

June 5, 1956

E. A. STALKER

2,749,025

COMPRESSORS

Original Filed Dec. 26, 1947

3 Sheets-Sheet 1

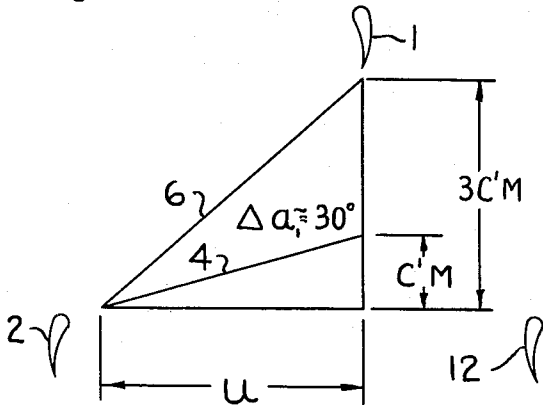


Fig. 1

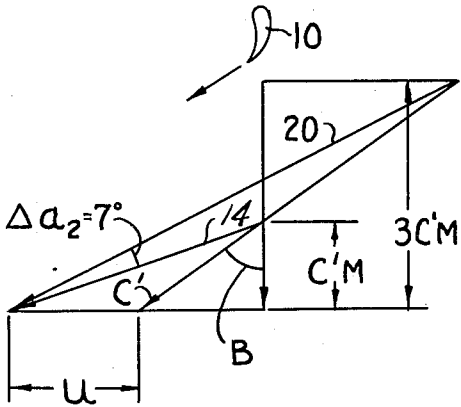


Fig. 2

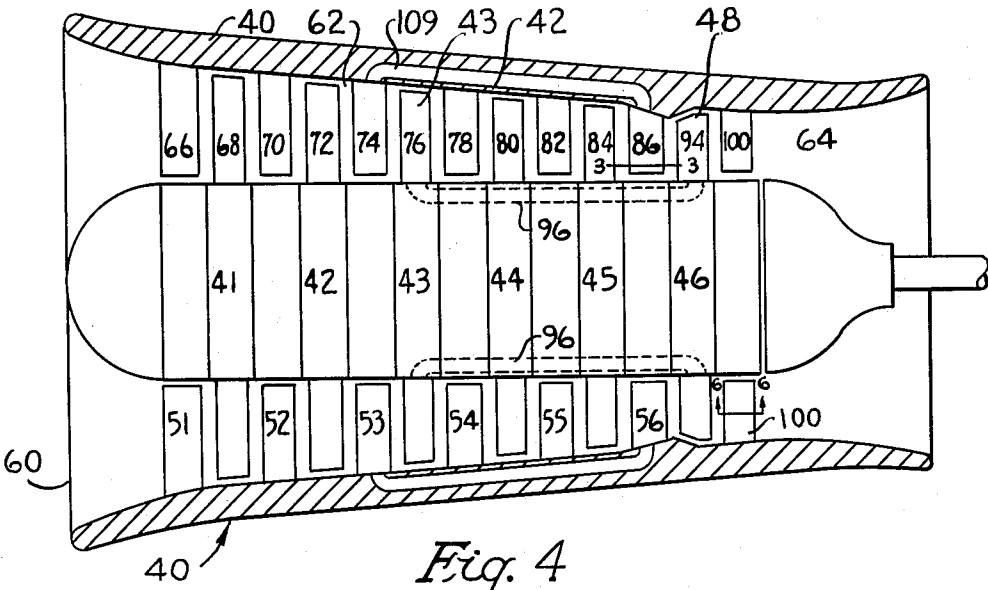


Fig. 4

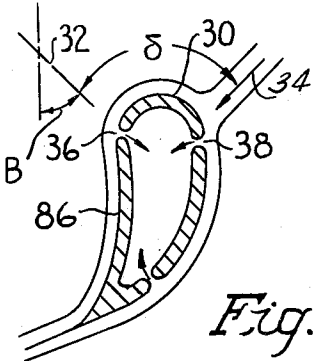


Fig. 3

INVENTOR.
Edward A. Stalker

June 5, 1956

E. A. STALKER

2,749,025

COMPRESSORS

Original Filed Dec. 26, 1947

3 Sheets-Sheet 2

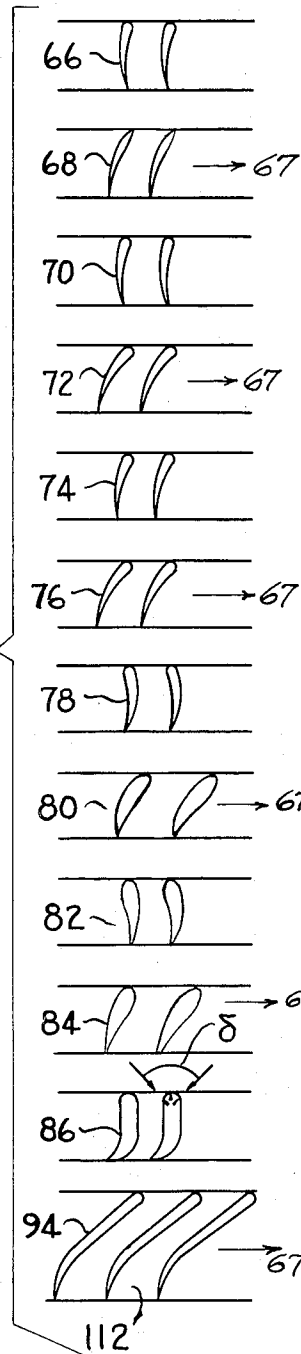


Fig. 5

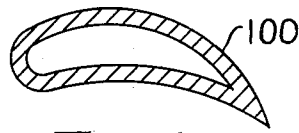


Fig. 6

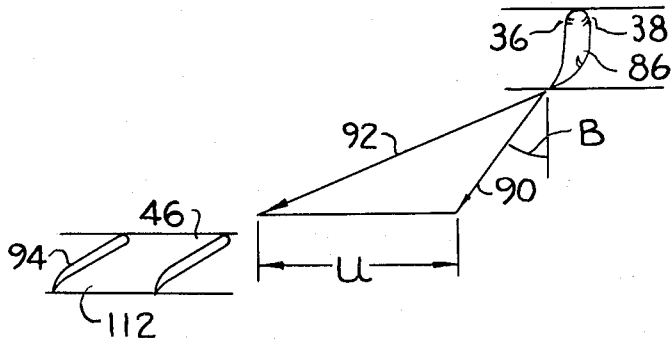


Fig. 7

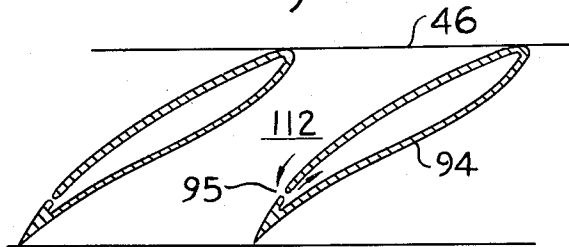


Fig. 8

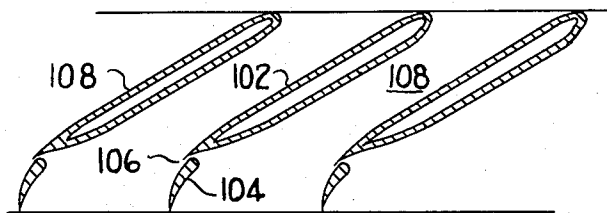


Fig. 9

INVENTOR.
Edward A. Stalker

