CENTRIFUGAL FANS.

A THEORETICAL AND PRACTICAL TREATISE

ON

Fans for Moving Air In Large Quantities At Comparatively Low Pressures.

BY

J. H. KINEALY, M. Am. Soc. M. E.

Formerly Professor of Mechanical Engineering at Washington University, St. Louis; Past-president American Society of Heating and Ventilating Engineers; Past-president Engineers' Club of St. Louis; Member Society of Arts, England; Member Boston Society of Civil Engineers, etc.

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Fully Illustrated and with Numerous Tables for Facilitating Calculations in Regard to Fans.

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PREFACE.

The matter set forth in this book was included in a series of articles written for the *Engineering Review* at the request of its editor. The favorable attention which the articles attracted lead the author to believe that there was a real demand for a book treating in a theoretical as well as a practical way of centrifugal fans. Hence the articles which appeared in the *Engineering Review* have been thoroughly revised, added to, and made as complete as possible for presentation to the public.

The subject of centrifugal fans has been dealt with by other writers, but the author has felt that the method of treatment has been in some instances entirely too theoretical and in other instances too intensely practical. Some writers have devoted their energies to evolving long complicated mathematical formulas, which were of little or no use to the engineer as they always contained constants to which the writers seemed unable to assign specific values. As a species of mental gymnastics the treatises were good, but as an aid to an engineer in laying out a system of heating and ventilating they were of little use.

Other writers have made certain statements in regard to fans without giving any explanation or reason for their statements. They have assumed the position of knowing whereof they speak, without giving any evidence that they really possess the knowledge they assume to have.

The author has tried to avoid being only theoretical, but at the same time has been careful to give a reason for every statement. Where mathematics has been needed, it has been used, and used as freely as the occasion demanded. No pains have been spared to explain fully the entire theory of centrifugal fans and to show clearly the way in which every formula and rule is derived.

In the theoretical matter and the derivation of the various formulas the author has had in mind the student and the mathematician, and has endeavored to show how the theory of the vortex applies to and is used every day in one branch of engineering. The development of the theory of vortexes as applied to fans is the result of several years of study and work, and to those who wish to indulge in mental gymnastics of a mathematical kind the author recommends a study of the vortex with an inward flow. This is the kind of a vortex that one sees when water flows out of a circular bowl through a hole in the middle of the bottom.

For the practicing engineer tables have been prepared, and they have been arranged in the way which the experience of the author in designing heating and ventilating plants has shown to be most convenient. The tables are full and complete, and it is hoped that few typographical errors have crept in. The calculations were carefully made and checked, and the proofs have been carefully read, but even in spite of care the author fears that here and there some error will be found, and he will be grateful to those who call his attention to any which they may find.

The author hopes that the book will find as much favor in the eyes of the public as did the articles in the *Engineering Review*, and trusts that it will be of interest to those who desire to study the theory of centrifugal fans, and of aid and value to those engineers who have to plan or erect works where fans must be used.

J. H. KINEALY.

St. Louis, Mo., January, 1905

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CHAPTER I.

Flow of Air. When air or any similar gas flows from a vessel into a space where the pressure is not more than five per cent. less than the pressure in the vessel, the velocity per minute at which the air or gas ought to flow if there be no friction is

$$(1) \qquad v = 60\sqrt{2gh}$$

v is the velocity in feet per minute; h is the height in feet of a column of the air or gas, having a base of one square foot, whose weight is equal to the difference between the pressures per square foot inside and outside the vessel; and g is the number 32.2, which represents the acceleration in feet per second of a freely falling body.

In all work connected with fans or blowers it is customary to express the pressures of air in ounces per square inch. Therefore if d be the density or weight in pounds of one cubic foot of the air under consideration; and p be the pressure in ounces per square inch equivalent to the height h of the column of air. we have

$$d\,h = \frac{144\,p}{16}$$

$$h = \frac{9\,p}{d}$$

If we put this expression for h in (1) and also put for g its value, we have

$$(2)\quad v = 60\sqrt{\frac{64.4 \times 9\,p}{d}} = 1443.6\sqrt{\frac{p}{d}}$$

From (2) it is seen that when air flows from a vessel into a space in which the pressure is p ounces less than in the vessel, the velocity in feet per minute depends not only upon the difference in pressures, p, but also upon the density of the air. Further, it is seen that the greater the density the less the velocity, and the less the density the greater the velocity. The density of air varies with the temperature: the higher the temperature the less the density. And hence (2) shows that air at a high temperature will have a greater velocity of flow, for the same value of p, than air at a lower temperature. The density of dry air at 32° is 0.0807, and this value in (2) gives as the expression for the velocity of flow of air at 32°,

$$v = 5081 \sqrt{p}$$

The density of dry air at 100° is 0.0710, and with this value of d the expression for v is

$$v = 5417 \sqrt{p}$$

The writer, in his practice, usually assumes that

$$v = 5200 \sqrt{p} \tag{3}$$

which corresponds to a value of 0.0770 for d, which is the density of air at a temperature of 56°. This expression while rigidly true only for a temerature of 56° and for dry air is true enough for all practical purposes for ordinary air and any temperature between 32° and 100°, and it has the further advantage of being simple and easily remembered.

Sometimes pressures of air in fan work are expressed in inches of water instead of ounces. And as one ounce per square inch is equivalent to a head of 1.73 inches, to convert pressures in ounces per square inches to inches of water, it is necessary to multiply the pressure in ounces by 1.73. Also, to convert inches of water to ounces per square inch, divide the inches of water by 1.73.

Table I. gives the velocity in feet per minute of air as calculated by equation (3) for various pressures in ounces per square inch.

TABLE I.

Velocities in feet per minute.

p Pressure in ounces per sq. in.	v Velocity in feet per minute	p Pressure in ounces per sq. in.	v Velocity in feet per minute	p Pressure in ounces per sq. in.	v Velocity in feet per minute
0.0	0	0.75	4500	3.0	9000
0.1	1645	0.8	4650	3.5	9720
0.2	2325	0.9	4940	4.0	10400
0.25	2600	1.0	5200	5.0	11600
0.3	2850	1.25	5810	6.0	12700
0.4	3290	1.5	6370	7.0	13800
0.5	3680	1.75	6880	8.0	14700
0.6	4030	2.0	7350	9.0	15600
0.7	4350	2.5	8220	10.0	16400

Volume of Air Flowing. It is evident that if the area in square feet of the opening through which an air or gas flows be multiplied by the velocity in feet per minute, the quotient ought to be the volume of the air which flows out per minute. Unfortunately, however, experiments show that when a gas flows through an opening in the side of a vessel the quantity that actually flows out per minute is less than the theoretical amount which the size of the opening and the difference in pressures would indicate ought to flow. This reduction in the quantity is due to two causes: First it is found that there is a reduction in the

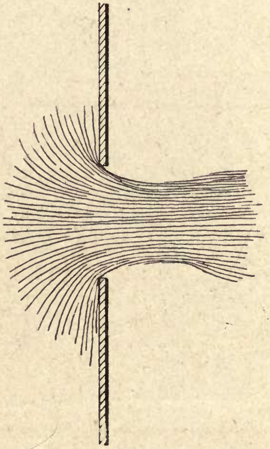


FIG. 1.

area of the cross-section of the stream flowing out, so that its area of cross-section where the velocity is greatest is not equal to the area of the opening; and second, the actual velocity is somewhat less than the theoretical velocity because of friction.

The reduction of the area of the cross-section of the stream of issuing gas and also the reduction of the velocity depends upon the kind of outlet through which the gas flows. If the outlet is in a thin wall the shape of the issuing stream is as

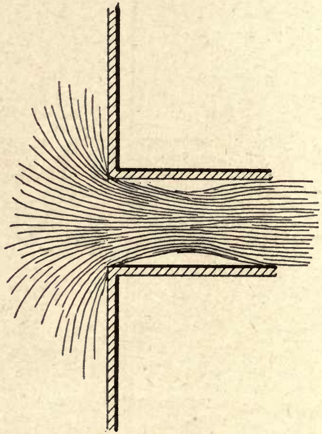


FIG. 2.

shown in Fig. 1, where the converging lines indicate the flow of the issuing gas, before it reaches the outlet. When the orifice is a short projecting tube, the form of the stream is as shown in Fig. 2. If the tube projects *inside* of the side of the vessel the form of the stream is as shown in Fig. 3. And if the tube is conical shaped the flow is as shown in Fig. 4. In all cases the velocity of the flow is greatest at the smallest section which is always of less area than the mouth of the outlet. The

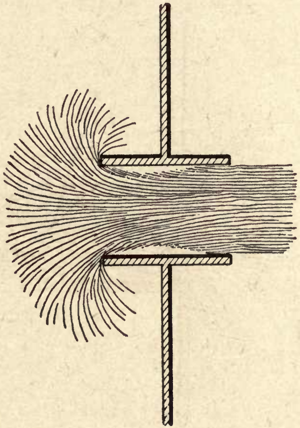


FIG. 3.

greatest velocity is usually almost but not quite equal to the theoretical velocity due to the pressure making the gas flow out.

If then a is the area in square feet of an opening through which air or gas flows, and v is the velocity in feet per minute of the flow as calculated by equation (3), the quantity Q in cubic feet which will flow out per minute is

$$Q = c\,a\,v \tag{4}$$

c is called the *coefficient of discharge*, and is the ratio of the actual discharge to the theoretical.

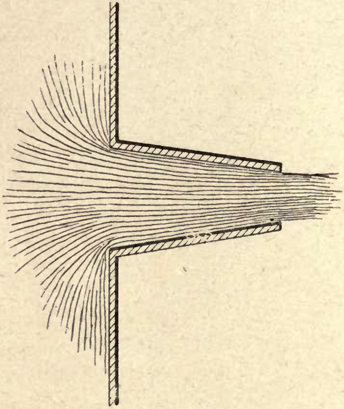


FIG. 4.

It is exceedingly difficult to give an exact value to c except for those particular conditions and size of openings which have been used in making experiments, and even for those conditions which are almost alike different experimenters have obtained different results. In some cases c is due almost wholly to a reduction of velocity, and in others to the fact that the cross-section of the stream of discharge where it has its maximum velocity is much less than the actual area of the opening in the vessel. The fact is that in choosing the proper value of c to be used in any particluar case it is necessary to exercise a great deal of that particular sense or judgment which goes with and makes the successful engineer.

For a circular opening in a thin plate or a plate whose thickness is small compared to the diameter of the opening, c may be taken as 0.62.

For a circular tube projecting from the flat side of a vessel, when the length of the tube is between 1.5 and 2.5 times the diameter of the tube, c may be taken as 0.82.

For a short tube which projects into the vessel instead of out of it, c may be taken as 0.50.

For a conical shaped tube whose length is about 5.5 the smaller diameter, whose large diameter is about 1.3 the small diameter, when the flow is from the larger diameter towards the smaller, and when the area of the smaller end of the tube is used in calculating the discharge, c may be taken as 0.94.

For a square opening c may be taken the same as for a circular opening whose diameter is equal to the side of the square.

The value of c for a rectangular opening which is not a square depends upon the relation of the two sides, and it is impossible to give any short rule for determining it.

It is evident that it makes no difference as to the actual result of our calculation whether we say that c is due to a reduction of velocity as well as to a reduction of the area of the stream flowing out, or whether we assume simply for the purposes of calculation that c is due entirely to a reduction of velocity. The result of the calculation will be exactly the same in either case. When the pressure under which the air flows, and the area of the outlet are both known, all that is necessary is to determine the velocity of the flow by equation (3) or Table I. and multiply it by the area of the opening and the proper value of c.

EXAMPLE:—How many cubic feet of air will flow per minute through a circular opening 2 feet in diameter in the side of an iron tank in which the pressure is 2 ounces?

Here a, the area of the outlet, is 3.14 square feet. And from Table I. it is seen that v, the velocity of the air flowing out, is 7350 feet per minute. The thickness of the metal of the tank will, of course, be small as compared to the diameter of the opening in this case and therefore we take c as 0.62.

These values of a, v, and c in equation (4) give

$$Q = c\,a\,v = 0.62 \times 3.14 \times 7350 = 14300.$$

Pressure Necessary for a Required Velocity. In what has been said before it has been assumed that the pressure is known and that the velocity or the quantity of flow is to be found. In many cases, however, the velocity or the quantity is given and it is required to find the necessary pressure.

From equation (3) we get the following equation for determining the pressure in ounces per square inch necessary to give a velocity or flow of v feet per minute.

$$p = \left[\frac{v}{5200}\right]^2 \qquad (5)$$

When we have given the quantity of air Q which is to flow per minute from the vessel and also the kind of outlet we can determine v by the following equation which comes from (4),

$$v = \frac{Q}{ca} \qquad (6)$$

Then knowing v we use equation (5) to find p or we may substitute in (5) the value of v as given by (6) and get at once the following expression for p, without determining v at all.

$$p = \left[\frac{Q}{5200ca}\right]^2 \qquad (7)$$

If the coefficient of discharge were equal to one, that is, if there were no reduction in either the velocity or the area of the outlet, the quantity actually discharged would be equal to the theoretical quantity, and the expression for p would be

$$(8) \qquad p = \left[\frac{Q}{5200a}\right]^2$$

If we divide equation (7) by equation (8) we get the ratio of the theoretical pressure to give a required velocity or to produce a given flow, to the actual pressure necessary. This ratio may be designated F, and its value is

$$(9) \qquad F = \left[\frac{Q}{5200ac}\right]^2 \div \left[\frac{Q}{5200a}\right]^2$$

$$= \frac{1}{c^2}$$

Example:—What is the theoretical and the actual pressure required in order to discharge 8000 cubic feet of air per minute through a short circular tube whose area of cross-section is 1 square foot?

Here Q is 8000; a is 1, and c for a short tube, may be taken as 0.82. From (8) we have the theoretical pressure is

$$p = \left[\frac{8000}{5200 \times 1}\right]^2 = 2.37$$

And from (7) we have the actual pressure is

$$p = \left[\frac{8000}{5200 \times 1 \times 0.82}\right]^2 = 3.52$$

That is, the theoretical pressure required is 2.37 ounces, while the actual pressure is 3.52 ounces. The difference is what is lost by friction and the cohesion of the particles of the air to one another.

The ratio of the actual pressure to the theoretical pressure required is 3.52 divided by 2.37 or 1.48.

We could have determined this ratio by (9), without having determined the theoretical and actual pressures, as follows,

$$F = \frac{1}{c^2} = \frac{1}{0.82^2} = \frac{1}{0.674} = 1.48$$

In other words, the actual pressure is nearly 50 per cent. greater than the theoretical pressure, and this 50 per cent. is simply lost in overcoming friction. There is always this kind of a loss when air or gas flows from one vessel into another, or from one space into another, and the actual amount of the loss depends upon the orifice through which the air or gas flows. When air flows into the inlet of a fan or from a chamber into a pipe there is this same kind of a loss which must be determined according to the size and shape of the orifice, and it must be allowed for when determining the actual pressure required for the discharge of a given quantity of air per minute.

CHAPTER II.

Vortex. If a cylindrical vessel such as is shown in Fig. 5 be partly filled with water or any similar liquid to the line *A B*, and then the water be made to rotate, it will be found that the surface of the water will be depressed at the center and raised at the circumference as shown in Fig. 5. The circumference will be raised the same height above the original surface that the center is depressed below it, and the surface will have the shape of a paraboloid with its vertex at *E* instead of the horizontal plane *A B* as it was before the rotation. The surface of the paraboloid is represented in Fig. 5 by the parabola *C E D*, and if this parabola were revolved about its vertical axis it would generate the surface which the water has when revolved.

As long as the water is kept rotating with the same velocity the surface will preserve the shape of the same paraboloid *C E D*, but the vertex *E* will *always* for all velocities of rotation be the same distance *below* the original surface $A\ B$ that the circumference $C\ D$ is above it. The distance

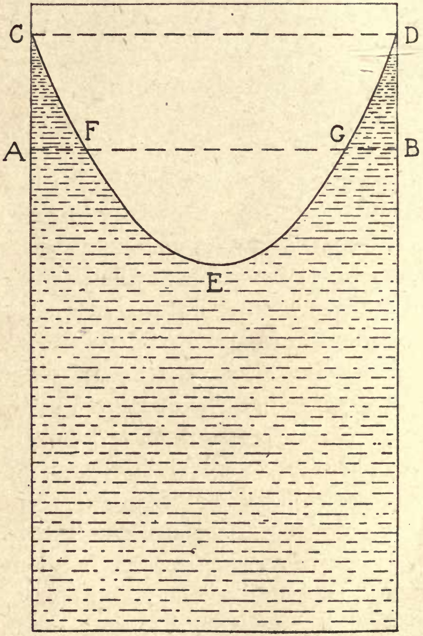


FIG. 5.

that E is below the surface $A\ B$ and the distance that the circumference $C\ D$ is above it depend upon the velocity of rotation of the water, but they are equal to one another for all velocities of rotation.

This making the surface of a liquid assume the shape of a paraboloid by rotation constitutes what is called a *vortex*.

It can be shown by mathematics, which would be out of place here, that the distance of the circumference $C D$ above the vertex E, is equal to the height, corresponding to h in equation (1), necessary to give to the particles of the liquid at the circumference the velocity which they have. And every point on the surface of the rotating liquid is above the lowest point E a distance equal to the height necessary to give the liquid at that point the velocity which it has. The velocity of rotation of the particles of liquid at the center is zero and it increases from the center towards the circumference where it is greatest. After the vortex is once formed, there is no radial flow of the particles of water, they have only a motion of rotation about the center. That is to say, the particles do not flow either towards or away from the center after the vortex is formed, they simply revolve about it. During the formation of the vortex, there is, of course, a flowing of the particles away from the center towards the circumference, but this flow ceases as soon as the vortex is formed.

The pressure per square inch on the bottom of the vessel is changed by the rotation of the water and the formation of the vortex. Before the formation of the vortex the pressure per square inch was uniform over the whole bottom and was that due to the depth from the surface of the liquid to the bottom; but after the vortex was formed the pressures on different points of the bottom became different from what they were before in exactly the same proportion that the depths at the different points were changed by the formation of the vortex. At the center, directly below *E*, the pressure becomes less than before the formation of the vortex by an amount corresponding to the distance of *E* below the original surface *A B*. And at the circumference the pressure becomes greater than before the formation of the vortex by an amount corresponding to the distance of *C D* above *A B*. Before the formation of the vortex the pressure at the circumference is exactly the same as that at the center; but after the vortex is formed, the pressure at the circumference is greater than that at the center by an amount corresponding to the distance of *C D* above *E*. But the distance of *C D* above *E* is, as has been stated, equal to the height corresponding to the velocity of the particles at the circumference; and hence the pressure at the circumference after the formation of the vortex is greater than the pressure at the center by an amount equal to the height which corresponds to the velocity of the particles at the circumference.

If instead of an open vessel such as is shown in Fig. 5, there be a thin closed vessel such as is shown in Fig. 6, and it be filled with a liquid and then revolved about a vertical axis *A B* passing through the center, there will be found to be the same tendency to form a vortex that there was in the vessel of Fig. 5. In this case, however, the

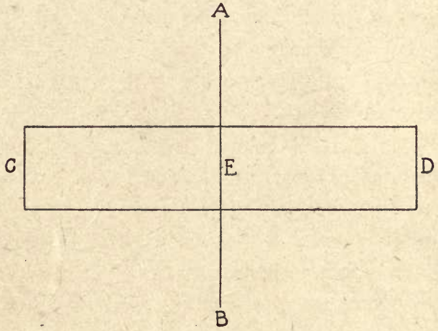


FIG. 6.

shape of the surface of the liquid cannot change because of the top of the vessel, but the pressures at various points from the center towards the circumference will change in a manner similar to what they did in the case of Fig. 5. The pressure on the bottom at the circumference will be greater than at the center by an amount exactly equal to the height of a column of the liquid corresponding to the velocity of rotation at the circumference. In the same way the pressure against the top of the vessel would change, it would become less at the center and greater at the circumference because of the rotation, but the difference between the pressure at the center and that at the circumference will be exactly equal to that corresponding to the velocity of rotation of the liquid at the circumference.

In what has been said nothing has been intimated as the effect of friction or the way in which the liquid is made to rotate in the vessel. The effect of friction is simply to make it more difficult to set the liquid to rotating and to reduce the velocity of the particles in direct contact with or near to the walls of the vessel. The means adopted to put the liquid in motion makes no difference as to the result or as to the difference between the pressures at different points provided the liquid is actually made to rotate. If, however, the liquid be made to rotate by means of a small paddle wheel such as is shown in Fig. 7, where the diameter *C D* of the vessel is greater than the diameter *A B* of the wheel, the velocity of rotation of the liquid between the ends of the paddles and the circumference of the vessel may be, and is very likely to be, somewhat greater than the velocity of the extremities of the paddles of the wheel. That is to say, because of the adhesion of the particles to the paddles some of the particles immediately outside of the wheel will be carried around just as if the diameter of the wheel were greater than it actually is. This apparent increase in the diameter of the wheel depends upon the adhesion of the particles to one another and to the ends of the paddles. The difference between the pressure at the center and that at the circumference will be equal to *the height*

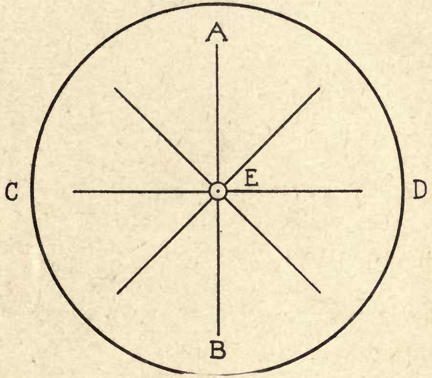


FIG. 7.

corresponding to the velocity of the liquid at the circumference and not the height corresponding to the velocity of the tips of the paddles, although the two velocities may happen to be the same. This is of importance when discussing the pressure produced in the casing of a fan or a centrifugal pump when the outlet is closed.

The results pointed out for the rotation of a liquid in a closed vessel are exactly the same as those obtained when a centrifugal fan wheel is revolved in its casing or housing with either the outlet or the inlet closed so as to prevent the flow of air through the wheel. The diameter of the wheel may apparently be increased and the difference between the pressure at the center and that of the circumference may be somewhat greater than that corresponding to the velocity of the circumference of the wheel. If the inlet of the casing is closed and the outlet open, the pressure at the circumference of the casing will be equal to that of the atmosphere, and the pressure at the center will be less; while if the inlet is open and the outlet is closed, the pressure at the center will be equal to that of the atmosphere, and the pressure at the circumference will be greater. But in both cases the difference between the pressure at the center and that at the circumference of the casing will be exactly equal to that corresponding to the velocity of rotation of the *air* at the circumference of the casing, and the velocity of rotation of the air may be greater than the velocity of rotation of the tips of the blades of the fan.

Vortex with Radial Flow. What has been said in the preceding article applies to vortexes in which there is no radial flow, either outward from the center towards the circumference or inward from the circumference towards the center, but the whole motion of the liquid in the vessel is one of rotation about the center. This, however, is not the condition which exists in a fan when it is discharging air. Such a condition exists in a fan only when either the inlet or the outlet is closed. When a fan is discharging air there is a flow of air outward from the central part towards the circumference, and the air in the fan wheel has a motion of rotation about the center and also a radial motion from the center towards the circumference.

As no centrifugal fan has been built in which the air traveled from the circumference towards the center, it is not necessary to discuss here vortices which have a radial velocity inward towards the center.

After a vortex is formed there may be a radial velocity of liquid from the center towards the circumference without effecting the vortex in any way, provided that there is no place where the radial velocity exceeds the velocity of rotation. If the radial velocity at any place does exceed the velocity of rotation there is a tendency at that place to break down the vortex, and the pressure due to the vortex is reduced at every point by an amount equal to the height corresponding to the difference in heads required for the radial velocity and the velocity of rotation at that point.

Now, in a vortex that has an outward radial flow the liquid enters near the center, where the velocity of rotation is less than at any point farther out towards the circumference, and hence what has just been said amounts to saying that the velocity of radial flow at the entrance must not exceed the velocity of rotation at the entrance. If the radial velocity at the entrance exceeds the velocity of rotation at the entrance, the pressure at the circumference will be less than that due to the velocity of rotation at the circumference by an amount equal to the difference between the heads required for the radial velocity and for the velocity of rotation at the entrance.

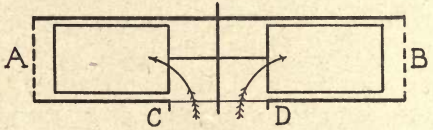


FIG. 8.

This can probably be made somewhat clearer by reference to Fig. 8, which represents a form of vortex-producing apparatus working upon air. That is to say, it is a fan whose inlet is *C D*. If the outlet at the circumference be closed, as indicated by the dotted lines *A* and *B*, and the paddle wheel be revolved, there will be no flow through the fan, and the pressure at the circumference will be greater than that at the entrance by an amount corresponding to the velocity of the air at the circumference. Let this pressure be *P*.

Now suppose an opening be made at A or B. The air will flow out of this opening with a velocity corresponding to the pressure at the circumference, and it will enter the fan with a radial velocity of a certain amount depending upon the quantity of air flowing through the apparatus per minute. Let the pressure corresponding to this radial velocity be p_1 and let the pressure corresponding to the velocity of the tips of the paddles at the point of entrance be p_2. Then as long as p_1 does not exceed p_2 the pressure at the circumference will be P; but if p_1 is greater than p_2 the pressure at the circumference will be

$$P - (p_1 - p_2) = P + p_2 - p_1$$

It has been assumed in the preceding that the velocity of radial flow is greatest at the point of entrance, since this assumption is in accordance with the facts as they actually exist in fans, as will be shown later.

