

- [54] ENERGY TRANSFER MACHINE
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- [73] Assignee: **The Garrett Corporation**, Los Angeles, Calif.
- [22] Filed: **Aug. 9, 1971**
- [21] Appl. No.: **169,997**
- [52] U.S. Cl. **415/53 T, 415/83**
- [51] Int. Cl. **F04d 5/00**
- [58] Field of Search **415/52, 53, 53 T, 415/83, 56, 57, 58**

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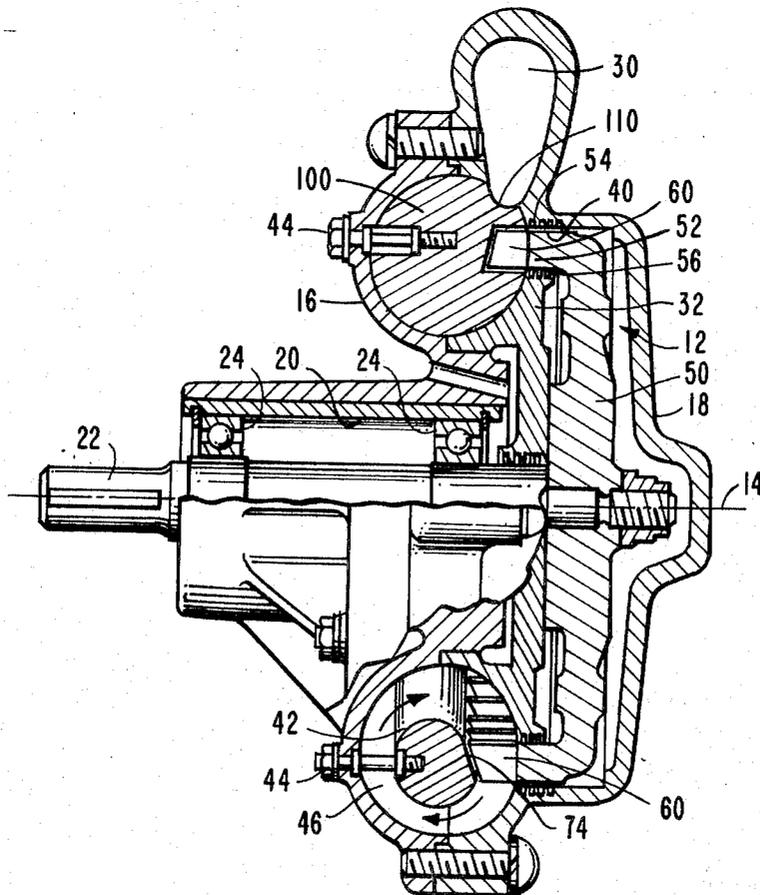
Primary Examiner—C. J. Husar
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[57] **ABSTRACT**

An energy transfer machine is disclosed including a casing enclosing a stator ring, the casing and ring conjointly defining a smooth wall, annular fluid passage extending about the axis of rotation between inlet and outlet passages formed in the casing. A rotor within the casing includes a blade cascade projecting in cantilever fashion into the annular fluid passage, each blade having leading and trailing edges extending generally parallel with the axis of rotation of the machine, the edge nearer the axis of rotation being longer than the edge further from the axis. The geometries and relative positions of the blades and annular fluid passage are such that the radial distance from the axis of rotation of the centroid of the meridional cross-section of the annular fluid passage is larger than the radial distance of at least the major portion of the edge nearer the rotational axis and smaller than the radial distance from the axis of rotation of at least the major portion of the edge further from the rotational axis.

Machines according to the invention are also characterized by constructional features which minimize the number of castings required and thereby greatly simplify the fabrication and structure to minimize cost.

19 Claims, 19 Drawing Figures



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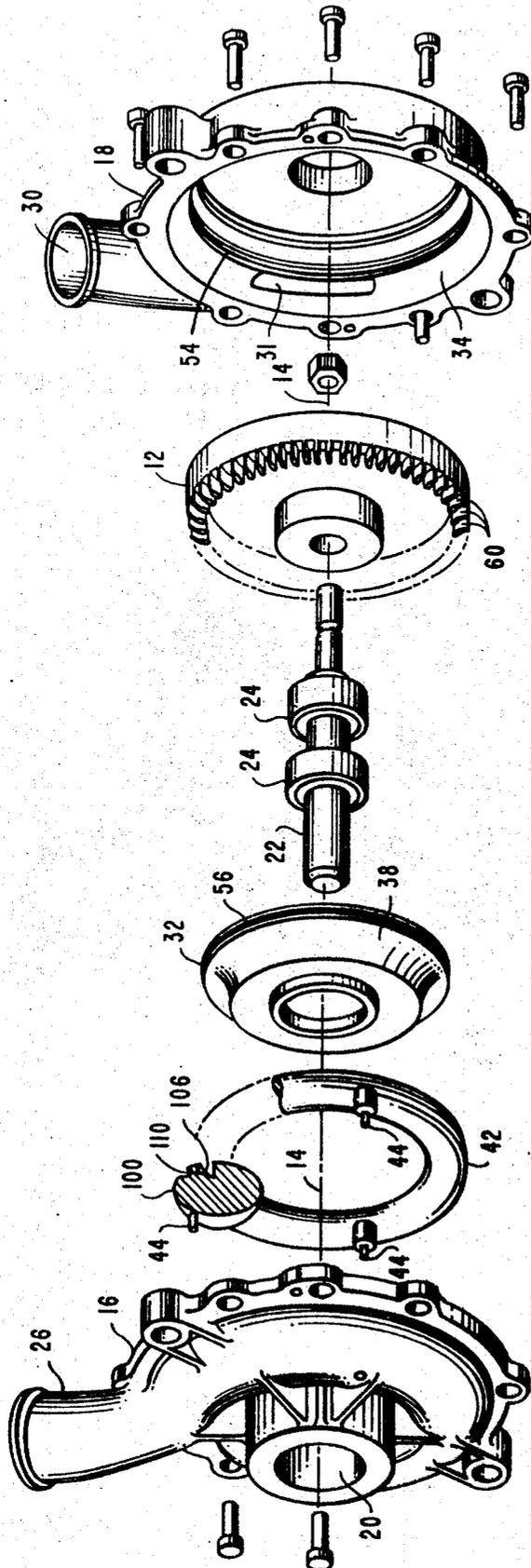


FIG.-1

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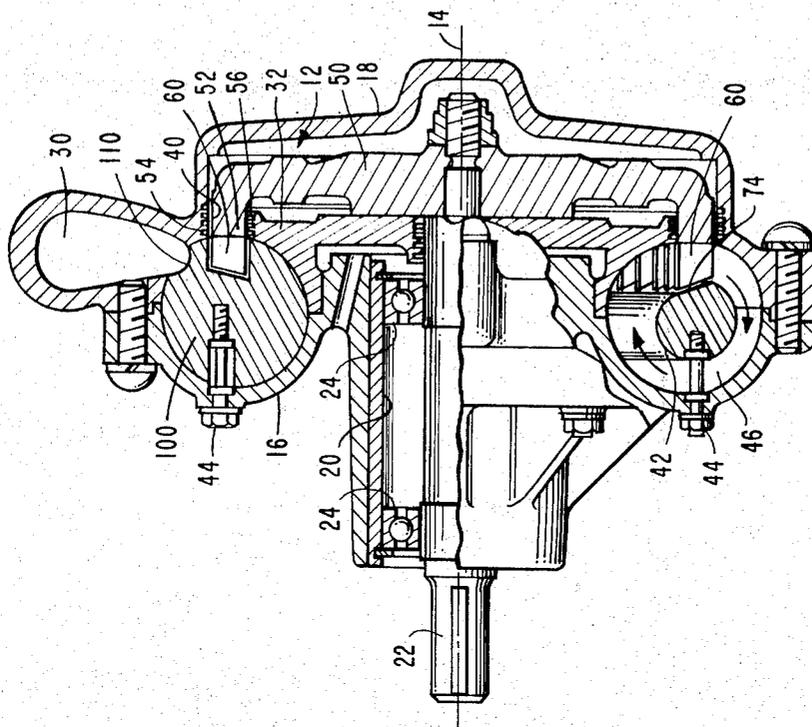


