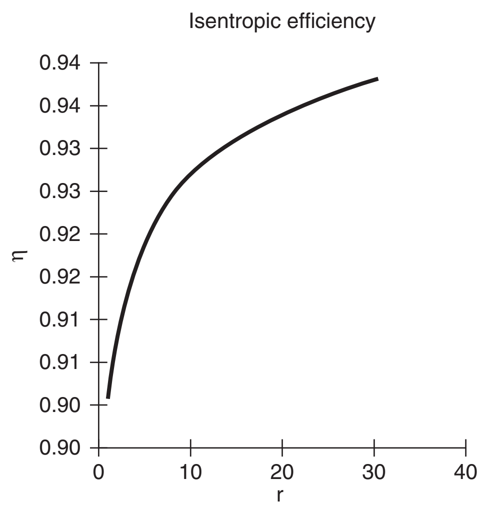

FIGURE 7.5.2
Isentropic efficiency function of expansion ratio

Knowing the process efficiency η (isentropic or polytropic), we can deduce the actual work of expansion:

$$\tau = \frac{\eta}{\eta_{pref}}(h - h_a)$$

The heat to provide is then:

$$Q = h_r - h_a - \tau = \left(1 - \frac{\eta}{\eta_{pref}}\right)(h_r - h_a) \tag{7.5.3}$$

While a polytropic compression leads to $k > \gamma$, we obtain in the case of a polytropic expansion: $k < \gamma$.

Unlike what happens in a compressor, given a constant polytropic efficiency (0.9 in Figure 7.5.2), the expansion isentropic efficiency increases when the expansion ratio increases.

This arises because the irreversibilities taking place at high temperature (and pressure) are partially recovered in the subsequent expansion stages, as they have the effect of warming the fluid.

7.5.2 Calculation of an expansion in Thermoptim

In Thermoptim the calculation of expansions is done in a manner analogous to that of compression, with the same screen. You should therefore refer to what was presented in section 7.1.5.

7.5.3 Turbines

Geometrical arrangement

The general layout of a turbine is given Figure 7.5.3. The stator is made of guide nozzles which accelerate the fluid, while the wheel (rotor) converts into mechanical energy at least part of the enthalpy available.

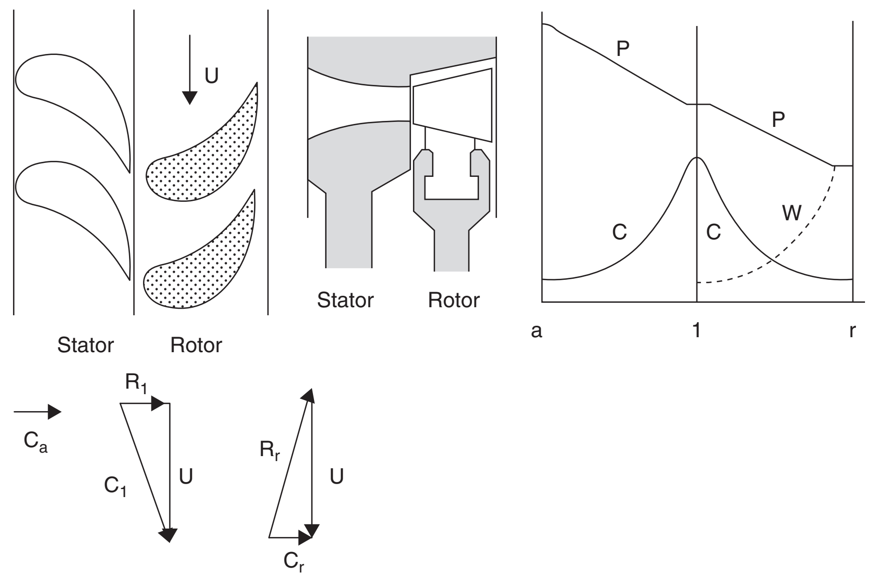

FIGURE 7.5.3
Velocity and pressure profile in an axial turbine stage

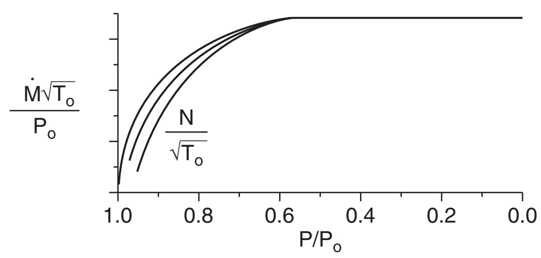

FIGURE 7.5.4
Turbine performance map

Changes in pressure and fluid velocity in a turbine

In a turbine, the evolution of the fluid is an expansion. The compressible flow equations indicate that for a subsonic regime, the section of the vein should decrease, and the velocity increase. This evolution takes place in two steps (Figure 7.5.3): in the stator, the absolute velocity increases, while in the impeller the relative velocity increases, and the absolute velocity decreases.

7.5.4 Turbine performance maps

Just as we did section 7.3.3.1, we will only present in this section some general results concerning performance maps of turbines, which will be completed in chapters 15 of Part 3 and 37 of Part 5 dealing with the off-design behavior of machines.

In the case of a turbine (Figure 7.5.4) it is conventionally the pressure ratio which is used as abscissa. As ordinate, we find the corrected mass flow or isentropic efficiency of the machine. The curve parameter is still the corrected rotation speed, which plays a secondary role.

We can see the flexibility of turbines to adapt to various operating conditions: efficiency tends to deteriorate only when we dramatically reduce the pressure ratio or the speed. This flexibility is particularly due to the flow stability in the blades due to the gradient of pressure therein. But what is most remarkable is the stability of the flow for the high pressure ratios, which comes from the supersonic regime, which takes place in at least part of the machine (the flow is choked at the place where the speed of sound is reached).

