and the machine in the manufacturer's range with the nearest V_i. The choice is then made according to the extra cost of the sliding drawer and its control.

7.2.3 Criteria for the choice between displacement compressors

Since the communication with the compression chamber is done through the automatic valve whose opening is triggered by the pressure difference on their faces, piston compressors can work at variable pressures. They can therefore operate through self-adaptation according to inlet and outlet conditions on a wide compression range.

Screw compressors however generally perform better than reciprocating compressors, if they are provided with a sliding drawer.

However, each of them corresponds to rather different capacity ranges.

In general, reciprocating compressors are suitable for low capacity (less than 75 kW mechanical drive), while screw compressors can match compression power of the order of 1 MW or greater.

Screw compressors are more expensive because they are more difficult to manufacture and the cost also increases with the sliding drawer and its control.

Reciprocating compressors are limited in pressure ratio, as we have seen, and do not tolerate the presence of liquid at the suction. On these two points, screw compressors have advantages.

7.3 DYNAMIC COMPRESSORS

7.3.1 General

Unlike positive displacement machines where the fluid is enclosed in a closed volume, in a dynamic compressor, a continuous flow of fluid takes place to which energy is communicated through moving blades driven by a rotor.

In this chapter we limit our ambition to establish the main results that are needed to understand the operation of dynamic compressors, and, given that the operation of the expansion turbines is quite similar to that of dynamic compressors, we will directly put up a number of results valid for all turbomachinery.

There are two main types of fluid flow relative to the rotor of a turbomachine: axial flow, almost always performed in turbojet dynamic compressors, and radial flow which is widely used for centrifugal dynamic compressors, including for refrigeration or automobile engine supercharging.

We will always consider the technical fluid to be compressible, either an ideal gas or a condensable real fluid. Consequently, the phenomena are governed by the fundamental laws of steady compressible flows which we begin by establishing in section 7.3.2. We will show that by varying the velocity of a gas, it is possible to change its pressure.

In general, a turbomachine consists of four elements in series (Figure 7.3.1).

- an inlet convergent C, or distributor, which is a fixed part whose function is to properly orient the fluid veins entering the impeller, and accelerate them slightly;
- the impeller I, or rotor, driven by a rotational movement around an axis. This wheel has vanes defining channels between which the fluid flow is distributed. It communicates to the fluid the mechanical energy of the blades in the form of kinetic, heat and pressure energy;
- the diffuser D is a fixed component that serves to transform in pressure part of the kinetic energy gained by the fluid when passing through the wheel. Depending on circumstances, the diffuser may or not involve blades. It is said to be partitioned or smooth;
- a volute V, also fixed, corrects the fluid lines on the periphery of the wheel, and guides them to the device outlet.

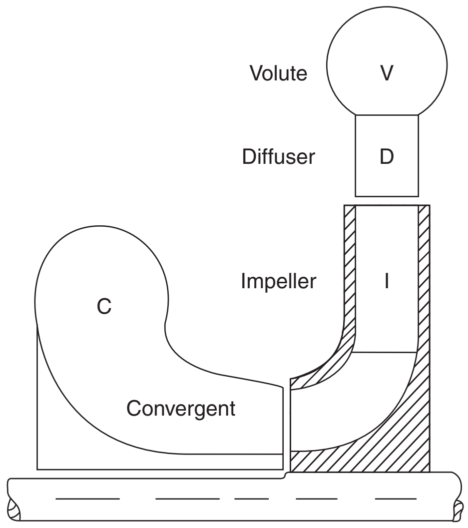

FIGURE 7.3.1
Sketch of a centrifugal compressor

For the reasons discussed in section 7.1.4, turbomachinery frequently has several stages, each having a limited pressure head. This is called multistage turbomachinery.

In axial turbomachinery, successive stages are juxtaposed by compact groups in which the volute is no longer necessary, and where the convergent and the diffuser situated between two moving wheels can be combined to form a single crown whose role is to straighten the fluid lines so they appear correctly on the input of the second impeller. Under such conditions, the middle stages have only two components.

It also happens that, for reasons of simplicity, a turbomachine even single staged may be composed of but two components, one fixed and one mobile.

In a turbine, it is essential to have expansion and guide nozzles at the inlet of the impeller, while the diffuser plays a secondary role and may eventually disappear.

In a dynamic compressor, conversely, the wheel inlet guide plays a secondary role in the recovery of the outlet kinetic energy. We can possibly suppress the inlet distributor.

7.3.2 Thermodynamics of permanent flow

In the previous sections, we assumed the kinetic energy variations of fluids undergoing processes were negligible, which allowed us to eliminate dK in our calculations. In dynamic compressors, this assumption is no longer valid, the practical effect being obtained by converting into pressure the kinetic energy of the fluid.

