COMPARISON OF CENTRIFUGAL AND AXIAL FLOW COMPRESSORS FOR SMALL GAS TURBINE APPLICATIONS

by

HEMENDRA JITENDRARAY BHUTA

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Approved by:

Alley H. Duncan
Major Professor

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NOMENCLATURE

A	Area
C_L	Lift coefficient
C_D	Drag coefficient
D	Drag force
E	Energy transfer per pound mass
E_O	Energy transfer
g_c	Universal constant
h	Enthalphy per pound of fluid
i	Angle of attack
L	Lift
M	Mass of fluid or Mach number
m	Mass flow rate
P	Pressure
Q	Volume flow rate or heat flow
r	Radius
s	Distance
T	Temperature
t	time
U	Linear velocity of rotor
u	Internal energy of fluid
V	Absolute velocity of fluid
V_a	Axial component of velocity V
V_m	Radial component of velocity V
V_u	Tangential component of velocity V
V_r	Relative velocity of fluid to rotor

v	Specific volume
c	Density
ω	Angular velocity
τ	Torque
η_h	Hydraulic efficiency
η	Overall efficiency

INTRODUCTION

Of various means of producing mechanical power, the turbine is in many respects the most satisfactory. For many years steam has proved to be a suitable working fluid for turbines, but for the last few years a far simpler arrangement has been used, eliminating the water-to-steam step and using hot gases to drive the turbine. This is the gas turbine.

The gas turbine has many successful applications in the jet propulsion of aircraft, in power plants for ships and locomotives, and in electrical power generating stations. More recent utilization for the gas turbine has been in smaller power generation such as that for some road vehicles and for helicopters. (See Plate I.)

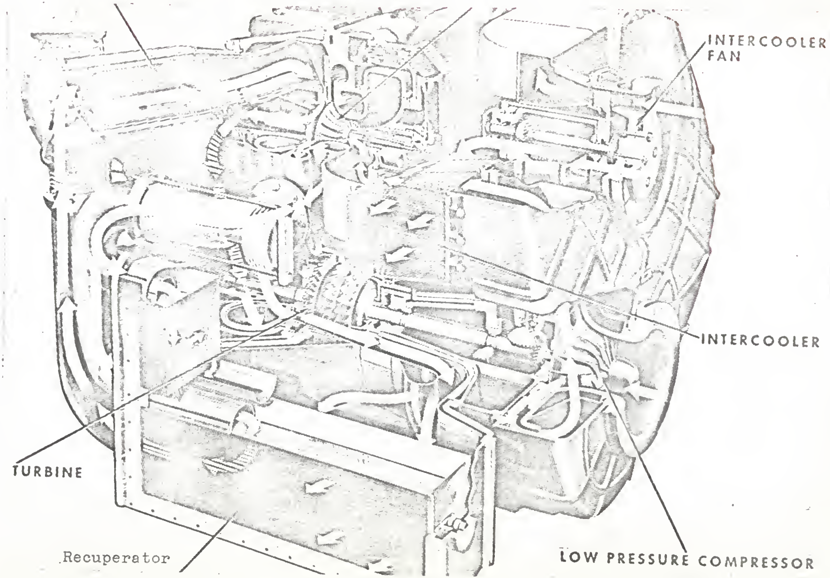
The major components of a gas turbine are a compressor, a combustion chamber, and a turbine. In order to produce expansion through a turbine, a pressure ratio must be provided. Therefore the first necessary step for the cycle of a gas turbine is compression of the working fluid. Two principle types of compressors are used for this purpose, the centrifugal and the axial, (Plates II, III, and IV).

The purpose of this report is to discuss the characteristics, usefulness, and limitations of centrifugal and axial flow compressors used in small gas turbines which develop less than 600 shaft horsepower. In the range of medium power each type of compressor has certain advantages. This information can be used

EXPLANATION OF PLATE I

Centrifugal compressor in Ford's 705 gas turbine engine.

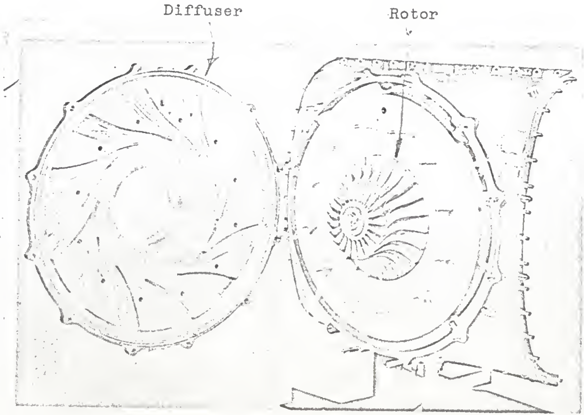
PLATE I


Primary combustor
INTERCOOLER FAN
INTERCOOLER
TURBINE
Recuperator
LOW PRESSURE COMPRESSOR

EXPLANATION OF PLATE II

Typical centrifugal compressor.

PLATE II



EXPLANATION OF PLATE III

Typical axial compressor and turbine.

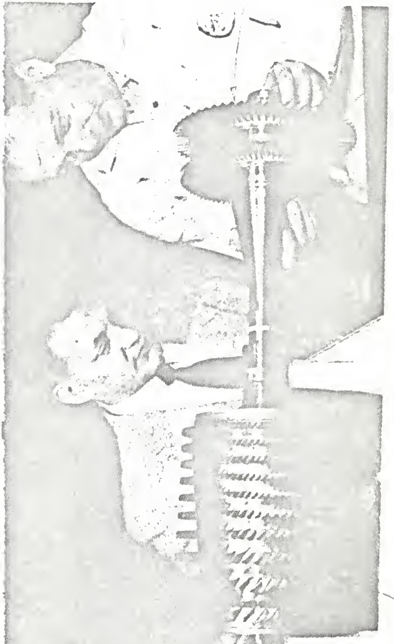
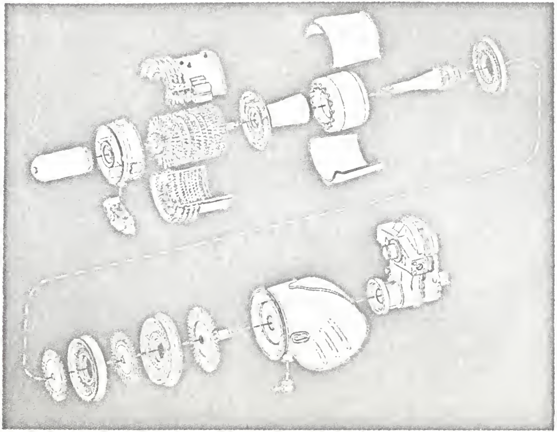


PLATE III

EXPLANATION OF PLATE IV

Exploded view of a gas turbine engine using axial flow compressor.

