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3,460,748

RADIAL FLOW MACHINE

Original Filed Dec. 10, 1962

2 Sheets-Sheet 1

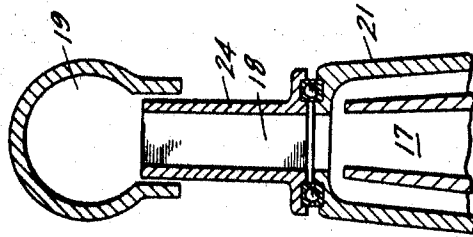


Fig. 1

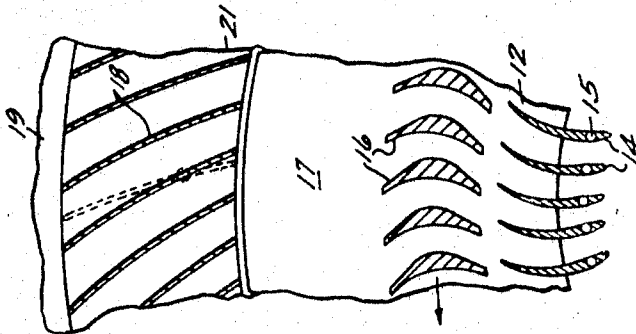


Fig. 2

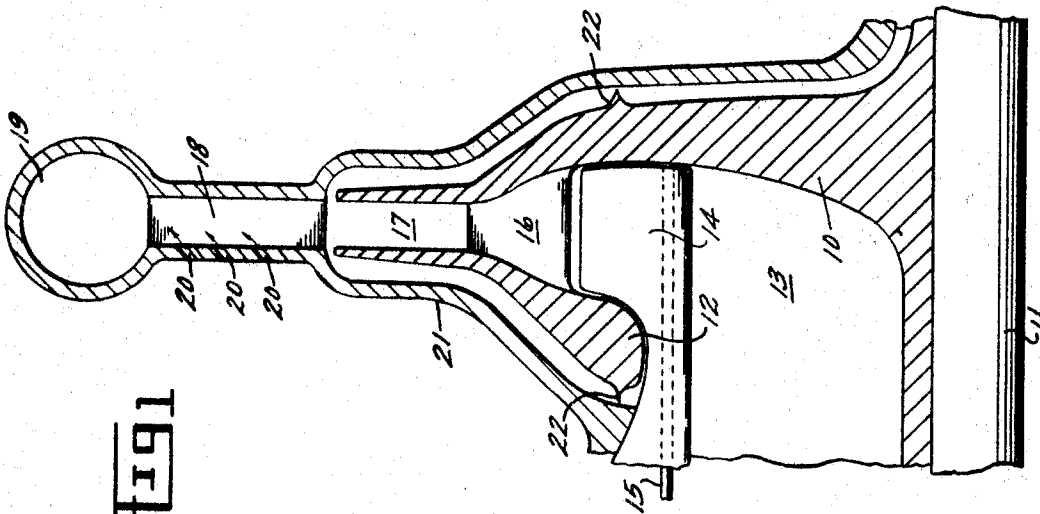


Fig. 3

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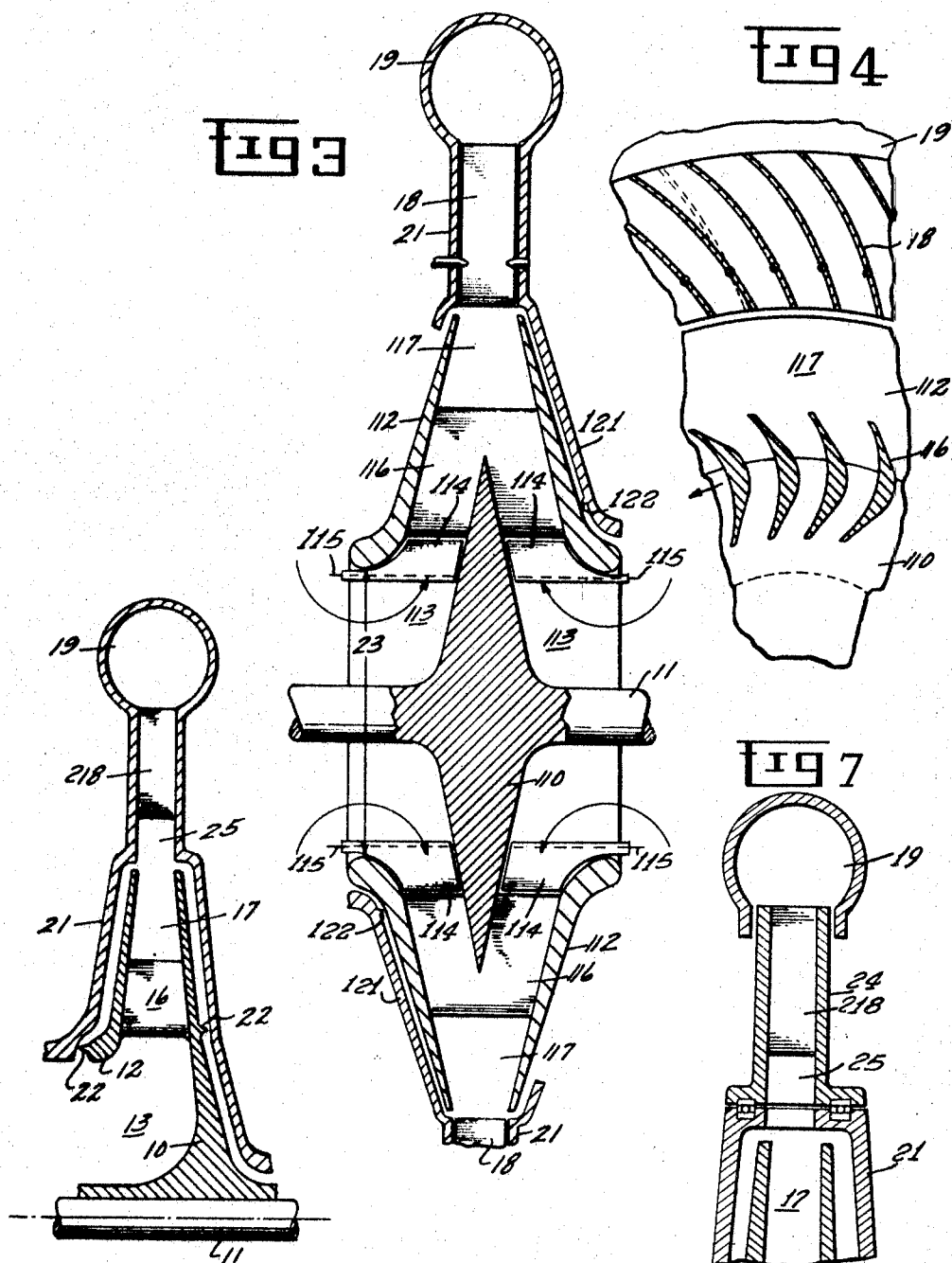
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RADIAL FLOW MACHINE

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Continuation of application Ser. No. 243,561, Dec. 10, 1962. This application Nov. 1, 1967, Ser. No. 684,585

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12 Claims

ABSTRACT OF THE DISCLOSURE

The disclosure shows a radial outflow compressor having two axially spaced rotor disks. A plurality of highly cambered blades extend between the disks and are disposed generally radially of the rotor axis with the blades being spaced from and concentric of the rotor axis. The disks define an inlet fluid flow path to the blades which directs the entering fluid essentially normal to the rotor axis and outwardly thereof. Guide vanes are provided to properly direct the air radially outwardly towards the highly cambered blades extending between the disks. The opposed surfaces of the disks, from the inward to the outward edges of the blades, are converging so as to form an annular rotating compressor section through which fluid flows from the inlet. The disks extend radially outwardly of the blades to form a rotating diffuser. A stationary, bladed, annular diffuser is disposed concentric of the rotating diffuser and is a continuation of the rotating diffuser. The pressurized fluid thus passes from the rotating diffuser to the stationary diffuser and is collected by appropriate means for direction to the point of utilization of the pressurized fluid. A second embodiment illustrates an inlet to the compressor wherein fluid may be directed thereto from opposite axial directions.

