

Feb. 2, 1960

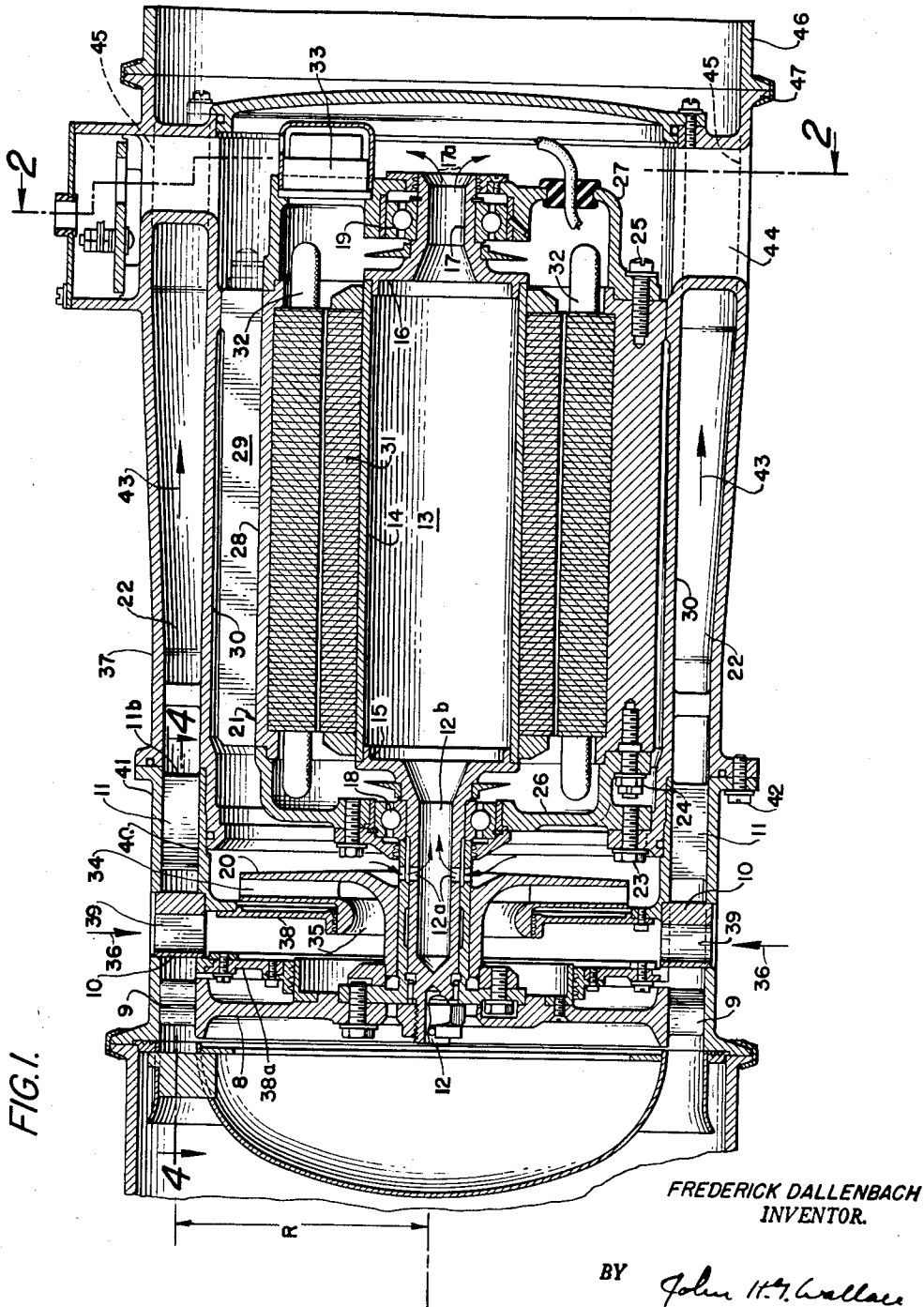
F. DALLENBACH

2,923,461

IMPULSE AXIAL-FLOW COMPRESSOR

Filed April 27, 1953

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FIG. 2.