Hence if we call p the difference between the pressure at the center and that at the circumference of a vortex having a radial flow, we find that its value depends upon the condition of radial flow as follows:

1. When the velocity of the radial flow at entrance is *equal to or less than* the velocity of rotation at entrance

$$p = P \qquad (10)$$

2. When the velocity of the radial flow at entrance is *equal to or greater than* the velocity of rotation at entrance,

$$(11) \qquad p = P + p_2 - p_1$$

It is to be noticed that if p_2 is equal to p_1, equation (11) becomes

$$p = P$$

which is the same as equation (10).

Conditions 1 and 2 are the only ones which can arise in the case of vortices with an outward radial flow, and the equations for these conditions, that is, equations (10) and (11), are the bases of the equations used in discussing centrifugal fans.

CHAPTER III.

Fans. Every centrifugal fan may be considered as an apparatus for producing vortexes of air or gases with a radial flow outward from the central part towards the circumference. And the more perfect are the vortexes produced the more efficient as a rule is the fan. The type and proportions of a fan should depend upon the work it has to do. If it is intended for the purpose of moving a large quantity of air against a comparatively low pressure its proportions and general construction should be decidedly different from what they should be if it were intended for the purpose of moving a rather small quantity of air against a high pressure. The earlier fans were intended principally for the ventilation of mines and they were required primarily for moving large quantities of air against comparatively low resistances. They were usually aspirator fans rather than blowers. That is to say, they were used for producing a partial vacuum in the air-shaft, and thus creating a flow of air through it. They usually discharged the air either into a large chamber surrounding the fan or directly into the surrounding atmosphere. They seldom had a housing or casing about the fan wheel except when it was necessary to protect the wheels from the weather. This type of fans was followed by a type which always had a housing or casing surrounding the fan wheel and the fans could be used either as aspirators for creating a suction or as blowers for forcing air against a pressure. This second type of fans was followed by a third type which differed from the second only in the shape of the housing or casing and which was the forerunner of what may be called the modern type of centrifugal blower so much used to-day for heating and ventilating work, for mechanical draft work and in fact wherever it is necessary to move large quantities of air against comparatively small resistances. Fans of this last type could be used equally well either as aspirators or blowers.

It is extremely difficult to say to whom belongs the honor of devising the first type of centrifugal fan, as the development of these fans seems to have been almost identical as to time and character on the continent of Europe and in England. As is usually the case in every art, men were working on the problem in different places very remote from one another and were reaching pretty much the same results by somewhat different methods.

The first type of fans may be considered as represented by any one of half a dozen different makes of fans erected in England or Europe previous to 1830.

The second type of fans may probably be ascribed to the ingenuity and inventive genius of Guibal of France.

The third type of fans was simply the outgrowth of the second or Guibal type. It is hard to say to whom belongs the honor of its invention, although it is sometimes spoken of as the Schiele type of fans.

The First Type of Fans. This type of fans was always used as an exhauster for moving air by suction and discharging it directly into the atmosphere. The fans were used primarily for producing a circulation of air for ventilating purposes through mines. Before its use mines had been ventilated by creating an upward draft in an air-shaft by means of a fire at the bottom. The fan was usually mounted on the surface of the ground near or at the air-shaft, with its inlet connected to the outlet of the air-shaft. When the fan was revolved it created a reduction of pressure at the top of the air-shaft, which in turn produced a flow of air or gases from the mine below up the air-shaft into the fan, by which it was discharged into the atmosphere. Fans of this type consisted simply of a fan wheel, often without any casing or housing except what was necessary to protect them from the weather, and they discharged the air or gases directly into the atmosphere from all points of the periphery of the wheel. The wheels were usually quite large, some even having a diameter as great as 25 feet. The diameter of the inlet was usually about 0.6 the diameter of the wheel. The width of the wheel, that is, the dimension in the direction of the axis, was usually about 0.50 of the diameter of the inlet. The wheel was usually of a uniform width from the inlet to the circumference. There was always only one inlet to the fans of this type; that is, air was admitted to the wheel from one side only. The wheels could be placed either in a vertical or a horizontal position. When in a vertical position, the fan wheel had its axis horizontal and the inlet was connected with the air-shaft of the mine to be ventilated by a horizontal duct. When in a horizontal position, the axis of the fan wheel was vertical and the inlet was usually placed directly over the outlet of the air-shaft.

The vanes or paddles of the earlier fans of this type were usually straight and were placed radially, but in the later fans they were often curved. It is possible that to Combes belongs the honor of first suggesting that the vanes of fans be curved. He required that the vanes of the fans designed by him be curved to a very marked degree; other designers required that the vanes be curved, but not to the same shape recommended by Combes.

The earlier fans of this type usually had but one movable disk which carried the vanes, and the fan was placed so that the vanes moved close to a stationary wall on the inlet side of the wheel. The movable disk was fastened to the shaft and extended out to the circumference of the wheel. The later fans had two movable disks between which the vanes were fixed as in the modern fans. One of these disks extended from the shaft to the circumference of the wheel, while the other, on the inlet side of the wheel, extended from the inlet opening to the circumference of the wheel.

Fig. 9 shows a vertical fan of this type. *A* is the movable disk which was fastened to the shaft *D*, and which carried the radial vanes *B*. The fan was driven by the pulley *E*, fastened to the shaft *D*. *C* is the inlet duct leading to the fan. It terminated at the vertical wall *H*, close to which the vanes *B* moved when the fan was made to revolve. The direction of the flow of air from the inlet *C* through the wheel is indicated by the arrows.

The disk *A*, with the location of the shaft *D*, and the position and arrangement of the vanes *B*, are shown in Fig. 10.

Figs. 9 and 10 are taken from a Treatise on

Ventilation, by Wyman, published at Boston in 1846.

It will be noticed that this fan has no casing or housing, and that since the vanes are radial

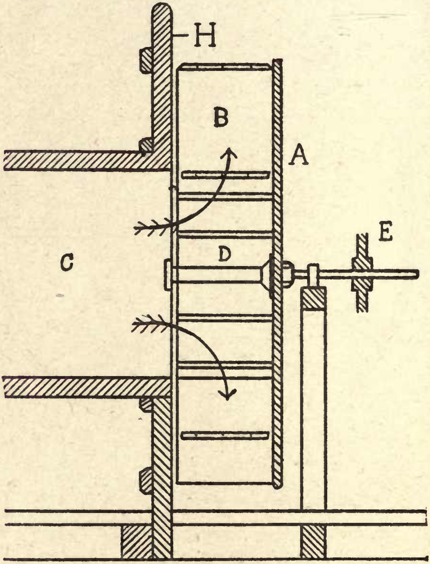


FIG. 9.

it will work equally well whichever way it be made to revolve. The outside diameter of the wheel, measured to the tips of the vanes, is about twice the diameter of the inlet; and the width of the vanes, measured from the inlet to the disk *A* is about one-half the diameter of the inlet, or one-quarter the diameter of the wheel.

The vanes *B* were made to revolve as close to the face of the wall *H* as possible in order that

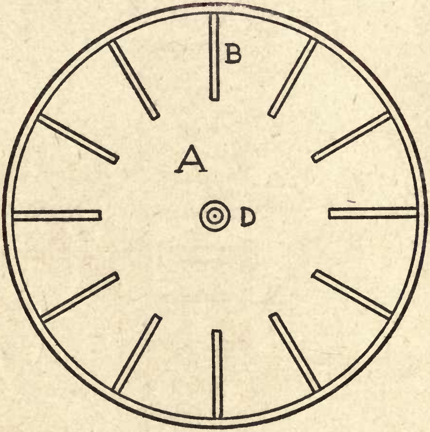


FIG. 10.

the leakage through the space between the vanes and the wall should be small.

Figs. 11 and 12 show an early form of a wheel of this type with vanes as recommended by Combes. This form of wheel is shown by Delaunay in his *Cours Elementaire de Mecanique*, published at Paris in 1854.

In the figures, *A* is the moving disk to which the vanes *D* were fastened, and which was mounted on the shaft *B*. *C* is the pulley, which was mounted on the shaft *B*, by which the fan was driven

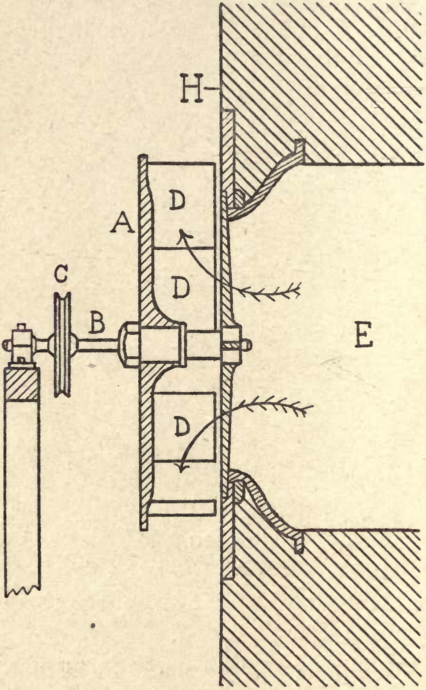


FIG. 11.

This fan, like the one shown in Figs. 9 and 10, had only one movable disk, and it was placed close to a vertical wall *H* through which the inlet duct

E passed. The air flowed from the inlet duct *E* through the fan and was discharged at all parts of the circumference.

The diameter of the inlet of the wheel was about 0.7 the diameter of the wheel, and the width of the wheel was about one-half the diam-

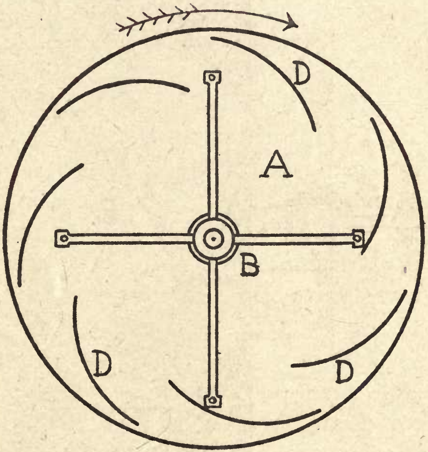


FIG. 12.

eter of the inlet. The vanes were very much curved, and the wheel was run with the convex part of the vanes forward, as indicated by the arrow in Fig. 12.

Tests made with some of the earlier types of wheels with vanes curved as recommended by

Combes show that vanes with much curvature were not so efficient as straighter vanes. It seems that they were not so effective in imparting a motion of rotation to the air as were the straighter and more nearly radial vanes.

The later fans of this type, most of which had two moving disks with the vanes between, resembled very much the modern fans of that type known as the "cone wheel."

The Second or Guibal Type of Fans. The peculiarity of the fans of this type is that they always had casings or housing in which the wheel carrying the vanes revolved. These housings were always fitted close to the wheels and had either one or two openings at the center through which the air entered. When there were two openings, there was one on ench side of the housing. The air was discharged from only one part of the periphery of the wheel instead of from every part as in the first type of fans.

Fig. 13 shows a sectional view of a Guibal fan taken from an illustration given in *Chauffage et Ventilation*, by Planat. *A* is the inlet; *B*, the vanes; *C*, the housing or casing; *D*, the outlet flue; and *E*, a movable screen which could be moved up or down in the groove *F*, thus increasing or diminishing the area of the outlet opening *H*, through which the air escaped from between the blades or vanes of the wheel into the outlet flue *D*.

It will be noticed that the housing is fitted close to the wheel, and that air can escape from the wheel only when one of the spaces between two consecutive vanes is opposite the opening *H*. It was found as the result of experience and experiments that for every fan there was a particular area of the opening *H* which gave the

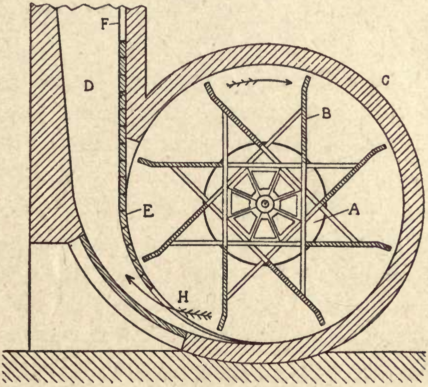


FIG. 13.

greatest calculated efficiency. And as it was impossible to calculate this best area of the opening *H*, the screen *E* was made so as to be adjustable in the groove *F*.

The vanes are shown flat and bent forward at the outer tips; and at the inlet end they are inclined backward at an angle or about 45° with the radius. The vanes were often put in so as to coincide with the radius, and at other times they were curved. For fans of this type there was never a moving disk to which the vanes were fastened and which moved with them. The sides of the housing took the place of the disks of the earlier type of fans, and these sides were always stationary and made to fit close to the wheels.

The diameter of the inlet of the fan shown in Fig. 13 is seen to be less than half the diameter of the wheel, and it is probable that there were two inlets, one in each side of the housing. As these fans could be used either as exhausters or blowers, they were placed either at the top of the air-shaft of the mine to be ventilated or at the bottom, depending upon which was the more convenient in each particular case.

Planat does not say how wide the particular fan shown in Fig. 13 was, although it is probable that, judging from other fans of the same type, the width was equal to about one-half the diameter of the inlet.

The outlet flue *D* is shown tapering, with the small end at the bottom. Some of the earlier Guibal fans were put up with square outlet flues, but Guibal conceived the idea of gradually reducing the velocity of the air as it passed up the flue from the fan to the flue outlet, in order to avoid the loss of head due to a sudden decrease in velocity. The result was that most of the Guibal fans of which one finds descriptions, have the tapering outlet flue mentioned in connection with them. In fact, this tapering outlet flue with the small end at the fan outlet is often mentioned as one of the characteristics of a Guibal fan installation.

The Third Type of Fans. This type of fans resembles very much the second or Guibal type. It had always a casing to the fan wheel, but the casing differed from that of the Guibal type of fans in that it was what was known to the earlier writers as an eccentric casing. The casing really consisted of a scroll which was close to the wheel at one point and gradually left the wheel so that it surrounded the wheel in the form of a spiral. The writer has been unable to determine whether this type of fan is earlier than the Guibal type or whether it is later. It is extremely difficult to get exact dates as to the introduction of the closed or encased type of fans, and it is especially difficult to determine whether the fan with the closely fitted casing antedated the fan with the spiral or eccentric casing.

Fig. 14 shows a fan of this type which is reproduced from an illustration in Ure's Mechanical Dictionary of Arts, Manufacturers and Mines, published in New York, in 1844, from the third London edition.

It will be noticed that this fan shows a casing in the form of a scroll or spiral, but Ure in his description does not say whether or not it is a true scroll or spiral. Later drawings of wheels of this type show the casing as a true spiral, and later writers in speaking of the casing say that the scroll should be of the form of an Archimedean spiral. Some drawings show the casing

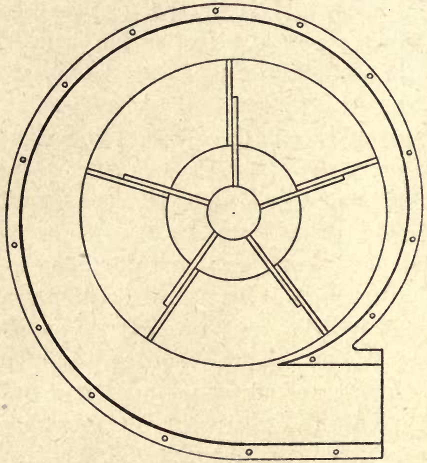


FIG. 14.

as circular but of larger diameter than the wheel and eccentric to it instead of concentric, and hence fans of this type were often called "eccentric" fans, to distinguish them from the fans of the Guibal type which had circular casings that were close fitting to the wheel.

The object of the scroll or spiral casing was to enable air to be discharged from all points of the periphery of the wheel, and there should be therefore a gradual increase of the distance between the spiral and the circumference of the wheel from the point at which the spiral is nearest the wheel to the opening of the discharge. When the scroll or spiral was of the proper proportions there was an almost uniform discharge from all parts of the periphery and, therefore, an almost uniform inward flow at the entrance and through the wheel between the vanes. In the fans of the Guibal type there was no flow except through the space between two consecutive vanes which were opposite the outlet opening, and as soon as this space was moved away from the outlet opening the flow through it ceased, but began through the space between two other vanes. There was, therefore, an intermittent flow through the space between each pair of vanes, and the starting and stopping of the flow through the spaces between the vanes made the fans less efficient than fans where the flow through the wheel was continuous and uniform at all times for the same speed of the wheel.

Fig. 15 shows diagrammatically the form of spiral which was used on the latter fans of this type. *A* is the entrance and *B* the outlet. The scroll began at the point *a* where it was quite close to the wheel, and ended at the point *e*. The wheel was supposed to revolve as indicated by the arrow. Following the scroll from the beginning to the end, in the direction of the motion of the wheel, the space between the wheel and the scroll gradually and uniformly increased. If

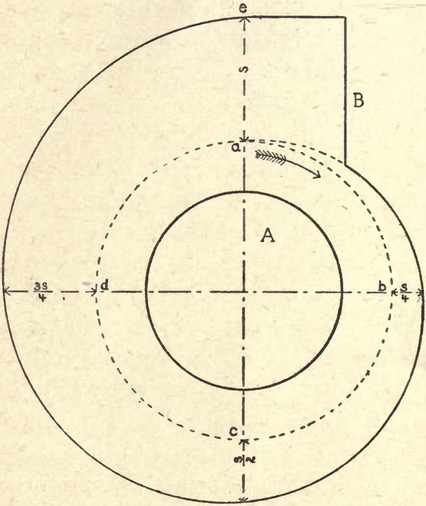


FIG. 15.

the space between the points *a* and *e*, the first and last points of the scroll, was *s*; the space between the wheel and scroll at *b*, one-quarter the circumference of the wheel from *a*, would be $\frac{s}{4}$; at *c*, one-half the circumference from *a*, the space between the wheel and the scroll would be $\frac{s}{2}$; and at d, the space would be $\frac{3s}{4}$.

A curve such as shown in Fig. 15, drawn about a circle in such a way that the distance between the curve and the circumference increases directly as the distance measured on the circumference of the circle, from the starting point, is an Archimedean spiral. This is the curve which was supposed to be used with fans of this type. Ordinarily, however, the true, exact, Archimedean spiral was not used, but approximations made up of arcs of circles were used. If the curve was made up of parts or arcs of two circles it was called a "two radius" scroll or spiral; if made up of three arcs it was called a "three radius" spiral; if of four arcs, a "four radius" spiral, etc.

Fans of this type were usually small as compared to fans of the preceding types and the wheels were run at a considerable number of revolutions per minute, so that the air was given the same velocity of rotation by means of the smaller wheels running at a high speed as it would be by means of the larger wheels running at a comparatively low number of revolutions. The width of the wheel in the earlier fans of this type was usually uniform from the center to the periphery; in the later forms, however, the width at the periphery was less than that at the center in almost the same proportion as that found in modern wheels.

When fans of this type were made with a double entrance, that is to say when there was an inlet opening in each side of the casing for the entrance of the air, the wheels were often provided with a central disk to which the vanes or paddles of the fans were attached and which really made of the double admission fan, two single admission fans joined together.

Modern Type. Fans of the modern type may be divided into two classes. One is an improvement on and in every way similar to the third type of fans described before. This class includes all of the ordinary centrifugal blowers and exhausters. The other class of modern fans includes what are known as "cone wheels," and is an improvement on the first type of fans.

The ordinary centrifugal blower or exhauster consists of a wheel carried on a shaft which revolves inside of a casing or as it is commonly called, a "housing." The housing is always in the shape of a spiral about the wheel and has either one or two inlets for the air. If there are two inlets, one in each side of the housing, the fan is called a "double admission fan," and if there is but one entrance or inlet for the air the fan is called a "single admission fan." In order that there may be as little impediment as possible to the flow of air into the fan during operation the single admission fans are often made so that both of the bearings for the shaft of the wheel are on the same side of the housing, and then the wheel is said to be overhung. In double admission fans it is impossible to avoid having the bearings in front of one of the inlets so there is usually a bearing on each side of the housing. The result of this is that the area of the inlets in a double admission fan is usually considerably reduced by the space occupied by the shaft and the bearings, and as the fans are usually driven by a pulley fastened to the shaft this pulley by obstructing one of the inlets interferes with the entrance of the air through that particular inlet to a very considerable degree. If the fan be driven by a direct attached motor or engine instead of a pulley there is then a still greater obstruction to the entrance of the air through one of the inlets. The result is that a double admission fan has its inlets so obstructed that it is little better than a single admission fan with an overhung wheel. Exhausters are always made with a single inlet, while blowers are made with either a single or a double inlet.

In the trade, fans are usually designated according to the shape of the housing; the number of inlets; the location of the outlet; and the direction of the flow of the air when leaving the outlet.

Fig. 16 shows a side view of what is known as a full housed, double admission, horizontal, top discharge fan; and Fig. 17 shows a section, in the direction of the axis of the wheel, of the same fan. This fan is called a "full housed" fan because all the casing or housing is shown above

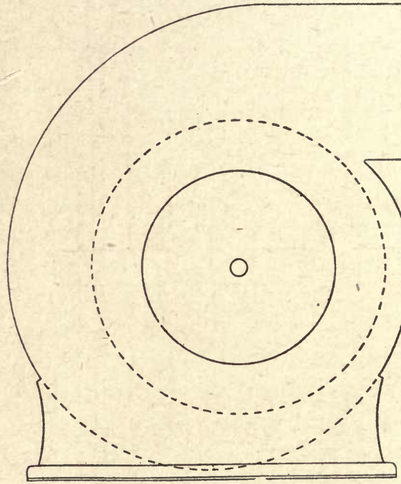


FIG. 16.

the foundation on which the fan rests. If a part of the housing or casing of the fan projects below the foundation so that the wheel revolves in a pit as shown in Fig. 18 the fan is called a "three-quarter housed" fan. It is evident, of course,

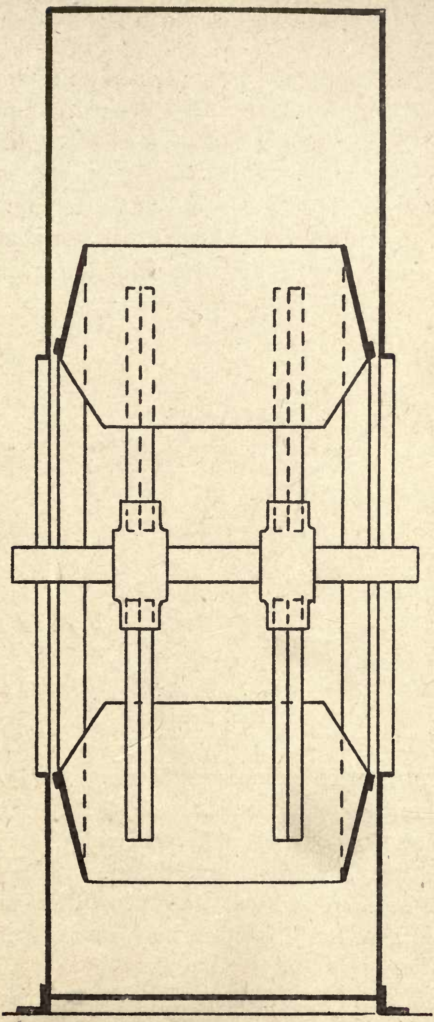


Fig. 17.

that in the case of a three-quarter housed fan all of the inlet or entrance opening must be above the foundation in order to give free admission to the air.

If the outlet of the fan is at the top of the housing and the air is blown in a horizontal direction as shown in Fig. 15, the fan is called a

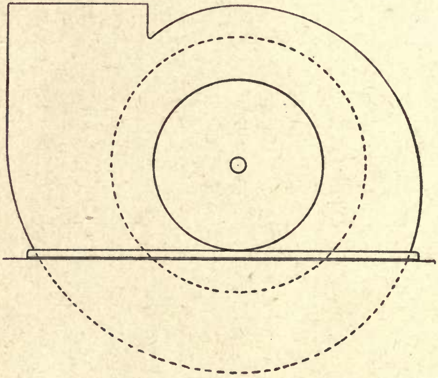


FIG. 18.

"top, horizontal discharge" fan. If, however, the outlet of the fan is at the top and the air is blown upward instead of horizontally, the fan is called a "top, up discharge" fan. If the outlet is at the bottom and the air is blown horizontally the fan is called a "bottom, horizontal discharge" fan.

The fan in Fig. 18 has a single inlet, the outlet is at the top and the air is discharged upward; it is called, therefore, a "three-quarter housed, single admission, top, up discharge" fan.

Fans are sometimes made with two outlets as shown in Fig. 19, where the air is discharged from

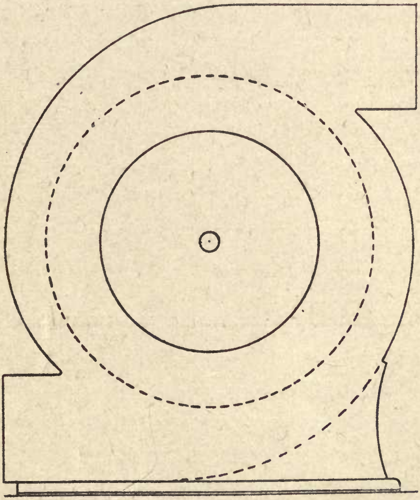


FIG. 19.

the housing at the top towards the right and at the bottom towards the left. Such fans are known as "double discharge" fans, and they are made so that the same or different quantities of air pass out through each outlet.

