

April 8, 1952

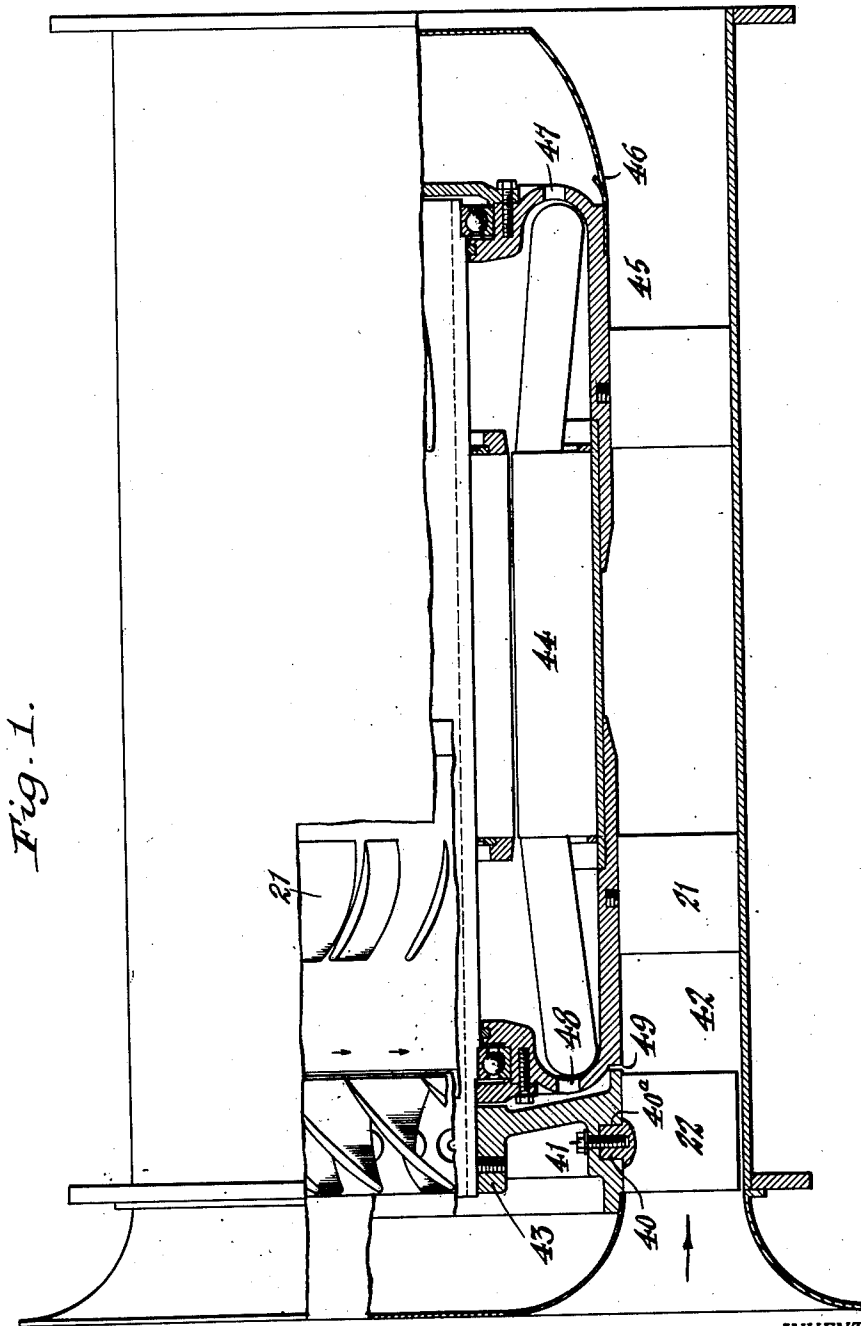
J. G. SAWYER

2,592,471

AXIAL FLOW FAN

Filed Aug. 22, 1946

3 Sheets-Sheet 1



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April 8, 1952

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AXIAL FLOW FAN

2,592,471

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3 Sheets-Sheet 2

Fig. 2.

