COMPRESSOR EMPLOYING RADIAL DIFFUSION

Filed Sept. 28, 1949 3 Sheets-Sheet 1

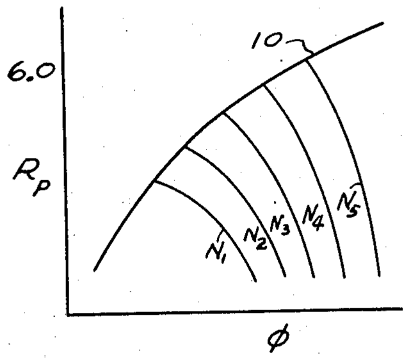

FIG. 1

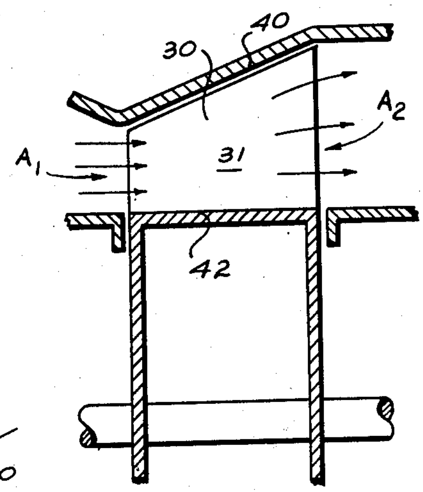

FIG. 3

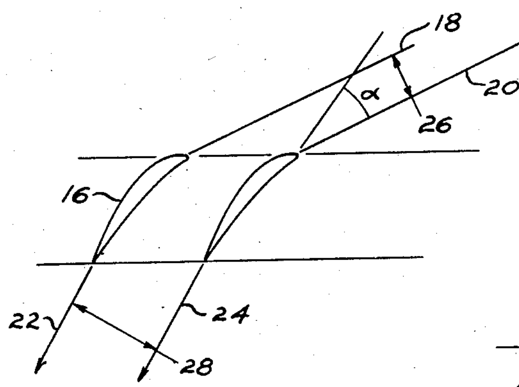

FIG. 2

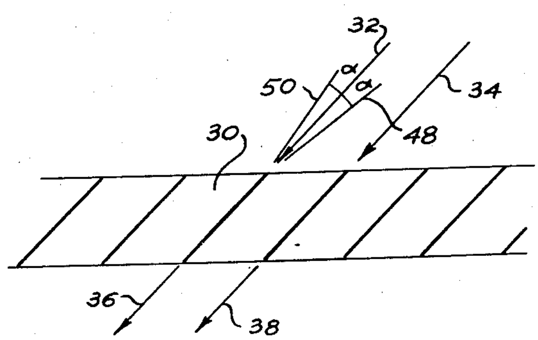

FIG. 4

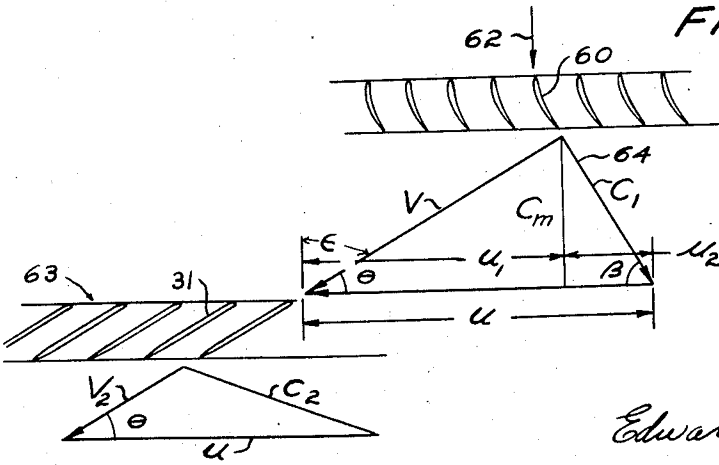

FIG. 5

