5

Axial Flow Compressors and Fans

5.1 INTRODUCTION

As mentioned in Chapter 4, the maximum pressure ratio achieved in centrifugal compressors is about 4:1 for simple machines (unless multi-staging is used) at an efficiency of about 70–80%. The axial flow compressor, however, can achieve higher pressures at a higher level of efficiency. There are two important characteristics of the axial flow compressor—high-pressure ratios at good efficiency and thrust per unit frontal area. Although in overall appearance, axial turbines are very similar, examination of the blade cross-section will indicate a big difference. In the turbine, inlet passage area is greater than the outlet. The opposite occurs in the compressor, as shown in Fig. 5.1.

Thus the process in turbine blades can be described as an accelerating flow, the increase in velocity being achieved by the nozzle. However, in the axial flow compressor, the flow is decelerating or diffusing and the pressure rise occurs when the fluid passes through the blades. As mentioned in the chapter on diffuser design (Chapter 4, Sec. 4.7), it is much more difficult to carry out efficient diffusion due to the breakaway of air molecules from the walls of the diverging passage. The air molecules that break away tend to reverse direction and flow back in the direction of the pressure gradient. If the divergence is too rapid, this may result in the formation of eddies and reduction in useful pressure rise. During acceleration in a nozzle, there is a natural tendency for the air to fill the passage

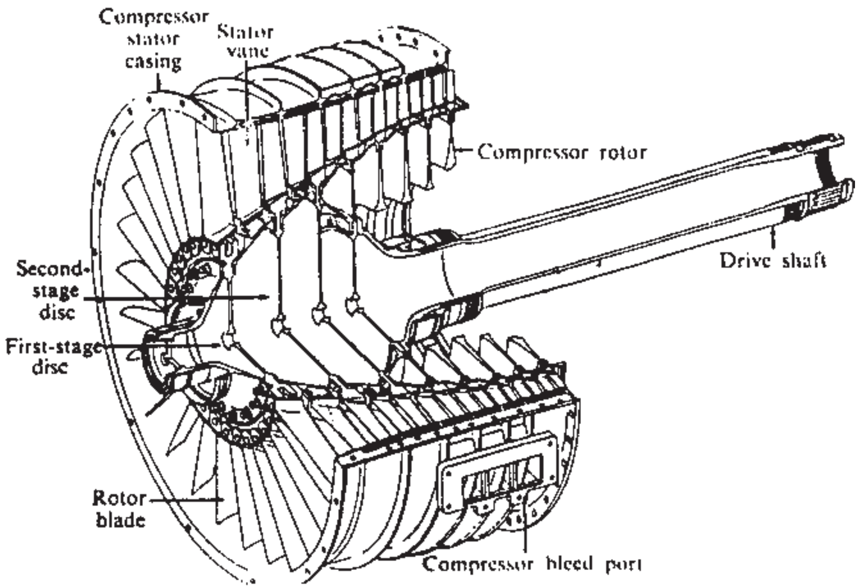

Figure 5.1 Cutaway sketch of a typical axial compressor assembly: the General Electric J85 compressor. (Courtesy of General Electric Co.)

walls closely (only the normal friction loss will be considered in this case). Typical blade sections are shown in Fig. 5.2. Modern axial flow compressors may give efficiencies of 86–90%—compressor design technology is a well-developed field. Axial flow compressors consist of a number of stages, each stage being formed by a stationary row and a rotating row of blades.

Figure 5.3 shows how a few compressor stages are built into the axial compressor. The rotating blades impart kinetic energy to the air while increasing air pressure and the stationary row of blades redirect the air in the proper direction and convert a part of the kinetic energy into pressure. The flow of air through the compressor is in the direction of the axis of the compressor and, therefore, it is called an axial flow compressor. The height of the blades is seen to decrease as the fluid moves through the compressor. As the pressure increases in the direction of flow, the volume of air decreases. To keep the air velocity the same for each stage, the blade height is decreased along the axis of the compressor. An extra row of fixed blades, called the inlet guide vanes, is fitted to the compressor inlet. These are provided to guide the air at the correct angle onto the first row of moving blades. In the analysis of the highly efficient axial flow compressor, the 2-D flow through the stage is very important due to cylindrical symmetry.

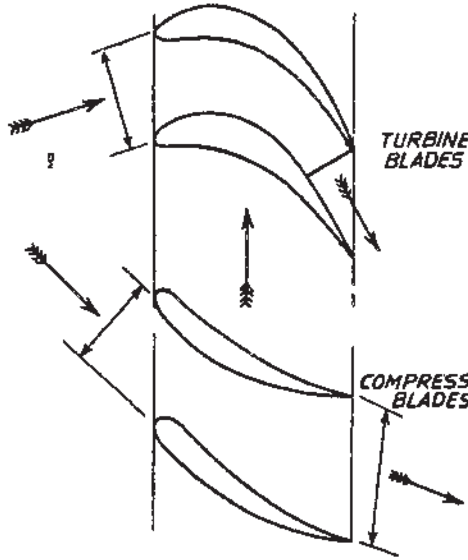

Figure 5.2 Compressor and turbine blade passages: turbine and compressor housing.

