May 17, 1960 B. C. ECK 2,936,948

AXIAL BLOWER WITH CONE-SHAPED HUB

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FIG. 1

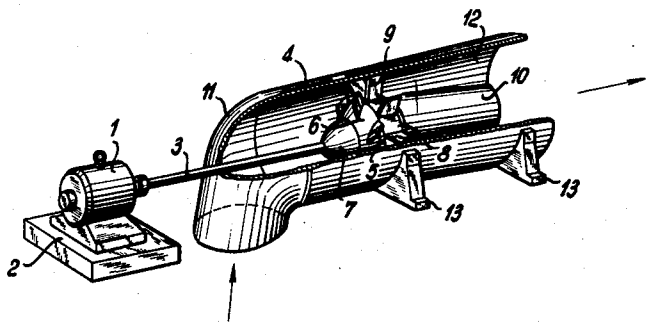


FIG. 2a

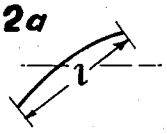
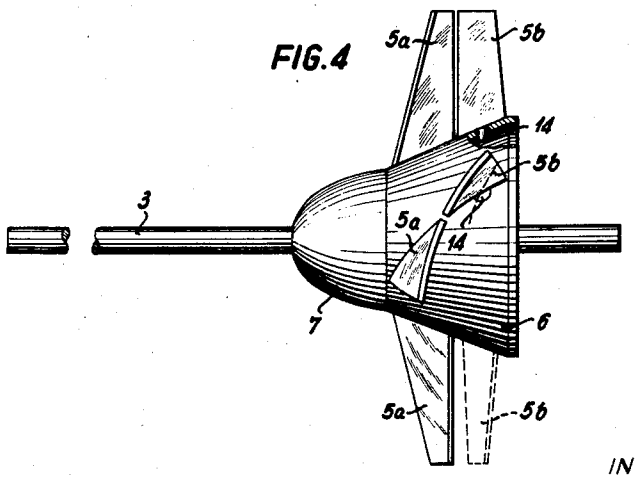


FIG. 2

FIG. 4



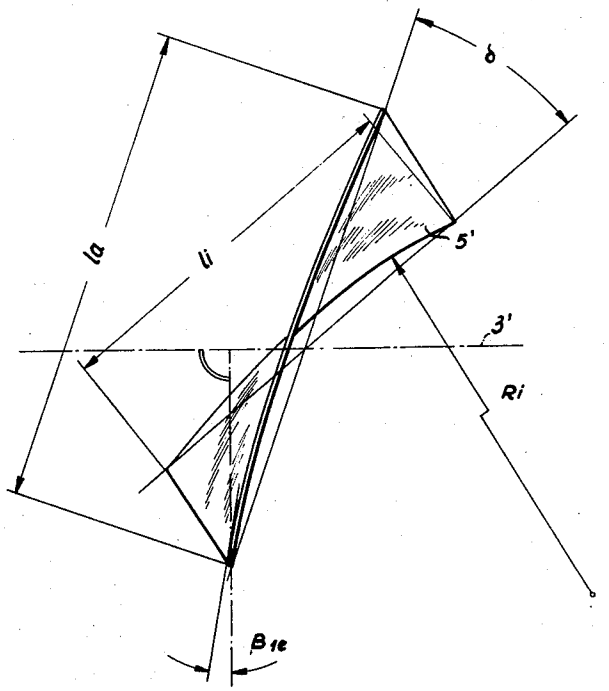
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FIG. 3



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AXIAL BLOWER WITH CONE-SHAPED HUB

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7 Claims. (Cl. 230—120)

The present invention relates to blowers and, more particularly, to an axial blower with a cone-shaped hub. Axial blowers having a meridian cross section which decreases in the direction of flow of the fluid delivered thereby have been known for a long time. Such blowers which usually are called meridian accelerated axial blowers are obtained by making the hub gradually to increase or by drawing in the outer mantle or by both. This type has the great advantage that without profiling, i.e. by providing simple sheet metal blades, pressures can be obtained which approach the values obtained with radial blowers. Non-profiled blowers are necessary if, for instance strongly corrosive dust particles are included in the fluid to be delivered. This is the reason why this particular type of blowers has been employed especially when suction blowers were required. These meridian accelerated axial blowers are not to be confused with axial blowers in which due to the noticeable change in density the meridian cross sections have to be changed (axial compressors). Blowers of this type with a considerably high degree of efficiency—above 80%—have been known. These heretofore known blowers, however, have the drawback that they yield a rather unfavorable characteristic of which practically a small range only is satisfactory which brings about the necessity of designing such blower very precisely. This fact enhances great drawbacks and makes necessary later corrections if, as is often the case, pressure and quantity were not precisely known at the time the blower was designed. Another drawback consists in that in spite of all efforts, the high maximum degrees of efficiency possible nowadays with normal axial blowers could not be obtained heretofore. Still another drawback of the heretofore known axial blowers consists in that, when the blower works in the neighborhood of the steeply declining section of the characteristics which declines with decreasing delivery, strong noises occur which have a rather disturbing effect.