PLATE IV



to help a design engineer make a selection of a centrifugal or axial flow compressor for a particular application.

Both the centrifugal and the axial compressors are analyzed by use of the same basic principle of fluid flow and thermodynamics. Each type of compressor transfers rotor energy to the fluid. The rate of energy transfer is equal to the product of torque and angular velocity.

This report discusses the basic principles for both compressors and the modifications inherent with each. Performance curves for each of the two types of compressors are presented and used as the topical background for analyzing the useful range of each type on the basis of efficiency, mass flow rate, and physical size. Examples of specific compressors are given, and technical data, properties, and limitations are examined, whereby the advantages and disadvantages of the two compressors may be compared. A discussion of the various applications and uses for both types of compressors is presented.

THE FUNDAMENTAL PRINCIPLE AND ENERGY TRANSFER OF ROTATING MACHINES

The rotating components of a compressor and a turbine have many features in common. They are both concerned with energy transfer. The turbine transfers the fluid energy to the rotor energy or mechanical energy. The compressor transfers the rotor energy to fluid energy. The basic design relationship for all turbomachines is a form of Newton's law of motion applied to a fluid traversing a rotor. So it is possible to set up a quite general equation for the mass flow, the inlet and outlet fluid velocities, the rotor velocities, and work exchange for all forms of rotating mechanisms, such as compressors, centrifugal pumps, and turbines.

First, consider the passage of a fluid through a rotor of any shape as shown in Fig. 1.

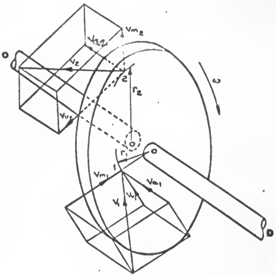


Fig. 1.

The rotor has an axis 0-0 and is turning at a steady rate of ω radians per second. Fluid enters the rotor at 1, passes through the rotor by any path and is discharged at 2. The directions of the fluid at 1 and 2 are at any arbitrary angles, and points 1 and 2 are at any radii r_1 and r_2. The following assumptions are made for the analysis.

1. Flow behaves according to the streamline theory. It follows that any particle of a frictionless fluid

passing through the rotor can be enclosed in a stream tube, and the changes of velocity and pressure can be followed by the Bernoulli equation. Practical coefficients can then be inserted to allow for friction.

2. Flow is continuous and uniform with respect to time at both the inlet and exit points, so interferences due to blades resulting in setting up of pressure and velocity disturbances do not matter so long as they do not interfere with the uniform inlet and exit velocities. This idea of uniform flow means that the uniform velocity determines the kinetic energy, whereas in case of non-uniform flow the mean velocity does not determine the kinetic energy since it varies as the square of the velocity. This assumption permits the determination of velocities from the volume flow with the use of the equation of continuity.
3. All losses outside the rotor are neglected. In general, some fluid will leak through the clearances at the blade tips. All such losses are assumed to be of negligible proportions.

Let V_1 be the velocity of the fluid entering the rotor at any radius r_1. The velocity vector v_1 can be resolved into three mutually perpendicular components. It is convenient to make these three components as follows:

1. Axial component Va_1, in a direction parallel to the axis 0-0.

2. Radial component Vm_1, in a direction normal to 0-0 and passing through it.

3. Tangential component Vu_1, in a plane normal to 0-0 and in a direction normal to a radius.

Similarly, let V_2 be the velocity of the fluid leaving the rotor at any radius r_2, and let Va_2, Vm_2, Vu_2 be the axial, radial, and tangential components of V_2, respectively. The change in magnitude of the axial velocity components through the rotor gives rise to an axial force, which must be taken by a thrust bearing to the stationary rotor casing. The change in the magnitude of the radial velocity components is carried in a similar manner as a journal load. Neither has any effect on the angular motion of the rotor.

The change in magnitude and of radius of the tangential components of velocity corresponds to a change in angular momentum or moment of momentum of the fluid. Newton's law of motion is applied to the fluid passing through the rotor, namely, the time rate of change of angular momentum of a particle about an axis is equal to the moment of the applied force. In other words, the change in angular momentum or moment of momentum of the fluid is equal to the summation of all the applied forces on the rotor, which is equal to the net torque of the rotor.

Let τ = net torque of the rotor

M_1 = mass of fluid entering the rotor at a radius r_1 during time "t"

M_2 = mass of fluid leaving the rotor at a radius r_2 during the same interval of time "t".

Then

Entering angular momentum in time interval "t"

$$= \frac{M_1}{g_c} Vu_1 \; r_1$$

Leaving angular momentum in the same time interval "t"

$$= \frac{M_2}{g_c} Vu_2 \; r_2$$

Thus the change of angular momentum per unit time is

$$\frac{M_1}{g_c t} Vu_1 \; r_1 - \frac{M_2}{g_c t} Vu_2 \; r_2$$

which is equal to the torque acting on the rotor. Therefore

$$\tau = \frac{M_1}{g_c t} r_1 \; Vu_1 - \frac{M_2}{g_c t} r_2 \; Vu_2$$

For steady flow conditions $\frac{M_1}{t} = \frac{M_2}{t} = m$, the rate of mass flow. Then the torque exerted by or acting on the rotor will be

$$\tau = \frac{m}{g_c} (r_1 \; Vu_1 - r_2 \; Vu_2) \qquad (1)$$

The rate of energy transfer, E_0, foot-pounds per second, is the product of torque, foot-pounds, and angular velocity of the rotor, ω, radians per second. So

$$\omega\tau = E_0 = \frac{m}{g_c} \omega(r_1 Vu_1 - r_2 Vu_2)$$

Now $\qquad \omega r = U$, linear velocity of the rotor at radius r.

