April 8, 1969 B. ECK ET AL 3,437,262

CROSS-FLOW FLUID MACHINES

Original Filed Sept. 5, 1962 Sheet 1 of 2

FIG.1.

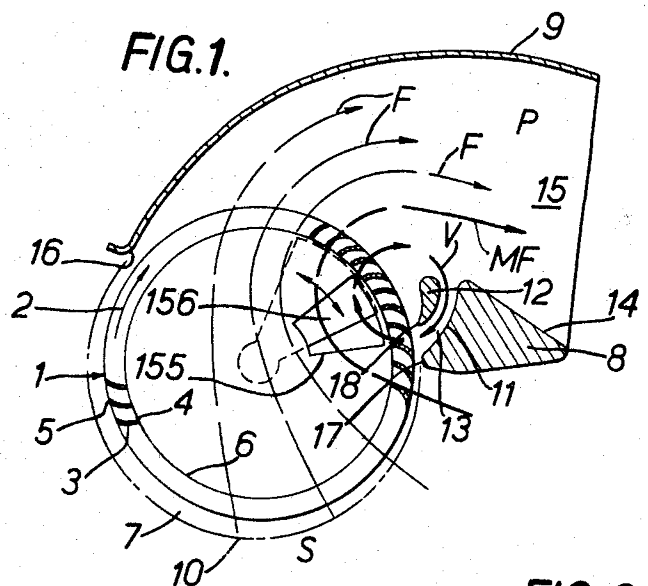

FIG. 2.

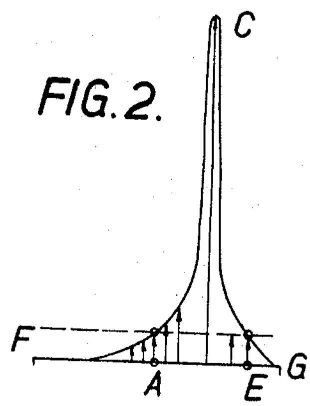

FIG. 3.

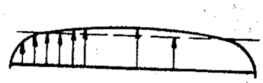

FIG.4.

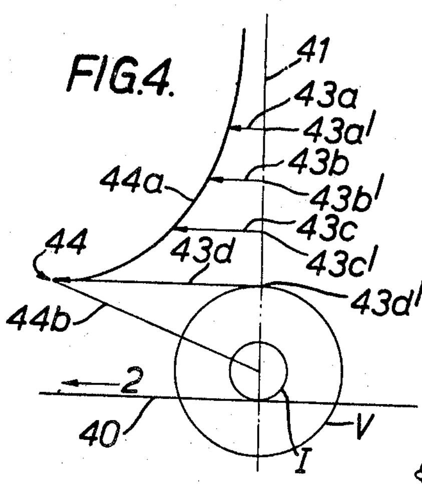

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FIG.5.

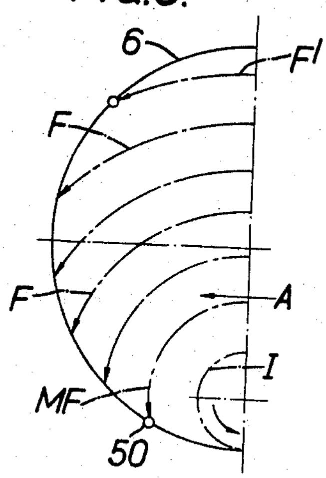


FIG.6.

