

Jan. 7, 1958

R. P. ATKINSON
CENTRIFUGAL COMPRESSOR

2,819,012

Filed Dec. 22, 1950

2 Sheets-Sheet 1

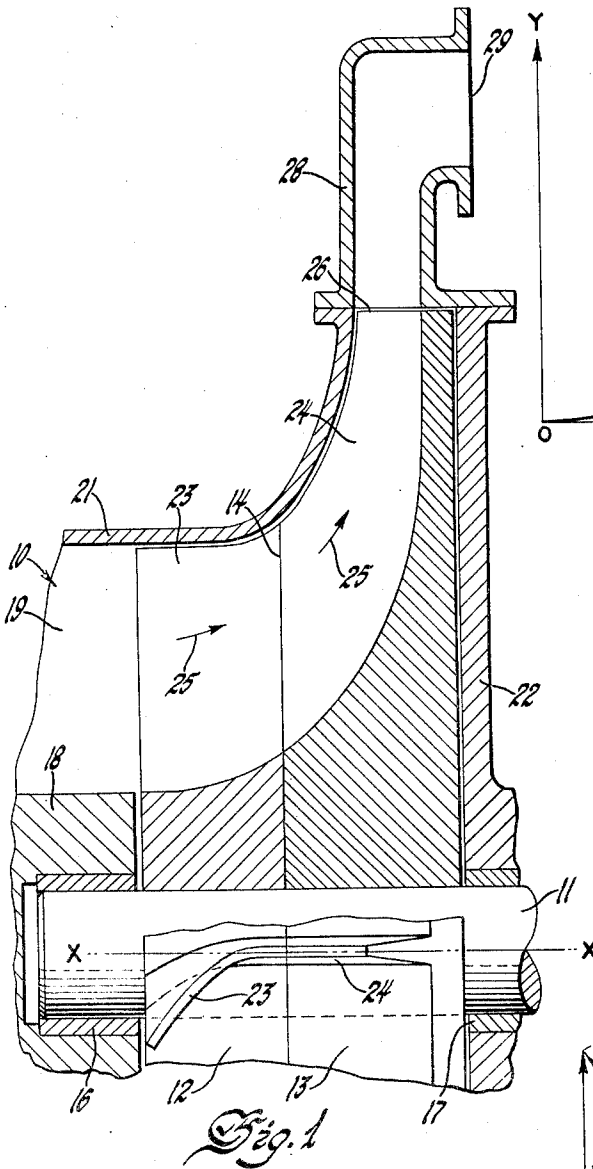


Fig. 1

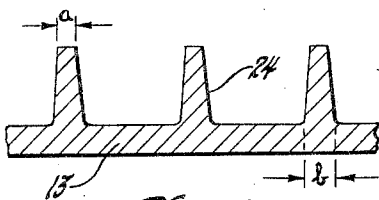


Fig. 2

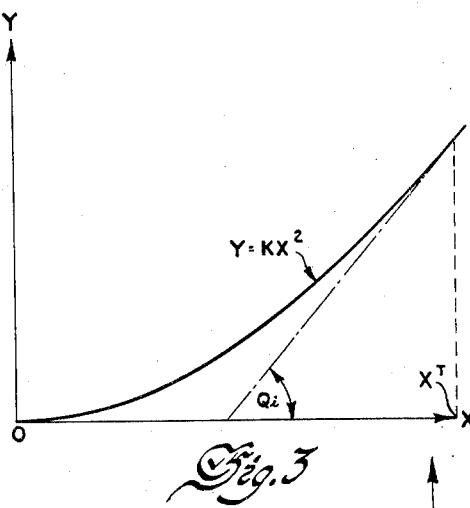


Fig. 3

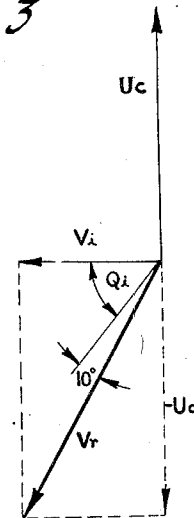


Fig. 4

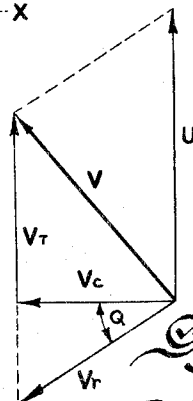


Fig. 5

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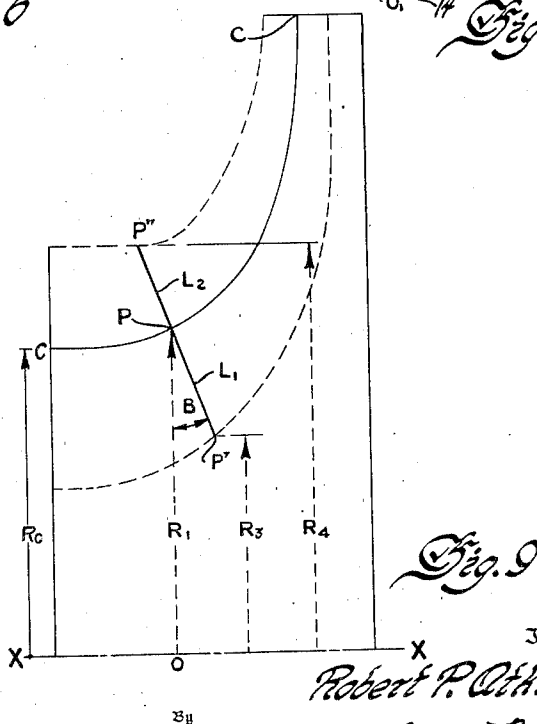
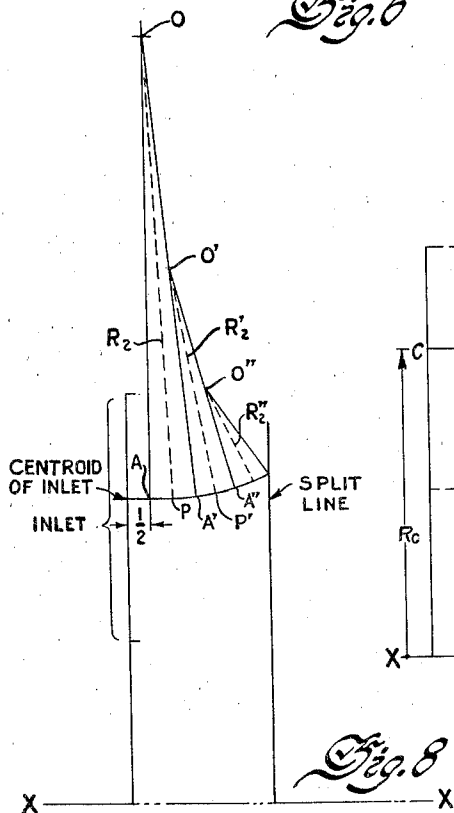
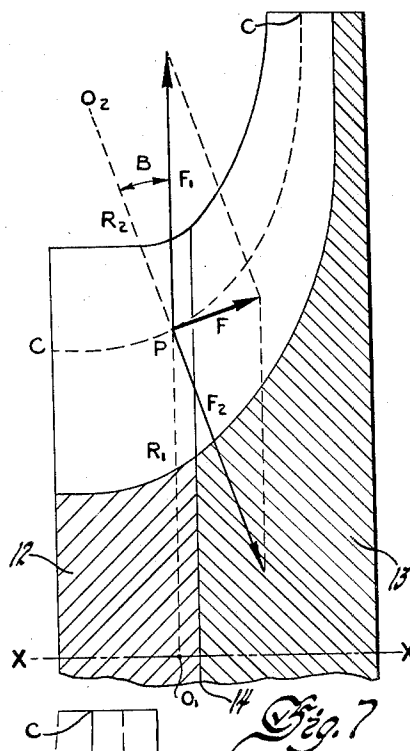
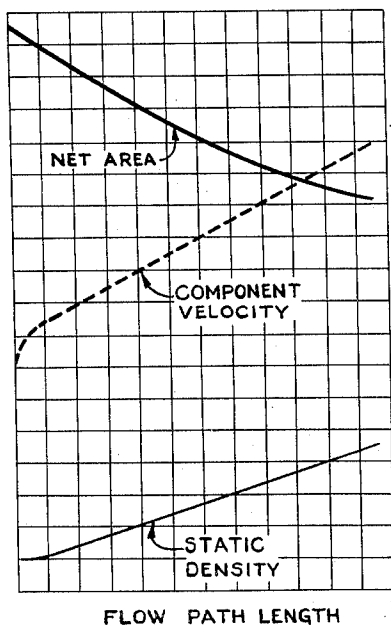
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CENTRIFUGAL COMPRESSOR

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Application December 22, 1950, Serial No. 202,273

11 Claims. (Cl. 230—119)

My invention relates to centrifugal compressors, more particularly to compressors of a known type in which air enters the compressor parallel to its axis of rotation, engages an inducer portion of the compressor rotor provided with generally helical blading in which the air flows in a predominantly axial direction and is accelerated tangentially, and then proceeds through an impeller portion of the rotor formed with substantially radial vanes in which the air is accelerated tangentially and radially, leaving the rotor with a high tangential velocity. In such a compressor, the air discharged from the rotor is received in a diffuser in which the velocity head of the air is largely converted to static head and the air is directed to the outlet or outlets of the compressor. It is to be understood, however, that this invention is concerned not with the diffuser as such but rather with the form of the rotor.

The invention is particularly intended for and highly valuable in high performance centrifugal compressors such as are used in gas turbine engines. It is well known that compressors for this service ordinarily must handle a large volume of air, must operate at a high compression ratio, and must be of high efficiency. The adoption of the principles of the invention in a previously known type of gas turbine engine has resulted in remarkable increases in the power and efficiency of the engine.

The principal objects of the invention are to provide a centrifugal compressor of high capacity, one of high compression ratio, and one of high efficiency. Further objects of the invention are to increase the flexibility and stability of centrifugal compressors and to improve the performance of such compressors and of motive power systems embodying compressors. A further object of the invention is to extend the range of usefulness of centrifugal compressors, especially single stage compressors, and thereby render more advantageous the use of the relatively simple and reliable centrifugal compressor in gas turbine engines, where heretofore the relatively temperamental, expensive, complicated, and delicate axial-flow compressor has been considered to be more suitable.