7.3.2.1 Compressible perfect fluid in steady-state

We restrict ourselves initially to the case of steady flows where the absolute pressure and the three velocity components are assumed to be constant over time. The flow occurs in fixed pipes, which implies the absence of moving walls. These conditions are obtained in the convergent and the diffuser of a turbomachine.

We assume initially the fluid is perfect from the hydrodynamic point of view, that is to say without viscosity (note that an ideal gas is not necessarily a perfect fluid from this point of view).

We obtain the law of a compressible fluid flow applying the first law of thermodynamics to a stream tube, taking into account the forces of gravity in calculating the enthalpy.

According to (5.4.7), we have : $\delta\tau = vdP + dK + gdz + \delta\pi$

The walls being fixed, $\delta\tau = 0$, and the fluid being perfect, $\delta\pi = 0$.

We have thus: $dK + vdP + gdz = 0$

which, in integral form, reads:

$$\frac{C^2}{2} + \int vdP + gz = \text{Const.} \tag{7.3.1}$$

This relationship is a generalization of the Bernoulli's equation of incompressible flows, which corresponds to the particular case when v = Const.

$$\frac{C^2}{2} + \frac{P}{\rho} + gz = \text{Const.}$$

In the form (7.3.1), it shows that even in the case of a reversible process, the flow law is complex because it depends on many factors (the initial state of the fluid, the final pressure, the external heat exchange).

Relationship (7.3.1) has been established in the case of an absolute steady flow. It can be generalized in the case of a relative steady flow, where the pressure and velocity values are fixed at every point of a system driven by a uniform movement. These are the conditions encountered in the impeller of a turbomachine.

This requires us to express that the useful work is equal to 0 (fixed walls), plus the work of inertial forces corresponding to the driving acceleration. These are the following laws:

- system driven by a uniform translation motion (axial turbomachinery case): driving acceleration is zero, equation (7.3.1) remains valid;
- system driven by a uniform rotation (centrifugal turbomachinery case): the acceleration drive is radial, and the work of the corresponding centrifugal inertia force is $\Delta\ \omega^2R^2/2$. Posing $U = \omega R$, drive velocity, equation (7.3.1) becomes:

$$\frac{C^2 - U^2}{2} + \int vdP + gz = \text{Const.} \tag{7.3.2}$$

7.3.2.2 Viscous steady compressible fluid

Viscosity changes the above relations only slightly. The useful work remains zero, the walls being stationary, and the work of viscous forces is zero, if one applies the first law to a vein extending to the wall, because the velocity of a viscous fluid is always zero at the wall.

According to (5.4.7), we have: $\delta\tau = 0 = vdP + dK + gdz + \delta\pi$

We deduce:

$$\frac{C^2}{2} + \int vdP + gz + \int \delta\pi = \text{Const.} \tag{7.3.3}$$

The effect of friction is ultimately a decrease in the head. We thus find the concept of "pressure drop" π.

$$\pi = \int \delta\pi$$

In the present state of our knowledge, as we do not *a priori* know how to compute π, it is experimentally determined.

7.3.2.3 Adiabatic Flows

Basic law

In the foregoing, no assumption was made on the heat exchange of the flow with the outside, which may vary from the perfect isolation (adiabatic process), to the infinitely developed contact with a source (isothermal process).

The real changes are always very close to the adiabatic since the gases are very poor heat conductors (see 5.3.6). Therefore it is assumed that the processes are always adiabatic, the actual process being irreversible, while the theoretical reference is isentropic.

In what follows, we neglect the potential energy gz, which is perfectly legitimate, given the relatively low altitude variations of the fluid, compared to the variations of kinetic energy and enthalpy (see 5.2.1).

The fundamental law of adiabatic flow in a fixed reference follows directly from (7.3.3). It reflects the conservation of total enthalpy h + K, and is written along a stream tube:

$$h + \frac{C^2}{2} = \text{Const.} \qquad (7.3.4)$$

Assuming further that the flow is reversible adiabatic (ds = 0) then:

$$\begin{aligned} &dh = v\,dP, \quad \text{and} \\ &v\,dP + C\,dC = 0 \end{aligned} \qquad (7.3.5)$$

It is possible to change the fluid pressure acting on its velocity and vice versa.

In the case where the reference is no longer fixed, but driven by a relative permanent rotation movement of drive velocity $U = \omega R$, the relation is written:

$$\frac{C^2 - U^2}{2} + h = \text{Const.} \qquad (7.3.6)$$

or

$$v\,dP + C\,dC - U\,dU = 0 \qquad (7.3.7)$$

This relation shows that in a dynamic compressor rotor, compression can be achieved by reducing the fluid velocity C or increasing the driving velocity U. In an axial compressor, the diameter of the wheel is approximately constant, and the first effect is dominant, while a centrifugal compressor is able to combine both, which allows it to provide much higher compression ratios in one stage.