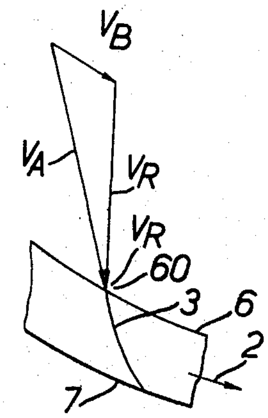
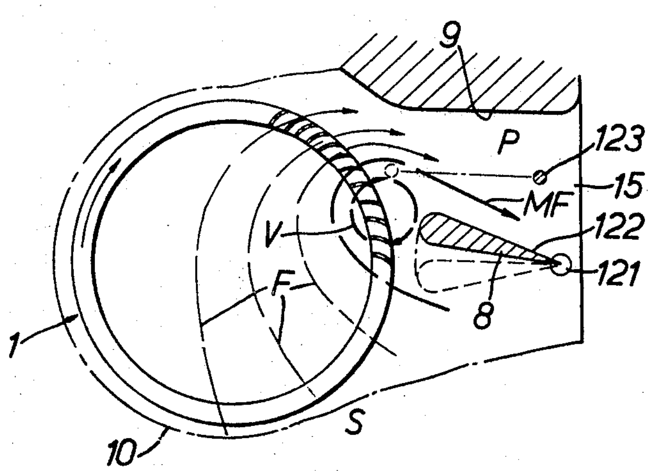


FIG.7.



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CROSS-FLOW FLUID MACHINES

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Original application May 5, 1965, Ser. No. 453,458, now Patent No. 3,249,292, dated May 3, 1966, which is a division of application Ser. No. 221,620, Sept. 5, 1962, now abandoned which in turn is a continuation-in-part of application Ser. No. 671,114, July 5, 1957, now abandoned. Divided and this application May 2, 1966, Ser. No. 546,811
Claims priority, application Germany, Dec. 7, 1956, L 26,388, L 26,389, E 13,334
Int. Cl. F04d 27/00, 17/08, 29/40
U.S. Cl. 230—114 6 Claims

ABSTRACT OF THE DISCLOSURE

A cross flow fluid machine having an auxiliary guide body in the discharge region of the machine to assist in positioning and stabilizing a fluid vortex within the machine.

This invention relates to cross-flow fluid machines for inducing movement of fluids which is to be understood as including both liquids and gases, and this application is a division of application Ser. No. 453,458 filed May 5, 1965, now Patent No. 3,249,292, itself a division of application 221,620 filed Sept. 5, 1962, now abandoned, which in turn is a continuation-in-part of application Ser. No. 671,114, filed July 5, 1957 and now abandoned. The invention relates more particularly to cross-flow machines of the type comprising a hollow cylindrical bladed rotor mounted for rotation about its axis and through which, in operation of the machine, fluid passes at least twice through the path of the rotating blades in a direction transverse to the axis of the rotor.

The invention concerns more especially fluid flow machines for operation under conditions of low Reynolds numbers. The Reynolds number of a particular fluid flow condition is a dimensionless number representing the ratio of the product of flow velocity and a characteristic linear dimension of the part under observation to the kinematic viscosity of the fluid. For the purpose of the present appilcation Reynolds number (Re) will be defined as

$$Re=d.c./\gamma$$

where d is the blade depth radially of the rotor, c is the peripheral speed of the rotor, and γ is the kinematic viscosity of the fluid, the latter being equal to the quotient of the dynamic viscosity and density. A Reynolds number is considered herein to be low if, as above defined, it is less than 5×10^4.

From the definition just given, it will be understood that the invention concerns more especially flow machines which are small dimensionally, run at low peripheral speeds, or are intended for use with air or other gas having a low density or used with a fluid having a high viscosity.

It is known that in a flow machine having bladed rotors, an initial acceleration and a subsequent deceleration of the flow occurs in boundary layers on the suction side of each blade as fluid passes over the blade. The higher the viscosity of the fluid in relation to its density or in relation to the relative velocity between the blade and fluid (i.e. the lower the Reynolds number) the greater is the deceleration of the boundary layer in the deceleration zone of the blade. If the boundary layer is slowed down sufficiently it separates from the blade and no longer follows the blade contour. The point at which separation occurs is known as the separation point. The separation point travels forward along the surface of the blade against the direction of flow in proportion to the increase in the effect of the viscosity relative to density or to the decrease in the relative velocity between the fluid and the blade.