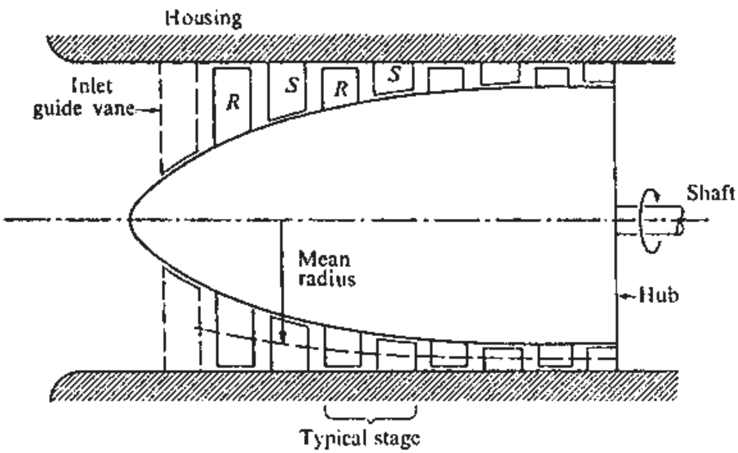

Figure 5.3 Schematic of an axial compressor section.

The flow is assumed to take place at a mean blade height, where the blade peripheral velocities at the inlet and outlet are the same. No flow is assumed in the radial direction.

5.2 VELOCITY DIAGRAM

The basic principle of axial compressor operation is that kinetic energy is imparted to the air in the rotating blade row, and then diffused through passages of both rotating and stationary blades. The process is carried out over multiple numbers of stages. As mentioned earlier, diffusion is a deceleration process. It is efficient only when the pressure rise per stage is very small. The blading diagram and the velocity triangle for an axial flow compressor stage are shown in Fig. 5.4.

Air enters the rotor blade with absolute velocity C_1 at an angle α_1 measured from the axial direction. Air leaves the rotor blade with absolute velocity C_2 at an angle α_2. Air passes through the diverging passages formed between the rotor blades. As work is done on the air in the rotor blades, C_2 is larger than C_1. The rotor row has tangential velocity U. Combining the two velocity vectors gives the relative velocity at inlet V_1 at an angle β_1. V_2 is the relative velocity at the rotor outlet. It is less than V_1, showing diffusion of the relative velocity has taken place with some static pressure rise across the rotor blades. Turning of the air towards the axial direction is brought about by the camber of the blades. Euler's equation

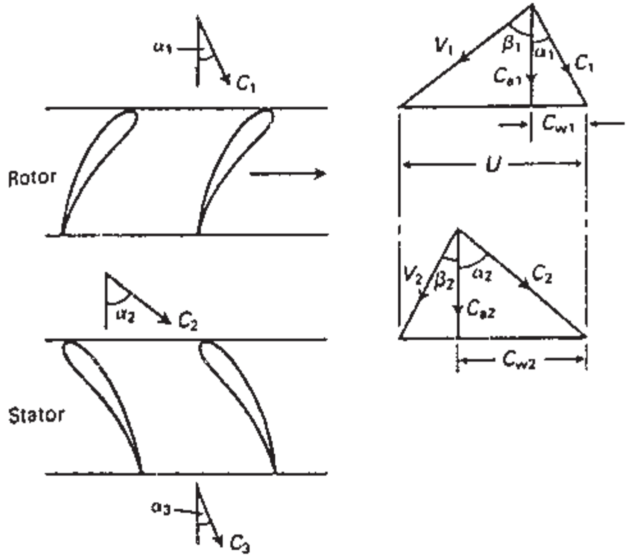

Figure 5.4 Velocity diagrams for a compressor stage.

provides the work done on the air:

$$W_c = U(C_{w2} - C_{w1}) \tag{5.1}$$

Using the velocity triangles, the following basic equations can be written:

$$\frac{U}{C_a} = \tan \alpha_1 + \tan \beta_1 \tag{5.2}$$

$$\frac{U}{C_a} = \tan \alpha_2 + \tan \beta_2 \tag{5.3}$$

in which $C_a = C_{a1} = C_2$ is the axial velocity, assumed constant through the stage. The work done equation [Eq. (5.1)] may be written in terms of air angles:

$$W_c = UC_a(\tan \alpha_2 - \tan \alpha_1) \tag{5.4}$$

also,

$$W_c = UC_a(\tan \beta_1 - \tan \beta_2) \tag{5.5}$$

The whole of this input energy will be absorbed usefully in raising the pressure and velocity of the air and for overcoming various frictional losses. Regardless of the losses, all the energy is used to increase the stagnation temperature of the air, $\triangle T_{0s}$. If the velocity of air leaving the first stage C_3 is made equal to C_1, then the stagnation temperature rise will be equal to the static temperature rise, $\triangle T_s$. Hence:

$$T_{0s} = \Delta T_s = \frac{UC_a}{C_p}(\tan \beta_1 - \tan \beta_2) \tag{5.6}$$

Equation (5.6) is the theoretical temperature rise of the air in one stage. In reality, the stage temperature rise will be less than this value due to 3-D effects in the compressor annulus. To find the actual temperature rise of the air, a factor λ, which is between 0 and 100%, will be used. Thus the actual temperature rise of the air is given by:

$$T_{0s} = \frac{\lambda UC_a}{C_p}(\tan \beta_1 - \tan \beta_2) \tag{5.7}$$

If R_s is the stage pressure ratio and η_s is the stage isentropic efficiency, then:

$$R_s = \left[1 + \frac{\eta_s \Delta T_{0s}}{T_{01}}\right]^{\gamma/(\gamma-1)} \tag{5.8}$$

where T_{01} is the inlet stagnation temperature.

