Chapter 1

INTRODUCTION

Compressors are commonly classified as either positive displacement or dynamic compressors. The positive displacement compressor achieves its pressure rise by trapping fluid in a confined space and transporting it to the region of higher pressure. The dynamic compressor develops its increase in pressure by a dynamic transfer of energy to a continuously flowing fluid stream. There are two basic types of dynamic compressors: axial-flow compressors and centrifugal (radial-flow) compressors. The flow streamlines through rotating rows in an axial-flow compressor have a radius that is almost constant, whereas they undergo a substantial increase in radius in a centrifugal compressor. For this reason, the centrifugal compressor can achieve a much greater pressure ratio per stage than the axial-flow compressor. But the axial-flow compressor can achieve a significantly greater mass flow rate per unit frontal area. Figure 1-1 compares normalized discharge pressure, P, versus flow rate, Q, for these two compressor types to illustrate the differences in their performance characteristics. The axial-flow compressor approximates a variable pressure ratio—constant flow machine, whereas the centrifugal compressor is closer to a constant pressure ratio—variable flow machine. The performance data displayed in Fig. 1-1 are for a single-stage centrifugal compressor and a five-stage axial-flow compressor, both of which have about the same design pressure ratio. This demonstrates the superior pressure ratio-per-stage capability of the centrifugal compressor. Traditionally, the centrifugal compressor has been the more rugged and lower-cost type, while the axial-flow compressor has offered better efficiency. Those differences have become much less significant in recent years due to advances in technology, particularly with regard to efficiency. Presently, the compressor type selected is more likely to be based on the performance characteristics, size and cost that is best suited to the application.

NOMENCLATURE

a = sound speed
C = absolute velocity
c_p = specific heat at constant pressure
c_v = specific heat at constant volume
H = total enthalpy and compressor head

h = static enthalpy
k = ratio of specific heats = c_p / c_v
M = Mach number = C / a
$\dot{m}$ = mass flow rate
N = rotation speed (rpm)
P = pressure
Q = volume flow rate = $\dot{m} / \rho_t$
R = Gas constant and stage reaction
Re = Reynolds number
r = radius
T = temperature
U = local blade speed, ωr
W = relative velocity
β = flow angle
γ = stagger angle
δ = P_{t0} / P_{ref}
η = efficiency
θ = polar (tangential) coordinate and T_{t0} / T_{ref}
μ = viscosity
ρ = density
ϕ = stage flow coefficient
ψ = stage work coefficient
ω = rotation speed (radians/second)

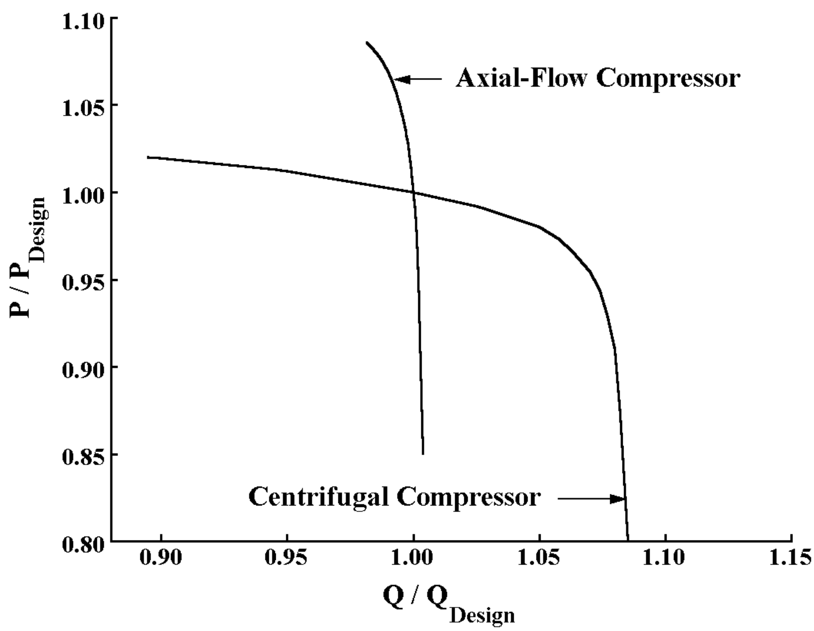

FIGURE 1-1 Performance of Dynamic Compressors

Subscripts

d = parameter at compressor discharge
ref = reference thermodynamic conditions
rev = reversible thermodynamic process
t = total thermodynamic condition
z = axial component
θ = swirl (tangential) component
0 = parameter at guide vane inlet or compressor inlet
1 = parameter at guide vane exit and rotor inlet
2 = parameter at rotor exit and stator inlet
3 = parameter at stator exit

