Oct. 31, 1950 E. A. STALKER 2,527,971

AXIAL-FLOW COMPRESSOR

Filed May 15, 1946

FIG. 9.

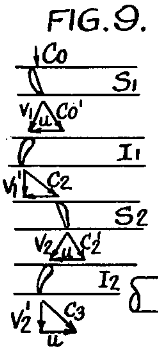
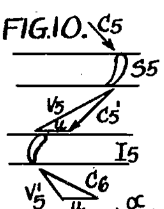


FIG. 10.



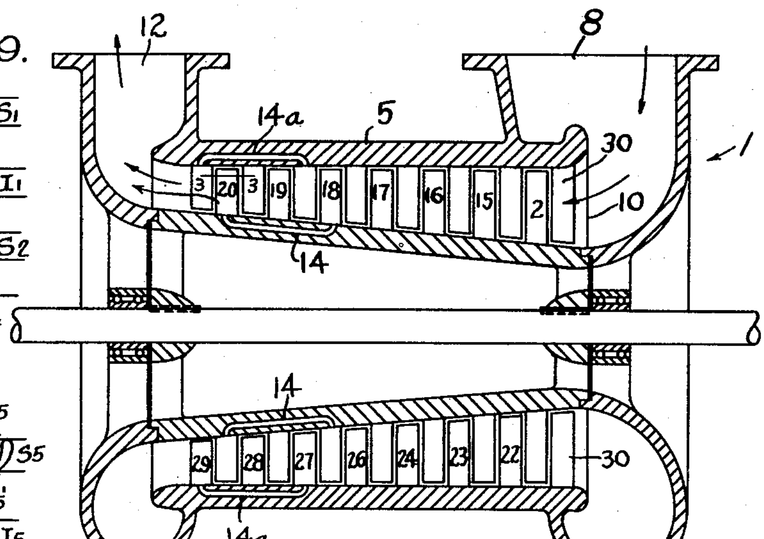


FIG. 1

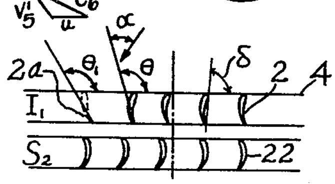


FIG. 2

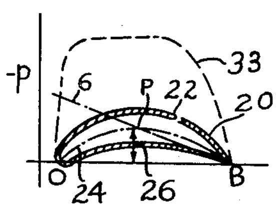


FIG. 3

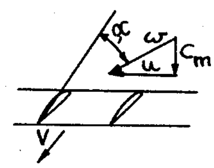


FIG. 4

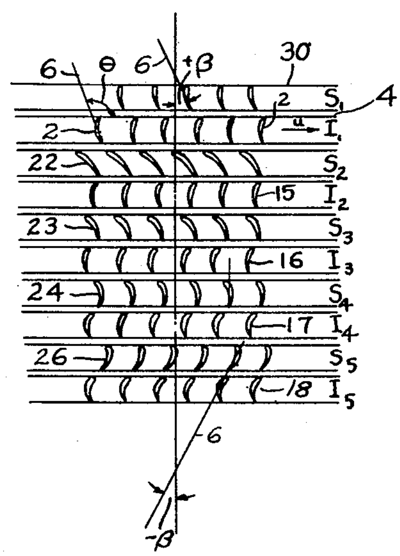


FIG. 8

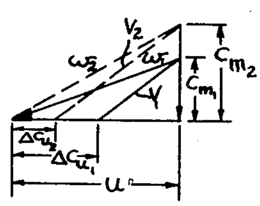


FIG. 5

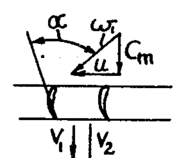


FIG. 6

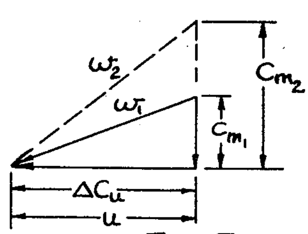


FIG. 7

INVENTOR.

Edward A. Stalker

Patented Oct. 31, 1950

2,527,971

UNITED STATES PATENT OFFICE

2,527,971

AXIAL-FLOW COMPRESSOR

Edward A. Stalker, Bay City, Mich.

Application May 15, 1946, Serial No. 669,856

5 Claims. (Cl. 230—122)

1

My invention relates to compressors particularly of the axial-flow type.