5.3 DEGREE OF REACTION

The degree of reaction, Λ, is defined as:

$$\Lambda = \frac{\text{Static enthalpy rise in the rotor}}{\text{Static enthalpy rise in the whole stage}} \tag{5.9}$$

The degree of reaction indicates the distribution of the total pressure rise into the two types of blades. The choice of a particular degree of reaction is important in that it affects the velocity triangles, the fluid friction and other losses.

Let:

ΔT_A = the static temperature rise in the rotor

ΔT_B = the static temperature rise in the stator

Using the work input equation [Eq. (5.4)], we get:

$$W_c = C_p(\Delta T_A + \Delta T_B) = \Delta T_S$$

$$\left.\begin{aligned} &= UC_a(\tan\beta_1 - \tan\beta_2) \\ &= UC_a(\tan\alpha_2 - \tan\alpha_1) \end{aligned}\right\} \tag{5.10}$$

But since all the energy is transferred to the air in the rotor, using the steady flow energy equation, we have:

$$W_c = C_p\Delta T_A + \frac{1}{2}(C_2^2 - C_1^2) \tag{5.11}$$

Combining Eqs. (5.10) and (5.11), we get:

$$C_p\Delta T_A = UC_a(\tan\alpha_2 - \tan\alpha_1) - \frac{1}{2}(C_2^2 - C_1^2)$$

from the velocity triangles,

$$C_2 = C_a\cos\alpha_2 \quad \text{and} \quad C_1 = C_a\cos\alpha_1$$

Therefore,

$$\begin{aligned} C_p\Delta T_A &= UC_a(\tan\alpha_2 - \tan\alpha_1) - \tfrac{1}{2}C_a^2(\sec^2\alpha_2 - \sec^2\alpha_1) \\ &= UC_a(\tan\alpha_2 - \tan\alpha_1) - \tfrac{1}{2}C_a^2(\tan^2\alpha_2 - \tan^2\alpha_1) \end{aligned}$$

Using the definition of degree of reaction,

$$\begin{aligned} \Lambda &= \frac{\Delta T_A}{\Delta T_A + \Delta T_B} \\ &= \frac{UC_a(\tan\alpha_2 - \tan\alpha_1) - \tfrac{1}{2}C_a^2(\tan^2\alpha_2 - \tan^2\alpha_1)}{UC_a(\tan\alpha_2 - \tan\alpha_1)} \\ &= 1 - \tfrac{C_a}{U}(\tan\alpha_2 + \tan\alpha_1) \end{aligned}$$

But from the velocity triangles, adding Eqs. (5.2) and (5.3),

$$\frac{2U}{C_a} = (\tan\alpha_1 + \tan\beta_1 + \tan\alpha_2 + \tan\beta_2)$$

Therefore,

$$\Lambda = \frac{C_a}{2U}\left(\frac{2U}{C_a} - \frac{2U}{C_a} + \tan\beta_1 + \tan\beta_2\right)$$

$$= \frac{C_a}{2U}(\tan\beta_1 + \tan\beta_2) \tag{5.12}$$

Usually the degree of reaction is set equal to 50%, which leads to this interesting result:

$$(\tan\beta_1 + \tan\beta_2) = \frac{U}{C_a}.$$

Again using Eqs. (5.1) and (5.2),

$$\tan\alpha_1 = \tan\beta_2, \quad \text{i.e.,} \quad \alpha_1 = \beta_2$$

$$\tan\beta_1 = \tan\alpha_2, \quad \text{i.e.,} \quad \alpha_2 = \beta_1$$

As we have assumed that C_a is constant through the stage,

$$C_a = C_1\cos\alpha_1 = C_3\cos\alpha_3.$$

Since we know $C_1 = C_3$, it follows that $\alpha_1 = \alpha_3$. Because the angles are equal, $\alpha_1 = \beta_2 = \alpha_3$, and $\beta_1 = \alpha_2$. Under these conditions, the velocity triangles become symmetric. In Eq. (5.12), the ratio of axial velocity to blade velocity is called the flow coefficient and denoted by Φ. For a reaction ratio of 50%, $(h_2 - h_1) = (h_3 - h_1)$, which implies the static enthalpy and the temperature increase in the rotor and stator are equal. If for a given value of C_a/U, β_2 is chosen to be greater than α_2 (Fig. 5.5), then the static pressure rise in the rotor is greater than the static pressure rise in the stator and the reaction is greater than 50%.