Whereas with normal axial blowers, the influence of the hub is determined only by the ratio of the hub diameter to the outer diameter, with meridian accelerated axial blowers a plurality of structural features may be varied which features are:

(*a*) The ratio of the rear diameter of the hub to the outer diameter.

(*b*) The ratio of the free meridian entrance annular or ring surface to the free meridian exit ring surface.

(*c*) The merging lines or contour of the hub. In this connection the difference between an increase along a cone or a more or less curved design is to be considered.

(*d*) A possible outer change in the contour of the mantle line for instance by drawing in the mantle or by enlarging the same.

In addition to the above, similar to normal axial blowers, the various blade features may be varied, i.e. the number of blades, the curvature of the blades, twist, etc.

It will be evident that in view of the numerous possibilities of variations possible with meridian accelerated

axial blowers, the optimum design of such meridian accelerated axial blower is considerably more difficult than the dimensioning of a normal axial blower. In addition thereto, with normal axial blowers a law is available by means of which nowadays almost all questions in connection with the normal axial blower can be solved by pure theoretical means. The application of this law to meridian accelerated axial blowers has up to date been unsuccessful. This failure is due to the fact that the Bauersfeld law as basis of said theory is not applicable to meridian accelerated blowers. Thus, when designing a meridian accelerated axial blower, the designer is rather helpless and has to do considerable guesswork. The experience with centrifugal pumps is of little value in this connection because due to cavitation, the blades of centrifugal pumps have to be laid out differently and would not yield optimum results with meridian accelerated axial blowers.

It is, therefore, an object of the present invention to provide a meridian accelerated axial blower which will overcome the above mentioned drawbacks.

It is another object of this invention to provide a meridian accelerated axial blower with specific structural features which have to be followed in order to obtain improved efficiency.

It is a still further object of this invention to provide a meridian accelerated axial blower which is of relatively simple construction and high efficiency.

These and other objects and advantages of the invention will appear more clearly from the following specification in connection with the accompanying drawings, in which:

Fig. 1 is a perspective view, partly in section of a meridian accelerated axial blower.

Fig. 2 illustrates on a scale somewhat larger than the scale of Fig. 1 the impeller with cone-shaped hub shown in Fig. 1.

Fig. 3 is a diagrammatic representation of an impeller blade.

Fig. 4 is a view of an impeller with blade adjustment.

In order to overcome the above outlined drawbacks and to materialize the objects set forth above, investigations have been carried on over a number of years and have been verified by numerous tests as a result of which it has been found according to the present invention that with axial blowers having a cone-shaped hub and in which the outer mantle is cylindrical or nearly cylindrical, the ratio of the diameter of the rear portion of the hub to the outer blade diameter must be less than two-thirds and must be greater than one-half, whereas at the same time the hub is to be designed as cone-shaped or approximately cone-shaped mantle with a generating agle γ between 10° and 20° while the ratio of the free meridian entrance annular or ring surface to the free meridian exit annular or ring surface should be within the limits of from 1.15 to 1.35.

Referring now to the drawings and Fig. 1 thereof in particular, the arrangement shown therein comprises a motor **1** mounted on a base **2** and drivingly connected through a shaft **3** with the cone-shaped impeller **6**. The cone-shaped impeller **6** is provided with impeller blades **5** and at its intake side is connected to shaft **3** by means of a head **7** the contour of which has been designed in conformity with the general factors governing the flow of a fluid along surfaces. Behind the impeller **6** there is provided the guiding wheel **8** which carries the guiding blades **9**. This guide wheel **8** simultaneously serves as bearing for shaft **3** and furthermore carries the diffuser core **10**. The blower proper is surrounded by a cylindrical mantle **4** which at its intake end merges with an intake elbow section **11** and at its discharge end merges with a discharge connection **12**, said discharge connection **12** being designed as diffuser mantle. The cylin-

drical blower mantle **4** and the diffuser mantle **12** are supported by supports **13**.

