2

First Steps in Thermodynamics: Absolute Beginners

Abstract: The purpose of this chapter is to discuss key concepts which should be presented to absolute beginners in order to enable them to understand and study the cycles of three basic energy technologies: steam power plant, gas turbine and refrigerating machine.

We are talking of lightweight educational presentation as we seek to minimize the background in mathematics and physics necessary for understanding these cycles, our goal being to make them accessible to readers unfamiliar with the language of specialists in thermodynamics. We show in particular that essential concepts can be presented without resorting to a function that can be difficult to master, entropy, which will be introduced only in the second part of this presentation.

The steam power plant and gas turbine are engines intended to convert heat into mechanical power, while the third cycle is a machine designed to extract heat at a temperature lower than the atmosphere thanks to a mechanical energy input.

2.1   ARCHITECTURE OF THE MACHINES STUDIED

Let us begin with a brief description of the operation and architecture of these machines.

2.1.1   Steam power plant

The basic cycle of a steam power plant is essentially a boiler where the fuel (solid, liquid or gas) is burned for generating steam (usually superheated) which is then expanded in a steam turbine whose shaft provides the motor work (Figure 2.1.1).
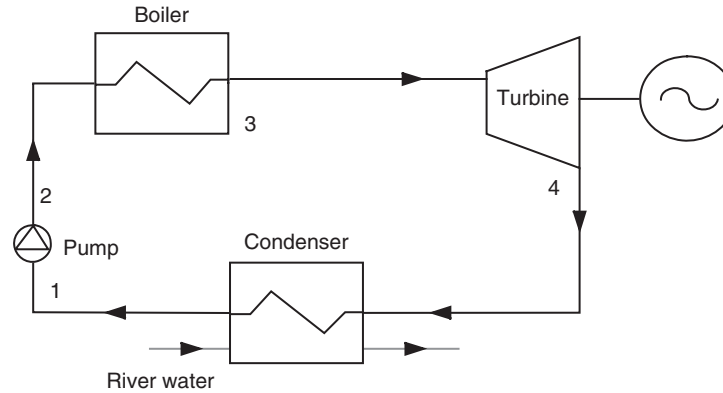
FIGURE 2.1.1
Sketch of a steam power plant

The steam leaving the turbine is completely liquefied (water) in a condenser before a pump restores the boiler pressure. Since water is compressed in the liquid state, the compression work is almost negligible compared to the work recovered on the turbine shaft.

The condenser cooling is provided by an external cold source, usually the outside air or water from a river as shown in Figure 2.1.1.

The pump is generally centrifugal, multistage as it must provide a significant compression ratio.

The boiler has three successive functions and behaves thus like a triple heat exchanger. It must:

- heat pressurized feedwater (in the economizer) to the vaporization temperature corresponding to the pressure;
- vaporize steam;
- and finally superheat steam at the desired temperature.

Steam turbines are mostly multistage axial turbines.

The condenser is a heat exchanger whose particularity is to work at a pressure lower than the atmosphere, given the low vapor pressure of water at room temperature.

We speak of cycle because, going through the four successive plant components, the working fluid undergoes a series of processes that lead it to return in its original state.

More specifically, it is an external combustion cycle, allowing use of a variety of fuels (including uranium). In most countries, over 90% of the stock of thermal power plants until recently was composed of such plants.

2.1.2 Gas turbine

In its simplest and most common form (Figure 2.1.2), a gas turbine is composed of three elements:

- a compressor, usually centrifugal or axial, which is used to compress the intake air at a pressure between 10 and 30 bar in modern machines;
- a combustion chamber in which fuel is injected under pressure, combusted with air previously compressed (the latter in large excess in order to limit the burnt gases temperature at the turbine inlet);
- a turbine, usually axial, in which are expanded the high temperature gases exiting the combustion chamber. A significant proportion (60–70%) of the recovered work on the turbine shaft is used to drive the compressor.
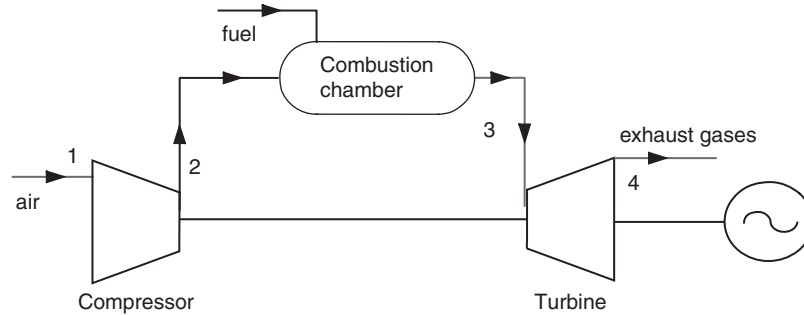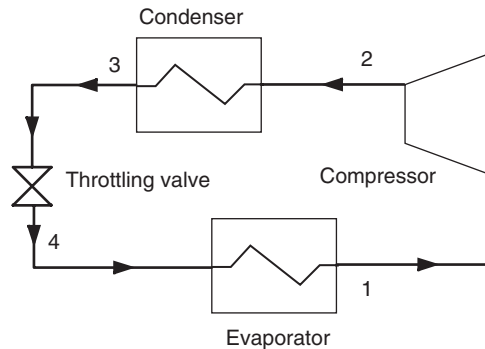
FIGURE 2.1.2

Sketch of a gas turbine

FIGURE 2.1.3

Sketch of a compression refrigeration cycle

In this form the gas turbine is a continuous stream internal combustion engine. Note that the gas turbine term comes from the status of the working fluid, which remains gaseous, and not the fuel used, which can be both gaseous and liquid (gas turbines typically use natural gas or light petroleum distillates).

To achieve compression ratios of 20 to 30, the compressor is multistage, with sometimes intermediate cooling to reduce the work consumed.

The combustion chamber is normally constructed of refractory alloy.

There are major technological constraints at the first stages of the expansion turbine, which are subject to the flow of burnt gas at high temperature. The parts most at risk are especially the rotor blades, which are very difficult to cool and in addition particularly sensitive to abrasion. It is therefore important to use a very clean fuel (no particles and chemicals that may form acids), and limit the temperature depending on the mechanical characteristics of the blades.

Exhaust gases being released into the atmosphere, it is inappropriate to speak of a cycle. However we do say it, in everyday language.

2.1.3   Refrigeration machine

In a refrigeration system (Figure 2.1.3), one seeks to maintain a cold chamber at a temperature below ambient. The idea is to evaporate a refrigerant at low pressure (and therefore low temperature) in a heat exchanger in contact with the cold chamber. For this, we need the refrigerant temperature to be lower than that of the cold chamber. The fluid is then compressed at a pressure such that its condensation temperature is greater than the ambient temperature. It is then possible to cool

the fluid by heat exchange with ambient air until it becomes liquid. The liquid is then expanded by isenthalpic throttling at low pressure and directed into the evaporator. The cycle is thus closed.

In a domestic refrigerator, the evaporator usually consists of two corrugated flat plates welded against each other, the refrigerant flowing in the channels formed by the corrugations. It generally lines the "freezing" compartment of the refrigerator (the frost layer forms on it). The plate between the flow channels operates as a fin to increase the thermal contact between the refrigerant and the cold compartment. This evaporator is connected to the rest of the machine by two tubes that cross the insulated wall. One of them is connected to the compressor suction, the other to the expansion valve.

The condenser is the black grill located on the refrigerator backside, consisting of coiled tubing supported by metal plates which firstly increase the heat exchange with the air, and secondly reinforce the mechanical rigidity. It is connected to the compressor outlet and expansion valve.

In most cases, the compressor is not directly visible because it is contained in a metal block mounted on rubber cushions to prevent vibrations, out of which exit an electric wire and two input and output fluid pipes. Such a compressor, generally of piston type, is called a hermetic compressor, which has the advantage that the engine is directly cooled and lubricated by the working fluid, without need for oil.

The expansion valve usually consists of a single capillary tube, and sometimes is a thermostatic valve.

2.2   FOUR BASIC FUNCTIONS

All three of these machines are crossed by thermodynamic fluids, steam or liquid water in the first, air and flue gases in the second, a refrigerant in the third.

What is particularly remarkable is that even if the technical solutions implemented are varied, as we have seen, these fluids are subjected to only four different types of processes:
- compression;
- expansion with work production;
- expansion without work production;
- temperature change (heating and cooling).

Therefore four functions are sufficient to describe the operation of these machines:
- compression can occur with the fluid being liquid or gaseous. In the first case the component is a pump, in the second a compressor;
- expansion with work production is generally made through turbines;
- expansion without work production in valves;
- heating can be generated either in combustion chambers or boilers, or in heat exchangers. Cooling is usually done in heat exchangers.

This finding has a very broad bearing: in all engines, the fluid is always successively compressed, heated, expanded and cooled, and, in all refrigeration cycles, it is compressed, cooled, expanded and heated.

Note finally, and this is very important in practice, that the flow of fluids through these components is either cyclical (in turbines, pumps and compressors), or continuous (in others).

In what follows, we will show how to characterize these processes in terms of thermodynamics. To do this we will focus on a small amount of fluid and we will seek to determine the evolution of its thermodynamic properties in these processes. In practical terms, it suffices to know what is called its state at the component inlet and outlet in order to calculate the performance of the entire machine.

2.3   NOTIONS OF THERMODYNAMIC SYSTEM AND STATE

It is necessary at this stage to introduce the concept of **thermodynamic system**, which represents a quantity of matter isolated from what we call the **surroundings** by a real or fictitious **boundary**. This system concept is very general in physics and is found especially in mechanics.

The notion of **state of a system** represents "the minimum information necessary to determine its future behavior". This state is defined by what is called a set of state variables allowing us to completely characterize the system at a given moment.

In mechanics, the position coordinates and speed determine the state of a system.

In thermodynamics, there are obviously several sets that meet this definition. The ones most used in the literature are the following pairs: (pressure, temperature), (pressure, volume), (temperature, volume).

A **state function** is a quantity whose value depends only on the state of the system, not its history.

In thermodynamics, we are led to distinguish two types of systems: **closed systems** that do not exchange matter with the surroundings, and **open systems** that do exchange.

The presentation made earlier of the operation of components involved in the machines we want to study has shown that they operate in open systems, as fluids pass through them.

This distinction is important because thermodynamic properties are expressed differently in closed and open systems.

2.4   ENERGY EXCHANGE BETWEEN A THERMODYNAMIC SYSTEM AND ITS SURROUNDINGS

It is essential to note that thermodynamic systems involved in the components we are interested in, only exchange energy with the surroundings in **two** distinct forms:
- heat by heat exchange on the system boundaries. It is usually denoted by Q;
- work, by action of pressure forces on the boundaries, the work of gravity forces being neglected. This work is usually denoted W in closed and τ in open systems.

One can also easily show that in an open system, for an infinitely small change, τ and Q are given by the following equations:

$$\delta\tau = v\,dP \tag{2.4.1}$$

$$\delta Q = C_p\,dT - v\,dP \tag{2.4.2}$$

The latter expression simply expresses an experimental fact, which is an essential basis of compressible fluid thermodynamics: heat δQ exchanged with the surroundings is reflected in a linear change in the thermodynamic state of the system.

2.5   CONSERVATION OF ENERGY: FIRST LAW OF THERMODYNAMICS

The fundamental law that governs the behavior of thermodynamic systems is the conservation of energy, known as the first law.

For a closed system, it can be stated as follows: the energy contained in a system which is isolated, or evolving in a closed cycle, remains constant whatever processes it undergoes. The various forms that the energy of a system can take: mechanical energy, heat energy, potential energy, kinetic energy, are all equivalent under the first law. Let us recall that, in our case, only heat and work are taken into account.

Calling W the work of external pressure forces and Q the heat exchanged with the surroundings, and neglecting the kinetic energies brought into play, the first law can be written for a closed system in the form:

$$\Delta u = W + Q \qquad\qquad 2.5.1$$

u is an extensive quantity called the internal energy of the system. It is a state function.

In this form, this law is very intuitive and readily accepted: it is a conservation law which states that energy (like mass) is neither lost nor created.

The internal energy, however, has meaning only if the system is closed, and needs to be generalized when considering a system in which matter enters and/or out of which it flows.

The work W exerted on a closed system can indeed be broken into two parts: one that is exerted on the mobile walls of the system if any, called shaft work τ, and one that is exerted on boundaries crossed by fluid coming out of the system and entering it. For open system components, which is the case for those that we are studying, this work, called transfer work, equals $-\Delta(Pv)$.

We therefore have: $W = \tau - \Delta(Pv)$

Introducing a function called enthalpy h, such as $\Delta h = \Delta u + \Delta(Pv)$, the first law is written in open systems:

$$\Delta h = \tau + Q$$

The first law is expressed as follows in open systems: the enthalpy change of an open system is equal to the amount of shaft work exerted on the moving walls and heat exchanged with the surroundings.