Superscripts

$'$ = relative condition

1.1 AXIAL-FLOW COMPRESSOR BASICS

Figure 1-2 illustrates the basic configuration of an axial-flow compressor. The first blade row shown is an inlet guide vane (IGV) to develop the swirl (tangential) velocity for which the first rotating row (R1) was designed. If the first rotating row is designed for no inlet swirl, the inlet guide vane will normally be omitted. This is followed by a series of stages (two in this illustration), where a stage refers to a rotating row, or rotor, in combination with its downstream stationary row, or stator (e.g., R1 and S1). The rotor row imparts energy to the fluid by increasing the swirl velocity. The stator row removes the swirl developed by the rotor to convert kinetic energy to static pressure and to establish the proper swirl velocity for the flow to enter the next rotor. Typically, an exit guide vane (EGV) follows the last stage to remove any residual swirl velocity to convert that kinetic energy to static pressure. Like the inlet guide vane, this row may be omitted if the last stator is designed to remove all of the swirl velocity. Although not shown on the figure, a diffuser-combustor (gas turbine) or diffuser-collector (industrial compressor) will follow the exit guide vane to recover as much kinetic energy as possible, as well as to direct the flow to its intended destination. Similarly, an inlet passage will precede the inlet guide vane. This can range from a smooth axial bell-mouth inlet to a complex side inlet, depending on the compressor's application.

Figure 1-3 illustrates the blade profiles for a stage viewed on a polar stream surface between adjacent blades. The rotor row is rotating with a velocity, $U = \omega r$, where ω is the rotation speed and r is the radius. Viewed in a frame of reference rotating with the rotor, the upstream velocity, W, is referred to as the relative velocity. The rotor deflects the flow such that the velocity in the stationary frame of reference of the stator (the absolute velocity), C, is properly aligned to enter the stator row. This process repeats in subsequent stages, with each stage adding energy to the fluid to achieve the overall pressure ratio required.

Axial-flow compressor design strategies are quite varied. Gas turbine compressors are normally intended for use in many identical units. Extensive design