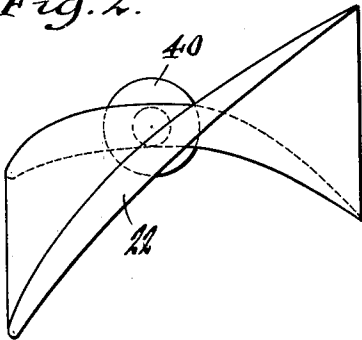


Fig. 3.

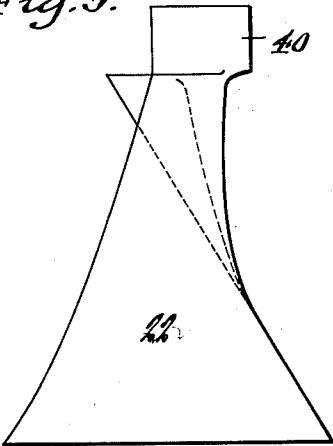


Fig. 4.

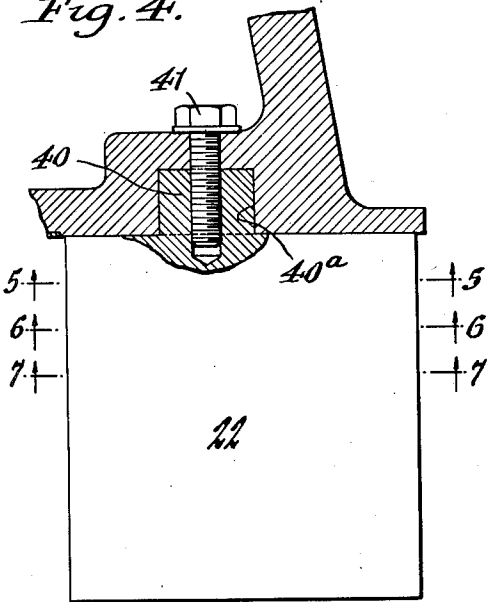


Fig. 5.

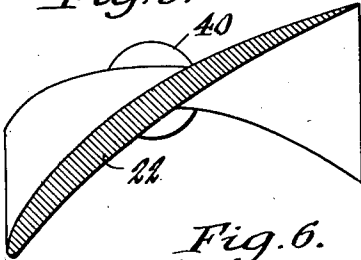


Fig. 6.

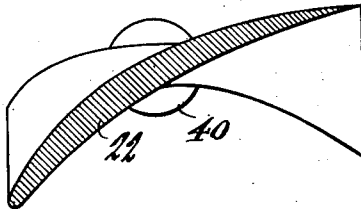
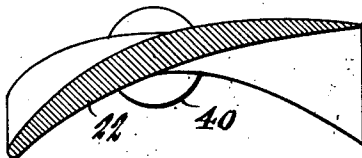


Fig. 7.



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3 Sheets-Sheet 3

Fig. 8.

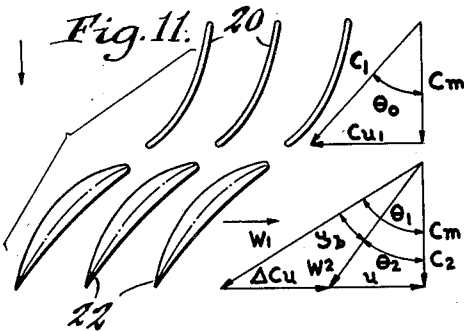
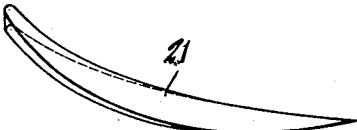


Fig. 9.