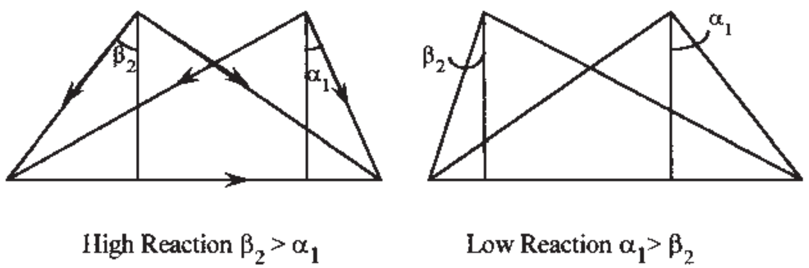

Figure 5.5 Stage reaction.

Conversely, if the designer chooses β_2 less than β_1, the stator pressure rise will be greater and the reaction is less than 50%.

5.4 STAGE LOADING

The stage-loading factor Ψ is defined as:

$$\begin{aligned}\Psi &= \frac{W_c}{mU^2} = \frac{h_{03} - h_{01}}{U^2} \\ &= \frac{\lambda(C_{w2} - C_{w1})}{U} \\ &= \frac{\lambda C_a}{U}(\tan\alpha_2 - \tan\alpha_1) \\ \Psi &= \lambda\Phi\,(\tan\alpha_2 - \tan\alpha_1)\end{aligned} \tag{5.13}$$

5.5 LIFT-AND-DRAG COEFFICIENTS

The stage-loading factor Ψ may be expressed in terms of the lift-and-drag coefficients. Consider a rotor blade as shown in Fig. 5.6, with relative velocity vectors V_1 and V_2 at angles β_1 and β_2. Let $\tan(\beta_m) = (\tan(\beta_1) + \tan(\beta_2))/2$. The flow on the rotor blade is similar to flow over an airfoil, so lift-and-drag forces will be set up on the blade while the forces on the air will act on the opposite direction.

The tangential force on each moving blade is:

$$F_x = L\cos\beta_m + D\sin\beta_m$$

$$F_x = L\cos\beta_m\left[1 + \left(\frac{C_D}{C_L}\right)\tan\beta_m\right] \tag{5.14}$$

where: L = lift and D = drag.

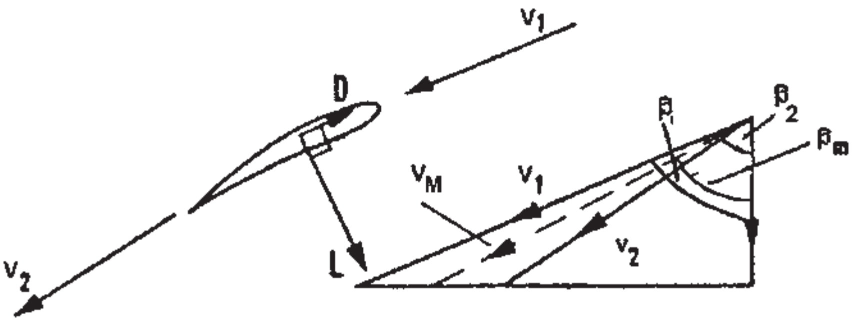

Figure 5.6 Lift-and-drag forces on a compressor rotor blade.

The lift coefficient is defined as:

$$C_{\mathrm{L}} = \frac{L}{0.5\rho V_{\mathrm{m}}^2 A} \tag{5.15}$$

where the blade area is the product of the chord c and the span l.

Substituting $V_{\mathrm{m}} = \frac{C_{\mathrm{a}}}{\cos\beta_{\mathrm{m}}}$ into the above equation,

$$F_{\mathrm{x}} = \frac{\rho C_{\mathrm{a}}^2 c l C_{\mathrm{L}}}{2} \sec\beta_{\mathrm{m}} \left[1 + \left(\frac{C_{\mathrm{D}}}{C_{\mathrm{L}}}\right) \tan\beta_{\mathrm{m}}\right] \tag{5.16}$$

The power delivered to the air is given by:

$$\begin{aligned} UF_{\mathrm{x}} &= m(h_{03} - h_{01}) \\ &= \rho C_{\mathrm{a}} l s (h_{03} - h_{01}) \end{aligned} \tag{5.17}$$

considering the flow through one blade passage of width s.

Therefore,

$$\begin{aligned} &= \frac{h_{03} - h_{01}}{U^2} \\ &= \frac{F_{\mathrm{x}}}{\rho C_{\mathrm{a}} l s U} \\ &= \frac{1}{2}\left(\frac{C_{\mathrm{a}}}{U}\right)\left(\frac{c}{s}\right) \sec\beta_{\mathrm{m}}(C_{\mathrm{L}} + C_{\mathrm{D}} \tan\beta_{\mathrm{m}}) \\ &= \frac{1}{2}\left(\frac{c}{s}\right) \sec\beta_{\mathrm{m}}(C_{\mathrm{L}} + C_{\mathrm{D}} \tan\beta_{\mathrm{m}}) \end{aligned} \tag{5.18}$$