Thus $\qquad E_0 = \frac{m}{g_c} (\omega r_1) Vu_1 - (\omega r_2) Vu_2$

$$= \frac{m}{g_c} (U_1Vu_1 - U_2Vu_2) \tag{2}$$

Equations (1) and (2) are forms of the Euler equation and are the basic relations for all forms of turbomachines, namely, pumps, compressors, and turbines.

All the energy transfer between the fluid and rotor must be accounted for by the difference of the two UVu terms; thus if Vu is the ideal tangential velocity as given by the ideal velocity triangles, then τ and E are corresponding ideal torque and energy transfer. If Vu is actual tangential velocity for real flow with friction, then τ and E must be actual torque and energy transfer.

Energy transfer implies a change of angular momentum of the fluid, which may be positive or negative. The usual thermodynamic convention is that the work done by the fluid is positive. Hence from the equation

$$E_0 = \frac{m}{g_c} (U_1Vu_1 - U_2Vu_2)$$

for a turbine $U_1Vu_1 > U_2Vu_2$, and for a compressor $U_2Vu_2 \quad U_1Vu_1$. However, in dealing with compressors alone it is convenient to consider positive work when $U_2Vu_2 > U_1Vu_1$.

Velocity Diagrams and Components of Energy Transfer

The Euler equation in terms of angular momentum is the basic form of the energy transfer equations. It is useful to transform this equation into a relationship which is helpful in understanding the physical basis of the energy transfer and is also convenient in certain aspects of design.

Figure 2 shows the ideal velocity diagrams for the inlet and discharge for a generalized rotor.

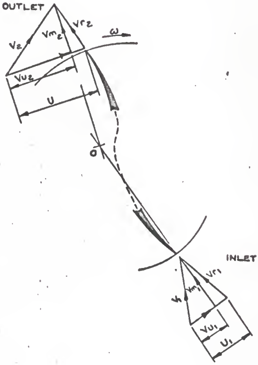


Fig. 2. Velocity diagrams.

Let V = the absolute velocity of the fluid

V_r = the relative velocity of the fluid (relative to the rotor)

U = the linear rotor velocity.

Let all these velocities be in the radial plane. The absolute velocity is resolved into a component Vm passing through the axis (a radial velocity) and a tangential component Vu. By vector principles, the combination of V and U gives the relative velocity V_r. Geometrically,

$$Vm_2^2 = V_2^2 - Vu_2^2$$

and $$Vm_2^2 = V_{r_2}^2 - (U_2 - Vu_2)^2$$

Therefore $$V_2^2 - Vu_2^2 = V_{r_2}^2 - (U_2^2 + Vu_2^2 - 2\,U_2Vu_2\,)$$

$$V_2^2 - Vu_2^2 = V_{r_2}^2 - U_2^2 - Vu_2^2 + 2\,U_2Vu_2$$

$$U_2Vu_2 = \frac{1}{2}\,(V_2^2 + U_2^2 - V_{r_2}^2)$$

Similarly, $$U_1Vu_1 = \frac{1}{2}\,(V_1^2 + U_1^2 - V_{r_1}^2)$$

From equation (2), $E_0 = E = \dfrac{1}{g_c}\,(U_1Vu_1 - U_2Vu_2)$; substituting the values of U_1Vu_1 and U_2Vu_2 in the above equation,

$$E = \frac{1}{g_c}\;\frac{1}{2}\left[V_1^2 + U_1^2 - V_{r_1}^2 - V_2^2 - U_2^2 + V_2^2\right]$$

$$E = \frac{1}{2g_c}\left[(V_1^2 - V_2^2) + (U_1^2 - U_2^2) + (V_{r_2}^2 - V_{r_1}^2)\right] \qquad (3)$$

Thus the energy transfer may be given by the sum of the differences of the squares of absolute fluid velocities V, rotor velocities U, and relative fluid velocities V_r at inlet and outlet of the rotor.

The first term represents energy transfer due to change of kinetic energy. So $\dfrac{(V_1^2 - V_2^2)}{2g_c}$ is the change of absolute kinetic energy or dynamic head occurring in the machine. The second term, $\dfrac{(U_1^2 - U_2^2)}{2g_c}$, represents the change of fluid energy due to the movement of the rotating fluid from one radius of rotation to another. This is centrifugal energy or the head produced by the centrifugal effect.

$\dfrac{(V_{r_2}^{\;2} - V_{r_1}^{\;2})}{2g_c}$ represents the change of kinetic energy due to change of relative velocity. Velocity V_r is relative to the rotor and any change of this velocity results in a change of static head or pressure within the rotor itself, similar to that for the absolute velocity in the casing.

Thus in equation (3), the first represents a change of dynamic head or pressure, and the other two represent a change of static head or pressure. The total head or pressure is thus the sum of the dynamic component and the static component. The relative proportion of each may vary considerably in different designs.

Thermodynamic Analysis

Let Fig. 3 represent a machine through which a gas is flowing at a steady rate of m pounds per second.

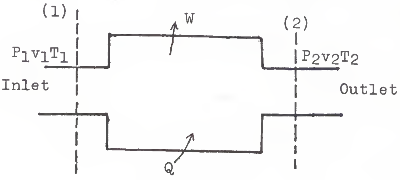


Fig. 3.

Let P_1 and T_1 represent the inlet pressure and temperature of the fluid, respectively, and let v_1, u_1, and h_1 be the specific volume, internal energy, and enthalpy per pound of fluid at the inlet condition. Similarly, P_2, T_2, v_2, u_2, and h_2 represent the pressure, temperature, specific volume, internal energy, and enthalpy at the exit. Let W - Btu per pound of flow, be the work done by the fluid, and Q - Btu per pound of flow, be the heat added during the process.

Applying the first law of thermodynamics to the boundary enclosing the fluid in the machine between sections 1 and 2,

$$dQ = du + dW \qquad (4)$$

(Changes in the kinetic and potential energy are neglected.)