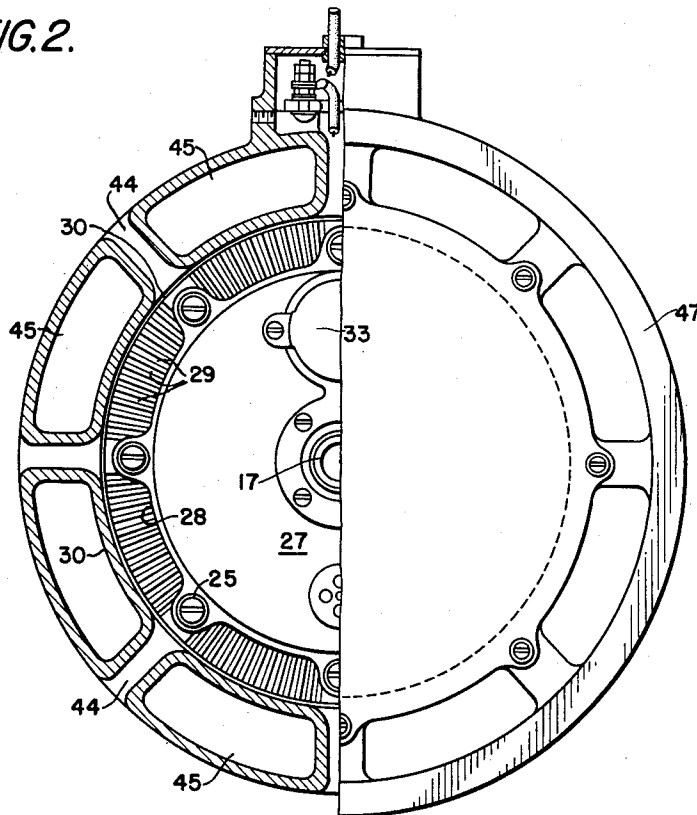
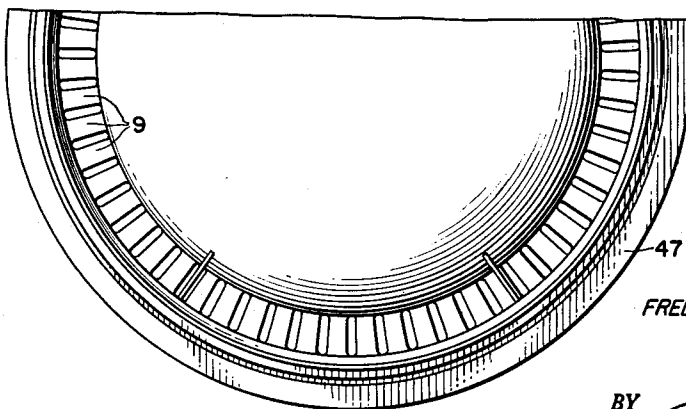


FIG. 3.



FREDERICK DALLENBACH,
INVENTOR.

