May 2, 1950 C. DE GANAHL ET AL 2,505,755

AXIAL FLOW COMPRESSOR

Filed June 10, 1946 4 Sheets-Sheet 1

Fig. 1,

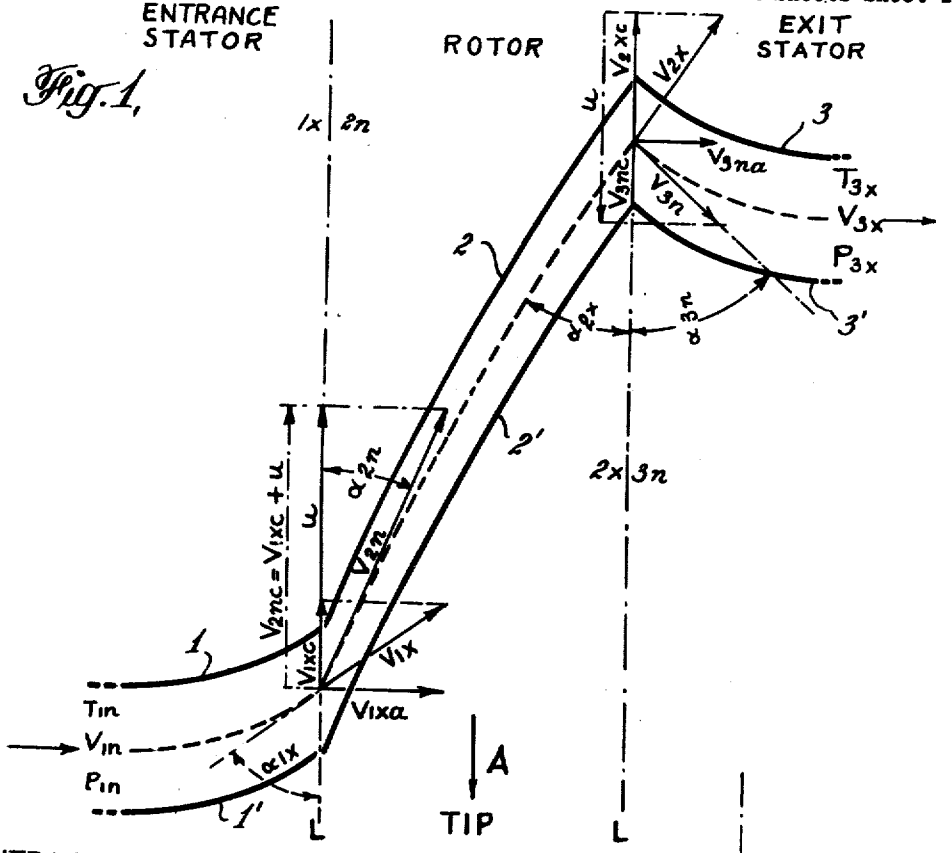
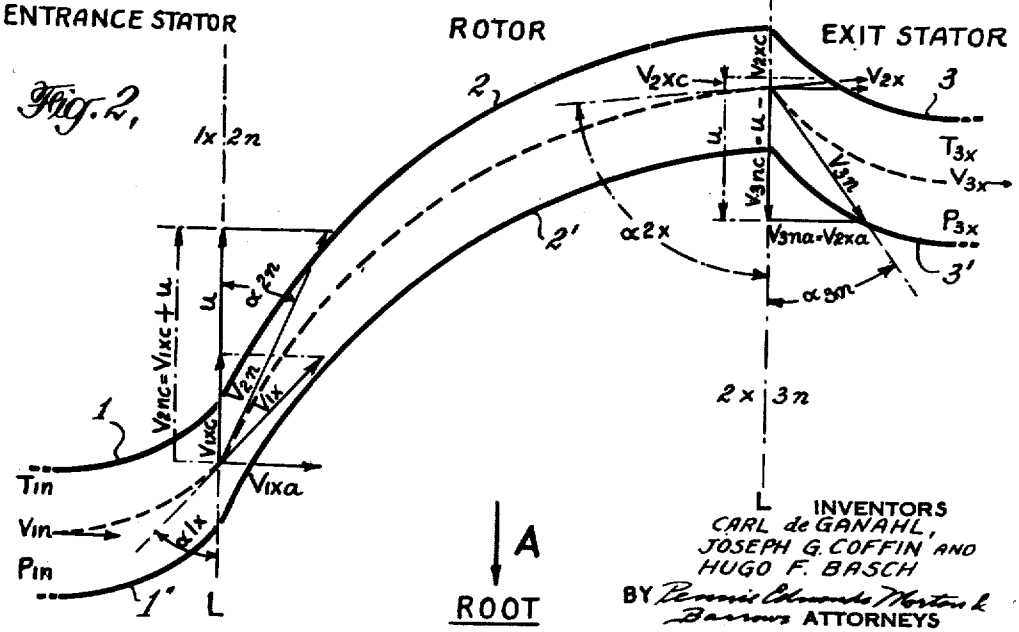