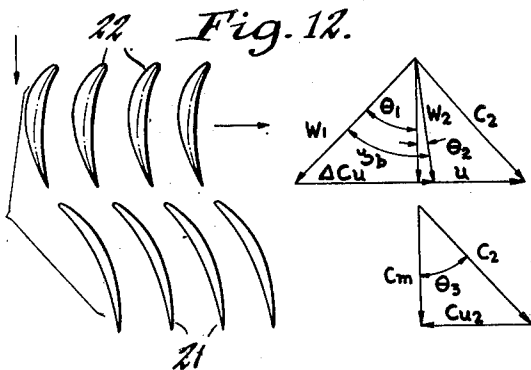
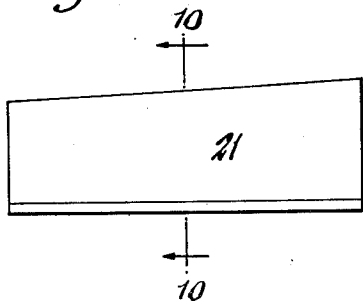
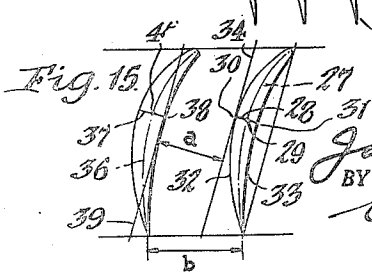
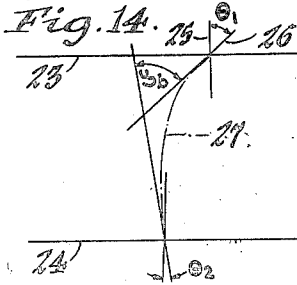
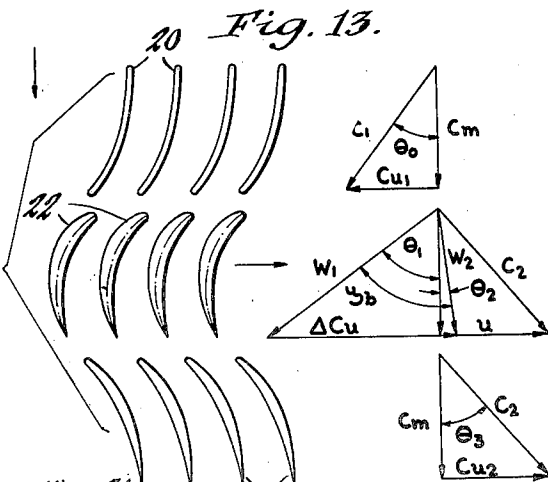
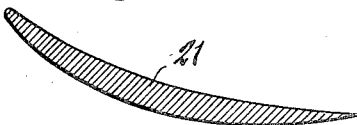


Fig. 10.



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2,592,471

AXIAL FLOW FAN

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Application August 22, 1946, Serial No. 692,207

7 Claims. (Cl. 230—122)

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This invention relates to certain new and useful improvements in fans or blowers of the axial flow type designed to produce a current of air or other fluid or to produce a pressure rise.

The capacity, or the ability of fans of this type to do work, depends upon two factors, that is, the difference in pressure between the inlet and outlet pressures of the fan, represented by the symbol Δp and the axial speed of flow (in cubic feet per second) of the gas delivered, represented by the symbol Q . The pressure difference obeys the following equation:

$$\Delta p = \psi \frac{\rho}{2} (\pi N D)^2 \quad (1)$$

where ψ is a dimensionless coefficient called the pressure coefficient, ρ is the density of the gas, N is the frequency of revolution of the fan runner, and D is the outside diameter of the fan runner. The flow of the gas delivered obeys the equation:

$$Q = \phi N D^3 \quad (2)$$

where ϕ is another dimensionless coefficient called the flow coefficient. It is easily apparent that, for a given size fan and a given rotational speed, the greatest capacity for work and the highest pressures are secured with fans having relatively high pressure and flow coefficients. In addition, the developed pressure obviously varies directly with the pressure coefficient. Both pressure and flow coefficients depend principally upon the shape, configuration and position of the fan blades, so that the proper design of the fan blades is a factor of the utmost importance in securing high pressures and high capacities.

One of the objects of the invention is to provide a fan of this character which is so designed and constructed as to develop maximum coefficients of pressure and capacity at minimum speeds and with a minimum of noise, and wherein the fan blades are so spaced and curved on the stream filament principle to develop pressure coefficients of the order of unity and higher.