BY *John H. T. Wallace*

Feb. 2, 1960

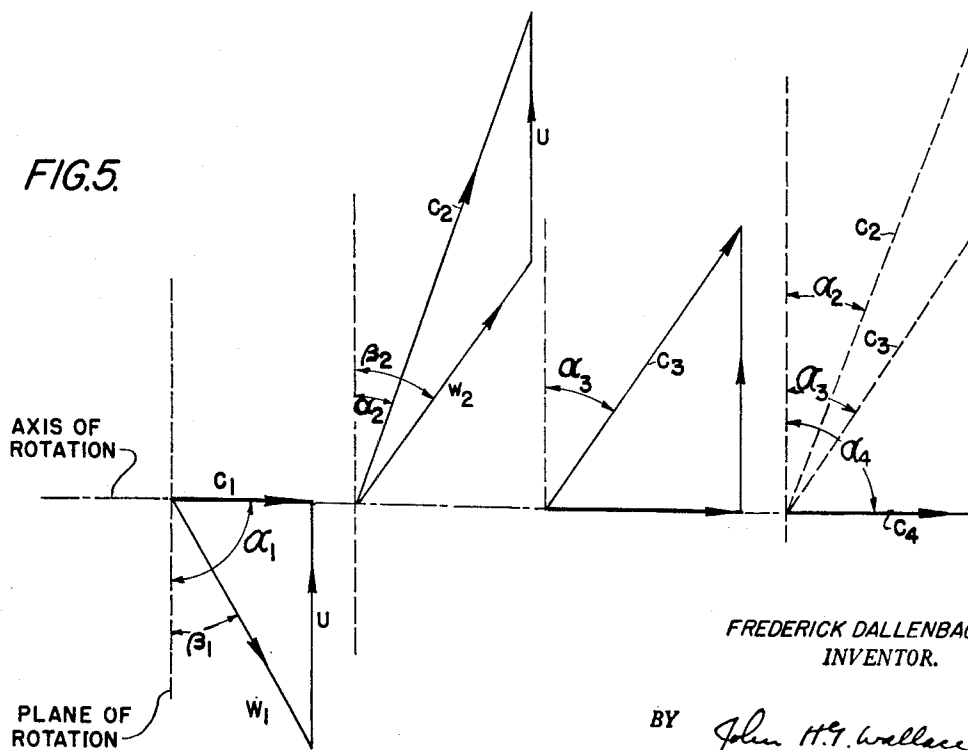
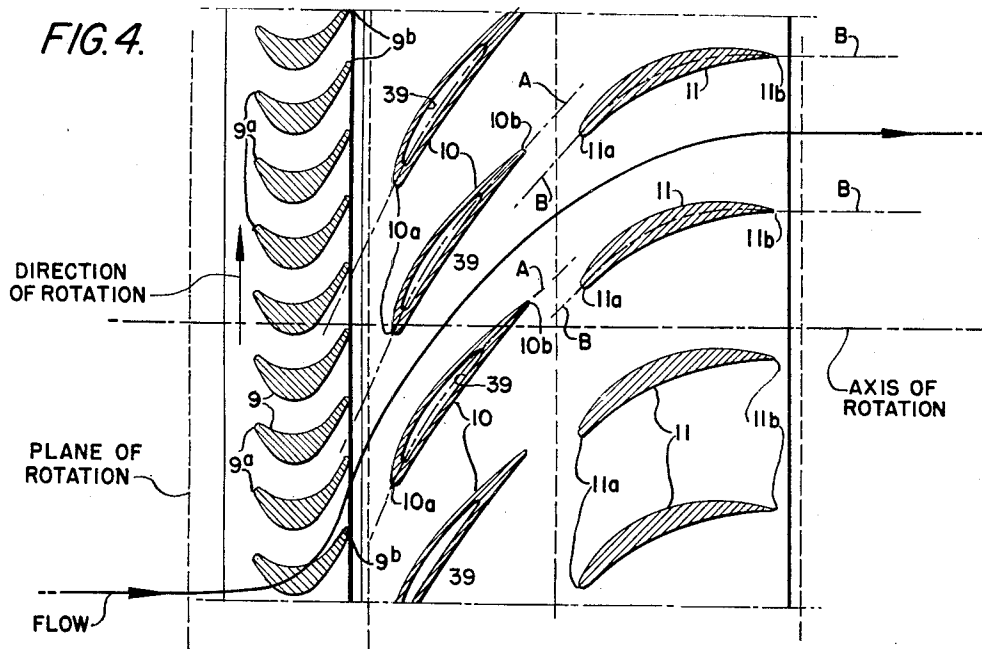
F. DALLENBACH

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4 Sheets-Sheet 3



FREDERICK DALLENBACH,
INVENTOR.

BY *John H. G. Wallace*

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2,923,461

IMPULSE AXIAL-FLOW COMPRESSOR

Frederick Dallenbach, Inglewood, Calif., assignor to The Garrett Corporation, Los Angeles, Calif., a corporation of California

Application April 27, 1953, Serial No. 351,155

9 Claims. (Cl. 230—120)

This invention relates to a compressor and more particularly to an impulse axial-flow compressor.