Expressed in another way, the objects of the invention may be defined as to provide a centrifugal compressor in which the flow path defined by the structure of the compressor is best accommodated to the natural path of flow of the gaseous medium, in which the distribution of pressure at any section of the passages of the compressor is as uniform as practicable, in which velocity distribution is uniform across the passages to a much greater extent than hitherto, and in which the tendency to supersonic flow and choking is suppressed to a greater extent and over a wider range of operating conditions than hitherto considered possible.

The principles of the invention and the manner in which they are applied to obtain the highly important advantages of the invention are difficult to summarize, but will be apparent to those skilled in the art from the subsequent detailed description of the principles of the invention and of a compressor embodying those principles.

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Referring to the drawings: Figure 1 is a diagrammatic sectional view of a single stage single entry radial flow compressor, taken on a plane containing the axis of rotation; Figure 2 is a diagram illustrating the blade thickness; Figure 3 is a diagram illustrating the preferred form of inducer blade; Figure 4 is a vector diagram illustrating conditions at the compressor inlet; Figure 5 is a vector diagram illustrative of flow conditions within the rotor; Figure 6 is a chart illustrating the variation of static density, component velocity, and net area of the path through the rotor; Figure 7 is a diagram expository of the principles of the invention; Figure 8 is a diagram illustrating the construction of the mean flow path or centroid of the rotor; and Figure 9 is a diagram illustrating the construction of the boundaries of the flow path.

Although compressors of the general type to which the invention relates are well known, it is believed advisable to describe briefly the structure of such a compressor in order to eliminate any possible misunderstanding as to terminology in the specification. Referring to Figure 1, the compressor illustrated diagrammatically comprises a rotor 10 mounted on a shaft 11 for rotation therewith. The rotor comprises an inducer portion 12 and an impeller portion 13. While these may be integral, they are ordinarily made separately for manufacturing reasons and are fixed together. The inducer abuts the impeller at a radial plane indicated at 14 and commonly called the "split line." The shaft 11 is mounted in bearings indicated by bushings 16 and 17 in the fixed structure of the compressor. The forward bearing 16 is mounted in a body 18 which defines the inner boundary of the compressor inlet 19. An annular casing 21 defines the outer boundary of the inlet and of the flow path through the rotor 10. The frame member 22 which supports the rear bearing 17 may be a disk closely adjacent the rear face of impeller 13. The inducer 12 is fitted with radial blades 23 which are of curved form to accelerate the air tangentially, as will be described more fully. The impeller 13 is provided with radial blades 24 which ordinarily constitute continuations of the impeller blades 23, a continuous passage 25 for flow of air from the compressor inlet 19 to the compressor outlet 26 being defined between each adjacent set of rotor and impeller blades. For greater clarity, one complete blade 23, 24 is illustrated as viewed at right angles to the plane of the impeller blade 24 in Figure 1. As will be apparent, the body of the rotor constitutes the inner boundary of the air flow path through the rotor. The outer boundary could of course be defined by a shroud fixed to the rotor but may be defined by the fixed casing structure 21 adjacent the tips of the blades, as illustrated. The air discharged from the rotor enters a diffuser 23 which may be of any suitable form, and is illustrated schematically as provided with a plurality of discharge outlets 29.

Although Figure 1 illustrates a single entry compressor, the principle of the invention is equally applicable to double entry and to multi-stage compressors. The form of the inlet may also be varied to suit the particular installation. The impeller blades are ordinarily radial, but may be curved ahead or back.

The nature of the invention may be indicated by stating that it involves providing a flow path in the compressor conforming to the natural flow path of the air or other fluid (referred to as air for conciseness).

The air entering the compressor is considered to have a known velocity parallel to the axis of rotation of the compressor, without radial or circumferential components.

In the inducer portion of the rotor, the air is accelerated circumferentially of the axis, or tangentially, in a uniform manner as practicable, by the inducer vanes. As the air acquires a tangential velocity, centrifugal force

acts upon the air in a radially outward direction. In the impeller, the angular velocity of the air about the axis remains constant, but the centrifugal force increases with the radius. There is thus a continuously increasing centrifugal force on a particle of air as it traverses the rotor from inlet to outlet.

The inertia of the air resists the radial acceleration, and a particle of air thus tends to follow a curved path, relative to the rotor as a frame of reference, under the action of the initial or entrance velocity and the centrifugal force exerted on the particle.

The problem of providing an optimum flow path is complicated by the fact that the air cannot be treated as a free particle under the action of centrifugal and inertia forces, due to the effects of friction, of back pressure in the rotor outlet, and of variations in the density of the air and the area of the flow path.

The rotor of the invention is formed so that the flow path provided for the air conforms to the path of free movement due to the entrance velocity and centrifugal force with allowance for the various modifying factors just described. As a result, the flow is not forced from its natural path by the physical boundaries of the channel defined by the compressor casing (or shroud, if present) and the surface of the rotor.

The mean path is determined for a prescribed relation between component velocity (velocity in the plane containing the axis) and distance along the mean path. The area of the passage is modified along the length of the passage to provide the abovementioned prescribed relation.

As will be apparent, the tangential velocity and the radial acceleration increase as the air moves radially outward through the impeller portion of the rotor, and the final path of the air (apart from the tangential component) approaches a radial direction. The air is discharged with a high absolute velocity, the major component of which is tangential, and the velocity head is largely converted to static pressure in the diffuser, which may be of known type.