The ordinary commercial fan used for heating and ventilating work or for mechanical draft has the housing made of sheet steel put together with angle irons, and well braced on the outside with angle irons so as to enable it to withstand the pressure and racking due to the movement of the wheel. The part that rests on the foundation is usually reinforced by heavy angle irons varying in size from the ordinary commercial 3 x 3 angle to the 6 x 6 angle. In some cases where very large wheels are used for ventilating work the housing is built up of wood and brick.

The housings and the shapes and dimensions of the scrolls or spirals will be discussed in a later chapter. It suffices to say here that the scrolls are usually either three or four radius scrolls or spirals.

The type of centrifugal fan that is used nowadays and known as a cone wheel is very similar indeed to the early fans described under the first type. It is seldom used where air is to be discharged against any considerable pressure but is very largely used for heating and ventilating work where the air is to be moved against little pressure and where the object is to move a large quantity of air against a low pressure. This fan will be discussed later.

CHAPTER IV.

Fan Wheel. A fan wheel of the ordinary type used for heating and ventilating work is shown in perspective in Fig. 20. *A* represents the side plates between which are fastened the blades *B*, commonly called vanes or "floats." To these vanes are riveted tee irons *C* which are cast into the hub of the wheel. One set of these tee irons constitutes what is called a "spider." It will be noticed that in Fig. 20 there are two sets, one near each side of the wheel. Such a wheel is said to have a double spider. Small wheels 4 feet or less in diameter usually have a single spider.

The side plates *A* are usually reinforced at the inlet of the wheel by a strip of iron or an angle iron as indicated by *D* in the figure.

When the wheel is in the housing supported on the shaft which passes through the hub, the air enters through the inlet and passes radially outward between the side plates *A* to the periphery of the wheel and from there passes into the housing. It will be noticed that the air is confined between the side plates *A* as long as it is passing

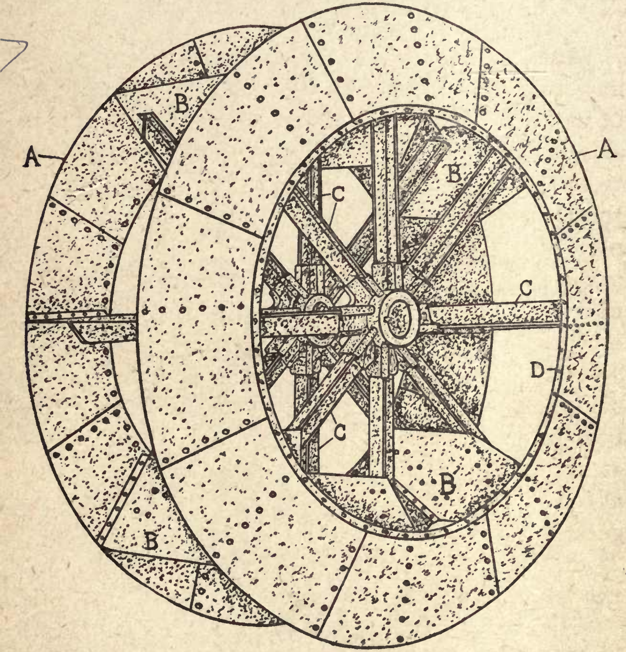


FIG. 20.

through the wheel and it is thus made to more readily take up the motion of rotation of the wheel. In this respect the fans of the modern type differ from those of the earlier types. In the earlier types the blades were not provided with the side plates *A* but they revolved directly between the sides of the casing or housing.

Vanes or Floats. There does not seem to be any rule or formula for determining the number of vanes or floats that a wheel should have. There should be enough to insure that the air passing from the wheel will be given the same velocity of rotation that the wheel has before it leaves the periphery, but there should not be so many as to make the space between two consecutive floats so narrow as to offer undue friction to the flow of the air from the center towards the periphery of the wheel. According to the old rule which was sometimes used there should be one float for every foot in diameter of the wheel. This meant that at the periphery of the wheel the floats would be about 37½ inches apart. That rule does not hold good with modern fans, as even the smaller sizes, 2½ or 3 feet in diameter, usually have 6 floats, and the larger wheels seldom have more than 12 floats.

Different manufacturers have different ideas about curving the floats. Some seem to think that they should extend from the center towards the periphery of the wheel in a radial direction as indicated in Fig. 21. Others give the floats a slight bend or curve at the outer end near the periphery of the wheel as shown in Fig. 22, in order, as they claim, to decrease the noise made when the wheel is working in the housing. Still others give the floats a considerable curve from the entrance to the outer periphery, as shown in Fig. 23. When the floats are put in radially it

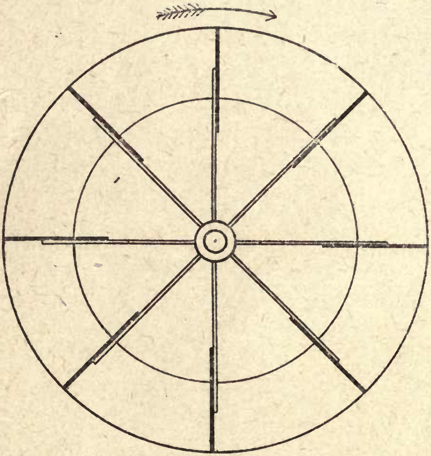


FIG. 21.

makes no difference which way the wheels revolve, but when they are curved as shown in Figs. 22 and 23 the wheels should be revolved so that the convex part of the float goes forward as indicated by the arrows. When the floats are curved as indicated in Fig. 23 and this curvature is very pronounced, it is found that the wheel will not work as well against a high pressure as a wheel with straighter and more nearly radial floats.

It will be noticed that the curved floats shown in Fig. 22 are tangent to the radius at the inlet. This is the way in which all the fan makers who use curved floats put them in. This is rather

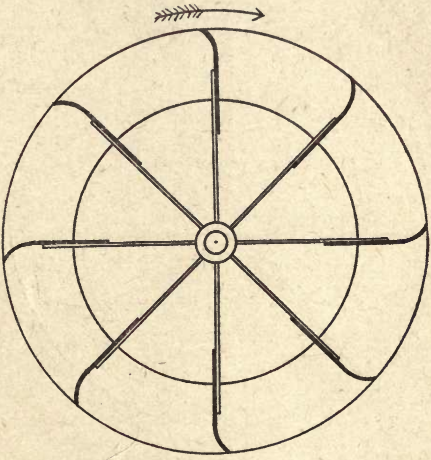


FIG. 22.

curious in the face of the fact that theoretical deductions indicate that the floats should be tangent to the radius at the periphery of the wheel and that at the inlet they should make an angle of about 45° with the radius. Most writers who have attempted to discuss the theory of fans have urged this point, and it seems to the author that it is well taken because the velocity of the air in the direction of the radius of the wheel is usually about equal to the velocity of rotation of the edges of the floats at the entrance, and hence the

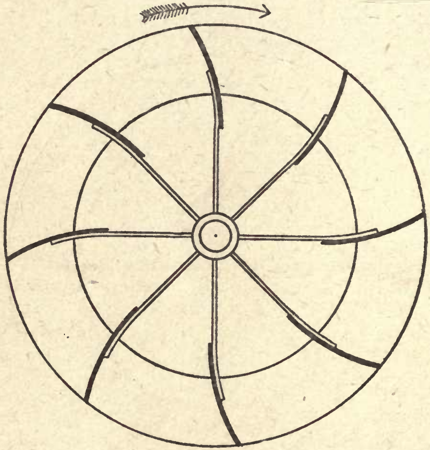


FIG. 23.

air would enter the space between the floats with less shock and probably with less loss of head if the floats were inclined at the entrance to an angle of about 45° with the radius. In order that the air may be given as nearly as possible the same velocity of rotation that the floats have, they should be radial at the outer end near the periphery, and when working against a pressure it is absolutely necessary that the floats be radial at the outer periphery of the wheel; hence, it seems that the floats should be curved as indicated in Fig. 24. Wheels with floats curved as indicated in Fig. 24 are illustrated in *Ventilationsmaschinen*

FIG. 24.

der Bergwerke by von Hauer, published at Leipsic, 1870. As far as the author knows, however, no modern fans have been made with blades curved as indicated in Fig. 24 nor are there any recorded tests of the fans of this type as shown by von Hauer.

The shape of the floats is shown in Figs. 25 and 26, where *A* represents the side plates; *B*, the float; *C*, tee irons of the spider; *E*, the heel or inner part of the float; and *F*, the tip or outer part

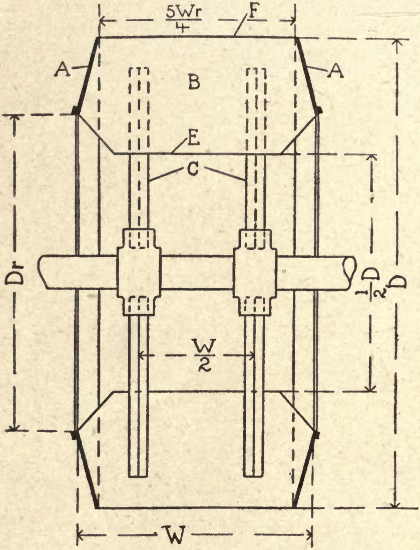
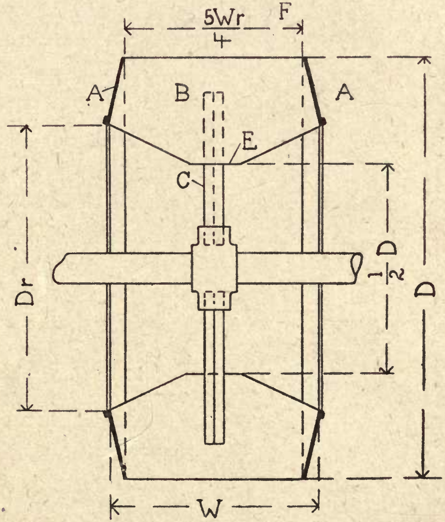


FIG. 25.

of the float. It will be noticed that the heel extends somewhat below the side plates *A* and as the side plates extend from the inlet to the circumference of the wheel the heel of the float extends below the inlet. The heel of the float is usually at a distance from the center of the wheel equal to one-quarter the diameter or one-half the radius of the wheel. The object of extending the floats towards the center is to set in motion the air between the circumference of the inlet and



Eig. 26.

the center of the wheel, and thus make it easier for the wheel to give the air during its passage through the wheel the same velocity of rotation that the floats have. The floats shown in Fig. 25 are for a wheel with a double spider, and the floats shown in Fig. 26 are for a wheel with a single spider.

Inlet. In the earlier types of fans the diameter of the inlet was almost always made equal to one-half the diameter of the wheel, but in the modern fans the diameter of the inlet is proportioned according to the use to which the fan is to be put. If the fan is to work against comparatively low pressures and is intended primarily for moving a large amount of air the diameter of the inlet is larger than it would be if the fan were intended primarily for moving small quantities of air against a considerable pressure. The ratio of the diameter of the inlet to the diameter of the wheel is usually designated by the letter r so that in Figs. 25 and 26 where D is used to indicate the diameter of the wheel, the diameter of the inlet is indicated by $D\,r$. In most fans used for heating and ventilating work it will be found that r is either $\frac{5}{8}$, equal 0.625; or $\sqrt{/2}$, equal 0.707. But for fans used to work against a considerable pressure, several ounces per square inch, r will often be 0.5 or even less. If r be made much greater than 0.7 it will be found that the wheel cannot move the air against a pressure, because the distance which the air travels in passing radially through the wheel is so short that it does not have time to acquire the velocity of rotation of the floats.

Width. In the earlier types of fans the width, W, of the fan and therefore the width of the wheel was always made such that the area for the radial flow of air through the wheel at the entrance was equal to the area of the entrance. Sometimes, of course, in order to allow for friction the area for the radial flow of air through the wheel was made slightly larger than the area of the entrance. The area for the radial flow of the air through the wheel at the entrance is equal to the area of the surface of a cylinder whose base is the diameter of the inlet and whose height is the width of the wheel. The area of the surface of such a cylinder is equal to $\pi r D W$; and the area of the inlet, assuming that there is only one inlet, is equal to $\frac{\pi r^2 D^2}{4}$. In order that these two areas may be equal to one another it is evident that W, the width of the fan, must be equal to one-quarter the diameter of the inlet. If the fan is a double admission fan and has two inlets, then the width of the fan must be at least one-half the diameter of the inlet, or double what it should be for a single admission fan.

Most fans used for heating and ventilating work are made with the width equal to one-half the diameter of the wheel, and this is the width of what is called a standard fan. What is known as a narrow fan is one for which the width is three-eighths the diameter of the wheel.

The width of the fan does not affect the quantity of air which it can handle per minute, or the pressure against which the air can be forced, so long as it is not less than one-quarter the diameter of the inlet for a single admission fan and one-half the diameter of the inlet for a double admission fan.

In modern fans the floats are made to narrow from the entrance to the periphery of the wheel in such a way as to make the radial velocity of the air passing through the wheel nearly uniform from the entrance to the periphery. The width at the periphery is usually about 25 per cent. greater than would be theoretically necessary in order that the radial velocity of the air through the wheel should be uniform from the inlet to the periphery of the wheel. The area through which the air passes radially at the inlet is $\pi r D W$; and calling w the width of the float at the periphery the area through which the air passes radially at the periphery of the wheel is $\pi D w$. In order that these two expressions may be equal to one another it is evident that

$$w = r W \qquad (12)$$

From this it is seen that the width at the periphery bears the same ratio to the width at the entrance that the diameter of the inlet bears to the diameter of the wheel.

As said before, it is usual to make the width at the periphery 25 per cent. greater than would be necessary in order that the velocity of the air in the direction of the radius of the wheel should be uniform from the inlet to the periphery, and hence the width at the periphery is usually

$$w = \frac{5\,r\,W}{4} \tag{13}$$

CHAPTER V.

Capacity. The capacity of a fan means the maximum number of cubic feet of air discharged by it per minute against a pressure corresponding to the velocity of the tips of the floats of the wheel.

As has been pointed out when discussing vortexes, the pressure in the housing of a fan will be that corresponding to the velocity of the tips of the floats, when the velocity in the direction of the radius at entrance is less than or equal to the velocity of rotation of the parts of the floats at the entrance. If the velocity of the air through the inlet be greater than the radial velocity at the inlet, the difference between the heads corresponding to these two velocities is lost. Hence the only advantage of making the wheels wider than necessary to make the radial velocity at the inlet equal to the velocity through the inlet, is that the air will be longer in the wheel and, therefore, be more likely to acquire the same velocity of rotation as the floats.

In order that the pressure in the housing shall be equal to that corresponding to the velocity of the tips of the floats, neither the velocity through the inlet nor the radial velocity at the inlet must be greater than the velocity of rotation of the points of the floats at the inlet.

Hence the maximum number of cubic feet of air discharged by a fan against a pressure corresponding to the velocity of the tips of the floats is equal to the product of the velocity of the parts of the float at the inlet multiplied by the area of the inlet, or by the area through which the air passes radially at the inlet if it be smaller than the area of the inlet.

Let us assume that we have a single admission fan, the diameter of whose wheel in feet is D. Then the diameter of the inlet will be $r\,D$, as explained before, and the area of the inlet will be $\frac{\pi\, r^2\, D^2}{4}$. This we will assume is less than the area through which the air passes readily at the inlet, as it will be if the width of the fan is greater than one-quarter the diameter of the inlet.

The velocity per minute of the points of the floats at the inlet will be $\pi\, r\, D\, N$, where N is the number of revolutions made per minute by the wheel.

And if, as explained in previous articles, we call c the coefficient of discharge, we have that the quantity of air C, in cubic feet entering the fan per minute is

$$(14) \qquad C = \frac{c\,\pi^2\,r^3\,D^3\,N}{4}$$

The value of c depends upon the shape of the inlet and is probably between 0.50 and 0.62, and may, therefore, be taken as 0.56.

The value of π is 3,1416, and hence

$$\frac{\pi^2\,c}{4}$$

is equal to 1.38, and (14) becomes

$$(15) \qquad C = 1.38\,r^3\,D^3\,N$$

Since the same quantity of air must leave the fan that enters it, equation (15) gives the capacity or number of cubic feet of air that will be delivered per minute by a fan when the pressure in the housing is that corresponding to the velocity of the tips of the floats of the wheel.

The velocity per minute of the tips of the floats is $\pi\,D\,N$, and from (3) we have, calling P the pressure in ounces per square inch corresponding to this velocity.

$$(16) \qquad \pi\,D\,N = 5200\,\sqrt{P}$$

From (16) we have

$$(17) \qquad D\,N = 1650\,\sqrt{P}$$

If now we substitute in (15) for $D\,N$ its value as given by (17) we get

$$(18) \qquad C = 1.38\, r^3\, D^2\, 1650 \sqrt{P}$$
$$= 2280\, r^3\, D^2 \sqrt{P}$$

As has been said before, one common value of r is $\frac{5}{8}$ or 0.625, and another common value is $\sqrt{0.5}$ or 0.707. Hence r^3 is usually either the cube of 0.625 or the cube of 0.707; that is, r^3 is either 0.244 or 0.353. Putting these values of r^3 in (18) we have

$$(19) \quad \begin{cases} C = 556\, D^2 \sqrt{P} & \text{when } r = \frac{5}{8} \text{ or } 0.625; \\ C = 804\, D^2 \sqrt{P} & \text{when } r = \sqrt{2} \text{ or } 0.707. \end{cases}$$

For all practical purposes we may say

$$(20) \quad \begin{cases} C = 550\, D^2 \sqrt{P} & \text{when } r = 0.625; \\ C = 800\, D^2 \sqrt{P} & \text{when } r = 0.707. \end{cases}$$

The equations for C given in (20) may be used for double admission fans as well as for single admission fans, because in the case of double admission fans one of the inlets has a pulley before it which impedes the flow into the fan; and both inlets are more or less obstructed by the shaft and the bearings which are usually placed in the inlets. The bearings and their supports take up so much of the inlet areas that the sum of the areas of the two inlets in the case of a double admission fan is very little if any greater than the area of the one inlet of the single admission fan.

Equation (20) applies to an exhauster as well as to a blower, the only difference being that in the case of an exhauster p is the *vacuum in ounces per square inch maintained at the inlet when the pressure at the outlet is that of the atmosphere*; while in the blower p is the pressure in ounces per square inch in the housing at the outlet when the pressure at the inlet is that of the atmosphere.

By transposing (20) and solving for D we get

$$(21)\quad \begin{cases} D = \sqrt{\dfrac{C}{550\sqrt{P}}} \text{ when } r = 0.625; \\[2ex] D = \sqrt{\dfrac{C}{800\sqrt{P}}} \text{ when } r = 0.707. \end{cases}$$

Equation (20) is to be used when we wish to find the capacity or number of cubic feet of air per minute a fan of a given diameter in feet will deliver when working against a given pressure in ounces per square inch; and equation (21) is to be used when we wish to find the diameter of the fan required to deliver a given quantity of air per minute against a given pressure in ounces per square inch.

Equation (20) shows how important it is to take the diameter of the inlet into account when ordering a fan, and it also explains why two fans of different makes, similar in almost every respect except as to the diameter of the inlet opening, will give such widely different results in actual use, so far as the quantities of air delivered by them are concerned. When we discuss the question of power required to run a fan and the question of efficiency, also, we shall see that the fan with a small inlet has certain advantages over a fan with a large inlet. And it will depend entirely upon the service for which the fan is to be used as to whether the inlet should be large or small.

EXAMPLE:—Determine the number of cubic feet of air delivered per minute by a fan having a wheel 8 feet in diameter with an inlet $68\frac{1}{2}$ inches in diameter, when working against a pressure of 0.5 ounce.

Here the diameter of the wheel is 96 inches and the diameter of the inlet is $68\frac{1}{2}$ inches, so that

$$r = \frac{68.5}{96} = 0.707$$

Hence we have from (20)

$$\begin{aligned} C &= 800 \times 8^2 \times \sqrt{0.5} \\ &= 800 \times 64 \times 0.707 \\ &= 36200. \end{aligned}$$

The same fan with an inlet 60 inches in diameter, making $r = 0.625$, would deliver only 22600 cubic feet of air per minute.

EXAMPLE:—Determine the diameter of fan required to deliver 12000 cubic feet of air per minute against a pressure of 0.25 ounce.

Let us assume first that r will be 0.625, then we have from (21)

$$D = \sqrt{\frac{12000}{550\sqrt{0.25}}} = \sqrt{\frac{12000}{550 \times 0.5}}$$

$$= \sqrt{43.6} = 6.6$$

Now let us assume that r will be 0.707, then we have from (21)

$$D = \sqrt{\frac{12000}{800\sqrt{0.25}}}$$

$$= \sqrt{30} = 5.5, \text{ about.}$$

The diameter of the inlet for the fan 6.6 feet in diameter would be $0.625 \times 6.6 = 4.13$ feet or 49.5 inches; and the diameter of the inlet of the smaller fan would be $0.705 \times 5.5 = 3.89$ feet or about 46.5 nches.

Table II. gives the capacities, as calculated by (20) for different pressures, of fans with inlets whose diameters are 0.707 of the diameters of the wheels.

TABLE II.

Capacities of centrifugal fans.

Diam. of wheel in ft.	Pressure per square inch in ounces.									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
3	2280	3220	3940	4550	5100	5570	6020	6440	6830	7200
3½	3100	4380	5360	6200	6930	7600	8200	8760	9300	9800
4	4050	5730	7010	8100	9050	9920	10700	11400	12100	12800
4½	5130	7250	8880	10200	11500	12500	13600	14500	15400	16200
5	6330	8950	11000	12700	14100	15500	16700	17900	19000	20000
5½	7650	10800	13200	15300	17100	18700	20200	21600	23000	24200
6	9100	12900	15800	18200	20400	22300	24100	25800	27300	28800
*6½	10700	15100	18500	21400	23900	26200	28300	30200	32100	33800
7	12400	17500	21500	24800	27700	30400	32800	35000	37200	39200
8	16200	22900	28000	32400	36200	39600	42800	45800	48600	51200
9	20500	29000	35400	41000	45800	50200	54200	58000	61500	64800
10	25300	35800	43800	50600	56500	62000	67000	71500	75900	80000
11	30000	43300	53000	61200	68400	75000	81000	86500	91700	96800
12	36500	51600	63200	73000	81500	89300	96500	100300	109000	115000

* This is an odd size, not made by all manufacturers.

TABLE IIa.

Capacities of centrifugal fans.

Diameter of wheel in feet.	Pressure per square inch in ounces.									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
3	1570	2220	2710	3130	3510	3830	4140	4420	4700	4950
3½	2130	3010	3630	4260	4760	5220	5640	6030	6400	6740
4	2780	3940	4830	5570	6230	6820	7360	7840	8320	8800
4½	3530	4980	6110	7020	7910	8600	9350	9970	10600	11100
5	4350	6150	7560	8730	9700	10700	11500	12300	13100	13700
5½	5260	7430	9070	10500	11800	12900	13900	14800	15800	16600
6	6260	8870	10900	12500	14000	15300	16600	17700	18800	19800
6½	7360	10400	12700	14700	16400	18000	19500	20800	22100	23200
7	8530	12000	14800	17000	19000	20900	22500	24100	25600	26900
8	11100	15700	19300	22300	24900	27200	29400	31500	33400	35200
9	14100	19900	24300	28200	31500	34600	37300	39900	42300	44600
10	17400	24600	30200	34800	38900	42600	46100	49200	52200	55000
11	21100	29800	36400	42100	47000	51600	55700	59500	63100	66600
12	25100	35500	43500	50200	56000	61400	66400	69000	75000	79100

To find from the table the capacity or number of cubic feet of air a fan will deliver per minute against a given pressure, find the diameter of the wheel in the first column, headed "Diameter of wheel in feet," and then look to the right to the column having at the top the given pressure, and the number in this column on the same line as the diameter of the wheel is the capacity of the fan at the given pressure. Thus, to find the capacity of a fan having a wheel 5 feet in diameter when working against a pressure of 0.4 of an ounce per square inch, look along the line on which 5 is in the first column, and under the column having 0.4 at the top is found the number 12700, which is the capacity of the fan under the given conditions.

The table may also be used to determine the diameter of a wheel required to deliver a given quantity of air per minute at a given pressure. Thus, to find the diameter of a wheel which has a capacity of 30000 cubic feet against a pressure of 0.5 of an ounce per square inch, look down the column headed 0.5 until 30000 is found, and the diameter of the wheel will be found on the same line but in the first column. In this particular case 30000 is not found in the 0.5 oz. column; 27700 is found opposite the 7-foot wheel and 36200 is found opposite the 8-foot wheel. It now becomes a question of engineering as to which wheel should be used. If the 7-foot wheel is used it will have to be speeded up so as to give a pressure of 0.6 of an ounce in the housing, and at this speed it will deliver 30400 cubic feet instead of 30000. If on the contrary, the 8-foot wheel be used it will have to be slowed down so as to give a pressure in the housing of about 0.35 of an ounce, but this pressure may not be sufficient to overcome the friction of the air after it leaves the fan. It would probably be better to use the 8-foot wheel, even if it gave more air than required, because, as will be seen later, the power required to run the 8-foot wheel and deliver 30000 cubic feet of air per minute will be less than that required for the 7-foot wheel.