FIG. -3

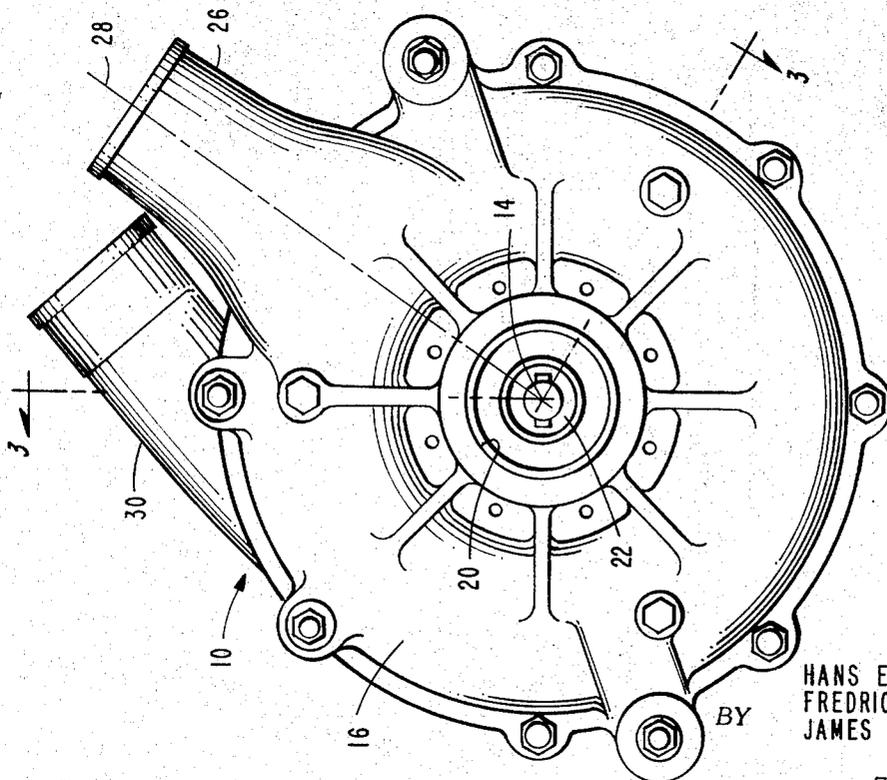


FIG. -2

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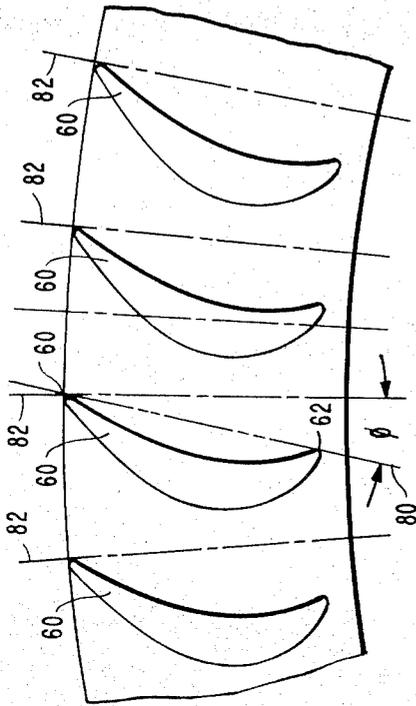


FIG. - 5

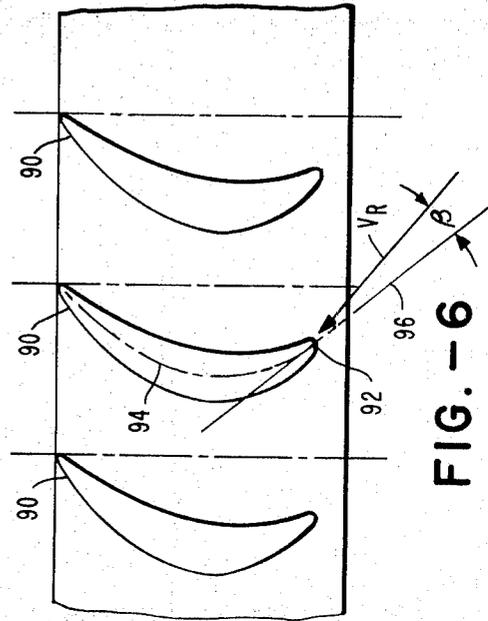


FIG. - 6

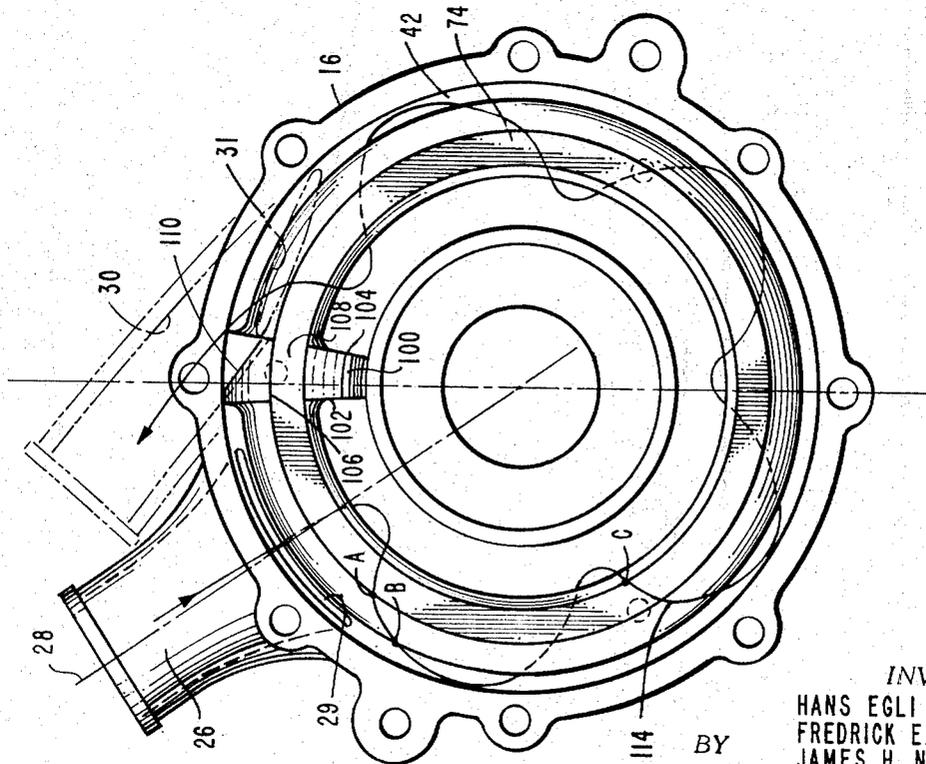


FIG. - 4

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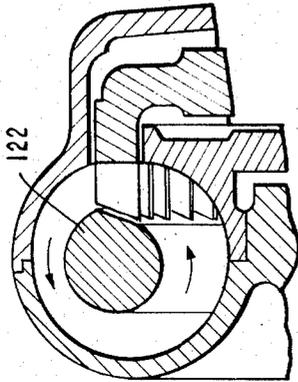