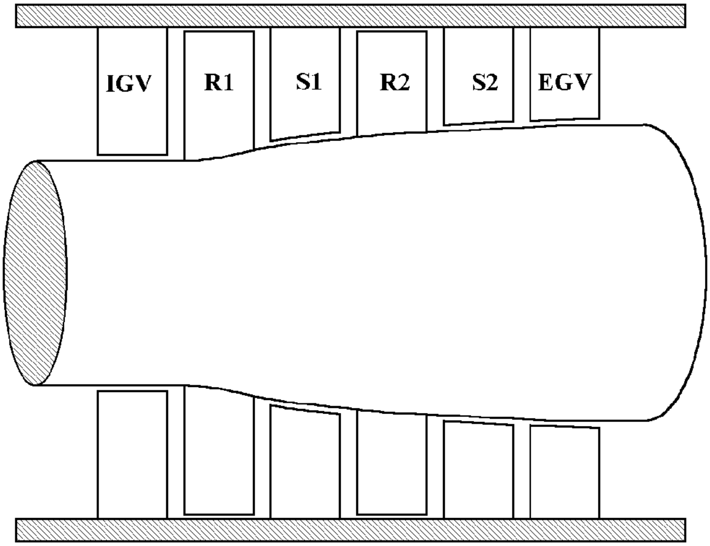

FIGURE 1-2 Axial-Flow Compressor Configuration

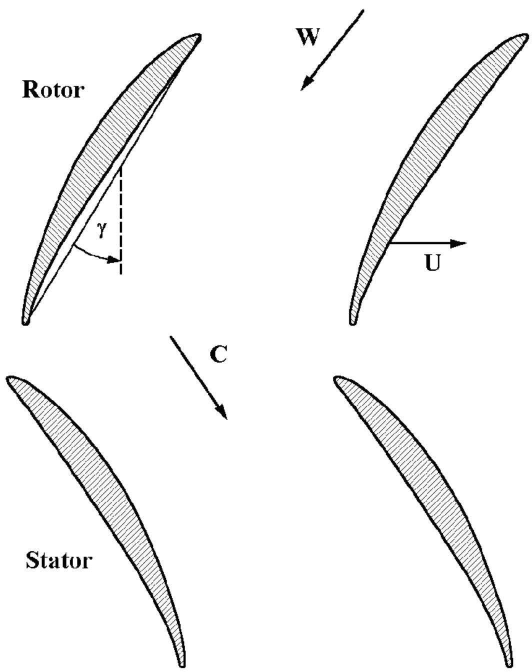

FIGURE 1-3 Polar Surface View of a Stage

and development programs are common and may include unique blade designs for all blade rows. By contrast, industrial compressors are usually designed specifically for a unique application, rarely involving any duplicate units. Here, a repeating stage approach is more common, where one or more basic stage designs are used for all compressors. Aerodynamic and mechanical flexibility are obtained by minor adjustments that do not compromise the basic stage's performance. Blades may be scaled to longer and thicker blades for mechanical integrity with a corresponding adjustment of the number of blades per row to preserve aerodynamic similarity. Blades usually must be restaggered, i.e., rotated on their base to change the stagger angle, γ, to achieve different performance levels. Otherwise, the intended duty would normally require a non-integer number of stages. Often the inlet guide vane and some of the stator blades may be adjustable so they can be restaggered by a control system while the machine is in operation to broaden the compressor's application range. This approach allows these "one-of-a-kind" compressors to be designed within practical cost. It also allows each compressor's design to be based on a well-established performance history. This is important, since these compressors cannot be confirmed by performance testing until after they are manufactured.

Figure 1-2 illustrates normal cantilevered blades that are attached at the root, with a clearance between the blade tip and the adjacent end-wall. Figure 1-4 shows a different style often used for stator blades. Here, a shroud band is attached to the blade tips to connect them together. This is often done for reasons of mechanical integrity. To reduce fluid leakage from the blade discharge back to the blade inlet, seal fins are normally attached to the shroud band. These provide a reduced clearance to retard leakage, yet are thin enough to minimize damage in the event that a rotor shaft excursion or "rub" causes the seals and shaft to come into contact. To minimize damage to the shaft, the stator blades and stator shrouds, the seal fins will be sacrificed in the event of a rub.

1.2 BASIC VELOCITY DIAGRAMS FOR A STAGE

The construction of velocity diagrams is a very useful concept for axial-flow compressor design. Here they will be used to illustrate the velocity vectors entering and leaving blade rows in a stage. It will be necessary to use both absolute and relative velocities, where relative velocities are viewed in a frame of reference rotating with the compressor's rotation speed, ω. Designating the relative and absolute tangential velocities as W_θ and C_θ, respectively, the two are related by

$$W_\theta = C_\theta - \omega r \tag{1-1}$$

where r is the local radius. The axial components of velocity are identical in both frames of reference, i.e.,

$$W_z = C_z \tag{1-2}$$

Therefore, the absolute and relative velocities are

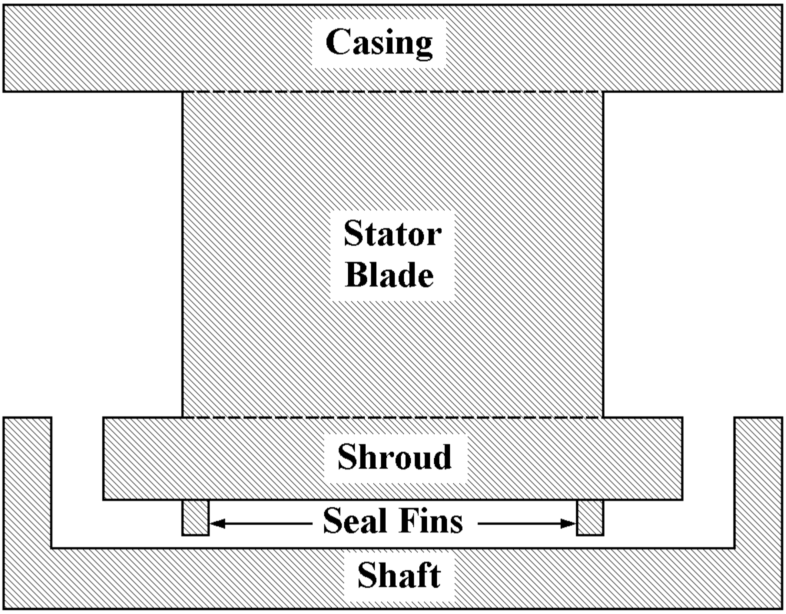

FIGURE 1-4 Shrouded Stator Blades

$$C = \sqrt{C_z^2 + C_\theta^2} \tag{1-3}$$

$$W = \sqrt{C_z^2 + W_\theta^2} \tag{1-4}$$

The absolute and relative flow angles are designated as β and β', respectively. They are defined by