Stagnation properties

We have already introduced in the previous section the total enthalpy h + K, and in section 5.3.1 the total energy u + K. We call stagnation pressure the pressure P_i indicated by a Pitot tube placed in an perfect gas flow, and stagnation temperature the isentropic temperature T_i measured by a thermometer placed in a flow so that the process bringing the flow to rest in front of the thermometer would be isentropic.

Given (7.3.4),

$\Delta h = C^2/2$ and the gas being perfect, we have: $\Delta h = c_p \Delta T$

The total or stagnation temperature is given by:

$$T_i = T + \frac{C^2}{2c_p} \tag{7.3.8}$$

The isentropic equation (5.6.8) gives the stagnation or total pressure:

$$PT^{\gamma/(\gamma-1)} = \text{Const.} \quad \text{or} \quad P_i = P\left(\frac{T_i}{T}\right)^{\gamma/(\gamma-1)}$$

Introducing the flow Mach number:

$$Ma = \frac{C}{\sqrt{\gamma r T}} \quad \text{and noticing that: } c_p = \frac{\gamma r}{\gamma - 1}$$

we get:

$$P_i = P\left(1 + \frac{\gamma - 1}{2} Ma^2\right)^{\gamma/(\gamma-1)} \tag{7.3.9}$$

These two relationships are interpreted as follows: in any isentropic flow of a perfect gas in a tube with fixed walls, the stagnation temperature and pressure are conservative.

In addition, the value of the stagnation properties is that they allow us to write the equations governing the flow by formally removing the velocity C, which greatly simplifies the expression.

Varying the section of a vein

Let us study what should be the section variation of a fluid stream undergoing an adiabatic expansion or compression in a reversible manner. In what follows, we assume that the fluid is a perfect gas.

In order for the flow to be reversible and adiabatic, three equations must simultaneously verified:

- the continuity equation, which indicates that the flow is constant in any section of the flow;
- If S is the section, $\dot{m}$ being the fluid flow, we get $SC/v = \dot{m}$;
  It comes: $dS/S + dC/C = dv/v$
- the isentropic equation $Pv^{\gamma} = \text{Const.}$;

Combined, and introducing the Mach number Ma, the first two equations lead to:

$$\frac{dS}{S} = \frac{v\,dP}{C^2}(1 - M_a^2) \tag{7.3.10}$$

- the kinetic equation (7.3.5) $v\,dP + C\,dC = 0$.

These last two equations, for various sonic operation modes, allow one to simply analyze the changes in pressure and velocity in the convergent and divergent, as shown in Figure 7.3.2.

This shows that, for subsonic flow, compression corresponds to a deceleration of the fluid, and expansion corresponds to an acceleration, while the reverse is true for supersonic regimes.

We find a well-known result of fluid mechanics: a sonic nozzle must be comprised at the throat inlet by a convergent of decreasing section, and at the throat outlet by a diverging increasing section. We call such a converging-diverging nozzle configuration a Laval nozzle (Figure 7.9.1).

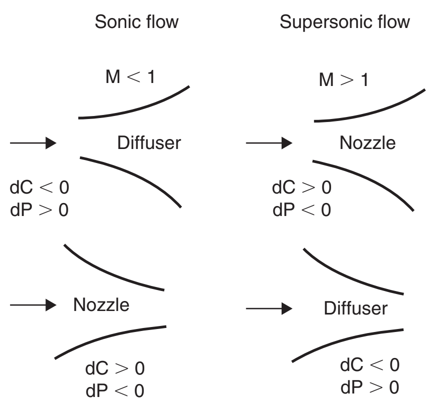

FIGURE 7.3.2
Adiabatic flow geometries

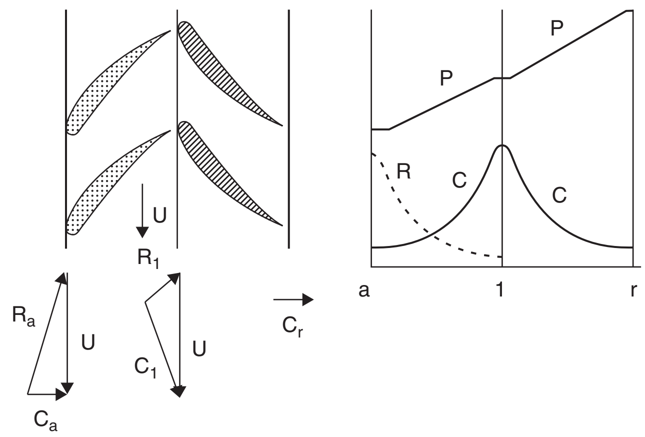

FIGURE 7.3.3
Velocity profile in an axial compressor

7.3.2.4 Changes in pressure and fluid velocity in a compressor

In a dynamic compressor, the evolution of the fluid is increased pressure, which, for a subsonic regime, requires that the section of the vein is increasing, while the velocity decreases. Evolution is a two time process (Figure 7.3.3): in the impeller, the relative velocity drops sharply, while the absolute velocity increases. The stator (diffuser) then slows the absolute velocity down.