Enthalpy thus appears simply as a generalization in open systems of the internal energy of closed systems. In practical terms, just consider this state function as the energy associated with the system under consideration, neither more nor less.

2.6 APPLICATION TO THE FOUR BASIC FUNCTIONS PREVIOUSLY IDENTIFIED

We will show in this section that the enthalpy variation of the fluid flowing through them is sufficient to determine the energy involved in these four elementary processes.

2.6.1 Compression and expansion with work

Expansion can be made with and without work. In the first case, the machine most commonly used is the turbine. In the second case, it is a simple valve or a filter (see section 2.6.2).

Machines doing the compression or expansion of a fluid have a very compact design for reasons of weight, size and cost. For similar reasons, they rotate very fast (several thousand revolutions per minute). Each parcel of fluid remains there very briefly.

Moreover, fluids brought into play in compressors and turbines are often gas whose heat exchange coefficients have low values.

Short residence time, small areas of fluid-wall contact, and low exchange coefficients imply that the heat exchange is minimal and that **the operation of these machines is nearly adiabatic**: Q = 0.

In a compression or expansion adiabatic machine, shaft work τ is thus equal to the change in enthalpy of the fluid Δh.

2.6.2   Expansion without work: valves, filters

There is a class of devices, such as the expansion valve of the refrigeration machine, where τ and Q are both zero, i.e. $\Delta h = 0$: they are static expanders such as valves and filters. The corresponding process is called an **isenthalpic throttling** or a **flash expansion**.

2.6.3   Heat exchange

Components which transfer heat from one fluid to another require large exchange areas, their heat fluxes being proportional to these areas. Technical and economic considerations lead to the adoption of purely static devices. For example, large bundles of tubes in parallel, traversed internally by one fluid while the other flows outside.

τ is zero because of the absence of movable walls.

In a heat exchanger, heat Q transferred or provided by one fluid to another is equal to its enthalpy change Δh.

2.6.4   Combustion chambers, boilers

In a combustion chamber or boiler, there are no movable walls either, and $\tau = 0$.

Heat Q transferred to the fluid passing through is equal to the enthalpy change Δh.

2.7   REFERENCE PROCESSES

The previous section showed that the determination of the enthalpy change of the fluid flowing through them is enough to calculate the energy involved in these four elementary processes.

But this information is not sufficient to characterize them completely. The physical analysis of their behavior helps to highlight the **reference processes** corresponding to the operation of components that would be ideal.

It is then possible to characterize the actual process by introducing an imperfection factor, often called effectiveness or efficiency, which expresses its performance referred to that of the reference process. This way makes it much easier to understand the processes undergone by the fluid.

The choice of reference processes is based on analysis of physical phenomena that take place in the components: **it is an absolutely essential modeling choice**.

Lastly, as we shall see, the reference process is very useful when trying to plot the cycle studied on a thermodynamic chart.

2.7.1   Compression and expansion with work

In section 2.6.1 we saw that the compressors and turbines are machines in which heat exchanges with the outside are usually negligible; this is referred to as adiabatic. The reference process for compression or expansion with work is the perfect or reversible adiabatic. Its equation can be obtained by integrating the differential equation expressing that the heat exchanged is zero at any

time, i.e. $0 = C_p \, dT - v \, dP$. For a perfect gas, for which $Pv = rT$, the corresponding curve is easily obtained. It is given by law $Pv^\gamma = $ Const. with $\gamma = C_p/C_v$.

2.7.2 Expansion without work: valves, filters

For throttling or expansion without work conserving enthalpy ($\Delta h = 0$), the reference process is isenthalpic.

2.7.3 Heat exchange

As a first approximation, as the pressure drops are relatively low, heat exchange can be assumed to be isobaric. The reference process is thus isobaric.

2.7.4 Combustion chambers, boilers

Similarly, combustion chambers and boilers can usually be regarded as isobaric. Combustion takes place at constant pressure, therefore, which still surprises a number of students, who believe that the combustion raises pressure, even in open systems.

To illustrate this, consider a fairly common example of boiler: a wall-mounted gas boiler in an apartment. The pressure will remain the same in both fluids, whether the hot water circuit at a pressure of 1 to 3 bar, or air circuit and smoke, at atmospheric pressure of course.

2.8 SUMMARY REMINDERS ON PURE SUBSTANCE PROPERTIES

Let us recall that a pure substance can be in one or more of three phases: solid, liquid or gaseous. When heating a liquid it turns to vapor, and we talk of vaporization. The temperature at which this change is realized depends on the pressure exerted on the substance considered. It remains **constant** as long as the vaporization is not complete, i.e. while its quality x is between 0 and 1. Let us recall that x is defined for a pure substance two-phase mixture as the ratio of vapor mass to total mass (vapor + liquid).

For this phase change to happen, it is necessary to provide or absorb energy, called **latent heat of change of state**. During the change of state, there are significant variations in the specific volume, the vapor being about 600 to 1000 times less dense than the liquid. This change in specific volume occurs at **constant pressure and temperature**.

The **critical point** represents the state where the phase of pure vapor has the same properties as the pure liquid phase. At higher temperatures and pressures (supercritical), it is not possible to observe a separation between liquid and gas phases: the disk surface which separates the liquid and vapor phases disappears at the critical point.

We call an ideal gas a gas whose internal energy and enthalpy depend only on the temperature, which simplifies the modeling: its equation of state is $Pv = RT$.

Many thermodynamic fluids in the vapor phase may be treated as ideal gases in a wide range of temperatures and pressures. This requires that the temperature-pressure combination deviates from the condensation zone as much possible (that is to say that the pressure is not "too" high or the temperature "too" low). Such conditions are commonly the case for gases known as "permanent"

at ambient temperature and pressure, as hydrogen, oxygen, nitrogen, the oxygen-nitrogen mixture that is dry air etc.

2.9   RETURN TO THE CONCEPT OF STATE AND CHOICE OF STATE VARIABLES TO CONSIDER

Now that we have introduced the first law, established its equation for open systems and showed just how it applies to the four basic changes experienced by fluids in machines that interest us, we can discuss the advantages and disadvantages of different state variables that one can consider using.

We have seen that several sets of state variables can be used to characterize a thermodynamic system. The most "natural" are temperature, pressure and volume, but there are others, like the enthalpy just defined, and secondly they are not fully satisfactory for of our goals, as we shall show.

Pressure P is essential, both because it directly determines the mechanical stress of components, and because, as we have seen, the reference process for fluid heating and cooling is isobaric.

Temperature T is also essential, but, unlike its predecessor, the isothermic process does not correspond to any process that concerns us.

Moreover, the **pair (P, T)** is not sufficient to represent the process experienced by a fluid as it changes state, the two variables being linked by the saturation pressure or temperature law: it lacks the vapor quality x.

Volume v intervenes very little in the analysis of interest, even in a closed system, because volume varies due to the existence of movable walls. In fact, its main practical interest is in the sizing of flow sections.

Enthalpy h is a fundamental variable too, because it is directly related to energy exchanges that take place in the machines. For an ideal gas, it is a linear function of temperature, very easily deduced, and in the liquid-vapor equilibrium zone, it provides additional information on the quality. Finally, remember that the isenthalpic process is the reference process for an expansion without work.

For our purposes, as part of this lightweight educational presentation, these remarks are sufficient to allow us to conclude that the pair **(P, h)** is a set of state variables of particular interest. We shall see the implications when we seek to plot the cycles of the machines we are studying in thermodynamic charts.

The analysis of processes undergone by fluids during compression and expansion with work showed that their reference process is reversible adiabatic. We indicated that for a perfect gas it follows law $Pv^{\gamma} = \text{Const}$.

Let us note, but incidentally given our focus for simplicity, that this law is that of the **isentrope**. **Entropy s** is also a state function widely used in thermodynamics, especially because the isentrope is the reference process for compression and expansion.

2.10   THERMODYNAMIC CHARTS

Thermodynamic systems which we consider can be characterized by two state variables; they are called bivariant. This means that their thermodynamic properties can be plotted in a plane in the form of a thermodynamic chart.

By highlighting the reference process of changes undergone by fluids and allowing us to calculate them, at least approximately, charts are among the basic tools of thermodynamics. Their interest is twofold:
* to help plot the cycles;
* to facilitate the estimation of the thermodynamic state of the various cycle points.

Because of the possibilities offered by software packages for calculating fluid properties that are increasingly common, the second interest tends to decline while the former retains its relevance. The display on a chart of a cycle calculated using a computerized tool helps ensure that it does not contain an abnormal point due to an error when entering data.

A chart is presented in a graphical map with the plot of a number of remarkable curves, including families of state function iso-values.

2.10.1 Different types of charts

In practical terms, the main processes that can occur in industrial processes involving pure fluids are, as we have seen, compression, expansion, heat exchange and throttling.

It is clear that the temperature T, pressure P and the steam quality x are state variables whose knowledge is necessary to study these processes and equipment design.

The above remarks show that the enthalpy h is also very important. Finally, knowledge of the specific volume v is necessary to size the pipes, since it allows us to convert mass flow rate into volume rate.

In conclusion, the most interesting quantities are in practice T, P, h, and additional information on x and v may be necessary. So the abscissas and ordinates of the charts that we can consider should be chosen among them (recall that we exclude here the entropy).

The pair (T, h) is rarely retained because the isobaric and reversible adiabatic are represented by curves with inflection points that make their use difficult. In addition, variations of T and h are proportional when the fluid follows the ideal gas law.

Except for permanent gases, i.e. whose state is very far from their saturation conditions, the pair (T, P) is insufficient, because T and P are bound by the saturation law in the liquid-vapor equilibrium zone. It would however be quite suitable to represent gas turbine cycles.

The pair (P, h) is increasingly used, usually with a logarithmic scale for pressures. Its widespread use has been promoted by refrigeration engineers, and it will be studied further.

Finally, the pair (P, v), Clapeyron chart, has a certain educational value, especially for the study of changes in closed systems. Its main drawback is its low visibility, the area of vapors being reduced and energy functions not appearing directly.

In conclusion, as this lightweight educational presentation does not include entropy, the pair (P, h) appears the best suited.
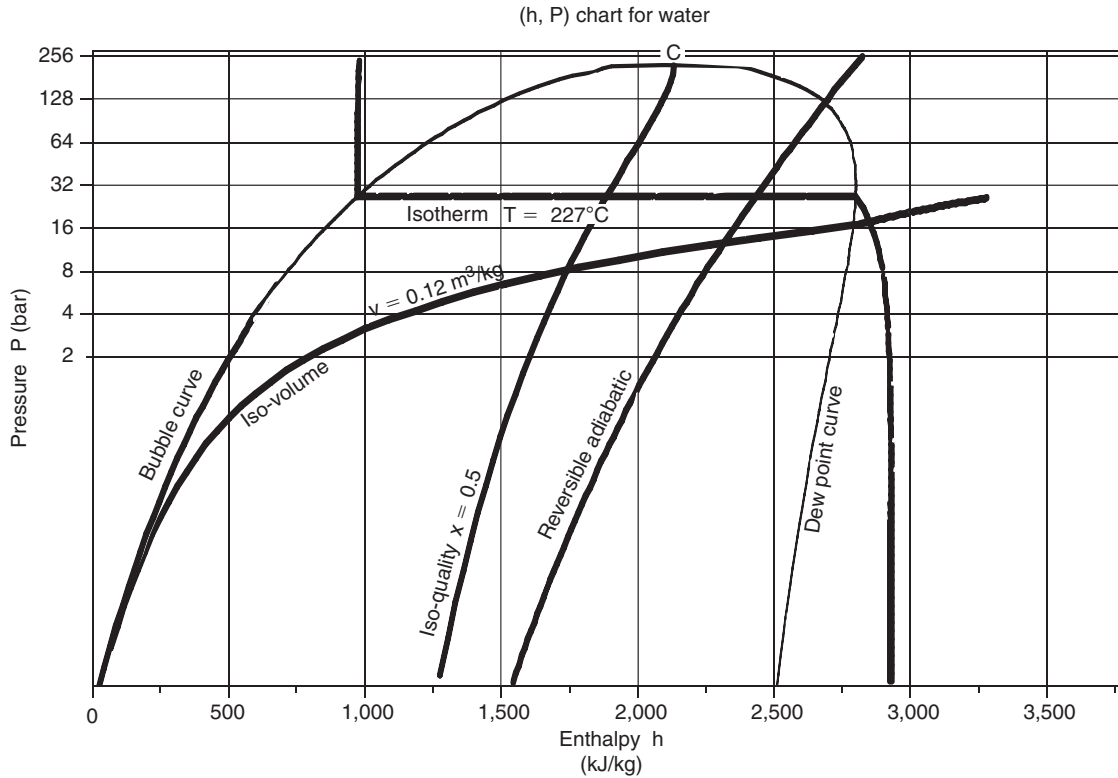
2.10.2 (h, ln(P)) chart

In the (h, ln(P)) chart (Figure 2.10.1) the enthalpy is the abscissa and pressure the ordinate, usually on a logarithmic scale.