FIG. - 10

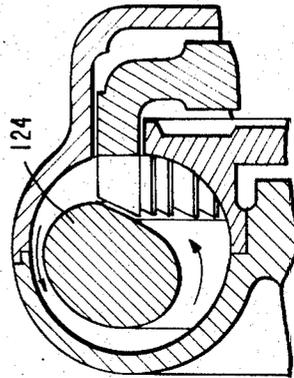


FIG. - 11

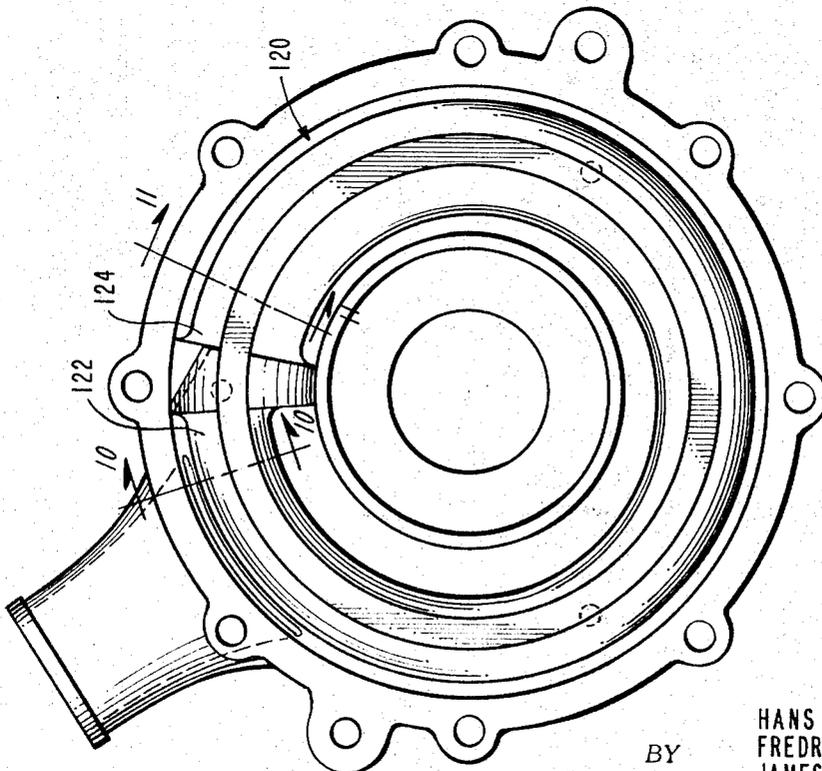


FIG. - 9

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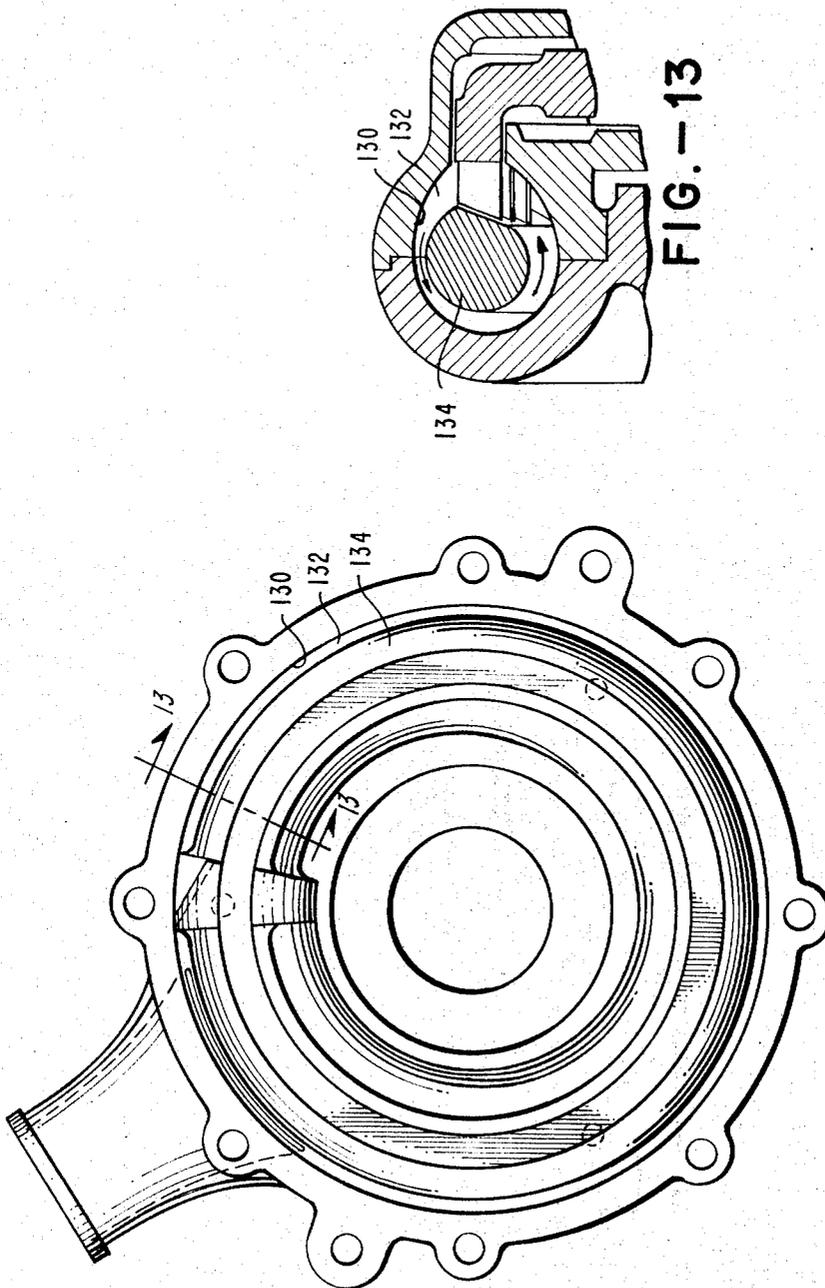


FIG.-13

FIG.-12

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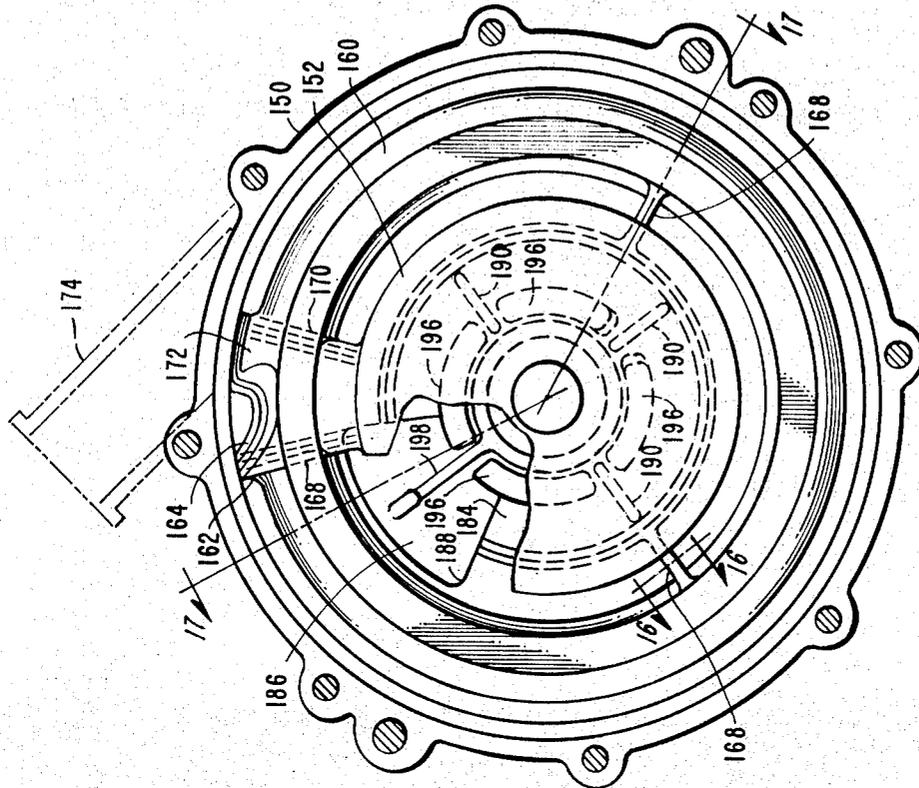