The movement forward of the separation point along the blade because of low Reynolds number conditions produces a number of undesirable effects in the type of flow machine described. A vorticity zone in which the kinetic energy of the fluid is converted into thermal energy is produced after the separation point with the result that the efficiency of the machine drops. The degree of deflection of the fluid in passing through the path of the rotating blades decreases owing to the fact that the flow does not follow the full extent of the blade profile. This results in less pressure gain in the machine since pressure gain is determined by the extent of the deflection of the stream tubes in the blade channel. Finally, the turbulent flow in back of the separation point effectively reduces a part of the cross-section of the blade channels so that the throughput through the rotor of the machine also diminishes.

For the reasons given, it has previously been considered that the operation of flow machines under conditions of low Reynolds numbers would necessarily and inescapably involve low efficiencies in comparison with efficiences obtainable under conditions of high Reynolds numbers. For example, although the inefficiency of the small blowers above referred to has been notorious, it has been tolerated simply because it has not hitherto been thought capable of improvement.

It has hitherto been thought that to avoid mixing losses a flow machine should always be designed to have a rectangular velocity profile at every section taken across the flow, that is, the graph of velocity of fluid flow at a given point plotted across the flow channel should rise rapidly from zero at one side of the channel to a steady value maintained over the greater part of the section and should then drop again rapidly to zero at the other side. It has also been assumed hitherto that a flow machine of the type described should always have the blades loaded approximately equally by the fluid in the circumferential zones where the fluid passes through the rotor blades. These two related conditions can normally be satisfied without much difficulty.

Following the principles hitherto generally adhered to in the art and enunciated above, one skilled in the art would normally prefer to design a cross-flow type blower, such as that shown in Patent No. 2,742,733, so as to work under conditions of high Reynolds numbers and would design the blade angles and ducting on the basis of, and with a view to producing a rectangular velocity profile throughout the blower and an equal loading on the rotor blades in the circumferential zones where the fluid passes these rotor blades. On the other hand, if operation at low Reynolds numbers could not be avoided, the same design principles would normally be applied and the resulting lower efficiency regarded as inevitable.

An object of the present invention is to provide a cross-flow machine capable of operating under conditions of low Reynolds numbers with better efficiency than has hitherto been regarded as acceptable.

The invention depends in part on the appreciation that contrary to what as previously been thought by those skilled in the art, it can be advantageous under flow conditions of low Reynolds numbers to bring about in a flow machinery a velocity profile having a pronounced maximum with a consequent very unequal loading of the blades in the circumferential zones through which fluid passes. This velocity profile with a pronounced maximum gives rise to some flow tubes within the blower having much greater velocity than the other flow tubes within the blower.

In the restricted circumferential zones of the rotor blades through which the high velocity stream tubes pass, correspondingly high relative velocities exist locally between the fluid and the blades, so that in these zones momentum is imparted to the fluid at efficiencies which could otherwise be obtained only with machines operating under conditions of correspondingly much higher Reynolds numbers. The velocity profile with a pronounced maximum leads to lower velocities than the mean velocity in other circumferential zones of the rotor blades and in these zones transfer of momentum occurs at an efficiency which is lower than it would have been had the velocity profile been rectangular. However, the available momentum in a stream tube issuing from the blades increases with the square of its velocity; thus the momentum of the fluid as a whole is substantially concentrated in the high velocity stream tubes so that the transfer efficiencies in the zones of slow throughflow have little effect on the overall efficiency.

The invention depends in part also on the appreciation that the above-mentioned velocity profile with a pronounced maximum can be obtained by setting up in the machine a cylindrical vortex including a field region with a velocity profile approximately that of a Rankine vortex and a core region eccentric to the rotor axis. A further object of the invention therefore is to provide various means for forming and stabilizing the vortex.