An object of this invention is to provide an impulse axial-flow compressor which is capable of producing a large pressure rise at low rotational speed.

Another object of this invention is to provide an impulse axial-flow compressor which is capable of delivering increasing pressure rises with increasing air flow at constant rotational speed.

Another object of this invention is to provide a novel rotor and stator combination having high efficiency.

Another object of this invention is to provide an impulse axial-flow compressor wherein the impulse rotor blades are cooperatively arranged with tandem cascade stator blades capable of deflecting the air leaving the rotor through large turning angles thereby converting the kinetic energy of the air leaving the rotor into a maximum useful static pressure rise.

Another object of the invention is to provide a rotor structure and tandem cascade stator blades cooperatively arranged therewith, for preventing separation of flow on the stator blades, and thereby accomplishing efficient and relatively high static pressure recovery in the operation of the impulse axial-flow compressor.

Another object of the invention is to provide an impulse axial-flow compressor which is particularly desirable for use in connection with supercharging, cooling and ventilating equipment, when operated at high altitude, since the present compressor is capable of delivering greater air weight flow, at low densities, than do conventional compressors.

It is another object of the invention to provide an impulse axial-flow compressor which is capable of operating with low power absorption at reduced flow deliveries and high inlet air densities.

Another object of the invention is to provide an impulse axial-flow compressor driven by an air-cooled electric motor wherein the stator and rotor structures thereof are both air-cooled, and in which a thermal protector switch is remote from direct influence of air used to cool the motor thereby permitting accurate control of the maximum allowable motor temperature to be maintained.

Another object of the invention is to provide an efficient mechanical arrangement for adequate cooling of both the rotor and the stator structures of an electric motor employed for driving the compressor.

Another object of the invention is to provide an efficient and compact impulse axial-flow compressor driven by an air-cooled thermally protected electric motor capable of reliable operation when subject to relatively high temperature motor cooling air and when the compressor is operating on high temperature air.

Another object of the invention is to provide an impulse axial-flow compressor, driven by an electric motor, wherein a motor cooling fan is capable of utilizing a separate source of air, such as ambient air, to cool the

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motor independently of the source of compressor inlet air.

Another object of the invention is to provide an effective sealing means, interposed between the compressor rotor and the motor cooling fan, which prevents substantial air leakage from the compressor to the cooling fan.

Still another object of the invention is to provide novel ducting means forming an integral part of the compressor structure and extending to the exterior thereof, for discharging the air employed to cool the electric motor.

A further object of the invention is to provide an impulse axial-flow compressor having a novel combined motor cooling duct and compressor delivery duct arrangement.

Further objects and advantages of the invention will appear from the specification and the accompanying drawings in which:

Fig. 1 of the drawing is a longitudinal sectional view of an impulse axial-flow compressor in accordance with the present invention showing parts and portions in elevation to facilitate the illustration;

Fig. 2 is a transverse half-sectional view taken on the line 2—2 of Fig. 1;

Fig. 3 is a fragmentary inlet end view of the impulse axial-flow compressor;

Fig. 4 is an enlarged fragmentary sectional view taken on line 4—4 of Fig. 1 showing the impulse compressor rotor blades and the tandem stator blades structure relative thereto;

Fig. 5 shows the velocity vector diagram of the air flowing progressively through the rotor and stator blades of the impulse axial compressor; said vector diagram being of equal scale to the structure shown in Fig. 4 of the drawing, and

Fig. 6 is an enlarged view of the compressor rotor and stator blades structure taken on substantially the same line as that of Fig. 4 of the drawing showing by broken lines relative dispositions of adjacent blades.