$$\tan\beta = C_\theta / C_z \tag{1-5}$$

$$\tan\beta' = W_\theta / C_z \tag{1-6}$$

Figure 1-5 illustrates the velocity diagrams for an inlet guide vane. The flow enters the guide vane with no swirl, i.e., $C_{\theta 0} = 0$, $C_0 = C_{z0}$. The guide vane deflects the flow by an angle, β_1. If C_{z1} is known, this defines the swirl velocity component, $C_{\theta 1}$. Then Eq. (1-1) is applied in vector form to subtract ωr from $C_{\theta 1}$ to define the swirl velocity component in the relative frame, $W_{\theta 1}$ and the relative flow angle, β'_1. Hence, the complete velocity diagram for the entrance to the downstream (rotating) rotor blade row is known. Figure 1-6 shows the velocity diagram construction for the rotor blade row. The inlet velocity diagram is the same as that determined for the guide vane exit. The rotor blade deflects the flow in the relative frame of reference from β'_1 to β'_2 to produce the discharge swirl velocity, $W_{\theta 2}$. If C_{z2} is known, $W_{\theta 2}$ can be computed. Then vector addition of ωr to

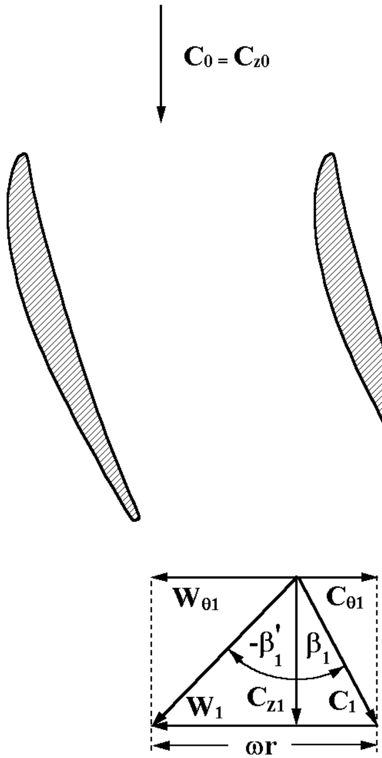

FIGURE 1-5 Guide Vane Velocity Diagrams

$W_{\theta 2}$ yields the absolute swirl component $C_{\theta 2}$. Hence, all velocity components and the flow angle in the absolute frame of reference can be computed to define the inlet conditions for the downstream (stationary) stator blade row. Construction of the stator and exit guide vane blade row velocity diagrams is accomplished in a similar fashion and will be left as an exercise for the reader. The important thing to note is that construction of this simple velocity diagram is a fundamental technique commonly used by turbomachinery aerodynamicists to convert between absolute and relative flow conditions. Here, C_z has been treated as known. In practice, values of C_z may be specified design conditions from which the flow passage areas will be computed to conserve mass. This will be referred to as the design mode. Alternatively, C_z may be computed from basic mass and momentum conservation for specified passage areas and the mass flow rate. This will be referred to as the analysis mode.

1.3 SIMILITUDE AND PERFORMANCE CHARACTERISTICS

Similitude or similarity is one of the most useful concepts in turbomachinery aerodynamics. Two turbomachines are completely similar if the ratios of all

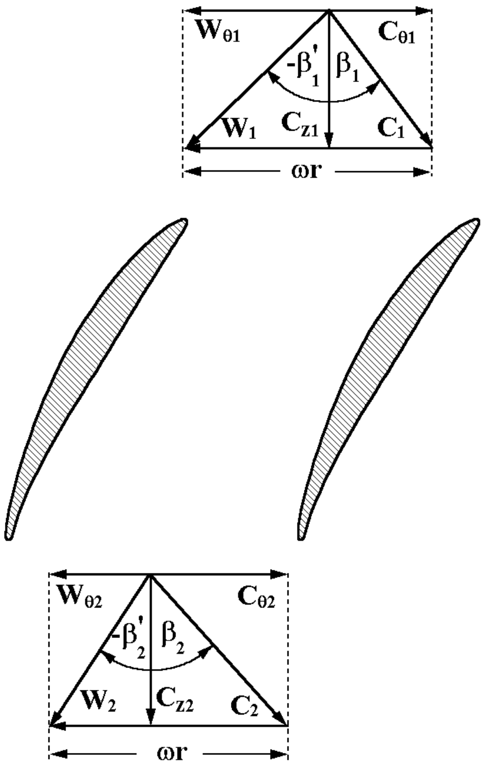

FIGURE 1-6 Rotor Blade Velocity Diagrams

corresponding length dimensions, velocity components and forces are equal (Sheppard, 1956). If two turbomachines are completely similar, it is possible to present their performance on a common performance map by selecting appropriate equivalent performance parameters. Equivalent performance requires that the two compressors have similar velocity diagrams throughout. To maintain similar velocity diagrams while conserving the mass flow rate, $\dot{m}$, the ratio of gas density-to-inlet density must also be similar throughout. This means that inlet volume flow, Q_0, is the relevant equivalent flow rate parameter, where