The present application is a continuation of co-pending application Ser. No. 243,561, filed Dec. 10, 1962, now abandoned.

The present invention relates to a radial flow machine and, more particularly, to a radial outflow compressor that pumps to high compression ratio in a single stage.

One of the problems in small lightweight jet type engines is the inability to provide a satisfactory compressor that obtains sufficient compression ratio and is still small and lightweight enough to be satisfactory for many applications. All multi-stage axial compressors, although capable of reasonable performance because of very light blade loadings, are prohibitively expensive for many engine applications. This is true because of the many rows of high precision blading required to obtain the compression ratio needed. Additionally, axial flow single stage supersonic compressors as well as centrifugal compressors of conventional geometry (whether inflow or outflow) have failed to develop the level of efficiency required to make a high performance engine. Their performance limitations are related to the basic axial inlet aerodynamics which generate strong spanwise variations in fluid velocity and flow angle into each blade row. These spanwise variations of fluid properties creating secondary flows not only restrict the attainable performance of the compressor but require complex twisted and tapered blade shapes that are expensive to manufacture.

A large contribution to rotor losses in axial and centrifugal fluid machines such as compressors is believed to be due to relative vorticity. In compressors, both the axial flow rotor blades and the conventional centrifugal impellers, which inducts air in an axial direction and directs it radially outward, operate with a secondary flow superimposed on the free stream through flow relative velocity. Although the inlet absolute vorticity may be zero, the

relative vorticity sets up a circulatory flow in a plane normal to the throughflow velocity within the confines of the rotating passage. This results in a secondary velocity which analysis has shown is of the same order of magnitude as the throughflow velocity and contributes significantly to rotor, diffuser and stator losses.

The main object of the present invention is to provide a radial outflow fluid machine as applied to a compressor which is so constructed as to substantially eliminate secondary velocity and thus prevent any effect on the main free stream throughflow velocity.

Another object is to provide such a compressor in which the construction and location of the compressor blading is such that the flow is substantially two-dimensional throughout the complete compressor and the rotational speed of any spanwise element of a rotor blade is substantially constant.

A further object is to provide such a compressor in which the air is handled only in a radial outflowing direction and boundary layers are centrifuged outward in the compressor.

Another object is to provide such a compressor which utilizes a rotating diffuser in combination with compressor blades accepting only radially directed air to provide a very high pressure ratio single stage machine.

A further object is to provide such a compressor in which the sealing problems are reduced to a minimum and the sealing structure is maintained on a small diameter of the rotor.

Briefly stated, the invention provides a radial outflow fluid machine illustrated and described for convenience as applied to a compressor that may be a single or dual entry compressor and which has a rotor disk mounted on a shaft for rotation. A second disk concentric with and axially spaced from the rotor disk is provided. Arranged serially outward then there are provided cambered compressor blades preferably of constant untwisted cross-section which join the disks together to provide a rotatable unit with an unobstructed curved inlet air passage to direct air radially outward between the disks. The location of the compressor blades at a sufficient radial distance from the center of rotation and the unobstructed inlet insure that only radial or two-dimensional airflow reaches the compressor blades. The disks extend radially beyond the compressor blades to form a rotating vaneless diffuser passage which is followed by a vaneless stationary diffuser. A scroll collecting means is radially spaced from the diffuser passage to collect the air from the vaneless diffuser for distribution. Modification permits a vaneless diffuser chamber or passage between the end of the rotating diffuser and the vaneless diffuser as well as a freely rotatable separate diffuser beyond the rotor. Further modification permits boundary layer bleed to be used when necessary.

