Jan. 31, 1956 E. A. STALKER 2,732,999

AXIAL FLOW ELASTIC FLUID TURBINE POWER PLANT, INCLUDING AN AXIAL FLOW RADIAL DIFFUSION COMPRESSOR

Filed July 31, 1946 2 Sheets-Sheet 1

Fig. 1

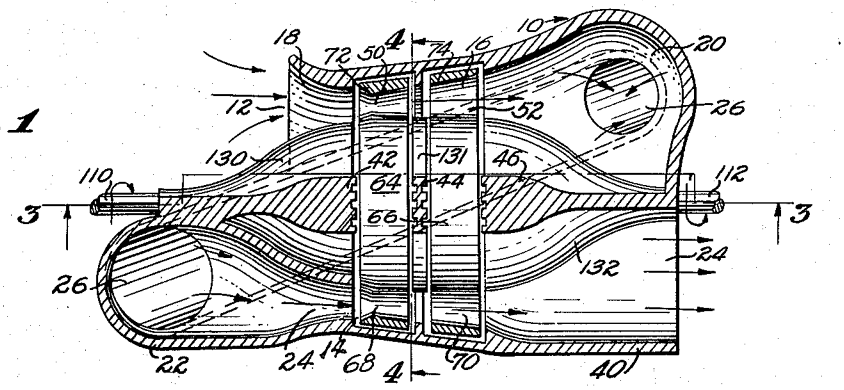


Fig. 3.

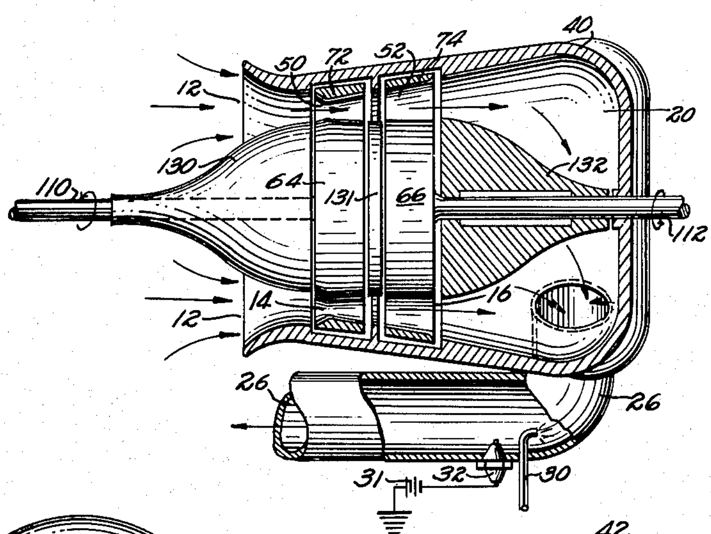


Fig. 2.

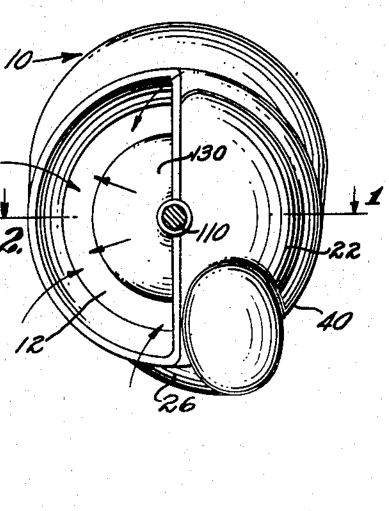
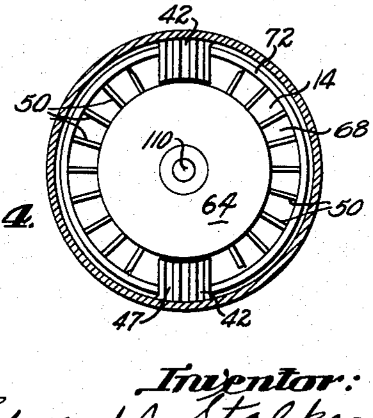


Fig. 4.