Table IIA gives the capacity of fans having wheels with inlets whose diameters are 0.625 of the diameter of the wheel. In every other respect it is exactly like Table II.

EXAMPLE:—Determine the capacity of a fan with a 5-foot wheel, whose inlet is 37.5 inches in diameter when working against a pressure of 0.4 of an ounce per square inch.

Here the diameter of the inlet is 0.625 of the diameter of the wheel, and from Table IIA we see that the capacity is 8730.

From Table II. we see that the same fan with an inlet whose diameter is 0.707 of the diameter of the wheel has a capacity equal to 12700; nearly 50 per cent. more with the larger inlet than with the smaller.

Blast Area. In the discussion of the capacity of a fan nothing has been said as to the size or area of the outlet orifice, although it has been assumed that it was large enough to allow the air to pass out. When the outlet orifice is small, the air will pass out of it with a velocity equal to that of the tips of the floats. If the opening be made larger and larger the air will continue to pass out with a velocity equal to that of the tips of the floats, until the area of the outlet multiplied by its proper coefficient of discharge becomes equal to what is known as the Blast Area of the fan. And if the area of the outlet be made larger, the pressure in the housing will become less and the air will pass out with a velocity less than that of the tips of the floats.

The *Blast Area* may be defined then as that theoretical area of outlet which will allow the maximum quantity of air to pass out while the pressure in the housing remains equal to that corresponding to the velocity of the tips of the floats.

By "theoretical area" is meant the area of an opening whose coefficient of discharge is 1.

When the pressure in the housing is equal to that corresponding to the velocity of the tips of the floats, the velocity of the air passing through the outlet is equal to the velocity of the tips of the floats. Hence the blast area in square feet multiplied by the velocity of the tips of the floats in feet per minute, must be equal to the capacity of the fan or the maximum quantity of air discharged by the fan when working against a pressure equal to that corresponding to the velocity of the tips of the floats.

Hence, calling A the blast area in square feet, and using the same notation as before, we have that the velocity of the tips of the floats in feet per minute is $\pi D N$; and the capacity C of the fan is

$$C = \pi D N A \tag{22}$$

But from (15) we know that

$$C = 1.38\, r^3 D^3 N$$

and putting this value of C in (22) we have

$$\pi D N A = 1.38\, r^3 D^3 N$$

From which we get

$$A = 0.44\, r^3 D^2 \tag{23}$$

Equation (23) gives the blast area in square feet for single inlet fans, but because of the impediments to the flow of air through the inlets of double inlet fans, as has been explained before, it is well to assume that for most commercial double inlet fans the blast area is the same as for single inlet fans. If it were possible to have a double inlet fan with each inlet as open and free for the admission of air as is usually the case with a single inlet fan, the capacity and also the blast area of such a fan would be double what it is for a single inlet fan. The only advantage to be gained in buying a double inlet fan for heating or ventilating work is the possibility of getting some increase of capacity and blast area by reason of the fact that the two inlets may offer a larger area for the admission of air than one, and whatever increase there may be, is simply so much increase in the factor of safety used.

If (23) be multiplied by 144 we have A', the blast area in square inches, is

$$A' = 144\,A = 63.4\,r^3\,D^2 \tag{24}$$

If r is equal to $\sqrt{2}$ or 0.707, (23) and (24) become,

$$A = 0.155\,D^2 \tag{25}$$

$$A' = 22.4\,D^2 \tag{26}$$

If r is equal to $\frac{5}{8}$ or 0.625, (23) and (24) become,

$$A = 0.107\,D^2 \tag{27}$$

$$A' = 15.5\,D^2 \tag{28}$$

Table III. gives the blast areas in square feet and in square inches for fan wheels of different diameters, for r equal to 0.707 and for r equal to 0.625. In this table the square feet are given to the nearest tenth; and the square inches to the nearest five below 1000, and to the nearest 50 above 1000.

TABLE III.

Blast areas.

Diameter of wheel in feet	$r=0.707$ Blast Area in sq. ft. A	$r=0.707$ Blast Area in sq. in. A'	$r=0.625$ Blast Area in sq. ft. A	$r=0.625$ Blast Area in sq. in. A'
3	1.4	200	0.96	140
3½	1.9	275	1.3	190
4	2.5	360	1.7	250
4½	3.1	455	2.2	315
5	3.9	560	2.7	390
5½	4.7	675	3.2	470
6	5.6	805	3.9	560
6½	6.5	945	4.5	655
7	7.6	1100	5.3	760
8	9.9	1450	6.9	990
9	12.6	1800	8.7	1250
10	15.5	2250	10.7	1550
11	18.8	2700	13.0	1900
12	22.3	3200	15.4	2250

In ordinary heating and ventilating work the blast area is a matter of small consequence, but in exhauster work for mills and factories where it is necessary to choose a fan to carry away shavings, dust and dirt, and where the velocity of the air in the ventilating ducts and flues must be quite large, the blast area is of great importance and plays a considerable part in the considerations which decide what size of fan to use.

The area of the inlet, as has been said before, is $\frac{\pi r^2 D^2}{4}$, and the coefficient of discharge for the inlet has been assumed to be 0.56; so that the theoretical area of the inlet or the product of the inlet area and its coefficient of discharge is

$$\frac{0.56\ \pi\ r^2\ D^2}{4} = 0.44\ r^2\ D^2$$

But from (23) we know that the blast area is $0.44\ r^3\ D^2$. Therefore, the blast area, A, is equal to the theoretical inlet area multiplied by r; and conversely, the theoretical inlet area is equal to the blast area divided by r, or $\frac{A}{r}$.

Effect of Outlet on Capacity. When the theoretical outlet area of a fan, that is the product of the area of the outlet multiplied by its proper coefficient of discharge, is less than the blast area, the pressure in the housing is that corresponding to the velocity of the tips of the floats; and the quantity of air discharged is equal to the velocity of the tips of the floats multiplied by the product of the area of the outlet and its coefficient of discharge. Let a be the area of the outlet, c its coefficient of discharge, and C_1 the number of cubic feet of air discharged per minute. The velocity in feet per minute of the tips of the floats is $\pi\ D\ N$, where as before, D is the diameter in feet of the wheel, and N is the number of revolutions per minute.

Hence the expression for the number of cubic feet of air delivered by a fan *when the product of the area of the outlet and its coefficient of discharge is less than the blast area*, is

$$C_1 = \pi D N c a \tag{29}$$

But from (22) we know that the expression for the capacity C, of the fan is

$$C = \pi D N A$$

From this we get

$$\pi D N = \frac{C}{A}$$

and this value of $\pi D N$ put in (29) gives

$$C_1 = \frac{c a C}{A} \tag{30}$$

If $c a$ is equal to A, C_1 becomes equal to C.

EXAMPLE:—Determine the number of cubic feet of air discharged per minute by a fan with a 9-foot wheel working against a pressure in the housing of 0.3 ounce per square inch and with an outlet whose area is 11 square feet and whose coefficient of discharge is 0.8. The diameter of the inlet is to be assumed to be 0.707 of the diameter of the wheel.

Here a is 11 and c is 0.8, so that $c a$ is 8.8.

From Table III. it is seen that the blast area of a 9-foot wheel is 12.6 square feet, and hence according to the conditions of the example, the area of the outlet multiplied by its coefficient of discharge is less than the blast area. The value of C_1 could be calculated direct by (29) if we knew N, but as N is not given and we have not yet explained how to find it when the pressure is known, we shall use (30) to find C_1.

From Table II. we find that C for a 9-foot wheel working against a pressure of 0.3 ounce is 35300. Hence putting the values of $c\,a$, A, and C in (30) we have

$$C_1 = \frac{c\,a\,C}{A} = \frac{8.8 \times 35300}{12.6}$$

$$= 24700.$$

When the product of the area of the outlet multiplied by its coefficient of discharge is *greater* than the blast area the problem of determining the number of cubic feet of air discharged per minute is much more complicated than when the product of the area of the outlet multiplied by its coefficient of discharge is less than the blast area. When the area of the outlet multiplied by its coefficient of discharge, or $c\,a$, is less than the blast area, A, the number of cubic feet of air discharged per minute is less than the capacity of the fan;

but when $c\,a$ is greater than A, the number of cubic feet of air discharged per minute, which we shall call C_2, is greater than the capacity of the fan. In order that the number of cubic feet of air discharged by a fan shall be greater than the capacity of the fan, the velocity through the inlet must be greater than the velocity of rotation of the points of the floats at the inlet, because the capacity of a fan, is as shown before, the number of cubic feet of air which will enter through the inlet per minute with a velocity equal to that of the points of the floats at the inlet.

As has been said when discussing vortexes with a radial flow from the center outward towards the periphery, when the radial velocity at the inlet is greater than the velocity of rotation at the inlet, the pressure in the housing or casing is less than the pressure corresponding to the velocity of the tips of the blades or floats. Further, when the velocity through the inlet is greater than the radial velocity at the inlet, and is also greater than the velocity of rotation of the points of the floats at the inlet, the pressure in the housing is less than that due to the velocity of rotation of the tips of the floats.

Let P be the pressure corresponding to the velocity of the tips of the floats; p_1, the pressure corresponding to the velocity through the inlet, or the radial velocity at the inlet if it be less than the velocity through the inlet; p_2, the pressure corresponding to the velocity of rotation of the points of the floats at the inlet; and p, the pressure in the housing making the air flow out through the outlet. Then from (11) we have

$$p = P + p_2 - p_1 \tag{31}$$

The air flows out of the housing with a velocity due to the pressure p, which according to (3) is equal to $5200\sqrt{p}$. And since the area of the outlet is a and its coefficient of discharge is c, the quantity of air flowing out per minute is

$$C_2 = 5200\, a\, c\sqrt{p} \tag{32}$$

Now the air flows into the fan through the inlet with a velocity due to the pressure p_1, which according to (3) is equal to $5200\sqrt{p_1}$. And since as has been shown when discussing the blast area, the inlet area multiplied by its coefficient of discharge is equal to the blast area, A, divided by r, or $\frac{A}{r}$, the quantity of air entering the fan which is the same as the quantity which flows out of the housing, is

$$C_2 = \frac{5200\, A\sqrt{p_1}}{r} \tag{33}$$

but from (31) we have

$$p_1 = P + p_2 - p$$

putting this value of p_1 in (33) we have

$$(34) \qquad C_2 = \frac{5200\, A \sqrt{P + p_2 - p}}{r}$$

From (32) and (34) we have

$$5200\, a\, c \sqrt{p} = \frac{5200\, A \sqrt{P + p_2 - p}}{r}$$

Solving this equation for p, we get

$$(35) \qquad p = \frac{P + p_2}{1 + \dfrac{a^2\, c^2\, r^2}{A^2}}$$

Now P is the pressure due to the velocity in feet per minute of the tips of the floats, which is $\pi\, D\, N$; and p_2, is the pressure due to the velocity in feet per minute of points on the floats at the inlets, which is $\pi\, r\, D\, N$. Therefore we have from (5)

$$P = \left(\frac{\pi\, D\, N}{5200}\right)^2$$

$$p_2 = \left(\frac{\pi\, r\, D\, N}{5200}\right)^2$$

And from these we get

$$P + p_2 = \frac{\pi^2\, D^2\, N^2\, (1 + r^2)}{5200^2}$$

If we put this value of $P + p_2$ in (35) and solve for p we get

$$(36) \qquad p = \frac{\pi^2 D^2 N^2 (1 + r^2)}{5200^2 \left(1 + \frac{a^2 c^2 r^2}{A^2}\right)}$$

And finally this value of p in (32) gives as the expression for C_2

$$(37) \qquad C_2 = \pi D N a c \sqrt{\frac{1 + r^2}{1 + \frac{a^2 c^2 r^2}{A^2}}}$$

But from (22) we get

$$\pi D N = \frac{C}{A}$$

and this value of $\pi D N$ in (37) give

$$(38) \qquad C_2 = \frac{C a c}{A} \sqrt{\frac{1 + r^2}{1 + \frac{a^2 c^2 r^2}{A^2}}}$$

If the area of the outlet multiplied by its coefficient of discharge is made equal to the blast area, that is if $a c$ is made equal to A, we see from (38) that C_2 becomes equal to C.

Equation (38) shows that, if a fan wheel be run at such a speed as to give a capacity C against a certain pressure when the outlet is the blast area, the quantity of air delivered by the fan without any change of speed but with a larger outlet will be increased by the value of the fraction

$$\frac{a\,c}{A}\sqrt{\frac{1+r^2}{1+\dfrac{a^2\,c^2\,r^2}{A^2}}}$$

This fraction involves only the values of a, c, r, and A, and it is therefore possible to make a table giving its value for different values of $\frac{a\,c}{A}$ and r. If we call this fraction B, we have from (38)

$$C_2 = C\,B \qquad (39)$$

Table IV. gives the value of B for different values of $\frac{a\,c}{A}$ and for r equal to 0.5, 0.625 and 0.707.

TABLE IV.

$\frac{a\,c}{A}$	B for $r=0.5$	B for $r=0.625$	B for $r=0.707$
1.0	1.00	1.00	1.00
1.2	1.15	1.13	1.12
1.4	1.28	1.24	1.22
1.6	1.40	1.33	1.30
1.8	1.50	1.41	1.36
2.0	1.58	1.47	1.41

Table IV. shows how the value of r affects the value of B for the same value of $\frac{a\,c}{A}$ If, for instance, $\frac{a\,c}{A}$ is 1.6, the table shows that for r equal 0.5, that is the diameter of the inlet is one-half the diameter of the fan, B is 1.40; while if r be 0.707, B is 1.30. It must be carefully borne in mind that, as has been said so often before, when the outlet is increased so that $a\,c$ is greater than A, the quantity of air delivered by the fan will be increased and made greater than the capacity of the fan, but the pressure in the housing near the outlet will be less than that due to the velocity of the tips of the floats.

As an example, we may consider the case of a fan with an 8-foot wheel for which r is equal to 0.707. Such a fan, when $a\,c$ is equal to A, working against a pressure of 0.3 of an ounce, has a capacity according to Table II., of 28000 cubic feet per minute. That is to say, C is 28000. Now suppose the speed of the wheel is not changed at all, but the outlet is made larger so that $\frac{a\,c}{A}$ is equal to 1.4. From Table IV. we find that when $\frac{a\,c}{A}$ is 1.4, and r is 0.707, B is 1.30. And from (39) we get

$$\begin{aligned} C_2 &= C\,B \\ &= 28000 \times 1.30 \\ &= 36400 \end{aligned}$$

Air Per Revolution. It is sometimes convenient to be able to determine the number of cubic feet of air which a fan of a given diameter of wheel and inlet will deliver per revolution.

From (22) we know that when the theoretical area, $a\,c$, of the outlet is equal to the blast area, A, the number of cubic feet of air delivered is equal to the capacity of the fan and is

$$C = \pi D N A \tag{40}$$

From (29) we know that when $a\,c$ is *less* than A, the number of cubic feet of air delivered is

$$C_1 = \pi D N a c \tag{41}$$

And from (37) we know that when $a\,c$ is *greater* than A, the number of cubic feet delivered is

$$C_2 = \pi D N a c \sqrt{\frac{1 + r^2}{1 + \frac{a^2\, c^2\, r^2}{A^2}}} \tag{42}$$

It is evident that if we divide (40), (41) and (42) by N, the number of revolutions per minute of the fan wheel, we shall have the number of cubic feet of air delivered per revolution for the different relative values of $a\,c$, and A. Therefore, let q be the number of cubic feet of air delivered per revolution, and from (40), (41) and (42) we get

$$(43)\quad q = \begin{cases} \pi D A, \text{ when } a\,c \text{ equals } A; \\ \pi D a c, \text{ when } a\,c \text{ is less than } A; \\ \pi D a c \sqrt{\dfrac{1 + r^2}{1 + \dfrac{a^2 c^2 r^2}{A^2}}}, \text{ when } a\,c \text{ is greater than } A. \end{cases}$$

If we let $\pi D A$ be represented by q', and remember that in (39) we have designated the fraction $\dfrac{a\,c}{A}\sqrt{\dfrac{1 + r^2}{1 + \dfrac{a^2 c^2 r^2}{A^2}}}$ by the letter B, from (43) we get

$$(44)\quad q = \begin{cases} q', \text{ when } a\,c \text{ equals } A; \\ \dfrac{q' a c}{A}, \text{ when } a\,c \text{ is less than } A; \\ q' B, \text{ when } a\,c \text{ is greater than } A. \end{cases}$$

The value of B for different values of $\dfrac{a\,c}{A}$ is given in Table IV.

q' is the air delivered per revolution of the wheel when the fan is working at its capacity and depends upon the blast area of the fan wheel;

and the blast area, A, as has been shown, depends upon the ratio, r, of the diameter of the inlet to the diameter of the wheel. To determine q' for any given fan all that is necessary is to multiply the blast area of the wheel as given by Table III. by π times the diameter of the wheel. The blast area must, of course, be in square feet, and the diameter of the wheel must be in feet.

Table V. gives the value of q' for fans with wheels of different diameters, and for r, the ratio of the diameter of the inlet opening to the diameter of the wheel, equal to 0.625 and to 0.707.

TABLE V.

Cubic feet of air per revolution.

Diameter of wheel in feet.	$r=0.625$ q'	$r=0.707$ q'	Diameter of wheel in feet.	$r=0.625$ q'	$r=0.707$ q'
3	9.1	13.2	6½	92	133
3½	14.3	20.9	7	116	167
4	21.4	31.4	8	173	250
4½	31	44	9	245	355
5	42	61	10	335	485
5½	55	81	11	450	650
6	74	105	12	580	840

Table V. may be used to determine the number of revolutions at which the wheel of a fan must be run to deliver a given number of cubic feet of air per minute when the theoretical outlet area, $a\,c$, is equal to the blast area, A.

EXAMPLE:—What must be the speed of a fan wheel whose diameter is 7 feet, and whose inlet diameter is $59\frac{1}{2}$ inches, in order to deliver 25000 cubic feet of air a minute.

Here r is $\frac{59.5}{84}$ or 0.707. We shall assume that the theoretical outlet area is equal to the blast area. From Table V. we see that when r is 0.707 a 7-foot wheel will deliver 167 cubic feet of air per revolution. Hence to deliver 25000 cubic feet of air per minute a 7-foot wheel must make as many revolutions per minute as 167 is contained in 25000, that is

$$N = \frac{25000}{167}$$
$$= 150.$$

CHAPTER VI.

Pressure. When the area of the outlet multiplied by its coefficient of discharge is equal to or less than the blast area, the pressure in the housing is P, that corresponding to the velocity of the tips of the floats; and since the velocity of the tips of the floats is $\pi D N$, we have from (3)

$$\pi D N = 5200 \sqrt{P} \tag{45}$$

From this we get

$$D N = 1650 \sqrt{P} \tag{46}$$

and also

$$N = \frac{1650 \sqrt{P}}{D} \tag{47}$$

EXAMPLE:—Determine the number of revolutions per minute which must be made by a fan with a 6-foot wheel to give a pressure of 0.6 of an ounce when the theoretical area of the outlet is equal to or less than the blast area.

Here P is 0.6 and D is 6, so that

$$N = \frac{1650 \sqrt{0.6}}{6}$$
$$= 213$$

If (46) be solved for P we get

$$(48) \qquad P = \left(\frac{1650}{D\ N}\right)^2$$

This equation enables us to determine the pressure in ounces per square inch in the housing of a fan when we know the diameter of the wheel and the number of revolutions made by it per minute.

Example.—Determine the pressure in ounces per square inch in the housing of a fan having an 8-foot wheel running at 150 revolutions per minute, when the theoretical outlet is equal to or less than the blast area.

Here D is 8, N is 150, and, therefore,

$$P = \left(\frac{8 \times 150}{1650}\right)^2$$
$$= 0.53$$

Table VI. is calculated from (47) and gives the number of revolutions per minute at which wheels of different diameters must be run in order to give various pressures in ounces per square inch when the theoretical outlet area is equal to or less than the blast area of the wheel.

TABLE VI.

Number of revolutions for different pressures.

Diam. of wheel in feet.	Pressure per square inch in ounces.									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
3	174	246	301	348	389	426	460	492	522	550
3½	149	211	258	293	334	365	394	422	447	471
4	130	185	226	261	292	320	345	369	391	413
4½	116	164	201	232	259	284	307	328	348	366
5	104	148	181	209	234	256	276	295	313	330
5½	95	134	164	190	212	232	251	268	284	300
6	87	123	151	174	195	213	230	246	261	275
6½	80	114	139	161	180	197	212	227	241	254
7	75	105	129	149	167	183	197	211	224	236
8	65	92	113	131	146	160	173	184	196	206
9	58	82	101	116	130	142	154	164	174	183
10	52	74	91	105	117	128	138	148	157	165
11	47	67	82	95	106	116	126	134	142	150
12	44	62	75	87	97	108	115	123	131	138

Table VI. should be used in connection with Tables II. and IIA. From Tables II. and IIA we can find the size of wheel required to deliver a given quantity of air per minute at a given pressure, then from Table VI. we can find the number of revolutions the wheel must make per minute.

EXAMPLE:—Determine the diameter of wheel required to deliver 50000 cubic feet of air per minute against a pressure of 0.7 of an ounce per square inch. The diameter of the inlet is 0.707 of the diameter of the wheel.

Turning to Table II. we find that a 9-foot wheel will deliver only 54200; and a 10-foot wheel will deliver 67000 cubic feet. We shall use, therefore, a 9-foot wheel. Turning now to Table VI., we find that a 9-foot wheel must make 154 revolutions per minute to give a pressure of 0.7 of an ounce.

When the area of the outlet multiplied by its coefficient of discharge is greater than the blast area, that is when *a c* is greater than *A*, we have from (36) that the pressure in the housing is

$$(49) \qquad p = \frac{\pi^2 D^2 N^2 (1 + r^2)}{5200^2 \left(1 + \dfrac{a^2 c^2 r^2}{A^2}\right)}$$

From which we get

$$(50) \qquad D N = 1650 \sqrt{\frac{p\left(1 + \dfrac{a^2 c^2 r^2}{A^2}\right)}{1 + r^2}}$$

$$(51) \qquad N = \frac{1650}{D} \sqrt{\frac{p\left(1 + \dfrac{a^2 c^2 r^2}{A^2}\right)}{1 + r^2}}$$

From (45) we get, by solving for *P*,

$$P = \frac{\pi^2 D^2 N^2}{5200^2}$$

Hence substituting in (49) for $\frac{\pi^2 D^2 N^2}{5200^2}$ its value we have

$$p = \frac{P\,(1+r^2)}{1 + \frac{a^2\,c^2\,r^2}{A^2}} \tag{52}$$

Equation (52) shows the effect on the pressure of increasing the outlet area without changing the speed or number of revolutions per minute. That is to say if a fan having a wheel of a given diameter is run at a certain number of revolutions, it will give a pressure in ounces per square inch equal to P when the outlet is equal to the blast area; but if the outlet is made larger than the blast area and the wheel be run at the same speed as before, the pressure then will be p as given by (52). In using (52) it must be remembered that P is the pressure the wheel would give if the theoretical outlet area $a\,c$ were equal to the blast area, and the value of P to be used in (52) must be calculated by (48) or taken from Table VI.

If we let F represent the fraction $\frac{1+r^2}{1 + \frac{a^2\,c^2\,r^2}{A^2}}$, (51) becomes

$$N = \frac{1650}{D}\sqrt{\frac{p}{F}} \tag{53}$$

and (52) becomes

$$p = P\,F \tag{54}$$

Table VII. gives the value of F for different values of $\frac{ac}{A}$ and for r equal 0.5, 0.625 and 0.707.

TABLE VII.

$\frac{ac}{A}$	Value of F for		
	$r=0.5$	$r=0.625$	$r=0.707$
1.0	1.00	1.00	1.00
1.2	0.92	0.89	0.87
1.4	0.84	0.79	0.76
1.6	0.76	0.70	0.66
1.8	0.69	0.61	0.57
2.0	0.63	0.54	0.50

EXAMPLE:—Determine the speed at which a 9-foot wheel must be run in order to give a pressure per square inch of 0.5 an ounce when the outlet is equal to the blast area, and also when the outlet is 1.6 times the blast area. Assume that the diameter of the inlet is 0.625 of the diameter of the wheel.

Here we have D is 9 feet; p is 0.5; $\frac{ac}{A}$ is 1.6; and r is 0.625. The number of revolutions the wheel must make when the outlet is equal to the blast area may be calculated by (47) or may be found from Table VI. From Table VI. we find that a 9-foot wheel must make 130 revolutions per minute to give a pressure of 0.5 of an ounce when the theoretical outlet is equal to the blast area.

Table VII. shows that when $\frac{a\,c}{A}$ is 1.6, and r is 0.625, F is 0.70.

