June 12, 1945. F. WHITTLE 2,378,372

TURBINE AND COMPRESSOR

Original Filed Dec. 15, 1938 3 Sheets-Sheet 1

Fig. 1.

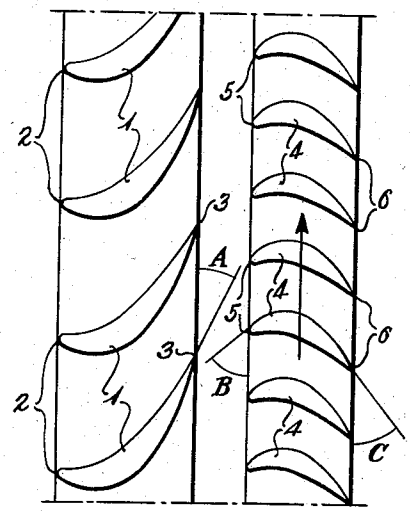


Fig. 2.

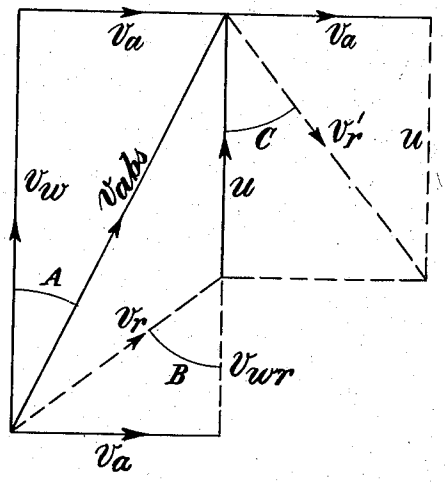
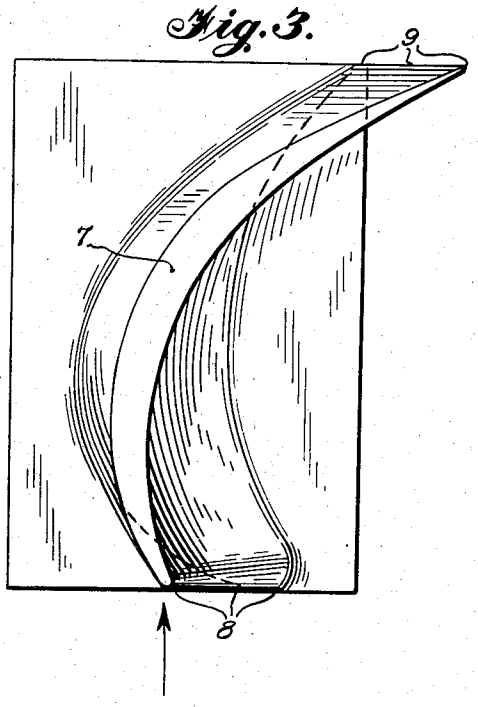


Fig. 3.



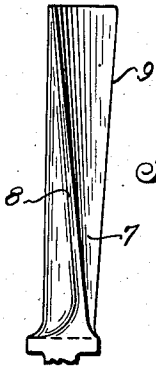


Fig. 4.

Inventor

Frank Whittle

By Stevens and Davis

Attorneys

June 12, 1945. F. WHITTLE 2,378,372

TURBINE AND COMPRESSOR

Original Filed Dec. 15, 1938 3 Sheets-Sheet 2

Fig. 5.

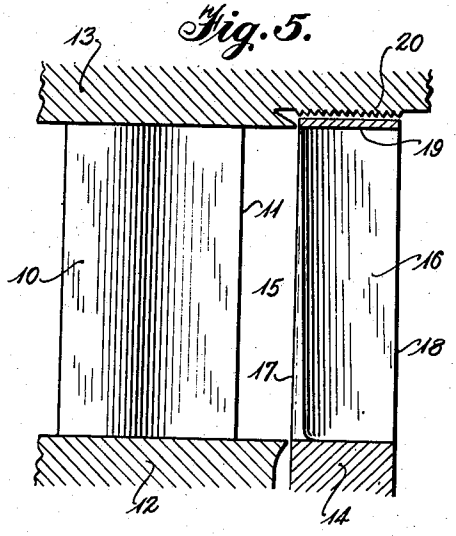
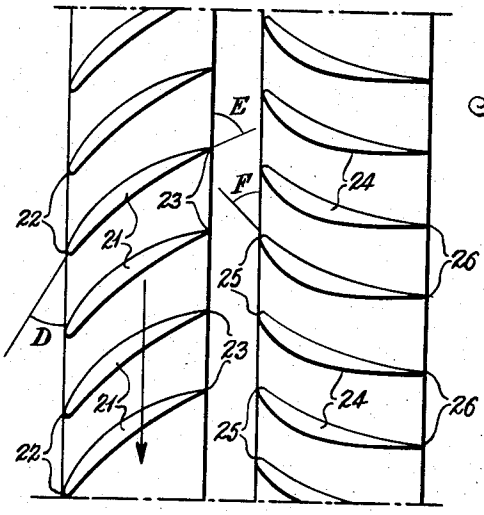


Fig. 6.