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AXIAL FLOW ELASTIC FLUID TURBINE POWER PLANT, INCLUDING AN AXIAL FLOW RADIAL DIFFUSION COMPRESSOR

Filed July 31, 1946 2 Sheets-Sheet 2

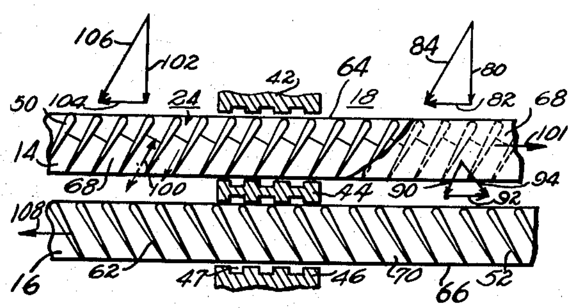

Fig. 5.

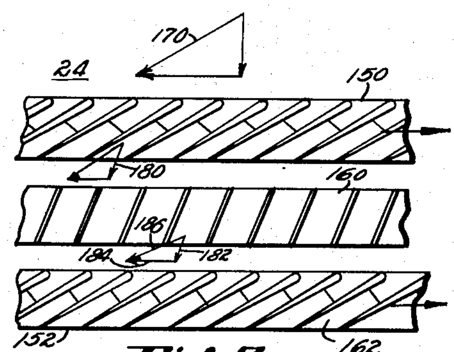

Fig. 7.

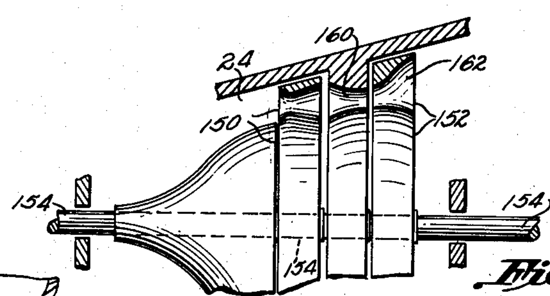

Fig. 6.

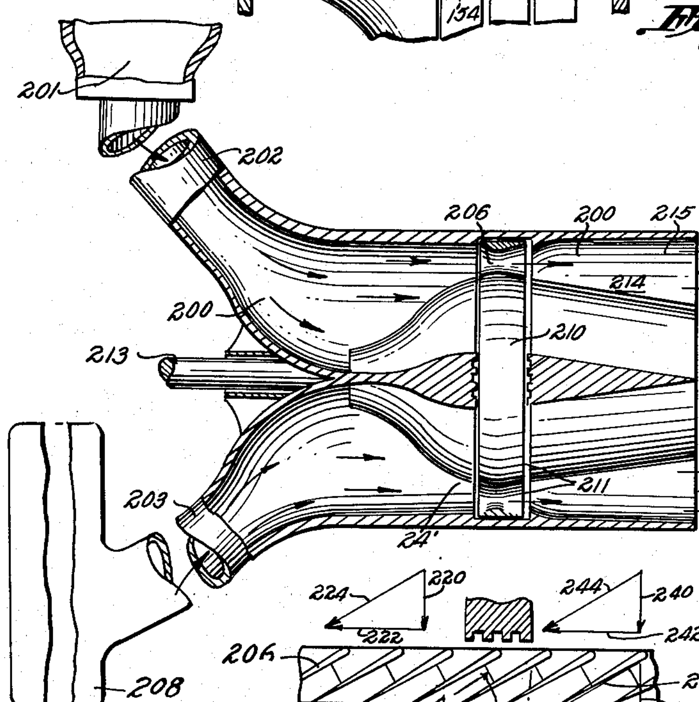

Fig. 8.

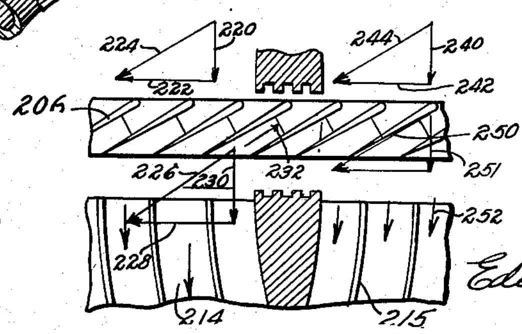

Fig. 9.

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AXIAL FLOW ELASTIC FLUID TURBINE POWER PLANT, INCLUDING AN AXIAL FLOW RADIAL DIFFUSION COMPRESSOR

Edward A. Stalker, Bay City, Mich.

Application July 31, 1946, Serial No. 687,385

4 Claims. (Cl. 230—122)

My invention relates to turbines and particularly to internal combustion turbines commonly called gas turbines. It also relates to compressors employing a novel diffusion principle of compressing fluid.