The saturation curve separates the plane in several zones. Its summit is the critical point C, the left side, bottom, representing the incipient boiling (bubble curve), and its right side down, the saturated vapor (dew point curve). Under this curve is the two-phase liquid-vapor equilibrium zone, and in the rest of the plan, that of simple fluid.

For this chart to be used, it is equipped with reversible adiabatic curves, isotherms, isovolumes, and in the mixed zone, iso-quality curves.

In the "liquid" zone on the left of the chart, isotherms have a very strong negative slope: the compression of a liquid involves a very small work.

In the two-phase zone, pressure and temperature are related by the law of saturation pressure, and isotherms are horizontal. The enthalpy increases enormously, corresponding to the heat of vaporization that must be supplied to the fluid.

FIGURE 2.10.1

Water (h, ln(P)) chart

In the zone to the right of the saturation curve, the isotherms are curved downward, close to vertical for values of low pressure. Indeed, the behavior of vapor then approaches that of an ideal gas, whose enthalpy depends only on temperature.

Heating or cooling (isobars) are reflected in this chart by a horizontal segment, an expansion without work (isenthalpic) by a vertical segment.

A reversible adiabatic is an upward curve with a slope equal to the inverse of specific volume. They are much less inclined in the vapor zone than in the liquid zone.

Iso-quality curves are contained inside the liquid-vapor equilibrium zone. They intersect at the critical point.

Iso-volume curves converge in the liquid zone where they become independent of pressure.

2.11   PLOT OF CYCLES IN THE (h, ln(P)) CHART

We will now explain how to draw in the (h, ln(P)) chart the steam power plant and refrigerating machine cycles. We assume initially that the compressions and expansions are perfect, that is to say they follow the reversible adiabatic, then we shall explain how to take into account the irreversibilities in these machines.

In practical terms, students should do the proposed activities in this section by hand, with diagrams on paper. These can either be printed directly from Thermoptim, but their accuracy is not great, or obtained commercially or in literature.

We have not yet specified the numerical data for these cycles. We shall do it now, selecting values which can be easily plotted, and build them step by step in this chart.
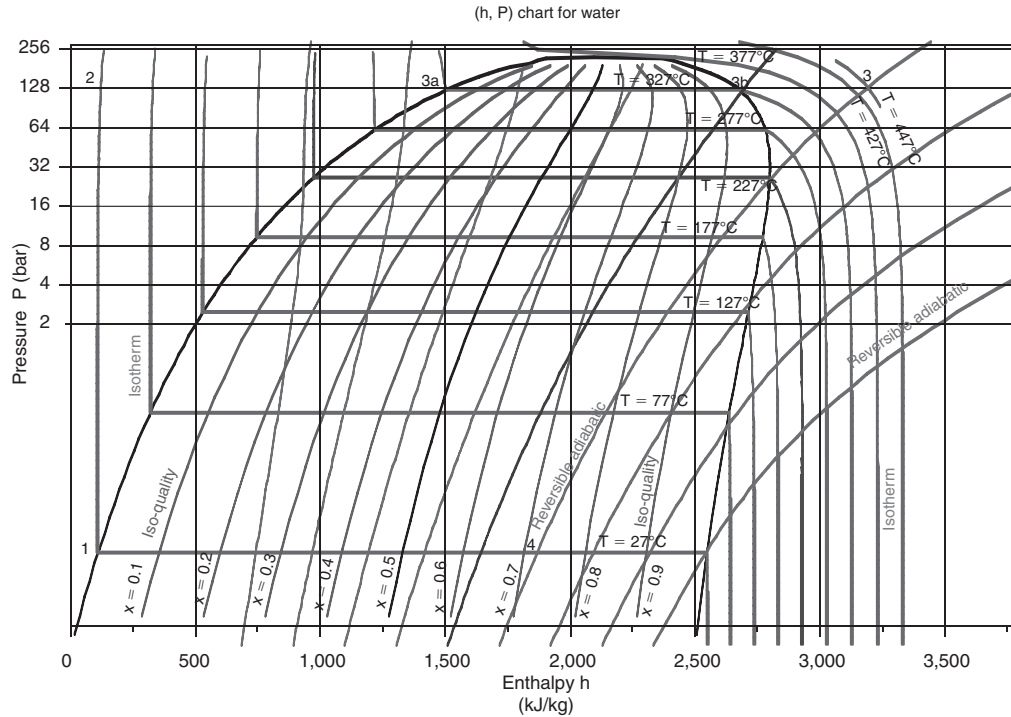
FIGURE 2.11.1
Construction of the steam power plant cycle

We will not seek to represent the gas turbine cycle for two reasons: firstly because the gas (h, ln(P)) charts are little used in practice, and secondly because to estimate the thermodynamic properties of combustion products requires special care. In our opinion it is much easier for a beginner to directly model the machine with a tool such as Thermoptim as shown section 2.12, and the plot of the cycle in a chart is not always needed.

2.11.1 Steam power plant

At the condenser outlet (point 1, Figure 2.1.1), water is in the liquid state at a temperature of about 27°C, under low pressure (0.0356 bar). In Figure 2.11.1, the point is easy to find, at the intersection of the isotherm T = 27°C and the saturation curve.

The pump compresses water at about 128 bar, which represents a significant compression ratio (around 3600).

The temperature T remaining approximately constant during compression (1-2), point 2 is located at the intersection of the isotherm T = 27°C and the isobar P = 128 bar (ordinate 128 bar).

The pressurized water is then heated at high temperature in the boiler, heating comprising the following three steps:

• liquid heating from 27°C to about 330°C boiling point at 128 bar, just above isotherm T = 327°C: process (2-3a). Point 3a lies on the vaporization curve at the same isobar;

• vaporization at constant temperature 330°C: process (3a-3b). Vaporization being carried out at constant pressure and temperature, it results on the chart in a horizontal segment 3a-3b. Point 3b is therefore on the descending branch of the vaporization curve, or dew point curve, at its intersection with the horizontal line of pressure 128 bar;

TABLE 2.11.1

ENTHALPY BALANCE OF SIMPLE STEAM CYCLE

Point	Flow-rate (kg/s)	h (kJ/kg)	
1	1	113	
2	1	126	
3a	1	1524	
3b	1	2672	
3	1	3189	
4′	1	1870	
Process	**τ (kW)**	**Q (kW)**	**Δh (kW)**
(1-2)	13		13
(2-3a)		1398	1398
(3a-3b)		1148	1148
(3b-3)		517	517
(3-4)	−1319		−1319
(4-1)		−1757	−1757
cycle	**−1306**	**1306**	
energy efficiency		42.64%	

- superheating from 330°C to 447°C, process (3b-3). Point 3 is still assumed to be the same pressure, but at a temperature T_3 of 447°C. It is thus at the intersection of the horizontal P = 128 bar and the isotherm T = 447°C (only partially shown in the figure).

Point 3 is also on an inclined downwards concave curve corresponding to a reversible adiabatic.

Process (3-4) is a reversible adiabatic expansion from 128 bar to 0.0356 bar. The point being in the mixed zone, the latter is within isotherm T = 27°C. Point 4 is at the intersection of the inclined downwards concave curve and this isotherm. Quality x is between 0.7 and 0.8. Linear interpolation allows us to estimate its value, equal to 0.72.

The enthalpies of the points can be read directly by projecting these points on the x-axis, and the energies involved can be easily deducted. This allows us in particular to determine the cycle efficiency. Table 2.11.1 provides these values.

Note that reading the compression work in this chart is very imprecise and it is best estimated from the integration of $\delta\tau = v\,dP$, very easy to make, v being constant. It is therefore $\Delta h = v\,\Delta P$, P being expressed in Pa, v in m^3/kg and h in J.

As shown in this example, the representation of the cycle in the (h, ln(P)) chart is very easy to understand: the heat exchanges, almost isobaric, correspond to horizontal segments, and the compression and expansion are reversible adiabatic, less steep as they are located far from the liquid zone. Figure 2.11.2 shows the cycle in the chart, the points being connected.

The efficiency here is the ratio of the mechanical work produced to the heat supplied by the boiler.

In reality, the turbine is not perfect, and expansion follows an irreversible adiabatic. It is customary to characterize the actual process by its isentropic efficiency η defined in the case of a turbine as the ratio of actual work to the reversible expansion work. Its value is typically about 0.85 to 0.9 in current applications.

To find the real point 4′, we first determine the value of the perfect machine work τ_s, then we multiply it by η, which gives the value of real work τ.

The enthalpy of point 4 is equal to that of point 3 minus τ.
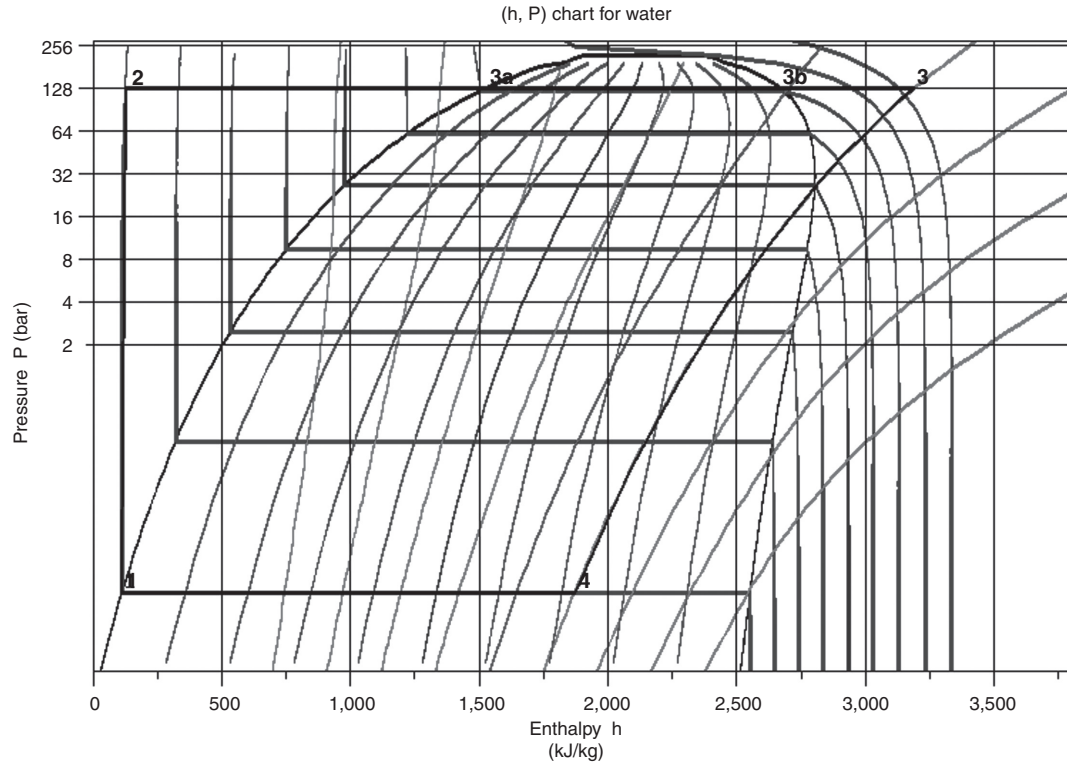
FIGURE 2.11.2

Steam power plant cycle in the (h, ln(P)) chart

In this case, taking $\eta = 0.9$, we have:

$$\tau_s = h_3 - h_4 = 1319 \, \text{kJ/kg/K}$$

$$\tau = \eta \, \tau_s = 0.9 \times 1319 = 1187$$

$$h_{4'} = 3199 - 1187 = 2002 \, \text{kJ/kg/K}$$

Point 4′, which is still in the liquid-vapor equilibrium zone, is thus on isobar P = 0.0356 bar, i.e. on isotherm T = 27°C with abscissa h = 2002 kJ/kg/K. Figure 2.11.3 shows the new look of the cycle (the previous cycle is plotted in dashed line).

Here we have not considered the pressure drops in heat exchangers. It would of course be possible to do so by operating on the same principle as described, and changing the pressure between the inlets and outlets of heat exchangers.

2.11.2 Refrigeration machine

The R134a compression refrigeration cycle operates between a suction pressure of 1.78 bar and a condenser pressure of 12 bar.

At the evaporator outlet , the fluid is fully vaporized and thus point 1 (see Figure 2.11.4) is located at the intersection of the saturation curve and the isobar P = 1.78 bar, or, equivalently, the isotherm T = −13°C.