FIG. - 15

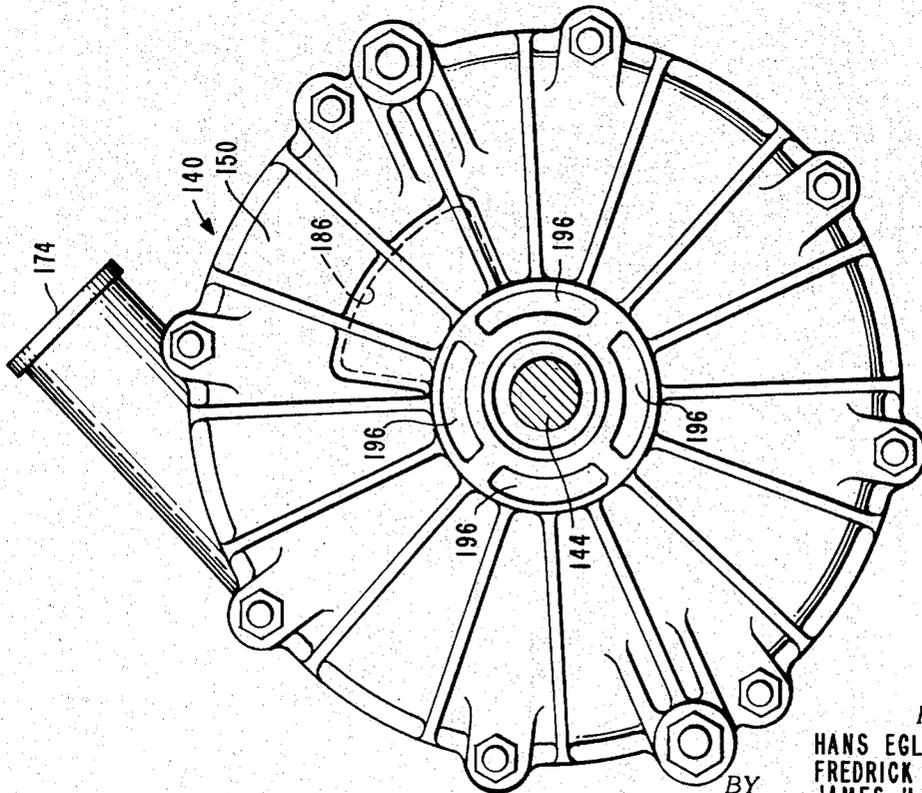


FIG. - 14

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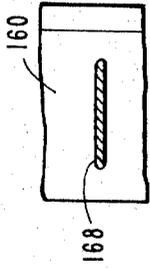


FIG. -16

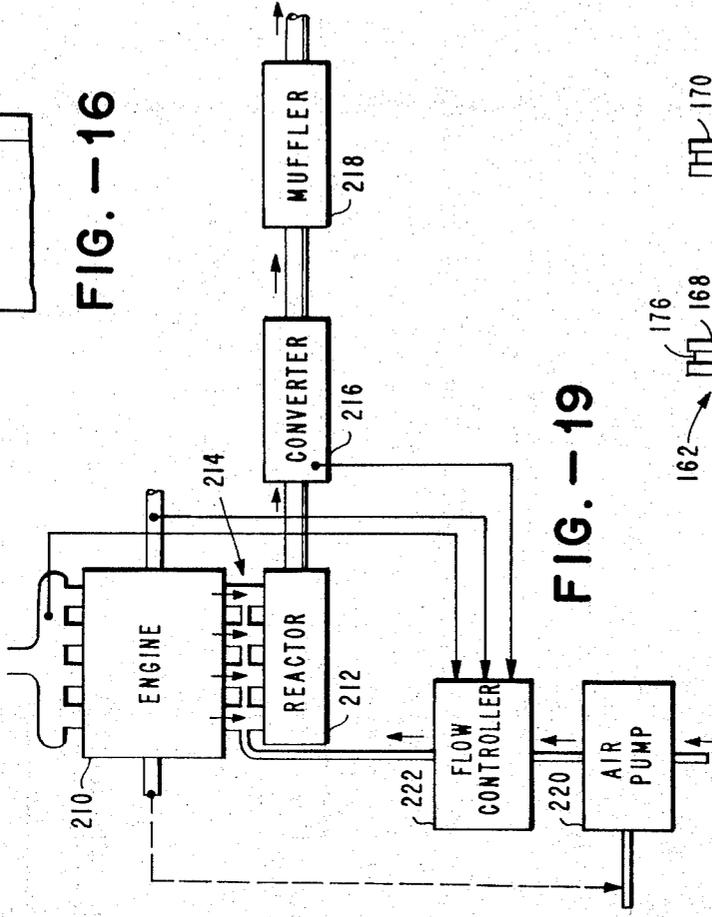


FIG. -19

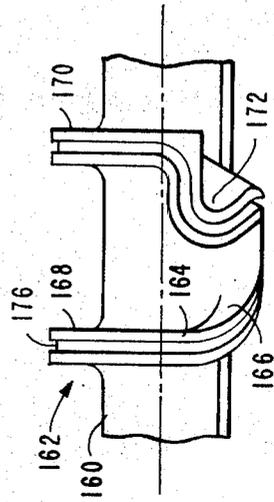


FIG. -18

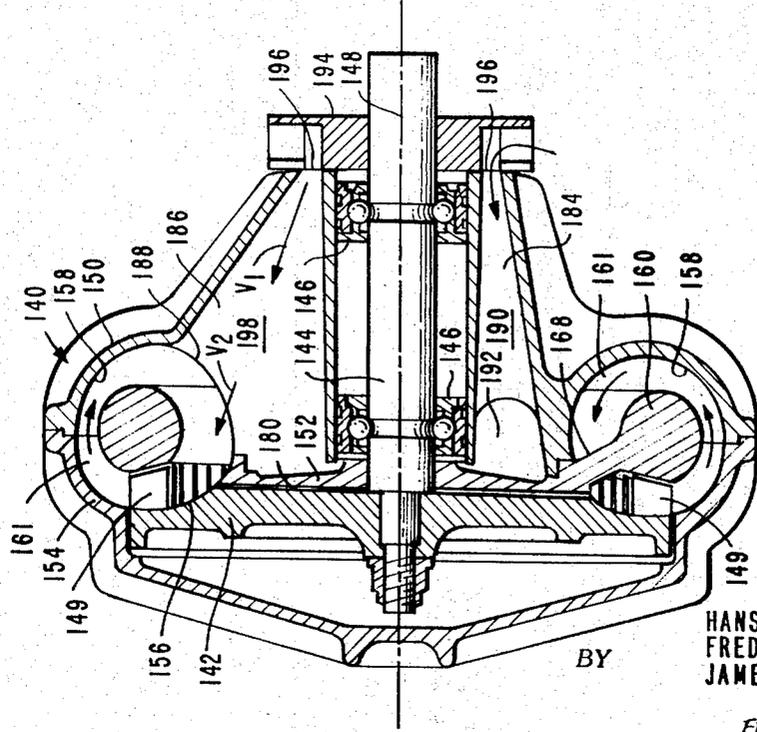


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ENERGY TRANSFER MACHINE

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates generally to turbomachines and more particularly to turbomachines in which the fluid is constrained to pass at least twice through the blades of a single rotor.