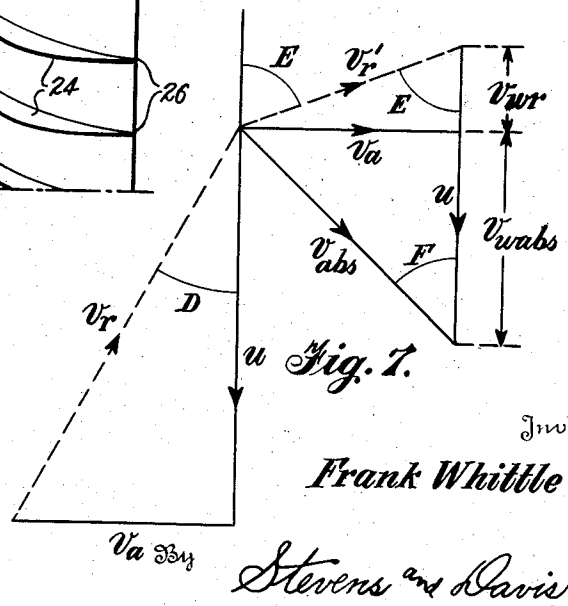


Fig. 7.

Inventor

Frank Whittle

By Stevens and Davis

Attorneys

June 12, 1945. F. WHITTLE 2,378,372

TURBINE AND COMPRESSOR

Original Filed Dec. 15, 1938 3 Sheets-Sheet 3

Fig. 8.

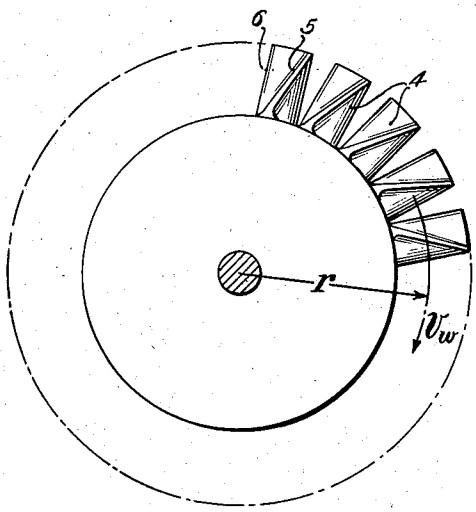


Fig. 9.

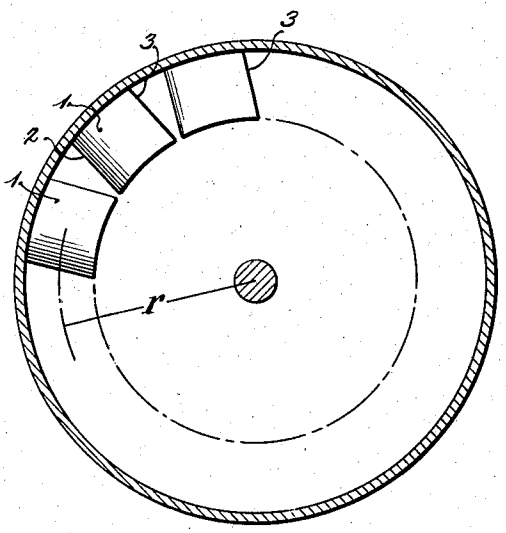
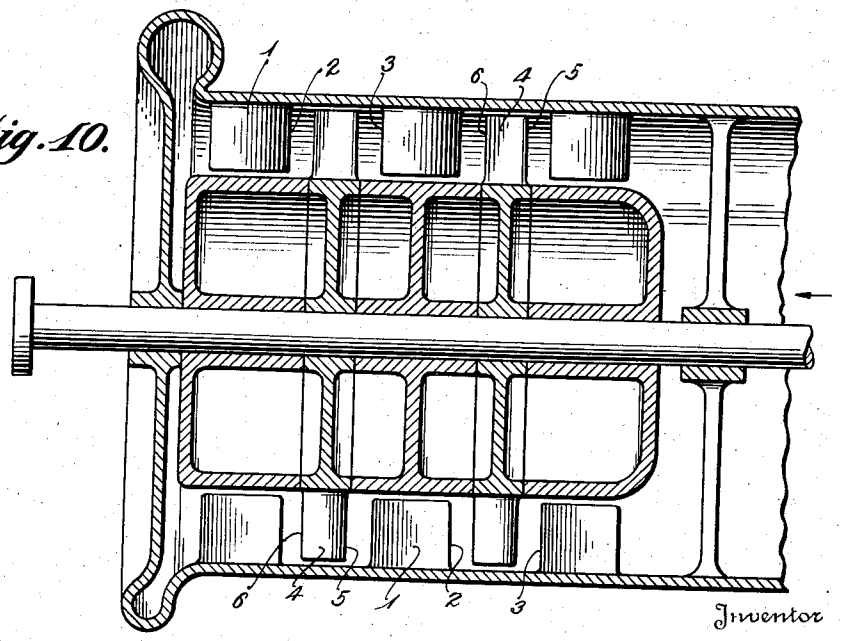