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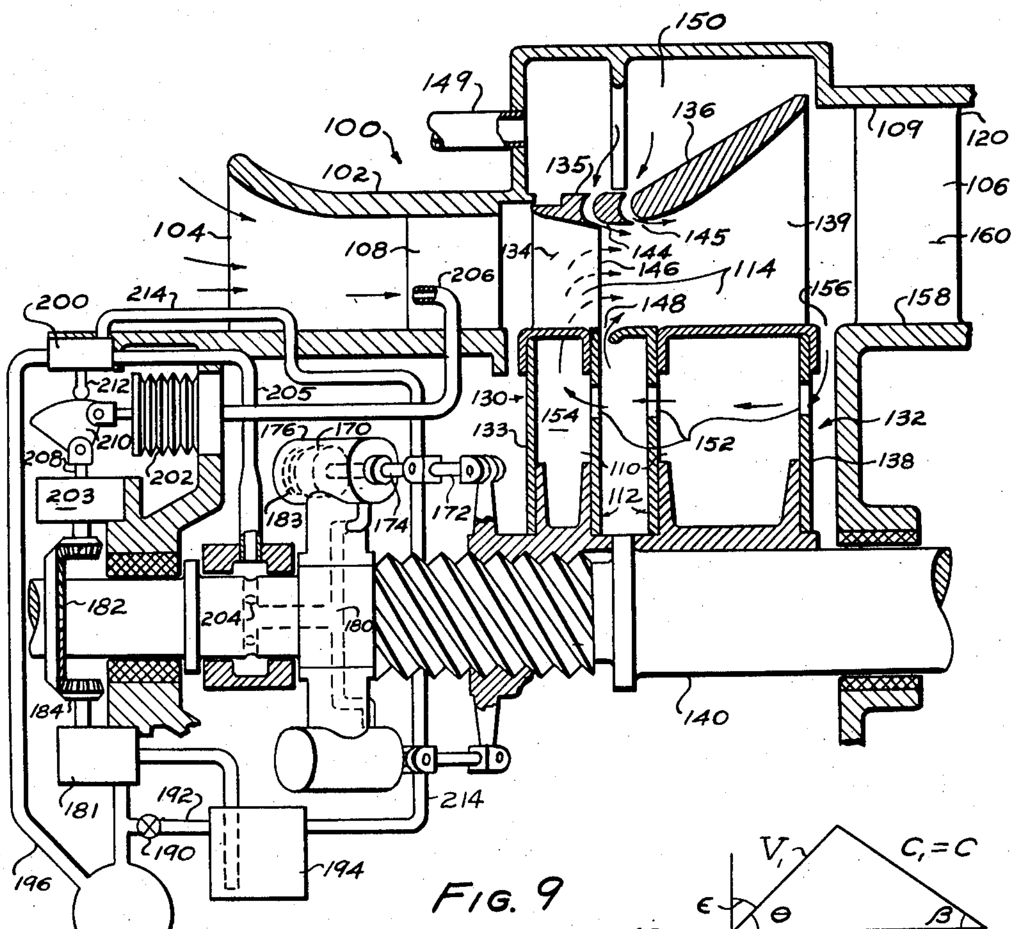

FIG. 9

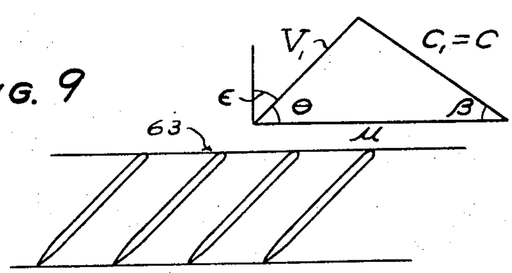

FIG. 6

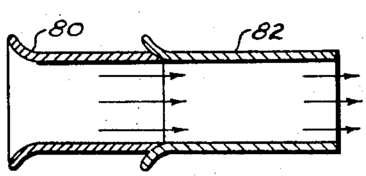

FIG. 7

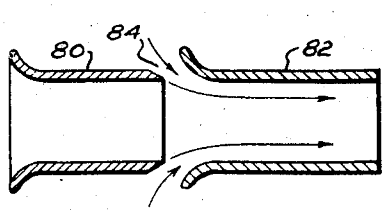

FIG. 8

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COMPRESSOR EMPLOYING RADIAL DIFFUSION