Fig. 2,



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AXIAL FLOW COMPRESSOR

Filed June 10, 1946 4 Sheets-Sheet 2

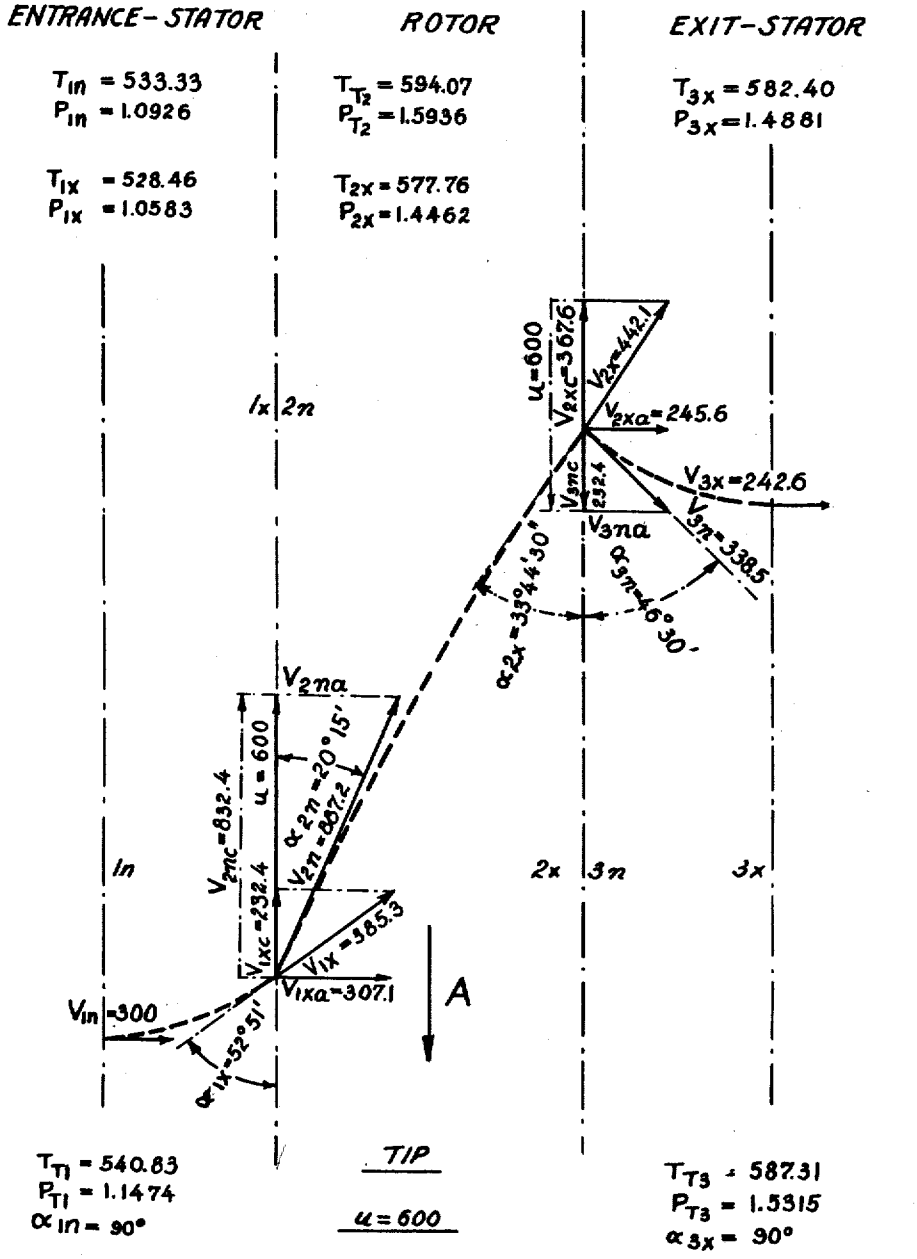


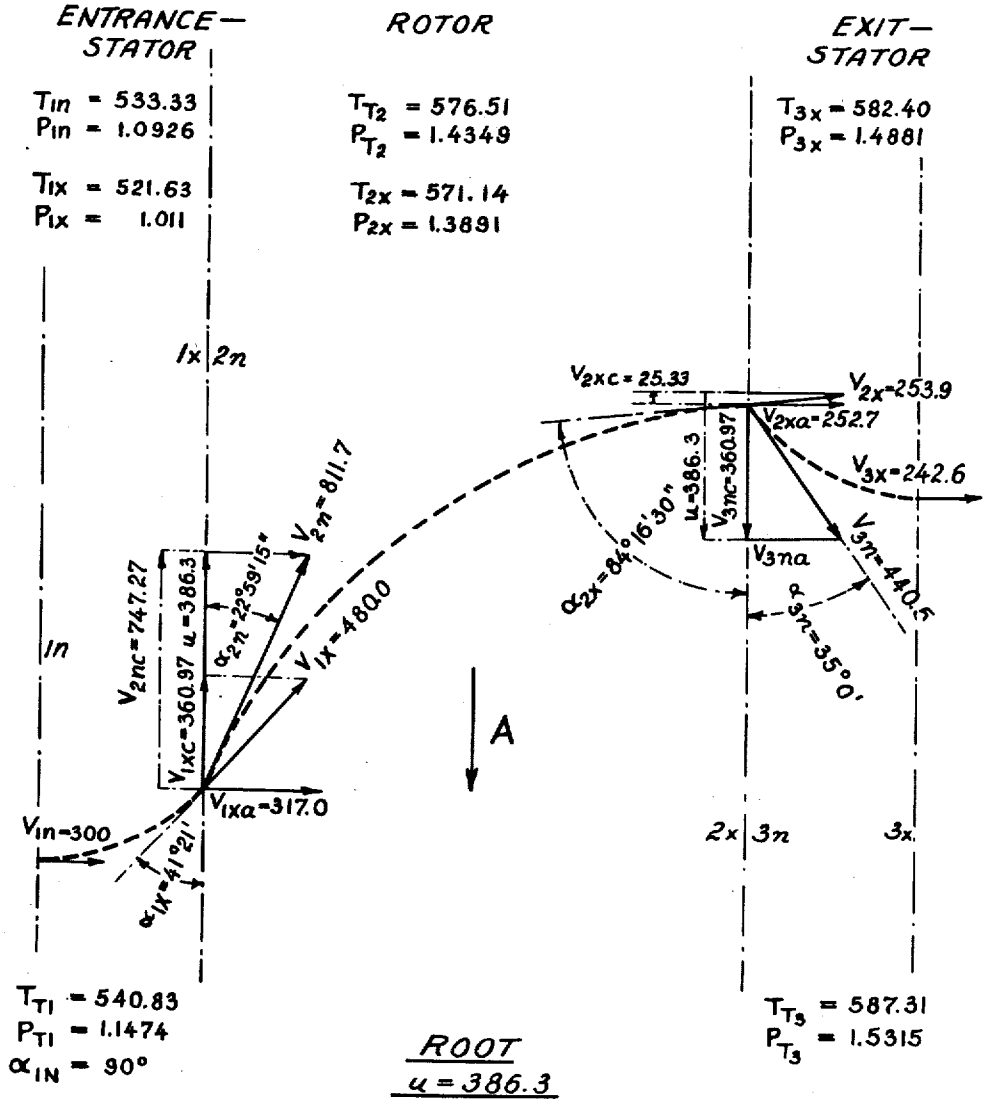
Fig. 3,

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AXIAL FLOW COMPRESSOR

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ROOT

u = 386.3

Fig. 4,

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AXIAL FLOW COMPRESSOR

Filed June 10, 1946 4 Sheets-Sheet 4

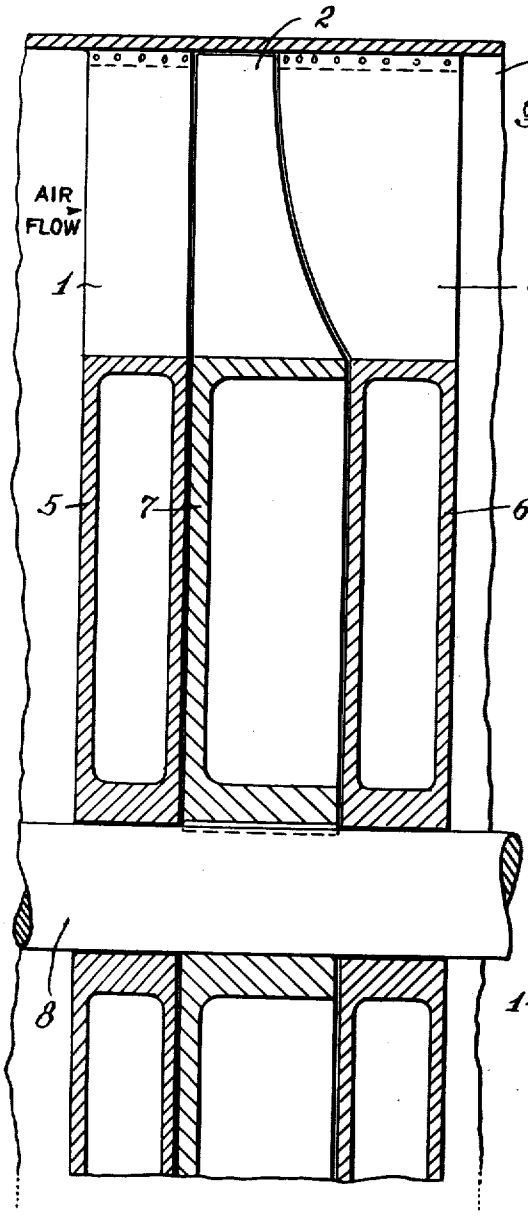


Fig. 5,

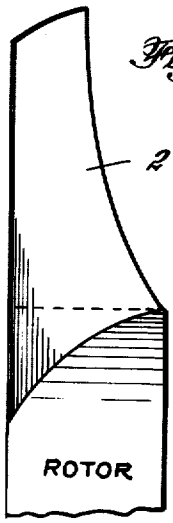


Fig. 6,

Fig. 7,

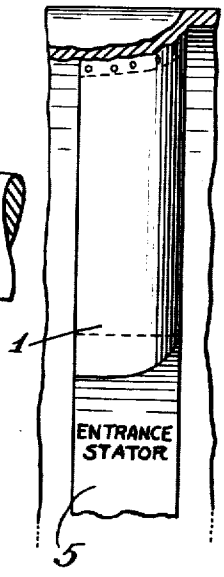
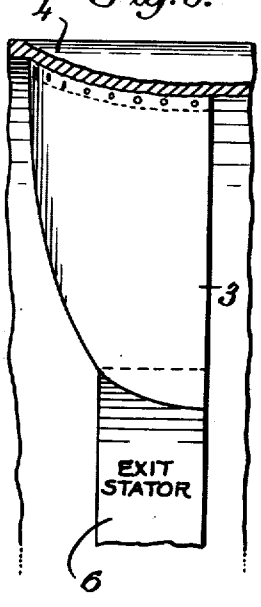


Fig. 8.



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Patented May 2, 1950

2,505,755

UNITED STATES PATENT OFFICE

2,505,755

AXIAL FLOW COMPRESSOR

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Application June 10, 1946, Serial No. 675,626

10 Claims. (Cl. 230—122)

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This invention relates to axial flow compressors for compressing air or other compressible fluid.

A single stage axial flow compressor, as distinguished from a centrifugal compressor, consists essentially of a compressor casing and a rotor comprising a central shaft or hub carrying a set of vanes or blades, the casing carrying a set of stationary blades constituting entrance stator blades and an additional set of blades which may be characterized as exit stator blades, the air flow being in a generally axial direction. A single stage compressor includes only one set of rotor blades associated with a set of entrance stator blades and with a set of exit stator blades. In a multistage compressor there are several sets of rotor blades and corresponding sets of stator blades.

This invention relates to improvements in axial flow compressors, the compressor being either a single stage or a multistage compressor. The principal objects of this invention include the provision of rotor and stator blades forming fluid ducts for conducting the fluid in a generally axial direction through the compressor, and the provision of blades of this character so formed or shaped that the compressed fluid is directed axially in the exit stator with the fluid pressure substantially constant from the root to the tip portions of the blades, the configuration of the compressor blades also being such that the fluid velocity does not have any radial component at any point. Also the pressure of the fluid as it is directed axially in the exit stator may have a value that is substantially constant from the root to the tip portions of the exit stator blades. Our invention makes it possible to provide an axial flow compressor having numerous advantages, including the following:

a. The compressed fluid, as it flows axially in the exit stator ducts, has substantially the same pressure at all radially spaced points between root and tip portions of the blades forming the exit stator ducts and accordingly the tendency to cross current flow from a relatively high pressure region to a relatively low pressure region can be avoided.

b. The over-all dimensions may be reduced for a given air capacity and pressure rise.

c. Only a few stages of compression need be employed to produce the required increase in pressure; pressure ratios per stage may be in the order of 1.5 to 1.6 with high efficiencies approaching 90% or better.

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d. The blades can be made much longer for a given diameter, thus greatly increasing the mass flow for a given diameter rotor.

e. The rotational speed of the rotor is greatly reduced, thereby reducing the stored energy in the rotor and the tension forces in the material.

f. The weight and cost of the compressor can be reduced to a minimum because of simplification of manufacture and the fact that the over-all dimensions may be reduced to a minimum for a given capacity.

The design of axial flow compressors has heretofore been based almost entirely on airfoil theory which involves two assumptions, i. e., that the fluid is incompressible, or in other words the density is constant throughout a stage, and that the axial fluid velocity is constant through the various stages.

In the conventional type of axial flow compressor, the energy put into the air by the rotating blades produces a pressure rise in the rotor and also imparts whirl velocity to the air, this whirl velocity being converted into a further pressure rise by the stator blades. The conventional type of axial flow compressor is based on airfoil theory, airfoils being used for both the rotating blades and the stator blades. At the tip the velocities should be kept below the values which would produce shock waves or local sonic velocities. At the root, stalling of the blades must be avoided, and accordingly the lift coefficient that can be applied at the root is limited. Since the pressure rise at the root should be kept as nearly equal as possible to the pressure rise at the tip, the limit of the pressure rise available is determined by the stalling of the blades at the root, and, for high efficiencies, by operating the blades at lift coefficients with low drag values. These two factors restrict the length of the blade that can be used and likewise the capacity of the machine. In the entrance stator ahead of the first rotor stage, the flow of air is directed with the direction of motion of the rotor in order to reduce the velocity relative to the rotor blades and also to approach symmetry of velocity diagrams in the rotor and the stator blades; that is to enable the rotor to add an amount of whirl velocity equal to that which can be taken out by the succeeding stator blades.

We have found that a greatly improved axial flow compressor can be made if the basic assumptions of the airfoil theory (i. e., that the fluid is incompressible and that the axial fluid velocity is constant for the various stages) are

ignored and the rotor and stator blades are constructed or formed in the manner hereinafter described whereby substantially constant pressure is obtained at all radially spaced points between the root and tip portions of the blades where the fluid flows axially, the blades also being preferably formed so that the fluid velocity does not have any radial component at any point.

In general, the principal object of this invention may be attained by forming the rotor and stator blades so that the same amount of energy per unit of compressible fluid is transmitted by the rotor to the fluid at the root portion of the rotor blades as at the tip portion of the rotor blades, even through the circumferential speed of the rotor blade tips is substantially greater than the circumferential speed of the root portions of the rotor blades. The required configuration of the rotor and stator blades to accomplish this purpose may be determined with the highest degree of precision by first selecting appropriate operating conditions for the entrance to the compressor, and then ascertaining by means of the established thermodynamic equations the proper exit angle for all entrance stator ducts and for all rotor ducts at all concentric zones or regions between the root and tip portions of the blades.

The amount of energy imparted to the fluid by the rotor is a function of the circumferential speed of the rotor and of the velocity of the fluid entering the rotor, and according to this invention the entrance stator blades and the rotor blades are shaped so as to provide progressively decreasing rotor entrance velocity from root to tip to compensate as completely as may be desired for the increase in circumferential rotor speed between the root and tip of the rotor blades, whereby the root portions of the rotor blades, travelling at relatively low circumferential speed, may impart to the fluid entering the rotor at higher velocity than at the tip, the same amount of energy, the exit portions of the rotor blades being correspondingly shaped to insure this result at the rotor exit so that as the compressed fluid flows axially in the exit stator, it may have the same pressure at all radially spaced points in the exit stator.

In a compressor embodying this invention, there is a pressure stage, or a multiplicity of pressure stages, in which each stage is a combination of an entrance stator, a rotor, and exit stator. In succeeding stages, the entrance stator to the next stage becomes a part of or is contiguous with the exit stator of the preceding stage. The flow is axial between stages, i. e., the flow is purely axial as the fluid enters the entrance stator from the exit stator. The rotor blades and stator blades are preferably made of thin sheet metal and these blades are not airfoils. The spaces between the blades form channels which are either converging to form nozzles or diverging to form diffusers, as hereinafter explained. The cross sectional area of the channel or duct is proportional to the sine of the angle the blade makes with a plane normal to the rotor axis. The solidity, i. e., the circumferential spacing of the blades, determines the angle of divergence or the angle of convergence of the duct, depending on whether the duct is a diffuser or a nozzle.

In the axial flow compressor of our invention, the entrance stator directs the fluid against the direction of rotation of the rotor rather than with this direction of rotation as in the conventional axial flow compressor, thereby increasing the relative velocity between the rotor blades and the entering air thus greatly reducing the required speed of the rotor as compared with the conventional compressor.

The angle of the blades at the rotor entrance is adjusted to be streamlined with the flow from the entrance stator. The channels formed by the blades in the rotor are diffusers having the greatest area ratio at the root and the smallest at the tip.

According to one embodiment of our invention, the circumferential velocity at the exit of the entrance stator and the circumferential velocity at the entrance of the exit stator can be kept equal.

The angle between the airflow leaving the entrance stator and the plane normal to the rotor axis is greatest at the tip and decreases toward the root. This progressive change in angle tends to increase the relative velocity between the rotor and the entering airflow from tip to root, compensating properly for the decreased rotor velocity toward the root of the blades. The entrance stator and the rotor are so formed that the rotor puts into the air the same amount of energy at the tip as it does at the root so that as the fluid is directed axially in the exit stator, there will be the same pressure and velocity from tip to root. To accomplish this end, the direction of the airflow is bent through a lesser angle of arc at the tip than at the root. This increases the ratio between exit and entrance area at each radius of the rotor towards the root, again properly compensating for the decreased rotor velocity at the root. It also increases the amount of whirl velocity delivered to the airflow at the root, further compensating for the decreased rotor velocity at the root.