An object of the invention is to provide a means of obtaining a maximum pressure rise through the compressor with a high efficiency. 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—

Figure 1 is an axial section through the compressor;

Figure 2 is a development of two of the stages showing the type of blade section and their pitch setting;

Figure 3 is a section of a blade taken along line 3—3 in Figure 1;

Figure 4 is a vector diagram of the flow shown in relation to two impeller blades;

Figure 5 is a vector diagram of the flow relative to the blades of Figure 4 for two different axial velocities and the same peripheral velocity;

Figure 6 is a vector diagram for the flow of fluid shown in relation to the type of blades of this invention;

Figure 7 is a vector diagram of the fluid flow relative to the blades of Figure 6 for two different axial velocities and the same peripheral velocity; and

Figure 8 is a development of several stages including the stages shown in Figure 2.

Figure 9 shows the vector diagrams for the first two stages.

Figure 10 shows the vector diagrams for the last stage.

I have found by theory and verified by experiment that, as a condition for maximum pressure rise in a stage, the increment of peripheral velocity added by the impeller, commonly called ΔC_u, should be equal to the peripheral speed u at the point on the radius corresponding to the mean value of the square of the relative fluid velocity. To achieve this value the stator and impeller blades require a large angle of pitch and a large maximum ordinate of the mean camber line.

Figure 1 shows a compressor 1 having a development of the stages as shown in Figure 2. It will be observed that each of the blades 2 of the first impeller I_1 has a blade section set with such a large pitch that the angle δ between the plane of rotation 4 and the tangent to the undersurface of the section is greater than 60 degrees and the zero lift line 6 of the airfoil section exceeds 90 degrees. This line gives the direction of the flow for zero lift on the section, and consequently it may at first glance appear that the blade set at such a large angle would pump the air toward the inlet of the compressor rather than the exit as it should. Actually it will be in the correct direction, that is toward the compressor exit even when the blades are set with a still greater value of pitch such as θ_1 for the blade 2a shown dotted. If the rotor blades give any rotation to the flow, as they of course do, the stators will convert this rotary flow into an axial one. By this arrangement an axial flow is begun and immediately grows to a large magnitude reducing the angle of attack between the blade and the true relative flow direction. In order to do this the stator blades are given a high camber.

2

The axial-flow compressor incorporating the arrangement of the blades of Figure 2 is shown in Figure 1. Fluid enters the compressor inlet 8 and flows through the annular passage 10 to the exit 12. Preferably both the stator and impeller blades have boundary layer control slots in their upper surfaces to enable the blades to operate at large angles of attack α relative to the resultant flow vector w and large lift coefficients as described in my U. S. Patent No. 2,344,835 issued March 21, 1944. It will not be further described here except to remark that the ducts 14 interconnect the interiors of the blades of spaced stages so that the pressure difference of the stages causes a flow through the blade slots and this flow controls the boundary layer. Thus, for example, the impeller blades 18 are connected by tube 14 to impeller blades 20. Stator blades are connected in like manner, for example blade 28 with blade 26 via tube 14a. In this arrangement the upstream blade has a discharge slot and the downstream blade has an induction slot.

The impeller blades are 2, and 15 to 20, while the stator blades are 22 to 30.

An induction blade is shown in Figure 3 where the induction slot is 22. The blade section has the mean camber line 24 with the maximum ordinate 26 above the subtending chord OB.

In order to achieve the large value ΔC_u imparted to the fluid in the peripheral direction the value of the maximum ordinate 26 should be greater than 5% of the length of the subtending chord OB and can be as large as 60%. The value used will depend on the type of machine but usually it will be of the order of 20%.

The zero lift line is found as the line drawn through the trailing edge B and the midpoint P of mean camber line. The pressure distribution curve is 33.

Figures 4 and 5 illustrate the mode of operation of the conventional compressor. Figure 4 shows an impeller blade in relation to the air velocity vectors. The axial velocity is C_m, the peripheral component is u giving the resultant velocity w. This is the air velocity relative to the blade. The vector for the air leaving the stage is V. These vectors are shown for two conditions in Figure 5, one for an axial velocity of C_{m1} and another for an axial velocity of C_{m2}. In the first case the change in peripheral velocity is ΔC_{u1}, being the peripheral difference in vectors w_1 and V_1.

When the axial velocity is increased to C_{m2} the leaving velocity V_2 still has the same direction as V_1. Hence ΔC_{u2} is less than ΔC_{u1} and the pressure rise of the compressor has decreased.