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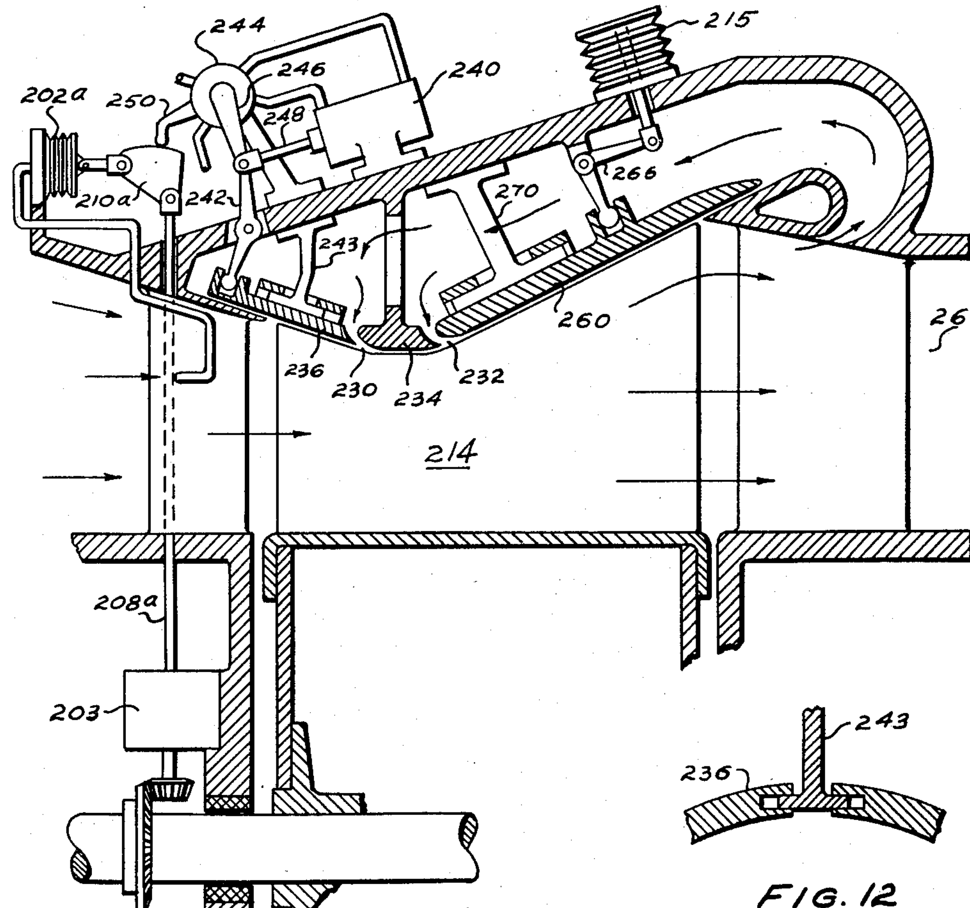

FIG. 11

FIG. 12

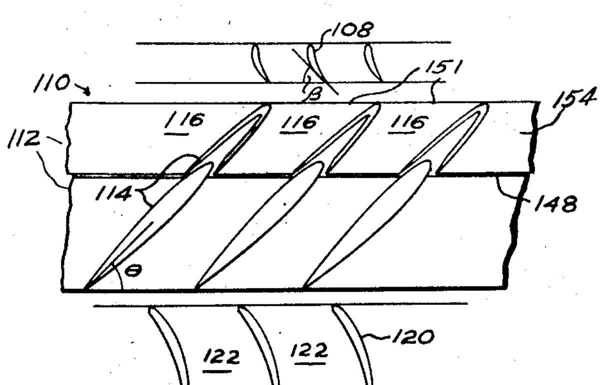

FIG. 10

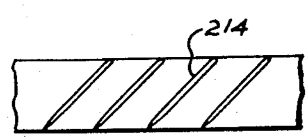

FIG. 13

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United States Patent Office 2,702,157
Patented Feb. 15, 1955

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2,702,157

COMPRESSOR EMPLOYING RADIAL DIFFUSION

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Application September 28, 1949, Serial No. 118,399

10 Claims. (Cl. 230—114)

This invention relates to compressors, particularly compressors of the axial flow type.

An object of the invention is to provide a compressor comprising a ring of inducer blades and a radial diffusion rotor with blade angles which are proper relative to said inducer blades.

Another object is to provide a compressor adapted for efficient operation over a range of supersonic velocities.

Still another object is to provide means of controlling the cross sectional areas of the rotor flow passages.

Still another object is to provide means responsive to the relative flow velocity for adjusting the effective flow cross sectional areas of the rotor flow passages.