The entrance to the exit-stator is adjusted to be streamlined with the flow from the rotor at each radius, and is bent through an arc to pure axial flow on leaving the exit stator. The angle between the airflow entering the exit stator and a plane normal to the axis of the compressor is largest at the tip, and decreases towards the root, providing the necessary difference of ratio between exit area and entrance area at each radius to convert the different velocities into pressure.

The general conditions above described can be repeated through succeeding stages.

In the accompanying drawings, we have illustrated in diagrammatic form, elements of an axial flow compressor embodying our invention. In these drawings:

Fig. 1 represents a development of a cylindrical section taken at or near the tip portion of the stator and rotor blades;

Fig. 2 represents a similar development of a cylindrical section taken at the root of the stator and rotor blades;

Fig. 3 is a vector diagram showing typical fluid velocity conditions obtaining at several points in a duct formed by the tip portions of the rotor and stator blades of a compressor representing one embodiment of our invention;

Fig. 4 is a vector diagram showing typical fluid velocity conditions obtaining at several points in a duct formed by the root portions of the rotor and stator blades of the embodiment of our invention to which Fig. 4 is applicable;

Fig. 5 is a longitudinal section of a portion of a single stage compressor embodying our invention;

Fig. 6 is an elevation of a rotor blade; and

Figs. 7 and 8 are fragmentary elevations of stator blades of a compressor embodying our invention.

In Figs. 1 and 2 of the accompanying drawings, a pair of entrance stator blades are shown at 1 and 1', a pair of rotor blades at 2 and 2' and a pair of exit stator blades at 3 and 3'. These Figs. 1 and 2 thus illustrate in diagrammatic form a single stage of an axial flow compressor and it will be understood that in a multistage compressor, the exit stator blades are extended to form the entrance stator for the next succeeding set of rotor blades. Figs. 1 and 2 show, respectively, the tip and root circumferential sections of the same pairs of stator and rotor blades and accordingly the same reference characters for the blades are used in both of these figures. The stator and rotor blades are preferably made of thin sheet metal stampings, the sheet metal being of uniform thickness throughout. All blades in any entrance stator are identical in shape and size and would accurately fit each other if placed close together. This is also true for any rotor set or any exit stator set.

Figs. 5–8, inclusive, show an entrance stator blade 1, a rotor blade 2 and an exit stator blade 3. The stator blades 1 and 3 have their tip portions secured to an outer casing 4 and their root portions secured to stationary members 5 and 6. The rotor blades 2, 2', etc., are secured to a hub 7 fixed to the compressor shaft 8.

The rotor and stator blades define ducts extending from the root to the tip and also extending in a generally axial direction through the rotor and stator blades. Each set of rotor blades and stator blades consists of blade elements bent to form ducts or channels and because of the fact that all of the blades in each set are identical in shape, consecutive blades are "parallel" in the sense that they are equidistant at all points from each other as measured along a circumferential direction. The cross sectional flow areas of the ducts formed by the blades are at all points proportional to the sine of the angle that the duct axis makes with a plane normal to the axis of the rotor.

The entrance stator blades 1 and 1' are shaped to form a nozzle or in other words, so that the cross sectional area normal to the duct axis at the exit of the entrance stator is smaller than the corresponding cross sectional area at the entrance to the entrance stator. The duct defined by each pair of rotor blades, as illustrated in the accompanying drawings, is a diffuser for the cross sectional area normal to the duct axis at the rotor exit is larger than the cross sectional area normal to the duct axis at the rotor entrance. Each pair of exit stator blades likewise defines a diffuser and these exit stator blades are shaped so that they direct the fluid axially.

The thermodynamic conditions of flow between two curved blades that are parallel in the sense above defined are such that the fluid temperature, pressure, and velocity conditions at the entrance and exit portions of the duct are determined solely by the ratio of the cross sectional areas normal to the duct axis at the entrance and exit respectively, if the shape of the ducts is such as to maintain streamlined flow between the entrance and exit. The angle of divergence (or convergence) between any pair of consecutive blades of the form described above is determined entirely by the circumferential distance between these blades, this distance being constant throughout the length (axially) of any pair of blades as explained above. Thus the cross sectional area of the duct defined by any pair of blades is a function of the shape of the blades,

whereas, the angle of divergence (or convergence) is a function of the circumferential distance between the blades and these two factors are entirely independent of each other.

In Figs. 1 and 2, the air or other compressible fluid is shown entering the entrance stator duct in a direction parallel to the axis of the rotor, the air at this point having a temperature T_{1n}, a vuelocity V_{1n} and a pressure P_{1n}. The air leaves the entrance stator nozzle with a velocity V_{1x} which may be considered as having an axial component V_{1xa} and a circumferential component V_{1xc}. It will be understood that the rotor section or element illustrated in Figs. 1 and 2 is moving in the direction indicated by the arrow A, with respect to the stationary entrance stator and exit stator, and in Fig. 1 the arrow u represents the reverse of the circumferential velocity or speed of the rotor tips. The velocity of the air entering the rotor, V_{2n}, is the vectorial sum of the velocity V_{1x} and the reverse of the rotor speed u. The air leaves the entrance stator nozzle at an angle with respect to a plane normal to the rotor axis, this angle being designated α_{1x} in Fig. 1 and it will be noted that the direction of the air entering the rotor duct at the rotor entrance, as indicated by the arrow V_{2n}, is parallel to the rotor blades at this point, thus providing streamlined flow into the rotor. In other words, as the air enters the rotor duct which is travelling at high speed in the direction indicated by the arrow A in Fig. 1, the air is not immediately subjected to any impact by the rotor blades. As the air progresses in the rotor duct, however, its direction of flow is progressively changed by the curved rotor blades, the air being discharged from the rotor duct in a direction indicated by the arrow V_{2x} (relative to the rotor). The air discharged from the rotating rotor ducts enters the stationary ducts formed by the exit stator blades and the direction of flow into the exit stator is represented by the arrow V_{3n} which represents the vector sum of the velocity V_{2x} and the rotor circumferential velocity u. It will be noted that the air entering the exit stator in the direction indicated by the arrow V_{3n} flows parallel to the exit stator blades at this point, thus providing streamlined flow into the exit stator. It will also be noted that the air entering the exit stator may be said to have a circumferential velocity component V_{3nc} and an axial component V_{3na}.

To illustrate the manner of selecting appropriate operating conditions, it may be assumed that the compressor is in an airplane traveling at V_0=500 feet per second; that the temperature of the air is T_0=520° R. (where °R. denotes °Rankine or °Fahrenheit absolute); and that the static pressure is P_0=1 atm. (2116 lbs. per square foot). It may be assumed that under these conditions, the air will enter the compressor at a velocity V_{1n} = 300 feet per second, and in any event the mouth or entrance of the compressor may be designed or adjusted so that the velocity of the air entering the compressor will be 300 feet per second when the plane is travelling at a speed in the neighborhood of 500 feet per second. An appropriate value is selected for the tip diameter of the rotor. Thus, for purposes of illustration, it may be assumed that the tip diameter is D=54", that the root diameter is 34.77" and the R. P. M. or N is 2543 revolutions per minute. With the tip and root diameters determined, the R. P. M. of the rotor is such as to produce a Mach number of air flow of approximately .788 entering the rotor tip, a satisfactory value.

The entrance stator blades may have their tip portions formed so that in this region the entrance stator ducts are shaped to form nozzles directing the air against the direction of movement of the rotor. The rotor blades will have a circumferential velocity

$$u=\frac{2\pi Nr}{60} \text{ ft. per sec.}$$

$$u=\frac{2\pi\times 2543\times 27}{60\times 12}=600 \text{ ft./sec.} \quad (1)$$

where N represents the rotor R. P. M. and r the radius of the compressor rotor at the tip portion of the rotor blades.

The total temperature T_{T1} at the entrance to the entrance stator is

$$T_{T1}=T_0+\frac{\overline{V_0}^2}{K}=520+\frac{\overline{500}^2}{12{,}000}=540.83^\circ \text{ R.} \quad (2)$$

This equation is derived from the conservation of energy equation and the constant K equals $2JgCp$, or 12,000, J being equal to 778, the mechanical equivalent of heat in foot pounds per B. t. u.; g equals 32.2 the acceleration of gravity in feet per second per second, and Cp equals .24 which is the specific heat of air at constant pressure in B. t. u. per °F. per lb. of air.

The total pressure P_{T1} is found from

$$\left(\frac{P_{T1}}{P_0}\right)=\left(\frac{T_{T1}}{T_0}\right)^{3.5} \quad (3)$$

where

$$3.5=\frac{k}{k-1}$$

k being the ratio

$$\frac{Cp}{Cv}$$

or 1.4, for air, and therefore

$$P_{T1}=1\times\left(\frac{540.83}{520.00}\right)^{3.5}=1.1474 \text{ atm.}$$

The actual temperature of the air entering the entrance stator, T_{1s}, is

$$T_{1s}=T_{T1}-\frac{\overline{300}^2}{K}=540.83-7.50 \quad (4)$$

$$=533.33 \text{ °R}$$

The actual pressure of the air entering the entrance stator, P_{1s}, is

$$P_{1s}=P_0\left(\frac{T_{1s}}{T_0}\right)^{3.5}=1\times\left(\frac{533.33}{520.00}\right)^{3.5}$$

$$=\left(1+\frac{13.33}{520}\right)^{3.5}=(1+.02563)^{3.5} \quad (5)$$

$$=1.0926 \text{ atms.}$$

With these values of T_{1s} and P_{1s} and V_{1s}, the airflow in pounds per second per square foot of entrance-stator entrance area is

$$W=\frac{A_{1s}V_{1s}P_{1s}}{RT_{1s}}$$

$$=\frac{1\times 300\times 1.0926\times 2116}{53.3\times 533.33} \quad (6)$$

$$=24.40 \text{ lbs./sq. ft./sec.}$$

For known or assumed values of T_{T1}, the total temperature of the entering air, and T_{T3}, the total temperature of the outgoing air, the angles α_{1x}, α_{2n}, α_{2x} and α_{3n} determine the temperatures, pressures and velocities (and their components) at the corresponding points in the compressor.