The wheel 8 carrying the cantilever blades 9 rotates in a direction as shown in Fig. 4 of the drawing. The rotor blades 9 are airfoil sections having respective leading and trailing edges 9a and 9b. These blades 9 are axial-flow compressor blades and may embody a variety of configurations peculiar to certain requirements and operating conditions of a compressor constructed according to the present invention. As shown in Fig. 6 the slope of the mean camber line 9c at the trailing edge 9b of each blade is disposed at an angle γ less than 90° to the plane of rotation and the trailing edge 9b is directed toward the direction of rotation. The preferred angle γ between the mean camber slope lines 9c of the trailing edges of the rotor blades 9 and the plane of rotation lies between 30° and 50° . The shape of the rotor blades 9 is such that air entering the blades has a velocity imparted thereto, and the direction of such air-flow defines an angle less than 90° to the plane of blade rotation. The air passing through the rotor blades, is turned and the relative velocities of the air therethrough are in the direction of wheel rotation.

The mean camber lines of the stator blades 10 and 11 are designated A and B respectively. The mean camber line slope at the leading edge 10a of each stator blade 10 is substantially parallel to the direction of air flowing from the rotor blades 9. The mean camber line slope at the leading edge 11a of each stator blade 11 is substantially parallel to the mean camber line slope at the trailing edge 10b of each stator blade 10. As shown in Fig. 4 the trailing edges of the stator blades are axially and tangentially spaced from the leading edges of the stator blades 11, whereby the wake of the trailing surfaces of the stator blades 10 does not flow over the stator blades

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11 but passes between successive ones of the blades 11, thereby avoiding high losses of efficiency in the second stage stators 11. The mean camber line slope at the trailing edges 11b of the stator blades 11 is substantially parallel with the axis of the rotor 8.

As shown in Fig. 4 of the drawing the air enters the rotor 8 at approximately axial parallelism therewith and after discharge from the rotor the air is turned by the tandem cascade arrangement of the stator blades 10 and 11 whereby it again flows in approximately axial parallelism with the rotor. Due to the aforescribed characteristics of the impulse type of rotor blading, air leaves the rotor with an increased kinetic energy with substantially no change of static pressure across the rotor. This kinetic energy of the air leaving the wheel is converted into a static pressure rise in the tandem cascade blade arrangement by turning the air leaving the rotor into essentially the same direction as that of the air entering the rotor. The tandem cascade stator blades 10 and 11 prevent separation of flow, thereby preventing turbulence and maintaining substantially laminar flow to accomplish efficient and relatively high static pressure recovery.

Referring particularly to Fig. 5 of the drawings it will be seen that specific relationships of velocities referred to are graphically illustrated. C_1 denotes the vectorial measure of the absolute velocity of the air entering the rotor blades while α_1 indicates the angle of the absolute velocity of the air entering the rotor blades. U denotes the peripheral velocity of the rotor blades while W_1 denotes the relative velocity of the air entering the rotor. The direction of the relative velocity of the air entering the rotor is denoted by β_1 . The absolute velocity of the air leaving the rotor blades 9 is represented by C_2 while the relative velocity of the air leaving the rotor blades 9 is represented by W_2 . The direction of the absolute velocity of the air leaving the rotor blades 9 is represented by α_2 while the direction of the relative velocity of the air leaving the rotor blades 9 is represented by β_2 . The vectorial measure of the absolute velocity of the air entering the stator blades 10, forming the first stage of the tandem stator cascade, is represented by C_2 while the vector of the absolute velocity of the air, leaving the stator blades 10, forming the first stage of the tandem stator cascade, is represented by C_3 .

Thus, the vectorial measure of the absolute velocity of the air entering the stators 11, forming the second stage of the tandem stator cascade, is substantially C_3 while the vectorial measure of the absolute velocity of the air leaving the second stage tandem stator cascade is represented by C_4 all as shown in Fig. 5 of the drawing. α_3 represents the direction of the absolute velocity leaving the first stage of the tandem stator cascade while α_4 represents the direction of the absolute velocity leaving the second stage of the tandem stator cascade. With reference to Fig. 5 of the drawing the absolute velocities may be readily compared with direct relationship to flow through various components of the compressor shown in Fig. 4. A comparison of C_2 with C_3 and C_4 provides a proportional comparison of the absolute velocities during decelerated flow through the tandem stator cascade.