Fig. 10.



Inventor

Frank Whittle

By Stevens and Davis

Attorneys

Patented June 12, 1945 2,378,372

UNITED STATES PATENT OFFICE

2,378,372

TURBINE AND COMPRESSOR

Frank Whittle, Rugby, England

Continuation of application Serial No. 245,980, December 15, 1938. This application September 12, 1942, Serial No. 458,122. In Great Britain December 15, 1937

11 Claims. (Cl. 230—122)

This invention relates to axial flow turbines, compressors, pumps and like rotary power conversion machines, operating with compressible viscous fluids, and is a continuation of my pending application, Serial No. 245,980, filed December 15, 1938, for Turbines and compressors. It is concerned to increase the efficiency obtainable in such machines by the employment therein of novel forms and arrangements of blades.

In determining the form of the blades employed in turbines of the kind above referred to, it has hitherto been assumed that the operative fluid issues from the blades forming the ring of nozzle jets in a series of discrete and separate jets, and that these jets tend to continue to travel in straight lines. The usual text books accept this assumption. As the result of the general acceptance of this assumption blade formations have been adapted to engage or discharge the operative fluid at constant fluid velocity and pressure at all radii. Further, it has been the practice in such turbines to keep the clearance between stator blades and rotor blades very small, with a view to inducing the fluid to pass from one to the other without escaping outside the periphery of the rotor, and for the same reason to make the rotor blades radially longer than their cooperating nozzle blades on the stator.

In the case of impulse turbines the acceptance of the assumption already mentioned has led to the further assumption that the fluid pressure is very little different on the two sides of the rotor wheel and no effort has been made to guard against tip leakage. Also in impulse turbines the assumption of substantially equal pressure on the two sides of the rotor wheel has materially affected the design of thrust bearings, and in many cases has been responsible for the provision of pressure equalizing holes through the wheel. From theoretical considerations I have found that this assumption of straight line flow from the nozzle ring is unjustified, and this has been proved to be the case in fact by experiment. I have also found that even from a nozzle ring with which there is not associated any bounding cylinder to constrain the radial flow of the fluid emerging from the nozzle ring, the flow is in fact substantially rotational and does not show the hitherto assumed tendency to flow tangentially, with the result that the velocity and the pressure of the fluid flowing from the nozzle ring vary in manner complementary each to the other as the radial distance from the axis varies. Thus, even when no special provision in the formation of the blades is made to produce this result, the characteristics of the flow from a stator nozzle ring of a turbine, or the discharge from the rotor of an axial-flow compressor, tend to approximate to the characteristics of a vortex. That is to say, even though the blading is unsuited to such an effect, the tendency exists for the product of the linear rotational velocity of the fluid and the radius to be uniform.

Wherever in this specification a quantity is said to be uniform, it is to be understood thereby that the magnitude of the said quantity is the same at all points along a radius within the path of flow of the operative fluid.

The invention involved in this application is the result of this applicant's discovery of a phenomenon that exists in the flow of viscous compressible fluids through an axial-flow turbine or similar machine and the application of this phenomenon to the design of turbine or compressor blades in a more efficient manner. The new design has actually proven astonishingly more efficient than previous designs, and indications are that at least a 1% greater efficiency will be achieved with turbines of the new design. This is truly a remarkable advance and improvement in this field, since turbine designs are already quite highly developed and a 1% increase in efficiency represents a large economy.

What the present applicant discovered is that steam or other viscous compressible fluid in flowing through a device of the nature of an axial-flow turbine tends strongly to assume a vortex type of flow even though the machine is not designed to permit it to do so. As a result the machine is to a certain extent resisting the normal flow of the fluid instead of removing energy from that normal flow and transforming that removed energy into usable energy.