The impeller **6** shown by itself on a larger scale in Fig. 2 has its cone-shaped hub arranged on shaft **3**. The head **7** which connects the cone-shaped hub **6'** of the impeller **6** with shaft **3** at the intake end, is designed in connection with principles governing the flow of fluid along surfaces. Half of the conical angle of the cone-shaped hub **6'** is designated with the letter γ as illustrated in Fig. 2 of the drawings.

The various relative dimensions or diameters critical for a blower according to the present invention and shown in connection with Fig. 2 are as follows: the letter d designates the pitch diameter of the impeller blades, whereas the outer diameter of the impeller blades is designated with the letter d_2. The front diameter of the hub **6'** is designated d_{e1}, and the diameter of the rear portion of the hub is designated with the letters d_{a1}. The diameter d_{e1} of the front portion of the hub and the outer diameter d_2 of the blades determine the free meridian entrance ring surface

$$\frac{\pi}{4}(d_2^2 - d_{e1}^2)$$

The diameter d_{a1} of the rear portion of the hub and the outer diameter d_2 of the blades determine the free meridian exit ring surface

$$\frac{\pi}{4}(d_2^2 - d_{a1}^2)$$

Fig. 2*a* shows the contour of a blade the length of which is designated with the letter *l*.

Fig. 3 diagrammatically illustrates an impeller blade **5** as seen from the top. In this diagrammatic illustration, the impeller is designated with the reference numeral **5'**, whereas the impeller axis shown as a dot dash line is designated with the reference numeral **3'**. It has been found that the rear hub diameter d_{a1} must be less than approximately two-thirds of the outer diameter d_2 of the blades. Preferably the outer mantle is cylindrical or approximately cylindrical, whereas the hub should have a cone shape or an approximate cone shape. Half the conical angle γ should be from 10 to 20°. The cone-shaped hub of the impeller **6** may sharply or in a somewhat rounded manner merge into an approximately cylindrical discharge or intake. This merging portion may be located within or outside the arrange of the blades or outside thereof as is shown in Fig. 2. The ratio of the free meridian entrance ring surface

$$\frac{\pi}{4}(d_2^2 - d_{e1}^2)$$

to the corresponding surface at the impeller exit, should be within the limits of 1.15 to 1.35. By "corresponding surface at the impeller exit" is meant the corresponding annular or ring surface at the exit side of the impeller which =

$$\frac{\pi}{4}(d_2 - d_{a1}^2)$$

The curvature of the blades is of importance. If thus the mean radius of curvature of the blades **5** is designated with R—by "profiling" is meant the radius of the mean skeleton line i.e. that line which is obtained if the profile of the profiled blades is halved at a plurality of portions and the thus found points are interconnected by a curve—the ratio of this radius of curvature to the outer blade diameter at the respective point should be less than about 1.1. This requirement should preferably be met along the entire length of the blade. The term mean radius has been employed because, when using a relatively thick sheet metal, the radii of curvatures of the inner and outer contour may differ somewhat from each other. It is necessary to find the curvature in an unequivocal manner. To this end, the central portions of the profile are connected by a curve. This curve will then have the so-called mean radius. The ratio of the respective blade chord to the respective diameter should preferably be within the limits of 0.3 to 0.75 whereas the blade chord should decrease from the outside *la* toward the inside *li*. This is all the more remarkable as with normal axial blowers this feature would reduce the degree of efficiency while simultaneously increasing the pressure. The twist of the blades as expressed by the difference in angle of the blade chord at the hub and at the outside (angle δ in Fig. 3) should be smaller than 25°, whereas the outer entrance angle of the blades β_{1e} measured with regard to the circumference should be less than 35°. The last mentioned feature which is known per se will in connection with the other above mentioned features bring about a further improvement of the degree of efficiency of the blower.

As to the blades, again numerous investigations and tests have been conducted in order to find the arrangement which will assure the most favorable degree of efficiency. It has been found that the most favorable degree of efficiency will be obtained with a blade number ranging from four to six.