June 5, 1956

E. A. STALKER
COMPRESSORS

2,749,025

Original Filed Dec. 26, 1947

3 Sheets-Sheet 3

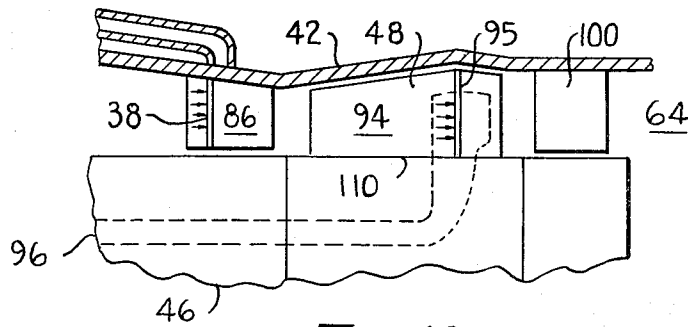


Fig 10

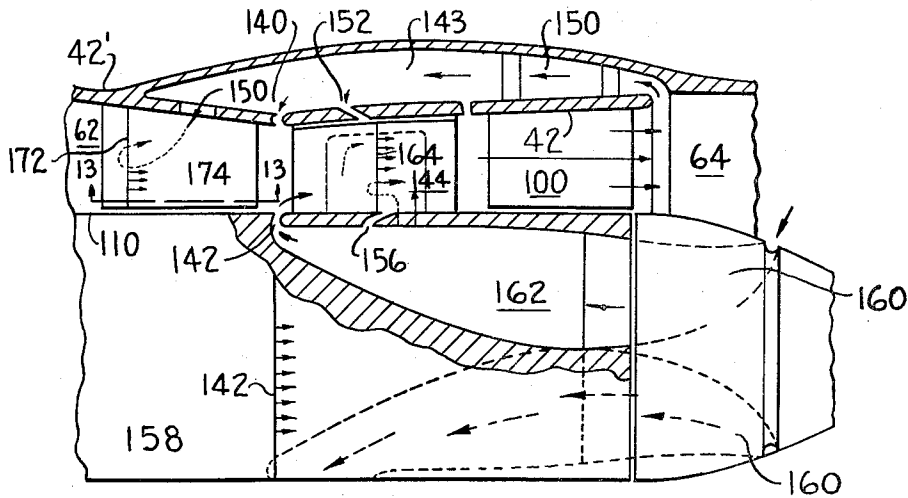


Fig.12

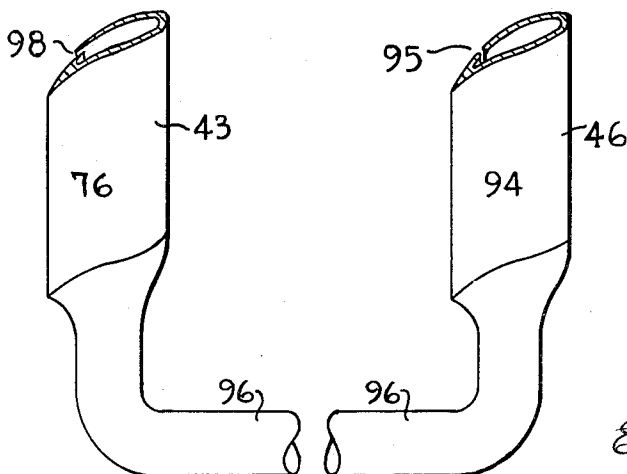


Fig. 11

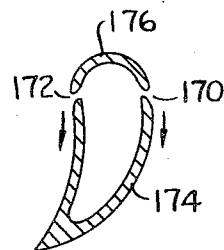


Fig. 13

INVENTOR.
Edward A. Stalker

1

2,749,025

COMPRESSORS

Edward A. Stalker, Bay City, Mich.

Original application December 26, 1947, Serial No. 794,018. Divided and this application June 2, 1952, Serial No. 291,252

3 Claims. (Cl. 230—122)

This invention relates to compressors and is a division of my pending application Serial No. 794,018, filed Dec. 26, 1947, entitled "Compressors."

An object of the invention is to provide a compressor which maintains its pressure and efficiency over a wider range of volume flow per revolution.

Another object is to provide blades of successively rounder and thicker noses in successive stages downstream to accommodate a wide range of angles of approach to the blade.

Still another object is to provide proper relations between stator and rotor blades to reduce the range of angles of approach which a blade requires.

Another object is to provide a combination of axial flow stages of one type with an axial flow stage of another type to ameliorate the great loss in efficiency at off-design conditions when the compressor has been designed for a high compression ratio.