The circumferential velocity component of the air leaving the entrance stator is

$$V_{1xc}=\frac{(T_{T3}-T_{T1})K}{2u}-\frac{u}{2} \quad (A)$$

This can be proved by combining the energy-equations

$$T_{T3}=T_{1s}+\frac{\overline{V_{2n}}^2}{K} \quad (7)$$

and

$$T_{T1}=T_{1s}+\frac{\overline{V_{1s}}^2}{K} \quad (8)$$

By subtraction,

$$(T_{T3}-T_{T1})K=\overline{V_{2n}}^2-\overline{V_{1s}}^2 \quad (9)$$

With the geometry in Fig. 1

$$\overline{V_{2n}}^2=\overline{V_{1s}}^2+u^2+2uV_{1s}\cos\alpha_{1s}$$

and

$$V_{1x}\cos\alpha_{1x}=V_{1xc}$$

so that

$$\overline{V_{2p}}^2=\overline{V_{1s}}^2+u^2+2uV_{1xc} \quad (10)$$

substituting in (9)

$$(T_{T2}-T_{T1})K=u^2+2uV_{1xc}$$

solving for V_{1xc} gives

$$V_{1xc}=\frac{(T_{T3}-T_{T1})K}{2u}-\frac{u}{2} \quad (A)$$

Likewise the circumferential velocity component of the air entering the exit stator is

$$V_{3nc}=\frac{(T_{T3}-T_{T2})K}{2u}+\frac{u}{2} \quad (B)$$

The derivation is similar to that above proved but using the energy equations

$$T_{T3}=T_{2x}+\frac{\overline{V_{3n}}^2}{K} \quad (11)$$

$$T_{T2}=T_{2x}+\frac{\overline{V_{2x}}^2}{K} \quad (12)$$

by subtraction

$$(T_{T3}-T_{T2})K=\overline{V_{3n}}^2-\overline{V_{2x}}^2 \quad (13)$$

since with the geometry in Fig. 1

$$\overline{V_{3n}}^2=\overline{V_{2x}}^2+u^2-2uV_3x\cos\alpha_{2x}$$

and

$$V_{2x}\cos\alpha_{2x}=V_{2xc}$$

so that

$$\overline{V_{3n}}^2=\overline{V_{2x}}^2+u^2-2uV_{2xc} \quad (14)$$

By substituting in (13)

$$(T_{T3}-T_{T2})K=u^2-2uV_{2xc}$$

solving for V_{2xc} gives

$$V_{2xc}=\frac{u}{2}-\frac{(T_{T3}-T_{T2})}{2u}K \quad (15)$$

and as $V_{3nc}=u-V_{2xc}$

$$V_{3nc}=\frac{(T_{T3}-T_{T2})}{2u}K+\frac{u}{2} \quad (B)$$

Adding the Equations A and B we find:

$$uV_{1xc}+uV_{3nc}=\frac{(T_{T3}-T_{T1})K}{2} \quad (C)$$

or

$$V_{1xc}\left(1+\frac{V_{3nc}}{V_{1xc}}\right)=\frac{(T_{T3}-T_{T1})K}{2u} \quad (C')$$

It follows from Equation C that according to our invention, the sum of uV_{1xc} and uV_{3nc} is a

constant for any predetermined values of T_{T1} and T_{T3}, this sum being exactly commensurate with the difference between the total temperature of the outgoing air T_{T3} and the total temperature of the entering air T_{T1}.

If values for T_{T1}, T_{T3} and u are assumed, as indicated above, it follows from Equations A and B that the values of V_{1xc} and V_{3nc} depend upon the value of T_{T2} alone. The value of T_{T2} may vary from below T_{T1} to above T_{T3} and accordingly there are an infinite number of possible thermodynamic solutions.

When $V_{1xc}=V_{3nc}$ then

$$V_{1xc}=\frac{(T_{T2}-T_{T1})K}{2u}-\frac{u}{2}=V_{3nc}=\frac{(T_{T3}-T_{T2})K}{2u}+\frac{u}{2}$$

and solving for T_{T2}

$$T_{T2}=\frac{T_{T3}+T_{T1}}{2}+\frac{u^2}{K} \qquad (16)$$

and substituting T_{T2} into V_{1xc}

$$V_{1xc}=\frac{(T_{T3}-T_{T1})K}{4u}=V_{3nc} \qquad (17)$$

Whether or not V_{1xc} is equal to V_{3nc}, then, as indicated above (see Equations A, B and C), the sum of uV_{1xc} and uV_{3nc} must remain a constant and V_{1xc} and V_{3nc} depend upon the value of T_{T2} alone.

As stated above, the entrance stator blades and rotor blades provide progressively decreasing rotor entrance velocities from root to tip to compensate as completely as may be desired for the increase in the circumferential rotor speed (u) between root and tip of the rotor blades, whereby the root portions of the rotor blades travelling at relatively low circumferential speed may impart to the fluid entering the rotor at higher absolute velocity than at the tip, the same amount of energy, the exit portions of the rotor blades being correspondingly shaped to insure this result at the rotor exit, so that as the compressed fluid flows axially in the exit stator, it may have the same pressure at all radial points in the exit stator.

The energy equation

$$uV_{1xc}+uV_{3nc}=\frac{(T_{T3}-T_{T1})K}{2} \qquad (C)$$

shows the energy relation above described, uV_{1xc} being the energy relation at the rotor entrance and uV_{3nc} that at the rotor exit.

To have the same energy imparted to the fluid entering the rotor from root to tip we must have

$$uV_{1xc}\Big|_r^t=c_1$$

and

$$uV_{3nc}\Big|_r^t=c_2$$

Thus

$$\frac{uV_{1xc}}{uV_{3nc}}\Big|_r^t=\frac{V_{1xc}}{V_{3nc}}\Big|_r^t=c_3 \text{ (a constant)} \qquad (D)$$

Also, as stated above, the air flow in pounds per second per square foot of entrance stator entrance area is

$$W=\frac{A_{1n}V_{1n}P_{1n}}{RT_{1n}} \qquad (6)$$

Thus the air flow at the entrance of the entrance stator is independent of the radius, or in other words, it is the same at all radial points of the entrance of the entrance stator. This means that the flow of air is vortex free as it enters the compressor. It is well known that if the flow of air is vortex free, i. e., irrotational (assuming an ideal compressible fluid), it remains vortex free throughout its motion. Accordingly, in the compressor of our invention the flow of air remains vortex free as it passes through the compressor (assuming ideal conditions). It follows from this fact that in the compressor the product of the circumferential component of the fluid velocity at the exit of the entrance stator (V_{1xc}) and the circumferential velocity of the rotor (u) is constant at all radial points between the root and the tip. Likewise the product of the circumferential component of the fluid velocity at the entrance of the exit stator (V_{3nc}) and the circumferential velocity of the rotor (u) is constant at all radial points from root to tip. Hence the ratio of

$$\frac{V_{1xc}}{V_{3nc}}$$

is constant from root to tip. Where $V_{1xc}=V_{3nc}$, then this ratio is 1. However, when a value of T_{T2} is chosen which results in values of V_{1xc} and V_{3nc} that are not equal, then the ratio of V_{1xc} to V_{3nc}, of course, has a value other than 1.

It will be noted that in a typical embodiment of our invention (see, for example, Figs. 3 and 4), the ratio

$$\frac{V_{1xc}}{V_{3nc}}$$

is the same at all points between the root and the tip. Thus in Figs. 3 and 4 this ratio at the root is

$$\frac{V_{1xc}}{V_{3nc}}=\frac{360.97}{360.97}=1$$

and at the tip

$$\frac{V_{1xc}}{V_{3nc}}=\frac{232.4}{232.4}=1$$

Also, at the root (Figs. 3 and 4):

$$uV_{1xc}=386.3\times 360.97=139443$$

and at the tip:

$$uV_{1xc}=600\times 232.4=139440$$

The choice of the ratio

$$\frac{V_{1xc}}{V_{3nc}}$$

in any specific case, at any given radius, determines the kind of compressor desired, i. e., whether it is of the more or less reactive type. If

$$\frac{V_{1xc}}{V_{3nc}}$$

is large, the compressor is mostly reactive, if small, it is less reactive. In any event, we prefer to have the blades formed so that

$$\frac{V_{1xc}}{V_{3nc}}$$

is constant from root to tip, for this condition prevents cross flow (i. e. radial flow) at all points.

The constancy of the ratio

$$\frac{V_{1xc}}{V_{3nc}}$$

in any one compressor design shows that

$$\left(T_{T2}-\frac{u^2}{K}\right)$$

must be constant from root to tip as is easily seen by taking the ratio of (A) to (B), giving

$$\frac{V_{1ae}}{V_{3ae}}=\frac{\left(T_{T2}-\frac{u^2}{K}\right)-T_{T1}}{T_{T2}-\left(T_{T2}-\frac{u^2}{K}\right)} \qquad (D')$$

or, solving,

$$T_{T2}-\frac{u^2}{K}=\frac{\frac{V_{1ae}}{V_{3ae}}T_{T2}+T_{T1}}{1+\frac{V_{1ae}}{V_{3ae}}}$$

Thus the values of V_{1ae}, V_{3ae} and T_{T2} are determinate for any radial calculation of the required angles.

The value of α_{1x} for any radial point is given by

$$\alpha_{1a}=\tan^{-1}\left\{\frac{KT_{T1}}{\overline{V_{1ae}}^2}\left(1-\frac{T_{1a}}{T_{T1}}\right)-1\right\}^{1/2} \qquad (F)$$

where

$$\frac{T_{1a}}{T_{T1}}$$

is the larger positive root of

$$\left(\frac{T_{1a}}{T_{T1}}\right)^6-\left(\frac{T_{1a}}{T_{T1}}\right)^5\left(1-\frac{\overline{V_{1ae}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

To derive this equation, we have

$$P_{1a}=P_{T1}\left(\frac{T_{1a}}{T_{T1}}\right)^{3.5} \text{ (for air)}$$

the adiabatic relation between pressure and temperature (see Equation 3)

$$\overline{V_{1a}}^2=K(T_{T1}-T_{1a})=KT_{T1}\left(1-\frac{T_{1a}}{T_{T1}}\right) \qquad (8)$$

$$V_{1ae}=\sqrt{\overline{V_{1a}}^2-\overline{V_{1ae}}^2}$$

as shown in Figs. 1 and 2.

Substituting Equation 8 for

$$\overline{V_{1a}}^2$$

we have

$$V_{1ae}=\sqrt{KT_{T1}\left(1-\frac{T_{1a}}{T_{T1}}\right)-\overline{V_{1ae}}^2} \qquad (18)$$

Using the continuity equation

$$W=\frac{AVP}{RT} \text{ (see Equation 6)}$$

$$W=\sin\frac{\alpha_{1a}V_{1a}P_{1a}}{RT_{1a}}$$

$$W=\frac{V_{1ae}P_{1a}}{RT_{1a}}$$

because $V_{1ae}=V_{1x}\sin\alpha_{1x}$

Substituting the above values of P_{1x} V_{1xa} in this last equation, squaring, and rearranging, we have

$$\left(\frac{T_{1a}}{T_{T1}}\right)^6-\left(\frac{T_{1a}}{T_{T1}}\right)^5\left(1-\frac{\overline{V_{1ae}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0 \qquad (G)$$

Also since

$$\frac{V_{1ae}}{V_{1ae}}=\tan\alpha_{1a} \text{ (see Fig. 1)}$$

and dividing Equation 18 by V_{1ae} we have

$$\tan\alpha_{1a}=\sqrt{\frac{KT_{T1}}{\overline{V_{1ae}}^2}\left(1-\frac{T_{1a}}{T_{T1}}\right)-1}$$

so that α_{1x}= the $\tan^{-1}$

$$\left\{\frac{KT_{T1}}{\overline{V_{1ae}}^2}\left(1-\frac{T_{1a}}{T_{T1}}\right)-1\right\}^{1/2} \qquad (F)$$

Similarly we find:

$$\alpha_{3a}=\tan^{-1}\left\{\frac{KT_{T2}}{(u-V_{3ae})^2}\left(1-\frac{T_{3a}}{T_{T2}}\right)-1\right\}^{1/2} \qquad (H)$$

where

$$\frac{T_{3a}}{T_{T2}}$$

is the larger positive real root of

$$\left(\frac{T_{3a}}{T_{T2}}\right)^6-\left(\frac{T_{3a}}{T_{T2}}\right)^5\left(1-\frac{(u-V_{3ae})^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0 \qquad (J)$$

where T_{T2} is known from Equation D and the arbitary chosen ratio

$$\frac{V_{1ae}}{V_{3ae}}$$

for the compressor. The smaller root involves supersonic flow.