With the compressor of the present invention, Figures 6 and 7, the resultant entering velocity is w_1 as shown in Figure 6 for a first case. The leaving velocity V_2 is axial in direction. For a second case the axial velocity becomes C_{m2}, and the leaving velocity V_2 is still axial. Hence ΔC_u is equal to u and remains constant. As remarked earlier this value of ΔC_u can be shown to offer the optimum pressure conditions.

It will thus be clear that the present invention makes possible an increase in volume pumped while maintaining the pressure. This is a very important characteristic.

It is usually desirable to obtain as large a pressure rise as possible from a given number of stages of an axial-flow compressor. To this end it is desirable to operate the machine with a tip peripheral speed as high as possible. However, the local velocity of relative flow must be less than the speed of sound or a compressibility shock occurs which limits the performance of the machine. This shock occurs whenever the local velocity attains substantially the velocity of sound in the local medium. The acoustic velocity is a function of the absolute temperature of the fluid, an increased temperature resulting in an increase of acoustic velocity.

Fluid passing over the upper surface of a blade increases in local velocity as is well known in aerodynamics. When this local velocity reaches the speed of sound in the local fluid, a compressibility shock will occur and seriously increase the blade resistance. To avoid the shock the ratio of local velocity to the acoustic velocity should be less than unity. The ratio is commonly called the Mach Number.

Since the fluid is coldest at entrance, the tip speed of the impeller blades is set by the speed of sound in the fluid at entrance. The tip speed can, however, be increased by giving the air an initial rotation at entrance in the same direction as the blade rotation. This expediency, however, reduces the pressure rise available from the machine. It has the advantage, however, of permitting the direct connection of the impellers to the shaft of the gas turbine which can operate at a greater tip speed because of the high temperature of the motive gas, giving a high value to the velocity of sound.

In the compressor the temperature rises due to the compression so that the blades of the later stages could be operated at a higher tip speed than the blades of the first stage. Where the stages are all on one shaft this is not practical. This invention discloses a means, however, of utilizing the increasing fluid temperature.

An initial vortex in the direction of rotation is induced at the entrance of the compressor by stator stage 30, Figures 1 and 8, to obtain a high tip speed relative to the compressor case but low relative to the fluid. This vortex is then dissipated in successive increments from stage to stage, preferably in the first half of the stages. Where a great many stages are used and the overall compression ratio is to be high, the last stages may be even subjected to a vortex of counter rotation.

To dissipate the initial vortex, I increase the camber of successive stator or guide vane stages. The camber increases the local velocity on the upper surfaces of the blades but since the temperature has been increased by the compression, the local velocity of flow can still be well below the local velocity of sound.

The cambers of the impeller blades are also increased from stage to stage to obtain greater lift coefficients. This also results in greater local velocity on the upper surface of the blade but again it is below the local velocity of sound because of the temperature rise.

In some of the downstream stages I utilize some of the increased temperature by arranging the stator blades to direct the fluid flow counter to the direction of rotation of the impellers. This increases the velocity of the fluid relative to the impeller and gives a greater pressure rise while at the same time not exceeding a Mach Number of one.

If the procedure of increasing the camber is followed to keep the local Mach Number just under unity, and if a large number of stages are used, the cambers of the downstream stages will become very large even though the cambers of the blades of the initial stages are quite small. The blades of real high camber will require boundary layer control but the earlier stages may not.

Figure 8 shows the development of the blades at a point somewhat out from the midpoint of the radius. This figure shows that the camber of the blades of successive stages is increased in the downstream direction. It also shows that succeeding stages deflect the fluid so as to give successively less component of velocity in the peripheral direction. This is indicated by the decreasing angle β between the zero lift lines 6 and the axis 32 of the machine. This figure shows that in the fourth stator stage S_4 the angle has changed to a negative angle $(-\beta)$ whereas in the first stage it was positive.

It will also be observed from Figure 8 that the camber of the blades of succeeding stages has increased in the downstream direction.

Figure 9 shows the vector diagrams for the first two stages where the air is discharged from the impeller in a substantially axial direction. This axial direction of discharge may be maintained for several other stages but since the size of the compressor is usually of great significance the vector diagrams are changed to the type shown for stage 5, Fig. 10, which may be typical of several neighboring stages.

The notation used places a prime on the symbol for the leaving velocity. The absolute velocities are indicated by C with proper suffix while the relative velocities are given by v with proper suffix. Thus the inlet velocity to the first stator is C_0 and the leaving vector is C_0'. The rotational velocity is u which combined with C_0' gives v_1 the vector relative to impeller I_1. It will be observed that C_0' has a component in the direction of rotation. The fluid leaves I_1 with the vector v_1' normal to the plane of rotation. The air enters the stator C_2 and leaves with C_2'. It is to be noted that this vector has a smaller component in the direction of rotation of I_2. In the compressor of Fig. 1 this component reverses direction at an intermediate stage and actually is directed against the direction of rotation. Stage 5 is typical of such stages.