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

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

Figure 1 shows a vector diagram for the air approaching a rotor blade;

Figure 2 shows a vector diagram for the air approaching a rotor blade at a flat angle;

Figure 3 is a chordwise section along the line 3—3 in Fig. 4.

Figure 4 is an axial section through an axial flow compressor according to this invention.

Figure 5 is a fragmentary diagrammatic development of some of the stages of the compressor of Fig. 4 with the blades shown solid although they are hollow in the machine;

Figure 6 is a section along line 6—6 in Fig. 4;

Figure 7 is a fragmentary development of the last stage of the compressor to show the vector relations;

Figure 8 is a fragmentary development of the last rotor showing the blades in section as they are in the machine;

Figure 9 is an alternate rotor construction to that of Fig. 8;

Figure 10 is an enlarged fragmentary axial section through a part of the last rotor and part of the case of the compressor of Fig. 4;

Figure 11 shows an isolated group of blades and a connecting duct, one blade having an induction slot and the other having a discharge slot;

Figure 12 is a fragmentary axial section through another compressor in which the rotor and stator blades have discharge slots and the other walls have slots for facilitating the entrance of a supersonic flow into the passages between blades; and

Figure 13 is a section along line 13—13 in Fig. 12.

When a multi-stage axial flow compressor is operating at a mass flow per revolution less than the optimum or design value with a back pressure that is relatively low the axial velocity in the downstream stages may be as much as three times the velocity which would prevail at the

2

optimum or design condition. This is so because the upstream stages do a certain amount of compressing at off design conditions and the lack of back pressure permits the flow compressed by the upstream stages to stream at greatly increased velocity through the later stages. This leads to a great change in the direction of the fluid approaching a later rotor or stator with respect to the direction for the optimum operating condition, reducing the angle of attack of the blades and their compressing ability.

For instance Figure 1 shows the vector diagram for a conventional axial flow compressor for a downstream stage when the compressor is operating under optimum condition, that is at about best efficiency and corresponding pressure ratio. In this instance the axial velocity for optimum operation is C_m' equal to a fraction of u the peripheral velocity. Under this condition the direction of the fluid leaving the stator blade 1 and approaching the rotor blade 2 is the vector 4. Now if the axial velocity is increased to 3 times C_m' the new direction is the vector 6 and the change in the angle of approach is $\Delta\alpha_1$ which is equal to about 30° . This is a greater range of angles of attack than a blade can accommodate.

Now consider a case as in Fig. 2 where the leaving velocity vector from blade 10 is C' directed at the positive angle B toward the rotor blade 12. The resultant vector is 14. If the axial component of C' is increased from C_m' as for Fig. 1 to $3C_m'$ the new resultant velocity vector is 20 whose peripheral component is much larger than that of vector 14. The peripheral component is not magnified as greatly as the axial since the new triangle of which 20 is the longer side is not symmetrical with respect to the triangle 14 $c'u$. The change in angle of approach to blade 12 is now $\Delta\alpha_2$ equal to about 7° . This is not only within the range of angles which the blade can accommodate but is also well within the range of angles of attack for best efficiency of the blade itself.

It is thus shown that deflecting the air toward the oncoming rotor blades reduces the range of approach angles of attack which the blade must accommodate when the compressor is operating at a pressure and speed substantially below optimum conditions provided the deflection through the angle B is accompanied by a rise in velocity.

The angle B for the vector representing the entering vector for a rotor (or stator) is positive when the vector has a peripheral component directed toward the concave face of the blade of the succeeding stage. Thus in Fig. 2, B is positive. Also in Fig. 3, B is positive since the vector 32 approaching the stator attacks the concave side of the blade 86.

The range of approach angles which can be accommodated by the downstream stages can also be extended to a considerable extent by making the noses of the blades successively thicker in successive stages in the downstream direction. Thus in Fig. 3 the nose 30 is substantially semicircular so that the relative flow will be able to flow about the nose without burbling when the approach vectors vary from vector 32 to vector 34 disposed angularly with respect to each other by the angle δ (delta). Thus the blade sections of the blades of downstream stages are to have noses defined by circular arcs of larger radii than the noses of blade sections of the upstream stages. The blade section shown at 86 in Fig. 5, for instance has a greater nose radius than that of the blade section shown at 82.