$$Q = \dot{m} / \rho_t \tag{1-7}$$

The local axial velocity is actually given by

$$C_{z0} = \dot{m} / (A_0 \rho_0) \tag{1-8}$$

where A is the passage area and ρ_0 is the inlet gas density, which is unknown. But the exercises in Chapter 2 will show that the ratio ρ_0 / ρ_{t0} is a function of C_{z0}, where ρ_{t0} is the known inlet total gas density. Hence, unique velocity diagrams are associated with a unique Q_0 / A_0, but can correspond to many values of $\dot{m}/A_0$ simply by altering ρ_{t0}. For this reason, all dynamic compressors are commonly referred to as volume flow machines. The exercises in Chapter 2 will show that the requirement for density ratio similarity requires that the Mach numbers be similar throughout, where the Mach number is the ratio of fluid velocity to the local sound speed, a. It will be shown that the ratio of a_0 / a_{t0} is, itself, a function of the Mach number, so the unknown a_0 can be replaced by the known inlet total sound speed, a_{t0}. Figure 1-7 shows an equivalent performance map based on these requirements. The flow parameter used is volume flow normalized by the inlet total sound speed, a_{t0}, and the inlet area, A_0. This ensures that the inlet axial Mach numbers will be similar. Three performance characteristics, or speed lines, are shown for three different rotation speeds: N, multiplied by a characteristic diameter, D, and normalized by the inlet total sound speed. This will ensure similarity of the tangential Mach numbers. If two axial-flow compressors are geometrically similar, and use the same working fluid, this performance map will apply to both machines. This, in turn, ensures that the pressure ratios will be the same for both, so pressure ratio is a reasonable choice for the other performance parameter. The situation becomes more complicated if the two compressors use different working fluids. In that case, complete similarity usually cannot be achieved, since different working fluids may produce different gas density (or specific volume) ratios for the same blade row velocity diagram. This "volume-ratio" effect will compromise similarity after the first rotor row, since

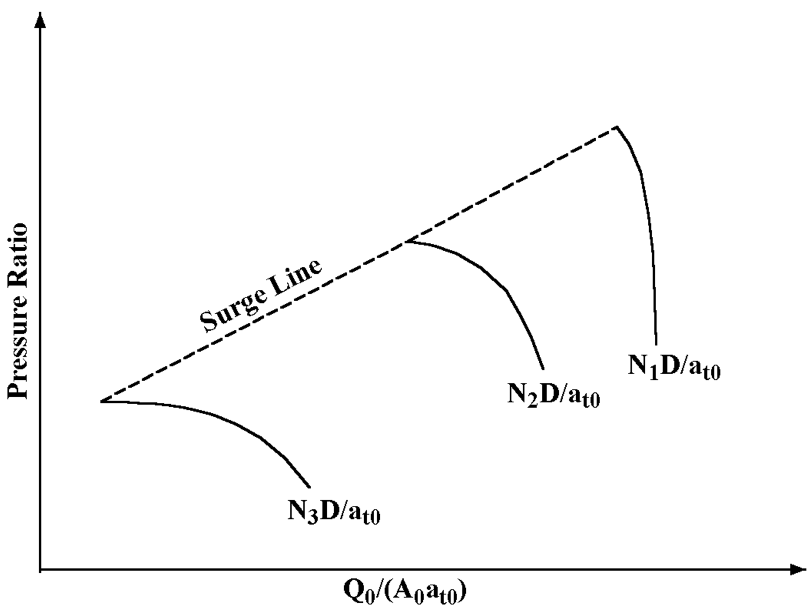

Figure 1-7 An Equivalent Performance Map

the differences in density will necessitate differences in axial velocity to conserve mass. Indeed, even with identical working fluids, non-ideal gas behavior may compromise similarity. True equivalent performance is assured only for working fluids that obey the perfect gas equation of state discussed in Chapter 2, which can be expressed as:

$$P = \rho RT \tag{1-9}$$

where P is pressure, T is temperature and R is the gas constant. An exercise in Chapter 2 will also show that achieving equivalent density ratios requires that the two working fluids have the same ratio of specific heats, $k = c_p / c_v$. Since axial-flow compressors are widely used in aircraft engine gas turbines, discussions on this topic often overlook these more subtle effects. That application of compressors deals exclusively with air as the working fluid, which is very nearly a perfect gas, offering little chance of any volume-ratio effect. Consequently, equivalent performance maps must be used with caution when a perfect gas model with a constant value of k cannot approximate the working fluid.