An object of my invention is to provide turbine rotors or runners adapted to be driven by jet reaction by jets passing between substantially straight blades.

Another object is to provide an effective and simple compressor operating by radial diffusion.

Another object of my invention is to provide a means of cooling the blades wherein the blades pass successively into a cooling passage.

Other objects will appear from the description, drawings and claims.

I accomplish the above objects by the means illustrated in the accompanying drawings in which—

Figure 1 is a fragmentary axial section through a gas turbine along line 1—1 in Figure 2 according to the present invention;

Figure 2 is a front view of the turbine;

Figure 3 is a fragmentary section taken along line 3—3 in Figure 1 omitting the wall separating the turbine and cooling passages;

Figure 4 is a section along the line 4—4 in Fig. 1;

Figure 5 is a development of the blade stages;

Figure 6 is a fragmentary axial section showing an alternate design of rotors;

Figure 7 is a development of the stages of the turbine of Fig. 6;

Figure 8 is a fragmentary axial section of another form of the invention; and

Figure 9 is a development of the stage of the turbine of Fig. 8.

Cross reference is made to Serial No. 593,631 filed May 14, 1945, now Patent 2,648,492 issued August 11, 1953, entitled Prime Movers, which discloses a compressor rotor employing radial diffusion. In the earlier application, the claims are directed to a compressor rotor employing radial diffusion generally while in the present application the radial diffusion rotor is disclosed in cooperation with a novel stator structure.

The gas turbine of this invention has a rotor whose blades are substantially straight in the direction of the flow between the blades. The blades pass successively through a hot motive gas and a cooling fluid. The straight blades and the mode of producing a force reaction between the fluid and the rotor leads to high efficiency in the turbine passage and the cooling passage.

In my patent application Serial No. 538,634, filed June 3, 1944, now abandoned, I have described a method of cooling blades where the blades pass successively out of a passage containing the flow of hot motive gas into another passage containing a flow of cooling fluid. That turbine had conventionally curved blades which are very satisfactory for extracting energy from the motive gas but are wasteful of energy in the cooling passage. This is so because the curvature of the blades for the turbine side is wrong for the cooling side or passage. It is wrong

for both possible conditions of operation of the cooling passage. That is, if the blades are correctly designed for the turbine passage, they will be incorrect for pumping air through the cooling passage if the machine is operated to perform such a function. It will also be improper and of low efficiency if it is desired to force air through the group of blades in the cooling passage without doing work on the rotor. Some work will always be done because of the curvature of the blades.

The present invention discloses a machine in which blading is provided which is just as efficient in pumping air in the cooling passage as in operating in the turbine passage.

The present invention also discloses blading which will absorb work in the turbine passage but will do no work on the cooling air flow in the cooling passage if this mode of operating is desired.

Conventional turbines of the curved blade type get a momentum reaction directed peripherally because the blades curve the flow. In the turbine of the present invention the gas flow proceeds along the straight axis between the blades with increasing velocity, thereby creating a thrust along the axis. A component of this thrust acts in the peripheral direction to rotate the rotor.

Referring now particularly to the drawings, the turbine is 10. Air enters the inlet 12 and flows through the cooling passage 18 and the two rotors 14 and 16 into the collector 20. From it the flow goes via duct 26 to the inlet collector 22 supplying the turbine passage 24. The air is heated in duct 26 by the injection of fuel from nozzle 30. This is ignited by the spark plug 32 served with electricity from a suitable source 31.

The case 40 of the turbine is divided into a turbine passage and a cooling passage by the partitions 42, 44 and 46, shown in Figs. 1 to 5. The rotors pass their blades 50 and 52 from the turbine passage 24 successively into the cooling passage 18 and vice versa through the gaps between the walls.

Figure 4 shows the blades on the hub 64 and blades have been cut away or removed at the top and bottom to show the partition 42.

The air is drawn into the cooling passage 18 by the pumping action of the blades therein. It is compressed and delivered to the turbine passage for the production of power by the blades passing through the turbine passage.

The blades are formed in a special manner so as to be satisfactory and efficient in both turbine passages. It will be observed from Figs. 1, 3 and 5 that the blades 50 and 52 are thin elements of sharp or rounded nose and sharp trailing edge. They are all fixed substantially parallel to each other on the rotor hubs, respectively 64 and 66. The passages 68 and 70 between the blades are straight when viewed in a radial direction as shown in Fig. 5 but are tapered when viewed in a peripheral direction, as shown in Figs. 1 and 3.