The principles of the invention may be most clearly explained by disclosing the procedure involved in designing a compressor according to the invention to meet given design conditions. Obviously, the size and specific form of the compressor will vary with such parameters as diameter, speed, air flow, pressure, and the like, but such variations do not affect the design procedure.

The following basic design conditions must be established or assumed: (1) pressure and temperature of air at the inlet, (2) air flow desired, weight per unit time, (3) rotor speed, revolutions per unit time, (4) assumed efficiency, and (5) assumed pressure ratio (total outlet pressure to total inlet pressure). Of these items, (1), (2), (3), and (5) depend upon the installation for which the compressor is built. For a gas turbine engine, pressure and temperature of the air may be assumed atmospheric conditions. The desired air flow would be determined by the engine air requirements. The rotor speed is usually that of the turbine, and is limited by the necessity to avoid excessive tangential velocities, and thereby by the diameter of the rotor. The value assumed for efficiency is based upon experience, and for the compressor of the invention experience indicates a value of 80% to be suitable for design purposes. The assumed pressure ratio is determined by the rotor tip speed and, of course, must not exceed what can be realized. The values assumed must be compatible, and the selection of design parameters may involve trial and error to some extent.

Certain dimensional values must also be assumed: (1) impeller radius, (2) outer and inner radii of the annular inlet, (3) impeller blade width at the outlet, (4) axial length of the inducer portion, (5) number of blades, and (6) blade thickness, normally expressed in terms of thickness at the tip and at the root of the blade.

The impeller radius is ordinarily limited by the dimen-

sions of the engine. It is related to speed, as excessive centrifugal forces in the rotor must be avoided. Pressure ratio is determined primarily by the speed and diameter of the rotor.

The inlet diameter is based on a compromise between having a large diameter to maintain a reasonably low inlet velocity and a small diameter to reduce the relative Mach number at the outer diameter of the inlet eye.

The outlet width of the impeller is based in the usual manner on the flow conditions at the outlet, that is, upon the area necessary for flow of the assumed weight of air and assumed outlet conditions. It is not critical, and may be varied over a rather wide range to suit installation requirements.

The length of the inducer may be limited by the allowable overall length of the rotor and machine tool requirements. In some installations, especially double-sided compressors and cases in which a compressor is substituted in a previously existing engine, it may not be possible to make the inducer of optimum length. In the inducer, the tangential velocity of the air is increased at roughly constant radius. In the impeller, the tangential velocity is increased as the air flows outward, tangential velocity being proportional to the radius. There is a gradual transition from axial flow at the inducer entrance to nearly radial flow at the impeller outlet. As indicated above, installation conditions may dictate an inducer shorter than optimum, but, in general, an effort should be made to provide an inducer portion of such length that tangential acceleration in the inducer approximately equals that in the impeller.

The number of blades is a matter of informed choice based upon test and experience. With too few blades, there is excessive turbulence; with too many blades, excessive friction. A small number of blades will give rise to greater pressure fluctuations. The use of a prime number of blades may reduce resonance.

The blade thickness is a matter of structural design to obtain sufficient mechanical strength combined with the required vibration characteristics and a structure adapted to manufacturing processes. Figure 2 illustrates blade thickness in the impeller, the thickness at the tip of the blade being indicated by *a* and at the root of the blade by *b*.

As will be apparent to those skilled in the art, the above discussion of design procedure largely relates to background material, as these or equivalent assumptions are necessary for the design of compressors in general of the type to which the invention relates. For this reason, the computations and the underlying theory are not discussed in detail.

From the design conditions and physical dimensions assumed, the area, velocity of flow, static density, total pressure, and total temperature, at the rotor inlet and outlet, may be computed in known manner.

The preferred inducer blading, upon which the subsequent discussion is based, is such that the line of intersection of a blade with any plane perpendicular to the axis is radial, and the intersection of a blade by a cylinder concentric with the axis is a parabolic curve in the cylinder. Otherwise expressed, the blade is of the form, in cylindrical coordinates, *T* is proportional to *X*², where *X* is the distance measured along the axis from the discharge end of the inducer and *T* is the angular displacement of a radial blade element relative to the discharge end of the blade. At any given radius, we may state this as *Y*=*KX*², where *Y* is the length of arc through which the blade is displaced, *K* is a constant to be determined, and *X* is as before. This relation is illustrated in Figure 3.

The first step in the determination of the constant *K* is the determination of the angle of attack of the inducer blade at the centroid of the inlet passage. In this specification, the term "centroid" is defined as the mean path; that is, the radius of the centroid is the radius which divides the flow path into two portions of equal area.

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The radius R_c of the centroid at the inlet may be determined for this purpose from the dimensions of an inducer passage, fixed by the inner and outer radii of the inlet and the number and thickness of the blades, by graphical methods. The tangential velocity U_c of the impeller at the centroid is the product of R_c in feet by the angular velocity of the rotor in radians per second. The air velocity, assumed parallel to the axis, equals the air flow in weight units divided by the product of the air density and the total area of the inlet annulus (the inlet 19) and is represented by V_i .

Given these values, the relative velocity of the air with respect to the rotor, V_r , may be determined as indicated by the vector diagram of Figure 4. Experience has shown that an angle of attack of approximately 10 degrees at the inducer entrance will give peak efficiency, due presumably to the resulting increase in effective inlet area. Too great an angle of attack will result in stalling at the entrance.