Another object is to provide a motor-driven axial flow fan which is designed to take the boundary layer air from the surfaces of the parts and direct it into the path of the motor to effectually cool it and improve the diffusing action of the fan.

Other features of the invention reside in the construction and arrangement of parts hereinafter described and particularly pointed out in the appended claims.

In the accompanying drawings: Figure 1 is a sectional elevation view of the fan embodying my

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invention. Figure 2 is a top plan view of one of the fan blades. Figure 3 is an end view thereof. Figure 4 is a sectional side view of one of the blades showing the manner of mounting the same. Figures 5, 6, and 7 are horizontal sections taken on the correspondingly numbered lines in Figure 4. Figure 8 is a top plan view of one of the guides or vanes. Figure 9 is an end view thereof. Figure 10 is a cross section taken on line 10—10, Figure 9. Figures 11, 12, and 13 are schematic profile views, with vector diagrams, of various fan blade sections and guide vane sections. Figures 14 and 15 are geometrical drawings showing the method of developing the fan blade and aft vane profiles and contours.

Similar characters of reference indicate corresponding parts throughout the several views.

In accordance with the present invention I have discovered a principle in the design of axial flow type fans, whereby pressure coefficients of the order of unity and above may be developed with efficiencies above those heretofore obtained. In obtaining these results I have found it necessary to so design the blade curvature and its thickness so as to accomplish most of the turning of the fluid medium and develop most of the pressure in the early stages of the interblade fluid passage; that is, most of the work is done upon the fluid in the inlet or up-stream portion of the blade.

In this type of fan, the blades act to guide or direct the flow of air through the runner section. This is known as the "laminar flow" or "stream filament" principle. By curving the fan blades, the air which initially enters the runner section in a direction opposed to the direction of motion of the fan blades, is caused to flow in laminar along a curved path roughly parallel to the walls defined by two adjacent fan blades, and to leave the runner section in a different direction.

The discharge guide vanes are also curved and they serve to transform any residual rotational component of flow of the air as it leaves the fan section into axial flow.

In designing a fan for commercial use, the desired pressure differential, the desired volume of air flow, the number of revolutions per minute, the density of the air, and the space available, including the fan diameter and thickness, are usually presented to the designer as conditions for which the fan is to be designed. From these design conditions, the pressure coefficient ψ is obtained from equation (1) above. If the pressure coefficient thus calculated proves to be higher than is theoretically or practically obtainable, it is then impossible to design a fan which will meet

the design conditions. As pressure coefficients up to 3 and above with good efficiency are obtainable with the fans of my invention, it is evident that it is possible to satisfy much more exacting design conditions than were heretofore possible.

Having determined that it is possible to attain the calculated pressure coefficient, the next step is to calculate the required angles of the fan blades and guide vanes, if the latter are used. The curvature of the fan blades imparts to the air stream passing through the runner section a change in component of absolute rotational velocity designated as ΔCu . The value of ΔCu is determined by the design conditions, according to the following equations:

$$-Cu = \frac{\Delta p}{\eta p u} \quad (3)$$

where η is the efficiency of the fan (assumed to be .75) or other reasonable value, determined according to experience and u is the speed of the fan blade tips (equal to πND). ΔCu , however, increases with the angle of air turn ϕ_a (assumed at the present stage to be equal to ϕ_b), as that the pressure differential Δp obtainable depends upon the angle through which the air can be turned.

In the preferred embodiment of this invention, guide vanes, either inlet or aft vanes or both, depending upon the particular installation are employed. With this construction, the air leaves the fan in an axial direction and the work done in turning the air manifests itself as a rise in pressure. The runner blades are set at the angle ϕ_1 , which is calculated according to the vector diagrams shown in Figures 11, 12, 13 depicting forms of the invention employing inlet vanes 29 only in Figure 11, aft vanes 21 only in Figure 12, and both inlet and aft vanes in Figure 13, in combination with fan runner blades 22. In these diagrams, the symbol C_m represents axial velocity, and W_1 and W_2 the air velocities relative to the fan blades at the inlet and outlet ends, respectively, of the runner section, while the remaining symbols are as previously indicated. The value of C_m is expressed in linear measure per second and is equal to

$$\frac{4Q}{\pi(D^2 - d^2)}$$

Accordingly, since the values of C_m , ΔCu and u are determined from the design conditions, it is a relatively simple matter to calculate the angles ϕ_1 and ϕ_b , and the values of ϕ_2 (equal to $\phi_1 - \phi_b$), W_1 and W_2 if desired.