In Fig. 3, assume that pistons are placed in the supply and exhaust pipes and that these pistons are exerting a pressure of P_1 and P_2, respectively. If, at the inlet, the piston moves a distance ds_1, it forces the fluid dm into the machine.

Work done = force x distance

So work done on the gas by piston at 1 = $-P_1\ a_1\ ds_1$, where a_1 is the area of the piston at 1, but $a_1 ds_1 = v_1 dm$, so work done at 1 = $-P_1\ v_1\ dm$. Similarly, work done by the gas at Section 2 is $+P\ v_2\ dm$. The machine is also doing work at the rate of W Btu per pounds of flow. Therefore

$$dW = -P_1\ v_1\ dm + P_2\ v_2\ dm + Wdm.$$

Substituting in dQ = du + dW and assuming the steady flow conditions,

$$Qdm = (U_2 - U_1)dm + Wdm + P_2v_2dm - P_1v_1dm$$

$$Q = U_2 - U_1 + W + P_2v_2 - P_1v_1$$

Rearranging the terms in the above equation,

$$Q = (U_2 + P_2 v_2) - (U_1 + P_1 v_1) + W$$

$$Q = h_2 - h_1 + W$$

or $$W = Q - (h_2 - h_1)$$

In the above analysis, the kinetic energy was neglected. If the kinetic energy is included in the analysis, then

$$W = Q - (h_2 + \frac{V_2^2}{2g_c}) + (h_1 + \frac{V_1^2}{2g_c}) \qquad (5)$$

The above equation gives work for a steady flow process by thermodynamic analysis. The same expression can be obtained by the steady flow energy equation. Consider the steady flow of unit mass of fluid. Then total energy <u>in</u> is equal to the total energy <u>out</u>. The energy of the fluid may be expressed by the internal energy u_1, the flow work Pv, the kinetic energy $\frac{V^2}{2g_c}$ and the potential energy due to position $\frac{gZ}{g_c}$. The change of this total energy is associated with shaft work, W_s, and heat, Q, which is transferred in or out of the rotor. Then by the energy balance

$$u_1 + P_1\, v_1 + \frac{V_1^2}{2g_c} + \frac{g}{g_c} Z_1 + Q$$

$$= U_2 + P_2\, v_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c} Z_2 + W_S$$

As in the thermodynamic analysis, any change of potential energy is neglected. By definition of enthalpy

$$u + P\, v = h$$

Thus

$$h_1 + \frac{V_1^2}{2g_c} + Q = h_2 + \frac{V_2^2}{2g_c} + W_s$$

Rearranging

$$W_s = Q - (h_2 + \frac{V_2^2}{2g_c}) + (h_1 + \frac{V_1^2}{2g_c})$$

Thus for unit flow rate, W_s is the rate of energy transfer. This must be the same as the rate of energy transfer E obtained by the dynamic analysis, that is, $W_s = E$. This is the basic relationship which equates the thermodynamic expression of the problem to that of the dynamics of the energy transfer by means of a turbomachine.

CENTRIFUGAL COMPRESSOR

Theory and Operation

A compressor is a device for increasing pressure of a flowing fluid. The flow of fluid is axial for an axial flow compressor and radial for a centrifugal compressor. A centrifugal compressor consists of four elements, as shown in Figs. 4 and 5.

1. A series of inlet buckets (or vanes) A. These buckets are attached to the shaft so they rotate with the shaft. They are designed in such a way as to guide the gas onto the impeller in an efficient manner.
2. A rotating impeller B. The impeller is fitted with radial vanes. The gas is supplied to these vanes by

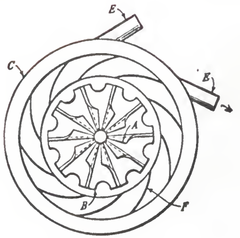


Fig. 4. Elements of a centrifugal compressor.

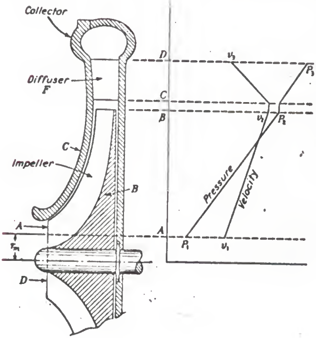


Fig. 5. Sectional view of a centrifugal compressor.

the inlet buckets A, near the axis of rotation.

3. An inlet duct and the outlet ducts E.

4. The diffuser element F.

The shaft and the impeller B are rotated at high speed. Thus the air contained in the passages of the rotating impeller is subjected to a centrifugal force. This force causes the air to flow radially outward, so that new air must enter the inlet of the impeller to take its place. This air is then subjected to the centrifugal force and is displaced radially. Thus a continuous flow results.

In case of an ideal centrifugal compressor it is assumed that the gas is accelerated in the impeller to impeller velocity. Thus the gas would leave the periphery of the impeller with a tangential velocity equal to that of the outer diameter of the impeller, and it will also have a low radial velocity of flow. Thus the effect of the impeller is to give the gas flowing through it a high velocity of flow and a small compression due to centrifugal force. This high velocity gas stream next enters the diffuser passage. The diffuser has gradually increasing passage areas. So as the gas stream passes through the diffuser, its velocity decreases, and the changes of the kinetic energy can be converted into a further increase of pressure.

The changes of pressure and velocity that occur as the air passes through the compressor can be represented schematically as shown in Fig. 5. The gas enters at the mean radius r_m with a low velocity V_1 and at approximately atmospheric pressure P_1. As the gas passes through the impeller, it is accelerated to a high velocity V_2 and a pressure P_2. Then in the diffuser its velocity decreases to V_3 but the pressure increases to P_3.