While the specification concludes with claims particularly pointing out and distinctly claiming the subject matter which is regarded as the invention, it is believed the invention will be better understood from the following description taken in connection with the accompanying drawings in which:

FIGURE 1 is a partial cross-sectional view of a typical single inlet compressor of the instant invention,

FIGURE 2 is a partial cross-sectional view showing the blade and vane layout of FIGURE 1,

FIGURE 3 is a view similar to FIGURE 1 showing a dual entry compressor,

FIGURE 4 is a partial cross-sectional view showing the blade and vane layout of FIGURE 3 omitting the guide vanes for clarity,

FIGURE 5 is a partial cross-sectional view of a typical single entry compressor like FIGURE 1 of a modified form employing a freely rotating vaneless diffuser,

FIGURE 6 is a partial cross-sectional view of a typical single entry compressor in a modified form, and

FIGURE 7 is a partial sectional view of a modified form of freely rotating diffuser structure.

In fluid machines such as axial compressors and conventional centrifugal compressors having an axial entrance for air and radial blades, secondary vortical flows are created. When the compressor wheel rotates, the incoming air that is induced into the passage between the blades rotates opposite to the wheel rotation and this occurs in each of the passages between blades to set up a vorticity. This imposes a vortex on the throughflow which is called secondary vorticity resulting in superimposed velocities on the blades and additional frictional losses are generated. The vorticity is set up by a difference in tangential blade velocity between the blade hub and tip because of the rotation of the blade around an axis. Additionally, the secondary vorticity represents energy which cannot be recovered in the form of pressure. Also in the conventional centrifugal compressor the incoming air that sets up the vortex is then turned to the radial direction. Turning the vortex through an additional turn into the radial direction sets up an even further vortex thus generating additional energy absorbing circulatory flows within the passages of the wheel creating undesirable frictional losses and non-uniform flow. Also, the performance of the vaneless or vaneless diffusers following the rotor is very sensitive to the uniformity of the flow which is received from such rotor. High efficiency in both rotor and diffuser is obtained by producing and maintaining two-dimensional flow.

In copending application Ser. No. 243,562 filed Dec. 10, 1962, and assigned to the same assignee as the instant invention, there is disclosed and claimed a compressor design that by-passes the vorticity problem by providing substantially two-dimensional flow. In that application a multi-blade arrangement is shown for a very high compression ratio.

The present invention is directed to an improvement combination on that design in the use of a vaneless diffuser with running side walls or a rotating diffuser. Rotating diffusers per se are known and the improvement in the instant invention lies in the use of a vaneless rotating diffuser in combination with substantially two-dimensional flow by particularly designing and locating and combining the compressor blades and other means serially as will be pointed out. Thus, with the instant invention, very high compression ratios are obtainable by a single row of rotating blades in addition to other advantages of boundary layer control and sealing as will be apparent as the description proceeds.

Referring first to FIGURE 1, the invention is shown in a typical single entry radial outflow modification. This comprises a rotor disk 10 rotatable around a central axis or shaft 11 which carries the disk. A second disk or hoop 12 is provided concentric with and axially spaced from rotor disk 10 and also spaced radially from shaft 11 to define, with the rotor disk 10, an unobstructed curved inlet air passage 13 between the disks to direct air radially outward between the disks. The term "unobstructed" means, as shown, that there is nothing between the fluid entering and the disk surface so that there is radial only entry of fluid, such as air, into the rotating system (or guide vanes when used) before any tangential velocity is imparted to the fluid. In order to provide the proper swirl to the incoming radial air, inlet guide vanes 14 are provided at a distance from the center of rotation and radially upstream of the rotor blades as shown to insure that they receive only radially flowing air thus insuring two-dimensional flow through the passage between the disks and eliminating the undesirable secondary vorticity and consequent losses. Inlet guide vanes 14 are preferably given a curvature for turning the incoming radial air in a direction opposed to the rotor or disk rotation to give the air a counter-swirl or tangential velocity as seen in

FIGURE 2. This permits a high relative velocity to the rotor and the resultant total pressure relative to the rotor also is increased. Thus, a higher pressure ratio is obtained. For higher efficiency, lower pressure ratio, the guide vanes can direct the air in the direction of rotation. While the guide vanes may in some instances be eliminated as illustrated in FIGURE 6, it is useful to have the guide vanes, for efficient off-design operation which is further assisted by making them adjustable by pivoting about axis 15 in any suitable manner not shown.