Having determined these angles, the camber line of the blade is next fixed and reference is had to Figure 14 wherein lines 23 and 24 represent the inlet and outlet planes, respectively, of the fan runner, and the distance between them is fixed by the design conditions. A line 25 is erected perpendicular to line 23 and a line 26 drawn through the intersection of 23 and 25 in such a fashion that the angle between 25 and 26 is equal to ϕ_1 . The object is to then draw a camber line curve 27 tangent to 26 at the point of intersection with 23 and intersecting line 24 at an angle of $90^\circ - \phi_2$, and such that the change of length S along the camber line with respect to the change of width a between adjacent blades reduces linearly to approximately 70% of the initial value. Mathematically, this is expressed as follows:

$$da/ds = k - KS \quad (4)$$

Where, k and K are positive constants.

Letting $\mu = da/ds$, these constants are expressed as follows:

$$K = \frac{A_2 - A_1}{S_2^2 \left[\frac{1}{1 - \mu_2/\mu_1} - \frac{1}{2} \right]} \quad (5)$$

and

$$k = \frac{KS_2}{1 - \mu_2/\mu_1} \quad (6)$$

$\mu_2/\mu_1 = .70$ approx.

In the course of my investigations, I have found that the above equations (4), (5) and (6) are most nearly satisfied by selecting as the camber line curve an ellipse having a properly chosen ratio of the minor axis to the major axis. For this purpose, axis ratios of 0.7, 0.8, 0.9 may be chosen, and having once drawn the curve 27, its radius of curvature R is variable but determinable at any particular point, while the value of a is determined at one point. I have found that a increases with R , and that the ratio varies somewhat from linearity. I have also found that optimum conditions are secured when R/a varies linearly from approximately 3 at the inlet to 4 at the outlet. In any case, having selected the values of R/a , it is easy to determine the blade pitch or blade spacing b from the following equation:

$$a = b \cos(\theta - \tau) \quad (7)$$

where τ is the thickness of the blade section. At the leading or trailing edge, θ is equal to θ_1 , or θ_2 , respectively, $a = 3R$ or $4R$ respectively, and τ is equal to c . From the latter, b is determined throughout, and the value of $-\tau$ throughout the range is then determined from Equation 7.

Referring now to Figure 15, the value of τ is calculated with reference to each particular point on the camber line 27, according to Equation 7. It is plotted at any point by erecting a perpendicular 28 to the camber line at a particular point 29 on such line, and laying off a distance $\frac{1}{2}\tau$ on each side of the camber line. The loci of the points 30 and 31 at distances of $\frac{1}{2}\tau$ from such camber line on the perpendicular line 28 then determine the blade profile, having a convex side 32 and a concave side 33.

The width a between adjacent blades may now be defined as follows, with reference to each particular point on the camber line 27. A line 34 perpendicular to the line 28 is erected at the point 30, and a point 35 on the camber line 26 of the adjacent blade is selected to be at the same location of such line 35 as the particular point 29 is on the line 27. A line 37 perpendicular to the line 36 is erected as before, and the point 38 laid off at a distance $\frac{1}{2}\tau$ from line 36. A line 39 perpendicular to 37 at point 38 is erected, and the perpendicular distance between lines 34 and 39 is defined as a .