To analyze in detail the operation of dynamic compressors, it would be necessary to make a detailed kinematic study of the evolution of fluid through the various organs.

Developments that would require such an analysis do not seem justified given our approach, oriented rather on systems than components, so we content ourselves at this stage to present the principle of these calculations, before considering the overall performance of compressors, based on dimensionless analysis. Further analysis will be provided in chapter 37 of Part 5. We refer interested readers to the books of L. Vivier (1965) and M. de Vlaminck & P. Wauters (1988) cited in the bibliography.

We denote with index a the fluid suction, r the discharge, 1 the rotor output, and R is the relative velocity.

In the impeller, the application of (7.3.6) gives:

$$2(h_1 - h_a) = (U_1^2 - U_a^2) + (R_a^2 - R_1^2)$$

In the diffuser:

$$2(h_r - h_1) = C_1^2 - C_a^2$$

Summing, we get:

$$2(h_r - h_a) = (U_1^2 - U_a^2) + (R_a^2 - R) + (C_1^2 - C_r^2)$$

Since C_r is somewhat equal to C_a, one can write:

$$2(h_r - h_a) = U_1^2 + C_1^2 - R_1^2 - (U_a^2 + C_a^2 - R_a^2)$$

However, as shown in the velocity triangle (Figure 7.3.3):

$$\overrightarrow{R} = \overrightarrow{C} - \overrightarrow{U} \quad \text{and} \quad R^2 = C^2 + U^2 - 2\,\overrightarrow{U} \cdot \overrightarrow{C}$$

$$\tau = h_r - h_a = \vec{U}_1 \cdot \vec{C}_1 - \vec{U}_a \cdot \vec{C}_a \qquad (7.3.11)$$

This relationship is Euler's theorem, which connects the enthalpy change and thus the useful work to the velocity profile in the compressor.

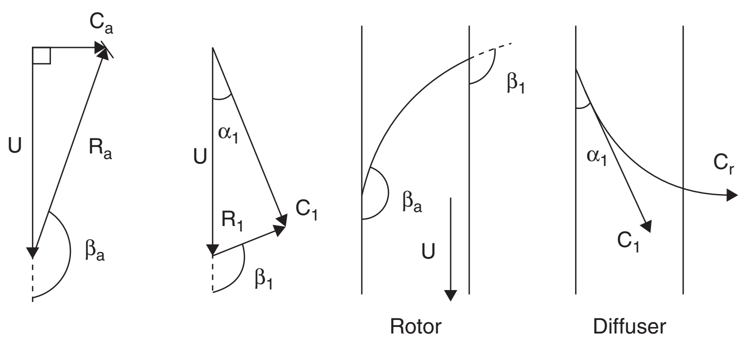

FIGURE 7.3.4
Velocity triangle

In an axial compressor, the drive velocity being substantially the same at the inlet and outlet of the wheel, this relationship is simplified, and yields:

$$\tau = h_r - h_a = \vec{U} \cdot (\vec{C}_1 - \vec{C}_a) \tag{7.3.12}$$

Generally one makes sure that $\overrightarrow{C_a}$ be perpendicular to $\overrightarrow{U}$ at the inlet of dynamic compressors, so that these relations become simpler yet:

$$\tau = \vec{U}_1 \cdot \vec{C}_1 \tag{7.3.13}$$

Velocity triangle and shape of the blades

Euler's relation is used to define the profile of the blades: in order for the fluid to flow in the rotor with the least possible irreversibilities, shocks must be avoided, whether from discontinuities in the fluid streams or from shock waves which can propagate if the velocity is too high. The thermodynamics of flows finds here one of its limits: the exact shape of the blade profile calls upon fluid mechanics and is thus outside the scope of this presentation. We will just give some indication of the results, only for axial compressors.

Given the Euler relationship, we seek to maximize the scalar product $\overrightarrow{U}_1 \cdot \overrightarrow{C}_1 = UC_1 \cos\alpha_1$. Therefore angle α_1 must be as small as possible.

Suppose the drive velocity U is known. To avoid shock in the blades, the relative velocity vector $\overrightarrow{R}$ must at all times be parallel to blades, which sets its angle β with the drive velocity $\overrightarrow{U}$.