2. Description of the Prior Art

The prior art includes reentry machines utilizing a single rotor to gain multi-stage performance. These machines are designed to deliver a high specific work output, that is, a high work output per unit weight of working fluid. After there has been a partial transfer of energy between the working fluid and the rotor, the fluid is returned to the rotor blades via a return duct or stator vanes for readmission to the blades and another energy transfer process takes place. This mode of operation can be continued so that multi-stage performance is approached with only one rotor. However, reentry machines have a number of disadvantages including the presence of interstage leakage and the restriction of the flow to a specific, single path of the return ducts or stator guide vanes for all load conditions which permanently fixes the number of passes made by the fluid through the blades.

In other apparatus of the prior art, the fluid moves in and out of the rotor blade passages in a generally uncontrolled and disorderly fashion due to lack of stator guidance. Because of the disorderliness of the flow, these machines are very inefficient and the net fluid flow through the machine ceases at a relatively low limiting value of back pressure.

U.S. Pat. No. 3,292,899, issued Dec. 20, 1966 to Hans Egli, one of the inventors of the present invention, discloses a multi-pass energy transfer machine representing a major improvement over the described prior art. Machines according to the referenced patent have specific diameter and specific speed characteristics falling between that of conventional turbomachines and conventional rotary, positive displacement machines and overlapping considerably into the latter category. A further advantage of machines of the cited patent is that they will produce, for a given tip speed, a much higher head than conventional turbomachines, the term "head" being defined as the energy transfer per pound of fluid.

The fluid energy transfer machine according to one embodiment of the referenced patent includes a casing having inlet and outlet openings and a plurality of blades arranged in cascade fashion on a rotor for movement in succession past the inlet and outlet openings. The housing has a smooth interior wall devoid of stator vanes and is so configured that the fluid in passing from the inlet opening to the outlet opening is constrained by the walls of the casing to flow in a generally helicoidal pattern about a stator ring and through the blade cascade a number of times. The geometry of the fluid path is primarily dictated by the shapes of the casing and stator ring, the manner in which the fluid is introduced into the housing (as determined by the shape and orientation of the inlet passage leading to the inlet opening) and the circumferential pressure gradient which, in turn, depends on the back pressure applied to the machine. Thus, the pitch of the helicoidal flow path at any specific meridional section of the machine and the total number of times that the fluid passes through the blade

cascade are functions of back pressure. In order to obtain a practical level of energy transfer, the fluid should pass at least twice through the cascade. In the case of a compressor, for example, for any fluid (whether compressible or incompressible), if the back pressure is increased while a given rotor speed is maintained, then the number of passes of the fluid through the blade cascade increases. The fluid flow pattern is orderly and remains so even though the back pressure is increased to relatively high levels.

In the prior art machines discussed above, comparatively little attention is paid, from a fluid dynamics standpoint, to the shaping of the rotor blades and their position in the flow passage. This is also true of the block seal separating the inlet and outlet openings. Thus, the overall efficiency of the machine is not optimized. The influence on the performance of the machine of the geometrical configurations and positions of the blades and block seal has not been adequately appreciated in the prior art.

SUMMARY OF THE INVENTION

Broadly, the present invention comprises an improvement of the turbomachine disclosed in U.S. Pat. No. 3,292,899, referenced above, and among its primary objects are the enhancement of the performance and the structural simplification of that type of machine.

According to one specific, exemplary form of the invention, there is provided a casing defining an interior, smooth-walled toroidal space positioned generally concentric of a central axis of rotation and having substantially circular meridional cross-sections. Inlet and outlet passages formed integrally with the casing communicate with the interior of the casing through inlet and outlet opening situated in close proximity to one another circumferentially.

The casing encloses a stator ring of generally circular meridional cross-section. The ring is positioned concentric of the rotational axis and is spaced from the interior wall surface of the casing to define a generally annular fluid passage extending circumferentially about the axis of rotation. The stator ring is disposed relative to the casing such that the meridional cross-section of the ring is eccentric with respect to the meridional cross-section of the toroidal space defined by the casing, the ring being situated somewhat closer to the outer extremity of the toroidal space with respect to the axis of rotation. According to one embodiment, the stator ring is supported at a plurality of circumferentially spaced points by axially extending studs attached to the casing.

The inlet and outlet openings are separated by a block seal formed integral with the stator ring and having side surfaces coinciding with meridional planes.

The block seal is recessed to form a notch blending smoothly with the outlet passage to minimize losses and disturbances and to permit passage of the blade cascade.

The inlet passage is shaped and positioned to direct the fluid into the annular fluid passage so that a helicoidal motion is initiated and the outlet is configured to receive the discharging fluid with minimal losses. In the exemplary form of the invention under consideration, the inlet passage is positioned to direct the fluid generally along meridional planes so that at the predominate operating condition, the absolute velocity vectors of

the fluid approach the blades at approximately 90° with respect to the direction of blade rotational velocity. The exhaust passage is oriented approximately in alignment with the absolute velocity vector of the exiting flow.

The casing also encloses a rotor having a blade cascade in close running clearance with the stator ring and extending in cantilever fashion from the rotor approximately parallel with the axis of rotation. The blades are cambered, aerodynamic sections with chord lines oriented at a predetermined stagger angle. The fluid flows in the above-mentioned generally helicoidal path about the stator ring and is impelled (in the case of a pump, blower or compressor) during each pass of the fluid through the blade cascade. In this fashion, the inlet and outlet passages introduce and receive the fluid from opposite sides of the cascade.

Each blade has a leading edge and a trailing edge extending generally parallel with the axis of rotation of the machine. According to an aspect of the invention, the edge nearer the axis of rotation is longer than the edge further from the axis of rotation. Thus, in the case of an "outflow" machine in which the fluid flows away from the axis of rotation during its passage through the blades, the leading edge of the blade is longer than the trailing edge. The reverse is true for an "inflow" machine.

According to another significant aspect of the invention, the geometries and relative positions of the blades and annular fluid passage are such that the radial distance from the axis of rotation of the centroid of the meridional cross-section of the annular passage is larger than the radial distance from the axis of at least the major portion of the blade edge closer to the axis and smaller than the radial distance from the axis of at least the major portion of the edge further from the axis. This geometry and relative positioning provides a way to assure that the angle of incidence, that is, the angle at which the relative velocity vector of the fluid approaches the blade cascade, will remain nearly constant over a wide operating range.

According to another aspect of the invention, in order to more closely satisfy continuity considerations in the case of compressible fluids, the meridional cross-sectional area of the annular fluid passage may be designed to decrease (in the case of a compressor or blower) toward the point of discharge. This may be accomplished by utilizing a stator ring having a meridional cross-sectional area which increases from inlet to outlet, or alternatively, a casing having an interior wall whose meridional cross-sectional area decreases between inlet and outlet, or a combination of such geometries.

The invention provides a broad operating range over which high efficiencies are obtained, and the relatively low operating speed of machines according to the invention permits the use, for many applications, of practical drive systems in contrast to conventional types of turbomachines which require rotational speeds too high for such drives.

By way of example, the machine of the invention may be designed as an air pump for use in the emission control systems of automobiles. Such pumps introduce pressurized, secondary air into the exhaust stream to provide additional oxygen to insure complete combustion of the exhaust products. Apparatus according to the invention meets the cost, minimum bulk, simplicity,

and functional and operational air requirements of such systems.