After a single blade profile is plotted as described above, and a second similar blade profile plotted from the similar camber line 36 at a distance b therefrom, the values of a with respect to S are checked at various points along the camber line, to see whether or not equations (4) and (5) hold. If these conditions are not satisfied, then the form of the camber line or the value of R/a have not been properly selected, and a different curve or a different value of R/a or both must be selected. Thus an ellipse of slightly different eccentricity or some other different curve may be tried. Again, the blade pitch b may be replotted with lower values of R/a . Fi-

nally, the value of b must be such that D is an integral multiple M thereof, since it is impossible to construct a fan with a fractional number of blades.

The calculations as described above determine the blade profile at the tip. The profiles at other points inward of the tip are determined in a similar fashion. In place of the value, u (tip speed) in equation (3) and the vector diagrams accompanying Figures 11, 12, and 13, a somewhat lesser value $2\pi N r_m$ is used, where r_m is the radius of the runner section at the profile point to be determined. This will result in different values of θ_1 , θ_2 , θ_3 etc., and consequently different blade profiles and blade spacings. All profiles should as closely as possible satisfy the conditions previously described except that instead of $b = \pi M D$, the relationship $b = 2\pi M r_m$ should hold, where M obviously must have the same value at all profiles. As profiles are determined nearer the axis of the fan runner section, the deviations from the prescribed or assumed conditions may be sufficiently great as to make it impractical to proceed further. If it has not been determined otherwise, this will determine the value of d . When a sufficiently representative number of blade profiles have been determined, they are combined into a smooth blade contour, the ends of the blade chords should preferably lie in two smooth curves, and to accomplish this may require some adjustment in their relative positions. A typical such blade construction (for a fan having runner section and aft vanes only) is illustrated in Figures 2 to 7 inclusive of the drawings. Each blade 22 is detachably anchored to the runner wheel of the fan and for this purpose has a stud 40 thereon seated in a companion socket 40^a in the wheel and secured thereto by a bolt 41.

If the inlet guide vanes are used, they should be curved so as to impart to the air a rotational component of absolute velocity opposite in direction to that imparted by the fan runner section. If the inlet guide vanes alone are used, the rotational component should preferably be equal in magnitude to that imparted by the fan runner section, in order that the air from the runner section may be delivered in an axial direction. As the mean flow of the air entering the inlet guide vanes is in a straight line in an axial direction, uncomplicated by rotational velocity, the problem of inlet guide vane design is simply only of duct design, to turn the air through the required angle. At the inlet side of the inlet guide vanes, therefore, the walls of the vanes should be parallel to the direction of flow or perpendicular to the inlet plane of the vanes. In the form shown in Figure 11, the angle θ_0 at the outlet of the inlet guide vanes is determined by the equation

$$\tan \theta_0 = \frac{\Delta C u}{C_m} \quad (8)$$

The perpendicular distance between the inlet and outlet planes of the inlet guide vanes is a design condition. It is sufficient to build the vanes out of ordinary sheet metal, tapered at the inlet and outlet edges to secure streamline flow. They are bent simply to form a contour whose profile is a segment of a circle. The chord spacing ratio need not be as high as in the case of the fan blades, but is preferably at least 0.75.

If aft vanes or discharge guide vanes are used, they should also be curved so as to impart to the air a rotational component of absolute velocity

opposite in direction to that imparted by the fan runner section. If aft vanes alone are used the rotational component should preferably be equal in magnitude to that imparted by the fan runner section, in order that the air from the aft vanes may be delivered in an axial direction. Unlike the inlet guide vanes, however, the contour should be developed in a fashion similar to that employed for the fan blades proper, except that the direction of curvature is opposite. The inlet angle θ_3 is negative and is determined from the vector diagram, as shown in Figures 12 and 13 while the outlet angle θ_4 is theoretically zero, since the flow at this point should be axial. In practice, however, the air turn is not quite as great as the blade turn, so that the outlet angle θ_4 is slightly positive, usually about 5 degrees. Having determined the angles θ_3 and θ_4 , the camber line, blade profile, and blade contour are determined in exactly the same fashion as for the fan blades proper. The chord-pitch ratio should be of the same order of magnitude as in the case of the fan blades proper. A typical construction, for use in combination with the fan blades illustrated in Figures 2 to 7 inclusive, is illustrated in Figures 8 to 10 inclusive.