One of the difficulties of compressors of any type is the presence of boundary layers in the flow passages. The present invention provides a built-in boundary layer control by means of a rotating diffuser in conjunction with the two-dimensional flow through the compressor. The air through the compressor is compressed by means of a single stage of compressor blades 16 which are highly cambered as shown in FIGURES 2 and 4 and are secured to the disks 10 and 12 so that the disks and blades rotate as a unit. It is to be noted that the blades 16 are particularly located at a radial distance from the center of rotation so that they receive substantially radial flowing air only so the flow is two-dimensional, thus reducing the losses. Because of the radial two-dimensional flow, blades 16 are of constant camber untwisted cross-section resulting in inexpensive manufacturing costs. It is to be noted also that blades 16 are so arranged that their leading edges are at constant radius in the spanwise direction or parallel to the center of rotation. The result is that the rotational speed of any spanwise cross-section is constant. This is desirable because the total pressure relative to the rotor remains constant across the span of the blades and the ability to put work into the airflow through the compressor is constant because the rotational speed of any radial point in the blade is constant. The high camber (over 30°) of the rotor blades 16 provides high strength and high work input at moderate rotational speed (less than 1500 feet per second). This provides higher pressure ratio at a given rotational speed as compared to a conventional compressor.

The compressor described is next provided with a rotating diffuser in combination with the two-dimensional flow by extending disks 10 and 12 radially beyond the periphery of blades 16 to form a rotating vaneless diffuser passage 17. As in any diffusing flow passage, boundary layer fluid becomes a problem and the present invention has a built-in boundary layer control by providing diffuser passage 17 with rotating walls. It can be seen that by rotation of diffuser passage 17, the rotating side walls of the disks 10 and 12 energize the boundary layer due to the centrifugal field on any particle, since all particles will have a tangential velocity equal to or greater than the rotational velocity of the rotor disks. The vaneless rotating diffuser passage 17 which diffuses the flow from a Mach number of about 2.5 down to about 1.6 is followed by a vaneless diffuser 18 radially spaced from the diffuser passage 17 for further diffusion and resulting pressure rise of the air from the rotor. Of course vanes 18 may be split to perform their function in two steps as shown and described in FIGURE 5 of said co-pending application. Any suitable collecting means such as scroll 19 may be employed to carry the pressurized air to a point of use by connecting it to the vaneless diffuser. Since there is a significant static pressure rise accomplished between the leading edge of the vaneless diffuser 18 and the collecting means 19, a boundary layer problem may be encountered. If so, control may be obtained by suitable boundary layer suction or blowing means 20 as shown in the walls of the vaneless diffuser or the walls may be porous surfaces, both means of boundary layer control being well known.

As shown in FIGURE 2, off-design operation may require variable inlet guide vanes and the outlet or vaneless diffuser vanes 18 may also be made variable or adjustable in the same manner. Vanes 18 being variable or adjustable may be freely rotated as shown dotted or rotation

in the opposite direction would leave a vaneless portion before the collector 19.

With the construction just described, sealing problems are also minimized. The whole rotating structure is surrounded by closely spaced casing 21 and sealing means 22 are provided at a small radius on each disk between the disk and casing well inwardly of the disk outer periphery. This puts the seal at a lower rotational speed of the rotor compared to that of the rotor periphery. Because of the rotation of the air in the spaces between 12 and 21 and between 10 and 21, the pressure increases outwardly, and the pressure gradient across the seal 22 is lessened. With this construction no sealing is required at the periphery of the disk between rotating diffuser passage 17 and the vaned diffuser 18 because the small sealed volume between casing 21 and the disks, once charged with air, is an effective seal itself for the air flowing through the compressor.

The radial velocity of the air is controlled by contracting of the passage through the rotor blades and this may continue on through the rotating diffuser if desired as shown in FIGURE 6. This contraction is an essential feature for the satisfactory operation of a single wheel radial flow compressor of the instant invention because as the static pressure and density increase, the flow area must decrease in order to prevent backflow in the compressor which would create an intolerable situation. Because of the increase in the radial distance, the passage width may be decreased or contracted to maintain a constant radial velocity. The actual contraction also depends on the ability of the passage to maintain the flow without separation from the disk walls. In other words, it may be necessary to use more contraction in the vaneless diffuser and less in the rotor as shown in FIGURE 6.