Again similarly we find

$$\alpha_{1a}=\tan^{-1}\left\{\frac{KT_{T2}}{(u+V_{1ae})^2}\left(1-\frac{T_{1a}}{T_{T2}}\right)-1\right\}^{1/2} \qquad (K)$$

where

$$\frac{T_{1a}}{T_{T2}}$$

is the larger positive real root of

$$\left(\frac{T_{1a}}{T_{T2}}\right)^6-\left(\frac{T_{1a}}{T_{T2}}\right)^5\left(1-\frac{(u+V_{1ae})^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0 \qquad (L)$$

We have also

$$\alpha_{3a}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{3ae}}^2}\left(1-\frac{T_{3a}}{T_{T2}}\right)-1\right\}^{1/2} \qquad (M)$$

where

$$\frac{T_{3a}}{T_{T2}}$$

is the larger positive real root of

$$\left(\frac{T_{3a}}{T_{T2}}\right)^6-\left(\frac{T_{1a}}{T_{T2}}\right)^5\left(1-\frac{\overline{V_{3ae}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0 \qquad (N)$$

The angle α_{1x} and the corresponding rotor angle α_{3x} producing streamlined flow into the rotor, are seen to be related by the equation

$$\tan\alpha_{3a}=\frac{V_{1ae}}{u+V_{1ae}}\tan\alpha_{1a} \qquad (P)$$

because from Fig. 1 we see that

$$\tan\alpha_{1a}=\frac{V_{1ae}}{V_{1ae}}$$

and

$$\tan\alpha_{3a}=\frac{V_{1ae}}{u+V_{1ae}}$$

Hence dividing,

$$\frac{\tan\alpha_{3a}}{\tan\alpha_{1a}}=\frac{V_{1ae}}{u+V_{1ae}}$$

Similarly

$$\tan\alpha_{3a}=\frac{V_{3ae}}{u-V_{3ae}}$$

$$\tan\alpha_{3a}=\frac{V_{3ae}}{V_{3ae}}$$

and

$$\frac{\tan\alpha_{3a}}{\tan\alpha_{3a}}=\frac{V_{3ae}}{u-V_{3ae}} \qquad (Q)$$

In the above Equations G, J, L and N, the exponents are simple whole numbers (6 and 5) be-

cause of the value used for k. The value used for k is $k=1.4$ which is the ratio of

$$\frac{C_p}{C_v}$$

for air. However, if k is not equal to 1.4, there will be a similar equation. The exponent

$$6=\frac{k+1}{k-1}$$

and the exponent

$$5=\frac{2}{k-1}$$

Of the two real positive roots, one is larger than

$$\frac{T_{1s}}{T_{T1}}=\frac{5}{6}$$

and one less than this value. The lesser one involves a supersonic value of T_{1x} which is extraneous to the subject matter of this invention. The greater root will come out to be greater than ⅚ which is the subsonic value for T_{1x}.

The angles $\alpha_{1n}=90°$, α_{1x}, α_{2n}, α_{2x}, α_{3n} and $\alpha_{3x}=90°$ thus determined will insure a constant flow across the compressor from root to tip of the blade without cross flow, and that the pressure at the exit of the exit stator is constant from root to tip.

The determination of the pressure, temperature, and velocity conditions obtaining at the exits X_1, X_2 and X_3, assuming streamlined flow, can be made by employing the usual thermodynamic relations. The ordinary method of making such computations is cumbersome because it involves a trial and error procedure. The solution can be simplified by utilizing a "B" table giving the ratio of the total pressures to the pressures at the exits, as

$$\frac{P_{T1}}{P_{1x}} \quad \text{or} \quad \frac{P_{T2}}{P_{2x}} \quad \text{or} \quad \frac{P_{T3}}{P_{3x}}$$

for various assumed values of T_T, P_T and outlet areas A_{1x} and A_{2x} and A_{3x}.

To find the conditions at the exit of the entrance stator

$$B_{1x}=\frac{WR\sqrt{T_{T1}}}{A_{1x}\sqrt{K}P_{T1}}$$

is used. ($A_{1x}=\sin\alpha_{1x}$) where W is the entrance air flow in pounds per square foot per second, R is the gas constant for 1 pound of air, i. e., 53.3, and K is the value of $2JgC_p=12{,}000$. T_{T1} is the total temperature of the air in the duct and is the same throughout the duct; P_{T1} is the total pressure in pounds per square foot.

Knowing

$$\frac{P_{T1}}{P_{1x}}$$

which is the corresponding value to B_{1x}

$$P_{1x}=\frac{P_{T1}}{P_{T1}/P_{1x}}=P_{2n}$$

and assuming an adiabatic process

$$T_{1x}=T_{T1}\left(\frac{P_{1x}}{P_{T1}}\right)^{\frac{k-1}{k}}=T_{2n}$$

The equation of conservation of energy

$$\frac{V_{1x}^2}{2g}+J_{cp}T_{1x}=\frac{0}{2g}+J_{cp}T_{T1}$$

Yields

$$V_{1x}=\sqrt{2gJcp(T_{T1}-T_{1x})}=\sqrt{K(T_{T1}-T_{1x})}$$

and by the geometry of the velocities the axial

and circumferential velocity components are determined

$$V_{1xc}=V_{1x}\sin\alpha_{1x}=V_{2nc}$$
$$V_{1xc}=V_{1x}\cos\alpha_{1x}$$
$$V_{2nc}=V_{1xc}+u$$

and

$$V_{2n}=\sqrt{V_{2nc}^2+V_{2na}^2}$$

The entrance angle to the rotor α_{2n} is a goniometric function of V_{2na}, V_{2nc} or V_{2n}

or

$$\sin\alpha_{2n}=\frac{V_{2na}}{V_{2n}}$$

or

$$\cos\alpha_{2n}=\frac{V_{2nc}}{V_{2n}}$$

or

$$\tan\alpha_{2n}=\frac{V_{2na}}{V_{2nc}}$$

For the case where $V_{1xc}=0$, $V_{2na}=V_{1n}$ and

$$\tan\alpha_{2n}=\frac{V_{2na}}{o+u}=\frac{V_{1n}}{u}$$

The total temperature T_{T2} of the rotor which is the same throughout the duct is found by using the equation of conservation of energy

$$\frac{\overline{V_{2n}}^2}{2g}+JcpT_{2n}=\frac{0}{2g}+JcpT_{T2}$$

$$T_{T2}=T_{2n}+\frac{\overline{V_{2n}}^2}{2gJcp}=T_{2n}+\frac{\overline{V_{2n}}^2}{K}$$

the thermodynamic adiabatic relation gives

$$P_{T2}=P_{2n}\left(\frac{T_{T2}}{T_{2n}}\right)^{\frac{k}{k-1}}$$

With $V_{1xc}=V_{3nc}$ and for example T_{T2}, P_{T2} W known, an $A_{2x}=A_0\sin\alpha_{2x}=\sin\alpha_{2x}$ has to be assumed that results in a known

$$V_{2xc}=u-V_{3nc}=u-V_{1xc}$$

The procedure, explained above, has to be repeated for different α_{2x} until the computation gives the desired V_{2xc}.

From

$$B_{2x}=\frac{WR\sqrt{T_{T2}}}{\sin\alpha_{2x}\sqrt{K}P_{T2}}$$

P_{T2}/P_{2x} is found

$$P_{2x}=\frac{P_{T2}}{P_{T2}/P_{2x}}=P_{3n}$$

$$T_{2x}=T_{T2}\left(\frac{P_{2x}}{P_{T2}}\right)^{\frac{k-1}{k}}=T_{3n}$$

$$V_{2x}=\sqrt{K(T_{T2}-T_{2x})}$$
$$V_{2xa}=V_{2x}\sin\alpha_{2x}=V_{3xa}$$
$$V_{2xc}=V_{2x}\cos\alpha_{2x}=\text{the desired } V_{2xc}$$
$$V_{3nc}=U-V_{2xc}$$
$$V_{3n}=\sqrt{V_{3nc}^2+V_{3na}^2}$$
$$\tan\alpha_{3n}=\frac{V_{3na}}{V_{3nc}}$$
$$T_{T3}=T_{3n}+\frac{\overline{V_{3n}}^2}{K}$$
$$P_{T3}=P_{3n}\left(\frac{T_{T3}}{T_{3n}}\right)^{\frac{k}{K-1}}$$

For the case where $V_{3uc}=0$, $V_{2zc}=V_{3z}$ and $V_{2uc}=u$. Therefore

$$\tan \alpha_{2z}=\frac{V_{2zc}}{V_{2uc}}=\frac{V_{3z}}{u}$$

From

$$B_{3c}=\frac{WR\sqrt{T_{T3}}}{\sin 90°\sqrt{K}P_{T3}}$$

P_{T3}/P_{3z} is found

$$P_{3c}=\frac{P_{T3}}{P_{T3}/P_{3z}}$$

$$T_{3z}=T_{T3}\left(\frac{P_{3z}}{P_{T3}}\right)^{\frac{k-1}{n}}$$

$$V_{3z}=\sqrt{K(T_{T3}-T_{3z})}$$

To show a numerical calculation to get a constant exit pressure throughout the exit of the exit stator, an extreme high u, say 792.1 ft./sec. was assumed and the computation made for the following conditions

$W=24.4$ lbs./sq. ft./sec.

$T_{T1}=540.83°$ R.

$P_{T1}=1.1474$ atms.

$\alpha_{1z}=60°$

$\sin \alpha_{1z}=A_{1z}=.8660$

Thus

$$B=\frac{24.4\times 53.3\times\sqrt{540.83}}{.8660\times\sqrt{12000}\times 1.1474\times 2116}$$

$$=.13132$$

From the B-table, for a value of B=.13132

$$\frac{P_{T1}}{P_{1z}}=1.0693$$

$$P_{1z}=\frac{1.1474}{1.0693}=1.0731$$

To find T_{1z} we have

$$T_{1z}=T_{T1}\left(\frac{P_{1z}}{P_{T1}}\right)^{m}, \text{ where } m=\frac{K}{K-1}=286$$

$$=530.56° \text{ R}$$

To find V_{1z} we have

$$V_{1z}=\sqrt{K(T_{T1}-T_{1z})}$$

$$=\sqrt{12000\times 10.27}$$

$$=351.1 \text{ ft./sec.}$$

The axial velocity component

$$V_{1zz}=V_{1z}\times\sin \alpha_{1z}$$

$$=351.1\times.866$$

$$=304.0 \text{ ft./sec.}$$

The circumferential velocity component,

$$V_{1zc}=V_{1z}\times\cos \alpha_{1z}$$

$$=351.1\times.5000$$

$$=175.5 \text{ ft./sec.}$$

For the rotor section for which

$$P_{2n}=P_{1z}=1.0731$$

$$T_{2n}=T_{1z}=530.56$$

$$V_{2nz}=V_{1zz}=304.0$$

then

$$V_{2nc}=V_{1zc}+u$$

$$=175.5+792.1=967.6 \text{ ft./sec.}$$

$$\tan \alpha_{2n}=\frac{V_{2nz}}{V_{2nc}}=\frac{304.0}{967.6}=.3143$$

$$\alpha_{2n}=17° 26' 30''$$

$$A_{2n}=\sin \alpha_{2n}=.29974$$

$$\cos \alpha_{2n}=.95401$$

$$V_{2n}=\frac{V_{2nz}}{\cos \alpha_{2n}}=\frac{967.63}{.95401}=1014.3 \text{ ft./sec.}$$

$$T_{T2}=T_{2n}+\frac{V_{2n}^2}{K}$$

$$=530.56+85.73=616.29° \text{ R.}$$

$$P_{T2}=P_{2n}\left(\frac{T_{T2}}{T_{2n}}\right)^{3.5}$$

$$=1.0731\left(\frac{616.29}{530.56}\right)^{3.5}=1.8124 \text{ atm.}$$

For the simple condition

$$V_{1zc}=V_{3zc}$$

for example, then

$$V_{3zc}=175.53$$

Accordingly

$$V_{2zc}=u-V_{3zc}$$

$$=792.1-175.53=616.57 \text{ ft./sec.}$$

With this value of V_{2zc}, we try various values of α_{2z} so that V_{2zc} shall equal 616.57.