For a stage in which $\beta_{\mathrm{m}} = 45°$, efficiency will be maximum. Substituting this back into Eq. (5.18), the optimal blade-loading factor is given by:

$$\Psi_{\mathrm{opt}} = \frac{\varphi}{\sqrt{2}}\left(\frac{c}{s}\right)(C_{\mathrm{L}} + C_{\mathrm{D}}) \tag{5.19}$$

For a well-designed blade, C_{D} is much smaller than C_{L}, and therefore the optimal blade-loading factor is approximated by:

$$\Psi_{\mathrm{opt}} = \frac{\varphi}{\sqrt{2}}\left(\frac{c}{s}\right) C_{\mathrm{L}} \tag{5.20}$$

5.6 CASCADE NOMENCLATURE AND TERMINOLOGY

Studying the 2-D flow through cascades of airfoils facilitates designing highly efficient axial flow compressors. A cascade is a row of geometrically similar blades arranged at equal distance from each other and aligned to the flow direction. Figure 5.7, which is reproduced from Howell's early paper on cascade theory and performance, shows the standard nomenclature relating to airfoils in cascade.

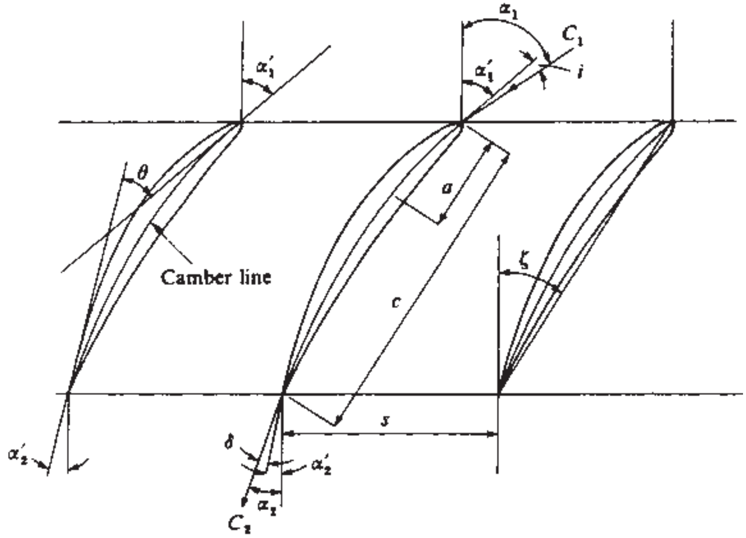

Figure 5.7 Cascade nomenclature.

α_1' and α_2' are the camber angles of the entry and exit tangents the camber line makes with the axial direction. The blade camber angle $\theta = \alpha_1' - \alpha_2'$. The chord c is the length of the perpendicular of the blade profile onto the chord line. It is approximately equal to the linear distance between the leading edge and the trailing edge. The stagger angle ξ is the angle between the chord line and the axial direction and represents the angle at which the blade is set in the cascade. The pitch s is the distance in the direction of rotation between corresponding points on adjacent blades. The incidence angle i is the difference between the air inlet angle (α_1) and the blade inlet angle $\left(\alpha_1'\right)$. That is, $i = \alpha_1 - \alpha_1'$. The deviation angle (δ) is the difference between the air outlet angle (α_2) and the blade outlet angle $\left(\alpha_2'\right)$. The air deflection angle, $\varepsilon = \alpha_1 - \alpha_2$, is the difference between the entry and exit air angles.

A cross-section of three blades forming part of a typical cascade is shown in Fig. 5.7. For any particular test, the blade camber angle θ, its chord c, and the pitch (or space) s will be fixed and the blade inlet and outlet angles α_1' and α_2' are determined by the chosen setting or stagger angle ξ. The angle of incidence, i, is then fixed by the choice of a suitable air inlet angle α_1, since $i = \alpha_1 - \alpha_1'$. An appropriate setting of the turntable on which the cascade is mounted can accomplish this. With the cascade in this position the pressure and direction measuring instruments are then traversed along the blade row in the upstream and downstream position. The results of the traverses are usually presented as shown

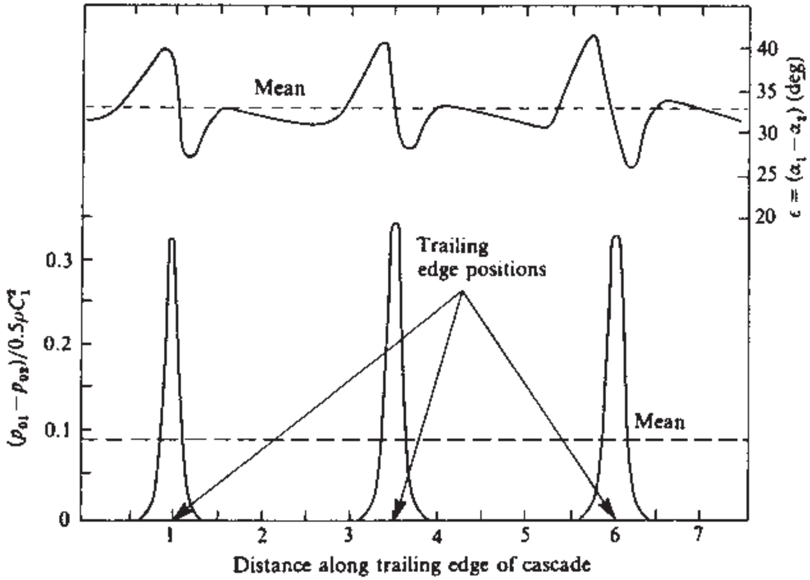

Figure 5.8 Variation of stagnation pressure loss and deflection for cascade at fixed incidence.