In previous turbine designs it has always been assumed, or, at least, the designs so indicate, that the fluid passed through the turbine in such a way that the rotational speed thereof was the same at all radii. Thus the turbine blades have been designed to operate upon a fluid, tending to rotate at the same speed at all radii. The present inventor has now discovered that the assumption as to the rotation of the fluid is in error and that the fluid has a strong tendency to rotate more rapidly toward the axis and less rapidly toward the tips of the blades. In fact, the flow tends to approach a vortex condition in which the rotational flow of the fluid is inversely proportional to the radius. As a consequence of this discovery, the present applicant has redesigned the turbine parts, particularly the blades

of the stator and/or rotor, so that the twist and/or entry angles and/or leaving angles vary in such a manner as to compensate for this difference in the rotative speed of the fluid at the various radii.

Some of the consequences of a failure to appreciate the true nature of the flow of a compressible and viscous fluid from the stator nozzle ring in, for example, an impulse turbine may be summarized as follows:

(*a*) It has not been appreciated that on entering the rotor blades the pressure of the fluid is in fact substantially higher near the tips than it is near the roots of the blades, and, as the result of this, the assumed rotational velocity of the fluid at the tips has been too high, and, at the roots, has been too low. The result of this has been that a degree of reaction existed at the tips of the blades for which no provision was made in the blade formation hitherto utilized. Thus, with the blade formations hitherto employed, the fluid has struck the back of the blades near the tips thereof, and, near the roots, has struck the blades with a substantially larger angle of attack than that for which the said blades were designed. At the tips, therefore, of impulse turbine blades there has been a substantial leakage of fluid unless other measures were taken to prevent it.

(*b*) The assumed pressure between the rotor wheel disc and the stator nozzle diaphragm has been higher than was the pressure which in fact existed, and this, in the case of an imperforate wheel disc, has produced a thrust towards the diaphragm, to meet which no provision was made, or, in the case of a perforated disc, has produced a wasteful flow of fluid from the exhaust side to the nozzle side of the wheel.

(*c*) Either the pressure at the blade roots, on the nozzle side, has been lower than that of the exhaust, in which case, there has been a root re-compression for which no provision was made, or, what is more probable, the exhaust pressure has been substantially lower than it might have been for a given output owing to a failure to provide for the necessary re-compression. This is probably the most serious loss, hitherto unrealised, which occurs in impulse turbines.

Similar consequences in the case of other types of turbines of the kind specified will be readily appreciated.

In the case of axial flow compressors considerable importance has in some cases of late been attached to the correct aerodynamic design of rotor blades. Yet the principal condition, namely, that the discharge from the rotor should be as nearly as possible a free circular vortex of uniform axial velocity seems to have been generally disregarded. Some workers have recognized that the ideal condition is one in which the rotor impresses uniform energy upon all the fluid passing therethrough, which leads to the condition that the rotor discharge should be a vortex; but for reasons best known to themselves they appear to have utilized blade formations which cannot satisfy this requirement.

Experiment has shown that in fact the rotational type of flow is extremely stable and tends to occur in the discharge from a stator nozzle ring, or from the rotor of a compressor, even if the formation of the blades is not one specially adapted to produce this kind of flow, but if the maximum of mechanical energy is to be obtained from the expansion of gas through a turbine, then it is theoretically necessary to arrange that the nozzle ring should conform to a flow of the fluid therefrom in a free vortex of uniform axial velocity, and that the rotor should produce a purely axial exhaust flow. In the case of a compressor, the corresponding condition to be achieved is that the discharge from the rotor should be a vortex of uniform axial velocity and the discharge from the stator should be purely axial.

It will be appreciated that the condition specified, namely, that the flow between a rotor and a stator should be either a vortex of uniform axial velocity or a purely axial flow means that the rotational component of velocity, if any, varies inversely as the radius.

The present invention is based fundamentally upon a realization of the fact that in order to increase the efficiency obtainable in machines of the kind referred to by avoiding losses due to undesired changes in the flow of the operative fluid through such machines, it is necessary to adopt formations of the rotor and/or stator blades which correspond and conduce to the type of flow above mentioned.

The invention therefore comprises an axial flow turbine, compressor, pump or like rotary power conversion machine, operating with compressible viscous fluid, wherein are employed blades which are so shaped that their effective entry and/or leaving angle or angles and pitch vary progressively at increasing radial distances along an individual blade in such a manner that the said effective angle or angles and pitch at each said radial distance correspond substantially to the motion of the operative fluid relative to said blade at each said radial distance upon the basis that the velocity of said fluid varies inversely with radial distance.