In most installations, either inlet guide vanes or aft vanes are used. It is usually preferred, however, not to use both inlet and aft vanes, because of space limitations and because the mathematical calculations required are more involved. Nevertheless, under certain conditions, particularly where extremely high pressures are desired, both inlet and aft vanes may be used. Such a structure is illustrated in Figure 13.

Referring now to Figure 1, this illustrates a conventional structure embodying my invention, in which a fan of the blade and vane arrangement shown in Figure 12 is employed. The fan is installed in a duct 42, the runner section 43 with its blades 22 being driven by an electric motor 44, in the direction indicated by the small arrows. The fan blades 22 and the aft guide vanes 21 are designed as hereinbefore indicated, the combination serving to force the air through the duct in the direction indicated by the large arrow.

While the invention has been generally described with reference to fans, I have found that the principle of my invention is particularly applicable to compressors where economy of space is a prime consideration. Hitherto, axial flow blowers have not ordinarily been used as compressors because of the low pressure coefficients available, but with my invention, two or more runner sections may be advantageously and obviously combined on a common shaft, to form a multi-stage compressor.

The described method of determining blade profile results in the accomplishment of most of the fluid turning and diffusion in the early part of the blade passage, where the boundary layer is small, and flow conditions are more favorable. A more generous R/a and a lower rate of diffusion da/ds are employed toward the outlet where the boundary layer conditions are less favorable. Then this portion of the blade is less taxed. This method will reduce losses to a minimum, and as tests have indicated, higher efficiencies are obtainable.

For the purpose of effectually cooling and improving the diffusing action of the fan, the boundary layer air is directed into the path of the motor, the air circulating from the duct 42 being directed into one end of the motor housing 45 and out of its opposite end, air intake ports

46, 47 being provided at one end of the housing and outlet ports and passages 48, 49 being provided at the other end thereof.

I claim as my invention:

1. An axial flow blower, comprising means forming a duct, a runner section disposed therein having a plurality of blades thereon spaced to define fixed passages therebetween, a stationary guide section disposed in said duct in axially-spaced adjacent relation to said runner section and having a plurality of vanes thereon spaced to define fixed passages therebetween, the runner blades being curved to form a concave advancing face and a convex retreating face with respect to the direction of rotation of said runner section and said guide vanes being curved with their convexity and concavity opposed to the convexity and concavity of said blades, the rate of change of width of the passage between adjacent blades with respect to the change of length of the blades' camber line being reduced linearly from its initial value at the blades' leading edge to 70% of said initial value at the blades' trailing edge, and the ratio of the blades' camber line radius to the width of the passage between blades varying linearly from approximately 3.0 at the leading edge to approximately 4.0 at the trailing edge.

2. An axial flow blower, means forming a duct, a rotatable runner section disposed therein and having a plurality of blades thereon spaced to define fixed passage therebetween, a stationary inlet guide section disposed in said duct in axially-spaced adjacent relation to said runner section and having a plurality of vanes thereon to define fixed passages therebetween, the runner blades being curved to form a concave advancing face and a convexing retreating face with respect to the direction of rotation of said runner section and said vanes being curved with their convexity and concavity opposed to the convexity and concavity of said blades, and the rate of change of width of passage between adjacent blades with respect to change of length of the blades' camber line being reduced linearly from its initial value at the blades' leading edge to 70% of said initial value at the blades' trailing edge, and the ratio of the blades' camber line radius to the width of blade passage varying linearly from approximately 3.0 at the inlet to approximately 4.0 at the trailing edge.

3. An axial flow blower, comprising means forming a duct, a rotatable runner section disposed therein and having a plurality of blades thereon spaced to define fixed passages therebetween, a stationary aft guide vane section disposed in said duct in axially-spaced adjacent relation to said runner section and having, a plurality of vanes to define fixed passages therebetween, the runner blades being curved to form a concave advancing face and a convex retreating face with respect to the direction of rotation of said runner section and said vanes being curved with their convexity and concavity opposed to the convexity and concavity of said blades, and the rate of change of width of passage between adjacent blades with respect to change of length of the blades' camber line being reduced linearly from its initial value at the blades' leading edge to 70% of said initial value at the blades' trailing edge, and the ratio of the blades' camber line radius to the width of blade passage varying linearly from approximately 3.0 at the inlet to approximately 4.0 at the trailing edge.