BRIEF DESCRIPTION OF THE DRAWINGS

Other objects, advantages and features of the invention will become apparent by reference to the following detailed description in conjunction with the accompanying drawings, in which:

FIG. 1 is an exploded, perspective view of an energy transfer machine, designed for operation as an air pump, in accordance with the invention;

FIG. 2 is a front elevation view of the machine of FIG. 1;

FIG. 3 is a side elevation view, partly in section, of the machine of FIG. 1, the section being taken along the broken plane 3—3 in FIG. 2;

FIG. 4 is a rear view of the machine of FIG. 1 with certain portions of the casing omitted for clarity and showing the path of a typical fluid particle;

FIG. 5 is a front view of a portion of the rotor of the machine of FIG. 1 showing details of the blade cascade;

FIG. 6 is a representation of a straight blade cascade showing certain, pertinent geometric interrelationships;

FIG. 7 is a somewhat schematic, meridional cross-section view of a portion of a machine according to the invention to illustrate certain geometric relationships for a first, exemplary blade configuration;

FIG. 8 is a meridional cross-section view similar to that of FIG. 7 showing the same geometric relationships for a second, exemplary blade configuration;

FIG. 9 is a rear view of an alternative embodiment of the energy transfer machine of the invention, designed for use as an air pump, with certain portions of the machine omitted for clarity;

FIG. 10 is a cross-section view of a portion of the apparatus of FIG. 9 as seen along the plane 10—10;

FIG. 11 is a cross-section view of a portion of the apparatus of FIG. 9 as seen along the plane 11—11;

FIG. 12 is a rear view of another alternative embodiment of the energy transfer machine of the invention, designed for operation as an air pump, with certain portions of the machine omitted for clarity;

FIG. 13 is a cross-section view of a portion of the apparatus of FIG. 1 as seen along the plane 13—13;

FIG. 14 is a front elevation view of an energy transfer machine according to still another embodiment of the invention;

FIG. 15 is a rear elevation view of the machine of FIG. 14 with certain portions thereof omitted for clarity;

FIG. 16 is a cross-section view of a portion of the machine of FIG. 14 taken along the plane 16—16 in FIG. 15;

FIG. 17 is a side elevation view in cross-section of the machine of FIG. 14 taken along the broken plane 17—17 in FIG. 15;

FIG. 18 is a top view of a portion of the machine of FIG. 14 showing details of the block seal; and,

FIG. 19 is a block diagram of an internal combustion engine and associated emission control system having a secondary air pump constructed pursuant to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Although it will be apparent to those skilled in the art

that machines according to the present invention can be constructed for operation as pumps, blowers, compressors, turbines, and so forth, utilizing either gas or liquid as the working fluid medium, for purposes of illustration the drawings and ensuing description are directed to an embodiment of the invention which has application as an air pump and is particularly useful as a source of pressurized secondary air in automobile emission control system.

Turning now to FIGS. 1-5 of the drawings, there is shown a machine exemplifying the present invention and including generally a casing 10 and a rotor 12 both concentric of a central axis of rotation 14.

The casing 10 is composite, split structure consisting basically of a front casing portion 16 and a rear casing portion 18 joined along an interface extending generally transverse of the axis 14. The front casing portion 16 defines an axially extending cylindrical bore 20 enclosing a shaft 22 supported by a pair of ball bearings 24. The rotor 12 is attached to the rear extremity of the shaft 22 for rotation about the axis 14. The front casing portion 16 also includes a radially oriented inlet passage 26 shaped and positioned symmetrically about a meridional plane 28. The inlet passage 26 communicates with the casing interior through an inlet opening 29. The rear casing portion 18 includes an outlet passage 30 and associated outlet opening 31. The passage 30 is oriented so as to be in approximate alignment with the absolute velocity vectors of the fluid approaching the outlet opening 31.

The casing 10 also encloses an annular, intermediate casing portion 32 concentric of the axis 14 and positioned between the front and rear casing portions 16 and 18. The casing portions 16, 18 and 32 have smooth, arcuate interior surfaces 34, 36 and 38, respectively. Except for an annular opening 40 between the rear portion 18 and the intermediate portion 32, the surfaces 34, 36 and 38 are contiguous and together define a smooth-walled, toroidal volume or space whose axis of revolution is the axis 14. In meridional cross-section, as shown in FIG. 3 for example, the toroidal space has a generally circular appearance, the exact geometry depending upon many parameters including flow rate, rotor speed, pressure ratio, type of fluid, and so forth. For particular applications, the shape of the toroidal space, in meridional cross-section, may be varied as required, for example, it may be elongated in a given direction such as in the direction of the axis of rotation.

Disposed inside the toroidal space defined by the casing 10 is a stator ring 42, which, in the example shown, has a generally circular meridional cross-section. The stator ring 42 is fastened to the front casing portion 16 by three studs 44 and is spaced from the interior wall surfaces 34, 36 and 38 so as to define conjointly with these surfaces an annular fluid passage 46 having a meridional cross-section that is uniform about the axis 14. The ring 42, as best seen in FIGS. 3 and 4, is positioned within the toroidal space so as to be somewhat closer to the outer extremities of that space. Like the toroidal space in which it is enclosed, the configuration of the meridional cross-section of the ring 42 may be varied from that shown in the drawings as required for particular applications.

The rotor 12 comprises a disk 50 transverse of the axis 14 and having along its outer periphery a forwardly projecting rim 52 extending parallel with the axis 14

into the annular opening 40. Outer and inner rim seals 54 and 56, respectively, on the rear and intermediate casing portions 18 and 32 preserve the circumferential pressure gradient within the casing during operation of the machine. Attached to forward extremity of the rim 52 in cantilever fashion is a cascade of spaced blades 60. As shown in FIGS. 3 and also with reference to FIG. 7, each blade 60 has an inner edge 62, an outer edge 64, a base 66 and a tip 68. In the particular example shown, the edges 62 and 64 are approximately parallel with the axis 14 but it is to be noted that the blades are not limited to that specific configuration. The edges 62 and 64 may to some extent be out of parallelism with respect to each other and the axis 14 to achieve specific performance characteristics and/or facilitate manufacture of the rotor. With respect to the latter in particular, the orientation of the blade edges approximately parallel with the axis of rotation makes possible the use of a simple, two-part die for casting the rotor as a single piece. To facilitate separation of such a die, the blade edges may be provided with "draft," that is, the edges may converge toward the tip at a small angle.

In the specific example shown, the stator ring 42 has a machined, conical face 74 positioned immediately adjacent the blade tips 68. The tips 68 have a linear configuration and slope relative to the axis 14 so as to be parallel with the conical face 74 as viewed in any meridional cross-section (see FIGS. 3, 7 and 8 in particular). Thus, a constant, minimum running clearance is maintained along the entire circumference of the ring 42 about the axis 14 between the face 74 and the blade tips 68. The surface 74 may be other than conical so long as it conforms to the contour of the blade tips and is separated therefrom by a constant, minimum running clearance.

As shown in FIG. 5, each blade 60 has a cambered, airfoil configuration and is set at a stagger angle ϕ defined by a typical blade chord line 80 and a meridional plane 82 tangent to the trailing edge 64 of the blade. According to one specific example, the stagger angle ϕ is approximately 12° and there are a total of 72 blades.

From the standpoint of optimizing the performance of the machine, it is desirable to maintain a constant angle of incidence over a broad operating range. For purposes of the present invention, the definition of "angle of incidence" may be understood by reference to FIG. 6 which shows a straight cascade of blades 90. The representation in FIG. 6 may be thought of as a portion of the blade cascade on a rotor of infinite radius with the plane of the drawing coinciding with a plane normal to the axis of rotation. Each blade 90 has a leading edge 92 and a camber line or mean line 94 about which the blade surfaces are constructed. The angle of incidence, β , is defined as the angle between a line 96 drawn tangent to the mean line 94 at the leading edge 92 and the vector V_R which is the component of the relative velocity of the fluid approaching the cascade in a plane normal to the axis of rotation.

With reference now to FIGS. 7 and 8, the maintenance of a nearly constant angle of incidence over a wide range of operation is achieved pursuant to this invention by positioning the blades so that, as viewed in a typical meridional cross-section, the radial distance from the axis of rotation of the centroid of the meridional cross-section of the annular fluid passage lies in between the radial distances from the axis of rotation of at least the major portions of the inner and outer edges

of the blades. The term "annular fluid passage," as used throughout this application, encompasses the entire passage about the stator ring including that portion of the passage traversing the blade cascade.