To further encourage the flow the nose is provided with the slots 36 and 38 (Figs. 3, 4, and 8) through which a flow may be inducted to control the boundary layer.

Since the fluid is compressed in successive stages the temperature rises along the compressor axis. Consequently the velocity of sound in the fluid increases in magnitude so that the velocity of the fluid relative to the blades

can be increased without precipitating a compressibility shock. In other words the local velocity on the blade surfaces can be higher on the downstream blades without reaching the critical Mach number of one.

Thickening the nose of the blades makes possible a wider range of angles δ (see Fig. 3) but increases the local velocity on the nose. However by taking advantage of the rise in temperature from stage to stage, the noses of the blades of successive stages may be thickened without the local Mach number exceeding the critical value.

Figs. 4 to 8 show a compressor incorporating the foregoing features.

In Fig. 4 the compressor is indicated generally by 40 comprised of the case 42, the rotors 41-46 and the stators 51-56. (See also Fig. 5.) Fluid enters the inlet 60 and is pumped through the annular or main flow passage 62 to the exit passage 64.

At the upstream end (Fig. 4) the stator 51 deflects the incoming air by means of the stator blades 66 in the direction of rotation 67 of rotor 41 composed of blades 68. The next stator 52 also deflects the fluid in the direction of rotation of rotor 42, but to a less extent, by blades 70. At the third stage the stator 53 deflects the fluid substantially axially toward the rotor 43. This stage is comprised of blades 74 and 76.

In the succeeding stages of Fig. 4 the stator blades deflect the flow with increasing peripheral velocity components against the direction of motion of the rotor blades.

The blades of the fourth stage are 78 and 80 and the blades of the fifth stage are 82 and 84. It is to be noted that in each of these stator stages (see Fig. 5) and in the sixth stator stage the stator blades are curved to give the flow a progressively greater peripheral component in successive downstream stages.

The stator blades 86 for instance in the sixth stage have tail portions directed substantially in the peripheral direction.

In Fig. 7 the velocity vector 90 leaving the blade 86 when combined with the peripheral velocity vector u of the rotor gives the velocity vector 92 acting relative to the rotor 46. The vector 90 makes the positive angle B with the axial direction and hence even for a great increase in axial velocity through the compressor the direction of 92 relative to the blades 94 of rotor 46 will change only a small amount.

The rotor blades 94, Fig. 8 are hollow and as shown in Figs. 4 and 11 each has its interior in communication by means of individual ducts 96 with the hollow blades 76 of the third stage. Since the fluid pressure is greater in the sixth stage than in the third stage fluid will enter the blades 94 through slot 95 and be discharged through the discharge slots 98 in blades 76. Thus the flow is induced to follow the curved portion of blade 94 making it possible to discharge the flow from the stage with a velocity direction perpendicular to the plane of rotation.

The last set of stators 100 (Fig. 4) takes out the peripheral component of velocity relative to the case 42 and directs the discharge of fluid axially along the passage 64.

As an alternate form the rotor may be formed as in Fig. 9. Here the blade is made in two parts, the fore part 102 and the aft part 104 spaced from the fore part to provide the slot 106. The flow through the slot provides a jet to control the boundary layer on the convex portion of the blade and induce the flow in the passage 108 between blades 102 to follow the blade surface.

The stators as shown in Fig. 4 are also interconnected by ducts such as 109 to provide for flows of fluid through the blade slots. This construction is similar to that shown in my U. S. Patent No. 2,344,835 issued March 21, 1944.

By providing the stator which gives a large positive angle B , the variation δ (Fig. 3) is kept small and consequently the blades 94 (Figs. 4 and 5) may be thin at

the nose and particularly efficient for high velocities of flow.

As shown in Figs. 4 and 10, particularly the latter, the case 42 diverges from the wall 110 of the rotor 48 so that each passage 112 between blades 94 is expanding in cross sectional area until the locality of the blade curvature is reached where the passage area is preferably made to contract slightly so that the flow about the curve is in a favorable pressure gradient. This facilitates an efficient flow about the curve.