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

The above objects are accomplished by the means illustrated in the accompanying drawings in which—

Fig. 1 illustrates the performance curves for an axial flow compressor;

Fig. 2 is a fragmentary development of rotor blading shown with the flow vectors;

Fig. 3 is a fragmentary axial section of a radial diffusion rotor;

Fig. 4 is a fragmentary development of the blading of the rotor of Fig. 3;

Fig. 5 is a vector diagram of a flow relative to the inducer and rotor blades of a radial diffusion compressor;

Fig. 6 is a fragmentary development of rotor blading shown in relation to a vector diagram;

Fig. 7 is an axial section of a flow nozzle and tube to illustrate the principle of controlling the effective flow cross sectional area;

Fig. 8 shows the sections of the nozzle and tube displaced axially;

Fig. 9 is a fragmentary axial section through a compressor according to the invention;

Fig. 10 is a fragmentary development of the compressor blading;

Fig. 11 is a fragmentary axial section through an alternate form of a compressor of the invention; and

Fig. 12 is a fragmentary section through a T-lug and the adjacent structure; and

Fig. 13 is a fragmentary development of the rotor blading of the compressor of Fig. 11.

A set of typical curves for an axial flow compressor is shown in Fig. 1 where R_p indicates the compression ratio and ϕ is the quantity of flow per revolution, commonly called the flow parameter. The curves N_1 to N_5 correspond to different rates of rotation. A characteristic of this type of compressor is the surge line **10** representing the lower value of the flow parameter at which surging begins. This value of ϕ corresponds to the burble angle of attack of the blades. In a compressor the angle of attack for burbling of the flow will not exceed about 10° from zero since there is little if any induced angle with the blade bounded at its root and tip by the hub and case walls respectively. Thus either the surge line or the maximum angle of attack of the blades defines one of the boundaries for efficient flow.

In the present invention this boundary is widened by the nature of the flow through the rotor.

In the contemporary axial flow compressor the compression produced in a rotor is obtained by the expansion of the flow peripherally by the blades. Thus in Fig. 2 the entering tube of flow passing between two rotor blades **16** is defined by the streamlines **18** and **20**. The leaving tube is defined by streamlines **22** and **24** and it is clear that the flow has been expanded since the width **28** is greater than the width **26**. According to Bernoullis equation the pressure will rise as the tube of flow is expanded. The process described represents peripheral diffusion or expansion of the flow. It depends on the angle of attack of the blades relative to the streamlines **18** and **20** to make **28** wider than **26**.

In the compressor of this invention the diffusion is radial as shown in Figs. 3 and 4. The cross sectional area A_1 of the inlet of each passage **30** is smaller than the exit area A_2 of each passage. Hence in passing through the rotor passages defined by blades **31** the flow is slowed and the static pressure rises. It is to be noted that the streamlines **32** and **34** are parallel to the blades at entering and the streamlines **36** and **38** are also parallel at leaving. In other words the angle of attack of the blades relative to the flow is not depended upon to provide the flow expansion, that is diffusion. This is due to the diverging of the peripheral walls, namely of the case **40** and the hub **42**. This process of pressure rise is herein called radial diffusion.

In Fig. 2 when the flow direction **20** (or **18**) is parallel to the flow direction **22** (or **24**) there is no pressure rise because the flow cross section is the same for the front and rear of the rotor. On the other hand in Fig. 4 the flow may depart to either plus or minus and there will still be an expansion of the flow since the expansion is built-in. Thus if the flow approaches along the direction of line **48** the diffusion is the sum of both peripheral and radial diffusion. If the flow approaches along **50** the radial diffusion is reduced but only by a small fraction since the ratio of A_2 to A_1 is large enough to preserve a net diffusion. Hence the radial diffusion rotor can operate over about twice the range of angles of attack of the peripheral diffusion rotor or over a far wider range of flow parameters ϕ.

The radial diffusion rotor develops about all of its pressure rise within the rotor at the design condition, that is when the flow is parallel to the blades **31**. This has an unique advantage as will be discussed later.

To accomplish the maximum pressure rise in the rotor, it must be operated at the highest possible tip speed which may be higher than the velocity of sound in the pumped fluid. In many compressor applications the velocity of the fluid relative to the blades should not exceed the velocity of sound in the fluid. To retain the high tip speed and subsonic relative fluid speed inducer blades **60** are employed to direct the entering flow with a peripheral component in the direction of rotation as illustrated in Fig. 5. The inducer blades deflect the flow **62** to the direction **64**. The velocity vector of this flow is commonly called C_1. The relative flow component due to the peripheral rotation of the rotor is u. The resultant fluid velocity relative to the blade **31** of the rotor **63** is V. The angle between the plane of rotation and vector C_1, is β and the angle between the vector V and the plane of rotation is θ, the pitch angle of the blade. The angle β is herein called the exit angle and is defined by the tangent to the mean camber line of the inducer or stator blade section at its trailing end. The exit angle is always the smaller angle between the tangent and the plane of rotation.