Entry speed $\overrightarrow{C_a}$ being perpendicular to $\overrightarrow{U}$, the inlet velocity triangle immediately gives angle β_a and defines vector $\overrightarrow{R_a}$.

At the outlet we obtain similarly β_1 and $\overrightarrow{R_1}$, knowing that the absolute velocity $\overrightarrow{C_1}$ must generally remain subsonic to avoid shock in the diffuser vanes.

By continuity the blade profile can be plotted (Figure 7.3.4).

However as discussed later, technological issues lead manufacturers to choose the type of wheel and to set some reduced design parameters. We do not, therefore, generally know U as has been assumed here, but a number of relationships exist between the velocity triangle quantities, so that it is determined by reasoning similar to that which was just presented.

To determine the blade thickness, whose distance between them defines the cross-section available for the fluid, we must give, in addition to the Euler relation, the equation of continuity. Moreover, even more than for the profile of nozzles, it is necessary to reduce the curvature to prevent detachment of the fluid streams along the compressor blades, because the pressure gradient exerted here is unfavorable, both in the rotor and in the diffuser. This constraint is much stronger for dynamic compressors than for turbines.

7.3.3 Similarity and performance of turbomachines

When the dynamic compressor is not working at its design conditions, performance generally suffers in larger proportion than in the case of positive displacement compressors. The reason is that dynamic irreversibilities (shock, detachment of fluid streams, etc.) are rising quite rapidly once the flow no longer corresponds to the geometry of the blades.

We will now further analyze the characteristics of turbomachinery, and the adaptation of a machine at different operating regimes. These regimes depend not only on basic variables considered in the design (N, $\dot{V}$, Δh), but also on the characteristic parameters of the fluid passing through the machine.

In the most common conditions, 7 independent physical variables may affect the efficiency of a turbomachine: a characteristic dimension (e.g. wheel diameter D), the rotation speed N, the fluid mass flow, the stagnation pressures at the inlet and outlet P_a and P_r and the total enthalpy at the entrance h_a and exit h_r.

As mentioned in section 7.3.2.3, we choose the stagnation properties to eliminate from the equations the fluid velocity at the entrance and exit of the machine.

Among these 7 variables are involved three basic units (M, L, T). The application of the Vaschy-Buckingham theorem reduces to $7 - 3 = 4$ the number of dimensionless variables characteristic of the operation of the machine.

The variables most commonly chosen are:

- a flow velocity C_f Mach number:

$$(M_a)_c = \frac{C_f}{C_s}$$

with

$$C_f = \frac{\dot{m}v_a}{A} = \frac{4\dot{m}v_a}{\pi D^2} = \frac{4\dot{m}rT_a}{\pi D^2 P_a}$$

and C_s speed of sound in the fluid ($C_s = \sqrt{\gamma r T_a}$ by equating the fluid to a perfect gas), thus:

$$(M_a)_c = \frac{4\dot{m}\sqrt{rT_a}}{\pi D^2 P_a \sqrt{\gamma}} \qquad (7.3.14)$$

- a wheel Mach number

$$(M_a)_u = \frac{U}{C_s} \qquad \text{with } U = \frac{\pi DN}{60}$$

$$(M_a)_u = \frac{\pi D N}{60\sqrt{\gamma r T_a}} \qquad (7.3.15)$$

- the ratio of inlet and outlet stagnation pressures:

$$\frac{P_r}{P_a}$$

- the isentropic efficiency of the stage:

$$\eta_s = \frac{(h_r)_s - h_a}{h_r - h_a} \quad \text{or} \quad \eta_s = \frac{h_r - h_a}{(h_r)_s - h_a}$$

depending on whether a compressor or a turbine.

When choosing a given machine and a particular fluid, the dimensionless numbers $(M_a)_c$ and $(M_a)_u$ become proportional to reduced variables of simpler expressions:

$$(M_a)_c \div \frac{\dot{m}\sqrt{T_a}}{P_a} = (\text{corrected mass flow } \dot{m}_c) \tag{7.3.16}$$

$$(M_a)_u \div \frac{N}{\sqrt{T_a}} = (\text{corrected rotation speed } N_c) \tag{7.3.17}$$

Other dimensionless quantities are also commonly used by manufacturers: flow factor φ and enthalpy factor ψ, or Rateau coefficients μ and δ.