The angle Q_1 (Figs. 3 and 4) of the inducer blades at entrance is therefore taken as ten degrees less than the angle of V_r at the centroid to the axis. Since the blade equation is

$$Y = KX^2, \frac{dY}{dX} = 2KX, \text{ or } \tan Q = 2KX_T$$

where X_T is the known length of the inducer. This equation may be solved for K , which determines the form of the inducer blading. It may be noted that, since the blades are radial, the angle Q and the angle of attack vary with the radius.

So much having been done, it is now in order to begin the determination of the mean path through the rotor and from the mean path the inner and outer boundaries of the path of air through the rotor.

The first stage in this operation is to prepare a plot of component velocity against path length, as illustrated in Figure 6. The path length is the length of the projection on a plane containing the rotor axis of the path through the entire rotor of a particle of air entering the compressor at the centroid of the inlet, and must be estimated from the dimensions of the rotor.

The component velocity V_c is velocity in a plane containing the rotor axis. In other words, component velocity is the resultant of the axial and radial components of the absolute velocity of a particle. This velocity at inlet and outlet is likewise computed from the stated design conditions. Component velocity at zero path length is that due to total area of the inlet. Component velocity at one half inch from the inlet is increased by the decrease in path area due to the inducer blades, and may be computed from total flow and net area of the path (static density assumed to equal that at the inlet). From the one half inch point, component velocity is assumed to increase linearly with path length.

Figure 5 illustrates the relationship between U , tangential velocity of the impeller, V_r , velocity of the air relative to the impeller, and V , absolute velocity of the air, which is the resultant of U and V_r . It also illustrates the rectangular components V_c and V_t of V taken respectively in a plane containing the axis of the impeller and normal to this plane. V_c is the component velocity and V_t the tangential velocity of the air.

It will be noted that the design assumption of linear increase in component velocity is based upon the desired result of smooth continuous changes in V_c , a condition believed to promote efficiency of compression. The ultimate form of the rotor will be such as to justify this assumption.

The principle underlying the construction of the optimum centroid of the mean air flow path is illustrated in Figure 7, which shows diagrammatically a single-sided impeller with axis of rotation $X-X$. The centroid of the air path, to be determined, is $C-C$. Assuming an elementary particle of air at a point P on the centroid, this particle will be subject to a centrifugal force F_1 due to

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its rotation about the rotor axis. F_1 for unit mass will equal tangential velocity V_t squared divided by the radius R_1 . The radius is defined by the position of P and equals O_1P . The tangential velocity may be calculated. As indicated in Figure 5, the tangential velocity equals $U - V_c \tan Q$. V is the product of R_1 and the angular velocity of the rotor in radians per second. V_c may be taken from the chart (Fig. 6). Q is determined by the form of the inducer blading and the radial and axial coordinates of point P .

The air flow path obviously curves outwardly, the problem being to define the path of natural flow of the air so as to form the impeller to avoid deflecting the air from this path. Also, the curve $C-C$ will clearly be a continuous curve. It will, therefore, have a center of curvature corresponding to the point P , indicated on the diagram at O_2 . The location of O_2 is fixed by the form of the curve $C-C$ and the location of P . The elementary particle of air at P is rotating about the instantaneous center O_2 at a radius R_2 equal to the radius of curvature of $C-C$ at point P . The velocity of P in this path is V_c , the component velocity (Figs. 5 and 6). The particle will therefore be subject to a centrifugal force F_2 directed away from O_2 , the magnitude of which, for unit mass, equals V_c^2/R_2 .

The resultant of F_1 and F_2 is indicated by F . If F is tangent to $C-C$ at P , the resultant force on the particle acts along the centroid and there is no tendency for the particle to depart from the assumed centroid. If the curvature of the centroid is too great, the resultant F will be directed inwardly from the tangent to $C-C$, and the actual air flow in the rotor will crowd against the inner boundary of the flow path. Conversely, if the curvature is too small, the air flow will trend toward the outer boundary of the rotor.

The centroid must therefore be determined in accordance with the principle illustrated by Figure 7. Conceivably, this problem might be solved by deriving an equation defining the locus $C-C$. However, the mathematical difficulties appear to be insuperable in view of the complication of the problem by such factors as friction, changing density of the air, and varying area of the flow path.

Various step-by-step methods of computation of the centroid may be employed. I presently prefer a partly graphical method, illustrated in Figure 8, in which the centroid is constructed by drawing short circular arcs about assumed centers. The resulting values for each increment are checked, and the assumed center varied, until the results are consistent. By repeating this process, the entire curve is constructed.

Referring again to Figure 7, since F_2 is normal to the centroid and F should be tangent to the centroid, the desired conditions will result only when F is normal to F_2 . The component of F_1 normal to F must therefore equal F_2 . This component equals $F_1 \cos B$, where B is the angle between F_1 and R_2 , which obviously equals the angle between the tangent to $C-C$ at P and the axis $X-X$.

Substituting in the equation $F_1 \cos B = F_2$ the values of F_1 and F_2 previously stated, and solving for R_2 , we have $R_2 = V_c^2 R_1 / V_t^2 \cos B$.