The taper of the passage is formed in rotor 14 by the shape of the rotor hub 64 and the shroud ring 72, in rotor 16 by the shape of hub 66 and shroud ring 74. If desired all the tapers of the passages could be accomplished in either the hubs or the shroud rings.

The passages between blades have a greater cross sectional area at exit than at inlet so that they will pump air in the cooling passage. How this pumping action is accomplished is best shown by the vector diagrams in Fig. 5. Air flows axially in passage 18 as indicated by vector 80. Due to the movement of the blades there is a peripheral relative vector 82 giving the resultant vector 84 directed along the axis of passage 68. Since the passage 68 is expanding to a larger exit, the velocity at the exit is reduced to the vector 90 relative to the rotor. There is also the peripheral vector 92 equal to 82 but directed oppositely so that the resultant is 94. It will be clear that the air leaving the first rotor has acquired an absolute net peripheral velocity component since the air was originally moving only axially. Hence the blades have added energy to the air or a pumping action has taken place. If the vector 90 had been as long as 84, the air would have had only an axial velocity upon leaving the rotor and no pumping would have been accomplished.

The air from the rotor 14 is directed along the vector 94 and the axes of the passages 70 of the rotor 16. The expansion of the air is continued in these passages and this rotor contributes further pumping action.

The passage 70 is divergent and continues the expansion of the gas in the divergent portion of 68.

The passages 68 of the first rotor have a converging portion preceding the diverging portion. This form is desirable in the turbine passage so as to produce a turning effort on the rotor.

The hot gases from duct 26 have sufficient temperature and pressure to achieve a supersonic velocity in passing through passages 68. The gases enter at a subsonic velocity, attain sonic velocity at the throat, and become supersonic at the exit. Since the same mass of gas passes all sections of a passage 68 and the exit velocity is higher than the inlet velocity, there is a thrust acting on the tapered walls of the passage, namely on the shroud and the hub 64. This force 100 is indicated on the hub 64. It is obvious that this force has a component in the peripheral direction and will turn the rotor in the direction 101. In order to bring the gas velocity to a supersonic velocity, it is necessary to first constrict the passage and then expand it. However, where the pressures are not very high the rounding of the leading edges of the walls may be sufficient for the inlet or converging portion.

Fig. 5 shows the vector diagrams for the motive gas flow. The axial vector is 102, the relative peripheral vector is 104 and the resultant is 106 directed along the axes of passages 68. At exit the gas is directed along the axes of the passages 70. Further expansion and increase of velocity occur in these passages so that rotor 16 receives a turning force rotating rotor 16 in the direction 108.

Rotor 14 is fixed to shaft 110 while rotor 16 is fixed to shaft 112. To each can be attached a power load such as a propeller. The two rotors and their respective shafts rotate in opposite directions.

Suitably contoured parts 130, 131 and 132 form the flow passages 18 and 24 and serve to house the rotor shafts. They are supported from the case by the walls 42, 44 and 46 respectively.

The walls such as 42, 44 and 46 are labyrinthed at 47 to reduce the flow of fluid from the turbine passage to the cooling passage and vice versa. See Figures 1 and 4.

The blades form radially directed walls for the blade passages and these should be substantially continuous from the inlet or front side of the rotor to the exit or rear side of the rotor since the flow in each passage is supersonic in the diverging portion of each passage. If free edges exist in the passage shock waves will be precipitated at each edge and cause large losses in efficiency.

The passage should expand gradually and continuously without abrupt changes in cross sectional area since such changes will also precipitate shock wave losses.

In another form of the invention shown in Fig. 6 both rotors 150 and 152 are fixed to the same shaft 154. Rotors 150 and 152 are of the type respectively of rotors 14 and 16 of Fig. 1. Between them is interposed a stator having the tapering passages 160 to reduce the supersonic velocity of the gas from rotor 150 to sonic. It is then expanded again in the diverging passages 162 of rotor 152. See Figs. 6 and 7. The entering resultant velocity vector in Fig. 7 for the first stage is 170. Leaving, the resultant velocity is 180. This is also the velocity entering passages 160. Leaving 160, the velocity is 182 giving with the relative peripheral velocity 184 of 152 a resultant 186 directed along the axes of passages 162. The tapered walls of the passages of rotors 150 and 152 receive a thrust having a peripheral component which turns them and shaft 154.