The invention also comprises an axial flow turbine, compressor, pump or like rotary power conversion machine wherein are employed blades which are so shaped and positioned as to conform to the condition that, when the machine is operating under its normal design conditions, in flowing through the said blades, the operative fluid being compressible and viscous, has its angular momentum about the axis of the machine changed from one value which is substantially uniform before engaging the blades, to another value which is also substantially uniform after leaving the said blades, whilst the axial velocity of said fluid is substantially uniform both before entering and after leaving the said blades, though the magnitude of the axial velocity is not necessarily the same before entering and after leaving the said blades.

Other features will appear from this specification.

Hitherto the accepted distinction between "impulse" turbines and "reaction" turbines has been that in the former for the highest diagram efficiency the ratio of the blade speed to the rotational component of the fluid speed, which has been commonly designated by the expression

$$\frac{u}{V_w}\left(\frac{u}{V_w}\right.$$

in the following description) is of the order of 0.5 whilst in the latter it is of the order of 1.0. In the light of my researches, however, this distinction is less marked, since it may be shown that a so-called "impulse" turbine which upon hitherto accepted assumptions as to fluid flow has a value for

$$\frac{u}{V_w}$$

at mid-blade of 0.5 may have in fact a certain degree of reaction at the blade tips, whilst it is theoretically possible to provide a single-row turbine, or one with a single row per stage, in which the value of

$$\frac{u}{V_w}$$

at mid-blade is much less than 0.5 and still have axial discharge.

It might be thought that it would only be important to allow for the rotational character of the flow in turbines where the blade length is very long radially in proportion to the mean diameter. It has, however, been shown by experiment with gaseous fluid that a substantial gain in efficiency can be obtained by providing in the design for rotational flow in a case where the blade length is approximately only 8% of the mean diameter.

From the foregoing it follows that, for impulse blading, to yield substantial efficiency there has to be recompression at the root. This is difficult to achieve without loss.

Impulse blading possesses the big advantage over reaction blading that for a given wheel speed, a much greater energy conversion per stage can be obtained. It is a corollary of the foregoing that the wheel which would obtain the highest energy conversion per stage whilst avoiding the losses associated with recompression, is one in which

$$\frac{u}{V_w}$$

at the root is 0.5 or slightly greater. This represents a well defined design which lies between the accepted form of impulses and reaction types, i. e.

$$\frac{u}{V_w}$$

at mid-blade is higher than for an impulse blade and lower than for a reaction blade. Moreover, such design is of a wheel on which the pressure difference across the disc is nearly zero. In such a blade the degree of reaction increases toward the tip, from a zero value at the root.

As a practical matter of axial-flow rotor blade design, such blades, whilst having a certain pitch angle, have also been previously given "twist," i. e., a progressive change of effective pitch angle from root to tip, in accordance with the different linear speeds of points on a blade at different radii; as hitherto it was assumed that the fluid velocity was constant, such twist was assumed to result in the blade meeting the gas at the same angle at all radii. According to this invention such blades are formed with substantially greater twist than that which is determined by the different linear velocities of different radial stations of the blade, such greater twist being that which takes account of the radial gradient of fluid velocity (which corresponds to, but is in inverse sense to, the radial pressure gradient).

Further, the last mentioned feature being adopted, a clearance is preferably left between the leaving edges of turbine stator nozzle blades and leading edges of the rotor blades, of at least a quarter of the chord of the rotor blades, and the clear space so formed is walled-in peripherally, preferably by cylindrical extensions of the rim of the nozzle diaphragm. Other novel features are also preferably provided, briefly indicated below.

A preferred construction of an axial flow turbine with blades formed as above, is such that

$$\frac{u}{V_w}$$

at the blade roots is .5 or slightly greater.

A preferred form of velocity-compounded turbine having the aforesaid basic feature, has the blades of successive rows so arranged, that only the last row creates axial discharge, whilst the angular momentum of the fluid in the spaces between rotor and stator blades is uniform in each said space.

In fully applying the invention both rotor and stator blades are formed in accordance with the basic principles hereinbefore enunciated. It is to be understood that there is a possibility that the pitch as between two sections at different radii (i. e., radial stations) of a blade may in effect be varied either by shaping the blade with a constant axial dimension, or by tapering the blade, e. g., by curtailing its leading edge.

In the drawings herewith:

Figure 1 is a formal diagram of turbine nozzle and rotor blades in conventional showing;

Figure 2 is a vector diagram related to Figure 1;

Figure 3 shows a turbine blade in end view from which can be seen a root section (thicker section) and a tip section (thinner section);

Figure 4 is an elevation of the blade of Figure 3, viewed axially;

Figure 5 is a diagrammatic and partial view of a blade arrangement showing stator and rotor blade relationships;

Figures 6 and 7 are diagrams corresponding to Figures 1 and 2, in the case of a compressor;

Figure 8 is a diagrammatic illustration of a turbine rotor showing the radius and direction of velocity;

Figure 9 is a diagrammatic illustration of the stator of a turbine showing the radius of the blades; and

Figure 10 is an illustration of the entire machine as taught by the present invention.