The flow factor φ

It is natural to look at both Mach numbers representative of flows in the machines, $(Ma)_c$ and $(Ma)_u$. The ratio of these two quantities, independent of fluid properties, determines the shape of the velocity triangle, and corresponds to a first dimensionless property: the flow factor φ which ensures kinematic similarity over the entire flow boundaries.

$$\varphi = \frac{(M_a)_c}{(M_a)_u} = \frac{4\dot{m}\sqrt{rTa}}{\pi\, D^2 P_a \sqrt{\gamma}} \frac{60\sqrt{\gamma r T_a}}{\pi DN}$$

$$\varphi = \frac{C_f}{U} = \frac{240\,\dot{V}}{\pi^2\, N\, D^3} = \frac{240}{\pi}\,\delta \tag{7.3.18}$$

This factor φ is proportional to another widely used dimensionless quantity: the second Rateau coefficient δ.

The enthalpy factor ψ

Turbomachinery is used either to expand a fluid to produce energy or to provide energy to a fluid. The flow being, as we noted earlier, close to adiabatic, because of low exchange surfaces and velocities, it is logical to take as a reference energy the expansion or compression isentropic work $|\Delta h_s|$.

We have:

$$|\Delta h_s| = \frac{\gamma}{\gamma - 1} P_a v_a \left| \left(\left(\frac{P_r}{P_a} \right)^{(\gamma-1)/\gamma} - 1 \right) \right| \tag{7.3.19}$$

The ratio of the reference energy to a well chosen kinetic energy is a dimensionless quantity of interest. A first idea is to use the speed of sound C_s. We get:

$$\frac{|\Delta h_s|}{1/2 C_s^2} = \frac{2}{\gamma - 1} \left| \left(\left(\frac{P_r}{P_a} \right)^{(\gamma-1)/\gamma} - 1 \right) \right| = \Omega \tag{7.3.20}$$

Usually this kinetic energy is taken equal to 1/2 U^2, corresponding to the maximum kinetic energy in the rotor. This way of doing is equivalent to multiplying Ω by (M_a). This defines the enthalpy factor ψ, equal to twice the first Rateau coefficient μ (itself equal to the power factor introduced in section 7.1.4).

In practice, the performance of a machine is usually given in the form of characteristic curves for constant values of the corrected rotation speed N_c:

$$\frac{P_r}{P_a} = f(\dot{m}_c) \qquad \eta_s = f\left(\frac{P_r}{P_a}\right) \quad \text{or} \quad f(\dot{m}_c)$$

Curves of equal efficiency can be directly plotted on charts:

$$\frac{P_r}{P_a} = f(\dot{m}_c)$$

Sometimes, the curves are presented in relation to reference values of the corrected mass flow rate or corrected speed.

Note that the performance maps of turbomachinery (as well as displacement machines) play a unique role in that they constrain intensive variables with extensive variables, whereas usually they are independent: the flow affects the compression ratio.

Examples of experimental results are presented in the following pages, corrected flows and corrected speeds being reported to their nominal values.

7.3.3.1 Performance maps of dynamic compressors

We content ourselves in this section to present some general results concerning the performance maps of dynamic compressors, which will be completed in Chapter 37 of Part 5 on the off-design behavior of these machines.

For dynamic compressors (Figures 7.3.5 to 7.3.7) the corrected mass flow is used as abscissa. The ordinate is the pressure ratio or the isentropic efficiency. The corrected rotation speed is still used as parameter.

They have the same shape as diffuser characteristics, for different rotation speeds.

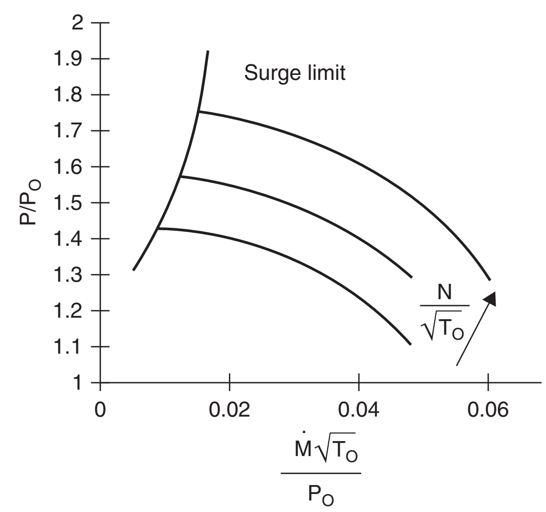

FIGURE 7.3.5

Dynamic compressor performance map

The corrected rotation speed strongly influences the performance of dynamic compressors. This is easily explained considering that in such a machine, energy is transmitted to the fluid by the rotor in the form of kinetic energy. At most, this energy is 1/2 U^2, that is to say, is proportional to N^2. It is therefore natural that the sensitivity of these machines to regime change is dramatic.

In addition, fluid mechanics tells us that the fluid flow in the blades is destabilized here by the pressure gradient. As soon as one deviates too much from the nominal operating conditions, there is a significant risk of stream separation along the blades. Besides a strong sensitivity of isentropic efficiency, this results in a double limitation of range of use of the machine: the risk of surge at low flow (which depends on the network in which the compressor discharges, cf. next section), and stalling on the side of higher flow rates. Some flexibility is indeed possible in this type of machine only if you can freely adjust the rotation speed and possibly the angle of incidence of certain rows of blades (guide vanes).