By expanding the rotor passages radially a far greater expansion can be employed than if the passages are expanded in the peripheral direction. The latter requires surfaces which are curved in the peripheral direction. This limits the amount of expansion since a flow trying to advance against an adverse pressure gradient separates from the curved surface for a relatively small pressure rise. On the other hand with radial expansion of the passage no peripheral curvature of blades is required so that separation of the flow on them is avoided. Any tendency to separate at the case wall is restrained by the centrifugal pressure which forces the flow against this wall as remarked above. At the inner or hub wall there is an outwardly increasing pressure gradient due to the whirl given the fluid and this also restrains the flow from separation from this wall. Consequently very great rates of expansion can be used. These conditions are borne out by test experience which show that for the same efficiency and blade tip speeds the pressure rise is about three times that of rotor passage employing peripheral expansion.

It is to be noted in Fig. 6 that the outer peripheral wall of each passage 162 departs from the axis of rotation at a greater rate than the inner peripheral wall. Since centrifugal force will throw the passage fluid against the outer wall, the fluid will not tend to separate from the wall even for a very large angle of divergence. This feature is particularly important when the rotor passages are acting to pump fluid since it makes possible a short and light weight rotor.

In still another form of the invention shown in Fig. 8 the rotor blades are designed to provide no pumping action in the cooling passage. The cooling air is forced through the cooling passage 200 by an external source 201 of compressed air delivered by the duct 202. The flow through the turbine or hot fluid passage 24′ is supplied from a suitable source 208 by duct 203. The cold flow is exhausted through exit 204 and the hot flow through exit 205.

In order to do no pumping in the cooling passage 200, the passages 206 between the blades on the rotor hub 210 are made with equal cross sectional areas at inlets and exits. The blades and hub constitute the rotor 211 mounted on shaft 213. The stator passages 214 are preferably made of increasing cross sectional area.

Referring to Fig. 9 the axial velocity of the gas in the turbine side is the subsonic velocity 220, the relative peripheral velocity is 222 and the resultant subsonic velocity is 224. Upon leaving the rotor the velocity is supersonic of magnitude shown by vector 226. The peripheral velocity vector is 228 giving a velocity vector 230 relative to the stator 215. Since the state mass passes the inlet and the exit of each passage 206 there has been an increase in the momentum at exit over that at the inlet so that there is a thrust 232 which has a component turning the rotor. The increase of momentum will always be present if the inlet velocity is subsonic and the exit velocity is supersonic, even through the inlet and exit areas are equal.

On the cool fluid side the axial velocity vector is 240 and the peripheral vector is 242 of the same magnitude as vector 222. The resultant velocity at entrance to the rotor passages is the subsonic vector 244. The magnitude of the vector 250 leaving the rotor is the same as that of vector 244 because the areas of inlet and exit are equal. Vector 251 is equal to vector 240. The velocity vector for the fluid leaving the exit 205 is 252. Each rotor passage lies between the front and rear faces of the rotor or rotor stage. Rotor passages rotate in whole with the rotor.

A rotor stage is comprised of rotor blades defining passages directing a flow of the gas from one side of the rotor to the other or from one group of stationary vanes to another.

The rotor passages having a converging portion succeeded by a diverging portion are suited to receive elastic fluid at either subsonic or supersonic velocity and discharge said fluid at supersonic velocity.

It will also be clear that I have provided a novel and useful gas turbine which can operate at high temperatures because of the effective blade cooling.

While I have illustrated a specific form of this invention it is to be understood that I do not intend to limit myself to this exact form but intend to claim my invention broadly as indicated by the appended claims.