Thus, in FIG. 7, which is an enlargement of a typical meridional cross-section of the machine shown in FIGS. 1-4, the radial distance, r_2 , from the axis of rotation 14 of the centroid 98 of the meridional cross-section of the annular fluid passage 46 is greater than the radial distance r_1 from the axis 14 to the inner blade edge 62 and less than the radial distance r_3 from the axis 14 to the outer blade edge 64.

In the embodiment of FIG. 7, the edges 62 and 64 are parallel with the axis 14 so that the centroid 98 lies between the entire extents of the edges 62 and 64. In contrast, in FIG. 8 there is shown a meridional cross-section configuration of a machine including blades 60a each having inner and outer edges 62a and 64a, respectively, that converge toward the blade tip so as to be non-parallel with respect to each other and the axis of rotation 14a. In this case, the radial distance, r_2' , of the centroid 98a of the meridional section of the annular fluid passage 46a intersects the inner blade edge 62a. However, in accordance with the principles of the invention, all points along at least a major portion, p , of the length of the inner edge 62a lie at a radial distance r_1' from the axis 14a that is less than the radial distance r_2' . Further, in the example of FIG. 8, the entire length of the outer edge 64a lies outside the radial distance r_2' to the centroid 98a.

Another geometric interrelationship forming an aspect of the present invention and which aids in achieving high performance and efficiency is that the inner edges of the blades (such as the edge 62) are longer than the outer edges (such as the edge 64). This, of course, is a reflection of the geometry of the annular fluid passage 46 whose radially outward portions are narrower than the radially inward portions. Stated another way, the blade cascade, in a general sense, separates the wider, radially inward regions of the annular passage 46 from the narrower, radially outward regions. This geometry, in cooperation with the circumferential pressure gradient, results in a rise of static pressure (in the case of a compressor) as the fluid moves from inlet to outlet.

Separating the inlet and outlet openings 29 and 31 is a block seal 100 which, as shown in FIGS. 3 and 4, fills and seals the entire fluid passage along a short sector, the length of which is at least equal to the spacing between adjacent blades. The inlet and outlet openings 29 and 31 are immediately adjacent the block seal 100 so as to be in close proximity to one another. The block seal 100 is preferably fabricated as an integral part of the stator ring 42 and in this specific embodiment has flat side faces 102 and 104 which coincide with meridional planes. The block seal 100 further has an arcuate recess 106 to permit passage of the blade cascade, the forward surface 108 of the recess 106 being flush with the face 74 on the stator ring. The recess 106 is dimensioned so that a minimum running clearance is provided for passage of the blade cascade.

The block seal 100 also has a U-shaped, angularly oriented notch 110 fairing smoothly into the outlet passage 30 to minimize losses.

With reference to FIG. 4 which shows, for a compressor, a typical streamline 114, that is, the path of a typical fluid particle, the fluid particle enters the inlet pas-

sage 26 and is directed thereby in a generally radial direction through the inlet opening 29 and under the stator ring 42. The fluid, following a generally helicoidal path, then enters the blade cascade at point A nearly perpendicular to the blade rotational direction. The energy level of the fluid is increased as a result of the work done by the blades on the fluid and this increase in energy level is seen mostly as an increase in absolute velocity and a small increase in static pressure. As the fluid leaves the blades at point B and circulates about the stator ring to the point C, a further pressure increase occurs as a consequence of the diffusing action which converts kinetic energy (energy associated with velocity) into static pressure. This diffusing action is a result of the increasing pressure as the fluid advances circumferentially about the axis 14. The fluid, entering at point C, then passes through a portion of the blade cascade a second time and the process may repeat itself several times before the fluid exits via the outlet passage 30. The fluid, at all times, follows a smooth, orderly, generally helicoidal path about the stator ring 42 and in the case of a compressible fluid processed by a machine having an annular fluid passage of constant cross-sectional area, that is, a passage defined by a surface of revolution about the rotational axis 14, the pitch of the generally helicoidal path 114 decreases as the fluid moves circumferentially about the axis 15 into regions of higher pressure.

The particular embodiment under discussion may be designated as an "outflow" machine in which the fluid moves through the blade cascade in a direction away from the axis 14. The inner edges 62 thus function as the leading edges of the blade while the outer edges 64 are the trailing edges. It will be appreciated that it is possible, with obvious modifications based on the disclosure herein, to reverse the fluid flow direction so that an "inflow" pattern is established in which the fluid moves toward the axis 14 during its traversal of the blade cascade. In that case, the outer edge becomes the leading edge and the inner edge functions as the trailing edge.

Turning now to FIGS. 9 through 13, there is shown a pair of alternative embodiments the construction of which is such that the meridional cross-section area of the fluid passage is gradually decreased between inlet and outlet so as to enhance the efficiency of the machine when compressible fluids, such as air, are processed. In the embodiment of FIGS. 9-11, the meridional cross-section area of the stator ring 120 is gradually increased between the inlet and outlet ends, the inlet end 122 having a relatively small cross-section area and the outlet end 124 having a relatively large cross-section area. All of the other geometric considerations discussed above are retained in this version for high performance and high efficiency operation.

In FIGS. 12 and 13, the meridional cross-section area of the interior wall 130 of the casing 132 is gradually decreased so as to obtain the same result as that accomplished with the embodiment of FIGS. 9-11. The meridional cross-section near the inlet end appears as shown in FIG. 10 and FIG. 13 shows a typical section near the outlet. In this embodiment, the stator ring 134 has a uniform meridional cross-section area along its entire length. It will be obvious that a combination of the configurations of FIGS. 9-11 and 12-13 may be used, that is, both the meridional cross-section area of the stator ring may be increased and the meridional

cross-section area of the interior wall of the casing may be decreased gradually between inlet and outlet.

The above-described geometry and orientation of the blades, the orientation and positions of the inlet and outlet passages relative to each other and to the block seal, and the small number or parts required, all contribute, in comparison to the prior art, to furnish a compact, high performance, simple and low cost fluid energy transfer machine.

Further structural simplifications may be achieved by fabricating the intermediate casing portion, the stator ring and the block seal as a one-piece casting. Such an arrangement, along with certain additional modifications, is shown in the embodiment of FIGS. 14-18. This embodiment of the invention comprises a casing 140 enclosing a rotor 142 mounted on a shaft 144 carried by ball bearings 146 for rotation about a central axis 148. The rotor has a cascade of blades 149 similar to those already described. The casing 140 includes a front portion 150, an intermediate ring-like portion 152 and a rear portion 154 joined to the front portion 150 about the outer periphery of the machine. The casing portions 150, 152 and 154 and an arcuate surface 156 on the rotor 142 together define a smooth-walled, interior wall surface 158 having a shape as described in connection with previously discussed embodiments.

The casing 140 encloses a stator ring 160 defining with the wall surface 158 an annular fluid passage 161. The ring 160 includes as an integral part thereof, a block seal 162 having an outer surface 164 conforming closely to the configuration of the interior wall surface 158. As shown in FIGS. 15 and 18, a part of the surface 164 comprises the periphery of a flange 166 having spaced, radially oriented portions 168 and 170 having side surfaces lying in meridional planes. The flange 166 further includes an undulation 172 which functions both to reduce the acoustic level during operation and to define a notch that blends into the outlet passage 174 to minimize losses. The surface 164 may be grooved along its length, as indicated in the drawings by the reference numeral 176, or alternatively, may be smooth; in either case, appropriate sealing means is applied to the surface 164 so that a leak-tight joint is provided and the stator ring is securely held in place.

