W once again reduces the resistance to the upward motion of the valve.

The exact shape of the cam face is, however, a matter for experimental determination. It cannot be obtained theoretically, as suggested by the patentees in their specification, without much experimental data on the resistance and throttling action of such valves. It is obvious, for instance, that if the amount of throttling necessary for point C (fig. 149, II.) has been determined and applied, the air then supplied to the compressor is in a more or less disturbed state and the characteristic pressure-volume curve for air in this condition would be somewhat different to the curve $A\ C\ F$. So that for any small increase in the flow $\delta\ v$ the increase in pressure $\delta\ p$ might be quite different to that obtained from a tangent to the curve $A\ C\ F$.

COMPRESSORS WITH INCLINED STRAIGHT BLADES.

Straight blades for compressors are cheaper and easier to manufacture than curved ones, but radial straight blades cannot make the proper angle at inlet. It is necessary to incline the blades in order to secure the proper inlet angle. Straight blades have already been used in compressors, but in this case the faces of the rotor destined to receive the blades had to be straight. Curved rotor faces, however, have the advantage over straight faces of forming a better guide for the air and of enabling a more favourable shape to be given to the rotor to counteract strains, and more particularly to avoid the bending strains which are liable to occur in rotors with one-sided admission. By the rotation of a straight line about an axis, without changing its

the wheel—shown in plan by $k\ a$. Draw $A\ B$ —fig. 151, IV.—equal in length to $k\ a$—fig. 151, III.—to show the thickness of the rotor. Make $A\ L$ and $B\ S$ perpendicular to $A\ B$—fig. 151, IV.—equal respectively to $k\ a^1$ and $k\ b^1$ —fig. 151, III.; then L and S are two points on the rotor profile. To obtain intermediate points, mark the positions of any three points 1, 2, 3, along $k\ a$—fig. 151,

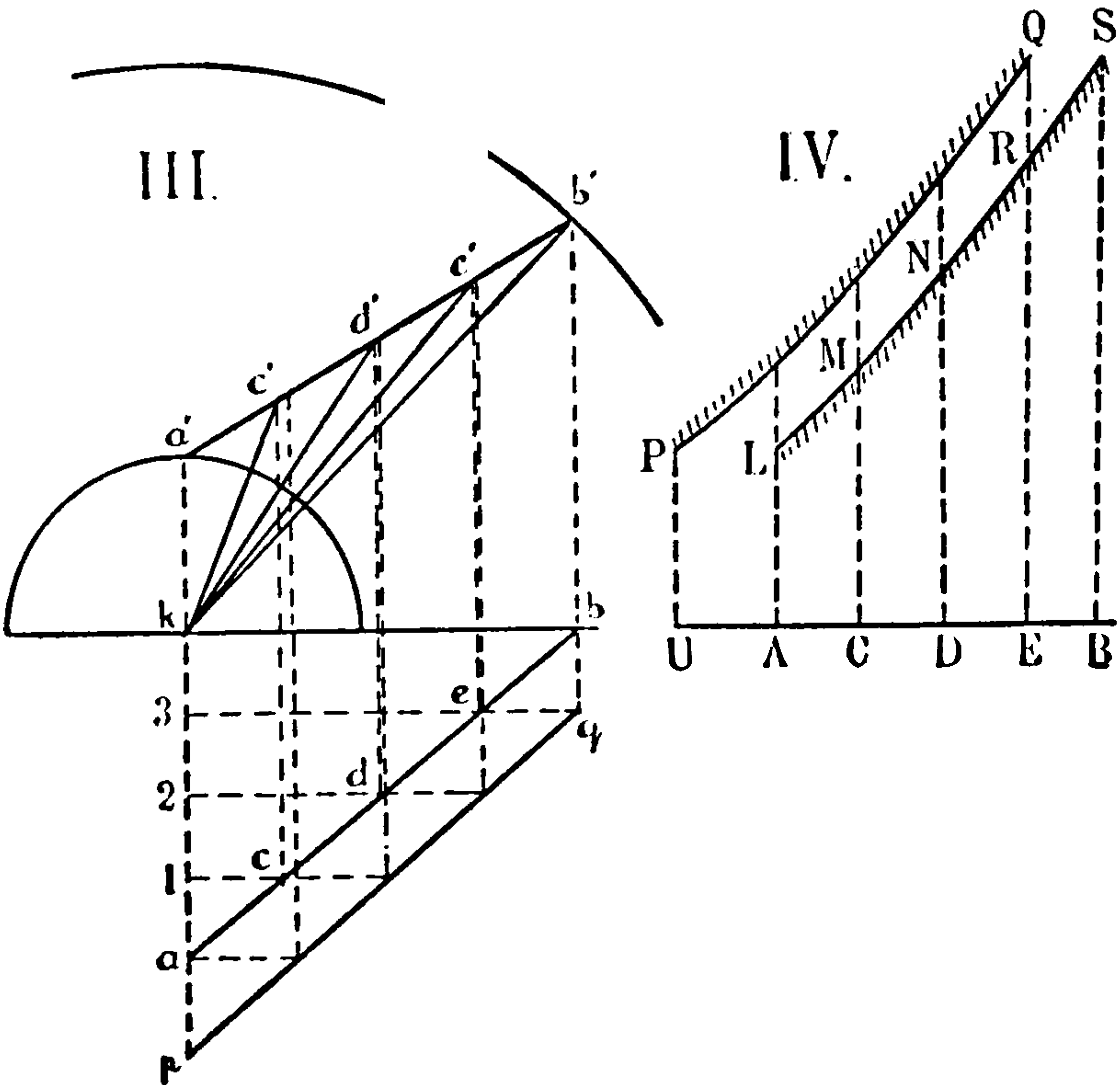

FIG. 151.—METHOD OF DRAWING THE HYPERBOIDAL FACE OF THE ROTOR OF A COMPRESSOR WITH STRAIGHT BLADES.