The invention is explained graphically with the aid of the accompanying diagrams, in which the method of arriving at the new blade formation is as follows, in the case of a single-row turbine to yield purely axial exhaust flow.

Figure 1 diagrammatically illustrates turbine nozzle blades 1 with leading edges 2 and leaving edges 3, and rotor blades 4 with leading edges 5 and leaving edges 6. Certain angles show the fluid-flow direction, referred to the plane of rotation; A is the angle of the flow from the nozzle blades at 3, B is the angle at which this flow meets the rotor blades at 5, and C is the angle at which the fluid leaves the edges 6. The rotor blades move according to the arrow in this figure.

Figure 2 is a vector diagram enabling the following formulae to be understood, in which the angles A, B and C, are reproduced. In regard to this:

$u=rw$=linear blade velocity at any given radius r.

v_w=absolute whirl velocity of fluid.

v_a=axial velocity of fluid.

v_{abs}=absolute velocity of fluid.

v_r=velocity relative to blades at which fluid enters blades 4.

v_{wr}=whirl velocity relative to blades at which fluid enters blades 4.

v'_r=velocity relative to blades at which fluid leaves blades 4.

(Though for simplicity v_a is shown the same at entry and exit, this is not necessarily always the case, and in general v_a at discharge will be greater than at entry.)

Then:

$$A=\tan^{-1}\frac{v_a}{v_w}$$

$$B=\tan^{-1}\frac{v_a}{v_{wr}}$$

$$C=\tan^{-1}\frac{v_a}{u}$$

Now,

$v_w r=M$ where M is the angular momentum of a unit mass of fluid about the axis of the machine, and it is a fundamental assumption for this invention that the value of M is constant over any one entry or exit plane.

$$\therefore v_{wr}=v_w-u=\frac{M}{r}-rw$$

where w=angular velocity of blade.

If it is desired to relate these angles with the conditions at any specific radius R, this can be done by the fact that $M=RV_w$ where V_w is the whirl velocity at that radius R.

The above laws apply to any row of a reaction or impulse turbine which is to yield purely axial flow (v_a in the foregoing case). To ascertain the correct twist the correct B and C are found for different values of r.

A numerical example will now be given which will serve to indicate the characteristics of a rotor blade form according to this invention as compared with a conventional form hitherto employed, based on

$$\frac{u}{V_w}=1$$

Data

Wheel mean diameter	inches	30
Blade length	do	2
Mean blade speed	feet per second	400
Mean gas speed (whirl)	do	400
Mean gas speed (axial)	do	80

Conventional blade form hitherto employed (which assumes a constant V_w):

Inner radius of blades 14″.
Outer radius of blades 16″.

Blade speed at tip $\frac{16}{15}\times 400=427$ ft. per second.

Blade speed at root $\frac{14}{15}\times 400=373$ ft. per second.

Relative gas speed at tip (whirl) 427–400=27 ft. per second. (Note: Blade is overtaking gas.)

Angle at tip $=90+\tan^{-1}\frac{27}{80}=108°\ 40'$.

Relative gas speed at root (whirl) =400–373=27 ft. per second.

Angle at root $=90-\tan^{-1}\frac{27}{80}=71°\ 20'$.

∴ Entry angle twisted through 37° 20′.

Blade form according to this invention (data as before):

Gas speed (whirl) is now inversely proportional to radius; axial velocity constant;

$$\text{gas speed (whirl) at tip}=\frac{15\times 400}{16}=375.$$

Blade speed=427 as before.

Relative gas speed at tip (whirl) =427–375=52.

(Note: Blade is overtaking gas.)

Angle at tip $=90+\tan^{-1}\frac{52}{80}=122°$

Gas speed at root (whirl) $=\frac{15\times 400}{14}=429.$

Blade speed at root=373 (as before).
Relative gas speed (whirl) at root=429—373=56.

Angle at root $=90-\tan^{-1}\frac{56}{80}=55°.$

∴ Entry angle twisted through 67°.

Figure 3 illustrates in end view an axial-flow gas turbine rotor blade designed in accordance with the principle of my invention, on the following relationships:

$\frac{u}{V_w}$ at mean section=.545

Axial velocity uniform=48.3% of the whirl velocity at mean section.

Blade length is 17.15% of mean diameter of rotor.