Broadly, a cross-flow machine constructed according to the invention comprises a hollow cylindrically bladed rotor mounted for rotation about its axis and having its interior clear of stationary guides and with the blades of the rotor being curved with their outer edges leading their inner edges. End wall means substantially aligned with the ends of thet rotor may be provided, the ends of the rotor being substantially closed. First and second guide walls may also be provided extending between the end walls over the length of the rotor and defining therewith an entry region and a discharge region said rotor and guide walls co-operating on rotation of the rotor to set up and stabilize a fluide vortex having a core region extending lengthwise of the axis and eccentric thereto adjacent the first guide wall and guiding a fluid throughput through the rotor from the entry region through the path of the rotating blades of the rotor to the interior thereof and thence again through the path of the rotating blades to the discharge region.

The invention includes also a body in said discharge region extending parallel to the rotor axis between the end walls and spaced from the guide walls for flow of fluid to either side of said body. This body may affect the position or intensity of the vortex to produce a desired effect on the operation of the machine.

Referring to the drawings in which several embodiments of the invention are illustrated:

FIGURE 1 is a cross-sectional view of a fluid flow machine constructed according to the invention;

FIGURE 2 is a graph illustrating velocity of fluid flow at the outlet of a cross-flow fluid machine constructed according to the invention;

FIGURE 3 is a graph illustrating velocity of fluid flow at the outlet of conventional cross-flow machines;

FIGURE 4 is a graph illustrating velocity of fluid flow within the field of a Rankine type fluid vortex;

FIGURE 5 illustrates the ideal fluid flow lines occurring in one half the cross-sectional area of a rotor of a machine of the type shown in FIGURE 1;

FIGURE 6 is a vector diagram illustrating flow of fluid contacting a blade on its second transversal of the path of the rotating blades or when the fluid passes from the interior of the rotor to the pressure side of the machine; and

FIGURE 7 is a cross-sectional view of a machine somewhat similar to that shown in FIGURE 1.

Reference is made to the figures wherein like parts have like identifying numerals and, in particular to FIGURE 1 which illustrates a flow machine having a cylindrically bladed rotor **1** which is mounted, by means not shown, for rotation about its axis in the direction of the arrow **2**. The rotor **1** has thereon blades **3** extending longitudinally thereof and having inner and outer edges **4** and **5** lying on inner and outer blade envelopes **6** and **7** formed when the rotor is rotated. The blades **3** are concave facing the direction of rotation and have their outer edges leading their inner edges.

First and second guide walls **8**, **9** extend the length of the rotor **1** between end walls **10** aligned with the ends of the rotor. The end walls **10**, only one of which is shown, cover the ends of the machine and may, although not necessarily, close the ends of the rotor. The walls **8** and **9** define entry and exit regions S and P for flow of fluid to and from the rotor. The guide wall **8** has an arcuate wall portion **11** on the exit side of the rotor converging towards the rotor in the direction of rotation to form a converging gap or recess. A rounded auxiliary body **12** extending parallel to the rotor axis between the end walls defines a passage **13** with the wall **11**. The guide wall **8** has an outlet wall portion **14** defining with the guide wall **9** an exit duct **15**, which, since it diverges in the direction of flow, acts as a diffuser.

The wall **9** terminates at **16** which is spaced from the rotor a minimum of one-half the blade depth and not more than three times the blade depth in order to minimize interference which causes an undesirable noise when the machine is operated, while at the same time providing a means to guide the flow leaving the machine. The zone **14** defines one end of both the entry arc and the exit arc. From the zone **16**, the wall **9** diverges steadily from the rotor in the direction of rotation indicated by the arrow **2** with increasing radius of curvature; remote from the rotor the wall may be straight. The exit region P accordingly consists of a channel the median line of which is of spiral form.

It is preferred that not only the wall **9** but also the guide wall **8** should be substantially spaced from the rotor; the machine can then be made without adhering to close manufacturing tolerance, while still effectively separating the pressure and suction sides and maintaining the relatively high efficiency of the machine. A machine constructed according to the invention and having such spacing lends itself readily to sheet metal construction.