I claim:

1. In combination in an axial flow elastic fluid compressor for increasing the pressure of a fluid flow therethrough, a rotor mounted for rotation about an axis and having a plurality of blades, said rotor having passages therethrough between adjacent blades from front to back, said passages being set peripherally obliquely to said axis and having an exit greater than the inlet in radial depth and cross-sectional area, the radial depth of each said inlet passage at the inlet end thereof being less than the radius from said axis to the radially inward side of each said passage at the inlet end of said passage, said exit facing rearward to discharge fluid rearward in the general direction of said axis, said rotor passages being arranged such that the angle between the forward portion of each blade and said axis is at least as great as that between the aft portion thereof and said axis, the axial length of the diverging portion of said rotor passages being not greater than the maximum radial depth thereof to provide a substantial angle of divergence between the inner and outer walls of said passage, the radii from said axis to the tips of said blades substantially increasing in the downstream direction, and a stator structure positioned adjacent said rotor on the downstream side thereof, said structure having a flow passage therethrough of rearwardly decreasing radial depth and cross sectional area measured normal to said axis, said passage being adapted to receive fluid thereinto from said rotor passage.

2. In combination in an axial flow elastic fluid compressor for increasing the pressure of a fluid flow therethrough, a rotor mounted for rotation about an axis and having a plurality of blades, said rotor having passages therethrough between adjacent blades from front to back, said passages being set peripherally obliquely to said axis and having an exit greater than the inlet in radial depth and cross sectional area, the radial depth of each said inlet passage at the inlet end thereof being less than the radius from said axis to the radially inward side of each said passage at the inlet end of said passage, said exit facing rearward to discharge fluid rearward in the general direction of said axis, said rotor passages being arranged such that the angle between the forward portion of each blade and said axis is at least as great as that between the aft portion thereof and said axis, the axial length of the diverging portion of said rotor passages being not greater than the maximum radial depth thereof to provide a substantial angle of divergence between the inner and outer walls of said passage, the radii from said axis to the tips of said blades substantially increasing in the downstream direction, and a stator structure positioned adjacent said rotor on the downstream side thereof, said structure having a flow passage therethrough of decreasing radial depth in the downstream direction, said passage being adapted to receive fluid thereinto from said rotor passage, the inner peripheral wall of said stator extending radially outward and rearward with its rear edge further out radially than the front edge thereof.

3. In combination in an axial flow compressor having an inlet duct for receiving a fluid flow and for the compression and impulsion of said fluid flow therethrough with increase in the static pressure thereof, a compressor rotor mounted for rotation about an axis, said rotor having a plurality of peripherally spaced streamlined blades defining a plurality of passages extending through from the front to the back of said rotor, the shape of said rotor blades being such that the angle between the forward portion of each blade and said axis is at least as great as that between the aft portion thereof and said axis, said blades being movable across the cross section of said duct, each said passage having increasing radial depth and cross-sectional area in the downstream direction with the cross-sectional area of the exit of each passage being greater than the inlet area thereof with resultant static pressure rise in the fluid flow, the axial length of the diverging portion of said rotor passages being not greater than the maximum radial depth thereof to provide a substantial angle of divergence between the inner and outer walls of said passage, the radii from said axis to the tips of said blades substantially increasing in the downstream direction, a stator structure positioned adjacent said rotor on the downstream side thereof, said structure having a plurality of stator passages therethrough adapted to receive fluid from said rotor, each said stator passage having a smaller exit than inlet cross-sectional area, and another rotor positioned adjacent said structure on the downstream side thereof, the last said rotor having a plurality of passages therethrough of increasing radial depth and cross-sectional area in the downstream direction and adapted to receive fluid from said stator structure passages, and means to rotate said rotors to impel a flow of fluid through said duct and passages.

4. In combination in an axial flow compressor for increasing the pressure of a fluid flow therethrough, a rotor mounted for rotation about an axis, said rotor having a plurality of blades and having passages therethrough between adjacent blades from an inlet at the front to an exit at the back thereof, said passages being set obliquely to said axis, said exit having a greater radial depth and area than said inlet, the radial depth of each said inlet passage at the inlet end thereof being less than the radius from said axis to the radially inward side of each said passage at the inlet end of said passage, said exit facing rearward to discharge fluid rearward in the general direction of said axis, said rotor passages being shaped such that the angle between the forward portion of each said passage and said axis is at least as great as that between the aft portion thereof and said axis, the axial length of the diverging portion of said rotor passages being not greater than the maximum radial depth thereof to provide a substantial angle of divergence between the inner and outer walls of said passage, the radii from said axis to the tips of said blades substantially increasing in the downstream direction, a stator structure positioned adjacent said rotor on the downstream side thereof, said structure having a flow passage therethrough of rearwardly decreasing cross-sectional area with its exit area less than its inlet area and adapted to receive fluid thereinto from said rotor passage.

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