There is also another advantage in the divergence of the hub and case walls. The increase in the cross sectional areas of the rotor passages in the downstream direction slows down the velocity of flow before the flow is turned by the blade. Hence the appearance of compressibility shock waves is delayed. That is, the peripheral tip speed of the blades can be higher before the shock wave appears in the passages between blades. This means that substantially greater pressure ratios can be obtained from a rotor.

The first shock waves appear at the leading edge of a blade but the critical shock wave which limits the mass flow through the rotor occurs in the passage downstream from the nose of the blade.

If the passages between blades begin to diverge radially opposite the blade noses, the radial expansion can compensate for the peripheral contraction due to the blade thickness. Hence there need not be a throat along the passages between blades or at least the throat may be placed far downstream from the inlet of each rotor passage. In this connection the blades may have substantially parallel sides as shown by blades 102 in Fig. 9.

The opposite sides of the blade sections such as blades 86 (Fig. 5) are substantially parallel along a substantial length between the nose portion and the aft portion. If it is desired to avoid supersonic velocities in the passages between such blades as 86 the hub and case walls can be made to diverge along the passages between stators as described for the rotor.

By making the last stage with thin blades and relatively sharp noses it can operate with very high fluid velocities without generating shock waves at the nose or in the passage. However in some applications the velocity may become supersonic in the last stage if the back pressure is reduced sufficiently when the rate of rotation of the rotor is near the optimum speed for the compressor as a whole. For this reason the type of rotor shown in the last stage is very advantageous since it can operate even at a supersonic velocity as has been disclosed in my application Serial No. 624,013 filed October 23, 1945, now Patent No. 2,648,493, entitled Compressors. Furthermore for a high performance compressor the last stage is preferably made to have a supersonic velocity of approach of the air at the optimum condition of operation. For such a compressor it is important that the angular range of the approach vector should be small to obtain the proper shock waves at the nose of the blades and within the rotor or stator passages. These are provided by this invention.

In an axial flow compressor if the pressure rise is great between inlet and exit for the design condition, then the machine will be much less efficient at a lower delivery, that is at a lower value of the mass of fluid delivered per revolution. The greater the pressure rise, the greater the drop in efficiency at an off-design delivery.

The compressor of this invention using the type of rotor 48 is provided to assuage this undesirable condition and places the axial flow compressor on a more favorable footing with respect to other compressors, such as for instance the centrifugal compressor, than heretofore existed.

When the fluid approaches the blades at supersonic values shock waves first appear at the leading edges of the blades and if the back pressure is substantial the shock

5

waves may occur ahead of the leading edges and the flow may refuse to enter the passages between the blades at high supersonic velocities. This difficulty can be overcome by discharge slots properly located with respect to the leading edges of the blades.

Figure 12 shows an alternate structure for the last rotor and the stator ahead of it. The balance of the compressor ahead of this stator would have a structure similar to that of Figs. 4 and 5.

The rotor blades 144 of the last rotor are encircled by the shroud ring and its leading edge forms the slot 140 with the case wall 42'. The slot 142 is formed in the peripheral wall of the hub 158.

Air for the case or outer wall slot 140 is bled from the passage 64 via the annular duct 150 formed in the case. The air is at a higher pressure in 64 than in the passage at the leading edge of blade 144 and hence can flow at a higher velocity from the slot 140 than the velocity of the local main flow.

Air is also supplied from duct 150 to the slot 152 positioned in the rotor passage 62 a substantial distance inward from the leading edge of blade 144.

Air is also supplied to the slot 142 and slot 156 from passage 64 via the annular ducts 160 and 162. Air also enters the hollow interior of blade 144 via 162 to serve the slot 164.

The discharge slots 170 and 172 of stator blade 174 is also served with air from duct 150. As shown in Fig. 13 this blade has well rounded nose 176 and the discharge slots located near the ends of the nose contour.

The passages 112 in the rotor between the blades in Fig. 12 are similar to those in Fig. 8 and are bounded by walls on four sides. The walls 110 of the hub of the rotor and the shroud ring 143 bound the passages on radially opposite sides while the adjacent blades bound the other two opposite sides. All of the walls may have slots therein but preferably only hub and case walls and one blade have slots. The slots in opposite walls within the passages are preferably not directly opposite each other.