It will be noticed that the guide wall **8** subtends only a small angle at the rotor axis, and since it presents a rounded nose **17** to the rotor it provides negligible obstruction to air flowing in to or out from the rotor. The auxiliary guide body **12** also does not obstruct such flow, as it too presents only a nose **18** to the rotor.

In operation of the FIGURE 1 machine, a vortex having a core whose periphery is defined by the stream line V and approximately a Rankine vortex is produced wherein the core is positioned eccentrically to the rotor axis. The whole throughput of the machine flows twice through the blade envelope in a direction perpendicular to the rotor axis as indicated by the flow lines F, MF.

FIGURE 4 illustrates an ideal relation of the vortex to the rotor **1** and the distribution of flow velocity in the vortex core and in the field of the vortex. The line **40** represents a part of the inner envelope **6** of the rotor blades **3** projected onto a straight line while the line **41** represents a radius of the rotor taken through the axis of the vortex core. Velocity of fluid at points on the line **41** by reason of the vortex is indicated by the horizontal lines **43**a, **43**b, **43**c and **43**d, the length of these being the measure of the velocity at the points **43**a^1, **43**b^1, **43**c^1 and **43**d^1. The envelope of these lines is shown by the curve **44** which has two portions, portion **44**a being approximately a rectangular hyperbola and the other portion, **44**b, being a straight line. Curve **44**a relates to the field region of the vortex and the curve **44*b*** to the core. It will be understood that the curve shown in FIGURE 4 represents the velocity of fluid where an ideal or "mathematical" vortex is formed, and that in actual practice, flow conditions will only approximate these curves.

The core of the vortex is a whirling mass of fluid with no translational movement as a whole and the velocity diminishes from the periphery of the core to the axis **42**. The core of the vortex intersects the blade envelope as indicated at **40** and an isotach I within the vortex having the same velocity as the inner envelope contacts the envelope. The vortex core is a region of low pressure and the location of the core in a machine constructed according to the invention can be determined by measurement of the pressure distribution within the rotor.

The velocity profile of the fluid where it leaves the rotor and passes through the path of the rotating blades will be that of the vortex. In the ideal case of FIGURE 4, this profile will be that of the Rankine vortex there shown by curves **44*a*** and **44*b***, and in actual practice, the profile will still be substantially that shown in FIGURE 4 so that there will be around the periphery V of the core shown in FIGURE 1 a stream tube of high velocity whose centre line is the stream line MF, and the velocity profile taken at the exit arc **13** will be similar to that shown in FIGURE 2 where the line FG represents the exit arc **13** and the ordinates represent velocity. The curve shown exhibits a pronounced maximum point C which is much higher than the average velocity represented by the dotted line.

It will be appreciated that much the greater amount of fluid flows in the flow tubes in the region of maximum velocity. It has been found that approximately 80% of the performance is concentrated in the portion of the output represented by the line AE which is less than 30% of the total arc **13**. A conventional velocity profile for fluid flow in a defined passage is illustrated by the way of contrast in FIGURE 3 where the average velocity of flow is represented by the dotted line. Those skilled in the art regard this profile as being approximately a rectangular profile which following the principles generally adhered to is the sort of profile heretofore sought in the outlet of a flow machine.

The maximum velocity C shown in FIGURE 2 appertains to the maximum velocity stream tube. With a given construction the physical location of the flow tube MF may be closely defined. The relative velocity between the blades and fluid in the restricted zone of the rotor blades **3** through which the maximum velocity stream tube passes is much higher than it would be if a flow machine were designed following the conditions adhered to heretofore in the art respecting the desirability of a rectangular velocity profile at the exit arc and even loading of the blades.