in Fig. 5.8. The stagnation pressure loss is plotted as a dimensionless number given by:

$$\text{Stagnation pressure loss coefficient} = \frac{P_{01} - P_{02}}{0.5\rho C_1^2} \tag{5.21}$$

This shows the variation of loss of stagnation pressure and the air deflection, $\varepsilon = \alpha_1 - \alpha_2$, covering two blades at the center of the cascade. The curves of Fig. 5.8 can now be repeated for different values of incidence angle, and the whole set of results condensed to the form shown in Fig. 5.9, in which the mean loss and mean deflection are plotted against incidence for a cascade of fixed geometrical form.

The total pressure loss owing to the increase in deflection angle of air is marked when i is increased beyond a particular value. The stalling incidence of the cascade is the angle at which the total pressure loss is twice the minimum cascade pressure loss. Reducing the incidence i generates a negative angle of incidence at which stalling will occur.

Knowing the limits for air deflection without very high (more than twice the minimum) total pressure loss is very useful for designers in the design of efficient compressors. Howell has defined nominal conditions of deflection for

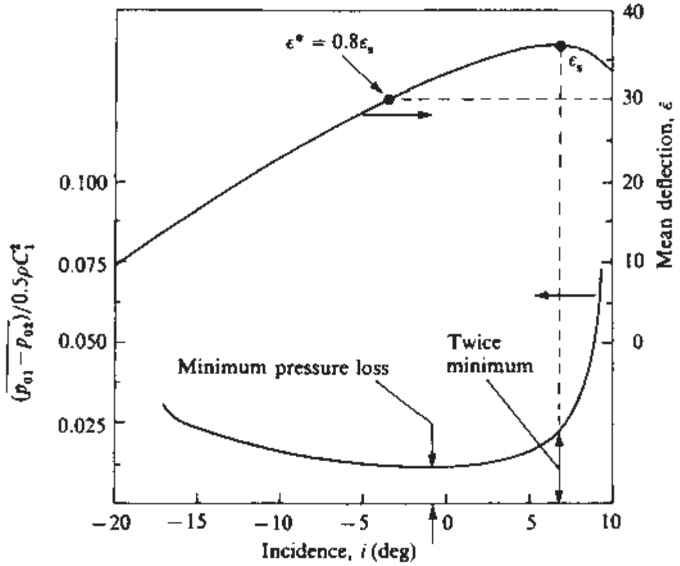

Figure 5.9 Cascade mean deflection and pressure loss curves.

a cascade as 80% of its stalling deflection, that is:

$$\varepsilon^* = 0.8\varepsilon_s \tag{5.22}$$

where ε_s is the stalling deflection and ε^* is the nominal deflection for the cascade.

Howell and Constant also introduced a relation correlating nominal deviation δ^* with pitch chord ratio and the camber of the blade. The relation is given by:

$$\delta^* = m\theta\left(\frac{s}{l}\right)^n \tag{5.23}$$

For compressor cascade, $n = \frac{1}{2}$, and for the inlet guide vane in front of the compressor, $n = 1$. Hence, for a compressor cascade, nominal deviation is given by:

$$\delta^* = m\theta\left(\frac{s}{l}\right)^{\frac{1}{2}} \tag{5.24}$$

The approximate value suggested by Constant is 0.26, and Howell suggested a modified value for m:

$$m = 0.23\left(\frac{2a}{l}\right)^2 + 0.1\left(\frac{\alpha_2^*}{50}\right) \tag{5.25}$$

where the maximum camber of the cascade airfoil is at a distance a from the leading edge and α_2^* is the nominal air outlet angle.

Then,

$$\alpha_2^* = \beta_2 + \delta^*$$

$$= \beta_2 + m\theta\left(\frac{s}{l}\right)^{\frac{1}{2}}$$

and,

$$\alpha_1^* - \alpha_2^* = \varepsilon^*$$

or:

$$\alpha_1^* = \alpha_2^* + \varepsilon^*$$

Also,

$$i^* = \alpha_1^* - \beta_1 = \alpha_2^* + \varepsilon^* - \beta_1$$

5.7 3-D CONSIDERATION

So far, all the above discussions were based on the velocity triangle at one particular radius of the blading. Actually, there is a considerable difference in the velocity diagram between the blade hub and tip sections, as shown in Fig. 5.10.