Referring to Figure 8, the boundaries of the inlet, the centroid of the inlet, the axis of rotation $X-X$, and the split line between the inducer and impeller are indicated. These are laid out, to any suitable scale, for the solution of the problem. The radius of the inlet centroid may be calculated by graphical methods from the physical dimensions and form of the impeller.

The flow is assumed to have no radial component for the first half inch of the path (in a particular large impeller), an assumption which introduces no significant error. Thus, the first half inch is laid out as a straight line from the centroid of the inlet parallel to the axis to the point A . At this point, R_1 is known, angle B is zero, V_c may be taken from the chart (Fig. 5), and V_t is derived

from V_c and the values of U and Q for the coordinates of point A, as indicated in Figure 5.

A chart showing the values of $\tan Q$ in terms of X and R throughout the inducer may be prepared for ready reference. $\tan Q$ is unity in the impeller. For the specific inducer described, $\tan Q$ is proportional to the product of X and R .

The value of R_2 may thus be computed, and is laid out as line AO normal to the axis in Figure 8. With O as center, an arc is drawn to A' . The extent of this arc must be small, as the computation proceeds by small increments. The arc is exaggerated in Figure 8 for clarity. An arc of the order of 2 degrees might be used. Obviously, the arc must approximate the segment of the non-circular locus which it represents.

The computed value of R_2 is checked for the interval AA' by calculating R_2 for the midpoint P of the arc. The radial and axial coordinates of P and the value of angle B at P are determined in the obvious manner, and the basic equation $R_2 = V_c^2 R_1 / V_T^2 \cos B$ is solved for the values of V_c and V_T for the actual coordinates of P . This value for radius R_2 , indicated by the dotted line OP , should closely approach the radius OA . If the discrepancy is greater than one percent, successive approximations may be made. Thus, if OP is greater than OA , a greater value of $R_2 = OA$ is assumed, a new arc is drawn, and the value of R_2 is recomputed for the new coordinates of P . When the equation for the midpoint of the arc is balanced within one percent, the arc AA' may be considered a sufficiently exact approximation to the segment of the desired centroid.

The process is repeated, computing $R_2 = O'A'$ from the coordinates of A' and the value of angle B , V_c , and V_T at A' . The arc $A'A''$ is drawn with center O' on OA' and radius equal to the new value of R_2 . The midpoint of this arc is checked and $O'A'$ modified if necessary, thus determining successive radii $R_2, R_2', R_2'',$ and so on. In this way, a close approximation to the mathematically exact centroid is built up segment by segment, and the final curve may appear substantially as indicated in Figure 7 by $C-C$.

When this has been done, the length of the centroid $C-C$ is measured. If this varies significantly from the value assumed, which is the abscissa of the chart (Fig. 6) of component velocity, a new assumption as to length of $C-C$ is made, the chart is corrected, and the centroid is recomputed. A discrepancy of one percent in the path length is acceptable.

The final operation in defining the form of the rotor consists of determining the inner and outer boundaries of the flow path, based upon the centroid as determined.

If, as illustrated in Figure 9, we draw a normal to the centroid at any point P , and rotate the centroid and the normal about the axis of rotation of the impeller, the centroid will generate a complex surface of revolution and the normal will generate a conical surface. The area of the conical surface between the inner and outer boundaries of the air path through the rotor is taken as the total area A_T of the flow path. The net area A_N of the flow path equals $A_T - A_v$, where A_v is the portion of A_T occupied by the vanes.

The desired value of A_N for any point on the centroid is derived as follows: A curve of static density of the air against path length is plotted, as illustrated in Figure 6. Static density at the inlet and outlet are computed from the stated design conditions. Static density is assumed to remain constant for the first half inch of the path (no compression as the air enters the spaces between the inducer blades) and then to increase linearly to the outlet. The desired net area A_N of the path at any point is equal to the air flow (pounds per second) divided by the product of static density and component velocity. A_N is computed and plotted against path length, as indicated in Figure 6.

The location of the inner and outer envelopes of the flow path is determined so that the portions of the net flow path outside and inside the centroid $C-C$ are of equal area. The desired area may be obtained by properly locating these envelopes. Since $A_T, A_N,$ and A_v do not vary in the same ratio with changes in the depth of the air passage, and no simple relation exists between $A_T, A_N,$ and A_v , a method of successive approximations is used.

Referring to Figure 9, the centroid $C-C$ has been established. To determine the inner and outer boundaries of the flow path, shown in broken lines, a series of points on each boundary are found, each point corresponding to a point on the centroid. With a number of points thus located, curves may be faired through these points.

For any point P , for example, the desired value of A_N at P is determined from the curve (Fig. 6) for the distance of point P from the inlet along the line $C-C$.

The value of A_T sufficient to give the required A_N is estimated. The angle B between the radius OP and the normal to the centroid is measured.

The values of $L_1 = PP'$ and $L_2 = PP''$ corresponding to the assumed value of A_T are determined. The areas of the conical surfaces generated by rotation of these line segments about the axis $X-X$ must each equal one-half A_T .

Therefore, from the geometry of Figure 9,

$$R_3 = \sqrt{R_1^2 - \frac{A_T \cos B}{2\pi}}$$

and

$$R_4 = \sqrt{R_1^2 + \frac{A_T \cos B}{2\pi}}$$

Therefore,

$$L_1 = (R_1 - R_3) / \cos B$$

and

$$L_2 = (R_4 - R_1) / \cos B$$

The total width of the passage, which equals the depth of the blade, is $L_1 + L_2$, which we may call L . Then $A_T = \pi L (R_3 + R_4)$, which relation may be used to check the computation.