Under low Reynolds number conditions, this unevenness of the velocity profile leads to beneficial results in that there will be less separation and energy loss in the restricted zone of the rotor blades through which the high velocity stream tubes pass than if these stream tubes had the average velocity of throughput taken over the whole exit arc **13** of the rotor. There is a more efficient transfer of momentum to the fluid by the blades in this restricted zone and while the transfer of momentum in the flow tubes travelling below the average velocity will be less efficient, nevertheless when all of the flow tubes are considered, there in a substantial gain in efficiency.

FIGURE 5 illustrates ideally a number of stream lines F characterizing stream tubes occurring within one half the rotor area defining by the inner envelope **6**, it being understood that the stream tubes in the other half of the rotor are similar. The centre line MF of the stream tubes of highest velocity is shown intersecting the envelope **6** at point **50** and the isotach I as being circular when the whole rotor is considered. It is seen that ideally the stream tubes of highest velocity undergo a change of direction of substantially 180° from the suction to the pressure sides when

the flow in the whole rotor is considered. It is also to be noted that the major part of the throughput, contained in these stream tubes, passes through the rotor blades where the blades have a component of velocity in direction opposite to the main direction of flow within the rotor indicated by the arrow A.

FIGURE 6 is a diagram showing the relative velocities of flow with respect to a blade at the point **50** referred to in FIGURE 5. In this figure V_B represents the velocity of the inner edge of the blade **3** at the point **50**, V_A the absolute velocity of the air in the flow tube MF at the point **50**, and V_R the velocity of that air relative to the blade as determined by completing the triangle. The direction of the vector V_R coincides with that of the blade at its inner edge so that fluid flows by the blade substantially without shock.

The character of a vortex is considered as being determined largely by the blade angles and curvatures. The position of the vortex, on the other hand, is considered as being largely determined by the configuration of the vortex forming means which forms and stabilizes a vortex in co-operation with the bladed rotor. The particular angles and curvatures in any given case depend upon the following parameters: the diameter of the rotor, the depth of a blade in a radial direction, the density and viscosity of the fluid, the disposition of the vortex forming means and the rotational speed of the rotor, as well as the ratio between overall pressure and back pressure. These parameters must be adapted to correspond to the operating conditions in a given situation. Whether or not the angle and shape of the blades have been fixed at optimum values is to be judged by the criterion that the stream tubes close to the vortex core are to be deflected approximately 180°.

It is to be appreciated that the flow lines of FIGURE 1 do not correspond exactly to the position of the vortex core as illustrated in FIGURES 4 and 5 which represent the theoretical or mathematical flow. These latter figures show that it is desirable to have the axis of the core of the vortex within the inner blade envelope **6** so that the isotach within the core is tangent to that envelope. Although this position is achieved in certain constructions hereinafter described, it is not essential, and in fact, is not achieved in the structure shown in FIGURE 1.

It is further to be appreciated that despite the divergence of the flow in FIGURE 1 from the ideal, the stream tubes of highest velocity which carry a major part of the throughput are nevertheless turned through an angle of substantially 180° in passing from the suction to the pressure side of the rotor and that these stream tubes pass through the rotor blades where the blades have a velocity with a component opposite to the main direction of flow through the rotor as indicated by the arrow A.

It will be seen that peripheral flow tubes of the vortex core region V flow through the passage **13** between the arcuate wall **11** defining the aforementioned recess and the auxiliary guide body **12**.

The end wall **10** closing one end of the rotor has an aperture **155** in the region where the vortex core is formed. The aperture **155** has the shape of a truncated sector which is adapted to be closed by a sector shaped cover plate **156** pivoted about an axis coinciding with the rotor axis. When the aperture **155** is completely opened, fluid enters through it from the exterior of the machine and spoils the vortex thus reducing the throughput of the machine.

The machine shown in FIGURE 7 is similar in many respects to that of FIGURE 1: the same references will be used for similar parts, which will not require further description. In this machine a first guide wall **8** is shaped as a thin sector having its larger end rounded and having its other end pivoted for movement about the axis **121**. The vortex core is formed in the same manner as in FIGURE 1. When the wall **8** takes the position shown in full line, the core takes the position shown. When the wall **8** is pivoted to the dotted position, the core is moved downwardly, thus changing the throughput of the machine.