It is then compressed at 12 bar along a reversible adiabatic. Point 1 being located approximately one third of the distance between two reversible adiabats on the chart, it is possible, by linear interpolation, to determine point 2 on isobar P = 12 bar, between these two curves.
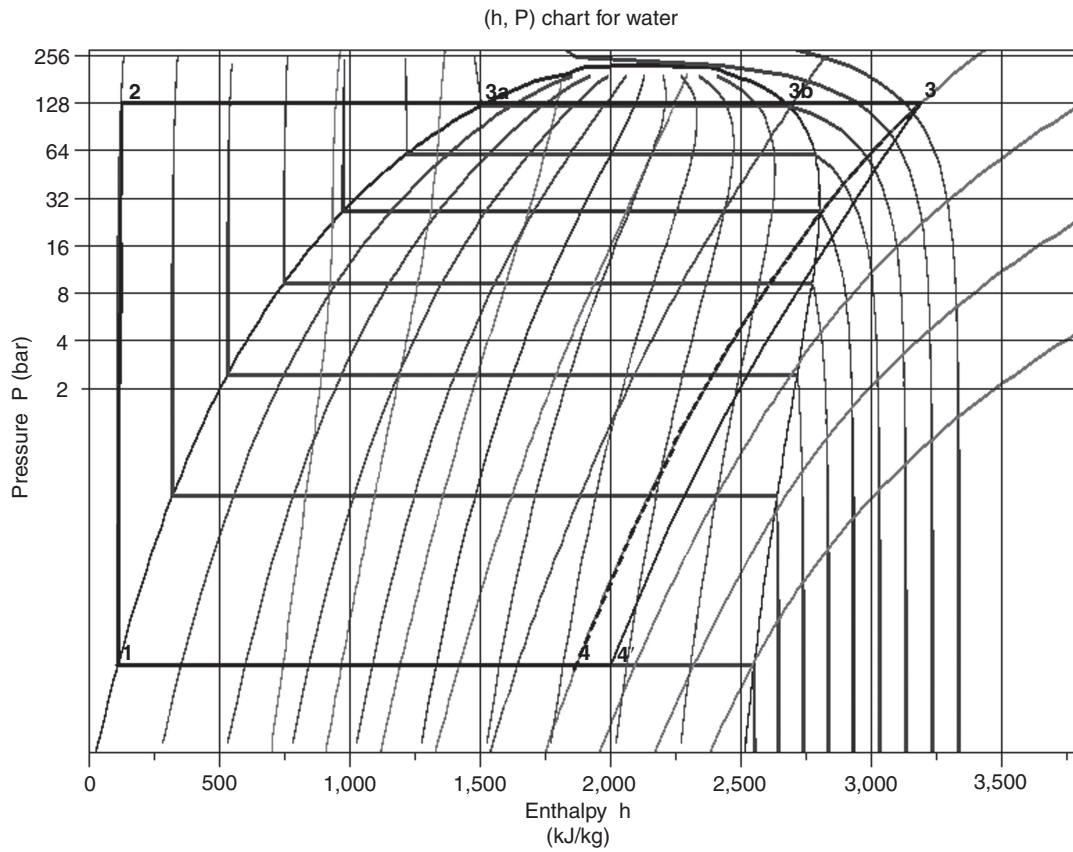
FIGURE 2.11.3

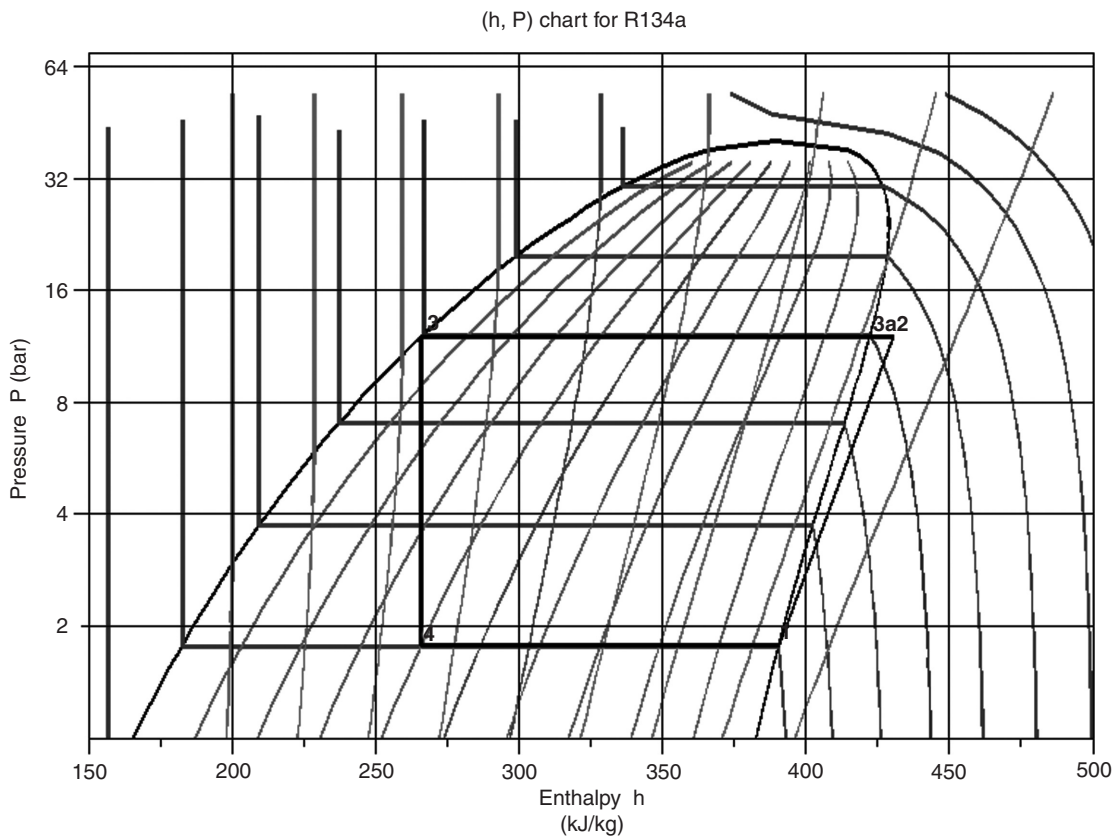Steam power plant cycle with irreversible expansion

FIGURE 2.11.4

Refrigeration machine simplified cycle

TABLE 2.11.2
SIMPLE REFRIGERATION CYCLE ENTHALPY BALANCE

Point	Flow rate (kg/s)	h (kJ/kg)	
1	1.00	391	
2	1.00	430	
3a	1.00	422	
3	1.00	266	
4	1.00	266	
Process	**τ (kW)**	**Q (kW)**	**Δh (kW)**
(1-2)	40		40
(2-3a)		−8	−8
(3a-3)		−157	−157
(3-4)			
(4-1)		**125**	125
cycle	**40**	−40	
Coefficient of Performance		3.14	

The refrigerant cooling in the condenser by exchange with outside air comprises two stages: de-superheating (2-3a) in the vapor zone followed by condensation along the horizontal line segment (3a-3). Points 3a and 3 lie at the intersection of the saturation curve and the isobar P = 12 bar, or, equivalently, the isotherm T = 47°C. Point 3a is located on the right, at the limit of the vapor zone, and point 3 on the left, at the limit of the liquid zone.

Expansion without work, and therefore an isenthalpic process, corresponds to the vertical segment (3-4), point 4 being located on the isobar P = 1.78 bar, or, equivalently, the isotherm T = − 13°C, at abscissa h = h_3. Its quality reads directly from the corresponding iso-quality: it is x = 0.4.

The energies involved can easily be determined by projecting these points on the x-axis. This allows us in particular to calculate the coefficient of performance COP of the cycle, defined as the ratio of useful effect (the heat extracted at the evaporator) to the purchased energy (here the compressor work). Table 2.11.2 provides these values.

This cycle differs from that of a real machine on several points:
- the actual compression is not perfect, so that the compression work is higher than that which would lead to reversible adiabatic;
- to prevent aspiration of liquid into the compressor, which could deteriorate it as the liquid is incompressible, the gas is superheated by a few degrees (typically 5 K) above the saturation temperature before entering the compressor;
- before entering the valve, the liquid is sub-cooled by a few degrees (typically 5 K), this first ensures that this device is not supplied with vapor, and second increases the refrigerator performance.

Here we can also characterize the actual compression by an isentropic efficiency, defined this time as the ratio of the work of the reversible compression to real work. Its value is typically around 0.7 to 0.85 in current applications.

To find the real point 2′, we determine the value of work τ_s in the perfect machine, then divide it by η, which gives the value of real work τ.
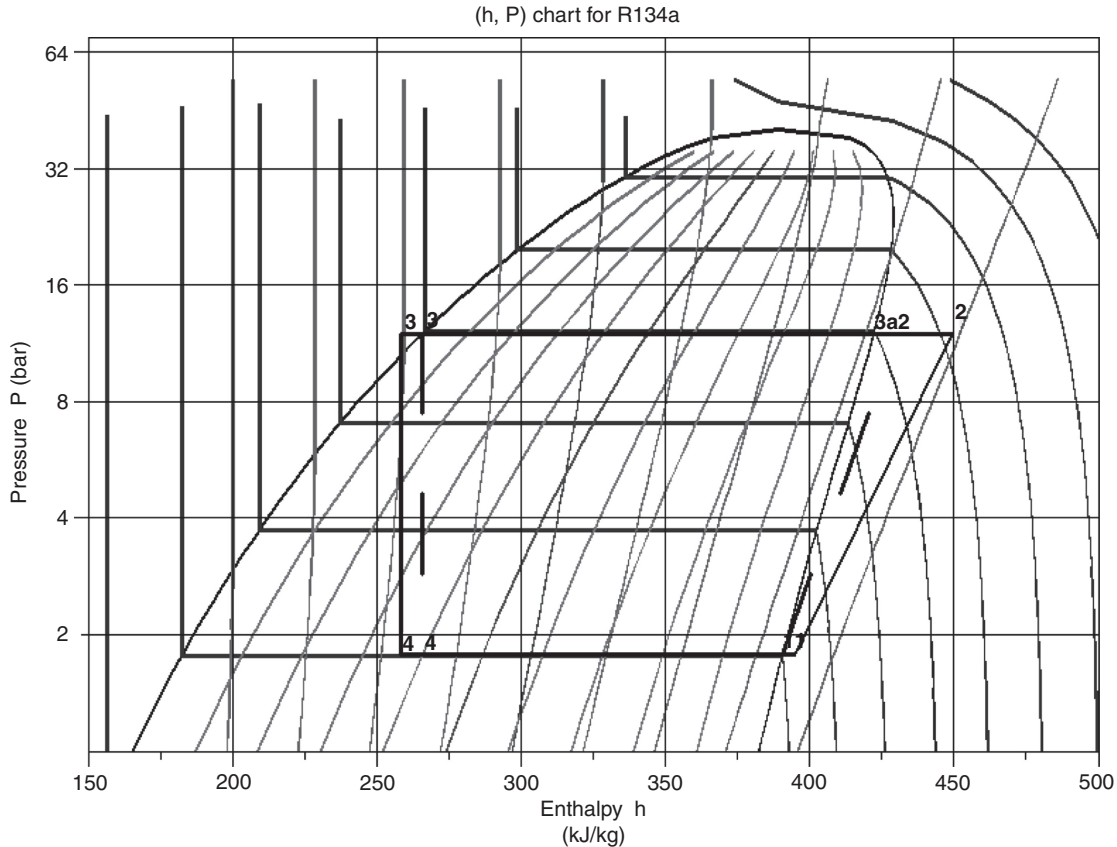
The enthalpy of point 2′ is equal to that of point 1 plus τ.

In this case, taking η = 0.75, we have:

$$\tau_s = h_2 − h_1 = 40 \, \text{kJ/kg/K}$$

$$\tau = \tau_s/\eta = 40/0.75 = 53$$

$$h_{2'} = 391 + 53 = 444 \, \text{kJ/kg/K}$$

FIGURE 2.11.5
Refrigeration machine with superheating and sub-cooling

The cycle changed to reflect superheating, subcooling and compressor irreversibilities is shown in Figure 2.11.5 (the previous cycle is plotted in dashed line).

The COP of the machine is of course changed: it drops to 2.57 because of the compressor irreversibilities.

The relevance of sub-cooling can easily be shown in the (h, ln (P)) chart because, for the same compression work, the effectiveness increases as the magnitude of the sub-cooling grows. It is however limited by the need for a coolant.

We did not consider the pressure drops in the exchangers. It would of course be possible to do so.