The pitch angle θ is always the smaller angle made with the plane of rotation and is positive for the blade nose directed in the direction of rotation of the rotor.

It may now be shown that there are critical angular relations between θ and β to achieve not only maximum pressure and efficiency, but to achieve even a significant compression.

The static pressure rise in passing through the rotor passages is according to the Bernoulli equation:

$$P_r=\frac{\rho}{2}V_1^2-\rho\frac{V_2^2}{2} \qquad (1)$$

and the velocity pressure added by the rotor is

$$\frac{\rho}{2}(C_2^2-C_1^2)$$

Then the total pressure rise contributed by the rotor can be written as

$$P=\frac{\rho}{2}(V_1^2-V_2^2)+(C_2^2-C_1^2) \quad (2)$$

where

ρ=mass density of the fluid
V_2=relative leaving velocity
C_2=absolute leaving velocity

According to trigonometry the relative velocities may be expressed as

$$V_1^2=u^2+C_1^2-2uC_1 \cos \theta \quad (3)$$

$$V_2=u^2+C_2^2-2uC_2 \cos \theta \quad (4)$$

If these are substituted into Equation 2 the pressure rise is expressed as

$$P=\frac{\rho}{2}[u(C_2-C_1) \cos \theta] \quad (5)$$

An examination of this equation shows that the pressure rise will be zero for θ equal to 90° since the cosine then will be zero. The pressure rise will be high when θ is substantially less than 90°. For very small flow through the rotor the value of θ could approach zero. However since the quantity of flow is always important the angle θ must be substantial in magnitude.

At the inlet of a compressor there is static pressure drop, as follows from Bernoulli's equation, to create the inflow velocity. Thus the flow through the stator or inducer vanes is accompanied by a static pressure drop.

Since the inducer vanes create velocity at the expense of static pressure, the amount of peripheral velocity supplied by the inducer vanes should be less than the peripheral component of V in order that the static pressure rise supplied in the rotor passages by the rotor will be greater than that lost by the action of the inducer vanes. Thus as in Fig. 5 the peripheral component u_1 of V is substantially greater than u_2 the peripheral component of C_1. This condition in the radial diffusion rotor is also satisfied if the blade angle θ is smaller than the exit or inducer vane angle β. Thus the blades of the rotor are to be set at a flatter exit angle than the inducer or stator vanes relative to the plane of rotation.

The conditions necessary for operating the compressor with the fluid velocity at or above supersonic speed may now be considered. Fig. 6 represents the rotor 63 of Fig. 5 but now $C_1=C$ the velocity of sound in the fluid and u has a supersonic value. The velocity achieved in the passages between the inducer blades is due to the upstream pressure and has a limiting value of the velocity of sound in the fluid at the conditions prevailing in the fluid at the exit of the inducer passages. Then for a given maximum value of u, the maximum value for angle θ is given for $C_1=C$. Any value of C_1 less than C will decrease the angle θ. Therefore if the vector is to approach the rotor blade substantially parallel thereto, the maximum angle of the blade relative to the plane of rotation is θ. With respect to the direction of the axis of rotation the angle is $\epsilon=90-\theta$. It is also clear that any reduction in the value of C_1 will make θ still less than β.

Thus for both subsonic and supersonic velocities θ is to be less than β.

In order for the compressor to provide a pressure rise in the rotor passages for a supersonic flow entering therein, each passage must converge and then diverge. The converging and diverging sections define a passage throat positioned between them. The throat cross sectional area should have such a magnitude that the supersonic flow should be reduced to sonic velocity c. Then if there is back pressure on the exit of the diverging section, a shock wave will ensue which will reduce the velocity aft of the wave to less than sonic and the diverging section can expand the flow to higher static pressures—as is well known. However for each velocity of approach there should be a different throat area. The higher the velocity of approach the smaller the throat area should be. A variable throat area presents great difficulties if done mechanically. This invention discloses a means of accomplishing the proper variation by simple and novel means.