Axial compressors

The compression ratios per stage that can provide axial compressors are relatively low, generally between 1.2 and 2.

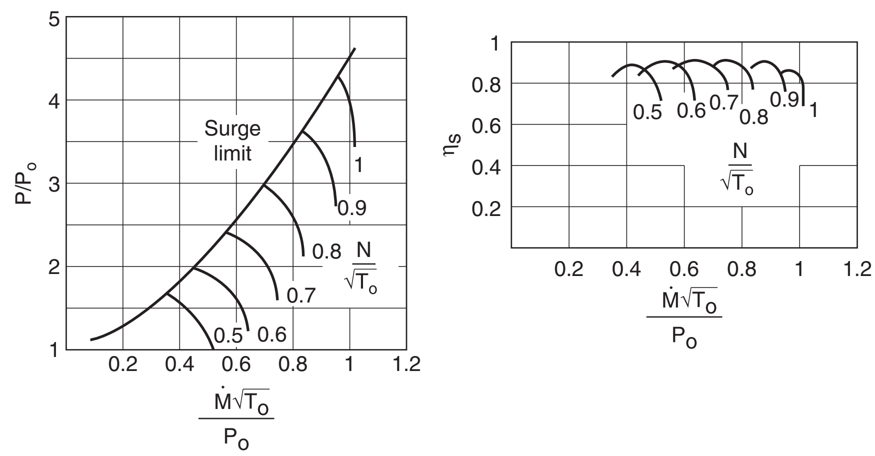

FIGURE 7.3.6

Axial compressor performance map

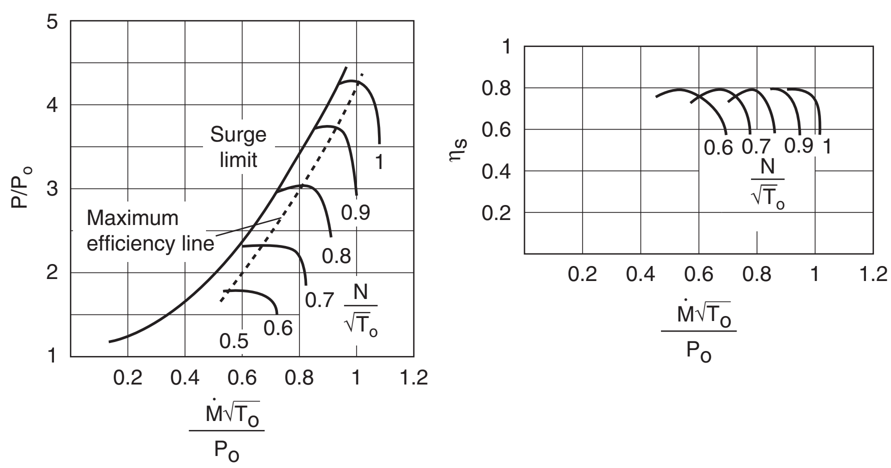

FIGURE 7.3.7

Centrifugal compressor performance map

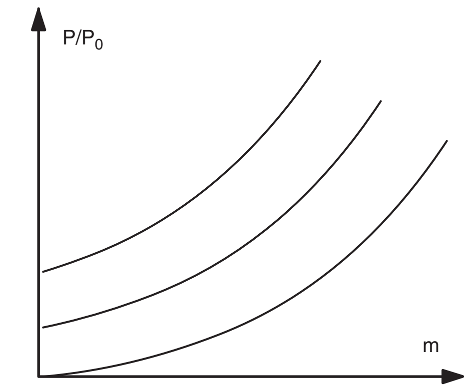

FIGURE 7.3.8
Circuit losses

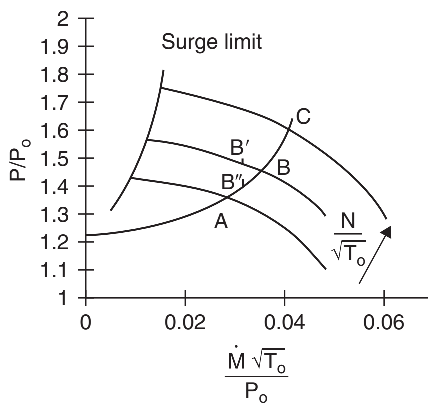

FIGURE 7.3.9
Stable operation point

Centrifugal compressors

Characteristics of centrifugal compressors are steeper than those of axial compressors, making them more stable. Due to the acceleration that the fluid receives during its passage through the rotor, the compression ratio per stage they can provide is much more important, between 2.5 and 9.