The shape of the velocity triangle will influence the blade geometry, and, therefore, it is important to consider this in the design. In the case of a compressor with high hub/tip ratio, there is little variation in blade speed from root to tip. The shape of the velocity diagram does not change much and, therefore, little variation in pressure occurs along the length of the blade. The blading is of the same section at all radii and the performance of the compressor stage is calculated from the performance of the blading at the mean radial section. The flow along the compressor is considered to be 2-D. That is, in 2-D flow only whirl and axial flow velocities exist with no radial velocity component. In an axial flow compressor in which high hub/tip radius ratio exists on the order of 0.8, 2-D flow in the compressor annulus is a fairly reasonable assumption. For hub/tip ratios lower than 0.8, the assumption of two-dimensional flow is no longer valid. Such compressors, having long blades relative to the mean diameter, have been used in aircraft applications in which a high mass flow requires a large annulus area but a small blade tip must be used to keep down the frontal area. Whenever the fluid has an angular velocity as well as velocity in the direction parallel to the axis of rotation, it is said to have "vorticity." The flow through an axial compressor is vortex flow in nature. The rotating fluid is subjected to a centrifugal force and to balance this force, a radial pressure gradient is necessary. Let us consider the pressure forces on a fluid element as shown in Fig. 5.10. Now, resolve

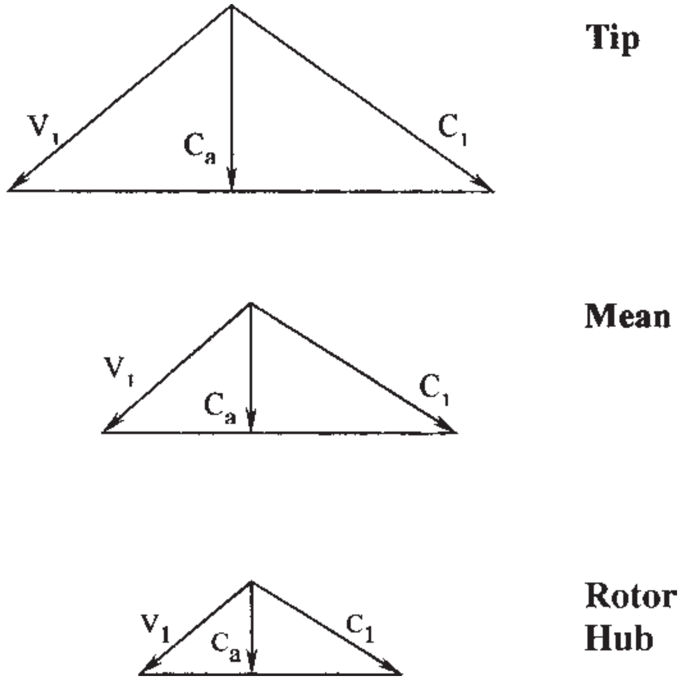

Figure 5.10 Variation of velocity diagram along blade.

the forces in the radial direction Fig. 5.11:

$$\mathrm{d}\theta(P+\mathrm{d}P)(r+\mathrm{d}r) - Pr\,\mathrm{d}\theta - 2\left(P+\frac{\mathrm{d}P}{2}\right)\mathrm{d}r\frac{\mathrm{d}\theta}{2}$$
$$= \rho\,\mathrm{d}r\,r\,\mathrm{d}\theta\frac{C_\mathrm{w}^2}{r} \tag{5.26}$$

or

$$(P+\mathrm{d}P)(r+\mathrm{d}r) - Pr - \left(P+\frac{\mathrm{d}P}{2}\right)\mathrm{d}r = \rho\,\mathrm{d}r\,C_\mathrm{w}^2$$

where: P is the pressure, ρ, the density, C_w, the whirl velocity, r, the radius.

After simplification, we get the following expression:

$$Pr + P\,\mathrm{d}r + r\,\mathrm{d}P + \mathrm{d}P\,\mathrm{d}r - Pr + \rho\,\mathrm{d}r - \frac{1}{2}\mathrm{d}P\,\mathrm{d}r = \rho\,\mathrm{d}r\,C_\mathrm{w}^2$$

or:

$$r\,\mathrm{d}P = \rho\,\mathrm{d}r\,C_\mathrm{w}^2$$

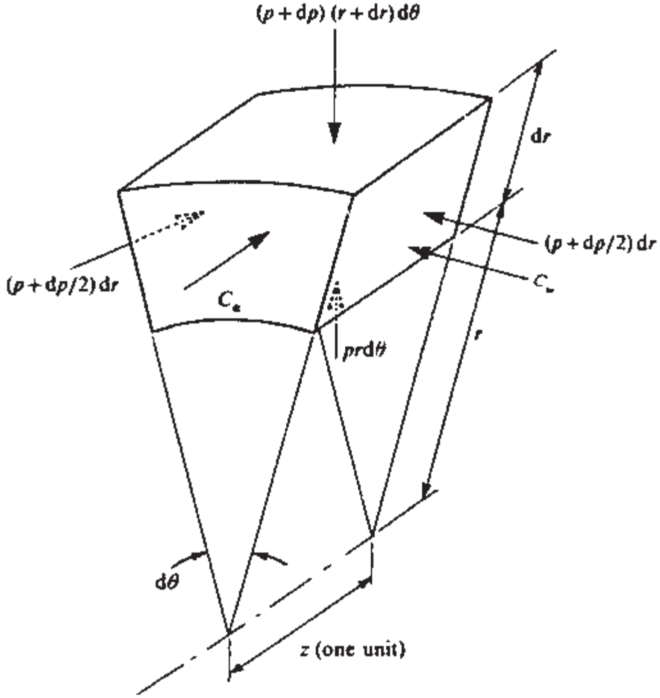

Figure 5.11 Pressure forces on a fluid element.