The principle of the control of the throat area is illustrated in Figs. 7 and 8. The primary flow passes through the inlet nozzle **80** whose exit area is the same as the cross sectional area of the tube **82**. When the tubes are displaced axially they define therebetween the annular slot **84**. If a secondary flow is introduced through slot **84** it will occupy an annular portion of the flow passage of **82** thereby reducing the magnitude of the cross sectional area otherwise available to the primary flow. The extent to which the cross sectional area is in effect reduced is determined by the width of the slot **84**. This principle is employed in the compressors.

Figs. 9 to 13 illustrate two forms of the invention incorporating radial diffusion rotor passages and inducer and stator blades. The passages have greater cross sectional areas at exit than at inlet and the pitch angles θ of the rotor blades are smaller than the inducer angles β. The rotors also include the means of controlling the effective flow area of the rotor passages.

The compressor in Fig. 9 is indicated generally by **100**. The case **102** admits fluid at the annular inlet **104** and emits it at the annular exit **106**. Inducer vanes **108** direct the incoming flow at the proper angle β (see Figs. 5 and 6) relative to the plane of rotation of rotor **110** comprised of the hub **112** and the blades **114** which define the rotor passages **116** therebetween as shown in Figs. 9 and 10.

The stator blades **120** are positioned downstream from the rotor and define the stator passages **122** therebetween.

The rotor is formed of a front section **130** and an aft section **132**. The front hub section **133** carries the front segments **134** of the blades and at their tips the front segment **135** of the shroud **136**. The aft hub section **138** carries the aft segments of the blades **139**.

As shown particularly in Fig. 10 the front segments of the blades are hollow and the aft segment is nested in the front segment.

The front section of the hub is threaded on the shaft **140** so that a rotary movement of the hub relative to the shaft moves the hub section axially along the shaft. The hub aft section is fixed to the shaft. The two sections of the rotor are adjustably spaced so that when they are separated they define a shroud slot **144**, the slots **146** in each blade and the hub slot **148** between the front and rear segments on each side of the blade.

The annular slots **144** and **145** in the shroud are supplied with fluid by the annular compartment **150** which in turn receives its fluid from a suitable source.

The blade and hub slots receive fluid from the openings **152** in the hub. The root ends of the blades are open to pass fluid from the region **154** into the blade interiors.

Slots **146** and **148** are supplied fluid via annular slot **156**. Slot **144** is supplied via **150** by the tube **149** from a suitable source which may be the discharge end of the compressor or even from a source of somewhat higher static pressure than that at slot **144** within the rotor passage.

The widths of the slots are controlled automatically in response to the velocity of flow entering the rotor passages.

Considering first the response to the velocity of the fluid approaching the rotor, the higher the fluid velocity the further open are the slots so that the slot flows reduce the cross sectional area available to the flow entering at the inlets **151** to the rotor passages **116**. As this inlet velocity declines the slots are reduced in width progressively.

The slot reduction is accomplished by rotating the section **130** on the threaded portion of the shaft. The threads have the same pitch as the blades so that the rotation causes the front segments of the blades to receive the rear segments further thereinto which reduces the widths of the slots, as will be obvious from Fig. 10.

The front section of the rotor is moved by two cylinder devices each comprised of a piston **170** connected to the rotor by the link **172** and piston rod **174**. The cylinders are fixed to shaft **140** and rotate therewith. A case **176** encloses each piston for its operation by control fluid pressure from duct **180**. When the fluid pressure increases the piston moves the rotor section **130** to increase the widths of the slots. When the control fluid pressure is decreased the spring **183** in the cylinder pushes the piston and decreases the widths of the slots.

The control fluid pressure for the cylinder is supplied by the pump **181** preferably driven from the compressor shaft **140** by gears **182** and **184**. The fluid goes from the pump to the accumulator **186** where the check valve **190** maintains a constant pressure by bypassing excess pressure through the bypass tube **192** to the reservoir **194**. From the accumulator the tube **196** carries the fluid to the valve **200** controlled by the sylphon **202** and the governor **203**. From the valve **200** the fluid goes via tube **205** to the ducts **204** of shaft **140** and thence to ducts **180** serving the cylinders **176**.

The sylphon responds to fluid velocity pressure received through tube **206** whose inlet is positioned just upstream from the rotor. As the main flow velocity increases in the compressor more control fluid pressure is supplied to the piston to increase the widths of the slots. As the main flow velocity declines less control fluid pressure is applied to the piston because valve **200** bypasses fluid to the reservoir via tube **214**.