The limiting value reached by the flow, when the pressure ratio exceeds the critical ratio is called critical flow. It is proportional to the throat section, which is obviously independent of the rotation speed, which explains the weak influence of this parameter.

In a supersonic nozzle the corrected mass flow is proportional to the throat section and independent of inlet conditions. When the nozzle is caused to operate outside the rated conditions, the fluid flow somehow adapts to the shape of the duct. The decrease in efficiency resulting from this adaptation is especially important when one wants to exceed the design pressure ratio.

As the expansion ratio remains above its critical value, the flow stays below the critical flow. When its value decreases beyond this value, the flow maintains a constant value.

If you want to go beyond the critical expansion ratio in a converging – diverging nozzle, the flow cannot increase sufficiently, and the expansion continues beyond the exit section with large oscillations at the outlet of the nozzle.

Indeed, if the flow is supersonic and the pressure in the outlet chamber does not match that used for the design, the following phenomenon takes place: if the outlet chamber pressure is greater than the nominal pressure, there is hyper-expansion in the nozzle, and thus sharp contraction at the outlet, otherwise, there is hypo-expansion in the nozzle, and thus sudden expansion at the exit.

The explanation for this phenomenon is clear: beyond the contracted section, the fluid velocity exceeds the speed of sound. Consequently, the conditions set on the outlet vein of this section may no longer influence the flow at the inlet of the throat. It continues downwards to the throat as if the final pressure was the rated outlet pressure, and if the actual pressure is different, a discontinuity is inevitable.

7.5.5 Degree of reaction of a stage

The study of the energy balance of a stage showed that the enthalpy change takes place partly in the rotor, and partly in the diffuser. We call the degree of reaction ε the fraction of the enthalpy change that takes place in the rotor. With the notation of Figure 7.3.3, we have:

$$\varepsilon = \frac{h_l - h_a}{h_r - h_a} = \frac{h_l - h_a}{\Delta h} \tag{7.5.4}$$

By definition, ε is between 0 and 1.

In a dynamic compressor stage $0.5 \leq \varepsilon \leq 1$, is commonly obtained. The lower limit ($\varepsilon = 0.5$) is approximated in the axial dynamic compressors. As for the upper limit (pure reaction: $\varepsilon = 1$), it is reached in the case of a single stage machine whose diffuser is nonexistent or of negligible effectiveness.

In a turbine stage, pure reaction is impossible. Two limiting cases frequently arise:

- **impulse turbine**s, in which $\varepsilon = 0$: any expansion of the fluid is then carried out in fixed blades or nozzles, at the inlet of the wheel, and the inlet and outlet pressures of the rotor are equal;
- **reaction turbines**, where $\varepsilon = 0.5$: the expansion is then evenly distributed between the nozzle and the wheel.

Each of these two types of turbine has advantages and disadvantages of its own. As we have said, it is not our intention to develop here in detail the theory of turbines. Simply note that the impulse turbines are generally used for head stages of multistage sections or for small capacity units, while reaction turbines are proving well suited for the low-pressure sections.

Indeed, a first advantage of using **impulse turbines** in the high pressure sections is that the entire expansion being made in the stator, the rotor is not subjected to a high pressure difference, which limits the mechanical constraints. A second advantage is that the flow in these turbines can be reduced by using partial injection, which is to only supply a fraction of fluid in the stator vanes. This type of operation is made possible in this case because the pressure is the same on both sides of the rotor, and no parasitic flow is expected in the non-injected parts.

However, the efficiencies of these turbines are slightly (2–3%) worse than those of **reaction turbines**, which are subjected to significant axial thrust, and do not use partial injection.

Because irreversibilities taking place in the top stages are partially recovered in subsequent stages, we can tolerate that their efficiency is slightly lower, which allows us to use impulse turbines in this case.

7.6 COMBUSTION

Combustion phenomena are of particular importance in the study of energy technologies, because they are the source of most heat and power production in the world: more than 90% of global consumption of primary commercial energy comes from burning coal, oil or natural gas.

The study of combustion is to determine the state and composition of combustion products, and consequently, their thermochemical properties including the amount of energy that is involved in the reactions. Moreover, the combustion conditions largely determine the quantities of pollutants emitted by energy technologies.

The purpose of this section is to determine as closely as possible the combustion characteristics that are important in terms of energy: the thermodynamic properties of the products (c_p, M, h, s), the energy released by combustion (ΔH_r), the concentration of unburned hydrocarbons (UHCs) and pollutants.

From these elements, it becomes possible to optimize the combustion, i.e. to obtain the best combustion efficiency by adjusting the air/fuel ratio or the excess air, which sets out the flame temperatures and losses by the fumes.

To achieve this goal it is necessary to know how to characterize fuels and write the complete reactions, stoichiometric and with excess or lack of air, taking into account the dissociation where it exists, which requires knowledge of general principles that govern equilibrium reactions (law of mass action), and finally calculate the energies released.

In the remainder of this book, we will always assume that the combustion reaction involves gaseous species, and that the various components are treated as ideal gases. Dalton's law then allows us to consider that the combustion products follow the ideal gas law.

The fundamentals you should make sure to understand are the following: air factor λ (7.6.2.4), adiabatic combustion temperature (7.6.4.6), CO_2 dissociation (7.6.3.2), quenching temperature (7.6.1.6) and heating value (7.6.4.5).

7.6.1 Combustion phenomena, basic mechanisms

Combustion reactions are rather unusual chemical reactions, typically brutal, where the non-equilibrium is the rule, equilibrium the exception. In these circumstances, it is still impossible to determine exactly during the reaction the evolution of thermodynamic functions characterizing the systems involved. They can only be calculated assuming that equilibrium conditions are met before and after the time of combustion.