Hence from (53) we have

$$N = \frac{1650}{D}\sqrt{\frac{p}{F}}$$

$$= \frac{1650}{9}\sqrt{\frac{0.50}{0.70}}$$

$$= 155$$

Equation (54) enables us to determine the effect on the pressure given by a fan wheel of increasing the outlet area without changing the number of revolutions. Thus in the last example we saw that a 9-foot wheel for which r is 0.625, would give a pressure of 0.5 of an ounce when making 130 revolutions. We also saw from Table VII. that for r equal to 0.625, and $\frac{a\,c}{A}$ equal to 1.6, F was 0.70. Now when F is 0.70 and P is 0.50, equation (54) shows that a 9-foot wheel making 130 revolutions and the theoretical area of the outlet equal to 1.6 the blast area, will give a pressure only of

$$p = PF = 0.5 \times 0.70$$

$$= 0.35$$

In most fan problems which arise in actual work the outlet of the fan housing is usually connected to a chamber in which it is desired a certain pressure shall be maintained. In all of such cases the outlet to be considered in calculating the value of $a\,c$ to be used in the equations for p, is *not the outlet from the fan housing* but the *outlet from the chamber* in which the pressure is desired. The outlet to be considered is always the one leading into the atmosphere. The pressure in the chamber will be somewhat less than in the housing because of losses due to friction, shape of housing, connections of duct between the housing and the chamber, etc., but ordinarily the chamber is located so close to the fan that the loss of pressure between the housing and the chamber is small.

Tables IV. and VII. enable us to determine the full effect of increasing the outlet of a fan so as to make the theoretical outlet greater than the blast area. For instance, Table IV. shows that if the outlet of a fan for which r is 0.625, be increased so that $\frac{a\,c}{A}$ is 1.2, the number of cubic air discharged per minute will be 1.13 what it is when the outlet is equal to the blast area; and Table VII. shows that the pressure will be only 0.89 of what it is when the outlet is equal to the blast area. In other words, to increase the outlet of a fan for which *r* is 0.625 so that the theoretical area is 20 per cent. greater than the blast area brings about an increase of 13 per cent. in the quantity of air delivered per minute by the fan with a decrease of 11 per cent. in the pressure. If *r* were 0.5, to increase the outlet so as to make it 20 per cent. greater than the blast area would bring about an increase of 15 per cent. in the quantity of air delivered and a decrease of only 8 per cent. in the pressure. While if *r* were 0.707, a corresponding increase in the outlet would mean an increase of 12 per cent. in the quantity of air discharged with a decrease of 13 per cent. in the pressure.

The pressure which can be produced by a centrifugal fan or the pressure under which air or gases can be delivered, is limited only by the velocity at which the wheel can be safely run; and this is limited by the materials used in the construction of the wheel and the care and skill with which the parts are put together.

Work. The work done per minute in moving air is equal to the product of the number of cubic feet of air moved per minute multiplied by the sum of the pressure in the housing near the outlet and the pressure corresponding to the velocity per minute of the air as it enters the fan. The pressures must be not ounces per square inch but pounds per square foot; and if the velocity of the air through the inlet is less than the radial velocity at the inlet, then the pressure corresponding to the radial velocity must be taken instead of the pressure corresponding to the velocity through the inlet.

As before, let p be the pressure in ounces per square inch in the housing; P the pressure in ounces per square inch corresponding to the velocity of the tips of the floats of the wheel; p_1, the pressure in ounces per square inch corresponding to the velocity through the inlet or the radial velocity at the inlet, whichever is the larger; and p_2, the pressure in ounces per square inch corresponding to the velocity of the points of the floats at the inlet. Then the sum of the pressure in the housing and the pressure corresponding to the velocity through the inlet is $p+p_1$. This sum, however, is in ounces per square inch; and to reduce it to pounds per square foot it must be divided by 16, since there are sixteen ounces to a pound, and multiplied by 144, since there are 144 square inches to a square foot. Therefore, the sum of the pressure in the housing and the pressure corresponding to the velocity through the inlet, when expressed in pounds per square foot is

$$\frac{144\ (p + p_1)}{16} = 9\ (p+p_1)$$

But it has been shown before that when the outlet is equal to the blast area, the number of cubic feet of air discharged by the fan per minute is the capacity C; p is equal to P; and p_1 is equal to p_2. Hence, the work done per minute is

$$9\,C\,(p+p_1) = 9\,C\,(P+p_2)$$

But from (48) we know that

$$P = \left(\frac{D\,N}{1650}\right)^2$$

and since the velocity through the inlet is $\pi\,r\,D\,N$, we have

$$p_2 = \left(\frac{\pi\,r\,D\,N}{5200}\right)^2$$

$$= r^2\left(\frac{D\,N}{1650}\right)^2 = r^2\,P$$

Therefore $P+p_2$ is equal to $P\,(1+r^2)$ and the expression for the work done per minute becomes

$$9\,C\,(P+p_2) = 9\,C\,P\,(1+r^2)$$

Now the horse power expended on the air is equal to the work done on it per minute divided by 33000. And if we designate this horse power by K, we have

$$(55) \qquad K = \frac{9\,C\,P\,(1+r^2)}{33000}$$

$$= \frac{C\,P\,(1+r^2)}{3670}$$

It must be remembered that K is only the horse power required to move the air and does not include any part of the power required to overcome the friction of the wheel in its bearings.

If we know the diameter of the wheel and the pressure against which the air is to be delivered we find C, the capacity, from Table II. if r is 0.707 or from Table IIA if r is 0.625. Then knowing C, P, and r we find the horse power expended on the air by (55).

EXAMPLE:—Determine the horse power required to move the air delivered by a fan for which r is 0.625 and whose wheel is 8 feet in diameter, when working at its capacity against 0.6 of an ounce per square inch.

From Table IIA we see that the capacity of an 8-foot wheel working against 0.6 of an ounce is 27200 cubic feet per minute. Hence we have that C is 27200; P is 0.6; and since r is 0.625,

$1 + r^2$ is 1.39. Therefore from (55) we get that the horse power expended on the air is

$$K = \frac{C\,P\,(1+r^2)}{3670}$$

$$= \frac{27200 \times 0.6 \times 1.39}{3670}$$

$$= 6.18$$

If we do not know P, but do know the number of revolutions the wheel makes per minute we must find P from Table VI., then find C from either Table II. or Table IIA, and then proceed as before.

EXAMPLE:—Determine the horse power required to move the air by a fan for which r is 0.707, and whose wheel is 7 feet in diameter and is making 150 revolutions per minute. The outlet is supposed to be equal to the blast area.

From Table VI. we see that a 7-foot wheel making 149 revolutions per minute gives a pressure of 0.4 of an ounce per square inch. Hence P is 0.4.

From Table II. we see that for a 7-foot wheel working against 0.4 of an ounce pressure, C is 24800.

And since r is 0.707, r^2 is 0.5 and $1 + r^2$ is 1.5. Hence

$$K = \frac{C P (1+r^2)}{3670}$$

$$= \frac{24800 \times 0.4 \times 1.5}{3670}$$

$$= 4.06$$

When the outlet area is less than the blast area the quantity of air discharged is less than the capacity, C, and is what we have called C_1. And since the quantity of air discharged by the fan is less than C, the velocity of the air through the inlet is less than $\pi\, r\, D\, N$ and the pressure corresponding to this velocity is less than $\left(\frac{\pi\ r\ D N}{5200}\right)^2$.

From (30) we know that when a is the area of the outlet in square feet, c its coefficient of discharge, and A the blast area, the expression for C_1, is

$$C_1 = \frac{a\, c\, C}{A}$$

The velocity through the inlet is directly proportioned to the quantity of air passing through it, so that when the quantity of air passing through the fan is C_1, the velocity through the inlet is

$\frac{a\,c\,\pi\,r\,D\,N}{A}$. And the pressure corresponding to this velocity is

$$\left(\frac{a\,c\,\pi\,r\,D\,N}{5200}\right)^2 = \left(\frac{D\,N}{1650}\right)^2 \frac{a^2\,c^2\,r^2}{A^2}$$

$$= \frac{P\,a^2\,c^2\,r^2}{A^2}$$

Therefore, the work done on the air alone when the theoretical area of the outlet is less than the blast area is equal to 9 times the quantity of air discharged per minute multiplied by the sum of P and the pressure $\frac{P\,a^2\,c^2\,r^2}{A^2}$ which corresponds to the velocity at the inlet. And this work divided by 33000, is the horse power expended on the air when $a\,c$ is less than the blast area A. Calling this work K_1 we have

$$K_1 = \frac{9\,C_1\,P\left(1 + \frac{a^2\,c^2\,r^2}{A^2}\right)}{33000}$$

(56) $$= \frac{C_1\,P\left(1 + \frac{a^2\,c^2\,r^2}{A^2}\right)}{3670}$$

Inasmuch as we seldom care to know the horse power necessary to move the air when the theoretical outlet area is less than the blast area it is not necessary to discuss (56). In actual practice we are seldom bothered about the horse power required to move the air unless the fan is working up to its capacity, because upon this work depends the size of the motor or engine which must be installed to run the fan.

When the theoretical outlet area is greater than the blast area the quantity of air discharged per minute is C_2; the pressure in ounces per square inch in the housing is p; and the pressure in ounces per square inch corresponding to the velocity through the inlet is p_1. Hence the work done on the air is $9\,C_2\,(p+p_1)$

But from (31) we have

$$p = P + p_2 - p_1$$

and, therefore,

$$p + p_1 = P + p_2$$

As has been shown before, $P + p_2$ is equal to $P\,(1+r^2)$. Hence the work done on the air is $9\,C_2\,P\,(1+r^2)$ and the horse power, K_2, exerted on the air is

(57) $$K_2 = \frac{9 C_2 P (1 + r^2)}{33000}$$

$$= \frac{C_2 P (1 + r^2)}{3670}$$

The expression for K_2 is very similar to the expression for K, the horse power when the fan is working at its capacity; the only difference being that in the expression for K we use C, while in the expression for K_2 we use C_2. If we divide (57) by (55) and solve for K_2 we get

(58) $$K_2 = \frac{C_2 K}{C}$$

From (39) we find that $C_2 = B\,C$, and therefore (58) becomes

(59) $$K_2 = B\,K$$

B, as has been explained before, depends upon the ratio, r, of the diameter of the inlet to the diameter of the wheel, and the ratio of the theoretical outlet area, $a\,c$, to the blast area A. Values of B are given in Table IV.

CHAPTER VII.

Horse Power Required to Run a Fan. The horse power required to run a fan is the sum of the horse power required to move the air and the horse power required to overcome the resistance to the wheel as it revolves in the housing. The resistance to the wheel as it revolves is made up of the friction of the shaft in the bearings and of the friction of revolution due to the air between the wheel and the housing. It is extremely difficult to determine the resistance to the wheel as it revolves, as it depends upon the bearings, and upon the fit of the wheel in the housing, and upon the size and weight of the shaft and wheel; and it is usually for each fan a constant per revolution of the wheel. That is the resistance at 100 revolutions per minute of the wheel is usually about twice what it is at 50 revolutions. It is probably safe, and near enough to the actual truth for all practical purposes, to say that the horse power required to overcome the resistance to the turning of the wheel is about 10 per cent. of the total power required to run the fan when it is working at its capacity or above. This means that the mechanical efficiency of the fan is about 90 per cent. and that K as given by (55) is 0.9 of the total horse power required to run the fan when it is working at its capacity.

If we call H the horse power required to run a fan when its theoretical outlet area is equal to the blast area, that is when the fan is working at its capacity, we have

$$(60) \qquad H = \frac{K}{0.9} = \frac{C\,P\,(1 + r^2)}{3300}$$

This equation has been used to calculate Tables VIII. and VIIIA.

Table VIII. gives the horse power required to run a fan when working at its capacity under different pressures, when the diameter of the inlet is 0.707 the diameter of the wheel, or when r is 0.707.

Table VIIIA gives the horse power required to run a fan when working at its capacity under different pressures, when the diameter of the inlet is 0.625 the diameter of the wheel, or when r is 0.625.

TABLE VIII.

Horse power required for centrifugal fans.

Diam. of wheel in feet.	Pressure in ounces per square inch.									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
3	0.10	0.29	0.54	0.83	1.16	1.52	1.92	2.34	2.80	3.28
3½	0.14	0.40	0.73	1.13	1.58	2.07	2.61	3.18	3.81	4.45
4	0.18	0.52	0.96	1.47	2.06	2.72	3.40	4.14	4.95	5.82
4½	0.23	0.66	1.21	1.85	2.62	3.41	4.33	5.27	6.31	7.36
5	0.29	0.81	1.50	2.31	3.20	4.23	5.32	6.51	7.77	9.10
5½	0.35	0.98	1.80	2.78	3.89	5.10	6.43	7.85	9.42	11.0
6	0.41	1.17	2.16	3.31	4.64	6.08	7.67	9.38	11.2	13.1
6½	0.49	1.37	2.53	3.89	5.43	7.15	9.01	11.0	13.2	15.4
7	0.56	1.59	2.93	4.51	6.30	8.30	10.4	12.7	15.2	17.8
8	0.74	2.08	3.82	5.89	8.32	10.8	13.6	16.7	19.9	23.3
9	0.93	2.64	4.82	7.46	10.4	13.7	17.3	21.1	25.2	29.5
10	1.15	3.26	5.97	9.20	12.9	16.9	21.3	26.0	31.1	36.4
11	1.39	3.94	7.23	11.1	15.5	20.5	25.8	31.5	37.6	44.0
12	1.66	4.70	8.62	13.3	18.5	24.4	30.7	36.6	44.6	52.3

TABLE VIIIA.

Diam. of wheel in feet.	Pressure in ounces per square inch.									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
3	0.07	0.19	0.34	0.53	0.74	0.97	1.22	1.49	1.79	2.09
3½	0.09	0.25	0.47	0.72	1.00	1.32	1.66	2.04	2.43	2.84
4	0.12	0.33	0.61	0.94	1.32	1.73	2.17	2.64	3.16	3.72
4½	0.15	0.42	0.77	1.19	1.67	2.18	2.76	3.36	4.03	4.68
5	0.18	0.52	0.96	1.47	2.04	2.70	3.39	4.15	4.97	5.78
5½	0.22	0.63	1.15	1.77	2.49	3.26	4.10	4.99	6.00	7.00
6	0.26	0.75	1.38	2.11	2.95	3.87	4.90	5.97	7.14	8.35
6½	0.31	0.88	1.61	2.48	3.46	4.55	5.75	7.03	8.38	9.78
7	0.36	1.01	1.87	2.87	4.01	5.28	6.64	8.13	9.72	11.3
8	0.47	1.33	2.44	3.76	5.25	6.88	8.68	10.6	12.7	14.9
9	0.59	1.68	3.08	4.76	6.65	8.75	11.0	13.5	16.1	18.8
10	0.73	2.08	3.82	5.88	8.21	10.8	13.6	16.6	19.8	23.2
11	0.89	2.52	4.61	7.10	9.90	13.1	16.5	20.1	24.0	28.1
12	1.06	3.00	5.51	8.48	11.8	15.5	19.6	23.3	28.5	33.4

When the theoretical outlet area is less than the blast area, that is when $a\,c$ is less than A, the horse power required to run the fan is equal to the horse power K_1, required to move the air, as given by (56), plus the horse power required to overcome the resistance to turning the wheel. If we assume the resistance to turning the wheel to be 10 per cent. of the horse power required to run the fan when working at its capacity, or 0.1 of H as given by (60), the horse power, H_1, required to run the fan will be

$$(61) \qquad H_1 = K_1 + 0.1\,H$$

$$= \frac{C_1\,P\left(1 + \dfrac{a^2\,c^2\,r^2}{A^2}\right)}{3670} + 0.1\,H$$

When the theoretical outlet area is greater than the blast area, that is when $a\,c$ is greater than A, the horse power, which we may call H_2, required to run the fan will be the horse power K_2, as given by (59), required to move the air plus the horse power required to overcome the resistance to turning of the wheel. We are probably safe in assuming here, as before, that the mechanical efficiency of the fan is about 90, or that K_2 is about 0.9 of H_2. Hence from (59) we have

$$(62) \qquad H_2 = \frac{K_2}{0.9} = \frac{B\,K}{0.9}$$

But from (60) we know that

$$\frac{K}{0.9} = H$$

and therefore (62) becomes

$$(63) \qquad H_2 = B\,H = \frac{C\ P\ (1 + r^2)\ B}{3300}$$

The value of H for a wheel whose inlet has a diameter equal to 0.707 or 0.625 of the diameter of the wheel, may be obtained from Table VIII. or Table VIIIA, and the value of B for various ratio of theoretical area of outlet to blast area may be obtained from Table IV.

EXAMPLE:—Determine the horse power required to run a fan whose inlet is 37½ inches in diameter, and whose wheel is 5 feet in diameter, when running at 275 revolutions per minute with a theoretical outlet area equal to the blast area; and, also, when the theoretical outlet area is 1.4 the blast area.

Here we have that r, the ratio of the diameter of the inlet to the diameter of the wheel, is

$$r = \frac{37.6}{60} = 0.625$$

From Table VI. we find that when a 5-foot wheel is making 275 (exactly 276 in the Table) revolutions per minute the pressure against which it is working, when the outlet is equal to the blast area, is 0.7 of an ounce per square inch.

And from Table VIIIA we find that a fan with a 5-foot wheel working at its capacity against a pressure of 0.7 of an ounce, requires 3.4 horse power to run it.

When the theoretical outlet area is made 1.4 the blast area we find from Table IV. that for r equal to 0.625, B is 1.24. We have just found that H is 3.4. Hence the horse power H_2, required to run the fan when the theoretical outlet area is 1.4 the blast area is from (63)

$$H_2 = B\,H$$

$$1.24 \times 3.4$$

$$= 4.2$$

It is interesting to look into the performance of this fan as to delivering air under the two conditions of outlet openings.

From Table IIA we see that a 5-foot wheel working at its capacity under a pressure of 0.7 of an ounce, delivers 16,700 cubic feet of air per minute. When the theoretical area of the outlet is greater than the blast area, we know from (39) that the quantity of air delivered, C_2, is B times the capacity as given by Table IIA. And from Table IV. we find that when $\frac{a\,c}{A}$ is 1.4, as in the example, B is 1.24 for r equal 0.625. Therefore, in this case

$$C_2 = B\,C = 1.24 \times 16700$$

$$= 20700$$

We also know that when the theoretical outlet area is 1.4 the blast area, the pressure in the housing at the outlet is, from (54), equal to $F\,P$. In this case P is 0.7 of an ounce per square inch, and F, from Table VII. for r equal 0.625, is 0.79.

Hence

$$p = F\,P = 0.79 \times 0.7$$

$$= 0.55$$

Therefore when the theoretical outlet area is equal to the blast area, we find that for this fan making 275 revolutions per minute, the pressure in the housing is 0.7 ounces per square inch; the cubic feet of air delivered per minute is 16700; the horse power required to run the fan is 3.4.

When the theoretical outlet area is 1.4 the blast area we find that, the pressure in the housing is 0.55 ounces per square inch; the cubic feet of air delivered per minute is 20700; the horse power required to run the fan is 4.2.

Engine Required to Run a Fan. The horse power of the engine required for a fan is equal to the horse power required to run the fan divided by the efficiency of the engine. The efficiency of engines used to run fans is usually between $\frac{2}{3}$ and $\frac{3}{4}$ for engines belted to the fans and between $\frac{1}{2}$ and $\frac{2}{3}$ for those direct connected to the fans. If we call the horse power of the engine E and assume that the efficiency is $\frac{2}{3}$, a fairly good average value, we have from (60) that the engine required to run a fan when working at its capacity is

$$(64) \qquad E = \frac{3\,H}{2} = \frac{C\,P\,(1 + r^2)}{2200}.$$

In the same way if we call E_2 the horse power of the engine required to run a fan when the theoretical area of the outlet is greater than the blast area, we find from (63) that

$$(65) \qquad E_2 = \frac{3\,H_2}{2} = \frac{3\,H\,B}{2}$$

$$= \frac{C\,P\,(1 + r^2)\,B}{2200}$$

When choosing an engine to run a fan it will usually be sufficient to choose one to run the fan when working at its capacity; and if proper care be used to see that the engine chosen is not too small to easily run the fan when working at its capacity, it will usually be large enough to run the fan when the outlet area is somewhat greater than the blast area. This method of chosing an engine to run a fan is a safe one in view of the fact that a fan is usually chosen of such a size that it will deliver the required quantity of air against a given pressure when working at its capacity. It is not necessary to bother about the size of engine required to run a fan when its theoretical outlet area is less than the blast area, since if the engine is large enough to run the fan at its capacity there will be no trouble when the fan is working at less than its capacity.

When the engine is belted to the fan or the fan is " belt connected " the problem of selecting the engine resolves itself into merely selecting that engine which will give the desired horse power when using steam at the pressure to be carried in the boiler. In this case the speed at which the engine is to run is a matter of no consequence, as whatever this speed may be the speed of the fan may be made anything desired by choosing the pulleys on the fan and engine of the proper sizes. When, however, the fan and engine are mounted on the same shaft or the fan is " direct connected," the problem becomes somewhat more difficult as the matter of torque or turning effort of the engine enters very largely. If a fan is direct connected to an engine, when steam is turned on and the engine started, the engine will speed up gradually and since the work done by the fan will gradually increase as the speed increased there will soon be reached a speed which is the maximum possible for the engine with the given conditions of steam pressure, cut-off, and back pressure.

If in (60) we make r equal to 0.707 we get that the power required to run a fan whose ratio of diameter of inlet to diameter of wheel is 0.707, when working at its capacity, is

$$H = \frac{C\,P}{2200} \tag{66}$$

From the well-known formulas in regard to steam engines* we know that the indicated horse power developed by a double acting engine whose length of stroke is l inches and whose diameter of cylinder is d inches, when making N revolutions minute and using steam whose mean effective pressure is P' pounds per square inch, is

$$\frac{2\,P'\,l\,\pi\,d^2\,N}{12 \times 4 \times 33000} = \frac{P'\,l\,d^2\,N}{251000}$$

*See Steam Engines and Boilers, by J. H. Kinealy.

Now let y be the efficiency of the engine, then the indicated horse power of the engine multiplied by y must equal the horse power required to run the fan. That is to say

$$(67) \qquad H = \frac{y\,P'\,l\,d^2\,N}{251000}$$

And from (66) we get

$$(68) \qquad \frac{C\,P}{2200} = \frac{y\,P'\,l\,d^2\,N}{251000}$$

From (15) we get by making r equal 0.707,

$$C = 0.49\,D^3\,N$$

where as before D is the diameter of the fan wheel in feet and N is the number of revolutions per minute at which it is run.

This value of C in (68) gives,

$$(69) \qquad \frac{0.49\,D^3\,P}{2200} = \frac{y\,P'\,l\,d^2}{251000}$$

From this we get

$$(70) \qquad l\,d^2 = \frac{60\,D^3\,P}{y\,P'}$$

y is seldom greater than $\frac{2}{3}$ and it may often be only $\frac{1}{2}$. In the case of an engine run by steam under 20 pounds by the gauge it is best to assume that y is $\frac{1}{2}$. Then (70) becomes

$$(71) \qquad l\,d^2 = \frac{120\,D^3\,P}{P'}$$

Direct connected engines seldom have governors, and they are usually of the throttling type and arranged to cut-off at about $\frac{2}{3}$ or $\frac{3}{4}$ the stroke. When the exhaust from the engine is not used for heating purposes so that there is no unusual back pressure on the engine we may say that the mean effective pressure P' of the steam in the engine is equal to 0.9 the boiler pressure, less 2. That is, if P'' is the boiler pressure and the exhaust steam is not used for heating purposes,

$$P' = 0.9\,P'' - 2$$

If the exhaust steam is used for heating purposes there will usually be a back pressure of about 2 pounds and then

$$P' = 0.9\,P'' - 4$$

Table IX. gives the mean effective pressure which may be expected in ordinary direct connected engines for different boiler pressures.

TABLE IX.

Mean effective pressures.

Boiler pressure in pounds.	Mean effective pressure when the exhaust is	
	not used for heating.	used for heating.
10	7.0	5.0
15	11.5	9.5
20	16.0	14.0
25	20.5	16.5
30	25.0	23.0
40	34.0	32.0
50	43.0	41.0
75	65.5	63.5
100	88.0	86.0

In the case of engines used with low pressure heating plants where the boiler pressure is about 15 pounds it is usually safe to say that when the exhaust steam is not used for heating purposes

$$l\,d^2 = 10\,D^3\,P$$

And when the exhaust steam is used for heating purposes

$$l\,d^2 = 12\,D^3\,P$$

EXAMPLE:—What should be the size of engine direct connected to a fan having a $6\frac{1}{2}$-foot wheel in order to run it at a $\frac{1}{2}$ ounce pressure with steam in the boiler at 15 lbs., if the exhaust is used for heating?

Table IX. shows that the mean effective pressure is 9.5 when the exhaust is used for heating and the boiler pressure is 15.

And since P is $\frac{1}{2}$ by the conditions of the problem, we have from (71)

$$l\,d^2 = \frac{120\,D^3\,P}{P'}$$

$$= \frac{120 \times 6\tfrac{1}{2}^3 \times \tfrac{1}{2}}{9.5} = 1740$$

Now we must choose from the catalogue list of sizes of the engine we desire to use one whose length of stroke in inches multiplied by the square of the diameter of the cylinder in inches will be equal to 1740.

In this particular case if l equal 14 and d equal 11, we have

$$l\,d^2 = 14 \times 121 = 1694$$

Hence an 11 by 14 engine would be the proper one to use.