Fluid from I_4 is discharged toward S_5 along C_5 and leaves S_5 along C_5'. This vector has a component in the plane of rotation of impeller I_5 counter to its direction of rotation. When combined with u the velocity relative to impeller 5 is v_5. It is to be noted that the vectors C_5' and u add to increase the magnitude of vector v_5 whereas in the first stage the vectors C_0' and u combined to give v_1 smaller than C_0'.

Since the fluid has been compressed in a plurality of stages ahead of the stage where the vector from the stator changes the direction of its horizontal component the temperature has risen and accordingly the velocity of sound in the fluid has increased in magnitude. Thus the increased velocity component such as v_5 can be accommodated without compressibility shock.

To recapitulate, I have disclosed a compressor which can operate at the best condition for producing a high compression ratio. That is, it can be operated to give the fluid a peripheral change in velocity equal to the peripheral speed of the blade. This is many times the value provided by conventional axial-flow compressors. To accomplish this, the compressor has blade sections of large arching of the mean camber line and the blades are set at very large pitch angles.

For operating at very large peripheral speeds relative to the case, while yet not at large speeds relative to the fluid pumped, stator blades are placed ahead of the first impeller to induce a vortex of the same rotational direction as the impeller. This somewhat reduces the maximum pressure ratio of the front stages but the loss is more than regained because the later stages can be operated at large speeds relative to the air because of the rising temperature of the compressed fluid and because the stators can be arranged to dissipate the initial vortex and may even advantageously introduce a vortex rotating counter to the impellers. By these procedures the number of stages for a given pressure ratio can be greatly reduced. For instance, an 18-stage compressor can be reduced to six stages.

The high cambers beyond certain values require boundary layer control to compel the flow to follow the blade surfaces.

I do not intend to limit the invention to blades requiring boundary layer control for high lift. In the case of the initial vortex, for instance, the first stages might have low cambers well below the need for boundary layer control. The camber could then be increased successively to values at the rear stages which would just escape the need for boundary layer control. Such a machine would not equal the pressure performance where the early stages are highly cambered and set at large pitch angles as described.

I have described one type of boundary layer control but I do not intend to limit myself to this. In particular, I intend to claim any type of slot in the blade surface supplied with air by any means.

I use the terms impeller and rotors interchangeably to designate the blade and hub structure to impel the flow through the plane of rotation.

I use the terms stator blades and guide vanes interchangeably to indicate the vanes which direct the fluid from one impeller to another. I do not intend to imply that stator blades or guide vanes are necessarily stationary. They might, for instance, be rotating oppositely to the impellers.

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.

What is claimed is:

1. In combination in a compressor, a case, hub means mounted in said case in spaced relation thereto forming therewith an annular flow passage within having an inlet and an exit and conveying a fluid flow, said hub means being mounted for rotation about an axis, a plurality of stages of impeller blades mounted on said hub means to impel said fluid flow, a plurality of stages of stator blades supported in said case with a stator stage ahead of each said impeller stage, the blades of a said stator stage ahead of an upstream impeller stage being angularly positioned to deflect said fluid flow in the direction of the rotation of the adjacent downstream impeller stage, a said stator stage ahead of a said downstream impeller stage having vanes angularly positioned to deflect said fluid flow against the direction of rotation of the adjacent downstream impeller stage, said annular passage between said upstream and downstream impeller stages having substantially continuously decreasing cross sectional areas in the downstream direction to increase the velocity of the flow from said downstream stator stages against said adjacent downstream impeller stage, a plurality of said impeller and stator stages being positioned ahead of the said downstream impeller stage first subjected to a counter-rotation inflow so as to increase the fluid temperature by compression to raise the velocity of sound in the fluid to a value higher than the local fluid velocity on the blades of said impeller stage subjected to said counter flow from the adjacent upstream stator stage, said impeller stage subjected to counter-rotation inflow having cambered blades with slots therein, and means inducing a flow of fluid through said slots for compelling the said flow to cling to the surface of said blade.