2.12   MODELING CYCLES WITH THERMOPTIM

At this point, all the notions allowing us to model these cycles with Thermoptim have been presented, and examples in Chapter 9 of Part 2 can be treated by changing the settings, which have been chosen slightly differently so that they can be easily hand-drawn in charts. Before you start entering models in Thermoptim, we recommend you to study Diapason initiation session S07En_init which introduces all the software concepts needed. The synoptic views that are obtained are given in Figures 2.12.1 to 2.12.3.
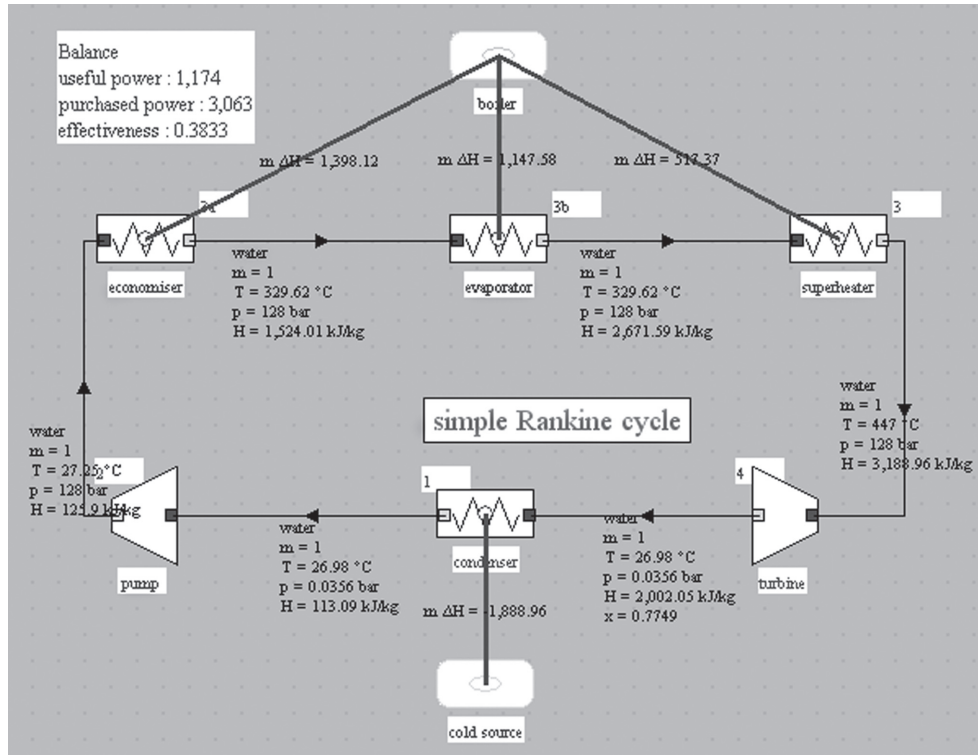
[CRC_In_9]

FIGURE 2.12.1

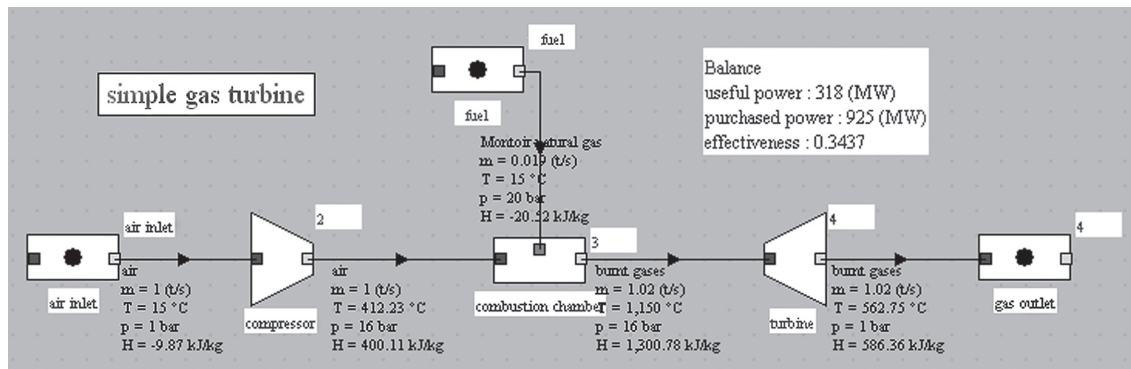Synoptic view of the steam power plant

FIGURE 2.12.2

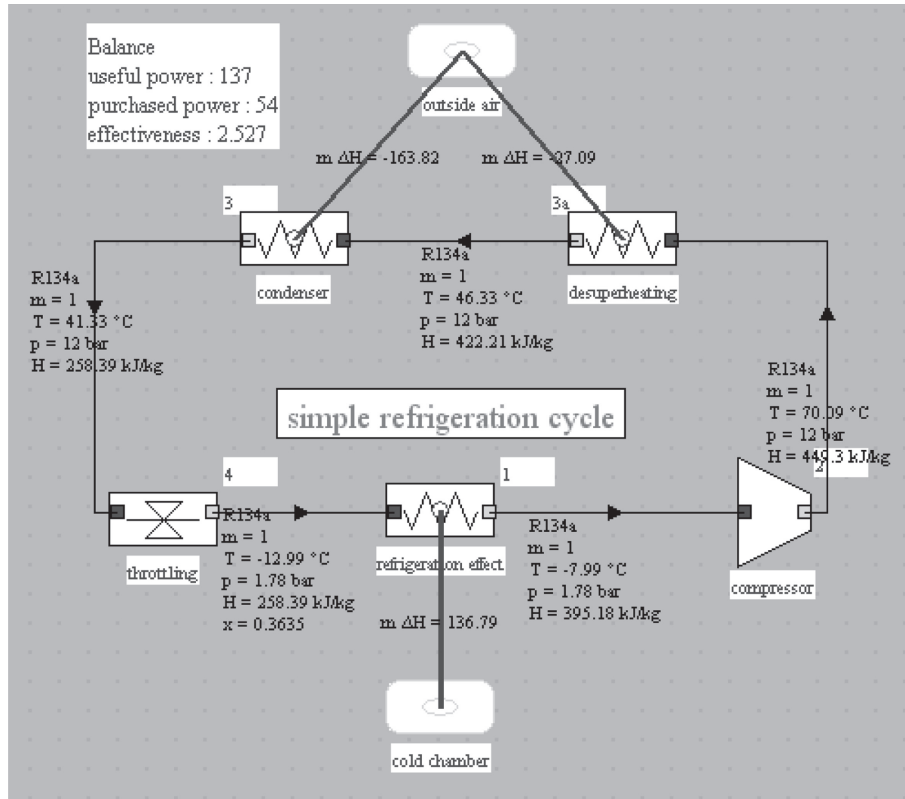Synoptic view of the gas turbine

2.12.1 Steam power plant

WORKED EXAMPLE

Modeling of a steam power plant with Thermoptim

The basic cycle of a steam power plant is essentially a boiler where the fuel is burned for generating steam (usually superheated) which is then expanded in a steam turbine whose shaft provides the motor work. The expanded steam is subsequently condensed and compressed by a pump before returning to the boiler.

The modeling of this cycle in Thermoptim is presented step by step in sections 9.1.1 to 9.1.5 of Part 2, with slightly different settings.

THDPT

[CRC_we_1]

Balance
useful power : 137
purchased power : 54
effectiveness : 2.527

outside air

m ΔH = -163.82   m ΔH = 27.09

3   3a

condenser   R134a   desuperheating
                   m = 1
                   T = 46.33 °C
                   p = 12 bar
R134a   H = 422.21 kJ/kg
m = 1
T = 41.33 °C
p = 12 bar
H = 258.39 kJ/kg   R134a
                                             m = 1
              simple refrigeration cycle   T = 70.09 °C
                                             p = 12 bar
                                             H = 449.3 kJ/kg
4   1   2

throttling   R134a   refrigeration effect   R134a   compressor
             m = 1   m = 1
             T = -12.99 °C   T = -7.99 °C
             p = 1.78 bar   p = 1.78 bar
             H = 258.39 kJ/kg   H = 395.18 kJ/kg
             x = 0.3635

                    m ΔH = 136.79

                     cold chamber

FIGURE 2.12.3
Synoptic view of the refrigeration machine with superheating and sub-cooling

2.12.2   Gas turbine

WORKED EXAMPLE
Modeling of a gas turbine with Thermoptim

A gas turbine is composed of three elements: a compressor, a combustion chamber and a turbine, in which are expanded the high temperature gases.

The setup that we will retain is: a gas turbine burning natural gas without dissociation sucks 1 t/s of air at 15°C and 1 bar, compresses it at 16 bar in a 0.85 isentropic efficiency compressor, and expands the burnt gases in a 0.85 isentropic efficiency turbine. The temperature of the gases at the turbine inlet is 1150°C.

The modeling of this cycle in Thermoptim is presented step by step in sections 9.3 of Part 2.

THDPT

[CRC_we_2]

2.12.3 Refrigeration machine

WORKED EXAMPLE
Modeling of a refrigeration machine with Thermoptim

In a refrigeration machine a refrigerant is evaporated at low pressure, then compressed at a pressure such that it is then possible to cool it by heat exchange with ambient air until it becomes liquid. The liquid is then expanded by isenthalpic throttling at low pressure and directed into the evaporator.

The modeling of this cycle in Thermoptim is presented step by step in sections 9.2 of Part 2, with slightly different settings.

THDPT

[CRC_we_3]

2.13   CONCLUSION

The essentially qualitative and phenomenological approach used in this presentation proves that it is possible to introduce thermal machine cycles to students with a limited number of thermodynamic concepts, especially not including entropy.

To sum up, the foundations of a lightweight educational presentation of thermodynamic cycles (Figure 2.13.1) appear to be the following (we have indicated in parentheses sections of Part 2 of
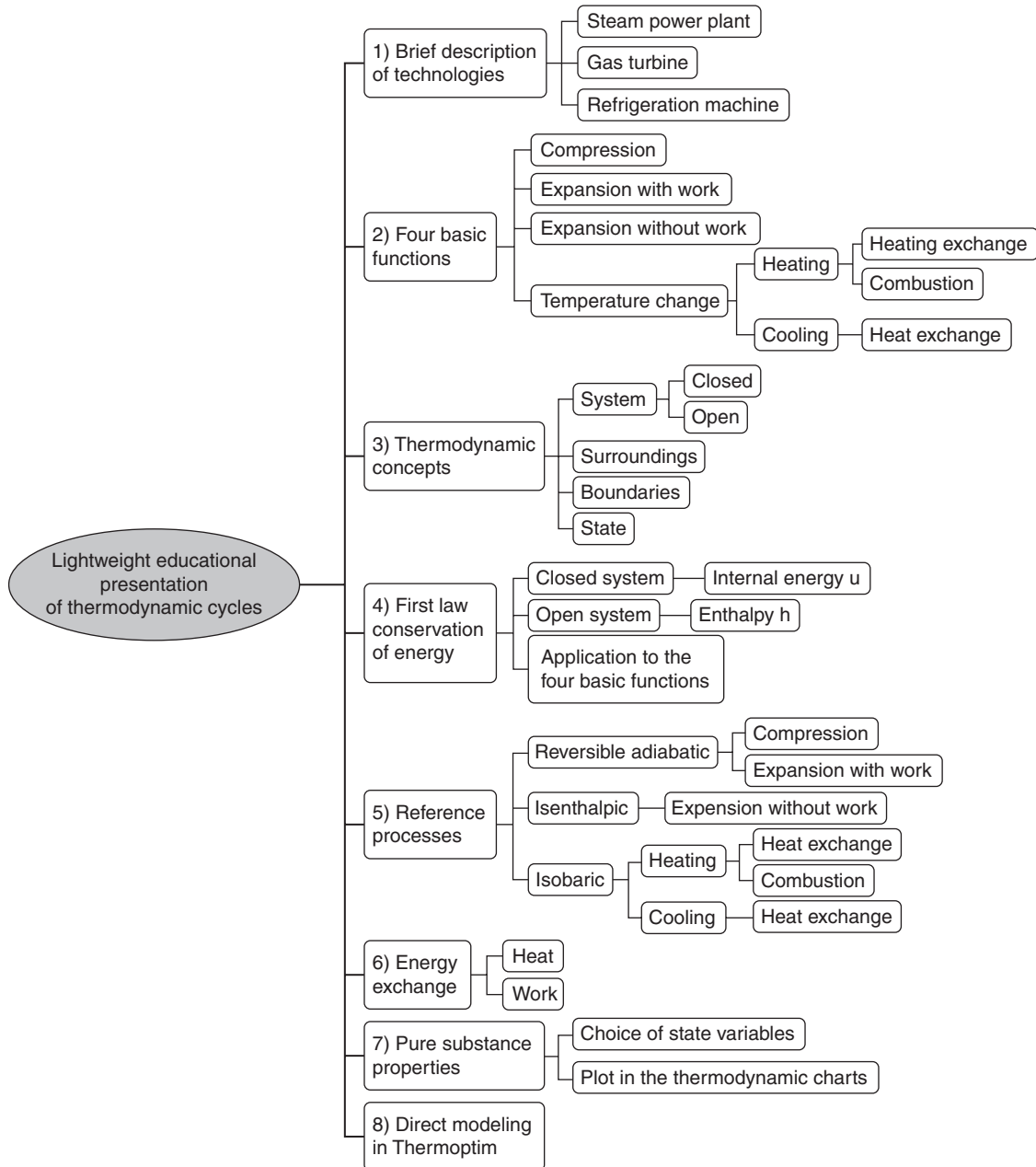

FIGURE 2.13.1

Map of the lightweight educational presentation milestones Part 1

the book where these points are developed in order to facilitate a further deepening of concepts discussed):

- Brief description of technologies, with their architecture (sections 9.1.1, 9.2.1 and 9.3.1);
- Highlighting the four basic functions: compression, expansion with work production, expansion without work production, heat exchange (section 5.3.6);
- Introduction of the concepts of thermodynamic system and state, distinction between closed systems and open systems (sections 5.1.1, 5.1.2);
- The thermodynamic systems we are interested in only exchange energy with their surroundings in two distinct forms: heat and work (Section 5.2);
- The fundamental law that governs the behavior of thermodynamic systems is that of conservation of energy, known as the first law (sections 5.3.1 to 5.3.4);
- Its application to the four basic processes shows that the determination of the enthalpy change of the fluid flowing through them is enough to calculate the energy they bring into play, whether work or heat (section 5.3.6);
- Brief reminders on pure substance properties and their graphic representation in thermodynamic charts (sections 5.6.1 and 5.6.6.1);
- Highlighting reference processes relevant to these functions for perfect machines (sections 7.1.2, 7.5, 7.7);
- Reflecting on the choice of state variables most appropriate for students with a light background in mathematics and physics, showing the educational potential of the (h, ln(P)) chart;
- Representation of the steam power plant and refrigeration machine cycles in these charts;
- Direct modeling of the cycles in Thermoptim (sections 9.1, 9.2 and 9.3).