The control action by the sylphon is superimposed on the control action supplied by the governor **203** which moves rod **208** vertically. This rod carries the cam **210** at its upper end on a pivot. The valve stem **212** rests on cam **210**. A vertical movement of rod **208** gives the valve stem a vertical displacement which is altered by the rotation of the cam about its pivot by the sylphon **202**.

The combined action of the governor and the sylphon is a measure of the main flow velocity relative to the rotor since this velocity is the resultant of the axial velocity ahead of the rotor and the peripheral velocity of the blades.

In still another form of the invention shown in Figs. 11–13 a forward slot in the case is controlled in response to the increase in velocity of the fluid approaching the rotor blades **214** while a rear slot is controlled by the back pressure in the compressor downstream from the rotor or rotor front face.

In Fig. 11 the front slot is **230** and the rear slot is **232**. The front slot is annular and is defined between the fixed part **234** of the case and the annular part **236** movable axially to vary the slot width. Part **236** is moved by the controllable jack **240** operating through the bell-crank **242** whose lower arm is articulated to the movable part **236**.

The part **236** is slidably supported by the T-lugs **243** whose heads fit slidably in the slots in the part as shown in Figs. 11 and 12.

The movement of jack **240** is controlled in response to the combined action of the sylphon and the governor as described earlier in connection with Fig. 9 except that in Fig. 11 valve **244** is a follow-up valve having its follow-up arm **246** articulated to the piston rod **248**. The governor **203** moves rod **208a** vertically and sylphon **202a** moves the cam **210a** so that the combination of sylphon and governor control the valve arm **250**. Thus the piston rod **248** of the jack and therewith the part **236** are correctly positioned as a junction of the axial velocity of the fluid and the rate of rotation of the blades of the rotor.

The rear slot **232** is defined between the case part **234** and the slidable member **260** supported by T-lugs **270** in a manner similar to the support provided for slidable part **236**. The sliding of member **260** is controlled by a plurality of sylphons **215** operating bell cranks **266** whose lower arms are articulated to the member **260**. The sylphons and their respective bell cranks are positioned about the periphery of the compressor.

As the pressure of the pumped fluid increases in the annular discharge duct **268** the top of the sylphon is moved outward which moves the sliding member **260** rearward increasing the slot width.

As the back pressure increases it is desirable to have the slot **232** widened to discharge ample fluid along the surface of member **260** to prevent the flow from separating from this surface.

Fig. 13 shows the blading of Fig. 11 in development. For high speed operation the blades are very thin with relatively sharp leading edges.

The provision of a shroud is of great importance since the pressure is a maximum along the tip of the blade due to the centrifugal pressure generated by the blades to keep the flow from separating from the outer wall of the passage. In the absence of a shroud there is a large tip leakage.

The outer wall whether it is that of a shroud or the case is preferably curved convexly with respect to the flow or the axis of rotation so that the angle of the expansion of the flow has desired values along the actual path of flow of a fluid particle. This path is curved relative to the case.

While I have illustrated specific forms of the invention, is is to be understood that variations may be made therein and that I intend to claim my invention broadly as indicated by the appended claims.

I claim:

1. In combination in a rotor adapted for the interchange of energy with a fluid, a rotor front section having a plurality of peripherally spaced blade front segments, each said segment having a spanwise extensive recessed interior, a rotor aft section having a plurality of peripherally spaced blade aft segments, means to adjustably support said sections in spaced tandem relation with each said aft segment positioned in a said interior in spaced relation with a said blade front segment to define a blade with a slot between said front and aft segments, said blades defining a plurality of rotor passages therebetween, each said slot leading out of the respective blade interior, means placing each said interior in communication with a source of fluid to supply flows through said slots into said passages, and means to move said rotor sections with an axial component of displacement relative to each other to vary the widths of said slots between respective front and aft segments.

2. In combination in a rotor adapted for the interchange of energy with a fluid, a rotor front section having a plurality of peripherally spaced blade front segments, each said segment having a spanwise extensive recessed interior, a rotor aft section having a plurality of peripherally spaced blade aft segments, means to adjustably support said sections in spaced tandem relation with each said aft segment positioned in a said interior in spaced relation with a said blade front segment to define a blade with a slot between said front and aft segments, said blades defining a plurality of rotor passages therebetween, each said slot leading out of the respective blade interior, means placing each said interior in communication with a source containing fluid which has passed through said rotor passages to supply flows through said slots into said passages, and means to move said front segments relative to their repective aft segments to vary the widths of said slots.