Mathematical Analysis

Using the momentum principle, it was found that the energy transfer $E = \frac{1}{g_c}(U_1Vu_1 - U_2Vu_2)$. It was transformed into another form

$$E = \frac{1}{2g_c}\left[(V_2^2 - V_1^2) + (U_2^2 - U_1^2) + (V_{r_1}^2 - V_{r_2}^2)\right] \qquad (3)$$

The last two terms in the above equation represent the change of static head or pressure in the rotor, where $(V_{r_1}^2 - V_{r_2}^2)$ is the static head due to the change in the relative velocity of the fluid, and $(U_2^2 - U_2^2)$ is the pressure due to the centrifugal effect. Thus in Fig. 5 the difference between the pressure P_2 and P_1 is given by the sum of the two terms $(U_2^2 - U_1^2)$ and $(V_{r_1}^2 - V_{r_2}^2)$. The pressure rise from P_2 to P_3 is given by the first term of the equation (3), namely, $(V_2^2 - V_1^2)$. That is, $(V_2^2 - V_1^2)$ represents the pressure rise for the diffusion process in the fixed casing following the rotor in the centrifugal compressor.

The Euler equation $E = \frac{1}{g_c}(U_1Vu_1 - U_2Vu_2)$ can be simplified for the centrifugal compressor. Figure 6 shows the inlet velocity diagram for an ideal centrifugal compressor. It is seen that the fluid enters the impeller in the axial direction, that is, the tangential component of the inlet velocity $Vu_1 = 0$.

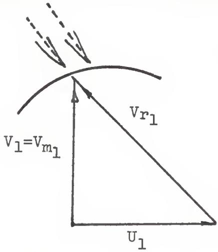


Fig. 6. Inlet velocity diagram for ideal centrifugal compressor.

Substituting $Vu_1 = 0$ in the Euler equation,

$$E = \frac{1}{g_c} (U_1 Vu_1 - U_2 Vu_2)$$

$$E = \frac{1}{g_c} (-U_2 Vu_2) = - \frac{U_2 Vu_2}{g_s}$$

Thus for the axial fluid entry, which is generally the case, the energy transfer is wholly represented by $\dfrac{U_2 Vu_2}{g_c}$.

Efficiency of a Compressor

The general definition of efficiency is the ratio of useful energy delivered to the energy supplied. For the compressor applications the following two efficiencies are considered.

1. Hydraulic efficiency η_h between fluid and rotor:

$$\text{hydraulic} = \eta_h = \frac{\text{useful fluid energy at final discharge}}{\text{mechanical energy supplied to rotor}}$$

2. Overall efficiency η,

$$\eta\,\text{overall} = \eta = \frac{\text{useful fluid energy at final discharge}}{\text{mechanical energy supplied to shaft}} .$$

For the comparison between the centrifugal and axial flow compressors it is convenient to use the overall efficiency η.

AXIAL FLOW COMPRESSOR

Theoretical Analysis

As mentioned earlier an axial flow compressor causes axial flow of fluid, so it has no significant radial component of velocity. Euler's equation gives

$$E = \frac{U_2 Vu_2 - U_1 Vu_1}{g_c} \qquad (2)$$

$$= \frac{1}{2g_c}\left[(V_2^2 - V_1^2) + (U_2^2 - U_1^2) + (Vr_1^2 - Vr_2^2)\right] \qquad (3)$$

For an axial flow compressor where the radial component of the velocity is absent, the centrifugal component of the energy transfer will be zero. Thus

$$E = \frac{1}{2g_c}\left[(V_2{}^2 - V_1{}^2) + (Vr_1{}^2 - Vr_2{}^2)\right]$$

Thus in this case the energy transfer is by means of change of absolute kinetic energy and by relative velocity diffusion effect. As in the case of a centrifugal compressor, a diffuser is needed to convert the kinetic energy or the dynamic head into static pressure. A row of stator blades is commonly used for this purpose. In general, a stage of an axial flow compressor consists of a rotor blade row and a stator blade row. Frequently inlet guide vanes are added before the rotor. These are similar to stator blades included before the rotor. The purpose of the inlet guide vanes is to direct the fluid onto the moving blades at a suitable angle. A stage of an axial flow compressor with inlet guide vanes is shown in Fig. 7. The velocity diagrams for the inlet and outlet conditions are shown. It can be seen from the velocity diagrams that the relative velocity of the fluid is reduced as it passes through the rotor. Thus static pressure increases in the rotor because of the

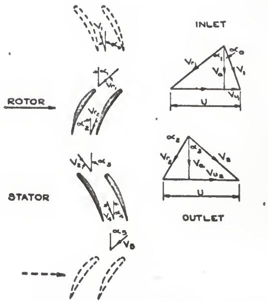


Fig. 7. A stage of an axial flow compressor.

decrease in the relative velocity. Mathematically,
$\frac{(Vr_1^2 - Vr_2^2)}{2g_c}$ gives the pressure increase due to the change in the relative velocity. The absolute velocity is increased in the rotor. This kinetic energy is then converted to static pressure in the stator. Mathematically, it is given by
$\frac{(V_2^2 - V_1^2)}{2g_c}$. Thus for an axial flow compressor the total pressure rise across a stage is given by the sum of the two pressure rises

$$\frac{(V_2^2 - V_1^2)}{2g_c} + \frac{(Vr_1^2 - Vr_2^2)}{2g_c} .$$

Introduction to Airfoil Theory

In designing an axial flow compressor, knowledge of the behavior of the flow of gas past blade elements is important. Airfoil theory is useful in studying this behavior. First, let the airfoil be parallel to the flow, as shown in Fig. 8(a). In this case the gas divides around the body rather symmetrically. It separates at the leading edge of the body and joins again at the trailing edge. The main stream does not suffer a permanent deflection. If the airfoil is not well designed, then, due to friction of the gas, some forces will be applied to the foil. And there will be some disturbance in the flow pattern of the gas. But if the airfoil is well designed, the flow will have only a negligible deflection.

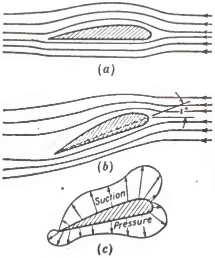


Fig. 8. Gas flow past an airfoil.

Consider the airfoil set at some angle i degree to the gas stream, as shown in Fig. 8(b). The angle i is called the angle of attack. It can be seen from the figure that the flow of the gas suffers a local deflection. And the streamline pattern is changed. By Newton's laws, the deflection of the gas stream is possible only if the blade exerts a force on the gas. Thus the reaction of the gas must produce an equal and opposite force on the airfoil. These forces appear in the form of pressure of the stream on the body. These forces will produce a resultant force R acting on the blade. This resultant force can be resolved into two mutually perpendicular components, as shown in Fig. 9.