The meridian accelerated axial blower according to the invention may be controlled by an arrangement according to Fig. 4. Impeller blades with adjustable pitch have been known heretofore, however, with these known arrangements, the blades were adjusted over their entire width with the result that the ends of the blades near the lower blade edge cause considerable disturbances in the flow when the edges extending into the flow, as well as the protruding edges of the holding bolt caused rather inconvenient disturbances in the flow. These drawbacks have been avoided according to the present invention by an arrangement which makes it possible to adjust the blades over a certain width only of said blades. Thus, for instance, each impeller blade may be sub-divided into two blade sections. In this instance either the blade section on the intake side is stationary and the other blade section is adjustable or vice versa. According to the embodiment shown in Fig. 4, the blade sections **5***a* are stationary at the intake side. The blade sections **5***b* which are located on the exit side are turnable about the holding bolt or pivot **14** which is rotatably journalled in the conical hub of the impeller **6**. The upper portion of Fig. 4 shows in full lines an impeller blade in normal position, whereas the lower portion of Fig. 4 shows an impeller blade in adjusted position indicated by broken lines. The adjusting mechanism itself does not form a part of the present invention and any standard adjusting mechanism known for adjusting the blades of an impeller of a blower may be employed in this connection. The control effect obtained by such a design is not much inferior to the control effect obtained when the blades are adjusted over their entire width. However, the drawbacks which as outlined above are inherent to the heretofore known arrangement with the adjustment of the blades over their entire width are not present with the control arrangement according to the invention.

It is, of course, understood that the present invention is, by no means, limited to the particular construction shown in the drawings but also comprises any modifications within the scope of the appended claims.

What I claim is:

1. In combination in an axial blower: a hollow at least approximately cylindrical member, and an impeller rotatably arranged within said cylindrical member and having a substantially cone-shaped hub provided with blades thereon and having a generating angle of from 10 to 20°, the ratio of the hub diameter at the rear end of the rear hub—d_{a1}—to the outer diameter of the blades—d_2—being less than 2:3 and greater than 1:2, and the ratio of the free annular meridian entrance surface

$$\frac{\pi}{4}(d_2^2 - d_{e1}^2)$$

to the free annular meridian exit surface

$$\frac{\pi}{4}(d_2^2 - d_{a1}^2)$$

being within the limits of from 1.15:1 to 1.35:1, d_{e1} representing the hub diameter at the front end of the hub.

2. An axial blower according to claim 1, in which the ratio of the radius of curvature are of each blade to the respective outer blade diameter pertaining thereto is less than 1.1:1.

3. In combination in an axial blower: a hollow at least approximately cylindrical member, and an impeller rotatably arranged within said cylindrical member and having a substantially cone-shaped hub provided with blades thereon and having a generating and γ of from 10 to 20°, the blades being so designed that the blade chords decrease from the outer diameter of the blades to the inner diameter thereof, the ratio of the rear hub diameter to the outer blade diameter being less than 2:3 and greater than 1:2, and the ratio of the free annular meridian entrance surface to the free annular meridian exit surface being within the limits of from 1.15:1 to 1.35:1.

4. In combination in an axial blower: a hollow at least approximately cylindrical member, and an impeller rotatably arranged within said cylindrical member and provided with blades thereon and having a generating angle of from 10 to 20°, the ratio of the respective blade chords to the respective blade diameters pertaining thereto being from 0.3:1 to 0.75:1, the ratio of the rear hub diameter to the outer blade diameter being less than 2:3 and greater than 1:2, and the ratio of the free annular meridian entrance surface to the free annular meridian exit surface being within the limits of from 1.15:1 to 1.35:1.

5. In combination in an axial blower: a hollow at least approximately cylindrical member, and an impeller rotatably arranged within said cylindrical member and provided with twisted blades thereon and having a generating angle of from 10 to 20°, the twist δ of said blades expressed by the difference in angle of the blade

chord at the hub and at the outer periphery of the respective blade being less than 25°, the ratio of the rear hub diameter to the outer blade diameter being less than 2:3 and greater than 1:2, and the ratio of the free annular meridian entrance surface to the free annular meridian exit surface being within the limits of from 1.15:1 to 1.35:1.

6. In combination in an axial blower: a hollow at least approximately cylindrical member, and an impeller rotatably arranged within said cylindrical member and having a substantially cone-shaped hub provided with blades thereon and having a generating angle of from 10 to 20°, the entrance outer angle β_{1e} of the blading being less than 35°, the ratio of the rear hub diameter to the outer blade diameter being less than 2:3 and greater than 1:2, and the ratio of the free annular meridian entrance surface to the free annular meridian exit surface being within the limits of from 1.15:1 to 1.35:1.

7. In combination in an axial blower: a hollow at least approximately cylindrical member, and an impeller rotatably arranged within said cylindrical member and having a substantially cone-shaped hub provided with from four to six blades thereon and having a generating angle of from 10 to 20°, the ratio of the rear hub diameter to the outer blade diameter being less than 2:3 and greater than 1:2, and the ratio of the free annular meridian entrance surface to the free annular meridian exit surface being within the limits of from 1.15:1 to 1.35:1.

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