2. In combination, in an axial-flow compressor having a fluid flow therethrough, a hub means supported for rotation about an axis, a plurality of blades supported on said hub means forming a plurality of impellers spaced along said axis, a plurality of guide vanes forming a plurality of axially spaced guide vane stages alternating with said impellers to direct a fluid flow from one impeller to another, said stages including a row of guide vanes ahead of the first impeller, said guide vanes being positioned to cause rotation of the fluid in the direction of rotation of said first impeller, a downstream said guide vane stage having vanes angularly positioned to direct said fluid flow in counter-rotation to the direction of rotation of an adjacent downstream said impeller, the first said impeller subjected to a counter-rotation inflow being preceded by a plurality of said impellers adapted to produce a substantial temperature rise in said fluid ahead of said impeller subjected to counter-rotating fluid, the blades of successive impellers in the downstream direction having blade sections of increasing maximum height of the mean camber arc above

the subtending chord to successively increase the temperature and pressure of said successive impellers, said temperature rise increasing the velocity of sound in the fluid to a value higher than the local fluid velocity on the blades of said impeller subjected to said counter rotation flow.

3. In combination in an axial flow compressor, a hub means supported for rotation about an axis, a plurality of blades supported on said hub means forming a plurality of impellers spaced along said axis, a plurality of guide vanes forming a plurality of guide vane stages interposed between said impellers to direct a fluid flow from one impeller to another, an upstream group of said guide vane stages disposed successively in the downstream direction being adapted to direct said fluid flow in the direction of rotation of said impellers with a successively smaller component of peripheral velocity for each said successive stage, and a downstream group of successive said guide vane stages being adapted to direct said fluid flow counter to the direction of rotation of said impellers adjacent thereto, the first said impeller subjected to a counter-rotation inflow being preceded by a plurality of said impellers adapted to produce a substantial temperature rise in said fluid ahead of said first impeller subjected to counter-rotating fluid, the blades of successive said impellers in the downstream direction having blade sections of increasing maximum height of the mean camber line above the subtending chord to successively increase the local velocity on successive blades thereby achieving augmented pressure rise per stage in the pumped fluid, said temperature rise increasing the velocity of sound in the fluid to a value higher than the local fluid velocity on the blades of said impeller subjected to counter-rotation in flow and on the blades of said successive impellers, said blades having slots in their upper surfaces, and means to induce a flow of fluid therethrough to control the flow on the blade surfaces.

4. In combination in a compressor having a main flow of fluid therethrough, a plurality of stages of impeller blades mounted for rotation about an axis, a plurality of stages of stator guide vanes supported with a said stage ahead of and adjacent each said impeller stage, said vanes of a stator stage ahead of an upstream impeller stage being angularly positioned to deflect said fluid flow in the direction of rotation of the adjacent downstream impeller stage, a said stator stage ahead of a downstream said impeller stage having vanes angularly positioned to deflect said fluid flow against the direction of rotation of the adjacent downstream impeller stage, a plurality of said impeller and stator stages being positioned ahead of said downstream impeller stage first subjected to a counter-rotation inflow so as to increase the fluid temperature by compression to raise the velocity of sound in the fluid to a value higher than the local fluid velocity on the blades of the said impeller stage subjected to said counter flow from said adjacent upstream stator stage, said impeller stage subjected to counter-rotation inflow having cambered blades with slots therein, and means to induce a flow of fluid through said slots for compelling the said main flow to cling to the surface of each said blade.

5. In combination in a compressor having a main flow of fluid therethrough, a plurality of stages of impeller blades mounted for rotation about an axis, and a plurality of stages of stator vanes supported with a stage ahead of each said impeller stage, said vanes of the stator stage ahead of an upstream impeller stage being angularly positioned to deflect said fluid flow in the direction of rotation of the adjacent downstream impeller stage, a said stator stage ahead of a downstream said impeller stage having vanes angularly positioned to deflect said fluid flow against the direction of rotation of the adjacent downstream impeller stage, a plurality of said impeller and stator stages being positioned ahead of the said downstream impeller stage first subjected to a counter-rotation inflow so as to increase the fluid temperature by compression to raise the velocity of sound in the fluid to a value substantially higher than the local fluid velocity on the blades of the said impeller stage subjected to said counter inflow from said adjacent upstream stator stage.

EDWARD A. STALKER.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
936,114	Gardner	Oct. 5, 1909
1,463,110	Worthen	July 24, 1923
2,234,733	Jendrassik	Mar. 11, 1941
2,314,058	Stalker	Mar. 16, 1943
2,344,835	Stalker	Mar. 21, 1944

FOREIGN PATENTS

Number	Country	Date
8,741	Great Britain	1906