The rotor 8 together with the blades 9 are driven at a constant speed by the electric motor which will be hereinafter described in detail.

As shown in Fig. 1 of the drawing the impulse axial-flow compressor is provided with a compressor rotor wheel 8 having impulse type blades 9, arranged to cooperate with the stator vanes 10 and 11. The rotor wheel 8 is supported on the shaft member 12 of the electric motor rotor 13. The rotor 13 is provided with an enlarged hollow cylindrical shaft member 14 fixed to the flange 15 of the shaft 12. The opposite end of the hollow cylindrical shaft member 14 is fixed to the flange 16 of the hollow shaft member 17. The shaft members 12 and 17 are supported in bearings 18 and 19 respec-

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tively which maintain concentric relationship of the rotor wheel 8, the motor rotor 13, and the impeller 20, relative to the casing structures of the compressor. These bearings 18 and 19 are retained in opposite ends, 26 and 27, of the heat exchanging casing 21. The heat exchanging casing is fixed internally of the compressor duct casing 22 by means of the bolts 23. The bolts 24 and 25 secure the ends 26 and 27 of the heat exchange casing to the cylindrical section 28 thereof, as shown best in Fig. 2 of the drawings. The cylindrical section 28 of the heat exchanging casing 21 is provided with a plurality of radially extending heat exchanging fins 29 which project therefrom and extend into close proximity to the inner wall 30 of the compressor duct casing 22. Supported on the outer side of the hollow cylindrical shaft portion 14 of the rotor 13 are the rotor windings 31, and fixed to the inner wall of the cylindrical casing section 28 of the heat exchanging casing 21 are the rotor stator windings 32. The electric motor of this impulse axial flow compressor is of the polyphase type having short circuited rotor windings.

Communicating with the stator windings internally of the heat exchange casing is the thermal protector switch 33. The switch embodies a conventional arrangement, including a thermally responsive element. The thermal protector switch is supported in the casing end member 27 of the heat exchange casing and is arranged to interrupt the flow of current to the electric motor in the event it is overheated to a predetermined degree. The heat exchanging casing portion 28, together with the inner wall 30 of the compressor duct casing 22, provides a duct surrounding the heat exchanging fins 29. This duct outwardly of the heat exchanging casing 21 provides a passage for air, which cools the motor without affecting the operation of the thermal protector switch while the latter senses the temperature of the stator windings. The duct defined by the casing walls 28 and 30 communicates with the outlet 34 of the impeller 20. This impeller is carried by the shaft 12 and is provided with an inlet 35 communicating with ambient air, flowing as indicated by the arrow 36. Positioned between the inner and outer walls 40 and 41 respectively, of the compressor duct casing 22, are the compressor stator blades 10 and 11. The blades 10 are provided with air passages 39, which extend radially therethrough, providing a passage for the air as indicated by the arrow 36, permitting direct communication with the inlet 35 of the impeller 20. Communicating with the outlet 34 of the impeller 20, are openings 12a which serve as passages to conduct air into the bore portion 12b of the shaft 12. Seal structures 38 and 38a provide confining walls for the entrance of air passing to the inlet 35 of the impeller 20. The seal 38a is arranged to prevent leakage of air from the compressor wheel 8 to the inlet of the impeller 20. The upstream section of the compressor duct casing 22 is connected to the downstream section thereof by means of the bolts 42. The inner and outer walls 40 and 41, respectively, align with the inner and outer walls 30 and 37, respectively, forming a continuous annular duct for air flowing in a direction as indicated by the arrow 43. The compressor duct casing 22 near the downstream end thereof is provided with radially extending passages 44. These passages communicate with the duct inwardly of the casing wall 30 and with the fins 29 of the heat exchanging casing 21. The compressor duct 22 is provided with outlet openings 45 which are disposed intermediate the passages 44 as shown in Fig. 2 of the drawing. The outlet openings 45 communicate with the interior of a duct 46 which is connected to a flange 47 of the compressor duct casing 22.