Motor Required to Run a Fan. Electric motors are usually supposed to be rated according to the power they will deliver on their own pulleys, which means that the horse power of the motor required for a fan would be the same as the horse power required to run the fan plus the slight loss between the fan and the motor. To make this assumption, however, when choosing a motor for a fan would mean "figuring" closer than can be considered good engineering. The writer usually assumes that the efficiency of transmission between the motor and the fan is about 75 per cent., which makes the horse power of the motor equal the horse power required to run the fan divided by 0.75. On this assumption the horse power of the motor, which we may call M for a fan working at its capacity is from (60)

$$(72) \qquad M = \frac{H}{0.75} = \frac{4\,H}{3}$$

$$= \frac{C\,P\,(1 + r^2)}{2450}$$

In the same way the horse power, M_2, of the motor for a fan when its theoretical outlet area is greater than the blast area is from (62)

$$(73) \qquad M_2 = \frac{H_2}{0.75} = \frac{4\,H\,B}{3}$$

$$= \frac{C\,P\,(1 + r^2)\,B}{2450}$$

The same remarks in regard to choosing an engine for a fan, apply to choosing a motor. It is much better to have the motor too large than too small, and when there is any doubt as to the power required to run the fan it is well to err on the side of safety and choose a motor of such a size that it can easily run the fan. Another reason for choosing a motor or engine of liberal size for a fan is that it often happens that for one reason or another it becomes necessary to speed up the fan, thus increasing the pressure against which it works or the quantity of air delivered per minute or both. And each one of these increases results in an increase in the horse power required to run the fan, and hence an increased demand on the motor or engine.

If the motor is direct connected to the fan it will, of course, have to run at the same speed as the fan and this will usually result in necessitating a motor of much larger size than would be required if the motor were belted to the fan. If the speed of a motor be decreased and made less than its normal speed or less than the speed for which the motor was originally designed, the power devoloped by the motor will be decreased in almost the same ratio as the speed. That is to say if a 10 horse power motor be designed to run at 800 revolutions a minute and it be desired that this motor be run at 200 revolutions instead of 800 then the motor instead of developing 10 horse power would develope about one-fourth of 10 horse power or 2½ horse power. When motors are direct connected to fans they are usually obliged to run at much less speed than if they are belted and hence because of the larger size motor required a direct connected motor is much more expensive for a fan than a belted motor. Thus, if a fan requires 8 horse power to drive it at 160 revolutions a minute which is about one-fifth of the speed at which a 8 horse power would ordinarily be expected to run, it would be necessary to get what would at an ordinary speed of about 800 revolutions a minute be a 40 horse power motor. That is because the speed of the motor is only about one-fifth that at which it would ordinarily be run, it would be necessary to get a motor about 5 times as large. This motor would not develop 40 horse power at 160 revolutions but would develop about 8 horse power and the cost of the motor would be pretty nearly the same as the ordinary 40 horse power motor.

A direct connected motor for a fan may usually be expected to cost from 4 to 6 times as much as a belted motor.

Width of Belt. As a matter of convenience in determining the size or width of a leather belt required to drive a fan, Tables X and XA have been calculated.

Table X is for single thick belts, and is based upon the supposition that one inch of width of belt

TABLE X.

Horse power transmitted by single belts.

Width of belt in in.	Speed of belt in feet per minute.									
	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
2	1.3	2.5	3.8	5.0	6.3	7.5	8.8	10.0	11.2	12.5
2½	1.6	3.1	4.7	6.3	7.8	9.4	11.0	12.5	14.1	15.6a
3	1.9	3.8	5.6	7.5	9.4	11.2	13.1	15.0	16.9	18.8
4	2.5	5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0
5	3.1	6.3	9.4	12.5	15.6	18.8	21.9	25.0	28.1	31.3
6	3.8	7.5	11.2	15.0	18.8	22.5	26.2	30.0	33.8	37.5
7	4.4	8.8	13.1	17.5	21.9	26.3	30.6	35.0	39.4	43.8
8	5.0	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0
9	5.6	11.2	16.9	22.5	28.1	33.7	39.3	45.0	50.6	
10	6.3	12.5	18.8	25.0	31.2	37.5	43.7	50.0		

TABLE XA.

Horse power transmitted by double belts.

Width of belt in in.	Speed of belt in feet per minute.									
	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
2	1.8	3.6	5.5	7.3	9.1	10.9	12.7	14.5	16.4	18.2
2½	2.3	4.5	6.8	9.1	11.4	13.6	15.9	18.2	20.5	22.8
3	2.7	5.5	8.2	10.9	13.6	16.4	19.1	21.8	24.6	27.3
4	3.6	7.3	10.9	14.5	18.2	21.8	25.4	29.1	32.7	36.4
5	4.5	9.1	13.6	18.2	22.7	27.3	31.8	36.4	40.9	45.5
6	5.5	10.9	16.3	21.8	27.2	32.7	38.2	43.6	49.1	
7	6.4	12.7	19.1	25.4	31.8	38.2	44.5	50.9		
8	7.3	14.5	21.8	29.1	36.3	43.6	50.9			
9	8.2	16.4	24.5	32.7	40.9	49.1				
10	9.1	18.2	27.2	36.3	45.4	54.5				

will transmit one horse power when travelling at a speed of 800 feet per minute.

Table XA is for double thick belts, and is based upon the supposition that one inch of width of belt will transmit one horse power when travelling at a speed of 550 feet per minute.

To use either table X or XA it is necessary to know the speed of the belt in feet per minute, and this is obtained by multiplying the circumference of the pulley on which the belt runs by the number of revolutions it makes per minute. The circumference must be in feet, and is always equal to π times the diameter in feet. But it is usual to express the sizes or diameters of pulleys in inches, and hence the diameter in feet will be equal to the size or diameter in inches divided by 12. Therefore, if d is the diameter of the pulley in inches, the circumference in feet will be

$$\frac{\pi d}{12} = \frac{3.14d}{12}$$

$$= \frac{d}{3.82}$$

For all practical purposes connected with the calculation of belts it is sufficiently exact to say that the circumference of a pulley in feet is equal to one-fourth of the diameter of the pulley in inches. And hence the speed of the belt used to drive a fan is equal to one-fourth the product of the diameter of the pulley on the fan, in inches, multiplied by the number of revolutions it makes per minute.

EXAMPLE:—Determine the width of a single belt running on a pulley 40 inches in diameter and making 200 revolutions per minute necessary to transmit 11 horse power.

Here, since the diameter of the pulley in inches is 40 and the number of revolutions per minute is 200, the speed of the belt in feet per minute is

$$\frac{40 \times 200}{4} = 2000$$

Looking now in Table X, since the belt is to be a "single" belt, we find under the column headed 2000, that a belt four inches wide will transmit 10 horse power and a belt 5 inches wide will transmit 12.5 horse power. We may, therefore, use a 4 inch belt at about 10 per cent. above its capacity or we may use a 5 inch belt at about 80 per cent. of its capacity. In most cases it would probably be best to use the 5 inch belt.

From Table XA we see that a 3 inch double belt will transmit 10.9 horse power when running at a speed of 2000 feet per minute.

CHAPTER VIII

Efficiency. The efficiency of a fan may mean either of two things; it may mean the quotient obtained by dividing the useful work by the work done on the air, or it may mean the quotient obtained by dividing the useful work by the work required to run the fan. The useful work is the product obtained by multiplying the quantity of air delivered per minute by the pressure in the housing at the outlet. The quantity of air must be expressed in cubic feet and the pressure must be in pounds per square foot.

It seems to the writer to be more reasonable and more in accordance with the meaning of the word efficiency when used in connection with other machines, to say that the efficiency of a fan is the quotient obtained by dividing the useful work by the work required to run the fan; and it is with this meaning that the word will be used here.

When the theoretical outlet area is less than the blast area, the pressure in ounces per square inch in the housing near the outlet is P, the pressure due to the velocity of the tips of the floats, and the number of cubic feet of air discharged per minute is C_1. As has been shown before in order to convert pressure in ounces per square inch into pressure in pounds per square foot it is necessary to multiply the pressure in ounces per square inches by 9. Hence in this case the pressure in pounds per square foot is $9P$. And since the number of cubic feet of air discharged per minute is C_1 the useful work is $9 C_1 P$. Divide this by 33000 and we get that the useful horse power is

$$\frac{9 C_1 P}{33000} = \frac{C_1 P}{3670}$$

The horse power required to run the fan is equal to the horse power expended on the air plus the horse power required to overcome the resistance to turning of the wheel. If we call k the horse power required to overcome the resistance to turning of the wheel, and remember that the horse power expended on the air as given by (56) is K_1, we have the horse power required to run the fan is $K_1 + k$

Then the efficiency, which we may call e_1, is the useful horse power divided by the horse power required to run the fan; that is

$$e_1 = \frac{C_1 P}{3670\,(K_1 + k)}$$

Substitute in this expression the value of K_1 as given by (56) and we get

$$(74) \qquad e_1 = \frac{C_1 P}{C_1 P\left(\frac{1 + a^2 c^2 r^2}{A^2}\right) + 3670\,k}$$

$$= \frac{1}{1 + \frac{a^2 c^2 r^2}{A^2} + \frac{3670\,k}{C_1 P}}$$

It must be remembered that k is probably not *exactly* the same for any two fans. It depends upon the construction and lubrication of the bearings of the shaft, the way the wheel is mounted, the weight of the wheel, the construction of the housing and its fit to the wheel, and a great many other details of the construction of the fan. If we assume as we did in (61) that k is equal to 0.1 of H as given by (60) we have

$$k = \frac{C\,P\,(1 + r^2)}{33000}$$

and

$$\frac{3670\,k}{C_1\,P} = \frac{3670\,C\,P\,(1+r^2)}{33000\,C_1\,P}$$

$$= \frac{C\,(1+r^2)}{9\,C_1}$$

But from (30) we have

$$\frac{C}{C_1} = \frac{A}{a\,c}$$

Hence

$$\frac{3670\,k}{C_1\,P} = \frac{A\,(1+r^2)}{9\,a\,c}$$

and this in (74) gives

$$(75) \qquad e_1 = \frac{1}{1 + \dfrac{a^2\,c^2\,r^2}{A^2} + \dfrac{A\,(1+r^2)}{9\,a\,c}}$$

When the theoretical area is equal to the blast area, as it is when the fan is working at its capacity, the efficiency, which we may call e in this case, becomes from (75)

(76) $$e = \frac{1}{1 + r^2 + \frac{1 + r^2}{9}}$$

$$= \frac{0.9}{1 + r^2}$$

It will be noticed that (76) shows that the efficiency of a fan working at its capacity depends only upon r, the ratio of the diameter of the inlet to the diameter of the wheel. To show the effect on the efficiency of a variation in the value of r, Table XI. has been calculated from (76).

TABLE XI.

Efficiencies.

r	e
0.25	0.85
0.35	0.80
0.50	0.72
0.625	0.65
0.707	0.60
0.75	0.58

Table XI. shows that the efficiency of a fan whose inlet is 0.5 the diameter of the wheel is 0.72, while the efficiency of a fan whose inlet is 0.707 the diameter of the wheel is only 0.60. In other words, the fan with the smaller ratio of inlet to diameter of wheel is 20 per cent. more efficient than the fan with the larger ratio of inlet to diameter. It is this difference in efficiency that makes it preferable to use a fan with a small ratio of inlet to diameter of wheel when working against comparatively high pressures.

Suppose it is required to deliver 20000 cubic feet of air per hour against a pressure of one ounce per square inch. Table II. shows that a fan with a 5-foot wheel with an inlet 0.707 the diameter of the wheel will do the work; and Table IIA shows that a fan with an inlet 0.625 the diameter of the wheel will require a 6-foot wheel. Table VIII. shows that the 5-foot wheel will require 9.1 horse power to run it, while Table VIIIA shows that only 8.4 horse power will be required to run the fan with a 6-foot wheel. By using the fan with a 6-foot wheel a saving of 8 per cent. will be made in the power required to run the fan as compared to the power which would be required for the fan with the 5-foot wheel.

The diameter of the inlet of the fan with the 5-foot wheel would be $60 \times 0.707 = 42.5$ inches; while the diameter of the inlet of the fan with the 6-foot wheel would be $72 \times 0.625 = 45$ inches.

When the theoretical outlet area is greater than the blast area the pressure in ounces per square inch in the housing at the outlet is p, and the pressure in pounds per square foot is $9p$. Since the number of cubic feet of air discharged is C_2, the useful work is $9p\,C_2$. But from (52) we have

$$p = \frac{P\,(1+r^2)}{1 + \dfrac{a^2\,c^2\,r^2}{A^2}}$$

and this value of p in the expression for the useful work gives

$$9\,p\,C_2 = \frac{9\,C_2 P\,(1+r^2)}{1 + \dfrac{a^2\,c^2\,r^2}{A^2}}$$

This divided by 33000 gives for the useful horse power,

$$\frac{9\,p C_2}{33000} = \frac{C_2\,P\,(1+r^2)}{3670\left(1+\dfrac{a^2\,c^2\,r^2}{A^2}\right)}$$

From (63) we find that the horse power required to run a fan when its theoretical outlet area is larger than the blast area is

$$H_2 = \frac{C\,P\,(1+r^2)\,B}{3300}$$

Calling the efficiency e_2, we have

$$(77) \qquad e_2 = \frac{9\,p\,C_2}{33000\;H_2} = \frac{0.9\,C_2}{C\,B\left(1+\dfrac{a_2\,c_2\,r_2}{A^2}\right)}$$

From (39) we have $C_2 = C\,B$, and hence (77) becomes

$$(78) \qquad e_2 = \frac{0.9}{1 + \dfrac{a^2\ c^2\ r^2}{A^2}}$$

Equation (78) is based upon the expression for H_2 as given in (63) which is based upon the assumption that the mechanical efficiency of the fan is 0.9. That is, that when a fan is working at its capacity or above its capacity only 0.1 of the total power applied to it is used in overcoming the resistance to the turning of the wheel due to friction, and 0.9 of the total power is expended on the air.

Equations (76) and (78) enable us to compare the efficiencies of two fans, one working at its capacity with an outlet area equal to the blast area, and the other working above its capacity with an outlet area greater than the blast area.

EXAMPLE:—It is necessary to supply 30000 cubic feet of air per minute against a pressure of 0.6 of an ounce per square inch. Determine the efficiency of a fan which will do this when working at its capacity, and, also, the efficiency of a smaller fan working beyond its capacity to do the work. Assume for the smaller fan that the ratio of the theoretical outlet area, $a\,c$, to the blast area, A, is 1.6.

From Table II. we find that a fan with a 7-foot wheel will deliver 30400 cubic feet of air against a pressure of 0.6 ounces when working at its capacity.

We know that when the theoretical outlet area is greater than the blast area the pressure p at the outlet is less than the pressure P corresponding to the velocity of the tips of the floats, and from (54) we have

$$p = F P$$

For $\frac{a c}{A}$ equal 1.6 we have from Table VII. that F is 0.66 when r is 0.707. We also know from the condition of the problem that p is 0.6. Therefore

$$P = \frac{p}{F} = \frac{0.6}{0.66} = 0.91$$

From Table IV. we find that when $\frac{a c}{A}$ is 1.6 and r is 0.707, B is 1.30. From (39) we get that the quantity of air delivered by a fan when its theoretical outlet area is greater than the blast area, is

$$C_2 = C B$$

From this we get, remembering that in our problem C_2 is 30000

$$C = \frac{C_2}{B} = \frac{30000}{1.3} = 23100$$

Now we must look in Table II. and find the diameter of the wheel whose capacity is 23100 when working against a pressure of 0.91. The table shows that a fan with a $5\frac{1}{2}$-foot wheel has a capacity of 23000 cubic feet per minute against a pressure of 0.9 ounces per square inch.

From Table VIII. the horse power required to run the fan with the 7-foot wheel is 8.28; and the horse power required to run the $5\frac{1}{2}$-foot wheel when working at its capacity against 0.9 of an ounce is 9.4. But the $5\frac{1}{2}$-foot wheel must be worked above its capacity, and by (63) we get that the horse power required to run it is then

$$H_2 = H B$$

H we have just found to be 9.4, and B we have found to be 1.3. Therefore the horse power required to run the $5\frac{1}{2}$ foot fan is

$$H_2 = 9.4 \times 1.3$$

$$= 12.2$$

The efficiency of the 7-foot fan, for which r is 0.707, is, from (76),

$$e = \frac{0.9}{1+r^2}$$

$$= \frac{0.9}{1+0.707^2} = 0.6$$

From (78) we get that the efficiency of the fan with the $5\frac{1}{2}$-foot wheel, for which r is also 0.707, is

$$e_2 = \frac{0.9}{1 + \frac{a^2\ c^2\ r^2}{A^2}}$$

$$= \frac{0.9}{1+1.6^2 \times 0.707^2} = 0.38$$

Air Per Horse Power. It is sometimes interesting to know the number of cubic feet of air, which we may call l, that a fan will deliver per minute per horse power applied to the wheel.

When a fan is working at its capacity we have, from (60), that the air delivered per minute per horse power is

$$(79) \qquad l = \frac{C}{H} = \frac{3300}{P\,(1 + r^2)}$$

and when the fan is working beyond its capacity so that the theoretical outlet area is greater than the blast area, the air delivered per minute per horse power is, from (62),

$$l_2 = \frac{C_2}{H_2} = \frac{0.9\,C_2}{K_2}$$

From (57) we have

$$K_2 = \frac{C_2\,P\,(1 + r^2)}{3670}$$

and hence we get for l_2

$$(80) \qquad l_2 = \frac{C_2}{H_2} = \frac{3300}{P\,(1 + r^2)}$$

Equations (79) and (80) show that the air delivered per minute per horse by a fan depends upon the pressure, P, corresponding to the tips of the floats and the ratio, r, of the diameter of the inlet to the diameter of the wheel. The equations further show that for given values of P and r, the air delivered per minute per horse power is the same whether the fan is working at or beyond its capacity.

In considering (79) and (80) care must be taken not to confuse the pressure P, corresponding to the tips of the floats, with the pressure in the housing at the outlet. In the example worked under the article on efficiency it was found that a 7-foot fan would supply the required quantity of air with an expenditure of 8.3 horse power, while the $5\frac{1}{2}$-foot fan required an expenditure of 12.2 horse power for the same work. This is not inconsistent with equations (79) and (80), because P for the 7-foot fan was 0.6 of an ounce, while for the $5\frac{1}{2}$-foot fan it was 0.91 of an ounce. Both fans were delivering 30000 cubic feet of air per minute. Therefore the 7-foot fan was delivering per horse power

$$\frac{30000}{8.3} = 3610$$

cubic feet of air per minute; while the $5\frac{1}{2}$-foot fan was delivering

$$\frac{30000}{12.2} = 2460$$

cubic feet of air per minute per horse power.

From (79) we get that the air delivered per horse power per minute by the 7-foot fan should be

$$l = \frac{3300}{P\ (1+r^2)}$$

$$= \frac{3300}{0.6(1+0.707^2)} = 3670$$

and for the 5½-foot fan we get

$$l = \frac{3300}{P\ (1+r^2)}$$

$$= \frac{3300}{0.91\ \ (1+0.707^2)} = 2420$$

There is a slight difference in the results obtained by the two methods, but it is due to the fact that tables were used, and it is not always possible to find the exact value desired in a table and we are obliged to take the nearest value given.

Equation (79) has been used to calculate the quantity of air delivered per minute per horse power by fans working at their capacity against different pressures, for *r* equal 0.5, 0.625 and 0.707; and the results are given in Table XII.

TABLE XII.

Air per minute per horse power.

r	Pressure in ounces per square inch.									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
0.500	26400	13200	8800	6600	5280	4400	3770	3300	2930	2640
0.625	23800	11900	7920	5950	4750	3980	3400	2970	2640	2380
0.707	22000	11000	7330	5500	4400	3670	3140	2750	2440	2200

CHAPTER IX.

Exhausters. Heretofore we have discussed fans as if they were all to be used as blowers, we have always assumed that the pressure in front of the inlet was that of the atmosphere, and that the pressure in the housing near the outlet was greater than that of the atmosphere. When a fan is used as an exhauster, its action so far as handling air is concerned is in no way different than when used as a blower. When a fan used as an exhauster is attached to a room or chamber and the inlet opening from the atmosphere to the chamber is closed, there is a *vacuum* in the chamber and a pressure equal to that of the atmosphere in the housing. The *vacuum* in the chamber is the same number of ounces below the pressure of the atmosphere that the pressure in the housing would be above that of the atmosphere in the case of a blower with its outlet closed. That is to say, the vacuum is expressed by the same number of ounces as the pressure corresponding to the velocity of the tips of the floats of the fan wheel. When the theoretical area of the opening from the atmosphere into the chamber is equal to or less than the blast area, the pressure in the chamber is less than that of the atmosphere by an amount equal to the pressure corresponding to the velocity of the tips of the floats; and when the theoretical area of the opening is greater than the blast area the pressure in the chamber is less than that of the atmosphere by an amount that is less than the pressure corresponding to the velocity of the tips of the floats.

In fan work, vacuums are expressed in ounces per square inch just as pressures are, although the word *vacuum* is seldom used. It is quite common to speak of an exhauster "working at a pressure of 0.5 of an ounce," when as a matter of fact it is not working against any pressure. What is meant is that the fan is being run at such a speed that if it were being used as a blower there would be a pressure of 0.5 of an ounce per square inch in the housing if the theoretical outlet area were equal to the blast area.

An exhauster is said to be working at its capacity when the theoretical inlet area from the atmosphere to the chamber from which the exhauster is taking air is equal to the blast area, and when the outlet of the fan is such that there is a perfectly free discharge of the air from the fan housing. When an exhauster is working at its capacity it will handle exactly the same quantity of air that it would handle if working at its capacity as a blower. That is to say, a fan us d as an exhauster and working at its capacity against a *vacuum*, of say, 0.6 of an ounce will handle exactly the same quantity of air that it would handle when working at its capacity as a blower against a *pressure* of 0.6 of an ounce. This means, of course, that Tables II. and IIA may be used for exhausters as well as for blowers. In fact, all of the tables given here apply to fans used as exhausters as well as to fans used as blowers.

When a fan is used at its capacity as a combination exhauster and blower the sum of the vacuum, expressed in ounces per square inch, in the chamber from which the air is being drawn, and the pressure, expressed in ounces per square inch, in the housing at the outlet is equal to the number expressing the pressure in ounces per square inch corresponding to the velocity of the tips of the floats. Thus if a fan is run at such a speed as to produce a pressure of 0.7 of an ounce in the housing when working at its capacity, it can be made to work at the same speed against a vacuum of 0.3 of an ounce per square inch and a pressure of 0.4 of an ounce per square inch. Or to put it another way, we may say that if a fan is run at such a speed that the pressure corresponding to the velocity of the tips of the floats is 0.7 of an ounce per square inch, the fan will work at its capacity for the same speed against any combination of vacuum and pressure, both expressed in ounces per square inch, whose sum is 0.7. Thus there may be a vacuum of 0.1 of an ounce and a pressure of 0.6 of an ounce; or a vacuum of 0.5 of an ounce and a pressure of 0.2 of an ounce. And if the inlet from the atmosphere to the chamber in which the vacuum is made, and, also, the outlet from the housing, be properly adjusted, the fan will deliver for each combination of vacuum and pressure an amount of air equal to its capacity.

CHAPTER X

Housing. In order to get good results from a fan used as a blower it is important to have a proper housing. The housing should fit close to the sides of the wheel, and be so made as to allow equal freedom for the discharge of air from all parts of the periphery. The air in the housing tends to revolve with the wheel and hence the best results will be obtained when the outlet opening is placed so that the air glides easily and smoothly into it without any abrupt change in the direction of its flow. A fan wheel placed in a rectangular or circular casing or housing, without a scroll, will not give as efficient results as if it were placed in a housing having a proper scroll by which the air is gradually led to the outlet. In the case of a circular or rectangular housing with the outlet opening projecting radially from the housing, the air must make a change of almost a right angle in its direction of flow in order to enter the outlet, and that it will not do, especially when the wheel is working at comparatively high pressures. At low pressures a rectangular or circular housing may be made to give fairly good results, because then the velocity of revolution of the air in the housing and at right angles to the outlet is not high. The higher the velocity of revolution of the air in the housing the greater is the difficulty of making the air change its direction abruptly, and therefore, the greater is the necessity of having a housing with a properly constructed scroll by which the air is gradually and smoothly led into the outlet opening without any abrupt change in either velocity or direction. The wheel of a fan should always revolve in the direction of the increase of the scroll, from the point where it is nearest the periphery of the wheel to the point where it is farthest from the periphery. There is probably no less efficient housing or one which reduces the air delivery capacity of a wheel more than a housing with a proper scroll but the wheel revolving backwards, that is in the direction opposite to the direction of the increase of the scroll.

The dimensions of the scroll of the housing of a fan wheel depend upon the diameter of the wheel and upon the diameter of the inlet opening. The space through which the air flows towards the outlet between the periphery of the wheel and the housing, should be proportioned between the place where the scroll is nearest the wheel and the place where the scroll is farthest from the wheel so that the velocity of the air flowing towards the outlet shall be the same at every point; and since the discharge from the periphery of the wheel into the housing is to be uniform at all points, this means that the scroll of the housing should be as nearly as possible a true Archimedean spiral.