A=22° 10′ root; 29° 15′ at tip.
B=33° 20′ root; 64° 40′ at tip.
C=46° 33′ root; 37° 20′ at tip.

A corresponding blade as hitherto employed would have had the variation on each side of the mean diameter of about half the above values of B.

Figure 4 shows this blade in elevation in the axial direction, on a smaller scale, the blade 7 having leading or inlet edge 8 and leaving or outlet edge 9.

In a turbine with blades shaped as above, there is preferably, according to the invention, a clearance space between the nozzle blades and rotor blades, and a typical construction is indicated by diagram in Figure 5.

In Figure 5 the nozzle blades 10 have trailing or leaving edges 11 and are mounted on the nozzle diaphragm 12 and within the nozzle diaphragm rim 13. The parts 12 and 13 are axially extended towards the plane of the turbine wheel 14, so as to form a cylindrical space at 15. The turbine rotor blades 16 have leading or entering edges at 17 and leaving edges at 18. The blades 16 are encircled by the blade shroud ring 19. An edge 20 is presented by the rim 13, close to the ring 19, a small working clearance being provided; the inner wall of the rim of casing opposite the ring 19, may be corrugated as indicated at 20, in known manner, the arrangement having the object of inhibiting fluid flow between the rotor of the turbine and the axial extension of the diaphragm, or the casing.

The axial dimension of the space 15 is as shown greater than half the chord of the blades 16; it is in any case greater than one quarter. The fluid flow in this space is as nearly as possible that of a free circular vortex, i. e. a concentric flow in which $v_w r$ is constant where v_w=rotational (whirl) velocity and r=radius.

Turning to the construction of a compressor in accordance with the invention, reference is made

to Figure 6 and Figure 7. These correspond generally to Figures 1 and 2. Figure 6 shows the rotor blades 21 with entering edges 22 and leaving edges 23; and stator blades 24 with like edges 25, 26. With reference to the allied vector diagram, Figure 7, the following are indicated:

v_a=axial velocity of fluid.
u=blade speed.
v_{wr}=whirl velocity of fluid relative to blades on discharge from blades 21.
v_r=velocity of fluid relative to blades on entering blades 21.
v'_r=resultant velocity of fluid relative to blades at discharge from blades 21.
v_{abr}=absolute velocity of fluid.
v_{wabs}=absolute whirl velocity on discharge from blades 21.

Then the angles to which the blades are designed, at any given radial station, are shown to be:

$$D=\tan^{-1}\frac{v_a}{u}$$

$$E=\tan^{-1}\frac{v_a}{u}-v_{wabs}$$

$$F=\tan^{-1}\frac{v_a}{v_{wabs}}$$

The predominant condition which is to be applied is that

$v_{wabs}=\frac{M}{r}$ where M=angular momentum of fluid

about the axis of the machine.

The foregoing examples illustrate the manner in which the angles defining blade formation in accordance with this invention may be ascertained with considerable accuracy. It will be appreciated, however, that the actual flow of the fluid may depart angularly from the angles so ascertained by small amounts. The blade angles thus ascertained may therefore be modified in practice by such small amounts to allow for such dixerence as will be understood by those skilled in the art.

I claim:

1. A rotary axial flow turbine-type fluid power conversion machine including a stator turbine element carrying fixed blades and a rotor turbine element carrying fixed blades and operatively positioned with respect thereto, the blades of at least one of said turbine elements having leading and trailing edges, the angles at said trailing edges from the element outward varying in conformity to a flow of fluid through the turbine in normal operation having constant angular momentum and with the said leading edges presenting a substantially zero angle of attack and the axial velocity of the fluid being substantially uniform at all radii.

2. A rotary axial flow turbine-type fluid power conversion machine including a stator turbine element carrying fixed blades and a rotor turbine element carrying fixed blades and operatively positioned with respect thereto, the blades of at least one of said turbine elements having leading and trailing edges, the angles at said trailing edges from the element outward varying in conformity with the flow of fluid through the turbine in normal operation and with the said leading edges presenting a substantially zero angle of attack, in which the angular rotational velocity of the fluid varies inversely with the radius and axial velocity of the fluid is substantially uniform at all radii.

3. A rotary axial flow turbine-type fluid power conversion machine including a stator turbine element carrying fixed blades and a rotor turbine element carrying fixed blades and operatively positioned with respect thereto, the blades of said turbine elements having leading and trailing edges, the angles at said trailing edges from the elements outward varying in conformity with the flow of fluid in normal operation and with the said leading edges presenting a substantially zero angle of attack, in which the angular rotational velocity of the fluid varies inversely with the radius and axial velocity of the fluid is substantially uniform at all radii.