Similarity is also compromised when the two compressors operate at substantially different Reynolds number, Re. Reynolds number is a measure of the inertia forces to viscous forces, $Re = \rho CL/\mu$, where L is a characteristic length and μ is the gas viscosity. The Reynolds number directly affects wall friction, which can alter the compressor's performance. In most cases, effects of the Reynolds number are small enough to be neglected. When that is not the case, suitable Reynolds number corrections (e.g., Wassell, 1968) may be applied to adjust the performance.

A common use of equivalent performance maps is to define performance of a specific compressor that is operated at various inlet total thermodynamic conditions. For that case, D and A_0 are constant and can be omitted on the map. Another common use is to relate performance of different compressor frame sizes, derived by directly scaling the geometry. Then, D and A_0 are included so the map defines the change in speed, N, needed to preserve Mach number equivalence and the flow rate supplied for each pressure ratio on a speed line.

The equivalent flow rate used in Fig. 1-7 is the true similarity parameter. When the application is to a perfect gas with constant R and k, Eq. (1-9) can be used to derive alternate equivalent flow parameters, i.e.,

$$Q / a_{t0} \propto \dot{m}\sqrt{T_{t0}} / P_{t0} \propto \dot{m}\sqrt{\theta} / \delta \tag{1-10}$$

where the sound speed has been replaced by a perfect gas relation from chapter 2,

$$a = \sqrt{kRT} \tag{1-11}$$

and θ and δ relate inlet total conditions to reference conditions (T_{ref}, P_{ref}), such as standard atmosphere conditions, i.e.,

$$\theta = T_{t0} / T_{ref} \tag{1-12}$$

$$\delta = P_{t0} / P_{ref} \tag{1-13}$$

Similarly, the equivalent speed parameter can be replaced, using

$$N / a_{t0} \propto N / \sqrt{T_{t0}} \propto N / \sqrt{\theta} \tag{1-14}$$

The alternate flow rate and speed parameters in Eqs. (1-10) and (1-14) are commonly used, but have less fundamental significance than Q_0 / a_{t0} and N / a_{t0}.

Similarly, compressor head, ΔH_{rev}, can be used in place of pressure ratio, where head is defined as the total enthalpy increase required to produce the actual pressure rise by an ideal, reversible process, i.e.,

$$\Delta H_{rev} = \int_{rev} \frac{dp}{\rho} \tag{1-15}$$

It can be shown that the appropriate equivalent head parameter is $\Delta H_{rev} / (a_{t0})^2$. The use of an equivalent head is common practice for centrifugal compressors, but is much less common for axial-flow compressors.

Figure 1-7 supplies only part of the performance information required. In addition to the pressure ratio and flow produced, it is necessary to know the work required to drive the compressor. Hence, a second equivalent performance map is required to completely define the compressor's performance. The most common parameter for this purpose is efficiency, η, defined as the compressor head or ideal (no loss) total enthalpy rise divided by the actual total enthalpy rise, i.e.,

$$\eta = \Delta H_{rev} / \Delta H \tag{1-16}$$

Alternate reversible processes that can be used to define ΔH_{rev} and η are discussed in Chapter 2. Figure 1-8 shows an equivalent efficiency map to be used in conjunction with Fig. 1-7. In some cases, it may be appropriate to use the exit static thermodynamic conditions rather than total values to define η and P_R. This is appropriate when the kinetic energy available at the compressor discharge serves no useful purpose for the specific application to which the compressor will be applied.

1.4 STAGE MATCHING AND STABILITY

Each blade row in a compressor will achieve its best performance for a specific inlet flow angle where losses are minimum. Basically, the designer seeks to "match" succeeding blade rows such that all operate close to their optimum inlet flow angles at a specific operating condition, commonly called the compressor's design point or match point, defined by the design flow rate and design speed. Hence, at design speed, losses can be expected to increase and performance to deteriorate as the compressor operates farther from its design flow rate. At flow rates less than the design flow rate, losses will increase to a point that the pressure-flow rate characteristic reaches a maximum. At lower flow rates, the characteristic will have a positive slope, which is theoretically unstable. The onset of