That is,

$$\frac{1}{\rho}\frac{dP}{dr} = \frac{C_w^2}{r} \tag{5.27}$$

The approximation represented by Eq. (5.27) has become known as radial equilibrium.

The stagnation enthalpy h_0 at any radius r where the absolute velocity is C may be rewritten as:

$$h_0 = h + \frac{1}{2}C_a^2 + \frac{1}{2}C_w^2; \left(h = c_p T, \quad \text{and} \quad C^2 = C_a^2 + C_w^2\right)$$

Differentiating the above equation w.r.t. r and equating it to zero yields:

$$\frac{dh_0}{dr} = \frac{\gamma}{\gamma - 1} \times \frac{1}{\rho}\frac{dP}{dr} + \frac{1}{2}\left(0 + 2C_w \frac{dC_w}{dr}\right)$$

or:

$$\frac{\gamma}{\gamma-1}\times\frac{1}{\rho}\frac{\mathrm{d}P}{\mathrm{d}r}+C_{\mathrm{w}}\frac{\mathrm{d}C_{\mathrm{w}}}{\mathrm{d}r}=0$$

Combining this with Eq. (5.27):

$$\frac{\gamma}{\gamma-1}\frac{C_{\mathrm{w}}^2}{r}+C_{\mathrm{w}}\frac{\mathrm{d}C_{\mathrm{w}}}{\mathrm{d}r}=0$$

or:

$$\frac{\mathrm{d}C_{\mathrm{w}}}{\mathrm{d}r}=-\frac{\gamma}{\gamma-1}\frac{C_{\mathrm{w}}}{r}$$

Separating the variables,

$$\frac{\mathrm{d}C_{\mathrm{w}}}{C_{\mathrm{w}}}=-\frac{\gamma}{\gamma-1}\frac{\mathrm{d}r}{r}$$

Integrating the above equation

$$\int\frac{\mathrm{d}C_{\mathrm{w}}}{C_{\mathrm{w}}}=-\frac{\gamma}{\gamma-1}\int\frac{\mathrm{d}r}{r}$$

$$-\frac{\gamma}{\gamma-1}\ \ln C_{\mathrm{w}}r=c \quad \text{where } c \text{ is a constant.}$$

Taking antilog on both sides,

$$\frac{\gamma}{\gamma-1}\times C_{\mathrm{w}}\times r=e^{\mathrm{c}}$$

Therefore, we have

$$C_{\mathrm{w}}r=\text{constant} \tag{5.28}$$

Equation (5.28) indicates that the whirl velocity component of the flow varies inversely with the radius. This is commonly known as free vortex. The outlet blade angles would therefore be calculated using the free vortex distribution.

5.8 MULTI-STAGE PERFORMANCE

An axial flow compressor consists of a number of stages. If R is the overall pressure ratio, R_{s} is the stage pressure ratio, and N is the number of stages, then the total pressure ratio is given by:

$$R=(R_{\mathrm{s}})^N \tag{5.29}$$

Equation (5.29) gives only a rough value of R because as the air passes through the compressor the temperature rises continuously. The equation used to find stage pressure is given by:

$$R_s = \left[1 + \frac{\eta_s \Delta T_{0s}}{T_{01}}\right]^{\frac{\gamma}{\gamma-1}} \tag{5.30}$$

The above equation indicates that the stage pressure ratio depends only on inlet stagnation temperature T_{01}, which goes on increasing in the successive stages. To find the value of R, the concept of polytropic or small stage efficiency is very useful. The polytropic or small stage efficiency of a compressor is given by:

$$\eta_{\infty,c} = \left(\frac{\gamma - 1}{\gamma}\right)\left(\frac{n}{n-1}\right)$$

or:

$$\left(\frac{n}{n-1}\right) = \eta_s\left(\frac{\gamma}{\gamma - 1}\right)$$

where $\eta_s = \eta_{\infty,c}$ = small stage efficiency.

The overall pressure ratio is given by:

$$R = \left[1 + \frac{N\Delta T_{0s}}{T_{01}}\right]^{\frac{n}{n-1}} \tag{5.31}$$

Although Eq. (5.31) is used to find the overall pressure ratio of a compressor, in actual practice the step-by-step method is used.

5.9 AXIAL FLOW COMPRESSOR CHARACTERISTICS

The forms of characteristic curves of axial flow compressors are shown in Fig. 5.12. These curves are quite similar to the centrifugal compressor. However, axial flow compressors cover a narrower range of mass flow than the centrifugal compressors, and the surge line is also steeper than that of a centrifugal compressor. Surging and choking limit the curves at the two ends. However, the surge points in the axial flow compressors are reached before the curves reach a maximum value. In practice, the design points is very close to the surge line. Therefore, the operating range of axial flow compressors is quite narrow.