All these milestones, with the exception of the two penultimate ones, are valid whether or not we use entropy.

It is not for us to minimize the relevance of this state function: we consider only that it may be preferable not to introduce it at the beginning of training when students do not have the required culture. If they do, we of course have no objection to use it, quite the contrary.

Without speaking of entropy, we insist instead on the adiabatic character of compressors and turbines, and the importance of the reversible adiabatic process as a reference for these machines. Without needing to say its name, the value of the entropy for cycle studies underlies therefore our approach, and its formal introduction at a later stage (e.g. in the second part of this presentation) is facilitated when the students have become familiar with all the new concepts presented here.

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First Steps in Thermodynamics: Entropy and the Second Law

Abstract: The purpose of this chapter is to discuss key concepts which should be presented to absolute beginners in order to enable them to understand and study the cycles of three basic energy technologies: steam power plant, gas turbine and refrigerating machine.

We are talking of lightweight educational presentation as we seek to minimize the background in mathematics and physics necessary for understanding these cycles, our goal being to make them accessible to readers unfamiliar with the language of specialists in thermodynamics.

In the first part of this presentation (Chapter 2), we introduced the basic concepts of thermodynamics without resorting to the notion of entropy, which can be difficult to master for some students. In this second part, we introduce it as simply as possible, and discuss the main implications of the second law of thermodynamics.

Keywords: lightweight educational presentation, steam power plant, gas turbine and refrigerating machine, heat exchange, enthalpy, entropy, second law of thermodynamics.

3.1 HEAT IN THERMODYNAMIC SYSTEMS

In the first part, we indicated that the heat exchanged by a system with the surroundings is, for an infinitesimal process, given by the following expression:

$$\delta Q = Cp\, dT - v\, dP \qquad\qquad 3.1.1$$

It expresses an experimental fact, the essential basis of the thermodynamics of compressible fluids: heat δQ exchanged with the surroundings is a linear function of the thermodynamic state of the system.

This equation is however valid only if there is no irreversibility inside or at the boundary of the fluid mass. If any, the relationship becomes:

$$\delta Q < Cp\, dT - v\, dP \qquad\qquad 3.1.2$$

We can then write:

$$\delta Q = Cp\, dT - v\, dP - \delta \pi \qquad\qquad 3.1.3$$

$\delta\pi$, essentially a positive term, has a very simple physical meaning: it is the heat generated by mechanical friction within the fluid. A straightforward interpretation for isobaric heating is that the fluid temperature rise dT is greater than $\delta Q/Cp$ because of irreversibilities.

Although it differs profoundly from heat received from the surroundings, it changes the thermodynamic state of the system in the same way.

δQ is the **heat exchanged with the surroundings**, counted positively if it is received by the system and negatively otherwise, and $\delta\pi$ the **heat dissipated by internal friction and shocks** if any. It is always positive or zero.

In practice, it is important to distinguish these two forms of heat, otherwise serious errors of reasoning can be made. In particular, processes without heat exchange with the surroundings, called adiabatic, are such that $\delta Q = 0$, whether the seat of irreversibility or not, that is to say whether $\delta\pi$ is zero or not.

3.2   INTRODUCTION OF ENTROPY

In the first part, we stressed how important in practice are **reference processes** corresponding to the processes that fluids would undergo in perfect machines.

We have also shown that, for compressors and turbines, the reference process is the reversible adiabatic, the corresponding law for a perfect gas being given by $Pv^\gamma = $ Const. with $\gamma = Cp/Cv$.

It is obtained by solving the differential equation $\delta Q = 0 = Cp\,dT - v\,dP$, which is quite simple replacing v by rT/P.

Given the importance of this law, it is interesting to try to generalize it to find a formulation that is valid on the one hand for all fluids and not only perfect gases, and secondly for all the processes, whether or not they bring into play irreversibilities.

Let us come back to the equation:

$$\delta Q = Cp\,dT - v\,dP - \delta\pi \qquad\qquad 3.2.1$$

which is also written:

$$\delta Q + \delta\pi = Cp\,dT - v\,dP \qquad\qquad 3.2.2$$

This equation is an inexact differential and not an exact differential. We call an integrating factor an expression by which one multiplies an inexact differential in order to turn it into an exact differential.

We can show that in the general case as for a perfect gas, 1/T is one of the simplest integrating factors for this equation. For the perfect gas, this integral expression is written:

$$s = s_0 + c_p \ln\frac{T}{T_0} - r\ln\frac{P}{P_0} \qquad\qquad 3.2.3$$

As can be seen, function s is a state function formally very close to the calorimetric equation providing heat exchanged by a system with its surroundings, as it can be deduced by using as integrating factor $I = 1/T$. It is called the **entropy of the system**.

In the (h, ln(P)) charts that we presented in the first part, the iso-value curves of the reversible adiabatic are those for which ds = 0. They are called **isentropic**.

The concept of entropy is thus introduced naturally, entropy being the state function closest to heat. There is no evidence in this approach, however, that this concept remains valid for non-perfect gases. With the more axiomatic definition we will give by introducing the second law, a perfect gas becomes a special case of the general theory.

3.3 SECOND LAW OF THERMODYNAMICS

3.3.1 Limits of the first law of thermodynamics

A major limitation of the first law of thermodynamics is the failure to take into account the quality of energy: indeed different forms of energy expressed in kWh are equivalent, but the possibilities of converting one form of energy into another are not.

Thus, work can still be fully converted into heat, but the converse is not true. The work is one form of energy whose quality is among the best, and which can therefore be taken as a reference.

We can rephrase this by saying that a possible indicator of the energy quality is its **ability to be converted into work**.

The first law postulates the equivalence of different forms of energy, but it does not take into account an essential experimental fact, which is that when a system interacts with its surroundings, the energy processes that it undergoes can only take place in a privileged sense, that cannot be reversed without a qualitative change in the system.

3.3.2 Concept of irreversibility

One can convert electricity into work using an electric motor of efficiency over 98%, or conversely convert mechanical work into electricity through a generator of equivalent efficiency, which means that these two forms of energy are about the same quality.

In the example of power conversion work we have just given, we stated that the machines used had excellent efficiencies, close to but slightly lower than 1: experience indeed shows that, whatever precautions are taken, some energy is degraded. The first law teaches us that the total energy is conserved, but some of its quality declines and ultimately ends up as heat, because of friction, Joule losses etc.

These losses are called irreversibilities, because the process work → electricity → work is not completely reversible: part of the initial work is converted into heat.

Irreversibilities encountered in energy facilities that interest us, except those taking place in combustion reactions, can be grouped into two broad classes that we discuss briefly a little later:
- irreversibilities stemming from temperature heterogeneity;
- mechanical irreversibilities due to viscosity.

3.3.3 Heat transfer inside an isolated system, conversion of heat into work

In the particular case of heat, it is always experimentally verified without exception, that the transfer of heat between two media at different temperatures is from the warmer body (the one whose temperature is higher), to the colder (whose temperature is lower).

Also a heterogeneous system composed of two media at different temperatures not isolated from each other always evolves towards a homogeneous state at intermediate temperature.

Moreover, when attempting to convert heat into work, experience proves that it is first necessary to have two heat sources, one at high temperature, and the other at low temperature (Figure 3.3.1).

Moreover, the larger the temperature difference between the sources, the greater the amount of work that can be converted from one kWh of heat. Temperature thus also appears as a possible indicator of the heat quality.

A further remark can be made at this stage: it is the existence of a temperature difference between two bodies that allows for work production.
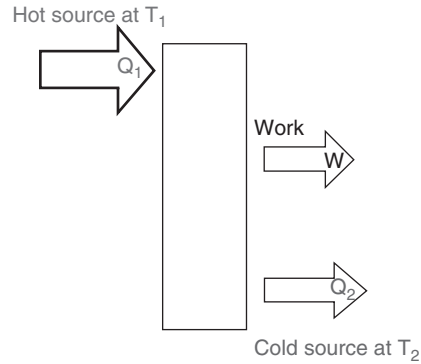
FIGURE 3.3.1
Conversion of heat into work

3.3.4 Statement of the second law

The second law complements the first by introducing a function called entropy, which is used to quantitatively characterize the effects of irreversibilities taking place in a system and explain the phenomena we have just discussed.

A rigorous and comprehensive presentation of the second law requires significant developments due to the precautions that must be taken first in writing assumptions and explanations of their connection with experience, and second in demonstrations which must be made.

Given our objectives, this is not justified here, especially as for applied thermodynamic calculations of practical interest, the main advantage of this second law can be summarized in two points:

• first, entropy is, as we have shown, the state function most closely related to the heat Q exchanged with the surroundings. It thus intervenes implicitly or explicitly in many equations governing the operation of energy components;

• second, the generation of entropy alone allows us to quantify all the irreversibilities taking place in these components and their boundaries, which is fundamental.

The second law states that entropy s has the following properties:

• s is a function of the system state variables;

• in any elementary process involving heat exchange δQ with the surroundings we have $\delta Q \leq T\,ds$, equality being satisfied if and only if the process is perfect (not irreversible).

Writing:

$$ds = \frac{\delta Q}{T} + d_i s \qquad\qquad 3.3.1$$

$d_i s$, positive or zero, is called "entropy generation".

This relationship can also be written, as we mentioned above:

$$\delta Q + \delta \pi = T\,ds \qquad\qquad 3.3.2$$

$\delta \pi$ is called "uncompensated work" or "uncompensated heat". It is positive for an irreversible process and zero otherwise.

3.4 (T, s) ENTROPY CHART

In the first part of this lightweight presentation, we excluded entropy from variables we have considered to represent the fluid properties. Now that we have introduced this state function, it is natural to question its relevance in this regard.
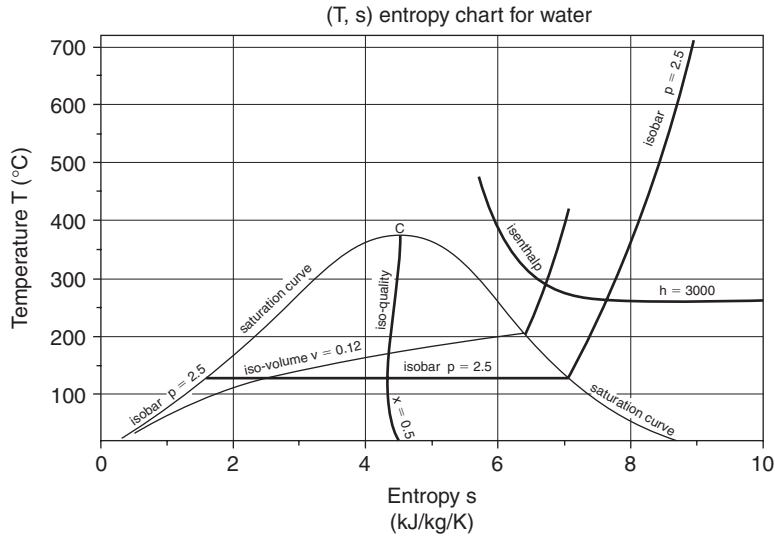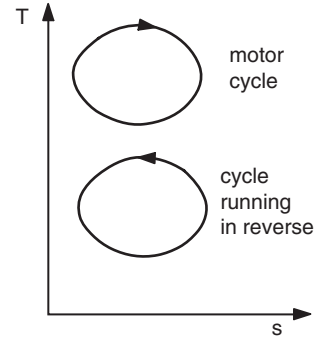
FIGURE 3.4.1
Water (T, s) entropy chart

FIGURE 3.4.2
Types of cycles in the entropy chart

Experience shows that the entropy and Mollier charts, which use as coordinates pairs (T, s) and (h, s), are particularly interesting, especially the first, to which we limit our analysis here.