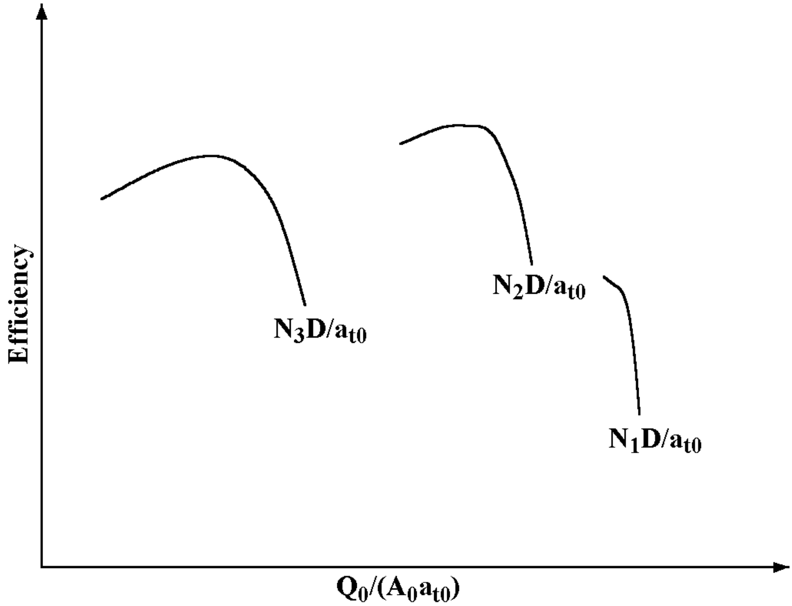

FIGURE 1-8 An Equivalent Efficiency Map

this severe unstable operation is commonly called surge. The limit of stable operation is referred to as the surge line as illustrated in Fig. 1-7. Surge is a very complex phenomenon, which depends on the entire system, not just the compressor. So associating it with a maximum on the pressure–flow rate characteristic is an oversimplification, but a useful one. In some cases, an approach to zero-slope near surge is evident, such as for speed line N_3 in Fig. 1-7. In other cases, the compressor may experience an abrupt stall, such that the characteristic appears to be quite steep at surge, similar to speed line N_1 in Fig. 1-7. This is mainly because the drop in pressure with reduced flow is so abrupt that it cannot be resolved in a performance test. Indeed, estimation of the onset of surge during the design phase is based more on the expected blade loading limits at the onset of stall than on the predicted shape of the pressure-flow characteristics. Similarly, at flow rates greater than the design flow rate, the increase in loss will eventually result in no rise in pressure. This condition is commonly referred to as choke, although it may be caused by large losses due to off-design operation rather than a true aerodynamic choke condition.

When the compressor is operated at off-design speeds, operation at different Mach number levels will compromise the stage matching, similar to the volume-ratio effect mentioned previously. Consequently, it is unlikely that all stages will be close to their optimum operating conditions at any flow rate for off-design speeds. Rather, optimum performance will occur at the flow rate offering the best compromise on stage matching. Performance will deteriorate for flows different from this optimum, much as described for the design speed performance.

The more speed deviates from the design speed, the greater the compromise of the stage matching. In general, at speeds lower than the design speed, the front stages are required to supply a greater portion of the rise in pressure while the rear stages become less effective. The inverse is true for speeds greater than the design speed. This stage mismatching can be alleviated to some degree if some of the stationary blade rows are adjustable during operation. Closing some of the stationary rows (i.e., increasing their stagger angles) in a controlled fashion will shift the optimum matching condition to lower speeds to reduce the mismatch at low speeds.

The Mach number level has a definite influence above and beyond its pronounced effect on stage matching. As a blade row's inlet Mach number increases, its low-loss operating range will decrease. At sufficiently high values, the blade row will start to experience aerodynamic choke in the blade row to significantly reduce its maximum flow capacity. Even the minimum loss levels will increase when the inlet Mach number becomes high enough to produce shock waves that are strong enough to induce boundary layer separation or to produce significant bow shock losses. Consequently, as the equivalent speed increases, pressure-flow characteristics become steeper, with less flow range from surge to choke as illustrated in Fig. 1-7, and the maximum achievable efficiency can be limited by Mach number levels, similar to speed line N_1 in Fig. 1-8.

1.5 DIMENSIONLESS PARAMETERS

In addition to the dimensionless parameters associated with similitude, axial-flow compressor design is often based on a number of other useful aerodynamic dimensionless parameters. To introduce these parameters, it is necessary to anticipate some results that are developed in more detail in Chapters 2 and 3. In particular, the total enthalpy rise for simple axial flow through a rotor blade row is expressed by the well-known Euler turbine equation, i.e.,

$$\Delta H = U(C_{\theta 2} - C_{\theta 1}) \tag{1-17}$$

where $U = \omega r$ is the local blade speed and H is the total enthalpy. Similarly, it will be shown in Chapter 2 that the static enthalpy, h, is related to H by

$$H = h + \tfrac{1}{2}C^2 \tag{1-18}$$

It is often useful to introduce dimensionless stage performance parameters expressed for a "repeating" stage, i.e., a stage designed to be followed by another identical stage. This means that the velocity diagrams for the rotor inlet (station 1) and the stator exit (station 3) must be identical. Then, the stage work coefficient, ψ, can be defined as