Since $A_N = A_T - A_v$, A_v must be determined. A_v may be found with sufficient accuracy from the formula

$$A_v = N L t / \sin Q$$

where N is the number of vanes, t is the average thickness of the vanes, and Q is taken as the value at the centroid. The term $\sin Q$ corrects for the inclination of the blades in the inducer. In the impeller, $\sin Q$ equals unity.

The value of $A_N = A_T - A_v$ obtained by subtraction is checked against the value of A_N originally assumed. If the discrepancy is greater than one percent, the assumed value of A_T is adjusted, and $L_1, L_2, A_v,$ and A_N are recomputed until a satisfactory agreement is reached.

A series of points on the inner and outer boundaries of the blades being thus determined, the form of the rotor is completely established. The casing 21 is formed for a small clearance from the rotor.

The rotor must, of course, be analyzed for centrifugal and other stresses, but this procedure is not material to my invention and need not be explained herein.

The foregoing explanation and description will enable those skilled in the art to practice the invention. It is unnecessary to an understanding of the invention to recite calculations based upon specific values of design parameters or to specify the resulting forms and dimensions; each compressor according to the invention must be designed for its particular environment.

It will be seen from the foregoing that the salient feature of compressors according to my invention is that the radial component of velocity of the air flow is developed

or, in other words, the air is deflected from axial flow to flow with a substantial radial component, by the free action of centrifugal force.

This distinguishes from previous compressors in which the deflection of the air flow from substantially axial to more or less preponderantly radial is either forced by the inner boundary of the flow path or restrained by the outer boundary. In such prior compressors, the forced deflection or forced restraint of deflection results in unequal pressure and velocity distribution across the flow path, reducing the total flow, pressure ratio, and efficiency of the compressor.

The compressor of the invention may be distinguished from prior art compressors by the term "free deflection" compressor, and the principle of the invention may be called the "free deflection" principle.

An example illustrating the benefits of the invention may be cited. The compressor of a previously existing gas turbine engine has been redesigned in accordance with this invention. The new compressor rotor is equal in diameter to the previous rotor. Notwithstanding this, striking improvements in the performance of the engine have been realized. The installation comprises a rotor of fifteen inch radius directly coupled to a turbine. The air flow has been increased twenty-two percent, the pressure ratio has been raised from 4.4 to 4.9, and the efficiency of the compressor from the former seventy-four percent to eighty percent. This improvement could not have been achieved by applying previously known principles of compressor design.

As a result, the specific thrust of the engine in pounds thrust per pound of air has been increased fifteen percent and the fuel consumption for unit thrust has been reduced by twelve percent. It may be noted that these gains involved an increase of less than one half of one percent in compressor R. P. M. The turbine of the engine was redesigned to fit the increased air flow, but the improvement of the engine is not based upon increase in turbine efficiency.

Since this degree of improvement was made in a compressor and an engine which had been highly developed for some years prior to the invention, the importance of the invention will be obvious.

The above description relates to a compressor in which component velocity increases linearly with path length. Obviously, other assumptions as to component velocity could be made, and the principles described are applicable to the design and construction of compressors based on any desired relation between component velocity and path length.

It will be apparent to those skilled in the art that my invention may be practiced in a variety of forms and that the principles thereof are readily capable of wide application. The invention is not to be considered as limited by the detailed description of an illustrative embodiment thereof.

I claim:

1. A centrifugal compressor comprising a vaned rotor rotatable about an axis, means defining an inlet to the rotor for entrance of air axially of the rotor, and means defining a diffuser for air discharged from the outlet of the rotor, the rotor being formed to discharge air radially relative to the rotor, the vanes of the rotor defining an air flow path from inlet to outlet the area of which varies from inlet to outlet to provide an even variation in air velocity along the path, the intercept of the mean surface of the path by a plane containing the axis of rotation being tangent to the resultant of the centrifugal force vectors due to tangential velocity of the air and to velocity of the air in the path defined by the said intercept of the mean surface at each point along the said mean surface.

2. A centrifugal compressor comprising a vaned rotor rotatable about an axis, means defining an inlet to the rotor for entrance of air axially of the rotor, and means

defining a diffuser for air discharged from the outlet of the rotor, the rotor being formed to discharge air radially relative to the rotor, the vanes of the rotor defining an air flow path from inlet to outlet the intercept of the mean surface of which by a plane containing the axis of rotation is tangent to the resultant of the centrifugal force vectors due to tangential velocity of the air and to velocity of the air in the path defined by the said intercept of the mean surface at each point along the said mean surface.

3. A centrifugal compressor of the axial-to-radial-flow type comprising, in combination, means defining an annular inlet and a vaned rotor comprising an inducer portion adapted to accelerate a gas stream circumferentially relative to the rotor axis and an impeller portion adapted to accelerate the gas stream radially and tangentially, the gas flow path between the rotor vanes varying from inlet to outlet of the rotor so as to provide substantially linear variation in the resultant of the axial and radial components of gas velocity, the inner and outer boundaries of the gas flow path through the rotor being so disposed that the centroid of the path substantially coincides with the free deflection path of gas flow through the rotor under design conditions.