3. In combination in a rotor having a plurality of blades having hollow interiors and adapted for the interchange of energy with a fluid, means defining a slot of adjustable width in each said blade, each said slot extending spanwise in the surface of its respective blade in communication with said hollow interior, means to sense the velocity of the fluid flow relative to said blades, means responsive to said sensing means to adjust said slot widths, and means at an end of each said blade to induce a spanwise flow of fluid in said interior to cause a flow through each said slot.

4. In combination in a compressor, a case having a peripheral slot therein, a rotor mounted in said case for rotation about an axis, said rotor having a plurality of blades spaced peripherally thereabout to define a plurality of rotor passages for a main flow of fluid therethrough, said slot being positioned between the front and aft ends of said blades, each said passage having an exit greater in radial depth and cross sectional area than the inlet thereof, a source of fluid in communication with said slot, and controllable means to adjust said slot to introduce a controlled flow of fluid from said source into said passages through said slot to effectively adjust the cross sectional area of each said passage available to said main fluid flow therethrough.

5. In combination in a rotor adapted for the interchange of energy with a main flow of fluid at supersonic relative speed, a plurality of blades spaced peripherally on said rotor to define a plurality of rotor passages to conduct said main flow through said rotor, each said passage having an exit greater in radial depth and cross sectional area than the inlet thereof adapting each said passage to the conversion of supersonic velocity to static pressure, a source of fluid of less total energy than said main flow of fluid, and controllable means to introduce a flow of fluid from said source into each said passage to effectively reduce each passage cross sectional area available to said main flow.

6. In combination in an axial flow rotor for use in a compressor, a hub means, a plurality of blades mounted on said means in peripherally spaced relation defining flow passages therebetween adapted for a supersonic flow of

fluid therethrough, each said blade having spaced walls defining a spanwise slot in communication with the blade interior, means to adjust said walls of each said slot relative to each other to vary the width of each slot, and means to provide a flow of fluid out each said slot to vary the flow cross sectional areas between blades available to a flow of fluid entering each said passage ahead of a said slot.

7. In combination in an axial flow rotor, a hub means, a plurality of blade nose segments mounted on said means in peripherally spaced relation, each said segment having a spanwise recess in its aft end, a plurality of blade aft segments mounted on said means in peripherally spaced relation with each said blade aft segment nested in a said recess defining with a said nose segment a blade having a slot leading out of the blade, and means to relatively move said nose and aft segments of each blade to vary the width of said slot therebetween.

8. In combination in an axial flow compressor, a case, a rotor mounted in said case for rotation about an axis, said rotor having a plurality of peripherally spaced blades defining a plurality of rotor passages for pumping fluid, each said passage having an exit of greater cross sectional area and radial depth than the inlet thereof, said rotor blades being set at a positive pitch angle less than 90 degrees with respect to the plane of rotation of said rotor, a stage of stator blades positioned in said case adjacent the upstream side of said rotor, each said stator blade having an aft end directed in the direction of rotation of said rotor to direct fluid in said direction, each said stator blade defining an exit angle with said plane of rotation, each said rotor blade being set at a pitch angle smaller than said exit angle.

9. In combination in an axial flow compressor, a case, a rotor mounted in said case for rotation about an axis, said rotor having a plurality of peripherally spaced blades defining a plurality of rotor passages for pumping fluid, each said passage having an exit of greater cross sectional area and radial depth than the inlet thereof, said rotor blades being set at a positive pitch angle less than 90 degrees with respect to the plant of rotation of said rotor, a stage of stator blades positioned in said case adjacent the upstream side of said rotor, each said stator blade defining an exit angle with the plane of rotation larger than said pitch angle.

10. In combination in a bladed rotor adapted for rotation for the interchange of energy with a fluid, a rotatable hub means, a blade means comprising a plurality of blade elements mounted on said hub means to define a plurality of peripherally spaced blades and a plurality of rotor flow passages therebetween, a plurality of said elements comprising a set adjustably positioned in tandem relation with respect to another set comprised of a plurality of said elements, and means to adjust the relative axial positions of said sets to vary the effective cross sectional areas of said passages to control the flow through said passages.

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