$$\psi = \Delta H / U^2 = (C_{\theta 2} - C_{\theta 1}) / U \tag{1-19}$$

where all data correspond to a constant, mean radius, or "pitch line" for the stage. The stage flow coefficient, ϕ, is defined by

$$\phi = C_{z1} / U \tag{1-20}$$

The stage reaction, R, is defined as the fraction of the stage static enthalpy rise that occurs in the rotor, i.e.,

$$R = (h_2 - h_1) / (h_3 - h_1) \tag{1-21}$$

Substituting P for h in Eq. (1-21) yields an alternate definition of reaction, in terms of static pressures, that is sometimes used. Requiring C_z to be constant through the stage, Eqs. (1-17), (1-18) and (1-21) can be combined to yield

$$R = 1 - (C_{\theta 2} + C_{\theta 1}) / (2U) \tag{1-22}$$

In Chapter 10, it will be seen that parameters ϕ, ψ and R provide useful guidance for stage design. Stage design involves defining blade geometry that will produce the desired performance. These dimensionless performance parameters define performance in a form general to any stage design problem. They are normally used to specify the performance objectives the stage should achieve at its mean radius or pitch line. While there are no fixed rules for selecting values for them, preferred values can normally be established based on the design goals most important to the designer, supported to some degree by simple logic. For example, 50% reaction stages (R = 0.5) are quite common, prompted mainly by the intuitive judgment that it is best to share the flow diffusion load equally between the rotor and the stator. Once specified, these parameters can be used to define the stage velocity diagrams from which the blades can be designed. For example, Eqs. (1-19) through (1-22) can be combined to yield

$$\tan\beta_1' = -(\psi / 2 + R) / \phi \tag{1-23}$$

$$\tan\beta_2' = (\psi / 2 - R)/\phi \tag{1-24}$$

$$\tan\beta_1 = (1 - R - \psi / 2)/\phi \tag{1-25}$$

$$\tan\beta_2 = (1 - R + \psi / 2)/\phi \tag{1-26}$$

and the velocity diagrams for the stator exit and rotor inlet are identical for a repeating stage. These parameters have defined the velocity diagrams at the pitch line only. It is necessary to supply additional design specifications and use fundamental fluid dynamics relations to generate the velocity diagrams at other radial locations.

1.6 UNITS AND CONVENTIONS

This book assumes consistent units throughout, such that the reader may use any set of consistent units preferred. For historical reasons, many turbomachinery organizations do not use consistent units, often using different units for different disciplines such as aerodynamics and thermodynamics. For example, it is not uncommon to find energy terms, fluid velocity and equation-of-state parameters expressed in inconsistent units, necessitating conversion factors in expressions such as Eq. (1-18). It will be left to the reader to recognize the need for those conversion factors. Flow angles and blade angles will be measured from the meridional plane, i.e., a plane of constant polar angle, θ, in a cylindrical coordinate system. These angles, and the associated swirl velocity components C_θ and W_θ, are considered to be positive in the direction of rotation. While the nomenclature is reasonably consistent throughout, the wide range of topics covered does not permit unique symbols for every parameter. Consequently, each chapter will include its own list of nomenclature to avoid confusion.

EXERCISES

1.1 An axial-flow compressor is to be operated with a different working fluid, which can be modeled as a perfect gas, but has values of gas constant, R, and ratio of specific heats, k, that are different from the normal working fluid. Develop new equivalent speed and flow rate parameters, in terms of $\dot{m}$, T_{t0} and P_{t0} that will ensure Mach number equivalence at the compressor inlet.

1.2 Free vortex stages with $\beta_1 = \beta_3 = 0$ are often used for axial-flow compressors. Derive an expression for work coefficient as a function of reaction for this type of stage. If the stage is also to have 50% reaction, specify the range of values for ψ and β_1' that can be used. If the resulting stage is to be used as a repeating stage in a multistage compressor, what type of inlet and exit guide vanes will be needed?

1.3 All dimensions of the compressor producing the performance map shown in Fig. 1-7 are scaled by a factor of 1.2 and both compressors are operated with the same inlet conditions and working fluid. If operating points for the original compressor are denoted as N_A and Q_A, develop expressions for equivalent operating conditions N_B and Q_B for the scaled compressor. If the original compressor operates at a speed of 3,600 rpm, what speed must be used for the scaled compressor? How much additional flow capacity will the scaled compressor have?

1.4 What scale factor should be applied to the compressor producing the performance map shown in Fig. 1-7 to increase the compressor's flow capacity by 20% for the same inlet conditions and working fluid? What adjustment in speed will be needed?