The same concept of the two-dimensional radial flow only in combination with the rotating diffuser may be used in a dual entry compressor as shown in FIGURE 3. Similar reference numerals in the one hundred series are applied to similar but not identical parts and, with dual air entry, it is believed that the arrangement is self-explanatory. Balancing problems are simplified in the dual arrangement and the problem of getting the airflow through the "eye" of the impeller, that is, the circular opening 23 is reduced. In a single entry impeller this dimension 23 becomes a limiting one. The opening 23 must be of a certain diameter in order to let the airflow pass into the rotor at a low enough velocity so that losses are small. By using the dual entry, this limitation is greatly alleviated. It provides the rotor blades at a smaller diameter as well as allowing the location of the rotor discharge at a smaller radius. In very small high speed compressors this can be significant in the overall efficiency of the machine.

FIGURE 4 illustrates partially the blade and vane layout of the dual entry arrangement showing the outlet guide vanes 18 pivotable as previously described although it is to be understood that this adjustable feature may or may not be used as necessary.

A further modification that might be employed in order to overcome boundary layer problems in vaned diffuser 18 is illustrated in FIGURE 5. As previously explained in connection with FIGURE 1, it may be necessary to provide suction boundary layer control means in vaned diffuser 18 for the reasons given. If suction boundary layer control is used as opposed to boundary layer blowing it will be apparent that the air removed is from the cycle itself. The result of bleeding air from the cycle is lost energy. Modification of FIGURE 1 to the structure of FIGURE 5 provides a freely rotatable vaned diffuser 24 that may rotate about the casing 21 in any suitable manner as diagrammatically illustrated. This achieves a boundary layer control without bleeding the air from the cycle. Again, rotation provides means for energizing the boundary layer in the vaned diffuser by centrifugal force on the boundary layer. Of course, the rotating diffuser 24

might be used to provide shaft power by connecting it to a suitable load.

In the event that the required diffusion is not obtainable in the rotating diffuser passage 17 and vaned diffuser 18 of FIGURES 1 and 3, a feature of the co-pending application may be used in combination with the structure herein as shown in FIGURE 6. In this figure, wherein the two hundred series is used to identify similar but not identical parts, a separate stationary vaneless diffusing chamber 25 as shown is provided between vane diffuser 218 and vaneless diffuser passage 17. It will be understood that it could be rotating as shown in FIGURE 7, if desired. Thus, FIGURES 5 and 7 may be the modified outer end of FIGURES 1 and 3. It is desirable that this chamber 25 have a radial depth of substantially ten percent of the radial distance from the center of rotation of the shaft to the periphery of the disks 10 and 12 in order to provide satisfactory flow characteristics by avoiding excessive friction loss on the walls due to boundary layer build-up. Additionally, rotation, as noted above, reduces the boundary layer problem. As previously stated, by varying the contraction, it may be desirable to provide more compression in the vaneless diffuser and less in the blade area and this construction is shown in FIGURE 6. Similarly, it will be understood that vaneless chamber 25 may be used in the freely rotatable diffuser of FIGURE 5 merely by shortening vanes 18 as shown at 218 in FIGURE 7. Thus, the diffusing chamber 25 with vaned diffuser 218 as shown in FIGURE 7 is made freely rotatable like that of FIGURE 5.

It can be seen that the advantages of the instant invention provide for all the advantages of the co-pending application of a two-dimensional fluid machine such as a compressor or pump flow in combination with a rotating diffuser resulting in a simple single stage construction to obtain pressure ratios as high as 12 to 1 in a design replacing the multi-stage device of the co-pending application. Both centrifugal pressure rise and the usual aerodynamic pressure rise are present with the rotating diffuser in combination with the two-dimensional flow to obtain this very high pressure ratio in a single stage machine. The sealing and tip leakage problems are reduced and boundary layer in the diffusing area is centrifuged and energized in the compressor.

While there have been described preferred forms of the invention, obviously many modifications and variations are possible in light of the above teachings. It is therefore to be understood that within the scope of the appended claims, the invention may be practiced otherwise than as specifically described.