In the entropy chart (Figure 3.4.1) entropy is the abscissa and temperature the ordinate. The vaporization curve again separates the plane into two areas, defining the two-phase zone and the zone of simple fluid. The critical point C is still at its maximum.

One of the advantages of such a chart (see Figure 3.4.2) is that any perfect cyclic process is reflected in the plane (T, s) by a contour (Γ), whose area A measures, depending on the sign, either the amount of heat Q brought into play, or the work provided or received τ.

Indeed, $Q = \int T\,ds$ by definition. As the cycle is closed, $Q + \tau = 0$, and $|\tau| = A$.

The rule of signs is as follows:

- if the cycle is described clockwise, work is negative, thus transferred by the fluid to the surroundings: we talk of a motor cycle or of a cycle running forward;
- if the cycle is described counterclockwise it is the opposite: we talk of a refrigeration cycle, or of a cycle running in reverse.

Form of isobars

To the left of the incipient boiling curve, liquid isobars are curved upward. Liquid isentropic compression having almost no effect on the temperature, liquid isobars are virtually merged with the ascending branch of the vaporization curve. The chart is very imprecise in this area and it is preferable to use a table or a program giving the thermodynamic properties along the vaporization curve.

Inside the two-phase zone, temperature and pressure are bound by the saturated pressure law, and isobars are horizontal.

Right of the vaporization curve, isobars are rising and, for an ideal gas, become exponential. They can be deduced from each other by horizontal translation.

If the pressure exceeds the critical pressure, isobars are strictly ascending curves, which do not intersect the vaporization curve.

Form of isenthalps

In the vicinity of the saturation curve, isenthalps are curved downward, with a strong negative slope.

Gradually as the distance from the vaporization curve increases, the gas approaches the corresponding ideal gas, and isenthalps become horizontal, as enthalpy is a sole function of temperature.

Of course, the plot of an isentropic process is very simple in this chart: it is a vertical segment.

3.5 CARNOT EFFECTIVENESS OF HEAT ENGINES

Motor thermal machines are designed to transform heat into mechanical energy. Let us assume that heat to transform is provided by an external source, said heat source, whose temperature T_1 is set. A second source, known as cold, is required to evacuate part of the heat. Its temperature T_2 is necessarily lower than T_1.

The second law is due to S. Carnot, who in 1824 showed that the efficiency of an ideal heat engine cycle is given by: $\eta = 1 - T_2/T_1$.

Carnot has shown that this efficiency does not depend on the nature of the machine and fluids used, but only T_1 and T_2.

In the (T, s) entropy chart, it is represented by the rectangle ABCD, described clockwise (Figure 3.5.1).

- AB is a segment of isotherm T_1 described from left to right. The fluid receives heat (isothermal expansion in the case of a perfect gas);
- CD is a segment of isotherm T_2 described from right to left. The fluid releases heat (isothermal compression in the case of a perfect gas);
- BC is a segment of isentropic S_2 described from top to bottom (reversible adiabatic expansion in all cases);
- DA is a segment of isentropic S_1 described from bottom to top (reversible adiabatic compression in all cases).

This relation is probably rightly one of the best known in thermodynamics. It is of great practical importance, given its implications:

- First, with the assumptions made (reverse cycle without friction and heat transfer without temperature differences), it is easy to show that this effectiveness is the highest that can be achieved by a simple fluid heat engine operating between the two sources at T_1 and T_2. It is therefore a maximum limit and real cycles generally have much lower efficiencies;
- Secondly, the Carnot effectiveness of all reversible machines operating between the two sources at T_1 and T_2 is the same and only depends on their temperatures, not on the working fluid used.
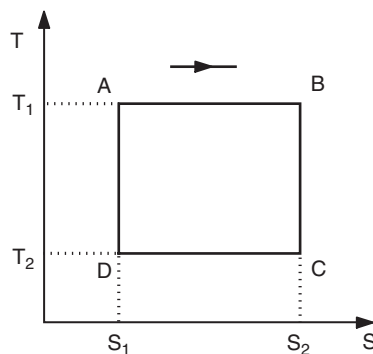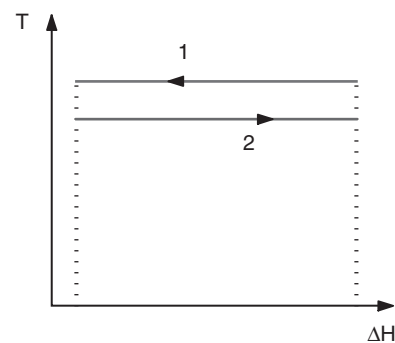
FIGURE 3.5.1

Carnot cycle

FIGURE 3.6.1

Constant temperature heat exchanger

3.6   IRREVERSIBILITIES IN INDUSTRIAL PROCESSES

3.6.1   Heat exchangers

In a heat exchanger, heat can only be transferred between two streams if a certain temperature difference exists between them. Indeed, for both technical and economic reasons, the exchange surfaces between these fluids are necessarily finite. It is then possible to show that the heat exchange is accompanied by an increase in entropy at the boundary between the two streams (at temperatures T_1 and T_2, Figure 3.6.1), given by:

$$d_i s = \left(\frac{T_1 - T_2}{T_1 T_2} \right) \delta Q$$

δQ being the heat absorbed by fluid 2. These irreversibilities are often described as external, because they take place at the boundary of the system.

The larger the temperature difference between the fluids, the greater the entropy creation. In a heat exchanger, the energy (enthalpy) transferred by the hot fluid to the cold fluid is conserved, but because the transfer is taking place with temperature decrease, entropy increases.

In energy systems, the temperature difference that must exist between two fluids that exchange heat generally constitutes an important source of irreversibility.

3.6.2   Compressors and turbines

We stated above that the compression and expansion devices are generally adiabatic ($\delta Q = 0$).

In an adiabatic reversible compressor or turbine, there is no creation of entropy. We say that evolution is **isentropic**, a very important concept in practice because it involves the reference process against which actual changes are characterized.

In real machines, as we saw in the first part of this presentation, we can take into account irreversibilities by introducing what we called an isentropic efficiency, equal to the ratio of isentropic work to real work for compressors, and the reverse for turbines, i.e. the ratio of actual work to isentropic work.

3.7   PLOT OF CYCLES IN THE ENTROPY CHART, QUALITATIVE COMPARISON WITH THE CARNOT CYCLE

The Carnot cycle being the one that leads to the best effectiveness, it is almost always interesting to compare actual cycles to it. In this section, we perform such a comparison for each of the cycles that we studied in the first part of this presentation.

3.7.1   Steam power plant

In the entropy chart (Figure 3.7.1), to increase readability, we have not shown iso-volumes. Points 1 and 2 showing compression in the liquid state are almost superimposed, and heating in the liquid state almost coincides with the liquid saturation curve. Vaporization is done in a horizontal line segment.

The isobaric superheating (3) is the maximum peak of the cycle and the irreversible expansion is reflected by an increase of entropy, the point 4 being located within the vapor-liquid equilibrium zone (quality equal to 0.775).
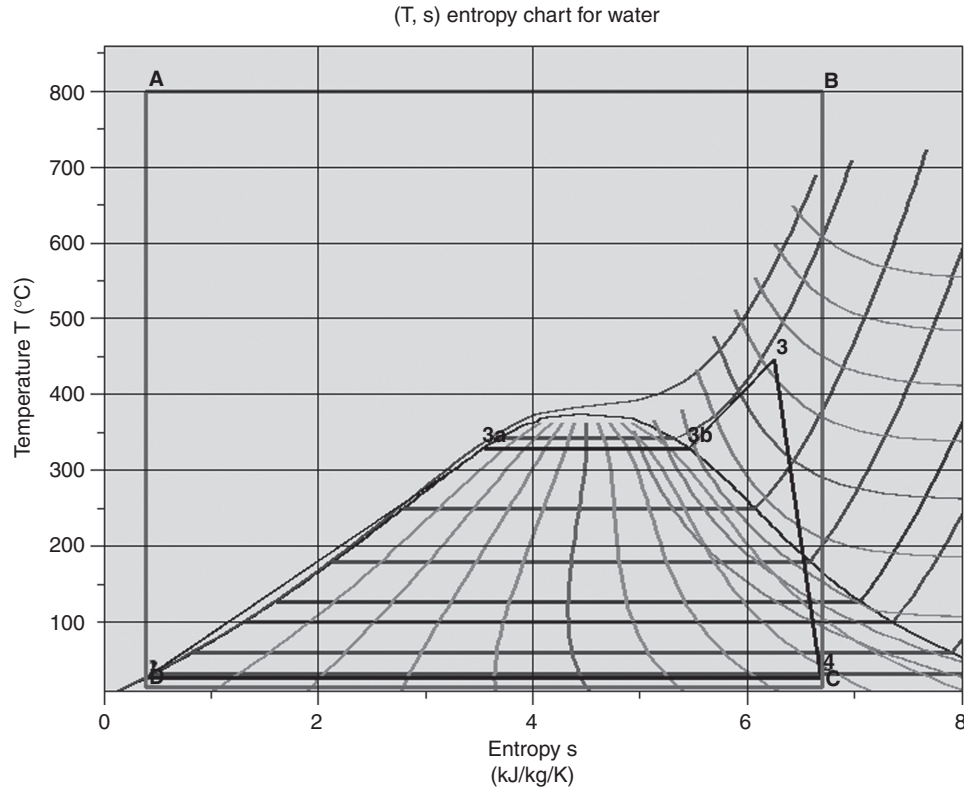
FIGURE 3.7.1
Steam power plant and Carnot cycles in the entropy chart

In order to plot the Carnot cycle (A-B-C-D), we have assumed that $T_1 = 800°C$ and $T_2 = 15°C$ (in practice T_1 is higher). The comparison of both cycles leads to the following comments:

1. the condenser (process 4-1) being a heat exchanger of finite size, condensing water cannot be at the same temperature as the cold source, which is a first difference with the Carnot cycle. However, we can consider that the heat exchange is nearly isothermal and irreversibilities are small;

2. compression (process 1-2) in the pump may at first approximation be assumed isentropic, its irreversibilities being low. However it deviates significantly from the Carnot cycle because the compression end temperature is about the same as that of the cold source, rather than that of the hot one;

3. we have seen that the isobaric pressurized water heating is done in three steps:
   - liquid heating in the economizer up to the saturation temperature at the pressure considered (process 2-3a);
   - vaporization at constant temperature in the vaporizer (process 3a-3b);
   - superheating up to maximum temperature in the boiler cycle (process 3b-3).

Given the shape of isobars in the entropy chart, this isobaric heating process deviates much from the Carnot cycle which states that the heat engine exchanges heat at constant temperature with the hot source. In particular the temperature difference with the hot source is maximum in the economizer: heat at high temperature is used to warm water below 100°C.

4. turbine (process 3-4) has an isentropic efficiency close to 0.85. Again the difference with the Carnot cycle is significant.