1. Lift L--at right angle to the undisturbed air stream
2. Drag D--tending to move the airfoil in the direction of the motion of flow.

By experiments it is possible to measure the lift and drag forces for different values of the flow velocity and angle of attack for various airfoil shapes. Using such values it is possible to define relations between the forces as follows:

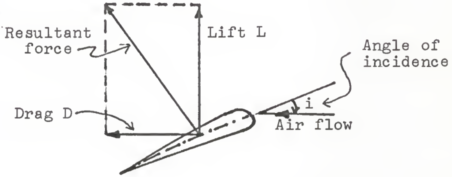


Fig. 9. Forces acting on an airfoil.

$$\text{Lift } L = C_L A \rho \frac{V^2}{2}$$

$$\text{Drag } D = C_D A \rho \frac{V^2}{2}$$

where L = lift force in pounds

D = drag force in pounds

C_L = lift coefficient

C_D = drag coefficient

A = area of surface, square feet

ρ = density of air, slugs per cubic feet

V = velocity of air, feet per second.

The coefficients C_L and C_D can be calculated from wind-tunnel tests. These could be plotted for different angles of attack for any desired blade section. A typical example is given in

Fig. 10. The drag coefficient increases rapidly as the highest lift coefficient is approached so that a larger percentage of available energy is lost in overcoming friction, so efficiency will be reduced. Thus there will be a point at which the most economical operation occurs. This is measured by the amount of effective lift for a given energy supply, which can be calculated for different angles of attack and can be used to select a blade profile for the best efficiency of an axial flow compressor.

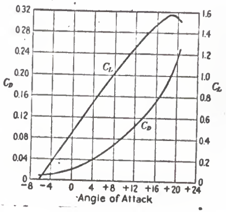


Fig. 10. Typical curves for lift coefficient C_L and drag coefficient C_D.

ANALYSIS AND COMPARISON OF CENTRIFUGAL AND AXIAL FLOW COMPRESSORS

The general form of overall characteristic of a compressor can be given as shown in Fig. 11. The ideal characteristic is given by the straight line H_E based on the Euler relationship where H_E is the energy transferred per unit mass, ft-lb f/lb m, or the change of head of the fluid. This theoretical head will be reduced because of the friction, turbulence, and incidence losses. So the actual characteristic will be of the form shown in Fig. 11. The friction, turbulence, and separation losses are proportional to Q^2, and generally increase rapidly at large values of Q^2 due to high Mach numbers.

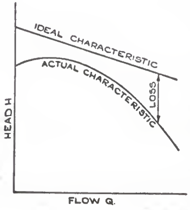


Fig. 11. Typical compressor characteristic.

The incidence losses are proportional to V^2, and hence Q^2, but are also proportional to a drag coefficient. This drag coefficient is minimum at the flow value near the design point and increases either way from the design point. So the incident losses increase either way from the design point.

The discharge losses in the diffuser casing are also proportional to Q^2 and to a critical value of Q where Mach number becomes excessive. Thus because of all these losses, the theoretical linear characteristic is modified to an approximately parabolic form. This actual characteristic may be considerably different for a particular compressor. It may have no maximum, rising continuously from maximum flow to shut-off head, or the characteristic may be steep.

There are some other important phenomena which affect the performance of a compressor. These are choking, stalling of blades, and surging. At high flow rates, it is possible to get a situation such that the mass flow remains fixed regardless of pressure ratio, that is, the characteristic becomes vertical. This is called "choking".

Figure 12 shows the stalling of compressor blades. The blade is said to stall when flow separation occurs on the blade surface as a result of an excessive angle of turn. Fluid

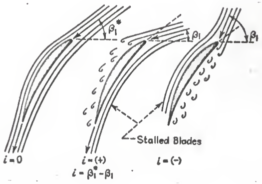


Fig. 12. Compressor blade stalling.

turbulence increases due to this separation of the flow and so the drag coefficient increases, which causes a greater loss in the total pressure. Such conditions are encountered during starting and low speed operation, particularly in axial flow compressors. At low speed there is very little pressure rise, and the design point of the compressor is for high speed and high pressure rise from the first to the last stage. Thus at low speed the absence of pressure rise in the last stages leads to an axial velocity higher than design velocity, and choking occurs. Choking limits the flow rate, which causes excessive angle of turn, and this creates the possibility of stall.

At the lower flow rates another unstable phenomenon known as surging occurs, which could be explained as follows. If surging could be prevented and the complete performance curve for a compressor plotted to zero, it would be of the type shown in Fig. 13. Let the compressor be operating at the surge point S. If the effect of any transient variables, such as a speed

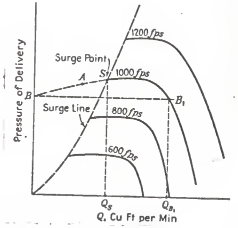


Fig. 13. Effect of surge on characteristic of compressor.

fluctuation or a change of rate of flow, is considered, then instantaneously the compressor may be required to deliver a flow below the surge limit, such as that represented by point A. Now the pressure P_A is less than the pressure P_S, the pressure at the surge point. Under these conditions there is a considerable volume of air already at pressure P_S on the delivery side of the compressor, which is now held in place only by a pressure P_A at the compressor outlet, so back flow from the delivery to the suction side will occur. In order for back flow to occur, the rate of delivery Q must pass through zero before it will reverse. For this the pressure of delivery will travel along the delivery line until point B is reached at pressure P_B. At this point forward flow will begin again at pressure P_B. The pressure then backs up along the curve until P_S is again reached. If the conditions are such that this performance point is stable, the flow will be steady. If, however, the disturbance is still present, the cycle will repeat again. This periodic flow and reversal of flow is the phenomenon of surging and may take place gently, with instability of pressures, or violently, with a series of loud bangs and fluctuations of pressure which can cause structural damage. This will also have the effect of decreasing efficiency of the compressor.

Typical performance charts of a centrifugal compressor and an axial flow compressor are given in Figs. 14 and 15.