For example, with

$$\alpha_{2z}=21° 35' \qquad V_{2zc}=615.3$$

This value is sufficiently close to the desired value. The computations required for $\alpha_{2z}=21° 35'$ are carried through below in detail.

$$A_{2z}=\sin 21° 35'=0.36786$$

$$\cos 21° 35'=0.9299$$

$$B=\frac{wR\sqrt{T_{T2}}}{\sqrt{K}A_{2z}P_{T2}}=.20891$$

From B-table,

$$\frac{P_{T2}}{P_{2z}}=1.2378$$

$$P_{2z}=\frac{1.8124}{1.2378}$$

$$=1.4643 \text{ atm.}$$

$$\left(\frac{T_{T2}}{T_{2z}}\right)=\left(\frac{P_{T2}}{P_{2z}}\right)^{m}=(1.2378)^{m}$$

$$=1.0678$$

$$T_{2z}=\frac{616.29}{1.0678}=579.81° \text{ R.}$$

$$T_{T2}-T_{2z}=36.48$$

$$V_{2z}=\sqrt{K(T_{T2}-T_{2z})}$$

$$=\sqrt{12000\times 36.48}=661.60 \text{ ft./sec.}$$

$$V_{2zc}=V_{2z}\times\cos \alpha_{2z}$$

$$=651.60\times.9299=615.3 \text{ ft./sec.}$$

With the value of $\alpha_{2z}=21° 35'$, then

$$V_{2zz}=V_{2z}\times\sin \alpha_{2z}$$

$$=661.6\times.3679=243.39 \text{ ft./sec.}$$

$$\tan \alpha_{3n}=\frac{V_{2zz}}{V_{2nc}}$$

$$=\frac{243.39}{175.53}=1.3865$$

$$\alpha_{3n}=54° 12'$$

$$A_{3n}=\sin \alpha_{3n}=.8111$$

$$V_{3n}=\frac{V_{2zz}}{\sin \alpha_{3n}}=\frac{243.39}{.8111}=300.1 \text{ ft./sec.}$$

$$T_{T2}=T_{2s}+\frac{\overline{V_{2N}}^2}{K}$$

$$=579.81+\frac{\overline{300.1}^2}{12000}$$

$$=579.81+7.50=587.31° \text{ R.}$$

$$P_{T2}=P_{2s}\left(\frac{T_{T2}}{T_{2s}}\right)^{3.5}=1.4643\left(\frac{587.31}{579.81}\right)^{3.5}$$

$$=1.5316 \text{ atms.}$$

To find P_{3x} note that α_{3x} is always 90° and sin $\alpha_{3x}=1=A_{3x}$

$$B=.08878$$

$$\frac{P_{T2}}{P_{3s}}=1.0292 \text{ from B-table}$$

$$P_{3s}=\frac{1.5316}{1.0292}=1.4881$$

$$T_{3s}=T_{T2}\left(\frac{P_{3s}}{P_{T2}}\right)^{m}=582.40° \text{ R.}$$

$$V_{3s}=\sqrt{K(T_{T2}-T_{3s})}$$

$$=\sqrt{12000\times4.9}=242.7 \text{ ft./sec.}$$

These detailed calculations are sufficient to draw all of the points on curves representing $u=792.1$ ft./sec. and $\alpha_{1x}=60°$ and pressure $P_{3x}=1.4881$.

Similar calculations can be made for α_{1x} from 90° to 40° and all values tabulated for $u=792.1$ ft./sec.

The next step is to assume a different u, say $u=600$ ft./sec. The problem is: What values of α_{1x} etc., will produce with this value of u the same values of P_{3x}, T_{3x}, V_{3x} as $\alpha_{1x}=60°$ produced for $u=792.1$ ft./sec.? The condition $V_{1xc}=V_{3nc}$ must still hold.

In order to do this, we use the following two relations which are generally true when

$$V_{1xc}=V_{3nc}$$

(A) $$T_{T2}=\frac{u^2}{K}+\frac{T_{T1}+T_{T2}}{2}$$

and

(B) $$V_{1xc}=\frac{(T_{T2}-T_{T1})K}{4u}=V_{3nc}$$

Using (B), with $T_{T2}=587.3$, the value for $u=792.1$, and $T_{T1}=540.83$ the entrance value, we have

$$T_{T2}+T_{T1}=1128.14$$

$$T_{T2}-T_{T1}=46.48$$

Then

$$V_{1xc}=\frac{46.48}{4\times600}\times12000=232.4 \text{ ft./sec.}$$

$$=V_{3nc}=V_{3xc}-u$$

so that

$$V_{3xc}=u-V_{1xc}=600-232.4=367.6 \text{ ft./sec.}$$

Using (A),

$$T_{T2}=\frac{1128.14}{2}+\frac{\overline{600}^2}{K}$$

$$=564.07+30=594.07° \text{ R.}$$

Then

$$P_{T2}=P_{T2}\left(\frac{T_{T2}}{T_{T2}}\right)^{3.5}=1.5316\left(\frac{594.07}{587.31}\right)^{3.5}$$

$$=1.5936 \text{ atm.}$$

All of these values correspond to a value for $\alpha_{1x}=60°$ for $u=792$ ft./sec.

We now try $\alpha_{1x}=52° 51'$.

$$A_{1x}=\sin 52° 51'=.79705$$
$$\cos 52° 51'=.60390$$

Knowing the values of T_{T1} and P_{T1}, which are the same from root to tip

$$B=.14268$$

and from B-table

$$\frac{P_{T1}}{P_{1s}}=1.0842$$

$$P_{1s}=\frac{P_{T1}}{1.0842}=\frac{1.1474}{1.0842}=1.0583 \text{ atm.}$$

Then

$$T_{1s}=T_{T1}\left(\frac{P_{1s}}{P_{T1}}\right)^{m}=528.46° \text{ R.}$$

$$V_{1s}=\sqrt{K(T_{T1}-T_{1s}}=385.3 \text{ ft./sec.}$$

$$V_{1xc}=V_{1x}\times\cos\alpha_{1x}$$
$$=385.3\times.60390$$
$$=232.4 \text{ ft./sec.}$$

which checks with the required value 232.4 for V_{1xc}.

$$V_{1xc}=V_{1x}\times\sin\alpha_{1x}$$
$$=385.3\times.79705=307.1 \text{ ft./sec.}$$

$$\tan\alpha_{1n}=\frac{V_{1xc}}{V_{1xc}+u}=\frac{307.1}{832.4}=.3689$$

$$\alpha_{1n}=20° 15'$$

$$V_{1n}=\frac{V_{1xc}+u}{\cos\alpha_{1n}}=\frac{832.4}{.9389}=887.2 \text{ ft./sec.}$$

$$T_{T2}=T_{1x}+\frac{\overline{V_{1n}}^2}{K}=528.46+\frac{\overline{887.2}^2}{12000}$$

$$=528.46+65.61$$

$$=594.07° \text{ R.}$$

This checks with the required value 594.07 for T_{T2} from Equation A.

We now have to find the right α_{3n} from which all the $2x$ values will develop. Try $\alpha_{3n}=46° 30'$.

$$A_{3n}=\sin\alpha_{3n}=.72537$$
$$\cos\alpha_{3n}=.68835$$

Knowing that T_{T2} must be 587.31 and P_{T2} must be 1.5315 in order to get the same P_{3x} as before

$$B=.12239$$

From the B table

$$\frac{P_{T2}}{P_{1s}}=\frac{P_{T2}}{P_{1n}}=1.0590$$

Hence,

$$P_{2s}=P_{2n}=\frac{1.5315}{1.0590}=1.4462 \text{ atm.}$$

$$T_{2s}=T_{2n}=T_{T2}\left(\frac{P_{2s}}{P_{T2}}\right)^{m}=577.76° \text{ R.}$$

$$T_{T2}-T_{2s}=594.07-577.76=16.31$$

$$V_{2s}=\sqrt{K(T_{T2}-T_{2s})}$$

$$=\sqrt{12000\times16.31}=442.1 \text{ ft./sec.}$$

$$T_{T2}-T_{2n}=587.31-577.76=9.55$$

$$V_{2n}=\sqrt{12000\times9.55}=338.5 \text{ ft./sec.}$$

$$V_{2nc}=V_{2n}\times\cos\alpha_{2n}$$
$$=338.5\times.6884=233.0 \text{ ft./sec.}$$

This value is sufficiently close to the desired value, 232.4 ft./sec.

$$V_{3nc}=V_{3xc}=V_{3n}\times\sin\alpha_{3n}$$
$$=338.5\times.7254=245.6 \text{ ft./sec.}$$

We also have

$$V_{2x}\times\sin\alpha_{2x}=V_{2x}=V_{3nc}$$

so that

$$\sin\alpha_{2x}=\frac{V_{2xc}}{V_{2x}}=\frac{245.6}{442.1}=.5554$$

and

$$\alpha_{2x}=33^\circ\ 44'\ 30''$$

We should find that $T_{T3}=587.3$ if calculations are correct.

$$T_{T3}=T_{3n}+\frac{\overline{V_{3n}}^2}{K}=T_{2x}+\frac{\overline{V_{3n}}^2}{K}$$
$$=577.76+9.55=587.31^\circ \text{ R.}$$

The above calculations illustrate our method of determining the entrance and exit angles for the rotor and stator blades to insure constant pressure P_{3x} at all radially spaced points in the exit stator where the air flows axially and to insure a circumferential velocity component at the exit of the inlet stator, V_{1xc}, that is equal to the circumferential velocity component V_{3nc} at the entrance of the exit stator. Thus, these calculations show that at a radius at which $u=792.1$ ft./sec.

$V_{1xc}=V_{3nc}=175.5$ ft./sec.
$P_{3x}=1.4881$
$\alpha_{1x}=60^\circ$
$\alpha_{2n}=17^\circ\ 26'\ 30''$
$\alpha_{2x}=21^\circ\ 35'$
$\alpha_{3n}=54^\circ\ 12'$
$\alpha_{3x}=90^\circ$

At a radius at which $u=600$ ft./sec. (which can be considered the tip, Fig. 3)

$V_{1xc}=V_{3nc}=232.4$ ft./sec.
$P_{3x}=1.4881$
$\alpha_{1x}=52^\circ\ 51'$
$\alpha_{2n}=20^\circ\ 15'$
$\alpha_{2x}=33^\circ\ 44'\ 30''$
$\alpha_{3n}=46^\circ\ 30'$
$\alpha_{3x}=90^\circ$

and at a radius at which $u=386.3$ ft./sec. (which can be considered the root, Fig. 4, having a diameter of 34.77'' as assumed above)

$V_{1xc}=V_{3nc}-361.$ ft./sec.
$P_{3x}=1.4881$
$\alpha_{1x}=41^\circ\ 21'$
$\alpha_{2n}=22^\circ\ 59'\ 15''$
$\alpha_{2x}=84^\circ\ 16'\ 30''$
$\alpha_{3n}=35^\circ\ 0'$
$\alpha_{3x}=90^\circ$

The same method as explained for $u=600$ was used, that is, to find values of α_{1x} etc. that produce with this value of $u=386.3$ the same values of P_{3x}, T_{3x}, V_{3x}, as $\alpha_{1x}=52^\circ\ 51'$ produced for $u=600$ or $\alpha_{1x}=60^\circ$ produced for $u=792.1$, while the condition

$$V_{1xc}=V_{3nc}$$

must still hold in this example.