As illustrated in the figure, the shape of the steam power plant differs appreciably from that of the Carnot cycle: it looks rather like a triangle than a rectangle, and its surface is much smaller. There is room for improvement by conducting reheats that lead to so-called para-isothermal expansion, and extractions which allow to perform a partial regeneration.
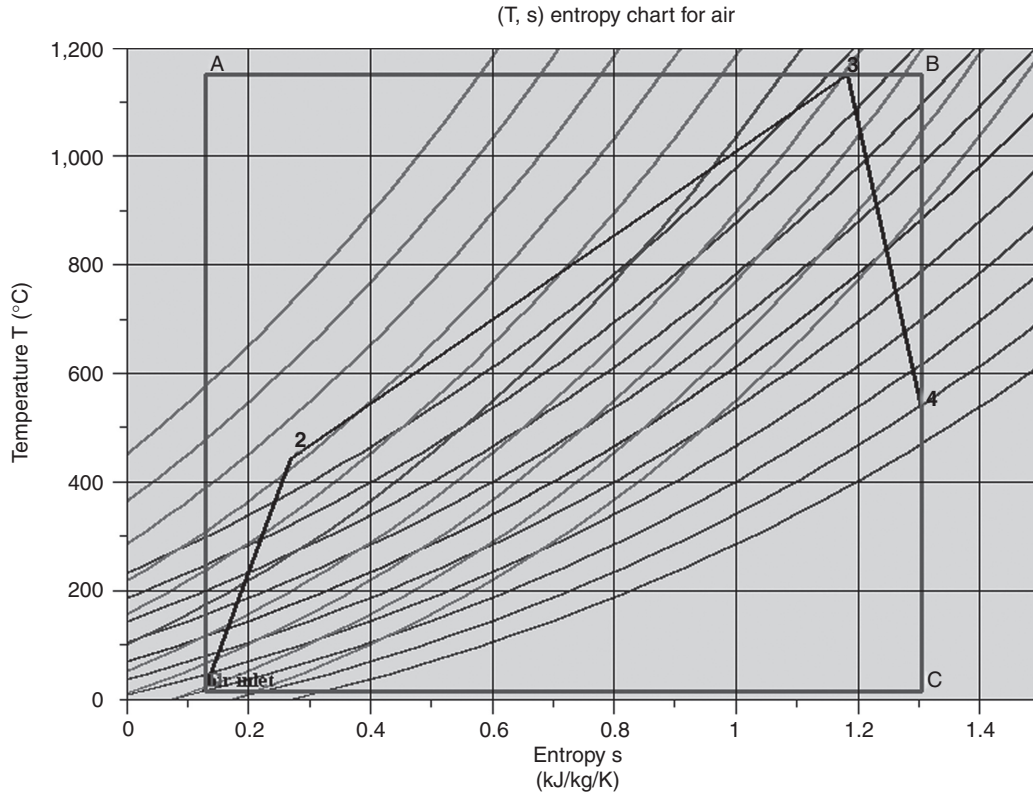
FIGURE 3.7.2

Gas turbine and Carnot cycles in the entropy chart

3.7.2 Gas turbine

In the entropy chart (Figure 3.7.2), irreversible compression and expansion result in an increase of entropy (air inlet-2, and 3-4). Heating in the combustion chamber is isobaric. However, as the working fluid composition changes during the combustion, the properties of the burnt gases are not exactly the same as those of air and we should not in principle plot points 3 and 4 on the same chart as points 1 and 2. There is indeed a change in entropy due to the combustion which explains why points 2 and 3 and points air inlet and 4 seem not to be on the same isobar.

In order to plot the Carnot cycle (A-B-C-D), we have assumed that $T_1 = 1150°C$ and $T_2 = 15°C$. The comparison of both cycles leads to the following comments:

1. compression (process 1-2) cannot be assumed to be isentropic, due to irreversibilities taking place in the compressor. It departs from the Carnot cycle;
2. combustion (process 2-3) takes place at constant pressure. Given the shape of the isobars on an entropy chart (similar to exponential), the difference is important between this process and the Carnot cycle which states that the heat engine exchanges heat at constant temperature with the hot source.
3. turbine (process 3-4) has an isentropic efficiency close to 0.9. Again the difference with the Carnot cycle is significant.
4. hot gases leaving the turbine are then discharged directly into the atmosphere, which is an important form of irreversibility, since their heat is lost.

As illustrated in the figure, the shape of the gas turbine cycle differs appreciably from that of the Carnot cycle: it looks rather like a diamond than a rectangle, and its surface is much smaller. Improvements can be obtained by conducting cooled staged compression, staged expansion with reheats, regeneration and combined cycle arrangement.

3.7.3 Refrigeration machine

In the entropy chart (Figure 3.7.3), to increase readability, we have not shown the isovolumes. Point 1 (partly masked by point C), slightly superheated, is placed on isobar 1.8 bar right of the saturated vapor curve. Irreversible compression results in an increase of entropy. Cooling with outside air has three stages: de-superheating (2-3a) in the vapor phase, condensation along the horizontal segment (3a-3b), and a slight sub-cooling (3b-3) which almost coincides with the liquid saturation curve.

Isenthalpic throttling (3-4) leads to an increase of entropy, point 4 being located within the zone of vapor-liquid equilibrium (quality equal to 0.363).

In order to plot the reverse Carnot cycle (A-B-C-D), we have assumed that $T_1 = 40°C$ and $T_2 = -8°C$. The comparison of both cycles leads to the following comments:

1. the evaporator being a heat exchanger of finite size, the refrigerant cannot be at the same temperature as the cold chamber, which is a first difference with the reverse Carnot cycle. However, we can consider that the heat exchange is nearly isothermal;

2. compression cannot be assumed to be isentropic, due to irreversibilities taking place in the compressor, which induces a new gap with the reverse Carnot cycle;

3. cooling and condensation of the refrigerant by heat exchange with ambient air generally cannot be isothermal, given the thermodynamic properties of refrigerants: you must first desuperheat the vapor, then condense it. Given the shape of isobars on an entropy chart, the difference with the reverse Carnot cycle is very important here;

4. the expansion of the condensed refrigerant could theoretically be close to isentropic, but the technological reality is different, and this for three reasons: firstly the expansion of a two-phase
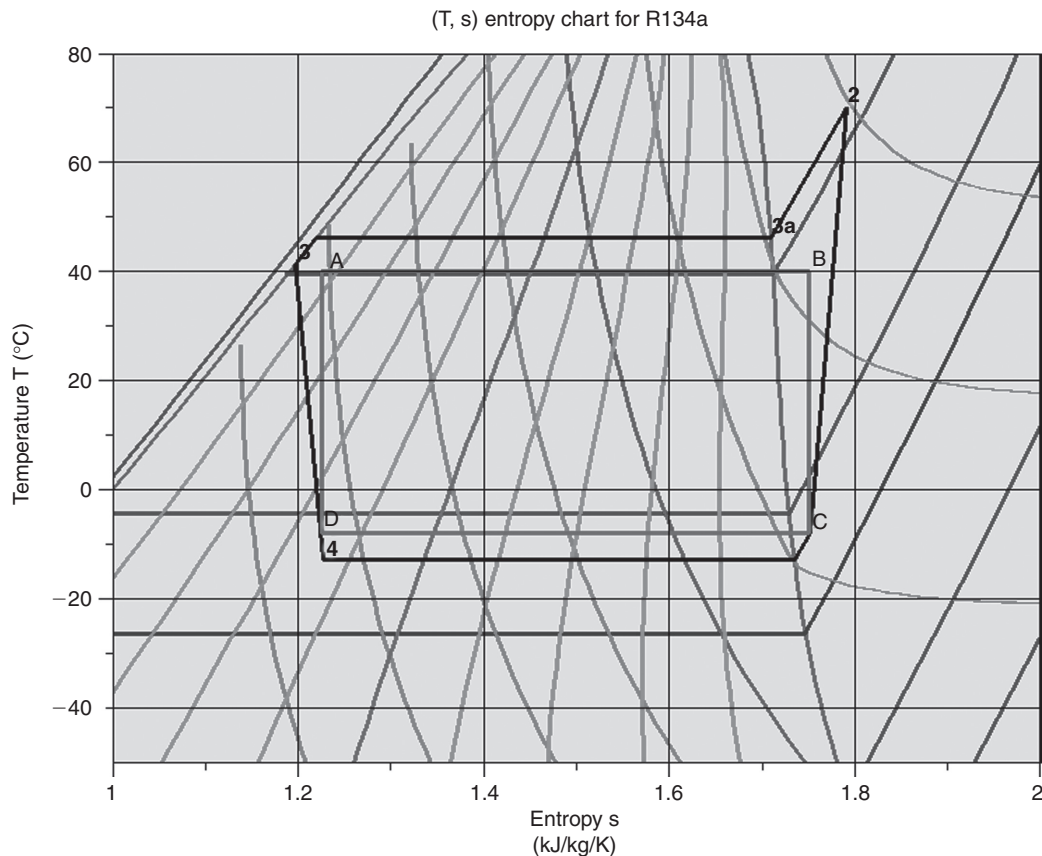

FIGURE 3.7.3

Refrigeration machine and reverse Carnot cycles in the entropy chart

mixture is done, unless special precautions are taken, with low isentropic efficiency; second, the work at stake is very low; and finally, especially for low-capacity refrigerators, there is no — suitable and cheap expansion machine. Therefore in practice one resorts to expansion valves or even simple capillary static devices that perform isenthalpic throttling. Again the gap with the reverse Carnot cycle is significant.

As illustrated in the figure, the reverse Carnot cycle appears inside the actual refrigeration cycle. The main difference is the compression curve (1-2-3a) due to the shape of isobars in the vapor zone. The surface of the actual cycle is larger.

3.8   CONCLUSION

In the first part of this presentation, we have shown that it is possible to introduce thermal machine cycles to students without referring to entropy. In this second part, we have shown that this notion is particularly interesting because it allows us to take into account irreversibilities that take place in actual processes.
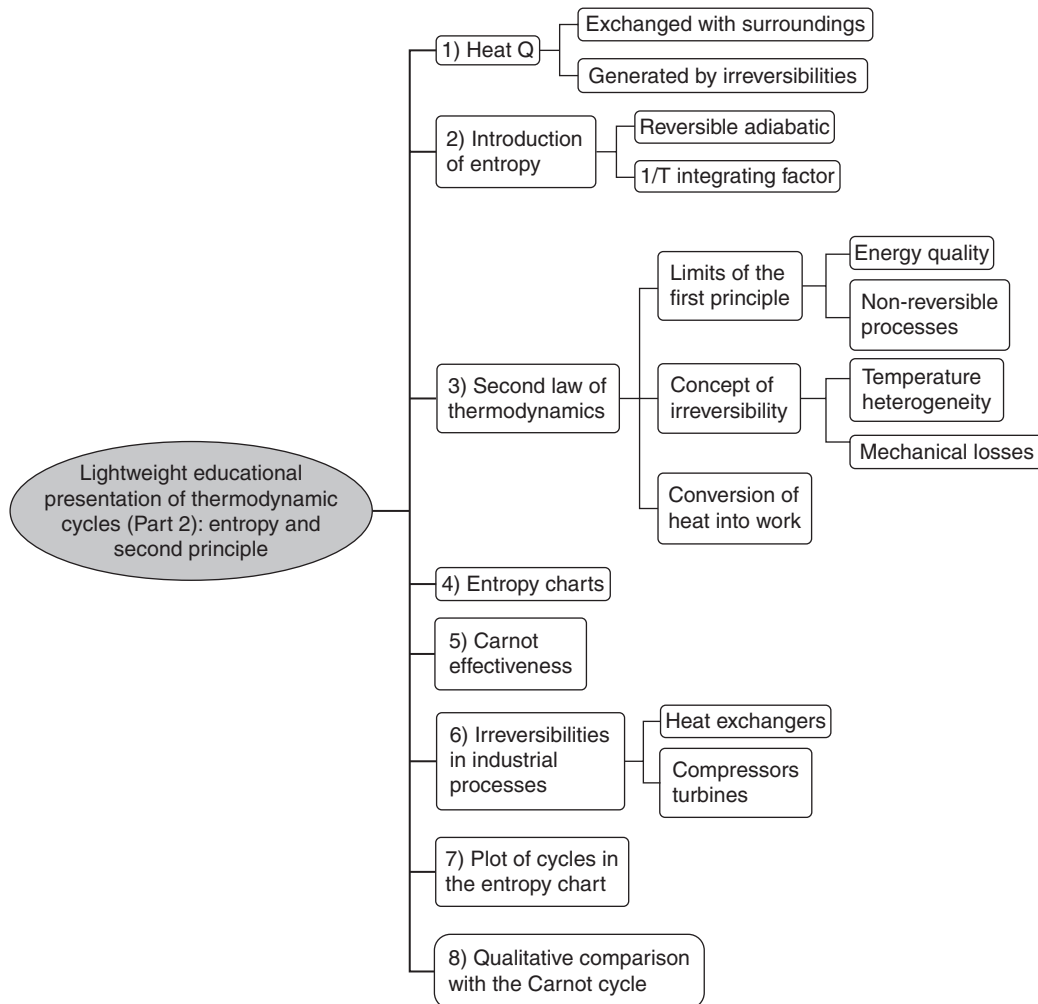

FIGURE 3.8.1
Map of the lightweight educational presentation milestones (Part 2)

Qualitative comparison with the Carnot cycle can be made in the (T, s) entropy chart and provide guidance on possible improvements.

The different steps that we recommend (Figure 3.8.1) are the following (we have indicated in parentheses sections of Part 2 of the book where these points are developed in order to facilitate a further deepening of concepts discussed):

- Heat in thermodynamic systems (section 5.2.2);
- Limits of the first law of thermodynamics (beginning of section 5.4);
- Concept of irreversibility (section 5.4.2);
- Heat transfer inside an isolated system, conversion of heat into work;
- Statement of the second law (section 5.4.1);
- Presentation of the (T, s) entropy chart (section 5.6.6.1);
- Carnot effectiveness of heat engines (section 5.4.3);
- Irreversibilities in industrial processes (section 5.4.2);
- Plot of cycles in the entropy chart (sections 9.1, 9.2 and 9.3);
- Qualitative comparison with the Carnot cycle.