The blades discussed herein are to be considered thin blades if their maximum thickness is less than 15 per cent of the blade section chord length.

In the preferred forms of the blades the chordwise length of the blade, that is the dimension along the direction of flow is preferably smaller than the spanwise dimension or length or at least the chord is not more than twice the span. The blades also have free leading and trailing edges extending in the same general radial direction.

Axial flow compressors have blade structures whose main flow passages extend in the general axial direction from an inlet at the front to an exist at the rear to discharge fluid in the general axial direction.

It will now be clear that I have provided a compressor which can operate efficiently over a wide range of mass flow rate per revolution. This is accomplished by thickening the noses of the blades in successive stages to take advantage of the increasing value of the velocity of sound in the compressed fluid; also by arranging the blades so that there is only a small range of approach angles for the blade to handle. This is very important for a supersonic rotor. It is also particularly advantageous in the last stage where it is desirable to add a large pressure increase and at the same time reduce the axial velocity. This is accomplished by the special rotor of the last stage which has expanding cross sectional areas of the passages between blades.

There are many applications, in aircraft particularly, where a short compressor is significant. For instance the velocity of flow through a combustion chamber of a gas turbine should be low but the discharge velocity of a compressor is high. Consequently the combustion cham-

6

ber must be connected to the compressor by an expanding tube to reduce the velocity. The special rotor of this invention expands the flow and lowers the velocity in one stage of the compressor while compressing, thus doing away with the need of a long diffuser.

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 compressor, a case, a plurality of stages of rotor and stator blades, and means mounting said stages of stator blades in said case, means mounting said rotor stages for rotation about an axis for pumping fluid, the blades of said stages each being defined by blade sections each having a relatively sharp trailing end and a circular arc nose defined by a selected radius, said rotor and stator stages being alternated in succession along the axis of rotation, the blades of a downstream said stage each having blade sections of larger nose radii than the blades of a stage on the upstream side thereof, said downstream stage having a plurality of stages on the upstream side thereof to increase the fluid temperature and therewith the velocity of sound in said fluid adapting said blades of said downstream stage to operate efficiently.

2. In combination in an axial flow compressor, a case, a plurality of stages of rotor and stator blades, and means mounting said stages of stator blades in said case, means mounting said rotor stages for rotation about an axis for pumping fluid, the blades of said stages each being defined by blade sections each having a relatively sharp trailing end and a circular arc nose defined by a selected radius, said rotor and stator stages being alternated in succession along the axis of rotation, the stator blades of a downstream said stator stage each having a larger nose radius than the stator blades of a stator stage on the upstream side thereof to increase the range of efficient operation of the compressor, said downstream stator stage having a plurality of rotor stages on the upstream side thereof to increase the fluid temperature and therewith the velocity of sound in said fluid.

3. In combination in an axial flow compressor, a case, a plurality of stages of rotor and stator blades, means mounting said stages of stator blades in said case, means mounting said rotor stages in said case for rotation about an axis for pumping fluid, said rotor and stator stages being alternated along the axis of rotation of said rotor blades and including a first downstream rotor stage preceded by a plurality of said rotor stages, said plurality of stator stages including a first downstream stator stage succeeding in the downstream direction a plurality of said rotor stages and having blades whose aft portions are directed rearward against the direction of rotation of said first downstream rotor stage which is positioned downstream adjacent to said first downstream stator stage to give fluid leaving therefrom a leaving velocity vector directed rearward and against the direction of rotation of said first downstream adjacent rotor stage, and a second downstream stator stage positioned downstream adjacent to said first downstream rotor stage for receiving fluid therefrom, the stator blades of said downstream stator stages each being defined by blade sections each having a relatively sharp trailing end and a circular arc nose defined by a selected radius, said second downstream stator stage having blades each of greater nose radii than the blades of said first downstream stator stage.

References Cited in the file of this patent

UNITED STATES PATENTS

2,431,592 Stalker ----- Nov. 25, 1947