The arrangement shown in FIGURE 7 also provides a means whereby a diffuser may be formed in one position of the wall **8** and where there will be no diffuser effect when this wall is in another position. Thus the wall **8** in the position shown in full lines diverges from a second guide wall **9** in the direction of flow so that a diffuser is formed. The wall **8** in the position shown in dotted lines, is substantially parallel to guide wall **9** so there is no diffuser effect at the outlet.

The throughput of a machine constructed according to the invention may be further regulated, by means of a rod **123** which extends parallel to the rotor along its length and is movable between the position shown in full lines and that shown in dotted.

We claim:

1. A cross flow fluid machine of the type comprising a hollow bladed cylindrical rotor mounted for rotation about its axis, a pair of end walls substantially aligned with the ends of the rotor, the ends of the rotor being substantially closed, first and second guide walls extending between the end walls over the length of the rotor and defining therewith an entry region and a discharge region said rotor and guide walls co-operating on rotation of the rotor to set up and stabilize a fluid vortex having a core region extending lengthwise of the axis and eccentric thereto adjacent the first guide wall and guiding a fluid throughput through the rotor from the entry region through the path of the rotating blades of the rotor to the interior thereof and thence again through the path of the rotating blades to the discharge region, and a body in said discharge region extending parallel to the rotor axis between the end walls and spaced from the guide walls for flow of fluid to either side of said body.

2. A fluid machine as claimed in claim **1**, wherein the body is of rounded form.

3. A fluid machine as claimed in claim **1**, wherein the body lies close to the first guide wall and defines therewith a passage for flow of peripheral flow tubes of said vortex core region.

4. A fluid machine as claimed in claim **1**, wherein the body is positioned within the discharge region between the periphery of the vortex core region and the adjacent flow tubes of the throughput.

5. A fluid machine of the type comprising a hollow bladed cylindrical rotor mounted for rotation about its axis, a pair of end walls substantially aligned with the ends of the rotor, the ends of the rotor being substantially closed, first and second guide walls extending between the end walls over the length of the rotor and defining therewith an entry region and a discharge region said rotor and guide walls co-operating on rotation of

the rotor set up and stabilize a fluid vortex having a core region extending lengthwise of the axis and eccentric thereto adjacent the first guide wall and guiding a fluid throughput through the rotor from the entry region through the path of the rotating blades of the rotor to the interior thereof and thence again through the path of the rotating blades to the discharge region, the first guide wall subtending a small angle at the rotor axis and having adjacent the rotor an arcuate portion defining a recess, and a rounded auxiliary guide body in said discharge region which is adjacent the rotor spaced from the guide walls and extends parallel to the axis between the end walls, the guide body inducing flow tubes of the vortex core region to circulate between said body and said recess with flow tubes of the throughput passing between the body and the second guide wall.

6. A fluid machine of the type comprising a hollow bladed cylindrical rotor mounted for rotation about its axis, a pair of end walls substantially aligned with the ends of the rotor, the ends of the rotor being substantially closed, first and second guide walls extending between the end walls over the length of the rotor and defining therewith an entry region and a discharge region said rotor and guide walls co-operating on rotation of the rotor to set up and stabilize a fluid vortex having a core region extending lengthwise of the axis and eccentric thereto adjacent the first guide wall and guiding a fluid throughput through the rotor from the entry region through the path of the rotating blades of the rotor to the interior thereof and thence again through the path of the rotating blades to the discharge region, the first guide wall presenting an edge to the rotor in spaced relation thereto, and a rod-like body in said discharge region extending parallel to the rotor axis between the end walls and spaced between the guide walls in a position adjacent and external to the vortex core region with flow tubes of said core region circulating between the body and the first guide wall and the throughput flow tubes passing between the body and said second guide wall.

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