4. A rotary axial flow turbine-type fluid power conversion machine including a stator carrying fixed blades and a rotor disc carrying fixed blades and operatively positioned with respect thereto, the rotor disc blades having leading and trailing edges, the angles at said trailing edges from the disc outward varying in conformity with the flow of fluid through the turbine in normal operation and with the said leading edges presenting a substantially zero angle of attack, in which the angular rotational velocity of the fluid varies inversely with the radius and axial velocity of the fluid is substantially uniform at all radii.

5. A rotary axial flow turbine-type fluid power conversion machine including a stator carrying fixed blades and a rotor carrying fixed blades and operatively positioned with respect thereto, the stator blades having leading and trailing edges, the angles at said trailing edges from the stator outward varying in conformity with the flow of fluid through the turbine in normal operation and with the said leading edges presenting a substantially zero angle of attack, in which the angular rotational velocity of the fluid varies inversely with the radius and axial velocity of the fluid is substantially uniform at all radii.

6. A rotary axial flow turbine-type fluid power conversion machine including a stator turbine element carrying fixed blades and a rotor turbine element carrying fixed blades and operatively positioned with respect thereto, the blades of said turbine elements having leading and trailing edges, the angles at said edges from the elements outward varying in conformity with the flow of fluid in normal operation, in which the angular rotational velocity of the fluid varies inversely with the radius and axial velocity of the fluid is substantially uniform at all radii, in which the ratio of the linear blade velocity at any given radius to the absolute whirl velocity of the fluid passing through the turbine in normal operation at the same radius is 0.5 or slightly greater at the roots of the blades and in which the degree of reaction is substantially zero at the root but increases toward the tip.

7. A rotary axial flow turbine-type fluid power conversion machine including a turbine element carrying blades, and a second turbine element carrying blades, said elements being operatively positioned and rotating with respect to each other, the blades of at least one of said turbine elements having leading and trailing edges, the angles at said trailing edges from the element outward varying in conformity to a flow of fluid through the turbine in normal operation having constant angular momentum and with the said leading edges presenting a substantially zero angle of attack and the axial velocity of the fluid being substantially uniform at all radii.

8. A rotary axial flow turbine-type fluid power conversion machine, including a stator carrying fixed blades and a rotor carrying fixed blades

and operatively positioned with respect thereto, wherein the operative fluid is admitted through the stator blades forming a nozzle ring, these blades being such that, substantially, the fluid is discharged therefrom at angles A which vary radially according to the formula

$$A=\tan^{-1}\frac{rV_a}{M}$$

when r is the radius, V_a the axial velocity and M the angular momentum of the fluid about the axis of the machine, and V_a and M are constant.

9. A rotary axial flow turbine-type fluid power conversion machine, including a stator carrying fixed blades and a rotor disc carrying fixed blades and operatively positioned with respect thereto, wherein the operative fluid is admitted through the stator blades forming a nozzle ring, and where the leading edges of the rotor blades are such that, at all radii r, the value of the

$$\text{entry angle } B=\tan^{-1}\frac{V_a}{\frac{M}{r}-u}$$

where r is the radius, V_a the axial velocity, u the blade speed, and M the angular momentum of the fluid about the axis of the machine, the algebraic sign of u being positive for a turbine and negative for a compressor, and V_a and M are constant.

10. A rotary axial flow turbine-type fluid power conversion machine, including a stator carrying fixed blades and a rotor disc carrying fixed blades and operatively positioned with respect thereto, wherein the operative fluid is admitted through stator blades forming a nozzle ring, and wherein the rotor blades are such that, at all radii r, the maximum value of the

$$\text{leaving angle } C=\tan^{-1}\frac{V_a}{u}$$

when u is the blade speed and when V_a is the axial velocity of the fluid which is constant.

11. A rotary axial flow turbine-type fluid power conversion machine, including a stator carrying fixed blades and a rotor disc carrying fixed blades and operatively positioned with respect thereto, wherein the operative fluid is admitted through stator blades forming a nozzle ring, these blades being such that, substantially, the fluid is discharged therefrom at angles A which vary radially according to the formula

$$A=\tan^{-1}\frac{rV_a}{M}$$

and where the leading edges of rotor blades are such that, at all radii r, the value of the

$$\text{entry angle } B=\tan^{-1}\frac{V_a}{\frac{M}{r}-u}$$

and the maximum value of the

$$\text{leaving angle } C=\tan\frac{V_a}{u}$$

where V_a is the axial velocity of the fluid, u is the blade speed and M is the angular momentum of the fluid about the axis of the machine, the algebraic sign of u being positive for a turbine and negative for a compressor, and V_a and M are constant.

FRANK WHITTLE.