Let Fig. 27 represent the housing of a fan wheel whose diameter is D, whose width is W, and whose blast area is A. Let the diameter of the inlet be $r\,D$ as indicated in the figure, and let the fan revolve from left to right as shown by the arrow. At the point a where the outlet begins, the distance of the scroll from the periphery of the wheel is x; at b, one-quarter round from a, the distance is $\frac{x}{4}$; at c, it is $\frac{x}{2}$, and at d, it is $\frac{3\,x}{4}$. In order that the fan may work at its capacity the outlet area multiplied by its coefficient of discharge must not be less than the blast area A. If we assume that the coefficient of discharge of the outlet is 1, we have, since the area of the outlet is $W\,x$,

$$W\,x = A$$

From which we get,

$$x = \frac{A}{W}$$

From (23) we have

$$A = 0.44\, r^3 D^2$$

And this value of A in the expression for x gives

$$x = \frac{0.44\, r^3 D^2}{W} \tag{81}$$

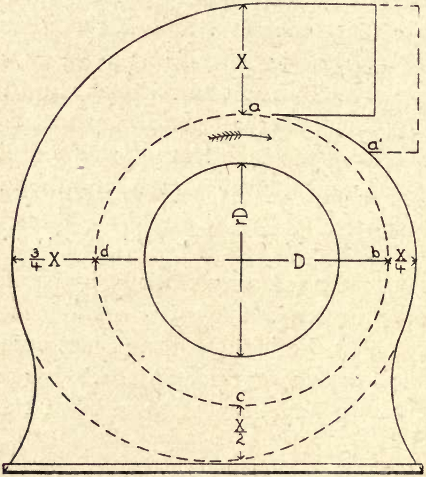


FIG. 27.

Since the distance from the periphery of the wheel to the housing at the bottom is $\frac{x}{2}$, as shown in Fig. 27, it is evident that the height of the housing cannot be less than $D+\frac{3x}{2}$. And it is evident that for a wheel of a given diameter, the height of the housing will be less the smaller we make x. But from (81) it is evident that the greater we make W, the width of the wheel, the smaller will be the value of x. Hence the wider we make the wheel the lower may the housing be.

It is usual to make the outlet opening square, and to make the actual area of the outlet greater than the blast area in order to allow for friction, and, also, because the coefficient of discharge is not always 1 as was assumed in deducing (81). Therefore, if we make x equal to W in (81) and make the actual outlet area equal to, say, 1.5 times the blast area, we get

$$W^2 = 1.5 \times 0.44\, r^3 D^2$$

$$= 0.66\, r^3 D^2$$

From which we get

$$W = \sqrt{0.66\, r^3 D^2} \qquad (82)$$

If now we make r equal 0.707 in (82) we get

$$W = \sqrt{0.66 \times 0.707^3 D^2}$$

$$= 0.48\, D$$

This shows the reason for the custom, which prevails with many manufacturers, of making the width of a fan equal to one-half of the diameter of the wheel when the diameter of the inlet opening is equal to 0.707 of the diameter of the wheel.

If r were 0.625 we should have

$$W = \sqrt{0.66 \times 0.625^3 D^2}$$

$$= 0.40\,D$$

Some manufacturers make what is called a "narrow" fan, whose width is about three-eighths of the diameter of the wheel; although most fans used for heating and ventilating work are made with a width equal to one-half the diameter of the wheel, irrespective of the ratio of the diameter of the inlet opening to the diameter of the wheel. When the outlet is made square and the width of the fan is one-half the diameter of the wheel the actual area of the outlet opening is about 1.6 the blast area, when r is 0.707.

Instead of making the bottom of the outlet at a in Fig. 27, it is usually made at a point a' about one-eighth of the circumference round from a. The outlet is then as indicated by the dotted lines, and the air which escapes from the periphery of the wheel between a and a' passes directly into the outlet without passing into the scroll at all. Hence instead of all of the air discharged by the fan being obliged to pass through the space between the point a and the scroll only $\frac{7}{8}$ of it must pass through this space. And in order that the velocity of the air which passes through this space shall be the same as the velocity of the air passing through the outlet, the area of the space should be $\frac{7}{8}$ of the area of the outlet. Since the area of this space is $x\,W$ and the area of the outlet is W^2, we have

$$x\,W = \frac{7\,W^2}{8}$$

and

$$(83) \qquad x = \frac{7\,W}{8}$$

If now we make W equal to one-half the diameter of the wheel we get

$$(84) \qquad x = \frac{7\,D}{16} = 0.44\,D\text{, about.}$$

From Fig. 27 we see that at b the distance of the scroll from the wheel is $\frac{x}{4}$ equal $0.11\,D$; at c it is $\frac{x}{2}$, equal $0.22\,D$; and at d it is $\frac{3\,x}{4}$, equal to $0.33\,D$.

The height of the housing is equal to $x + D + \frac{x}{2}$

equal $1.66\,D$; and the length of the housing is equal to $\frac{3}{4}x + D + \frac{x}{4}$, equal $1.44\,D$.

The value of x given above is for a width of fan equal to one-half the diameter of the wheel, and we have shown that this is the width proper for a fan whose inlet opening is 0.707 the diameter of the wheel. We have shown that for a fan with a square outlet, and an inlet equal to 0.625 the diameter of the wheel, the width of the fan should be about 0.4 the diameter of the wheel, although it is often made $\frac{3}{8}$ the diameter of the wheel. When the width of the fan is $\frac{3}{8}$ the diameter of the wheel x becomes

$$x = \frac{7\,W}{8} = \frac{7 \times 3\,D}{8 \times 8}$$

$$= \frac{21\,D}{64} = 0.328\,D$$

Fans are sometimes made with a width equal to half the diameter of the wheel, an inlet equal to 0.625 the diameter of the wheel, and a square outlet. These fans are usually made with housings so proportioned that x is equal to about $0.30\,D$. The height of housings of such fans is equal to $1.45\,D$. In general, we may say that for fans whose width is equal to one-half the diameter of the wheel, x is 0.44 D when the inlet is 0.707 the diameter of the wheel; and x is 0.30 D when the inlet is 0.625 the diameter of the wheel.

Having decided upon the value of x the next thing is to draw the scroll. The scroll is usually

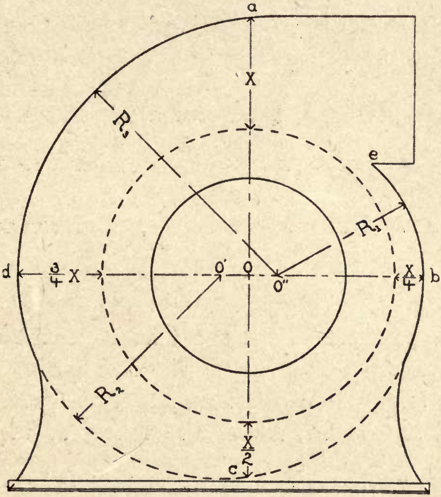


FIG. 28.

a three arc scroll, that is to say it is made up of arcs of three different radii. Referring now to Fig. 28 it is seen that the distance of the point b from the center c of the wheel is $\frac{D}{2} + \frac{x}{4}$; the distance of c from o is $\frac{D}{2} + \frac{x}{2}$, and the distance of d from o is $\frac{D}{2} + \frac{3x}{4}$. The distance from b to d is $\frac{3x}{4} + D + \frac{x}{4}$, which is just twice the distance of c below the point o. Hence we may make the part $b\,c\,d$ of the scroll a semicircle whose radius is $\frac{D}{2} + \frac{x}{2}$. The center of this circle will be at o', to the left of o a distance $o'\,o$, which is equal to $\frac{D}{2} + \frac{x}{2}$ minus the distance $o\,b$. But $o\,b$ is $\frac{D}{2} + \frac{x}{4}$, and hence $o'o$ is $\frac{x}{4}$.

The part $d\,a$ of the scroll must be the arc of a circle whose center is on the line $o\,b$ and whose radius is equal to the distance of the point a, the top of the outlet, above the center of the wheel, which is $\frac{D}{2} + x$. The center of this arc will be at o'', to the right of o a distance $o\,o''$ equal to $\frac{D}{2} + x$ minus the distance $o\,d$. But $o\,d$ is $\frac{D}{2} + \frac{3x}{4}$, and hence $o\,o''$ is $\frac{x}{4}$.

The part $e\,b$ of the scroll may be drawn about

o'' as a center with a radius equal to the distance $o\,b$ minus $o\,o''$, which is $\frac{D}{2}+\frac{x}{4}-\frac{x}{4}$, equal $\frac{D}{2}$.

Hence if we call R_1 the radius of the arc $e\,b$; R_2 that of the arc $b\,c\,d$; and R_3 that of the arc from d to the top of the fan, we have

$$R_1=\frac{D}{2}$$

$$R_2=\frac{D}{2}+\frac{x}{2}$$

$$R_3=\frac{D}{2}+x$$

The height, T, of the scroll is equal to R_2+R_3; and the height of the fan from the foundation to the top of the scroll is usually one or two inches greater than the height of the scroll.

For a fan whose width is one-half the diameter of the wheel and whose inlet is 0.707 the diameter of the wheel x is equal to $0.44\,D$, and hence we have the distance $o'o$, equal distance $o\,o''$, is equal $\frac{x}{4}$ or $0.11\,D$, and

$$R_1 = 0.5\,D$$

$$R_2 = 0.72\,D$$

$$R_3 = 0.94\,D$$

$$T = 1.66\,D$$

For a fan whose width is about one-half the diameter of the wheel and whose inlet is 0.625 the diameter of the wheel we have said that x is about 0.30 D, and hence we have the distance $o'\,o$, equal distance $o\,o''$, is equal $\frac{x}{4}$ or 0.075 D, and

$$R_1 = 0.5\,D$$

$$R_2 = 0.65\,D$$

$$R_3 = 0.80\,D$$

$$T = 1.45\,D$$

The side pieces and the scroll piece of the housing of a fan when made of sheet steel must be braced with angle irons and made sufficiently thick to resist the pressure of the air and the straining action due to the movement of the wheel. The heavier the braces and the more there are, the thinner may be the plates of the side pieces and the scroll piece. There seems to be no particular rule or formula followed in determining the thickness of the plates used or the size of the angle irons used for bracing them, and yet there seems to be a rather remarkable agreement in the practice of the different manufacturers.

Dimensions of Housings. In laying out fan work it is often necessary to know approximately at least the space that will be occupied by a fan having a wheel of a certain diameter with a given ratio of inlet to diameter of wheel. And in order to facilitate the work of the designer in such cases the following tables, showing the various dimensions of the housings of fans are given. The dimensions given in the tables have been calculated not taken from the catalogue of any manufacturer, and hence it is not likely that they will be found to agree exactly with the dimensions of any manufacturer. They are only approximate, and are intended only to enable a designer to obtain a fairly accurate estimate of the space that a fan will occupy. The exact dimensions of the housing of a fan of a certain make will depend upon the details of construction adopted by the maker and must be obtained from him if they are desired. The dimensions are given in inches in all cases.

Tables XIII., XIV. and XV. are for fans for which the ratio, r, of diameter of inlet to diameter of wheel is 0.707; and Tables XVI., XVII. and XVIII. are for fans for which r is 0.625.

The dimensions in Table XIII. have been calculated by the following equations, in which D is the diameter of the wheel in feet.

$$I = 8.48\,D \qquad O = 6\,D$$

$$G = 17.3\,D \qquad T = 19.9\,D+1$$

$$J = 9.96\,D \qquad L = 7.32\,D$$

$$Q = 8.64\,D+1$$

In Table XIV. all dimensions except T and Q are the same as in Table XIII., and

$$T = 15.5\,D+z$$

$$Q = 4.2\,D+z$$

z is two inches for wheels 4 feet or less in diameter; 2½ inches for wheels larger than 4 feet up to 6 feet; 3 inches for wheels larger than 6 feet up to 9 feet; and 4 inches for wheels larger than 9 feet.

In Table XV. all the dimensions except T and Q are the same as in Table XIV., and here

$$T = 12.84\,D + Z$$

$$Q = 4.2\,D + Z$$

z has the same value as in Table XIV.

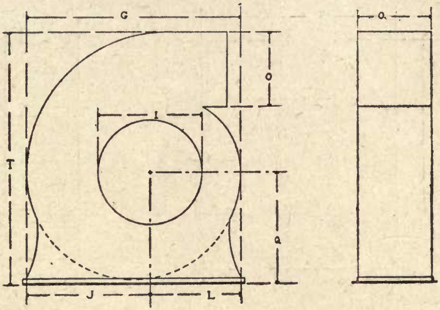


TABLE XIII.

Full housed, top, horizontal discharge fan, $r = 0.707$.

Diameter of wheel in feet.	Dimensions of housings in inches.						
	I	*O*	*G*	*T*	*J*	*L*	*Q*
3	25	18	52	61	30	22	27
3½	30	21	61	71	36	25	31
4	34	24	69	81	40	29	36
4½	38	27	78	91	45	33	40
5	42	30	87	101	50	37	44
5½	47	33	95	110	55	40	49
6	51	36	104	120	60	44	53
6½	55	39	113	130	65	48	57
7	59	42	121	140	70	51	61
8	68	48	139	160	80	59	70
9	76	54	156	180	90	66	79
10	85	60	173	200	100	73	87
11	93	66	191	220	110	81	96
12	102	72	208	240	120	88	105

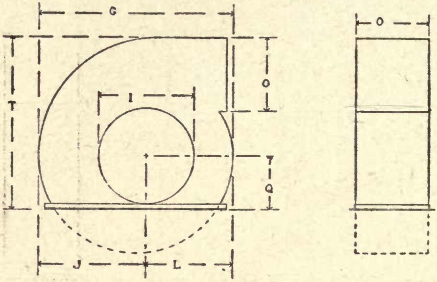


TABLE XIV.

Three quarter housed, top, horizontal discharge fan, $r=0.707$.

Diameter of wheel in feet.	Dimensions of housings in inches.						
	I	*O*	*G*	*T*	*J*	*L*	*Q*
3	25	18	52	48	30	22	15
3½	30	21	61	56	36	25	17
4	34	24	69	64	40	29	19
4½	38	27	78	71	45	33	21
5	42	30	87	80	50	37	24
5½	47	33	95	88	55	40	26
6	51	36	104	96	60	44	28
6½	55	39	113	104	65	48	30
7	59	42	121	112	70	51	32
8	68	48	138	127	80	58	37
9	76	54	156	143	90	66	41
10	85	60	173	159	100	73	46
11	93	66	190	175	110	80	50
12	102	72	208	190	120	88	54

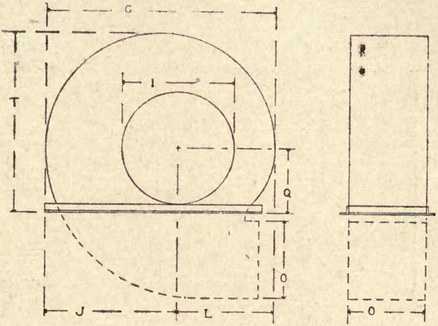


TABLE XV.

Three quarter housed, bottom, horizontal discharge fan, $r=0.707$.

Diameter of wheel in feet.	Dimensions of housings in inches.						
	I	*O*	*G*	*T*	*J*	*L*	*Q*
3	25	18	52	41	30	22	15
3½	30	21	61	47	36	25	17
4	34	24	69	53	40	29	19
4½	38	27	78	60	45	33	21
5	42	30	87	67	50	37	24
5½	47	33	95	73	55	40	26
6	51	36	104	80	60	44	28
6½	55	39	113	87	65	48	30
7	59	42	121	93	70	51	33
8	68	48	138	106	80	58	37
9	76	54	156	119	90	66	41
10	85	60	173	132	100	73	46
11	93	66	190	146	110	80	50
12	102	72	208	158	120	88	55

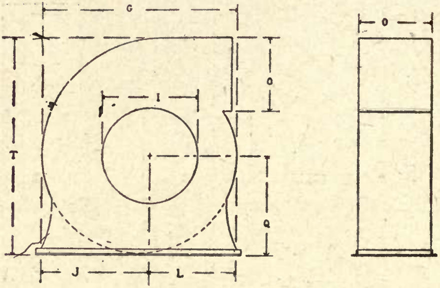


TABLE XVI.

Full housed top, horizontal discharge fan, $r=0.625$.

Diameter of wheel in feet.	Dimensions of housings in inches.						
	I	*O*	*G*	*T*	*J*	*L*	*Q*
3	23	18	47	53	26	21	24
3½	26	21	55	62	31	24	28
4	30	24	63	71	35	28	32
4½	34	27	70	79	39	31	36
5	38	30	78	88	43	35	40
5½	41	33	86	97	48	38	44
6	45	36	94	105	52	42	48
6½	49	39	101	114	56	45	52
7	53	42	109	123	61	48	56
8	60	48	125	140	70	55	63
9	68	54	140	158	78	62	71
10	75	60	156	175	87	69	79
11	83	66	172	193	96	76	87
12	90	72	187	210	104	83	95

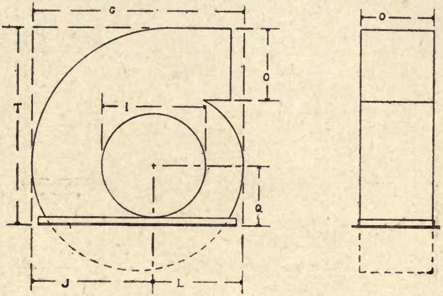


TABLE XVII.

Three quarter housed, top, horizontal discharge fan, $r=0.625$.

Diameter of wheel in feet.	Dimensions of housings in inches.						
	I	*O*	*G*	*T*	*J*	*L*	*Q*
3	23	18	47	42	26	21	13
3½	26	21	55	49	31	24	15
4	30	24	63	55	35	28	17
4½	34	27	70	63	39	31	19
5	38	30	78	69	43	35	21
5½	41	33	86	76	48	38	23
6	45	36	93	83	52	41	25
6½	49	39	101	90	56	45	27
7	53	42	109	96	61	48	29
8	60	48	125	110	70	55	33
9	68	54	140	123	78	62	38
10	75	60	156	138	87	69	41
11	83	66	172	151	96	76	45
12	90	72	187	164	104	83	49

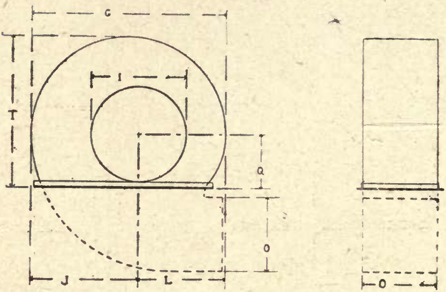


TABLE XVIII.

Three quarter housed, bottom, horizontal discharge fan, $r = 0.625$.

Diameter of wheel in feet.	Dimensions of housings in inches.						
	I	*O*	*G*	*T*	*J*	*L*	*Q*
3	23	18	47	38	26	21	13
3½	26	21	55	43	31	24	15
4	30	24	63	48	35	28	17
4½	34	27	70	54	39	31	19
5	38	30	78	60	43	35	21
5½	41	33	86	66	48	38	23
6	45	36	93	72	52	41	25
6½	49	39	101	78	56	45	27
7	53	42	109	84	61	48	29
8	60	48	125	97	70	55	33
9	68	54	140	107	78	62	37
10	75	60	156	120	87	69	41
11	83	66	172	131	96	76	45
12	90	72	187	143	104	83	49

The dimensions in Table XVI. have been calculated by the following equations:

$I = 7.5\,D \qquad O = 6\,D$

$G = 15.6\,D \qquad T = 17.4\,D + 1$

$J = 8.7\,D \qquad L = 6.9\,D$

$Q = 7.8\,D + 1$

In Tables XVII. and XVIII. all the dimensions except T and Q are the same as in Table XVI. The equations for T and Q in Table XVII. are

$$T = 13.35\,D + z$$

$$Q = 3.75\,D + z$$

The equations for T and Q in Table XVIII. are

$$T = 11.55\,D + z$$

$$Q = 3.75\,D + z$$

The values of z used in Tables XVII. andXVIII. are the same as those given in the explanation of Table XIV.

Shaft. When the power necessary to drive a fan is known it is an easy matter to calculate the size of shaft required for it, but a manufacturer designing a line of fans is never sure exactly what will be the worst conditions under which a fan will run and as it will not pay to design a fan for every condition he designs all of the ordinary commercial fans for what is probably the worst conditions of ordinary use. And as the ideas of different manufacturers are not the same as to what are probably the worst conditions under which an ordinary fan will be used, it results that the fans of different manufacturers differ more in such details as the size of shaft, the kind of bearings, diameters of bolts, etc., than in other things.

The shafts of small fans are usually made larger in proportion to the diameter of the wheels than are shafts of large fans. This is so because it is not advisable to have a shaft less than an inch and a half on even a small fan having a wheel 3 feet in diameter, even although such a shaft may be larger than is actually necessary for the work the fan will be called upon to do; and, also, because large fans are seldom worked at as high a pressure as the smaller ones.

A safe practice for fans having wheels not smaller than 3 feet in diameter and not larger than 7 feet, is to make the diameter of the shaft in inches equal to one half the diameter of the wheel in feet.

This rule if applied to large fans will give a shaft which is unnecessarily large, so for fans with wheels not less than 7 feet nor more than 12 feet in diameter we may make the diameter of the shafts equal to $1\frac{3}{4}$ inches plus one quarter of the diameter of the wheel in feet.

Thus, we should use for ordinary heating work a 2-inch shaft with a 4-foot wheel, and a $4\frac{1}{4}$-inch, equal $\frac{D}{4} + 1\frac{3}{4}$, shaft for a 10-foot wheel.

CHAPTER XI.

Cone Wheels. The name "cone wheel" is applied to a form of single inlet centrifugal fan wheel that is used either without a housing or with a housing that does not fit close to the wheel, has no scroll and is large as compared to the wheel. The proportions of the cone wheels are entirely different from those of other centrifugal fans, but the general formulas which have been deduced for centrifugal fans apply to the cone wheels as well as to the other forms of centrifugal fans. These fans, however, do not give good results when the pressure against which they have to work is at all high. They stand intermediate between the ordinary disc fan which will work against very low pressures only, probably not over $\frac{1}{4}$ or $\frac{3}{8}$ of an ounce, and the regular centrifugal blower with close fitting housing having a proper scroll, which will work against a pressure of several ounces per square inch. It is, of course, impossible to say what is the limit of pressure against which a cone wheel will work. Tests have been made which show that they will work against a pressure of an ounce or an ounce and a half. Even for low pressures they are not as economical to operate as the ordinary centrifugal fan, and, in the opinion of the writer, they are not to be used for heating or ventilating work when it is possible to use an ordinary centrifugal with a close fitting housing and a proper scroll. Where they are used, care should be taken to see that they are properly erected with a free inlet and a housing so arranged and porportioned that there can be a free discharge of the air from all points of the periphery. A free entrance for the air at the inlet and a free discharge at the periphery are as absolutely necessary for fans of this type as for those of the ordinary centrifugal type; and a lack of these essential features has been the cause of failure to give satisfactory results.

Fig. 29 shows a section parallel to the axis of a cone wheel in place ready for operation; and Fig. 30 shows a view looking at the inlet. The housing in this case consists of a rectangular room *A* in which are the wheel, one bearing of the shaft *B*, and the driving pulley *C*. The air enters through the inlet *E* and passes along the cone *F* to the floats *G* which are fastened to the side plates *H*. The air passes between the side plates *H* into the housing, and from there passes out

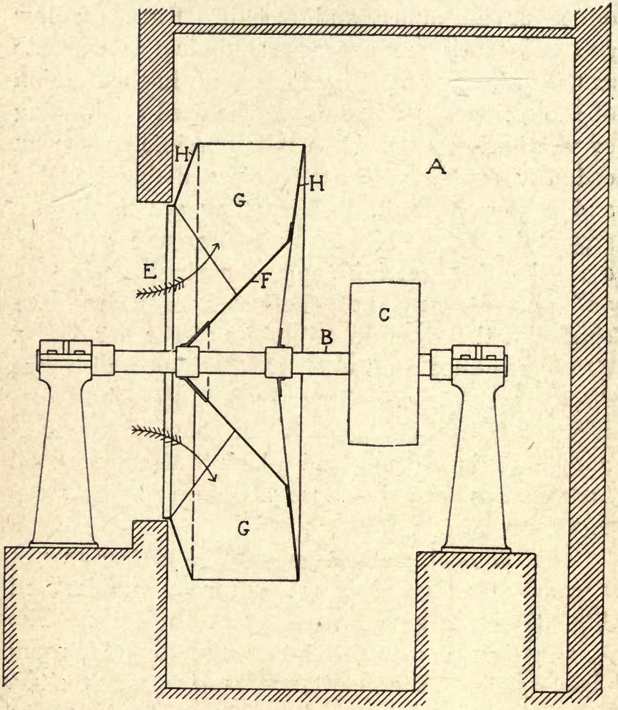


FIG. 29.