The intermediate casing portion 152 is supported by two or more ribs 178 projecting inwardly from the stator ring. In this way, the ring 160, the intermediate casing portion 152 and the ribs 178 may all be formed as a single casting. Alternatively, studs (such as studs 44 in the embodiment of FIGS. 1-4) may be used in place of, or in conjunction with, the ribs 178. As a further structural simplification, the inner rim seal (reference numeral 56 in FIG. 3) is replaced by a transverse face seal 180 between the rotor and intermediate casing portion extending normal to the axis 148.

The front casing portion 150 defines a circumferential, axially extending chamber 184 including an enlarged inlet passage 186 communicating with the annular fluid passage 161 through an inlet opening 188. The opening 188 extends a short arcuate distance (for example about 60°) about the axis 148 and is positioned immediately adjacent the block seal 162. The chamber 184 tapers outwardly toward the rear of the machine and is divided into sections by three longitudinal partitions 190, the chamber sections communicating with each other and the inlet passage 186 through openings 192 behind the partitions 190. Fluid is introduced into

the chamber 184 through a centrifugal dirt separator 194, the operation and structure of which are well known in the art, and a series of arcuate intake ports 196. A web 198, centrally positioned within the intake passage 186 strengthens the front casing portion and assists in directing the incoming fluid.

The fluid entering the chamber 184 has a relatively high absolute velocity as represented by the vector V_1 . The outwardly tapering configuration of the chamber 184 provides a diffusing characteristic so that the fluid entering the annular fluid passage 161 will have a lower absolute velocity, represented by the vector V_2 . Thus, one function of the chamber 184 is to minimize losses by making the velocity V_2 required at the inlet opening 188 compatible with the velocity V_1 produced by the dirt separator 194. The configuration of the chamber 184 and the presence of the web 198 serve to direct the fluid approaching the inlet opening 188 generally along meridional planes so that the fluid enters the blades 149 approximately perpendicular to the direction of blade movement, similar to that described in connection with the inlet passage 26 of the embodiment of FIGS. 1-4. The passage 174 is oriented in the same direction as the outlet passage 30 of the above-mentioned embodiment.

To illustrate an application of the present invention, FIG. 19 is a schematic representation of an automobile internal combustion engine 210 and associated emission control system. The emission control system shown is typical of the systems currently under consideration for general use by the automotive industry to comply with clean air standards and includes a thermal reactor 212 connected to exhaust passages 214, a catalytic converter 216 and a muffler 218. An air pump 220, which may be of the type disclosed herein, is directly driven by the engine 210 and supplies secondary air under pressure to the exhaust passages 214 via a flow controller 222.

What is claimed is:

1. A machine for transferring energy between a fluid medium and a shaft rotatable about an axis, comprising:

a casing having inlet and outlet openings and enclosing a stator ring and defining therewith a vaneless fluid passage extending circumferentially about said axis; and

a rotor mounted on said shaft and having blades projecting into said fluid passage, said blades having inner and outer edges relative to said axis, the radial distance from said axis to the centroid of the meridional cross-section of said fluid passage lying between the radial distances from said axis to at least major portions of said inner and outer edges.

2. A machine, as defined in claim 1, in which said inner edges are longer than said outer edges.

3. A machine, as defined in claim 1, in which at least one of said inner and outer edges extends parallel with said axis.

4. A machine, as defined in claim 1, in which the casing includes inlet and outlet passages communicating with said fluid passage through said inlet and outlet openings, said inlet passage being oriented to direct the fluid so that the absolute velocity thereof approaches the blade cascade approximately perpendicular to the direction of movement of said cascade, said outlet passage being oriented in approximate alignment with the

direction of the absolute velocity of the exiting fluid approaching said outlet passage.

5. A machine, as defined in claim 1, in which the fluid moves away from said axis of rotation during its passage through the blade cascade.

6. A machine as defined in claim 1, in which the fluid moves toward said axis of rotation during its passage through the blade cascade.

7. A machine for transferring energy between a fluid medium and a rotatable shaft, comprising:

a casing having a smooth interior wall surface defining a smooth-walled toroidal space about a central axis of rotation, said casing including fluid inlet and outlet openings;

a block seal within said toroidal space separating said inlet and outlet openings;

a stator ring mounted within said toroidal space and defining with said casing an annular fluid passage extending circumferentially about said axis of rotation, the meridional cross-section of said annular fluid passage having a centroid; and

a rotor mounted on said shaft for rotation about said axis and including along its outer periphery a blade cascade, each blade having an inner edge and an outer edge, said inner edge being positioned closer to said axis than said outer edge, at least a major part of said inner edge lying radially inward of said centroid and at least a major part of said outer edge lying radially outward of said centroid.

8. A machine, as defined in claim 1, in which the projection of each blade on a meridional plane has a generally trapezoidal configuration and includes a tip, said stator ring including a surface immediately adjacent said blade tips and having a configuration corresponding to said tips.

9. A machine, as defined in claim 7, in which said inlet and outlet openings are in close angular proximity to one another and said block seal has side surfaces generally coinciding with meridional planes.

10. A machine, as defined in claim 7, in which said casing includes inlet and outlet passages and said block seal has a portion fairing smoothly into said outlet passage.

11. A machine, as defined in claim 7, in which the meridional cross-section of said annular fluid passage varies between said inlet and outlet openings to compensate for compressibility of the fluid.

12. In a machine for transferring energy between a fluid medium and a shaft rotatable about an axis, said machine including a casing and stator ring enclosed within said casing, said casing and ring defining an annular fluid passage extending about said axis, inlet and outlet passages in said casing in communication with said fluid passage, a rotor mounted on said shaft and carrying a blade cascade extending into said fluid passage in close proximity to said stator ring, each blade of said cascade having leading and trailing edges, one of which edges is closer to the axis than the other, the improvement in which the edge closer to said axis is longer than the other edge.

13. A machine, as defined in claim 12, in which the centroid of the meridional cross-section of said annular

passage lies at a radial distance from said axis that is between the radial distances from said axis to at least major portions of said leading and trailing edges.

14. A machine, as defined in claim 12, in which said edges extend generally parallel with said axis of rotation.

15. A machine for transferring energy between a fluid medium and a shaft rotatable about an axis, comprising:

a casing having inlet and outlet openings and enclosing a stator ring and defining therewith a fluid passage extending circumferentially about said axis; and

a rotor mounted on said shaft and having blades projecting into said fluid passage, said blades having leading and trailing edges generally parallel with said axis, one of said edges lying closer to said axis than the other of said edges, the edge closer to said axis being longer than said other edge.

16. A machine for transferring energy between a fluid medium and a shaft rotatable about an axis, comprising:

a casing having inlet and outlet openings and enclosing a stator ring and defining therewith a fluid passage extending circumferentially about said axis; and

a rotor mounted on said shaft and having blades projecting into said fluid passage, said blades having leading and trailing edges generally parallel with said axis and in which the radial distance from said axis to the centroid of the meridional cross-section of the fluid passage lies between the radial distances to said leading and trailing edges.

17. A machine for transferring energy between a fluid medium and a shaft rotatable about an axis, comprising:

a casing enclosing a stator ring and defining therewith a fluid passage extending circumferentially about said axis, said casing including an inlet passage and an outlet passage, said passages communicating with said fluid passage through inlet and outlet openings; and

a rotor mounted on said shaft and including a blade cascade projecting into said fluid passage and separated from said stator ring by a close running clearance, the blades of said cascade having leading edges and trailing edges, said inlet passage being oriented to direct the fluid generally radially to enter the blade cascade approximately perpendicular to the direction of movement of said cascade past said inlet opening, said outlet passage being oriented in approximate alignment with the direction of the absolute velocity vectors of the fluid exiting from said blade cascade.

18. A machine, as defined in claim 17, in which said inlet and outlet openings are in close circumferential approximation and separated by a block seal having a notch conforming to the contour of said outlet passage.

19. A machine, as defined in claim 18, in which said block seal has side surfaces generally coinciding with meridional planes.

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