An examination of Fig. 3 and Fig. 4 will show the values listed above for a tip where the radius and R. P. M. are such that $u=600$ ft./sec. and a root where the R. P. M. is the same but the radius is such that $u=386.3$ ft./sec. These values apply to the case where the condition that $V_{1xc}=V_{3nc}$ has been imposed, i. e., the circumferential velocity at the exit of the entrance stator equals the circumferential velocity at the entrance to the exit stator.

In Figs. 1 and 2, the entrance stator blades are shown at 1 and 1', the rotor blades at 2 and 2' and the exit stator blades at 3 and 3' as explained above, and the dotted line in each figure represents the center line or axis of the duct formed by the stator and rotor blades. In Figs. 3 and 4, the center line or axis of the duct formed by the blades is likewise illustrated by a dotted line. In each figure of the drawings, vertical dot and dash lines represent the entrance stator exit 1_x, the rotor entrance and exit 2_n and 2_x, respectively, and the exit stator entrance 3_n. In Figs. 3 and 4, vertical lines also indicate the entrance stator entrance 1_n and the exit stator exit 3_x, the latter denoting that portion of the exit stator at which the fluid is directed axially by the exit stator blades. At this point in the exit stator, the angle of the exit stator blades with respect to a plane normal to the axis of the compressor, α_{3x}, is 90°. The blades of the entrance stator and of the exit stator preferably consist of thin sheet metal of uniform thickness and the rotor blades may also be formed of thin sheet metal of uniform thickness throughout, although in some instances, it is appropriate to have the rotor blades of greater thickness near the middle portion of the rotor than at the rotor entrance and exit. This latter construction is desirable where the solidity of the rotor blades is such that the rotor ducts would not constitute diffusers extending from the rotor inlet to the rotor outlet unless the blade thickness is increased near the central plane of the rotor. Figs. 1 and 2 illustrate a construction in which the stator blades have the same solidity as the rotor blades but it is to be understood that this condition is not essential, for in some instances it is desirable to have one or more of the stator elements comprising blades having a solidity different from that of the rotor blades. In general it may be stated that a single stage or multistage compressor embodying our invention comprises entrance and exit stator elements and a rotor element interposed between these stator elements, the stator and rotor blades all forming fluid ducts extending from the root to the tip portions of the blades and in a generally axial direction through the stator and rotor blades, the entrance stator blades being shaped to provide axial flow of the fluid into the entrance stator and the exit stator blades being shaped to direct the fluid axially. Also, according to our invention, the entrance stator has an exit angle α_{1x} having a value that is a function of the total temperature at the entrance of the entrance stator, T_{T1}, the temperature of the fluid as it leaves the entrance stator, T_{1x}, and the circumferential component of the velocity of the fluid as it leaves the entrance stator, V_{1xc}, as set forth in Equation F, and the rotor has an exit angle α_{2x} having a value that is a function of the total temperature in the rotor, T_{T2}, the temperature at the rotor exit, T_{2x}, and the circumferential component of the fluid velocity at the rotor exit V_{2xc} as set forth in Equation H, with the rotor and exit stator having entrance angles (α_{2n}, α_{3n})) having values providing streamlined flow of the fluid into the rotor and from the rotor into the exit stator, with

$$\frac{V_{1xc}}{V_{1nc}}$$

constant and V_{1xc} determined by Equations A and C'.

For making a typical computation of the several blade angles it may be assumed that

$$\frac{V_{1xc}}{V_{2nc}}$$

is chosen to give the type of compressor desired, i. e., the desired degree of reaction. Thus, if

$$\frac{V_{1xc}}{V_{2nc}}=.2$$

the degree of the reaction at the center of the rotor blade, i. e. for a u of 500 feet per second, where u at the tip is 600 feet per second and u at the root is 386.3 feet per second, is approximately 60%. With the entering conditions as indicated in Figs. 3 and 4 of the drawings we have, for $W=24.4$:

$$T_{T1}=540.83 \qquad P_{T1}=1.1474$$
$$T_{T3}=587.31 \qquad P_{T3}=1.5315$$

For the values at the tip portions of the blades we have from Equation C':

$$V_{1xc}=77.47$$
$$V_{2nc}=600+77.47=677.47$$
$$V_{2xc}=\frac{77.47}{.2}=387.33$$
$$V_{2xc}=600-387.33=212.63$$

and from Equation A:

$$T_{T2}=578.58$$

and then

$$P_{T2}=P_{T1}\left(\frac{T_{T2}}{T_{T1}}\right)^{3.5}=1.4529 \text{ (for air)}$$

Similarly for the root values we have from Equation C':

$$V_{1xc}=120.32$$
$$V_{2nc}=120.32+386.3=506.62$$
$$V_{2xc}=\frac{120.32}{.2}=601.60$$
$$V_{2xc}=3863.-601.60=-215.3$$

and from Equation A

$$T_{T2}=561.02$$

and then

$$P_{T2}=P_{T1}\left(\frac{T_{T2}}{T_{T1}}\right)^{3.5}=1.3043$$

Substituting these values in the equation for T_{1x} we find for the tip

$$\left(\frac{T_{1x}}{T_{T1}}\right)^6-.99908\left(\frac{T_{1x}}{T_{T1}}\right)^5+.01293=0$$

the solution of which is

$$\frac{T_{1x}}{T_{T1}}=.98520$$

and for the root

$$\left(\frac{T_{1x}}{T_{T1}}\right)^6-.99777\left(\frac{T_{1x}}{T_{T1}}\right)^5+.01293=0$$

the solution of which is

$$\frac{T_{1x}}{T_{T1}}=.98377$$

These values for

$$\frac{T_{1x}}{T_{T1}}$$

when substituted in the Equation F (for α_{1x}) give for the tip

$$\alpha_{1x}=\tan^{-1}3.8741$$
$$=75.53°$$

and for the root

$$\alpha_{1x}=\tan^{-1}2.5052$$
$$=68.24°$$

Similarly when these values are substituted in the equation for T_{2x} we have for the tip

$$\left(\frac{T_{2x}}{T_{T2}}\right)^6-.99349\left(\frac{T_{2x}}{T_{T2}}\right)^5+.00863=0$$

the solution of which is

$$\frac{T_{2x}}{T_{T2}}=.98414$$

and for the root

$$\left(\frac{T_{2x}}{T_{T2}}\right)^6-.99311\left(\frac{T_{2x}}{T_{T2}}\right)^5+.01038=0$$

the solution of which is

$$\frac{T_{2x}}{T_{T2}}=.98172$$

These values of

$$\frac{T_{2x}}{T_{T2}}$$

when substituted in the Equation H (for α_{2x}) give for the tip

$$\alpha_{2x}=\tan^{-1}1.1982$$
$$=50.15°$$

and for the root

$$\alpha_{2x}=\tan^{-1}1.2865$$
$$=52.14°$$

(pointing toward the axis because V_{2xc} is negative, -215.3).

For the condition of streamlined flow of the fluid into the rotor and from the rotor into the exit stator we have the condition that

$$\tan\alpha_{2n}=(\tan\alpha_{1x})\frac{V_{1xc}}{u+V_{1xc}} \qquad (P)$$

which gives tan α_{2n} at the tip

$$\tan\alpha_{2n}=.44301$$
$$\alpha_{2n}=23.89°$$

and for the root

$$\tan\alpha_{2n}=.59498$$
$$\alpha_{2n}=30.75°$$

In like manner the values of α_{3n} may be determined from Equation Q with the following results:

At the tip $\alpha_{3n}=33.34°$
At the root $\alpha_{3n}=24.72°$

Corresponding values of the blade entrance and exit angles can be found in the manner set forth above for any other value of u along the blade.

In the typical computation set forth above the angles α_{1x} and α_{2x} for the root and tip were computed by the direct Equations F and H and α_{2n} and α_{3n} deduced for the root and tip by using the Equations P and Q which define the condition for streamlined flow of the fluid into the rotor and into the exit stator. It will be understood, however, that the angles α_{2n} and α_{3n} may be determined by the direct Formulas K and M

respectively. Thus, using Equations M and N, the value for

$$\frac{T_{2x}}{T_{T2}}$$

has for its solution at the tip

$$\frac{T_{2x}}{T_{T2}}=.96950$$

and

$$\alpha_{3n}=\tan^{-1}.65789$$

or

$$\alpha_{3n}=33.34°$$

at the tip, i. e., the same angle determined from Equations H and Q.

At the root

$$\frac{T_{2x}}{T_{T2}}$$

has for its solution

$$\frac{T_{2x}}{T_{T2}}=.93780$$

giving

$$\alpha_{3n}=\tan^{-1}.45956$$

or

$$\alpha_{3n}=24.68°$$

at the root i. e., the same angle determined above, by using Equations H and Q.

Similarly the angles α_{1x} and α_{3n} may be determined by the direct formulas F and M and then the angles α_{2n} and α_{2x} may be deduced by using the Equations P and Q.

As explained above, our invention makes it possible to provide an axial flow compressor in which the pressure of the fluid as it flows axially in the exit stator has substantially the same value from the root to the tip portion of the exit stator.

Furthermore the fluid velocity does not have any radial component at any point in the compressor. The entrance stator blades and the rotor blades are shaped so as to provide progressively decreasing rotor entrance velocity from root to tip to compensate as completely as may be desired for the increase in circumferential rotor speed between the root and the tip of the rotor blades, whereby the root portions of the rotor blades, travelling at relatively low circumferential speed, may impart to fluid entering the rotor at higher absolute velocity than at the tip, the same amount of energy, the exit portions of the rotor blades being correspondingly shaped to insure this result at the rotor exit. Thus the rotor and stator blades are of such shape that the rotor and stator ducts accommodate the vortex free flow of the fluid through the compressor. A characteristic of our compressor is that the velocity of the fluid entering the entrance stator and as it is directed axially in the exit stator is truly axial in direction, (i. e., without any circumferential velocity component and without any radial velocity component), the magnitude of the velocity of the fluid entering the entrance stator being constant from root to tip and likewise the magnitude of the fluid velocity as it is directed axially in the exit stator is constant from root to tip. At the rotor entrance and exit, however, the fluid has a velocity varying progressively from root to tip and which may be regarded as the resultant of circumferential and axial velocity components, at least one of these components varying progressively from root to tip.

It will be understood by those skilled in the art that the equations set forth above and appearing in the appended claims are based on the assumption of an ideal compressible fluid and that in the design of any specific compressor embodying our invention the usual slight modification of the theoretically correct blade shapes may be made to compensate as completely as may be desired for the fact that the compressible fluid does not have the precise physical properties of the assumed ideal fluid.

It is to be understood that our invention is not limited to the particular illustrative embodiments illustrated in the accompanying drawings and described in detail above, but includes such modifications thereof as fall within the scope of the appended claims.

We claim:

1. An axial flow fluid compressor comprising entrance stator blades, rotor blades, and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow of the fluid into the entrance stator, the exit stator blades being shaped to direct the fluid axially, the entrance stator having an exit angle α_{1x} varying progressively from root to tip and having the value

$$\alpha_{1x}=\tan^{-1}\left\{\frac{KT_{T1}}{\overline{V_{1xc}}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T1}}$$

is the largest real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{1xc}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

the exit stator having an entrance angle α_{3n} varying progressively from root to tip and having the value

$$\alpha_{3n}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{3nc}}^2}\left(1-\frac{T_{2x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{2x}}{T_{T2}}$$

is the largest real positive root of

$$\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{3nc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

with

$$\frac{V_{1xc}}{V_{3nc}}$$

constant from root to tip and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})K}{2u\left(\frac{V_{3nc}}{V_{1xc}}+1\right)}$$

and the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left(\frac{V_{1xc}}{u+V_{1xc}}\tan\alpha_{1x}\right)$$

and an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left(\frac{V_{3nc}}{u-V_{3nc}}\tan\alpha_{3n}\right)$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and as it enters the exit stator and whereby the velocity of the fluid as it leaves the rotor varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the fluid as it is directed axially by the exit stator is constant from root to tip.