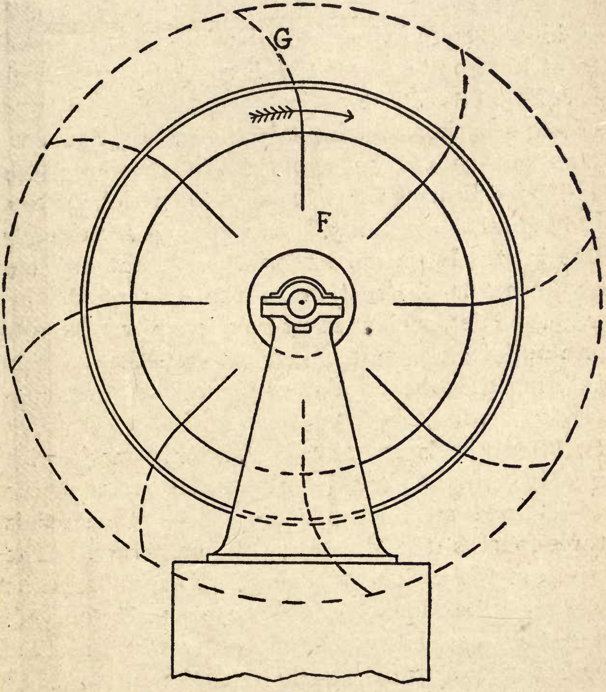


FIG. 30.

through proper openings. The distance between the periphery of the fan wheel and the ceiling, side walls, or floor of the housing should be ample for the free discharge of the air at every point, and it will be found well to make this distance equal to at least one-quarter the diameter of the wheel. In Fig. 29 it will be noticed that one of the shaft bearings is shown in front of the inlet. This construction should be avoided whenever possible and both bearings put back of the wheel, making the wheel "overhung," and giving a perfectly free inlet. It must be remembered that any impediment or obstruction to the flow of the air into the inlet, such as must be offered by a bearing or a pulley close to the inlet, prevents the air from entering the wheel and thereby reduces the capacity of the fan.

Cone wheels are almost uniformly made with a width equal to one-quarter the diameter of the wheel, and an inlet opening whose diameter is three-fourths the diameter of the wheel. The width of the periphery is usually three-fourths the width of the fan. The floats are usually curved, and the wheel is supposed to be revolved so that the convex side of the floats move forward, as shown by the arrow in Fig. 30. It is probable that this form of float is not so good as a properly designed float with its concave side forward, but it is the form generally used.

Since the diameter of the inlet is three fourths the diameter of the wheel r in (18) is 0.75. Hence the expression for the capacity of a cone wheel is

$$(85) \qquad C = 2280\, r^3\, D^2 \sqrt{P}$$

$$= 962\, D^2 \sqrt{P}$$

For all practical purposes it is near enough to the truth to say that

$$(86) \qquad C = 950\, D^2 \sqrt{P}$$

Table XIX. giving the capacities of cone wheels of various diameters working against different pressures per square inch, has been calculated by (80).

When a cone wheel is working at or less than its capacity the number of revolutions which it must make in order to give a certain pressure in ounces per square inch may be determined by (47) or by means of Table VI.

The horse power required to run a cone wheel when working at its capacity is obtained by making r in (60) equal to 0.75. Doing this we get

$$(87) \qquad H = \frac{C\,P}{2100}$$

Table XX. gives the horse power as calcu-

TABLE XIX.

Capacities of cone wheel fans.

Diameter of wheel in feet.	Pressure in ounces per square inch.							
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
3	2700	3800	4700	5400	6100	6600	7200	7600
3½	3700	5200	6400	7400	8200	9000	9700	10400
4	4800	6800	8300	9600	10700	11800	12700	13600
4½	6100	8600	10500	12200	13600	14900	16100	17200
5	7500	10600	13000	15000	16800	18400	19800	21200
5½	9100	12800	15700	18200	20300	22300	24100	25700
6	10800	15300	18700	21600	24200	26500	28600	30600
6½	12700	17900	22000	25400	28400	31100	33600	35900
7	14700	20800	25500	29500	33000	36100	39000	41600
8	19200	27200	33300	38500	43000	47100	50800	54400
9	24300	34400	42100	48600	54400	59600	64400	68800
10	30000	42500	52000	60000	67200	73500	79500	84800
11	36300	51400	63000	72600	81300	89000	96000	103000
12	43300	61200	75000	86500	96800	106000	114000	122000

TABLE XX.

Horse power required for cone wheels.

Diameter of wheel in feet.	Pressure in ounces per square inch.							
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
3	0.13	0.36	0.67	1.03	1.45	1.89	2.40	2.90
3½	0.18	0.50	0.91	1.41	1.95	2.57	3.24	3.96
4	0.23	0.65	1.19	1.83	2.55	3.37	4.23	5.18
4½	0.29	0.82	1.50	2.32	3.24	4.26	5.36	6.55
5	0.36	1.01	1.86	2.86	4.00	5.26	6.60	8.08
5½	0.43	1.22	2.24	3.47	4.83	6.37	8.03	9.80
6	0.51	1.46	2.67	4.12	5.76	7.57	9.53	11.7
6½	0.63	1.70	3.14	4.84	6.75	8.90	11.2	13.7
7	0.70	1.98	3.64	5.62	7.86	10.3	13.0	15.9
8	0.91	2.59	4.75	7.33	10.2	13.5	17.0	20.7
9	1.16	3.28	6.02	9.26	13.0	17.0	21.5	26.2
10	1.43	4.05	7.43	11.4	16.0	21.0	26.5	32.3
11	1.73	4.90	9.00	13.9	19.4	25.4	32.0	39.2
12	2.07	5.83	10.7	16.5	23.0	30.3	38.0	46.5

lated by (87) required to run cone wheels of different diameters when working at their capacities against various pressures.

The size of engine of motor required for a cone wheel is determined by dividing the horse power required to run the wheel, as found from Table XX, by the efficiency of the engine or motor. The efficiency of the ordinary small engine may be taken as $\frac{2}{3}$, and that of the ordinary motor either direct or alternating current may be taken as $\frac{3}{4}$.

Tables VI, XIX and XX enable us to determine all that it is necessary to know in order to select a cone wheel for a given work. Table XIX unables us to determine the diameter of the wheel required; Table VI enables us to determine the number of revolutions at which it must be run; and Table XX enables us to determine the horse power required to run it.

EXAMPLE:—Determine the size of a cone wheel required to deliver 30,000 cubic feet of air per minute against a pressure of 0.3 of an ounce per square inch. Also determine the number of revolutions at which the wheel must be run, and the horse power required to run it.

Looking in Table XIX under the column headed 0.3, we find that the nearest number to 30000 is 33300, which is opposite an 8-foot wheel. Hence we will use an 8-foot wheel.

Now looking in Table VI under the column headed 0.3 and opposite the 8-foot wheel we find

113. Hence the 8-foot wheel must be run at 113 revolutions per minute.

From Table XX we find that the horse power required to run an 8-foot cone wheel when working at its capacity against a pressure of 0.3 of an ounce per square inch is 4.75.

The horse power of the engine required to run the fan will be

$$\frac{4.75}{0.66} = 7.15$$

The diameter of the cylinder and the length of stroke of the engine will depend upon the pressure of steam carried and the number of revolutions the engine is to make per minute. If the engine is to be of the direct connected type, so that the fan wheel will be mounted directly on the shaft of the engine, it must make the same number of revolutions that the fan does.

The size of motor required to run the fan will be

$$\frac{4.75}{0.75} = 6.35 \text{ horse power.}$$

It is interesting to compare the cone fan required by the conditions of the problem with an ordinary centrifugal fan of the blower type, of the size required for the work and having an inlet whose diameter is 0.625 the diameter of the fan wheel.

Table IIA shows that for a fan in which r is 0.625, a 10-foot wheel will be required to deliver 30000 cubic feet of air per minute against a pressure of 0.3 of an ounce per square inch. Table VI. shows that this wheel must make 90 revolutions per minute. And Table VIIIA shows that 3.82 horse power will be required to run the wheel.

The difference between the power stated as required for the cone wheel and the blower wheel is greater than it should be because the cone wheel chosen was capable of delivering 10 per cent. more air per minute than was actually required. The horse power actually required by the cone wheel when delivering 30000 cubic feet instead of 33300 is about $\frac{4.75}{1.1} = 4.32$. This, however, is still greater than the power required for the blower fan by about 13 per cent. In other words the cone wheel would be of smaller diameter than the blower wheel, and would therefore probably cost a little less; but the cone wheel would cost 13 per cent. more to operate.

CHAPTER XII.

Disk Fans. While this book is intended as a treatise on centrifugal fans it would not be complete without some reference to disk fans which operate in a totally different way from centrifugal fans. The centrifugal fan as we have shown operates upon the principal of a vortex, but the disk fan operates like a screw. The disk fan consists essentially of disks or blades set so that their center lines are at right angles to an axis and their planes inclined at an angle to the axis. When the blades are made to revolve about the axis the air is forced forward by the action of the disks or blades, in the direction of the axis. The planes of the blades really form parts of helicies, and these helicies move the air by their action on it.

It is evident that as the blades of a disk fan are made to revolve, they tend to create a vortex just as do the floats of a centrifugal fan, but not to the same degree, because the blades or floats of a centrifugal fan are in the same plane as the axis, while the blades of a disk fan are in a plane inclined to the axis. Fig. 31 represents diagrammatically the relative position of a blade and the axis of a disk fan. The upper drawing represents a view looking at the end of the axis, and the lower one represents a plan. In both views *A* represents the axis and *B* the blade. The angle which the plane of the blade makes with the axis is X, shown in the lower drawing of Fig. 31. The blades are usually wider at the outer end than at the end nearest the axis.

It is evident that when the angle X is zero, the blade is in exactly the same position in relation to the axis as the float of the ordinary centrifugal fan. And when the angle X is 90 degrees, the blade will move through the air edgeways and have no appreciable effect in moving the air. When X is between zero and 90 degrees there are two effects on the air: one is the centrifugal effect which makes the air move radially from the center towards the outer point of the blades, and the other is the screw effect which makes the air move in the direction of the axis of the wheel. The angle which makes the centrifugal effect least and the screw effect the greatest is the one which will give the best results.

The ordinary disk fan has usually perfectly straight, flat blades, but many manufacturers

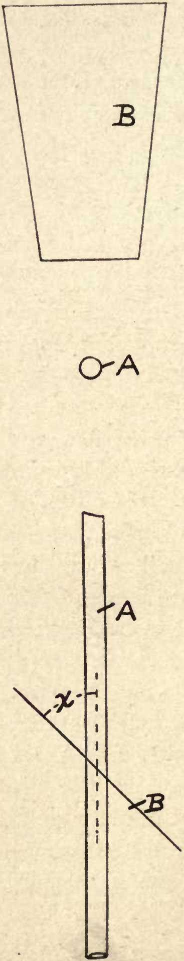


FIG 31

make special disk fans which have the blades curved in different ways so as to increase the screw effect and diminish the centrifugal effect. In order to reduce the centrifugal effect with straight, flat blades the blades are often made to revolve inside of a tube-like casing which prevents the air from escaping at the outer edge of the blades. Even when this is done, however, it is found that there is always more or less escape of air from the blades at the outer ends, and this escape reduces the delivering capacity of the fan. If the fan is made to work against a pressure, the slip or leakage of the air from the blades at the outer edge is increased, and if the pressure be greater than a certain amount it is found that the fan will not deliver any air at all. The air will enter the fan at or near the axis and then be blown *backwards* near the outer edge of the blades; the air will simply circulate back and forth through the fan, entering at the central part and leaving at the outer part, on the same side as that on which it entered.

This circulating of the air through the fan without actually delivering any at the place when it is to go, begins at different pressures for different fans, and depends upon the shape of the blades, their size, and the angle which they make with the axis of the fan.

Disk fans may be divided into two general classes; those having straight, flat blades and those having curved blades.

Those having blades curved at the outer edge so that some advantage is obtained from the centrifugal action are generally said to be of the Blackman type, because they resemble a form of fan first put on the market by the Blackman Fan Co. of England. Fans of this type will work against a slightly higher pressure and will deliver more air than fans of the straight, flat blade type.

It is probably safe to say that disk fans are essentially ventilating fans and should be used only for drawing air from a practically free space and discharging it against no resistance. The slightest resistance to the movement of the air either into the fan or away from it tends to reduce the quantity of air delivered by the fan. These fans cannot discharge air against a wind pressure and when used for ventilating purposes must have the outlet leading from the fan so arranged that the wind cannot blow directly into it.

The number of blades seem to have very little effect upon the results obtained by the use of a disk fan, and fans with a few blades, say five or six, will give about as good as, if not slightly better results than, fans with a larger number of blades.

Many attempts have been made to deduce theoretical formulas for disk fans, but because of the great number of variables that must be considered, and because of the extreme mobility of air and gases, the attempts can hardly be said to be successful. In all the formulas deduced there have appeared certain terms or constants whose values could be determined only by experiment. Again it seems probable that even with a correct and proper formula, any calculation made as to the working of a disk fan must be very carefully used because, as has been said before, a slight resistance to the movement of the air materially affects the amount delivered by the fan, and this may be sufficient to upset all the calculations.

In 1897 Mr. William G. Walker, of London, read a paper on Propeller Fans before the Institution of Mechanical Engineers, in which he gave the results of a great number of experiments made by him with disk fans, or, as he called them, propeller fans.

These experiments showed that the best results were obtained when the blades of straight blade disk fans were set at an angle of 35 or 40 degrees with the axis.

They also showed that the velocity of the tips of the blades was about 4.35 times the mean velocity of the air leaving the fan when working with a free, unobstructed inlet and outlet, or when working at its maximum capacity.

Fig. 32 shows a side view of a disk fan which is belt driven. The air enters on the left and flows through the fan towards the right. That

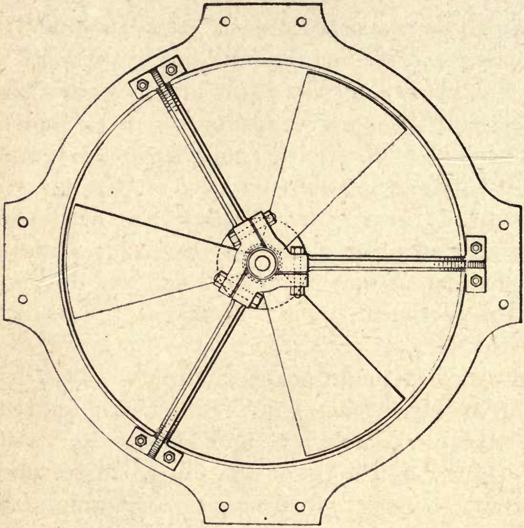


FIG. 33.

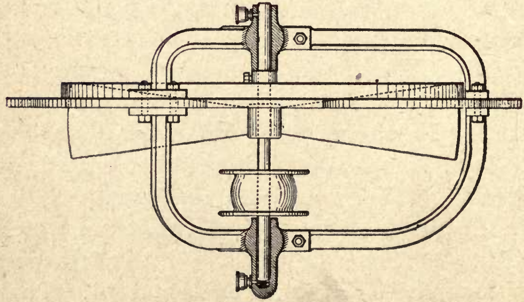


FIG. 32.

is the alr flows *into* the fan on the side next to the pulley.

Fig. 33 shows a view of the fan shown in Fig. 32 when looking at it from the side on which the pulley is. The fan is supposed to be run from left to right, or right handed.

This fan is arranged to be fastened before an opening in a wall so that the air may be taken from a room on one side of the wall and be discharged into a space on the other side.

Number of Revolutions per Minute. The velocity of the tips of the blades of a fan is limited by the strength of the fan, and by the fact that if the velocity be too great the fan will hum and be noisy. There is no danger of breaking and little or no noise if the velocity of the tips of the blades be about 5500 feet per minute. Hence if D be the diameter of the fan in *inches*, and N the number of revolutions made by the fan per minute, we have, assuming that the velocity of the tips of the blades shall be 5500 feet per minute,

$$\frac{\pi D N}{12} = 5500 \tag{88}$$

From which we get

$$D N = 21000 \tag{89}$$

This equation may be used to determine the number of revolutions a disk fan of a given diameter should make, and it applies equally well to all forms of disk fans.

Capacity of a Disk Fan. The capacity of a disk fan or the number of cubic feet of air which it can discharge when working with a free inlet and a free outlet may be determined by equations founded upon experiments.

When a disk fan is working freely the stream of air discharged by it is of the same diameter as the fan and has a mean velocity which bears a constant ratio to the velocity of the tips of the blades. If we call this ratio x, we have that the mean velocity in feet per minute of the stream of air flowing from the fan is

$$\frac{\pi x D N}{12}$$

The area of cross-section of the stream is the area of a circle whose diameter is the diameter of the wheel, that is the area in square feet is

$$\frac{\pi D^2}{4 \times 144}$$

Now calling C the capacity of the fan in cubic feet per minute we have

$$(90) \qquad C = \frac{x\, \pi^2\, D^3\, N}{4 \times 1728}$$

If now we put for $D\,N$ its value as given by (89), we get

$$(91) \qquad C = \frac{x\, \pi^2\, D^2\, 21000}{4 \times 1728}$$

$$= 30.4\, x\, D^2$$

From the experiments by Walker, already referred to, we see that for a disk fan with plane straight blades set at an angle of about 40 degrees with the axis, x is equal to $\frac{1}{4.35}$. Hence the capacity of such a fan is

$$(92) \qquad C = \frac{30.4\, D^2}{4.35} = 7\, D^2$$

From the results of experiments quoted by Mr. Geo. E. Babcock in the Transactions of the American Society of Mechanical Engineers, Vol. VII., 1886, it is found that x for a disk fan of the Blackman type is $\frac{1}{2.68.}$ Hence the capacity of a disk fan with curved blades of the Blackman type is

$$(93) \qquad C = \frac{30.4\, D^2}{2.68} = 11\, D^2$$

It must be remembered that equations (92) and (93) are for fans working with a perfectly free inlet and also a perfectly free outlet; conditions which never do and never can exist in the actual use of a fan. How much the amount of air actually delivered under working conditions will be less than that given by the equations it is impossible to predict, although it is probable that the air delivered by a disk fan under ordinary good working conditions may be taken as about two thirds of the capacity as given by either (92) or (93).

That is we may say that under ordinary working conditions the working capacity or C' is

$$(94) \quad C' = \begin{cases} 5\,D^2 \text{ for a straight blade disk fan.} \\ 7\,D^2 \text{ for a Blackman type fan.} \end{cases}$$

Horse Power Required. The result of Walker's experiments show that the work required to run a disk fan is about three times the work required to give the mean velocity of the leaving stream to the air delivered. And the mean velocity of the leaving air is, for fans with straight blades from Walker's experiments, equal to the velocity of the tips of the blades divided by 4.35; and for fans of the Blackman type, from the experiments quoted by Babcock, equal to the velocity of the tips of the blades divided by 2.68.

We have supposed that the velocity of the tips of the blades shall be 5500 feet per minute, and hence the mean velocity of the air leaving the fan, in the case of a fan with straight blades, is

$$\frac{5500}{4.35} = 1260;$$

and in the case of a fan of the Blackman type it is

$$\frac{5500}{2.68} = 2050.$$

For a fan with straight blades the pressure in ounces corresponding to a velocity of 1260 feet per minute, from (5), is

$$p = \left(\frac{V}{5200}\right)^2 = \left(\frac{1260}{5200}\right)^2$$
$$= 0.059$$

And for a fan of the Blackman type the pressure in ounces corresponding to a velocity of 2050 feet per minute, from (5), is

$$p = \left(\frac{V}{5200}\right)^2 = \left(\frac{2050}{5200}\right)^2$$
$$= 0.155$$

The work required to give a quantity of air a certain velocity is, as has been explained, equal to the number of cubic feet of air moved multiplied by the pressure in *pounds* per *square foot* corresponding to the velocity of the air. That is the work done per minute to give C cubic feet of air per minute a velocity corresponding to a pressure of p ounces per square inch is

$$\frac{144\,C\,p}{16} = 9\,C\,p$$

And the work per minute divided by 33,000 gives the horse power.

For a straight blade disk fan p has been shown is 0.059, and hence the horse power K required to give the air its mean velocity, is

$$(95) \qquad K = \frac{9\,C\,p}{33000} = \frac{9\,C \times 0.059}{33000}$$

$$= \frac{C}{62200}$$

For a fan of the Blackman type p has been shown to be 0.155 and hence the horse power required to give the air its mean velocity is

$$(96) \qquad K = \frac{9\,C\,p}{33000} = \frac{9\,C \times 0.155}{33000}$$

$$= \frac{C}{23600}$$

The horse power required to run the fan is from the results of Walker's experiments equal to $3\,K$ as given by (95) and (96).

Hence the horse power H required for a straight blade disk fan is from (95)

$$H = 3\,K = \frac{3\,C}{62200} \tag{97}$$

If now we put for C in (97) its value as given by (92) we get

$$H = \frac{3\,C}{62200} = \frac{21\,D^2}{62200} \tag{98}$$

$$= \frac{D^2}{3000} \text{ about.}$$

In the same way from (96) and (93) we get that the horse power required for a disk fan of the Blackman type is

$$H = 3\,K = \frac{3\,C}{23600} \tag{99}$$

$$= \frac{33\,D^2}{23600} = \frac{D^2}{700} \text{ about.}$$

Table XXI. gives the revolutions per minute, capacity, working capacity, and horse power required for disk fans with straight, plane blades, as calculated by (89), (92), (94), and (98).

Table XXII. gives the revolutions per minute, capacity, working capacity, and horse power required for disk fans of the Blackman type as calculated by (89), (93), (94), and (99).

It must be remembered that Tables XXI. and XXII. are based upon the supposition that the velocity of the tips of the blades will be equal to 5500 feet per minute. If the velocity be made greater the amount of air delivered by the fan will be increased and the horse power required to run the fan will also be increased. That is if the fan be run at a speed of say 6000 feet per minute, or 10 per cent. greater than the speed used in calculating the tables, the capacity and the working capacity will both be increased by 10 per cent., but the horse power will be increased by about 20 per cent. This is so because horse the power varies as the square of the speed while the capacity and the working capacity vary only as the speed.

An inspection of Tables XXI. and XXII. shows that in order to deliver the same amount of air, a smaller fan of the Blackman type can be used than can be used if the fan has straight blades, but the horse power required for a Blackman fan will be much greater than the horse power

TABLE XXI.

Disk fans with straight, plane blades.

Diameter of wheel in inches	Revolutions per minute	Capacity	Working capacity	Horse power required
18	1170	2270	1620	0.11
24	875	4040	2870	0.18
30	700	6300	4500	0.30
36	585	9100	6500	0.43
42	500	12300	8800	0.59
48	435	16200	11600	0.77
54	390	20500	14600	0.97
60	350	25300	18100	1.20
66	320	30500	21800	1.46
72	290	36400	26000	1.73

TABLE XXII.

Disk fans of the Blackman type.

Diameter of wheel in inches	Revolutions per minute	Capacity	Working capacity	Horse power required
18	1170	3560	2270	0.46
24	875	6350	4040	0.82
30	700	9900	6300	1.28
36	585	14200	9100	1.76
42	500	19400	12300	2.52
48	435	25400	16200	3.30
54	390	32200	20500	4.17
60	350	39700	25300	5.15
66	320	48000	30500	6.20
72	290	57000	36400	7.40

required for the fan with straight blades. This is so because the velocity of the air leaving the Blackman fan is much greater than the velocity of the air leaving the fan with straight blades. And because of the fact that the velocity of the air leaving the Blackman fan is so much greater than the velocity of the air leaving the fan with straight blades, the Blackman fan may be used for moving air against low pressures, pressures in the neighborhood of about 0.10 of an ounce per square inch.

CHAPTER XIII.

Choosing a Centrifugal Fan. The size of wheel and the proportions of the wheel and housing for a centrifugal fan depend entirely upon the conditions under which the fan is to work. In all cases, however, the diameter of the wheel and the number of revolutions per minute at which it must be run depend solely upon the quantity of air to be delivered per minute and the pressure against which the fan must work. The quantity of air to be delivered per minute is always determined by the known conditions of the problem in hand, and the pressure against which the fan must work must be calculated from the conditions under which the air must be moved. This pressure is equal to the head required to give to the air the velocity which it has when it finally leaves the ducts or flues through which it is forced by the fan, plus the head required to overcome the resistance due to friction of the air in its passage from the fan outlet to the outlets of the flues or ducts. If there is any resistance to the entrance of the air into the fan, the pressure required to overcome it must also be added to the sum of the other pressure in order to get the total pressure against which the fan must work.

Having determined the quantity of air to be moved per minute and also the pressure against which the fan must work, the next thing to decide is the ratio, r, of the diameter of the inlet to the diameter of the wheel. For comparatively high pressures choose a fan with a small value of r, and for low pressure choose a fan with a rather large value of r. For ordinary heating and ventilating work r should be taken as 0.625 or 0.707, but for foundry work or for certain kinds of gas work r should be taken less than 0.625. The larger the value of r the smaller will be the diameter of the wheel required, but the greater will be the horse power required to run the fan.

The value of r will also affect the size of the housing.

Whether the fan shall be a full housed, top, horizontal discharge; or a three-quarter housed, bottom, horizontal discharge; or even some other style of fan will depend upon the space available for the apparatus, and the peculiarities of the place in which it is to be set up.

Having determined the quantity of air to be delivered, the pressure in ounces per square inch against which the fan must work, and the ratio of the diameter of the inlet to the diameter of the wheel, *then take as the size of wheel to be used, that wheel which when working at its capacity against the given pressure will deliver the required quantity of air per minute.* In other words, the pressure due to the velocity of the tips of the floats should be equal to the pressure against which the air must be delivered.

The considerations which govern the size and proportions of a fan for heating and ventilating work are different from those which are encountered in mechanical draft work, or in work pertaining to drying, or to the removal of dust or waste products from a factory, and it is impossible to discuss each different class of work here. It suffices to say that if the fan be chosen with proper reference to the work it is to do and the conditions under which it is to be run, and then be properly set up, it will be found to give most satisfactory results and to require very little care or attention.

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