4. A centrifugal compressor comprising a vaned rotor with an inducer portion and an impeller portion, the mean flow path through the rotor conforming substantially to that defined, for design conditions of operation, by the following procedure: Assumption of substantially linear increase through the rotor of the component of velocity of the air in a plane containing the rotor axis; and derivation of a flow path originating at the centroid of the inlet and corresponding to the free path of flow of a particle under the effect of centrifugal force and with the component of velocity varying as stated.

5. A centrifugal compressor comprising a vaned rotor with an inducer portion and an impeller portion, the boundaries of the flow path through the rotor conforming substantially to those defined, for design conditions of operation, by the following procedure: assumption of substantially linear increase through the rotor of the component of velocity of the air in a plane containing the rotor axis; derivation of a centroid of the flow path originating at the centroid of the inlet and corresponding to the free path of flow of a particle under the effect of centrifugal force and with the component of velocity varying as stated; assumption of substantially linear increase through the rotor of air density; determination of the required net area of the path as a function of path length for component velocity and density varying as stated; and location of the inner and outer boundaries of the flow path so that the path is divided into two equal parts by the centroid and the net area of the path conforms to that determined as stated above.

6. A centrifugal compressor of the axial-to-radial-flow type comprising, in combination, means defining an annular inlet and a vaned rotor comprising an inducer portion, with vanes decreasing progressively in angular relation to the axis of the rotor from a maximum angle at the inlet to substantially a zero angle at the discharge end of the inducer, and an impeller portion with vanes substantially at a zero angle to the axis, the axial length of the inducer being such that the tangential acceleration of gas flowing through the inducer accords with the tangential acceleration of the gas in the portion of the impeller adjacent the inducer.

7. A centrifugal compressor of the axial-to-radial-flow type comprising, in combination, means defining an annular inlet and a vaned rotor comprising an inducer portion, with vanes decreasing progressively in angular relation to the axis of the rotor from a maximum angle at the inlet to substantially a zero angle at the discharge end of the inducer, the vanes being curved to provide substantially uniform tangential acceleration of gas in the inducer, and an impeller portion with vanes substantially

at a zero angle to the axis, the axial length of the inducer being such that the tangential acceleration of gas flowing through the inducer accords with the tangential acceleration of the gas in the portion of the impeller adjacent the inducer.

8. A compressor comprising, in combination, a rotor rotatable about an axis, means defining an annular inlet for directing gas into the rotor substantially parallel to the axis, and a diffuser for receiving gas discharged from the rotor, the rotor comprising a body in the form of a body of revolution about the axis with vanes extending from the surface of the body substantially normally to the surface, the spaces between adjacent vanes constituting gas flow paths from the inlet to the diffuser trending smoothly from a direction substantially parallel to the axis at the inlet ends thereof to a direction approximately normal to the axis at the discharge ends thereof, the depth of the vanes varying from inlet to outlet so that the area of each path varies as a function of distance along the path according to the relation that the area is the quotient of a constant by the product of two quantities each varying substantially linearly with distance along the path, the said quantities being component velocity of gas in the path and static density of gas in the path.

9. A compressor comprising, in combination, a rotor rotatable about an axis, means defining an annular inlet for directing gas into the rotor substantially parallel to the axis, and a diffuser for receiving gas discharged from the rotor, the rotor comprising a body in the form of a body of revolution about the axis with vanes extending from the surface of the body substantially normally to the surface, the spaces between adjacent vanes constituting gas flow paths from the inlet to the diffuser trending smoothly from a direction substantially parallel to the axis at the inlet ends thereof to a direction approximately normal to the axis at the discharge ends thereof, the form of the surface of the body being such and the depth of the vanes varying from inlet to outlet so as to define a path the locus of the geometrical center of which is tan-

gent at any point along the path to the resultant of the centrifugal force vectors at that point due to tangential velocity of the gas about the said axis and to velocity of the gas in the said path.

10. A compressor as recited in claim 9 in which the area of the path varies with distance along the path so that the velocity of the gas in the path varies linearly with distance along the path.

11. A compressor comprising, in combination, a rotor rotatable about an axis, means defining an annular inlet for directing gas into the rotor substantially parallel to the axis, and a diffuser for receiving gas discharged from the rotor, the rotor comprising a body in the form of a body of revolution about the axis with vanes extending from the surface of the body substantially normally to the surface, the spaces between adjacent vanes constituting gas flow paths from the inlet to the diffuser trending smoothly from a direction substantially parallel to the axis at the inlet ends thereof to a direction approximately normal to the axis at the discharge ends thereof, the form of the surface of the body being such and the depth of the vanes varying from inlet to outlet so as to define a path the locus of the geometrical center of which is the free deflection path for flow of gas from the inlet to the outlet.

References Cited in the file of this patent

UNITED STATES PATENTS

1,097,729	Rice	May 26, 1914
2,228,194	Birkigt	Jan. 7, 1941
2,398,203	Browne	Apr. 9, 1946
2,399,852	Campbell et al.	May 7, 1946

FOREIGN PATENTS

279,426	Great Britain	Aug. 9, 1928
460,985	Germany	June 8, 1928
512,098	Great Britain	of 1939
619,617	Great Britain	of 1949