2. An axial flow fluid compressor comprising entrance stator blades, rotor blades, and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow of the fluid into the entrance stator, the exit stator blades being shaped to direct the fluid axially, the entrance stator having an exit angle α_{1x} varying progressively from root to tip and having the value

$$\alpha_{1x}=\tan^{-1}\left\{\frac{KT_{T1}}{\overline{V_{1xc}}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T1}}$$

is the largest real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{1xc}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

the exit stator having an entrance angle α_{3n} varying progressively from root to tip and having the value

$$\alpha_{3n}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{3nc}}^2}\left(1-\frac{T_{3x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{3x}}{T_{T2}}$$

is the largest real positive root of

$$\left(\frac{T_{3x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{3x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{3nc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{2nc}}^2}\left(1-\frac{T_{1x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T2}}$$

is the largest real positive root of

$$\left(\frac{T_{1x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{2nc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

and an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{2xc}}^2}\left(1-\frac{T_{2x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{2x}}{T_{T2}}$$

is the largest real positive root of

$$\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{2xc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

with

$$\frac{V_{1xc}}{V_{2nc}}$$

constant from root to tip and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})\,K}{2u\left(\frac{V_{2nc}}{V_{1xc}}+1\right)}$$

and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})\,K}{2u}-\frac{u}{2}$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and as it enters the exit stator and whereby the velocity of the fluid as it leaves the rotor varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the fluid as it is directed axially by the exit stator is constant from root to tip.

3. An axial flow fluid compressor comprising entrance stator blades, rotor blades, and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow of the fluid into the entrance stator, the exit stator blades being shaped to direct the fluid axially, the entrance stator having an exit angle α_{1x} varying progressively from root to tip and having the value

$$\alpha_{1x}=\tan^{-1}\left\{\frac{KT_{T1}}{\overline{V_{1xc}}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T1}}$$

is the largest real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{1xc}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

the rotor having an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{2xc}}^2}\left(1-\frac{T_{2x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{2x}}{T_{T2}}$$

is the largest real positive root of

$$\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{2xc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

with

$$\frac{V_{1xc}}{V_{2nc}}$$

constant from root to tip and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})\,K}{2u\left(\frac{V_{2nc}}{V_{1xc}}+1\right)}$$

and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})\,K}{2u}-\frac{u}{2}$$

with the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left(\frac{V_{1xc}}{u+V_{1xc}}\tan\alpha_{1x}\right)$$

and with the exit stator having an entrance angle α_{3n} varying progressively from root to tip and having the value

$$\alpha_{3n}=\tan^{-1}\left(\frac{u-V_{3xc}}{V_{3xc}}\tan\alpha_{1x}\right)$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and as it enters the exit stator and whereby the velocity of the fluid as it leaves the rotor varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the fluid as it is directed axially by the exit stator is constant from root to tip.

4. An axial flow air compressor comprising entrance stator blades, rotor blades, and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow of the air into the entrance stator, the exit stator blades being shaped to direct the air axially, the entrance stator having an exit angle α_{1x} varying progressively from root to tip and having the value

$$\alpha_{1x}=\tan^{-1}\left\{\frac{KT_{T1}}{V_{1xc}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T1}}$$

is the larger real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^6-\left(\frac{T_{1x}}{T_{T1}}\right)^5\left(1-\frac{V_{1xc}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

the exit stator having an entrance angle α_{3n} varying progressively from root to tip and having the value

$$\alpha_{3n}=\tan^{-1}\left\{\frac{KT_{T3}}{V_{3xc}^2}\left(1-\frac{T_{3x}}{T_{T3}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{3x}}{T_{T3}}$$

is the larger real positive root of

$$\left(\frac{T_{3x}}{T_{T3}}\right)^6-\left(\frac{T_{3x}}{T_{T3}}\right)^5\left(1-\frac{V_{3xc}^2}{KT_{T3}}\right)+\left(\frac{WR\sqrt{T_{T3}}}{\sqrt{K}P_{T3}}\right)^2=0$$

with

$$\frac{V_{1xc}}{V_{3xc}}$$

constant from root to tip

and

$$V_{1xc}=\frac{(T_{T3}-T_{T1})\,K}{2u\left(\frac{V_{3xc}}{V_{1xc}}+1\right)}$$

and the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left(\frac{V_{1xc}}{u+V_{1xc}}\tan\alpha_{1n}\right)$$

and an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left(\frac{V_{3xc}}{u-V_{3xc}}\tan\alpha_{3n}\right)$$

whereby streamlined flow of the air is provided as the air enters the rotor and as it enters the exit stator and whereby the velocity of the air as it leaves the rotor varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the air as it is directed axially by the exit stator is constant from root to tip.

5. An axial flow fluid compressor comprising entrance stator blades, rotor blades, and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow of the fluid into the entrance stator, the exit stator blades being shaped to direct the fluid axially, the entrance stator having an exit angle α_{1x} having the value

$$\alpha_{1x}=\tan^{-1}\left\{\frac{KT_{T1}}{V_{1xc}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T1}}$$

is the largest real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{2}{k-1}}\left(1-\frac{V_{1xc}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

the exit stator having an entrance angle α_{3n} varying progressively from root to tip and having the value

$$\alpha_{3n}=\tan^{-1}\left\{\frac{KT_{T3}}{V_{3xc}^2}\left(1-\frac{T_{3x}}{T_{T3}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{3x}}{T_{T3}}$$

is the largest real positive root of

$$\left(\frac{T_{3x}}{T_{T3}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{3x}}{T_{T3}}\right)^{\frac{2}{k-1}}\left(1-\frac{V_{3xc}^2}{KT_{T3}}\right)+\left(\frac{WR\sqrt{T_{T3}}}{\sqrt{K}P_{T3}}\right)^2=0$$

with

$$\frac{V_{1xc}}{V_{3xc}}$$

constant from root to tip and

$$V_{1xc}=\frac{(T_{T3}-T_{T1})\,K}{2u\left(\frac{V_{3xc}}{V_{1xc}}+1\right)}$$

and the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left(\frac{V_{1xc}}{u+V_{1xc}}\tan\alpha_{1x}\right)$$

and an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left(\frac{V_{3xc}}{u-V_{3xc}}\tan\alpha_{3n}\right)$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and as it enters the exit stator and whereby the velocity of the fluid as it leaves the rotor varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the fluid as it is directed axially by the exit stator is constant from root to tip.

6. An axial flow fluid compressor comprising entrance stator blades, rotor blades, and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow of the fluid into the entrance stator, the exit stator blades being shaped to direct the fluid axially, the entrance stator having an exit angle α_{1x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left\{\frac{KT_{T1}}{\overline{V_{1xc}}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{1x}}{T_{T1}}$$

is the larger real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{1xc}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}PT_1}\right)^2=0$$

the rotor having an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{2xc}}^2}\left(1-\frac{T_{2x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{2x}}{T_{T2}}$$

is the larger real positive root of

$$\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{2x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{2xc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

with

$$\frac{V_{1xc}}{V_{3nc}}$$

constant from root to tip and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})K}{2u\left(\frac{V_{3nc}}{V_{1xc}}+1\right)} \text{ and } V_{1xc}=\frac{(T_{T2}-T_{T1})K}{2u}-\frac{u}{2}$$

with the rotor having an entrance angle α_{2n} varying progresively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left(\frac{V_{1xc}}{u+V_{1xc}}\tan\alpha_{1x}\right)$$

and with the exit stator having an entrance angle α_{3n} having the value

$$\alpha_{3n}=\tan^{-1}\left(\frac{u-V_{3nc}}{V_{3nc}}\tan\alpha_{2x}\right)$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and as it enters the exit stator and whereby the velocity of the fluid as it leaves the entrance stator varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the fluid as it is directed axially by the exit stator is constant from root to tip.

7. An axial flow fluid compressor comprising rotor blades and exit stator blades, all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the rotor and exit stator blades, the blades of the exit stator being shaped to direct the fluid axially, the exit stator

having an entrance angle α_{3n} varying progressively from root to tip and having the value

$$\alpha_{3n}=\tan^{-1}\left\{\frac{KT_{T2}}{\overline{V_{3nc}}^2}\left(1-\frac{T_{3x}}{T_{T2}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\frac{T_{3x}}{T_{T2}}$$

is the largest real positive root of

$$\left(\frac{T_{3x}}{T_{T2}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{3x}}{T_{T2}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{3nc}}^2}{KT_{T2}}\right)+\left(\frac{WR\sqrt{T_{T2}}}{\sqrt{K}P_{T2}}\right)^2=0$$

where

$$V_{3nc}=\frac{(T_{T2}-T_{T1})K}{2u}$$

and the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\frac{V_{1n}}{u}$$

and an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\left(\frac{V_{3nc}}{u-V_{3nc}}\tan\alpha_{3n}\right)$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and as it enters the exit stator and whereby the velocity of the fluid as it leaves the rotor varies progressively from root to tip and consists of circumferential and axial components each varying progressively from root to tip, and the pressure of the fluid as it is directed axially by the exit stator is constant from root to tip.

8. An axial flow fluid compressor comprising entrance stator blades and rotor blades all forming fluid ducts extending from the root to the tip portions of the blades and extending in a generally axial direction through the entrance stator and rotor blades, the blades of the entrance stator being shaped to provide axial flow into the entrance stator, the entrance stator having an exit angle α_{1x} varying progressively from root to tip and having the value

$$\alpha_{1x}=\tan^{-1}\left\{\frac{KT_{T1}}{\overline{V_{1xc}}^2}\left(1-\frac{T_{1x}}{T_{T1}}\right)-1\right\}^{\frac{1}{2}}$$

where

$$\left(\frac{T_{1x}}{T_{T1}}\right)$$

is the largest real positive root of

$$\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{k+1}{k-1}}-\left(\frac{T_{1x}}{T_{T1}}\right)^{\frac{2}{k-1}}\left(1-\frac{\overline{V_{1xc}}^2}{KT_{T1}}\right)+\left(\frac{WR\sqrt{T_{T1}}}{\sqrt{K}P_{T1}}\right)^2=0$$

and

$$V_{1xc}=\frac{(T_{T2}-T_{T1})K}{2a}$$

and the rotor having an entrance angle α_{2n} varying progressively from root to tip and having the value

$$\alpha_{2n}=\tan^{-1}\left(\frac{V_{1xc}}{u+V_{1xc}}\tan\alpha_{1x}\right)$$

and an exit angle α_{2x} varying progressively from root to tip and having the value

$$\alpha_{2x}=\tan^{-1}\frac{V_{3n}}{u}$$

whereby streamlined flow of the fluid is provided as the fluid enters the rotor and whereby the velocity of the fluid as it leaves the entrance stator varies progressively from root to tip and

consists of circumferential and axial components each varying progressively from root to tip and the absolute velocity of the fluid as it leaves the rotor is axial.

9. An axial flow fluid compressor according to claim 1 characterized by the ratio

$$\frac{V_{1ac}}{V_{1ac}}$$

being equal to 1.

10. An axial flow fluid compressor according to claim 4 characterized by the ratio

$$\frac{V_{1ac}}{V_{1ac}}$$

being equal to 1.

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