The operation of the impulse axial-flow compressor is substantially as follows: when the electric motor is energized, the rotor 13 thereof revolves in the bearings 18 and 19 thereby rotating the rotor wheel 8 causing

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the blades 9 thereof to impel air toward and past the stator blades 10 and 11 and through the compressor duct casing 22 as indicated by the flow line in Fig. 4 of the drawings. The air passes into the duct 46 which conducts it to the desired point.

Under certain operating conditions, the flow of air through the compressor duct 22 is at fairly high temperature, therefore, it is necessary to provide very efficient means for cooling the electric motor internally of this compressor duct. The passages 39 extending through the stator blades 10 provide air inlets communicating with the impeller outlet 34. The air flow through the passages 39, as indicated by the arrows 36, at times may be heated, but is preferably cool ambient air for cooling the motor. The air entering the impeller 20 is relatively dense as compared with the air which has dissipated heat from the motor. This air is acted upon by the impeller 20 before the air is heated by the motor, in order to maintain efficient operation of the impeller. The impeller is centrifugal in operation, and its peripheral outlet communicates with the heat exchanging fins 29 projecting from the heat exchanging casing 21. The hollow cylindrical portion 28 of the heat exchanging casing 21 is arranged in thermally conductive relationship with the motor stator windings 32 in order to exchange heat from the motor without introducing cooling air into the area of the stator windings. The thermal protector switch 33 communicates with the interior of the heat exchanging casing 21 and senses the temperature of the stator windings for the purpose of shutting off the supply of electricity to the motor in the event it becomes overheated. Air which passes in heat exchange relationship with the fins 29 is exhausted through the radially extending passages 44 together with the air forced through the hollow rotor structure of the electric motor as indicated by the arrows 17a. The thermal protector switch 33 having its temperature sensing element internally of the heat exchanging casing 21 is, therefore, capable of responding to a predetermined temperature rise in the stator winding without any direct thermal influence of the cooling air which absorbs heat from the stator and rotor structures of the motor. The cooling air passing internally on the inner wall 30 of the compressor duct 22 also prevents heat exchange from the compressor duct 22 to the electric motor.

I claim:

1. In a compressor, a rotor having axial flow blades each provided with leading and trailing edges, the mean camber line slope of said blade at their trailing edges extending at an angle between 30 and 50° to the plane of rotation of said rotor and disposed toward the direction of rotation thereof, and stator blades having a leading edge mean camber slope line substantially parallel to the mean camber lines of the trailing edges of said rotor blades.

2. In a compressor, a rotor having axial flow blades each provided with leading and trailing edges, the mean camber line slope of said blades at their trailing edges extending at an angle between 30 and 50° to the plane of rotation of said rotor and disposed toward the direction of rotation thereof, stator blades having their leading edge mean camber slope line substantially parallel to the mean camber lines of the trailing edge portions of said rotor blades, and a second stage stator blade cascade having blades in which the camber line slope of the blades at their leading edges is substantially parallel to the mean camber line slope of the trailing edges of the first stage stator blades.

3. In a compressor, a rotor having axial flow blades each provided with leading and trailing edges, the mean camber line slope of said blades at their trailing edges extending at an angle between 30 and 50° to the plane of rotation of said rotor and disposed toward the direction of rotation thereof; stator blades having their leading

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edge mean camber line slope substantially parallel with the mean camber lines of the trailing edge portions of said rotor blades, a second stage stator blade arrangement having a plurality of blades in which the camber line slope of the blades at their leading edges is parallel to the mean camber line slope of the trailing edges of the first stage stator blades, said second stage stator blades, at their leading edges, being spaced axially and tangentially from the trailing edges of said first stage stator blades.