4. An axial flow blower, comprising means

forming a duct, a rotatable runner section disposed therein and having a plurality of blades thereon spaced to define fixed passages therebetween, a stationary inlet guide section disposed in said duct in axially spaced relation at one side of said runner section and having a plurality of vanes thereon to define fixed passages therebetween, a stationary aft guide section disposed in said duct in axially spaced relation at the other side of said runner section and having a plurality of aft vanes thereon to define fixed passages therebetween, said blades being curved to form a concave advancing face and a convex retreating face with respect to the direction of rotation of said runner section and both sets of said vanes being curved with their convexity and concavity opposed to the convexity and concavity of said blades, and the rate of change of width of passage between adjacent blades with respect to change of length of the blades' camber line being reduced linearly from its initial value at the leading edge of the blade to 70% of said initial value at the blades' trailing edge, and the ratio of the blades' camber line radius to the width of blade passage varying linearly from approximately 3.0 at the inlet to approximately 4.0 at the trailing edge.

5. An axial flow fan, comprising means forming a duct, a rotatable runner section disposed therein having blades thereon to define fixed passages therebetween, a stationary inlet guide section disposed in said duct in axially-spaced adjacent relation to said runner section and having vanes thereon, the blades being curved to form a concave advancing face and a convex retreating face with respect to the direction of rotation of said runner section and the guide section vanes being of opposing curvature to said runner-blades, and the rate of change of width of passage between adjacent blades with respect to change of length of the blades' camber line being reduced linearly from its initial value at the leading edge to approximately 70% thereof at the trailing edge as determined by the following formulas:

$$da/ds = k - KS$$

where k and K are positive constants and letting $\mu = da/ds$, these constants are expressed as

$$K = \frac{A_2 - A_1}{S_2^2 \left[\frac{1}{1 - \mu_2/\mu_1} - \frac{1}{2} \right]}$$

and,

$$k = \frac{KS_2}{1 - \mu_2/\mu_1}$$

$$\mu_2/\mu_1 = .70 \text{ approximately.}$$

6. An axial flow fan, comprising means forming a duct, a rotatable runner section disposed therein having blades thereon to define fixed passages therebetween, a stationary inlet guide section disposed in said duct in axially-spaced adjacent relation to said runner section and having vanes thereon, the blades being curved to form a concave advancing face and a convex retreating face with respect to the direction of rotation of said runner section and the guide section vanes being of opposing curvature to said runner-blades, and the rate of change of width of passage between adjacent blades with respect to change of length of the blades' camber line being reduced linearly from its initial value at the leading edge to approximately 70% thereof at the trailing edge as determined by the following formulas:

$$da/ds = k - KS$$

where k and K are positive constants and letting $\mu=da/ds$, these constants are expressed as

$$K=\frac{A_2-A_1}{S_2^2\left[\frac{1}{1-\mu_2/\mu_1}-\frac{1}{2}\right]}$$

and,

$$k=\frac{KS_2}{1-\mu_2/\mu_1}$$

$\mu_2/\mu_1=.70$ approximately.

and the blade spacing being determined, after having selected the radius of curvature value R/a , from the following formula:

$$a=\beta \cos (\theta-\tau)$$

7. An axial flow fan, comprising a rotatable runner section having blades thereon, a stationary inlet guide section having vanes thereon, the blades being curved to form a concave advancing face and a convex retreating face with

respect to the direction of rotation of said runner section and the guide section vanes being of opposing curvature to said runner-blades, the guide vanes having their inlet sides substantially parallel to the direction of flow and their outlet sides being shaped as determined by the formula,

$$\tan \theta_0=\frac{\Delta C u}{C_m}$$

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