It will be noted on the characteristic of Figure 7.3.11 that we can equip the (pressure − flow) performance map by including the iso-isentropic efficiency η lines. The locus of these values corresponds substantially to portions of ellipses. They are commonly called efficiency islands.

7.3.3.2 Surge in dynamic compressors

When a dynamic compressor delivers in a receiver circuit, its operating point is determined by the intersection of its characteristic curve and that of the circuit.

The losses of the circuit being in first approximation proportional to the square of flow rate, the shape of the circuit characteristic is given in Figure 7.3.8. The ordinate intercept, sometimes called static load, corresponds to terms independent of flow in the generalized Bernoulli equation (7.3.3).

The operating point of a compressor evolves thus according to its rotation speed, so that at any time, the power dissipated in the circuit equals the shaft power of the machine, and passes from A to C when the rotation speed increases (Figure 7.3.9).

Things do not always happen as simply, as a zone of unstable operation may appear in the left part of the dynamic compressor performance map maximum for the reasons explained below.

Let us first check that a point such as B is stable (Figure 7.3.9). Suppose that the machine being in nominal operation at point B at constant speed, the flow drops suddenly. Given the shape of the characteristic of the machine, its operating point tends to move to B′, while the resistance of the circuit drops to B″.

It follows that the compressor has more power (B′ − B″) than that required by the resistance of the circuit, which has the effect of increasing the flow through it, and thus brings the point to B. The system is naturally stable.

If the operating point lies to the left of the compressor characteristic maximum, two cases may arise:

- if the slope of the characteristic of the circuit is greater than that of the machine, a reasoning similar to the previous shows that the system is stable.
- if, as in the chart of Figure 7.3.10 it is the compressor characteristic which is the steepest, we see that any flow-rate reduction translates into a reduction of the power of the machine faster than circuit resistance. So there is amplification of the initial disturbance, and the system is unstable. Complex phenomena occur then, depending on the mechanical inertia of the entire system. The flow rate decreases very rapidly (20 to 80 milliseconds) and may even reverse.

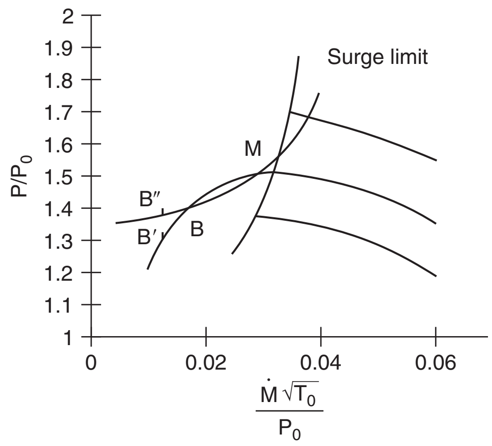

FIGURE 7.3.10
Unstable operation point

Circuit resistance falls accordingly, the discharge manifold empties and the compressor available power becomes surplus, which leads to a very rapid increase in flow. Very quickly, the point of functioning returns to the right of B′. The discharge pipe is filled, the pressure rises, and flow lowers again: cyclic conditions are met. We call this instability a surge cycle, whose period varies between 0.5 and 5 seconds. Surge is generally very detrimental to the mechanical compressor, which is subjected to intense vibration that can put rotor and stator in touch.

The phenomena involved are very complex because of the inertial effects associated with filling and emptying the collector network. They are still poorly known. So manufacturers recommend users to avoid operating in the left of the performance map maximum of their machines, bounded by the surge limit called surge line.

As it is not always possible to meet this constraint, it is often necessary to provide check valves at the outlet of the machine, or more complex devices when it is not enough. Instrumentation and modern control provide opportunities to detect the onset of surge, which is accompanied by changes in pressure and flow as well as in suction temperature for axial compressors. The anti-surge control must be able to react very quickly given the phenomenon speed, increasing the flow through the opening of a bypass valve.

7.3.4 Practical calculation of dynamic compressors

To be able to size their compressors quickly with sufficient accuracy, manufacturers take full advantage of the similarity rules. The main difficulty, however, is that the situation is not as simple as in the case of positive displacement compressors.

In the latter case, in fact, an entire range of machines can be represented with sufficient accuracy by the same set of charts giving the volumetric efficiency and the compression ratio as a function of compression ratio.

In the case of turbomachinery, this is approximately true, and manufacturers prefer to use a characteristic per wheel, if they want good accuracy, because the shapes of efficiency islands may vary significantly from one machine to another, even for a given range. However they use similarity rules for the flow, and, very often, performance maps are plotted in a particular system of axes, combining the enthalpy and flow factors (or Rateau coefficients) and the wheel Mach number.