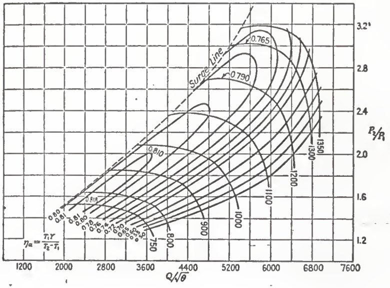


Fig. 14. Typical centrifugal compressor performance chart (axial inlet, radial engine collector case). (Campbell and Talbert, Trans. S.A.E.)

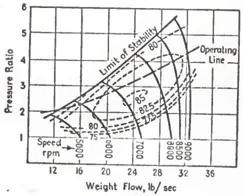


Fig. 15. Typical performance curves of an axial flow compressor showing operating line. (Ponomareff, Westinghouse Engineer.)

Efficiency, Mass Flow, and Pressure Ratio

The centrifugal and axial compressors both have limitations as to pressure ratio per stage. Energy is put into the gas only by accelerating it in the direction of rotation of the machine. The higher the boost per stage, the higher its rate of deflection. There is a limit to the maximum allowable speed at the tip of an inlet vane. Mach number M is defined as the ratio of the local velocity to the sonic velocity. If the Mach number at the tip of the inlet vane is unity or greater, there is a possibility of shock waves which reduces efficiency of the compressor. If the impeller is designed correctly for the desired pressure ratio, then the high Mach number can be avoided. In an actual impeller, however, mechanical limitation assumes the role of controlling factor on the allowable tip speed; this is the stress resulting from the centrifugal forces of the rotating mass, particularly in centrifugal compressors. Also, boundary-layer growth and separation losses tend to increase with increased boost per stage, with resulting tendency toward lower efficiency.

Pressure ratio per stage is a relative quantity depending on which type of compressor is considered. In general, axial-stage pressure ratio is much smaller than that for a centrifugal-stage. In general, centrifugal-stage pressure ratios run between 1.5 and 3.0 for highest efficiency, while optimum axial pressure ratios run between 1.1 and 1.25.

In an axial flow compressor a large number of stages keeps down tangential acceleration and high local Mach numbers, thus avoiding resultant increased boundary-layer separation and shock. This also offers the high overall efficiency desired while achieving high pressure ratios.

In general an axial flow compressor can handle a higher volume flow more effectively than a centrifugal compressor. The axial flow compressor has practical lower limits of flow as a function of pressure ratio. This is shown in Fig. 16. Capacities lower than those indicated in Fig. 16 cannot be effectively met by the axial flow machine and are best handled by the centrifugal type compressor of lower capacity.

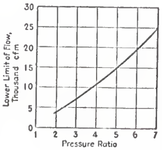


Fig. 16. Practical lower limits of flow for axial flow compressors. (Ponomareff, Westinghouse Engineer.)

The centrifugal compressor has lower efficiency in comparison with the axial compressor. In modern designs of centrifugal compressors for small gas turbines, overall efficiency ranges from 77 to 79 per cent for a pressure ratio of 4.1. An example is the low pressure compressor of gas turbine engine 705 of The Ford Motor Company. (See Plate I, and Fig. 17.) A two-stage centrifugal compressor will operate up to pressure ratio values of 6.0:1 to 7.0 and with efficiencies of 75 to 77 per cent.

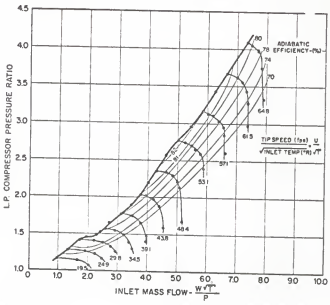


Fig. 17. Compressor characteristics of Ford 705 engine.

The axial compressor has a maximum overall efficiency of 85 to 88 per cent with pressure ratios of 4:1 to 6:1. The pressure ratio attainable per stage of an axial compressor is much smaller than for a centrifugal type. Thus for a pressure ratio of 8.0:1, from 8 to 12 stages will be required. Therefore the cost of the axial compressor is higher than that of a comparable centrifugal type.

Operating Range

An operating range of a compressor may be defined as the ratio of that volume larger than design value at which the efficiency is reduced five points from peak value, to that volume less than design value at which the same five-point reduction in efficiency takes place, or the volume at which surge takes place if reached first.

The axial compressor usually has a shorter range of efficient operation than the centrifugal compressor. In other words, for constant speed operation the centrifugal compressor has a greater range of volume flow for a specified efficiency variation than the axial flow compressor. The typical performance charts of the two compressors show the difference in ranges. This can be shown mathematically as follows.

Figure 15 shows that at a speed giving 2.2:1 maximum pressure ratio and .82 efficiency, the range of a typical axial compressor is approximately $\frac{21}{18} = 1.17$. The range of a typical single-stage centrifugal compressor at the speed to give the same pressure ratio of 2.2:1 is, from Fig. 14, $\frac{5000}{3650} = 1.37$, approximately. The shorter range of axial compressors is mainly due to mismatching of the first and last stage other than the design point. Limitations in the range of flow for the centrifugal compressor result principally from off-design flow conditions at the diffuser inlet and entrance to the impeller vanes. Similar effects of the adverse angles of attack on the rotor and stator blades for off-design operation limit the range of the axial flow compressor.

Physical Size

Physical size is an important factor in considering an application of a particular type of compressor. In general, the centrifugal compressor has a larger diameter than the axial type, but it is considerably shorter in the axial direction. Table 1 gives comparison between the axial compressors and the centrifugal compressors for about the same volume flow and pressure ratios.

Table 1. Comparison of relative physical size for centrifugal and axial compressors.