Having thus described the invention, what is claimed as novel and desired to be secured by Letters Patent of the United States is:

1. A radial outflow compressor comprising:
 - a rotor disk,
 - a shaft carrying said disk for rotation thereon,
 - second disks concentric with and axially spaced from said rotor disk one on each side thereof,
 - said second disks being spaced radially from said shaft and formed to define with said rotor disk an unobstructed curved inlet air passage between the rotor disk and each second disk for dual entry of air to flow radially outward between the disks,
 - seriate means arranged radially outward including highly cambered compressor blades secured to all said disks at a radial distance from said shaft so that the blades receive substantially radial flowing air only,
 - said air passage contracting through said compressor blades,
 - said disks and blades rotating as a unit in the direction of camber of the blades,
 - a casing surrounding said unit,
 - said second disks extending radially beyond the periph-

ery of said blades to form a rotating vaneless diffuser passage,
 a vaned diffuser radially spaced from said diffuser passage, and
 collector means following said vaned diffuser for collecting the air therefrom. 5

2. Apparatus as described in claim 1 wherein said compressor blades are of constant untwisted cross-section.

3. Apparatus as described in claim 1 including stationary guide vanes carried by said casing in each inlet air passage upstream of said compressor blades to receive radially flowing air only and impart tangential velocity thereto. 10

4. Apparatus as described in claim 1 having adjustable guide vanes carried by said casing in each inlet air passage upstream of said compressor blades to receive radially flowing air only and impart tangential velocity thereto, and 15

seal means on each second disk between the disk and casing and disposed radially inward of the periphery of the disks. 20

5. Apparatus as described in claim 1 wherein the vanes in said vaned diffuser are adjustable.

6. Apparatus as described in claim 1 wherein said vaned diffuser is freely rotatable about said shaft. 25

7. Apparatus as described in claim 1 wherein a vaneless diffusing chamber is provided between the periphery of the disks and said vaned diffuser,
 said chamber having a radial depth of substantially 10% of the radial distance from the shaft center to the periphery of said disks. 30

8. Apparatus as described in claim 7 wherein said vaneless diffusing chamber and said vaned diffuser are freely rotatable about said shaft.

9. A radial outflow compressor for handling fluid flow at supersonic velocity and converting the same to a high static pressure, said compressor comprising, 35

a pair of axially spaced rotor discs, rotatable in a predetermined direction about a common axis,
 a plurality of blades, highly cambered in the direction of disc rotation and extending between said discs, 40
 and disposed generally radially of the rotor axis, said blades being spaced from and concentric of said axis, the opposed surfaces of said discs, from the inward to the outward edges of the blades, converging toward each other to form an annular rotating compressor section through which the fluid flows at supersonic velocity in normal operation, 45
 means defining an inlet fluid flow path to said blades, characterized in that these means define a flow path which is essentially normal to said common axis and directs fluid flow outwardly from said axis toward the inner edges of said blade prior to engagement therewith, 50

said discs extending radially outwardly of said blades to form a rotating diffuser for converting supersonic flow to a substantially reduced velocity,
 a stationary, annular diffuser concentric of and formed as a continuation of said rotating diffuser, means for providing a fluid seal between said stationary and rotating diffusers, 5
 means for collecting pressurized fluid from said stationary diffuser,
 whereby upon rotation of said discs in the direction of blade camber, fluid flowing from the inlet, through the compressor section, to the rotating and stationary diffusers, is pressurized to a high level.

10. A radial outflow compressor as in claim 9 wherein: the inner and outer edges of the highly cambered blades are parallel to said common axis, and
 said blades have an essentially constant cambered section from one disk to the other.

11. A radial outflow fluid machine as in claim 10 wherein: 10
 the stationary diffuser comprises a pair of annular spaced disks, generally aligned with said rotor disks and a plurality of blades extending between said disks and disposed generally tangentially relative to said common axis.

12. A radial outflow fluid machine as in claim 9 wherein inlet guide vanes are disposed in said inlet passageway means to provide a controlled tangential direction to the fluid passing radially outwardly through said inlet.

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