4. In a compressor, a rotor having axial flow blades each provided with leading and trailing edges, the mean camber line slope of said blades at their trailing edges extending at an angle between 30 and 50° to the plane of rotation of said rotor and disposed toward the direction of rotation thereof; stator blades having leading edge mean chamber line slope substantially parallel with the mean camber lines of the trailing edge portions of said rotor blades; a second stage stator blade arrangement having a plurality of blades in which the mean camber line slope of the blades at their leading edges is parallel to the mean camber line slope of the trailing edges of the first stage stator blades, said second stage stator blades being spaced axially from the first stage stator blades and also spaced circumferentially, at the leading edges thereof, from the trailing edges of the first stage stator blades, the mean camber line slope at the trailing edges of said second stage stator blades being substantially parallel to the axis of the compressor.

5. In an impulse axial-flow compressor: a rotor having blades of a particular shape adapted to impart to the air leaving the rotor blades, a relative velocity in a direction angle between 30 and 50° to the plane of rotation of the rotor and disposed toward the direction of rotor rotation; and a tandem stator cascade blade structure adapted to turn the air flow into a direction substantially parallel with the axis of the rotor, the trailing edge and the leading edge camber line slope of adjacent stage blades of the tandem stator cascade being substantially parallel.

6. In an impulse axial-flow compressor: a rotor having blades of a particular shape adapted to impart, to the relative velocity of the air leaving the rotor blades, a direction angle between 40 and 60° to the axis of the rotor and disposed toward the direction of rotor rotation; and a multiple stage cascade stator structure adapted to turn the air flow into a direction substantially parallel with the axis of the rotor, the mean camber slope line of the trailing edges of one set of stator blades in said cascade structure being substantially parallel to that of the leading edges of an adjacent set of stator blades.

7. In an impulse axial-flow compressor: a rotor having radially extending curved blades with leading and trailing edges extending in the direction of rotation, the mean camber line of the trailing edges of said blades extending at an angle between 30 and 50° to the plane of rotation of said rotor; and a plurality of axially spaced rows of curved stator blades at the output side of said rotor, said stator blades having leading and trailing edges, the mean camber lines at the leading edges of the stator blades adjacent said rotor being substantially parallel to the mean camber lines of trailing edges of the rotor blades, the mean camber lines at the trailing edges of the last row of stator blades being substantially parallel with the axis of rotation of said rotor, and the mean camber lines of the leading and trailing edges of adjacent rows of stator blades being substantially parallel.

8. In an impulse axial-flow compressor: a rotor having radially extending curved blades with leading and trailing edges extending in the direction of rotation, the mean camber line of the trailing edges of said blades extending at an angle between 30 and 50° to the plane of rotation of said rotor; and a plurality of rows of curved stator blades spaced axially at the output side of said rotor, said stator blades having leading and trailing edges, successive rows of stator blades being progressively curved to dispose the mean camber lines of the leading edges of the

stator blades adjacent said rotor substantially parallel to the mean camber lines of the trailing edges of said rotor blades and the mean camber lines of the trailing edges of the last row of stator blades substantially parallel to the axis of rotation of said rotor.

9. In an impulse axial-flow compressor: a rotor having radially extending curved blades with leading and trailing edges extending in the direction of rotation, the mean camber line of the trailing edges of said blades extending at an angle between 30 and 50° to the plane of rotation of said rotor; and a plurality of rows of curved stator blades spaced axially at the output side of said rotor, said stator blades having leading and trailing edges, successive rows of stator blades being progressively curved to dispose the mean camber lines of the leading edges of the stator blades adjacent said rotor substantially parallel to the mean camber lines of the trailing edges of said rotor blades and the mean camber lines of the trailing edges of the last row of stator blades substantially parallel to the axis

of rotation of said rotor, the stator blades in a succeeding row being offset from and spaced circumferentially a greater distance than the stator blades in the preceding row.

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CERTIFICATE OF CORRECTION

Patent No. 2,923,461

February 2, 1960

Frederick Dallenbach

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 5, line 50, for "blade" read -- blades --; column 6, line 51, for "radically" read -- radially --.

Signed and sealed this 26th day of July 1960.

(SEAL)

Attest:

KARL H. AXLINE
Attesting Officer

ROBERT C. WATSON
Commissioner of Patents

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