Pressure ratio	6.0		3.0	
Volume flow, cfm	5300		5300	
Type of compressor	Centrifugal	Axial	Centrifugal	Axial
Number of stages	2	10	6	6
Rotor diameter, in.	13	$8\frac{3}{4}$	13	$8\frac{3}{4}$
Outer diameter, in.	25	$8\frac{3}{4}$	25	$8\frac{3}{4}$
Length, in.	12	30	3.5	20
Geometrical volume, cu.ft.	3.42	1.04	1.0	7

The centrifugal compressor has about three times larger diameter than the axial compressor, but the length of the two-stage centrifugal compressor is about three times shorter than the axial compressor of about the same capacity. The length of the single-stage centrifugal compressor to get a pressure ratio of 3.0 is only 3.5 inches, whereas that for the axial compressor is 20 inches. Thus in smaller sizes the centrifugal compressor is much smaller than the axial compressor. Also, smaller single-stage centrifugal compressors are lighter in weight and relatively simple to design and manufacture, but they have relatively large diameter and bulky multi-stage construction.

There are some other advantages of a centrifugal compressor for small gas turbine applications. It is much less vulnerable to the effects of dust in the intake air than the axial type, where the stator and rotor blades are liable to erosion effects. For an aircraft gas turbine, which operates for most of its life in the air, screening of the intake air may not be essential, but for a ground engine screening of the intake air is essential, especially for the axial type. The single-stage centrifugal compressor is also cheaper to manufacture as it has no multi-stage rotor and stator system.

Applications

Centrifugal and axial gas compressors are widely used in a variety of small gas turbine applications. For very small gas turbine applications the centrifugal compressor is generally used, and for somewhat bigger or medium power applications the axial compressor or the combination of the two is used. Small gas turbines for electric generator sets use centrifugal compressors. An example is the gas turbine power unit Model GTP 70-50 Series made by AiResearch Manufacturing Division of

Arizona of the Garrett Corporation. This 120-horsepower shaft power unit uses a single-stage centrifugal compressor. Centrifugal compressors are also used in gas turbines for missile support units and aircraft support units. A good example of this is Model GTCP85-291, a 90-horsepower unit also made by Garrett Corporation. This unit is also used to provide compressor bleed air for starting the main engines of turbojet or turboprop aircraft engines. It is also used for aircraft cabin air conditioning or for ground applications such as to remove snow, frost, and ice. A centrifugal compressor is also used for gas pumping and marine applications. The Boeing 551 engine, which is a 400-horsepower unit used for this purpose, has a centrifugal compressor. The compressor is 23 inches in diameter and weighs 385 pounds. By comparison, the Solar Aircraft Company's Jupiter engine is a 400-horsepower unit and has a 10-stage axial compressor and weighs around 800 pounds. This engine was fitted in the destroyer the U.S.S. Timmerman in 1951. Similar axial compressors used in turbine engines made by Solar Aircraft Company are used in pleasure boats, tow boats, and commercial fishing boats. These engines are also used in remote pumping stations. Centrifugal and axial compressors are also used for small aircraft and helicopter applications. Messrs. Napier Limited, of England, use a 12-stage axial compressor for small aircraft applications of gas turbines which develops 750 horsepower, whereas AiResearch of Arizona uses a two-stage centrifugal compressor for its gas turbine developing 600 horsepower, for turboprop, turboshaft, or helicopter applications.

Small gas turbines are used in many automobile trucks. The centrifugal compressor is generally used for this purpose; for example, General Motor's GT-309 engine and the Ford Motor Company's 705 gas turbine engine.

Also, centrifugal compressors are used in small gas turbine units for drainage systems, fire fighting, and other water-pumping purposes. Axial flow compressors are generally used for bigger aircraft units or locomotive units where higher power and better efficiency are required.

CONCLUSION

The centrifugal compressor has some advantages over the axial compressor for a small gas turbine unit. It is light-weight and relatively simple to design. It is more stable in operation over a range of speeds and (limited) pressures and less liable to appreciable variations in efficiency with changes of air flow. It has larger diameter than the axial type but it is considerably shorter in the axial direction. It is much less vulnerable to the effects of dust in the intake air than the axial type.

The centrifugal compressor has lower efficiency than the axial compressor, but the pressure ratio attainable per stage of the axial compressor is much smaller than for the centrifugal type. Thus more stages are required for the axial compressor which makes it heavier and more complicated to design.

Centrifugal compressors will be used for smaller gas turbines where the volume flow requirement is low and the pressure ratio is such that it can be obtained by a single stage and where the layer diameter is not a problem.

ACKNOWLEDGMENT

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COMPARISON OF CENTRIFUGAL AND AXIAL FLOW COMPRESSORS FOR SMALL GAS TURBINE APPLICATIONS

by

HEMENDRA JITENDRARAY BHUTA

B. S., Kansas State University, 1964

AN ABSTRACT OF A MASTER'S REPORT

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requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1966

The purpose of this report is to discuss the characteristics, usefulness, and limitations of centrifugal and axial flow compressors used in small gas turbines.

Gas turbines are widely used in power outputs ranging from a few horsepower in portable or auxiliary units, to outputs of over 30,000 horsepower in straightforward shaft units or in jet engines.

The major components of a gas turbine are a compressor, a combustion chamber, and a turbine. The two principle types of compressors used are centrifugal and axial. Both compressors are analyzed by use of the same basic principle of fluid flow of thermodynamics. Each compressor transfers rotor energy to the fluid. The rate of energy transfer is equal to the product of torque and angular velocity.

A general equation for energy transfer between a fluid and a rotor is derived, and, from that, useful equations for each of the two compressors are developed. Basic principles for both compressors are discussed.

Performance charts for both compressors are presented. Efficiency, mass flow rate, useful range, and physical size of the two compressors are discussed and compared.

Centrifugal compressors have lower efficiency than axial compressors but have wider operating range for the same speed. Centrifugal compressors have higher pressure ratio per stage than axial compressors. They are larger in diameter but are considerably shorter in length than axial compressors. Single-stage centrifugal compressors are lighter in weight and simpler to design. Thus for small gas turbines, centrifugal compressors have the advantages of light weight, simplicity, short length, high-pressure ratio per stage, and wider range of operation. And so in many small gas turbine applications such as for driving generators, in missile and aircraft support units, in automobile or truck engines, or for smaller marine applications, centrifugal compressors are more widely used. Axial compressors are widely used in applications where higher power and higher efficiency are required and also where the large diameter of a centrifugal compressor is at a big disadvantage, as in high power jet engines.