III.—and draw horizontal lines $1c$, $2d$, $3e$, to meet $a\ b$ in the points c, d, e respectively. Draw vertical lines cc^1, dd^1, ee^1 to meet $a^1\ b^1$ in the points c^1, d^1, e^1 respectively. In fig. 151, IV. make $A\ C$, $C\ D$, $D\ E$, $E\ B$ equal respectively to $a1$, 12, 23, $3k$, of fig. 151, III. Draw vertical lines through these points, and make $C\ M$, $D\ N$, $E\ R$ equal respectively to $k\ c^1$, $k\ d^1$, and $k\ e^1$ of fig. 151, III. Then the curve $S\ R\ N\ M\ L$ through these points is the one required. Note that the lengths $k\ a^1$, $k\ c^1$, $k\ d^1$, $k\ e^1$, $k\ b^1$ are the true lengths of lines drawn through the several points of the generating line, and perpendicular to the axis of revolution; consequently, they represent the radii of the curve at these points.

The curve $P\ Q$ for the rear face of the rotor may be obtained in an exactly similar manner. Let the axial width of blade $Q\ S$—fig. 151, IV.—at outlet be less than the width $P\ L$ at inlet. Make $k3$ or $b\ g$—fig. 151, III.—in plan equal to $Q\ S$—fig. 151, IV.—and $p\ a$—fig. 151, III.—equal to $V\ A$—fig. 151, IV. Assuming that the inlet and outlet circles of revolution are equal to those of the front face, and that the planes of the blades are parallel to the axis of revolution, then $a^1\ b^1$ is the elevation of $p\ q$ as well as $a\ b$—fig. 151, III.—and the shape is obtained by projection from $a^1\ b^1$ as in the last case. The vertical lines from the intermediate points—fig. 151, III.—lie a little to the right of the verticals cc^1, dd^1, ee^1. It will be noticed that inclined straight blades cut the inner and outer circles in such a manner as to make the angle at inlet Y—fig. 150, II.—between the blade and the radius greater than the same angle at outlet. For compressors running at high speeds it is necessary that this angle at inlet should be large, but it is not at all necessary that the outlet angle should be large; in fact, in many cases the blade is made radial at outlet. It has already been demonstrated that in fans and compressors the curving back of the blade at outlet, regarded as so essential to efficiency in centrifugal pumps, secures only a trivial increase in mechanical efficiency at the expense of a real loss of manometric efficiency.

NOTES

[1] *Phil. Mag.*, vol. xvii., p. 389.
[2] "Hydraulics," by W. M. Wallace, p. 194.
[3] *Journ. Amer. Soc. Eng.*, Sept., 1912, p. 1354.
[4] "Physics: Experimental and Theoretical," by R. H. Jude.
[5] See paper on Centrifugal Fans, by Heenan and Gilbert, in the Proc. Instit. of Civil Engineers, vol. cxxii., fig. 8, plate 5.
[6] See fig. 41 further on.
[7] British Patent, 1913—1,382.
[8] Proceedings of Institution of Civil Engineers, vol. cxxiii., part 1.
[9] The table giving these experiments will be found in Appendix II. of Heenan and Gilbert's paper.
[10] Proceedings of the Institution of Civil Engineers, vol. cxxii.
[11] "Les Ventilateurs de Mines," *Revue Universelle des Mines*, vol. xx., 1892.
[12] From "The Theory of the Centrifugal Pump and Fan," N.E.C. Inst. of Engineers and Shipbuilders, vol. xiv.
[13] Proceedings of the Institution of Civil Engineers, vol. xlvii.
[14] Proceedings of the Institution of Civil Engineers, vol. liii.
[15] Bodmer's "Hydraulic Motors, Turbines," etc.
[16] Proc. Inst. Mining Engineers, vol. xl.
[17] "Du Calcul des Ventilateurs," by M. Lelong, Ingénieur des Constructions Navales.
[18] From "Die Gebläse," by von Ihering.
[19] Annales des Pontes et Chausées, 1888.
[20] See vol. xliii., Minutes of the Proceedings of the Institute of Mining Engineers.
[21] "Bulletin de la Société de l'Industrie Minérale."
[22] See Rapport au Congrès International de Mécanique Appliqué.
[23] It is obvious that the centre does less work per pound than the outer part, since $a\,v \div g$ is the work per pound done by the fan.
[24] "Hydraulics," by W. M. Wallace, p. 134.
[25] British Patent, 1913—24,233.
[26] British Patent, 1913—1,182.

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